3RD EDITION

Commercial Energy Auditing Reference Handbook

Steve Doty, P.E., C.E.M.

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Thank you, Katy

Table of Contents

Introduction

This handbook is a valuable collection of reference material related to commercial energy auditing, written for practicing energy professionals. The main goal of this book is to increase audit effectiveness so that more audits become projects. It includes proven solutions and practical guidelines, as well as the essential information needed for winning energy audits and projects.

The motivation for writing this book:

- Having accurate facts and figures in **one convenient location** will accelerate the research phase of each audit while maintaining quality.
- The benefit of hands-on practical experience is found throughout this book, including things that work and things to watch out for.
- Audit strategies that are **tailored to each unique business sector** are more effective than general guidelines, and the first step is to learn where the energy is used in the business.
- **Automatic control systems** are often underutilized and represent untapped potential for savings. To leverage this technology, a separate section provides specific automatic control applications.
- **Quantifying savings** is a skill that is essential for success, but one that is treated lightly in many texts and so a section is devoted to identifying the fundamentals at work and how to quantify them.
- Identifying **maintenance** activities that reduce energy use represent immediate low cost savings opportunities for customers.
- The first step to reducing energy use in the commercial building population is to encourage **changes in how *new* buildings are built** and operated, and there is a section on this subject. By curbing the energy use of new buildings, we can eventually catch up.

The measure of the book's success will be its acceptance and use. Your feedback and suggestions are welcome.

Steve Doty

To the Reader

Here are the two main ways I use to find information in this book, with examples.

Method 1: (Table of Contents or TOC): Identify the general category to narrow the search, then browse the sub categories using the Table of Content and Appendix.
Method 2 (Index): Identify unique key words or phrases for your subject matter. Using the Index, find the key word directly.

Example: What are typical measures that make sense to a school?
(TOC): Look in the section "SPECIFIC INFORMATION" and find "ENERGY SAVING OPPORTUNITIES BY BUSINESS TYPE"; then find "Education—Colleges and Universities" or "Education—Schools K-12"
(Index): Look for "schools" or "universities" or "K-12."

Example: What is the conversion between ton-hours and kWh?
(TOC): Look in the section "GENERAL INFORMATION" and find "FORMULAS AND CONVERSIONS"; then find "HVAC Formulas and Conversions."
(Index): Look for "ton-hours."

Note on Sources:

There are hundreds of third-party sources in this handbook, which will add credibility when using the statements, facts, and figures it contains. However, not all of the information has a third-party source to rest on—which either means I couldn't find one, or the information represents an original thought. In any case, where there is no source listed, the source should be considered by default to be:

Source: Doty, Steve, Commercial Energy Auditing Reference Handbook, Fairmont Press.

Note on Corrections:

Continuous Improvement is a way of life for engineers. If you have a question or comment, or if you find an error within this book, please contact the publisher and they will forward it to me. Thank you!

User Guide

This handbook contains a great deal of information in condensed form and its success will depend in large part on the ability of the reader to navigate through the material. A condensed Table of Contents below shows the basic organization of subject matter. In the full Table of Contents, each item is expanded as shown in this example. There is also a generous index of key words for quick access. Note that the Appendix items are not indexed.

Suggested References

This is a condensed collection of information for professionals engaged in energy engineering. It is mostly without explanatory material and relies on readers with an understanding of the underlying principles and applications. It will also serve as a companion reference for other texts on individual subjects. The following are suggested sources of additional information and detail.

Energy Engineering and Energy Management

Energy Management Handbook, Doty / Turner, Fairmont Press
Handbook of Energy Engineering, Thumann / Mehta, Fairmont Press
Information Technology for Energy Managers, Capehart, Fairmont Press

HVAC

Principles of Heating, Ventilating and Air-Conditioning, ASHRAE
Pocket Guide for Air-Conditioning, Heating, Ventilation and Refrigeration, ASHRAE

Lighting

Lighting Management Handbook, DiLouie, AEE
IES Lighting Ready Reference, IES

Automatic Controls

Fundamentals of HVAC Control Systems, Taylor, ASHRAE
Optimization of Unit Operations, Liptak, Chilton

Process Diagramming and Mass-Energy Balancing

Material and Energy Balance, Guidebook 1—General Aspect of Energy Management and Energy Audit, Chapter 4,
Bureau of Energy Efficiency, New Delhi, India (Govt. issued text)

Section I

Specific Information

Chapter 1

Benchmarking

GENERAL

A good place to begin an energy audit is to compare the use of the facility with similar facilities. For the purpose of this book, benchmarking refers this initial comparative step. It is a rough gage of whether the existing use is more than, less than, or about the same as what would be expected.

It provides a basic indicator of potential for savings. For **example**, if the annual energy use per SF is 25% higher than a trusted benchmark value, then a 25% savings for the facility may be a reasonable goal. Similarly, if the energy use is about equal to the available benchmark, a customer desiring to reduce energy use by 50% will likely require extreme measures and may not be practical.

The comparison may be in any meaningful and available units that are pertinent to the business. Often customers or performance contractors are interested in dollars instead of energy units, since dollars are what drive the business. This is understandable, however benchmarking in dollars introduces additional variables of the cost of energy—and this can vary substantially by area and over time. For this reason, benchmarks that are in terms of energy units instead of dollars introduce less uncertainty and are easier to work with. Just remember that, at the end of the day, it's usually about money.

Examples:

Energy use per SF per year	(kBtu/SF-yr)
Energy use per part produced	(kBtu/part)
Energy use per ton of material processed	(kBtu/ton)
Energy use per month per hotel guest	(kBtu/month-guest)
Energy use per 100 meals served	(kBtu/100 meals)
Energy use per year per student	(kBtu/student-yr)

DIFFERENTIATING BY ENERGY SOURCE

Benchmarking is possible for electric use vs. natural gas use or other fuels, however this introduces more variables. For **example**, if one building is heated with electric resistance, and one is heated with natural gas, and one with a combination of electric and gas, the benchmarks would be different with electric and gas energy evaluated separately. But if the benchmark is simply Btus of energy, then buildings heated by any means would be measured against the same standard.

ENERGY USE INTENSITY

The term *energy use intensity* (EUI) is a common measure of energy consumption in commercial buildings. It is synonymous with energy use index and energy use intensiveness. It is derived from the total consumed energy per year, divided by the square foot area of the building. It represents how concentrated the energy use is within the building. Buildings with high EUI also tend to be self heating. A variety of EUI data is provided in the **Appendix**.

LIMITATIONS OF EUI

EUI is a simple measure, with limitations. For **example**, using the standard metric of floor area (square feet), the building volume is not considered. A warehouse with high stacked storage and high ceilings will have increased energy use through envelope heat loss/heat gain compared to shorter warehouse building with less wall area.

CALCULATING EUI

Note: for these examples, natural gas is used. The same method applies to other fuels.

Single EUI

For defined business sectors with available data, comparing actual to 'reasonable' energy intensiveness (Btu/SF-yr) is a logical first step.

Step 1: Convert one year's energy consumption data, gas and electric, to the common unit of kBtu (1000s of Btus).

1 kBtu = 1000 Btu
1 kWh = 3.413 kBtu
1 therm = 100,000 Btu
1 ccf= _____ Btu **

***This varies by locale,* typically around 100,000 Btu/100 cubic feet (cf).
some as high as 1050/cf or 105,000 per ccf
some as low as 800 Btu/cf or 800,00 per ccf

Step 2: Determine building square footage that is occupied. Exclude garages, crawl spaces, breezeways, etc.

Step 3: Energy Use Intensiveness (EUI) = kBtu per year / SF

Step 4: Compare to available benchmark data.

See the **Appendix** for common EUI benchmark data.

MIXED EUI

It is common to create an "adjusted EUI" for facilities with defined constituent parts, using proportions.

Example: A commercial building is 70% lodging and 30% restaurant. From the **Appendix**, Lodging EUI=100; Restaurant EUI=231.
(0.7)*(100)+(0.3)*(231) = 139 kBtu/SF-yr combined (ans.)

PRODUCTION EUI

A similar approach can be used for manufacturing or other volume-related activity.

For example: If 500,000 screwdrivers are manufactured in a year, the energy cost per screwdriver can easily be calculated. In this example, energy is seen as an ingredient to the produced item, similar to steel or labor, and savings equate to reduced part cost and increased profit. Restaurants (meals per day) and hotels (average percent occupancy) are similar applications of production metrics.

EUI ADJUSTMENT FOR OCCUPANCY

Occupancy is a variable that affects energy use in offices, motels, multi-family, schools, etc. It is desirable to adjust the benchmark for energy use according to occupancy, but the relationship is not simple. See **Chapter 8—Building Operations and Maintenance, "Vacancy."**

Energy use effects from reduced occupancy are more significant at very low occupancy levels, although zero occupancy does not equate to zero energy use. At 0% occupancy, the building is vacant and energy use intensity can be estimated using CBECS 'vacant' category, with reasonable accuracy.

It is also important to recognize that few buildings are 100% occupied for sustained periods, and CBECS values of EUI do not represent 100% occupancy. For example, industry trade associations survey offices and report statistics including average occupancy—in one such survey (BOMA) the average occupancy was reported to be 87%.

One approach to establishing an approximate EUI where occupancy is significantly low (say, half or less) is to consider the building to be two pieces. For example, an office building operating at 50% occupancy could have an EUI that is a blend of 'office' and 'vacant', blending the two EUIs proportionally on area. Where occupancy levels vary significantly throughout the year, regression can correlate occupancy to energy use.

ENERGY END USE DISTRIBUTION

Sometimes called energy use "pies," these predict the percent of total use going to which function, such as heating, lighting, etc. This can be used as a benchmark if a constituent piece of energy use is known.

Example: If the domestic water heaters are electric and the only thing using natural gas is the heating system, then the natural gas bills represent the entire heating energy use. If this amount of energy represents 40% of total energy but the end use pie shows 20%, this may indicate excess heating energy is being used for some reason.

Chapter 3 shows energy end use distribution pie diagrams for some common business sectors.

ENERGY END USE VARIATIONS OVER TIME

Note the striking jump in lighting energy in the 70's and 80's, and the high heating costs characteristic of older buildings (envelope).

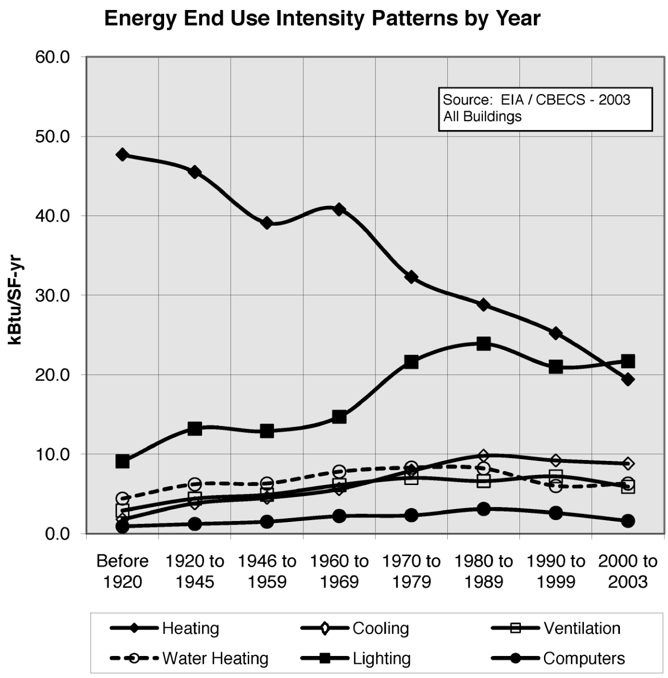


Figure 1-1. Energy End Use Patterns by Year

ENERGY COST AS A PERCENT OF TOTAL OPERATING COST

For operating budgets, "utilities" is a line item and often most of that is energy. Each percent of savings on utilities translates to increased profit. This is shown in table form in the **Appendix** item: **"Equating Energy Savings to Profit Increase."** If the operating cost percentage is high compared to other similar companies, then profit will usually be proportionally lower.

Information on operating expenses is available from private industry groups, and also from the U.S. Census Bureau "Business Expenses" report. To get energy cost as a percent of total operating cost, divide the utility cost by the total operating expenses. Some typical energy cost percentages are shown in the **Appendix** item: **"Operating Expenses: Percent that are from Utility Costs"**

LIMITATIONS OF USING BENCHMARK DATA

While very useful, remember that benchmarks are averages. When a test value is significantly different than a benchmark, it prompts questions and may lead to opportunities. However when a test value is only slightly different than the benchmark, the conclusion may be only that the numbers are close. For **example**, if a building EUI is calculated at 102 kBtu/SF and the benchmark for comparison is 100 kBtu/SF, this is not proof positive that the building energy use is 2% higher than normal.

National averages are just that. Local climate differences can be substantial. Where available and valid, local data will be more realistic than national data. However a local data sampling of 1 or 2 other buildings may not be valid, compared to national averages that are include thousands of data points. On the other hand, the data collected by EIA for the CBECS reports are not verified – simply entered, and anomalies exist. The best advice for the use of tables and local sources is that they suggest patterns and are valid for general conclusions only.

With additional limitations, the CBECS data can be parsed to yield answers to specific questions on energy use, and the data tables are set up to do that, with directions on the CBECS website.

See also **Chapter 9, *Normalizing Data*.**

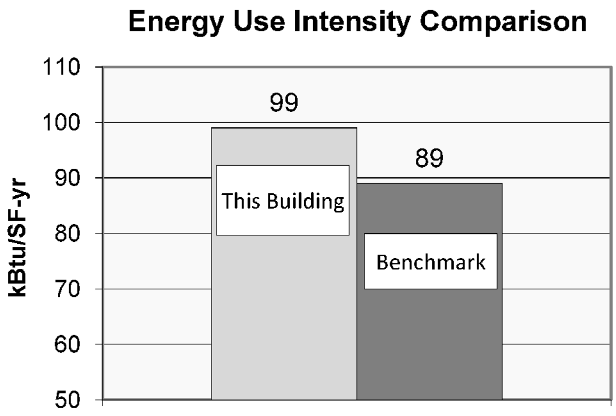


Figure 1-2. Typical EUI Benchmarking for an Office Building

Chapter 2

Analyzing Energy Use Graphs

GENERAL

Energy use profiles, by month, are very useful in most cases, especially when patterns or abnormalities suggest issues and possible energy saving opportunities. Studying these is a good initial step. Graphing is also very effective for presenting complex information to customers. Finally, graphing is an excellent tool for displaying the results of changes made—before and after. Graphing with two vertical axes can be insightful for things happening concurrently.

INTUITIVE INFORMATION

Some things are obvious from how they look.

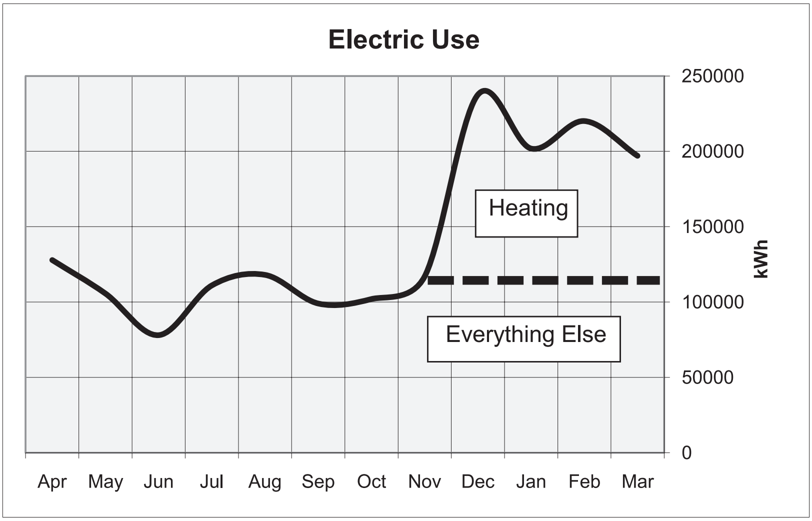


Figure 2-1a.

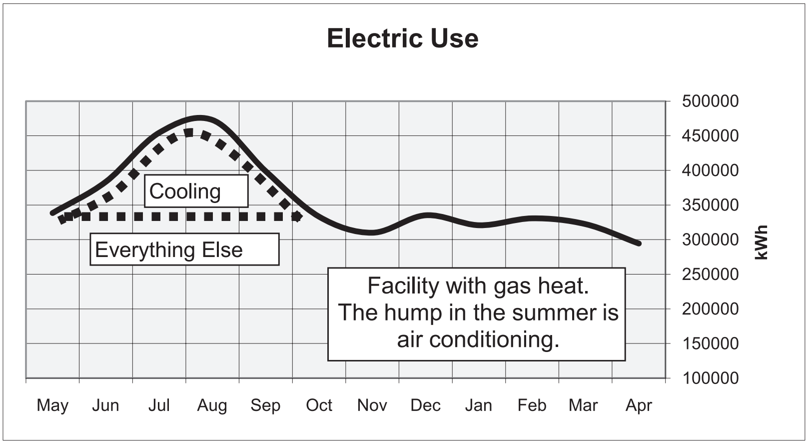


Figure 2-1b.

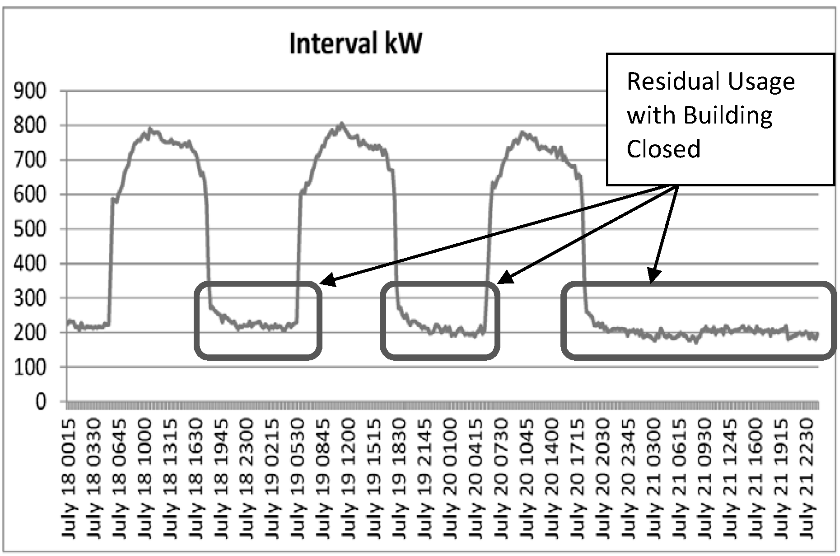


Figure 2-1c.

Interval Data to Identify a Prominent End Use

This manufacturing facility includes a large quantity of refrigerated storage. There are two shifts of productions except for Saturday afternoon through Sunday afternoon, when the plant is shut down. Remaining loads are mostly refrigeration and some lights. Interval data provides a useful sanity check to the baseline magnitude of the refrigeration load.

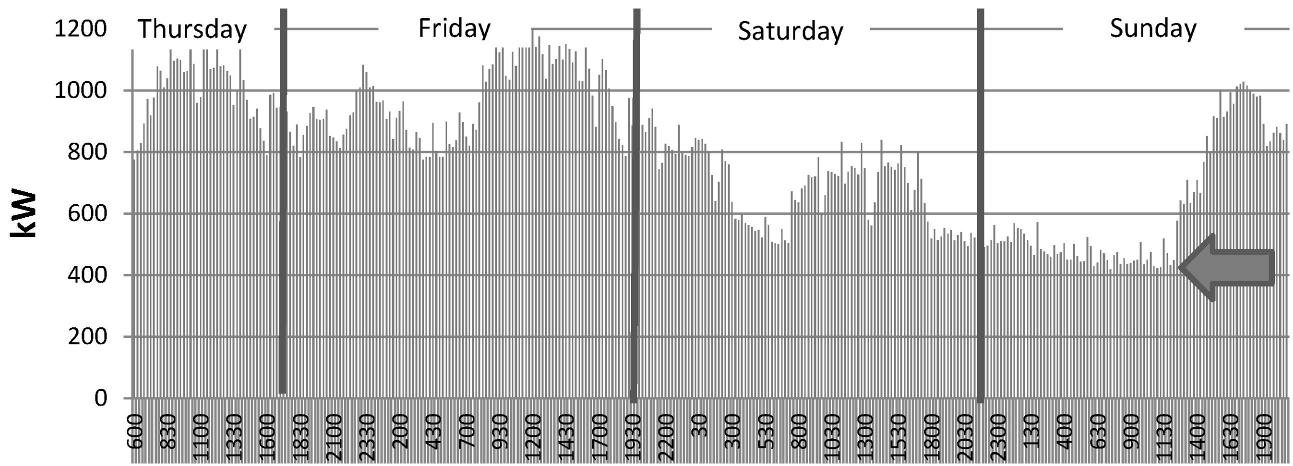


Figure 2-1d.

YEAR BY YEAR COMPARISONS

Same measurements overlaid in different years.

Example showing consistent use year-to-year.

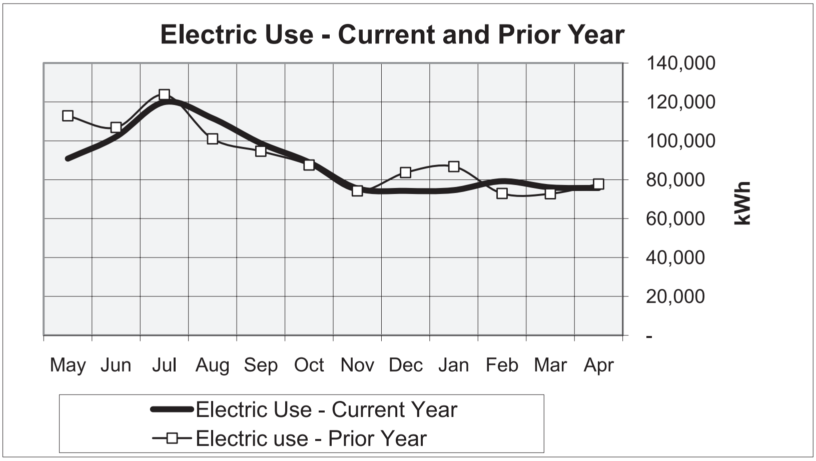


Figure 2-2a.

Energy usage creeping up year-to-year.

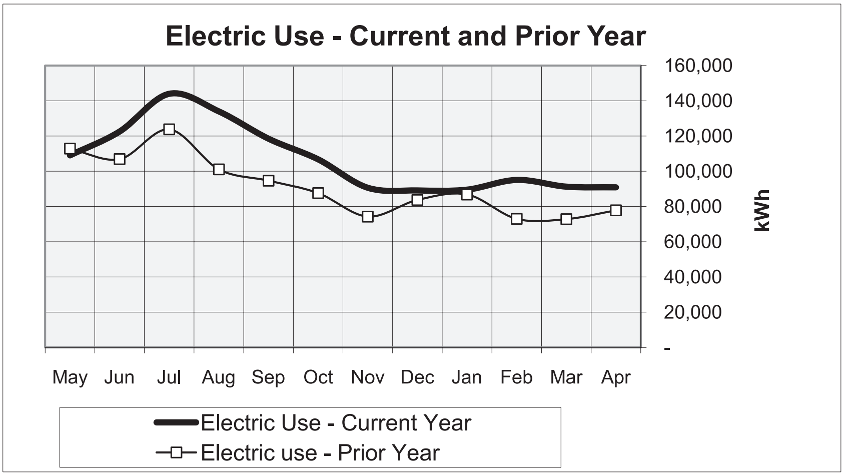


Figure 2-2b.

Example showing anomalies and prompting questions. This may have been a construction event, utility bill irregularity, or some other one-time event.

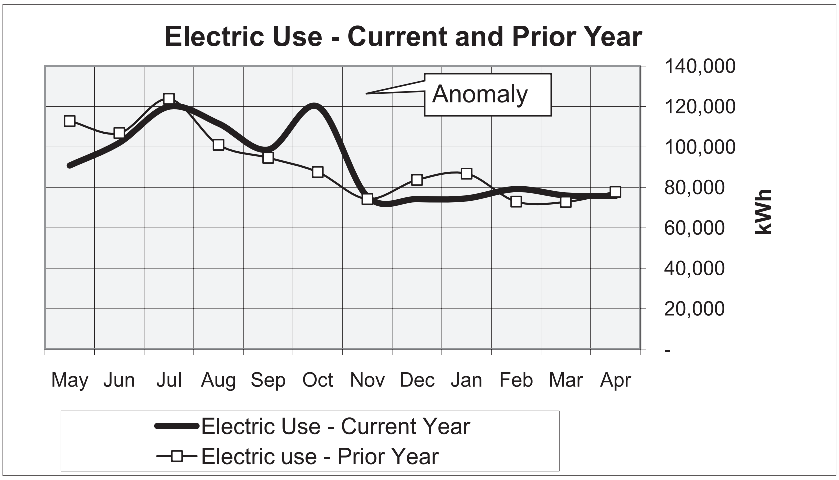


Figure 2-2c.

Example showing anomalies and offering explanations. The heavy dashed line suggests what energy use might have been without summer break.

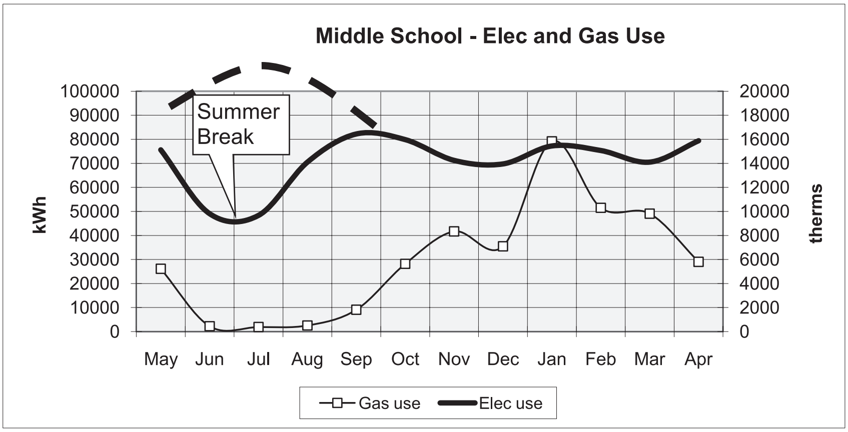


Figure 2-2d.

Example showing results of projects

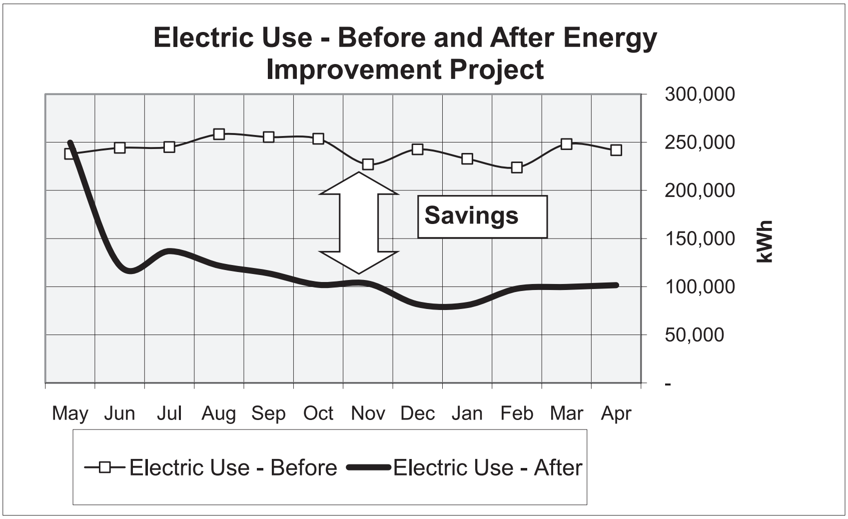


Figure 2-2e.

Comparative information:
Showing results of projects, baseline, expected, actual.

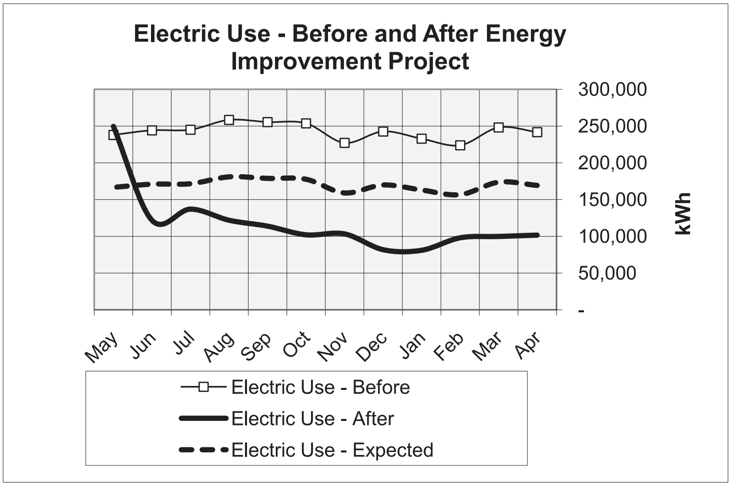


Figure 2-2f.

WEATHER DEPENDENCE

Weather dependent example

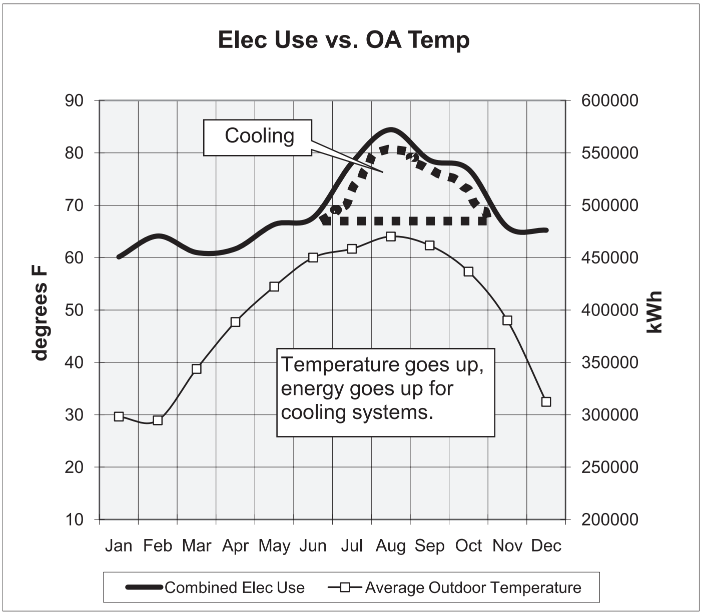


Figure 2-3a.

Weather dependent example.

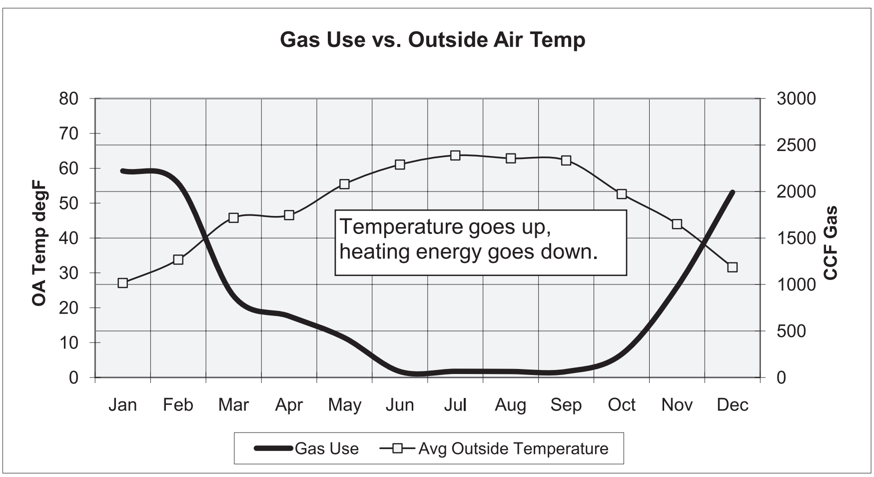


Figure 2-3b.

Weather dependent example.

Electric heat and electric cooling produce two characteristic 'humps'. The size of the two humps depends on the relative severity of the seasons, but all things equal the cooling hump will be smaller since the COP>1 and COP=1 for electric resistance heat.

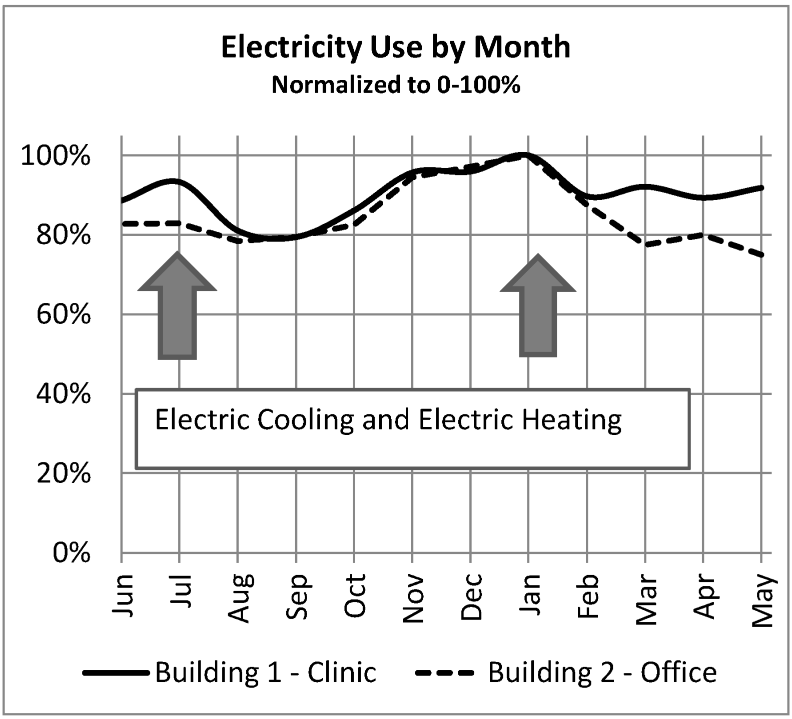


Figure 2-3c.

Weather independent example.

A common conclusion from weather independence is that energy use is driven by process and improvements related to envelope may be trivial.

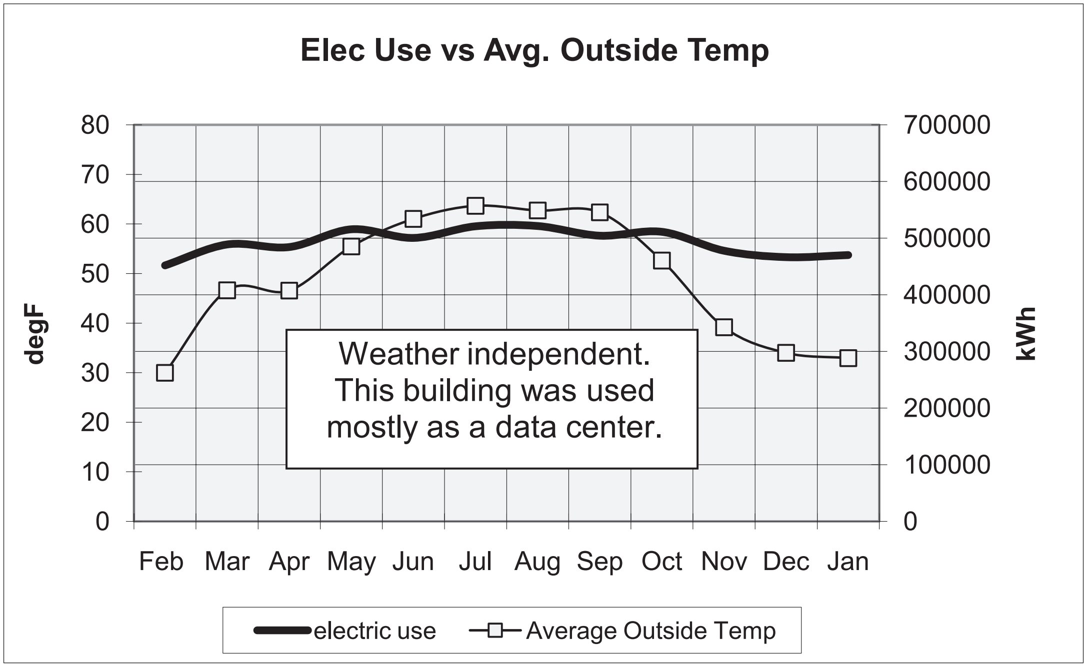
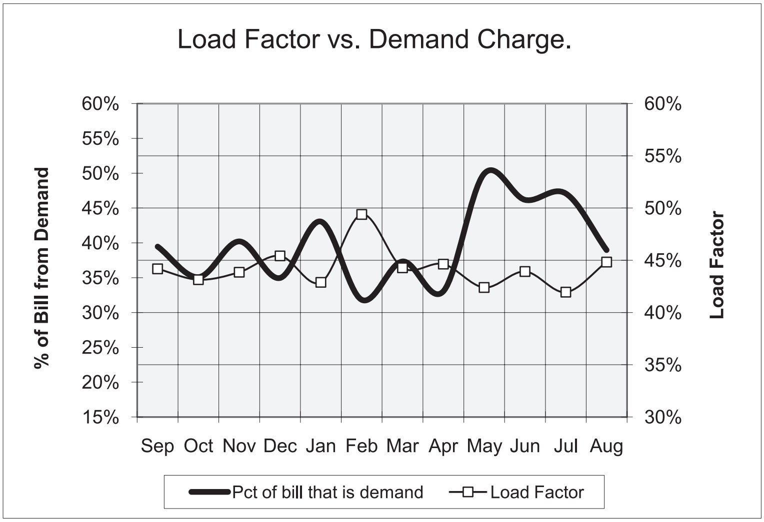


Figure 2-3d.

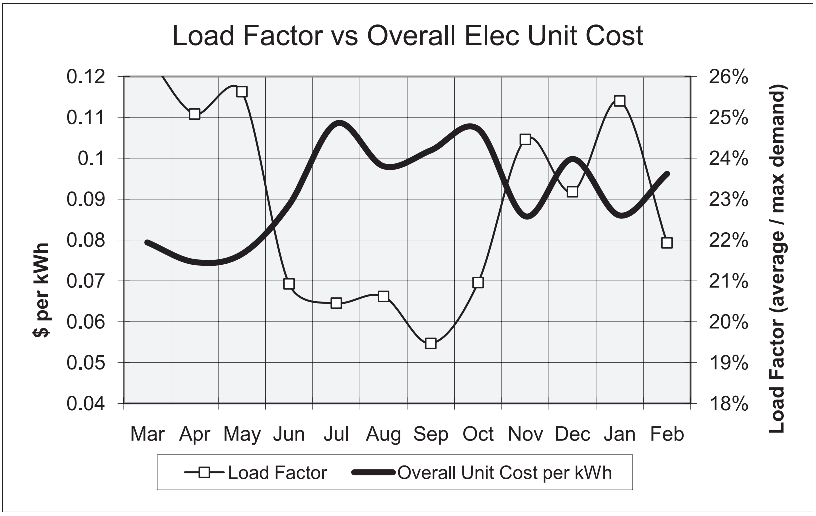
LOAD FACTOR

The example graph shows the mirror image effect of poor load factor on electric cost.

"Load Factor" is the ratio of average to maximum electric demand; overall electric cost will be inversely proportional to load factor with standard utility rates that favor steady electric usage.



Figures 2-4a.

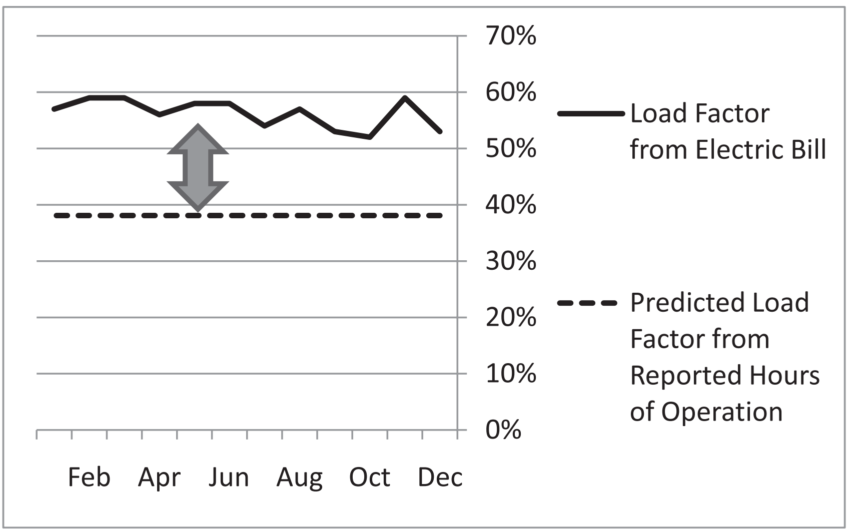


Figures 2-4b.

The effect of "low load factor" on the electric bill can be easily explained—short periods of high power demand raise the monthly demand charge by the same amount whether used for an hour or all month. Changes that improve load factor, such as steady use throughout the day and avoiding brief periods of high electrical demand, can reduce electrical charges while energy use remains the same. ECMs in this category involve planning such as changes in process schedules.

See also **"Load Factor" in Chapter 6**.

Load factor is a good sanity check for this all-electric building. **See Figure 2-4c.** The customer reported hours of 7:30am-5:30pm Monday through Friday and 8am-noon Saturday suggests a load factor less than 40% but actual load factor is much higher. This indicates that equipment is probably running after hours and may lead to savings opportunities.



Figures 2-4c.

BUSINESS VOLUME (PRODUCTION RATES)

Data points are normalized, 100% being the most production and most energy during the graphed period. When energy use does not go down proportionally with production, energy use per unit of product output increases and so the "ingredient cost" of energy in the part increases—and the profit margin for the part decreases. This concept applies equally to manufacturing, hotels, restaurants, etc.

Example: Strong correlation (manufacturing)

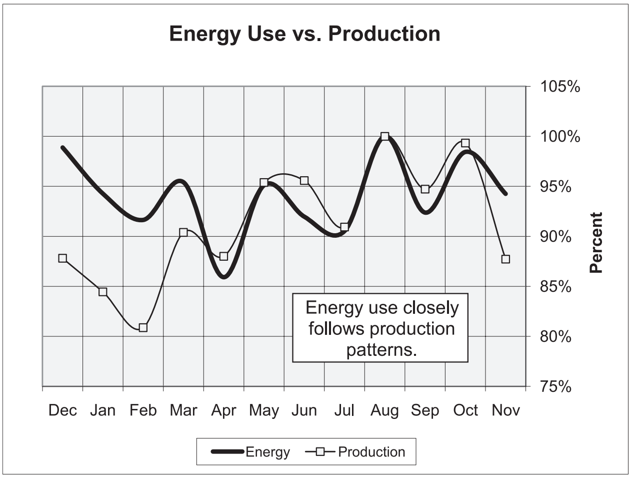


Figure 2-5a.

Example: Weak correlation (manufacturing)

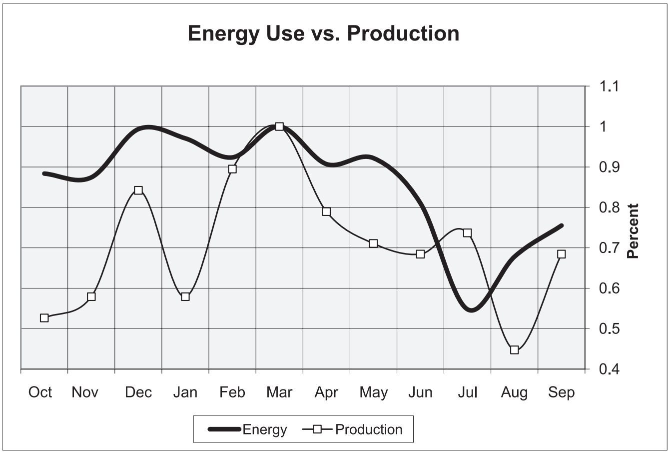


Figure 2-5b.

Manufacturing Example

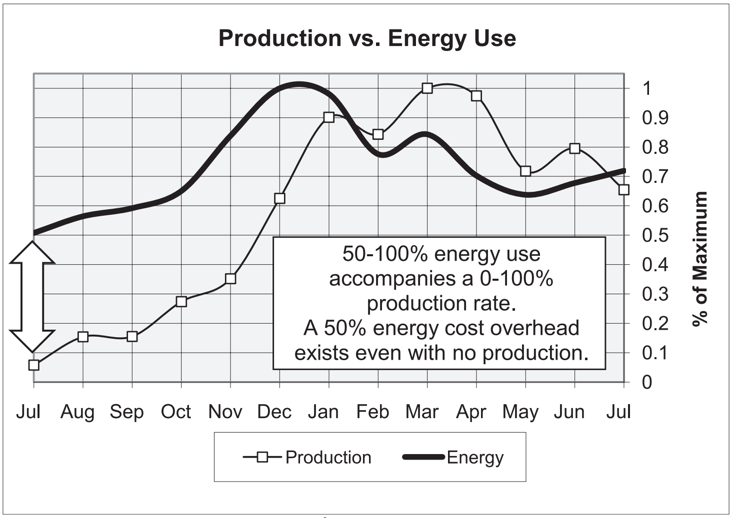


Figure 2-5c.

Hotel Example

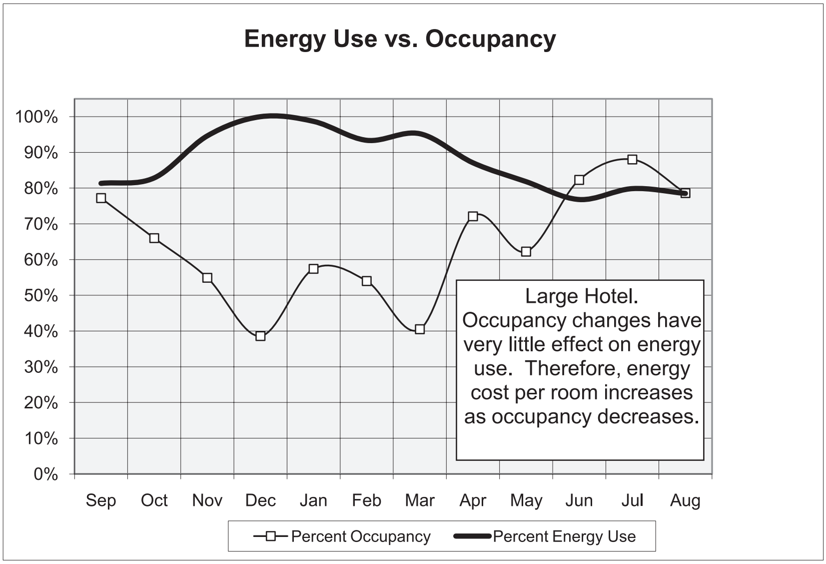


Figure 2-5d.

Energy cost per unit for this manufacturing facility increases with reduced output, from idling equipment energy use.

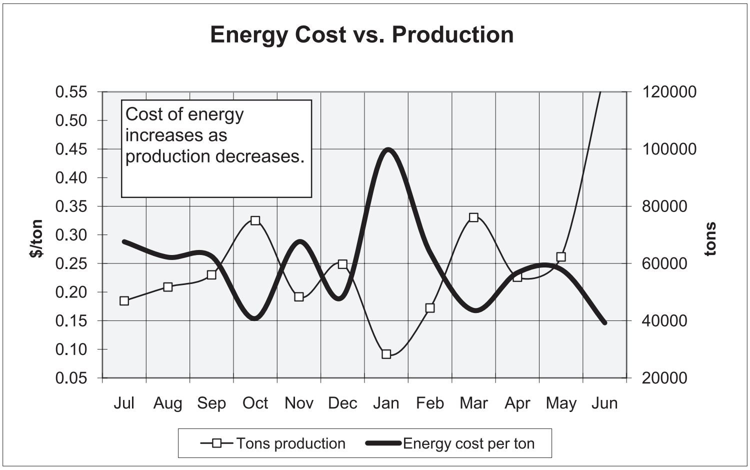


Figure 2-5e.

SAVINGS OPPORTUNITIES

Savings opportunity: Boiler running in summer, standby losses.

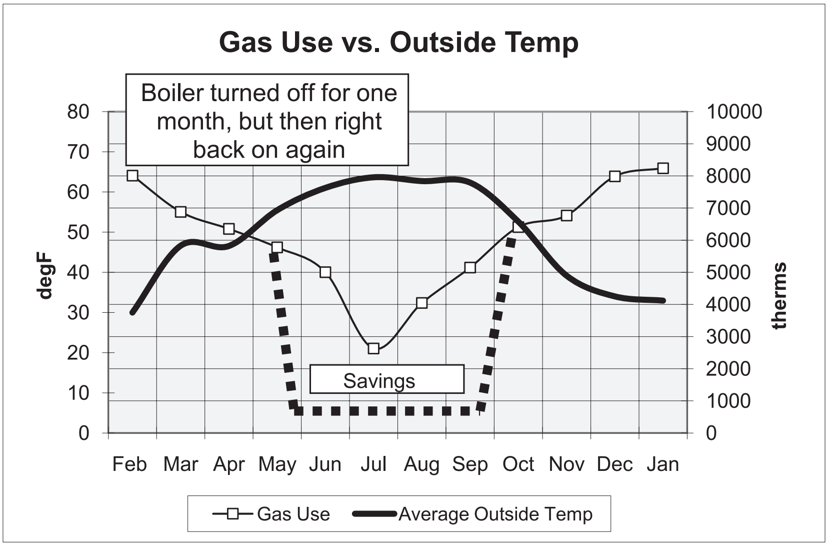


Figure 2-6a.

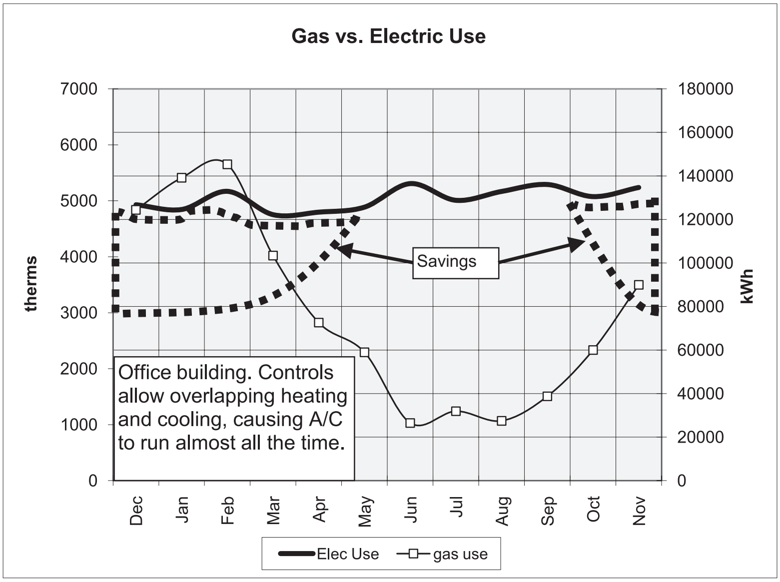


Figure 2-6b.

Savings opportunity: Overlapping heating and cooling.

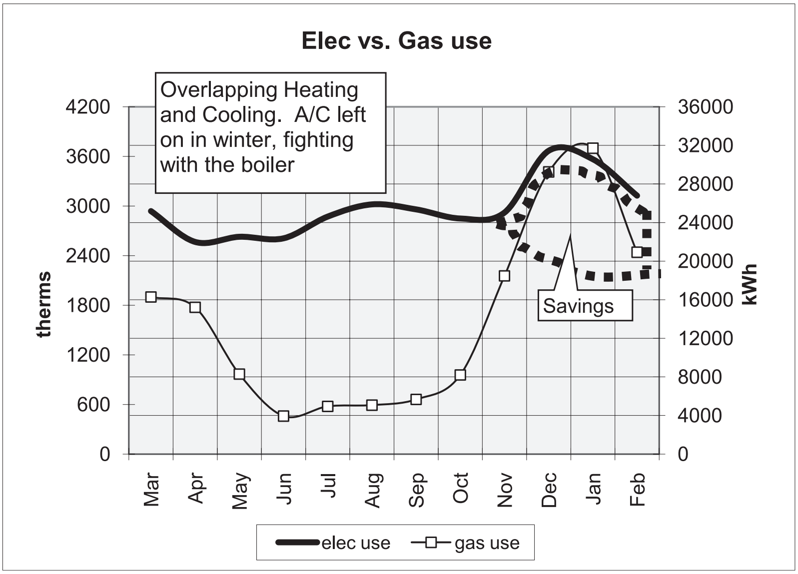


Figure 2-6c.

Savings opportunity: Economizer not functioning.

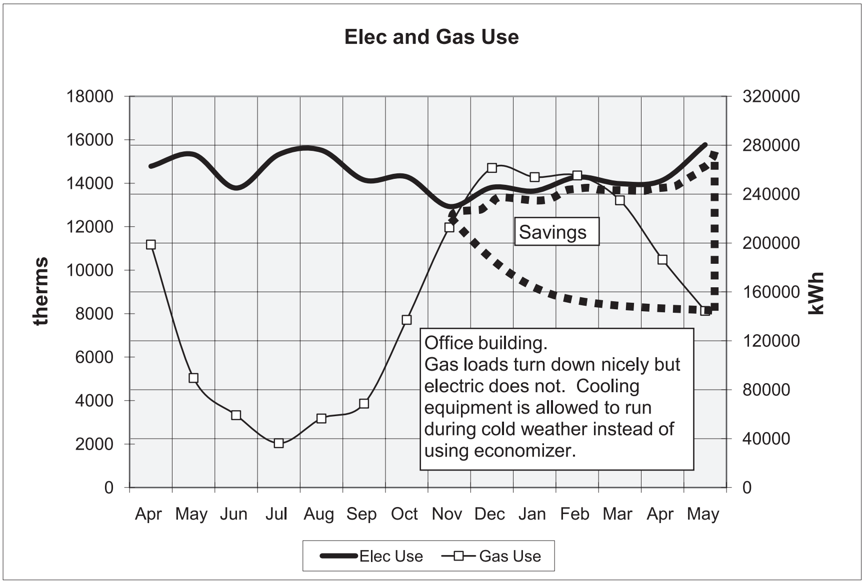


Figure 2-6d.

Chapter 3

Energy Saving Opportunities by Business Type

The dominant energy users are listed first, which are the drivers of utility costs and the most likely categories to bring the largest results. There will be other measures that can be suggested at each facility.

Common measures are organized in groups, which are:

- Controls
- Maintenance
- Low-cost/No Cost
- Retrofit—Or Upgrade at Normal Replacement
- May Only Be Viable During New Construction

A preface to this section is a listing of energy end use pies, which are excellent visual aids and also serve well for customer presentations. Data for the pie charts is from CBECS End Use reports. These CBECS reports are from modeling and not survey data, and reasons for variations between survey years are unknown.

PBA CONCEPT

The **PBA concept** (Principal Business Activity) is very important in using CBECS data. For example, a university campus includes multiple PBAs; the "Education" piece is only for the classrooms. Lecture Halls would be "Public Assembly," dormitories would be "Lodging," and the cafeteria would be "Food Service."

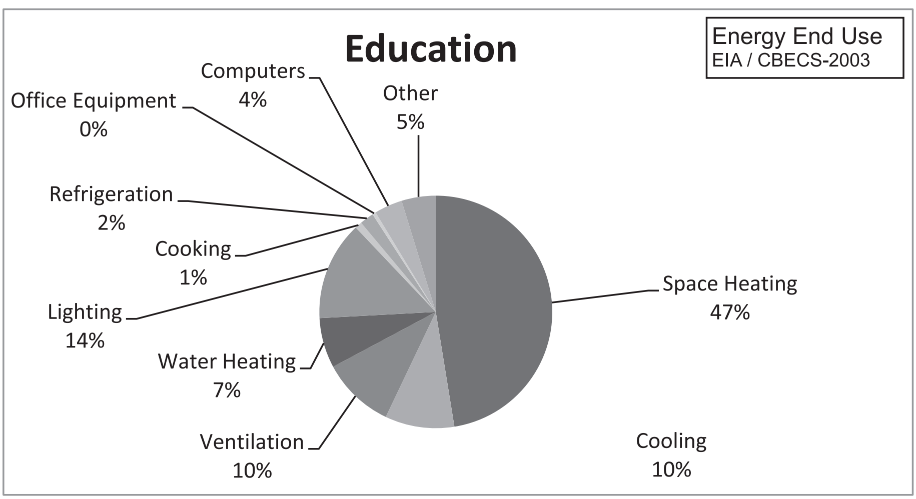
Naming of the pies is consistent with CBECS naming (PBA). A definition of each CBECS PBA name is located in the **Appendix**. Naming of the segment ECM sheets in this text is not always the same as the PBA, and not all segment ECM sheets have a corresponding PBA category. A reasonable correlation is shown in **Figure 3-1.**

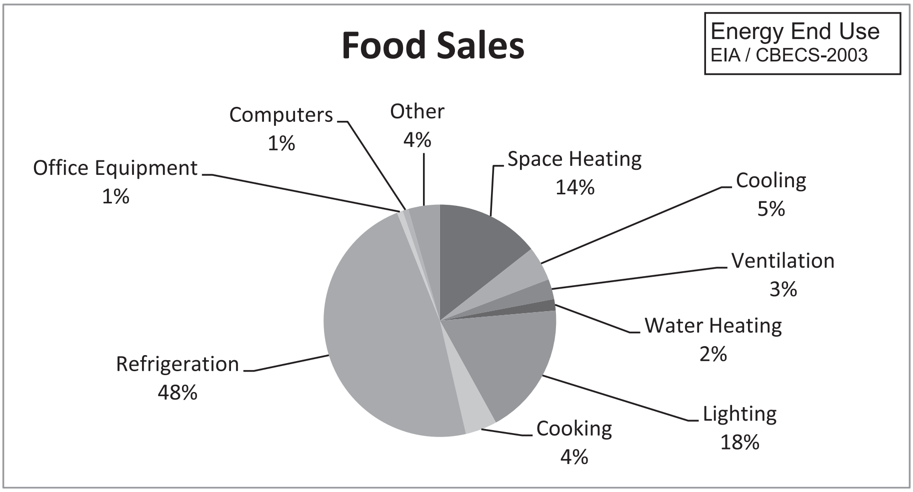
Segment ECM Sheet Name (This Text)	CBECS End Use Pie Name (PBA) – Best Fit
Apartment Buildings / Multi-Family / Dormitories	Lodging
Churches / Worship	Religious Worship
Data Centers	----
Education – Colleges and Universities	Education
Education – Schools K-12	Education
Food Sales – Grocery Stores	Food Sales
Food Service - Restaurant	Food Service
Health Care - Hospital	Health Care – Inpatient
Health Care – Non Hospital	Health Care - Outpatient
Laundries - Commercial	----
Libraries / Museums	----
Lodging / Hotels / Motels	Lodging
Office Buildings	Office
Retail / Sales	Mercantile – Retail Mall and Non-Mall
Warehouses	Warehouse and Storage

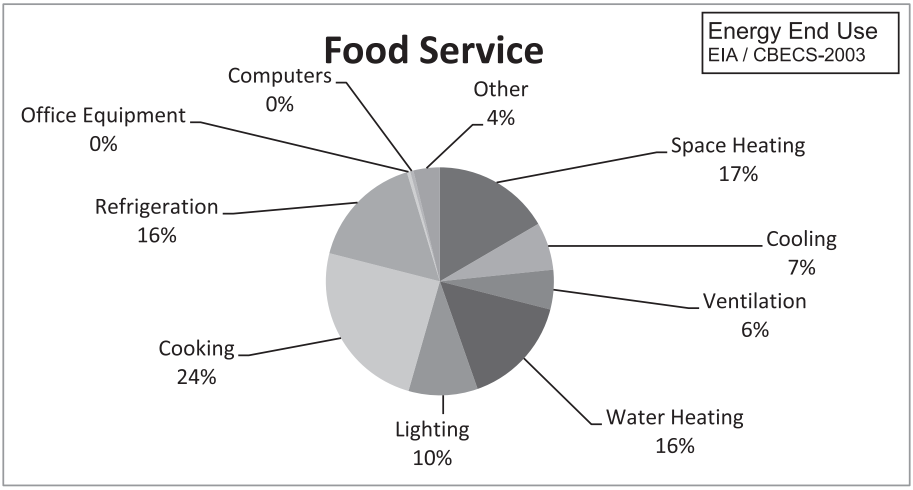
Figure 3-1. CBECS End Use Pie Diagrams by Segment ECM Sheet

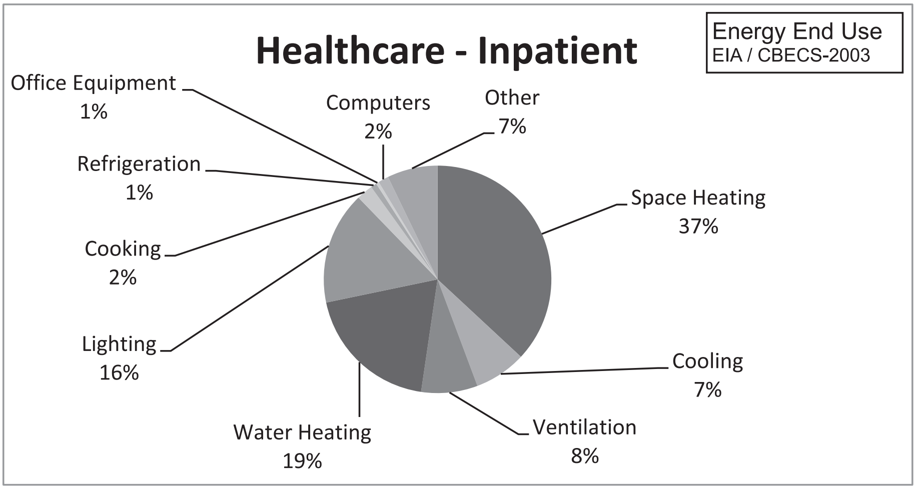
ENERGY END USE PIES

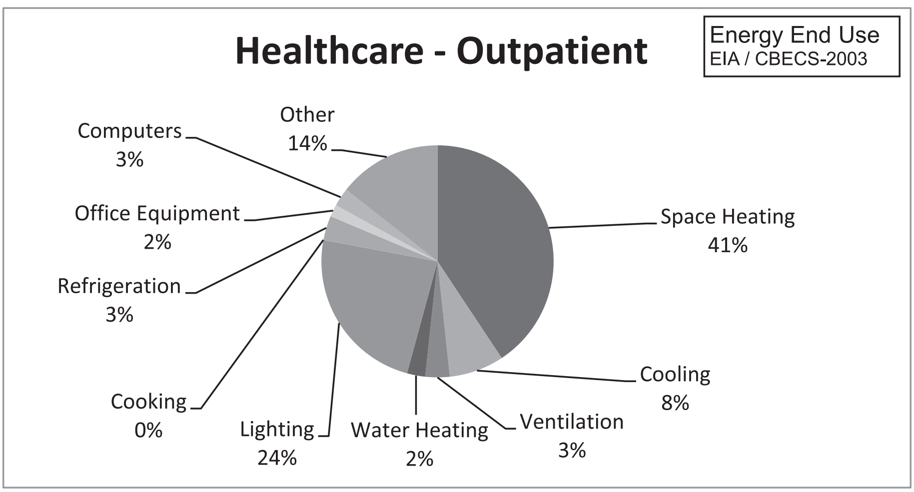
2003 End Use Data

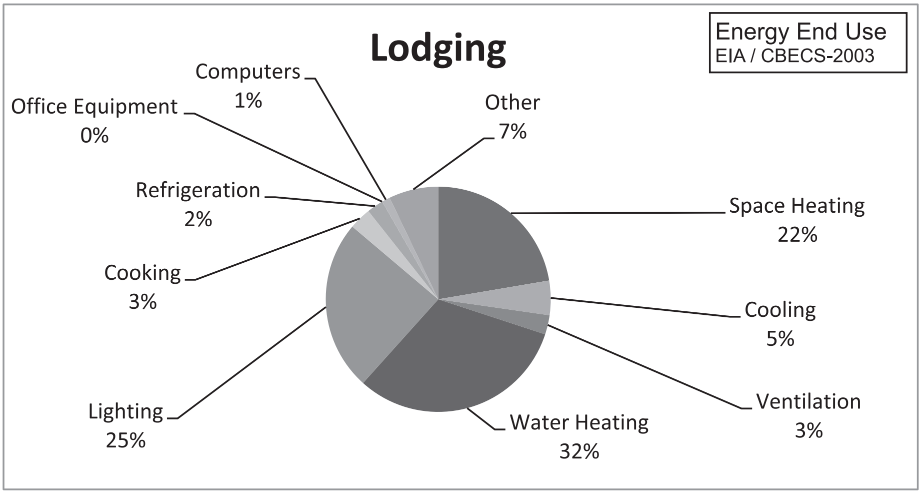


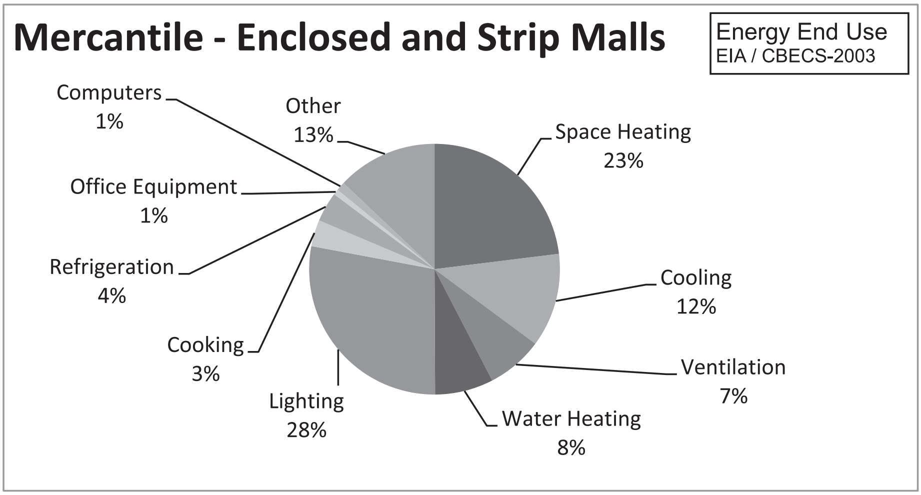

Food Sales
Energy End Use
EIA / CBECS-2003
Computers 1%
Other 4%
Office Equipment 1%
Space Heating 14%
Cooling 5%
Ventilation 3%
Water Heating 2%
Refrigeration 48%
Lighting 18%
Cooking 4%

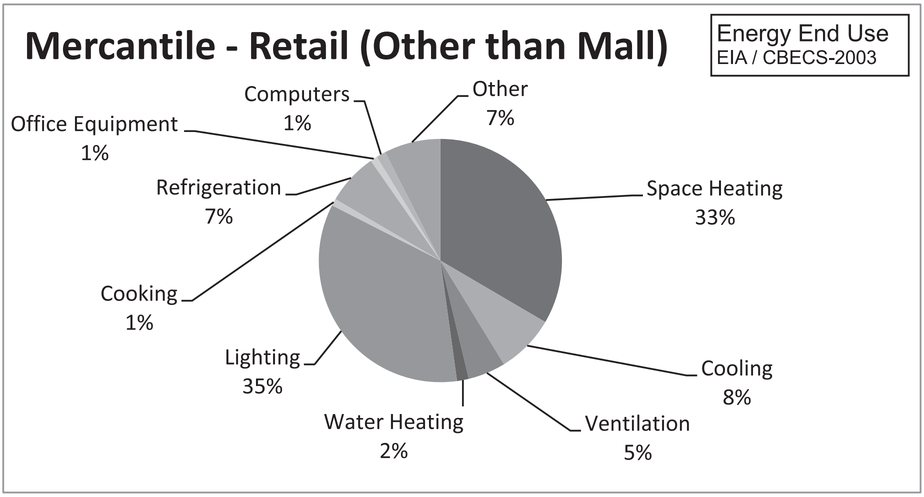

Food Service
Energy End Use
EIA / CBECS-2003
Computers 0%
Other 4%
Office Equipment 0%
Space Heating 17%
Refrigeration 16%
Cooling 7%
Ventilation 6%
Cooking 24%
Water Heating 16%
Lighting 10%

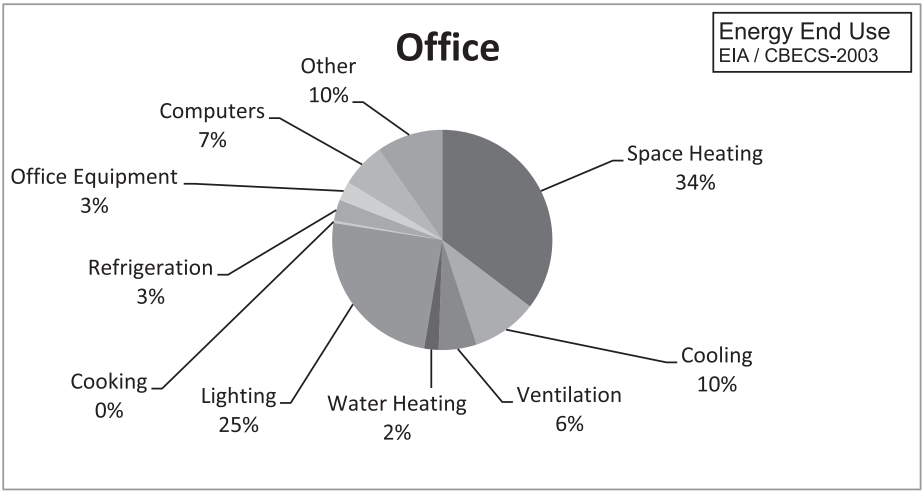

Healthcare - Inpatient
Energy End Use
EIA / CBECS-2003
Office Equipment 1%
Computers 2%
Other 7%
Refrigeration 1%
Space Heating 37%
Cooking 2%
Lighting 16%
Cooling 7%
Water Heating 19%
Ventilation 8%

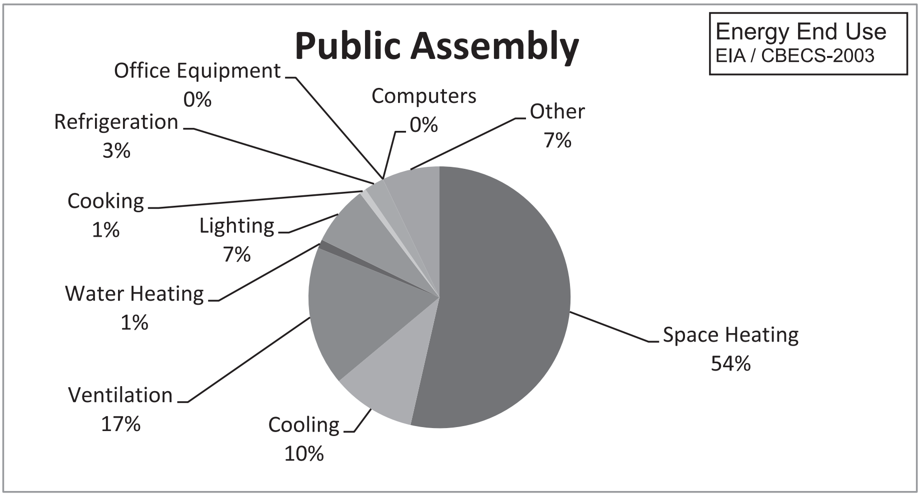

Healthcare - Outpatient
Energy End Use
EIA / CBECS-2003
Computers 3%
Other 14%
Office Equipment 2%
Space Heating 41%
Refrigeration 3%
Cooling 8%
Cooking 0%
Lighting 24%
Water Heating 2%
Ventilation 3%

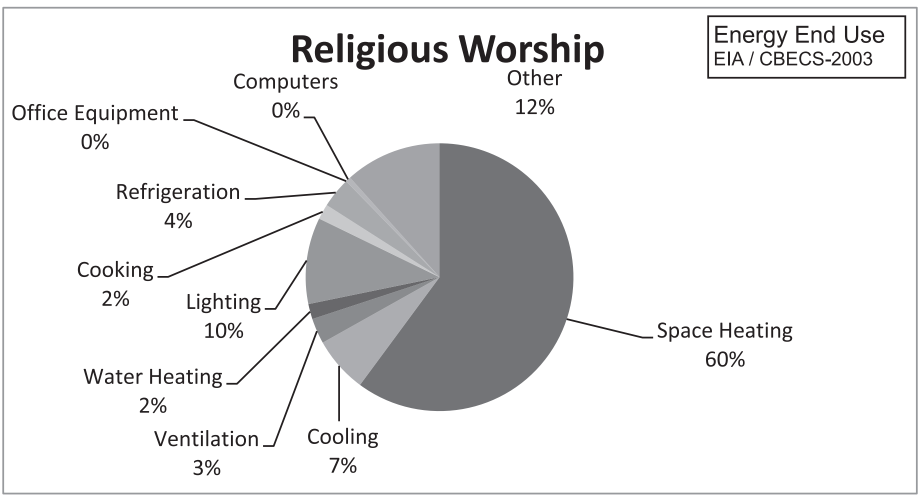

Lodging
Energy End Use
EIA / CBECS-2003
Computers 1%
Office Equipment 0%
Other 7%
Refrigeration 2%
Space Heating 22%
Cooking 3%
Cooling 5%
Ventilation 3%
Lighting 25%
Water Heating 32%

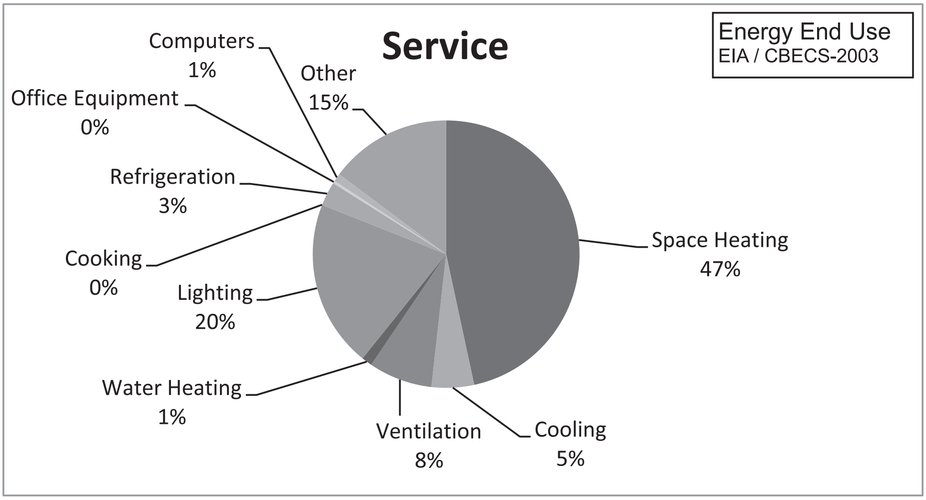

Mercantile - Enclosed and Strip Malls
Energy End Use
EIA / CBECS-2003
Computers 1%
Other 13%
Space Heating 23%
Office Equipment 1%
Refrigeration 4%
Cooling 12%
Cooking 3%
Ventilation 7%
Lighting 28%
Water Heating 8%

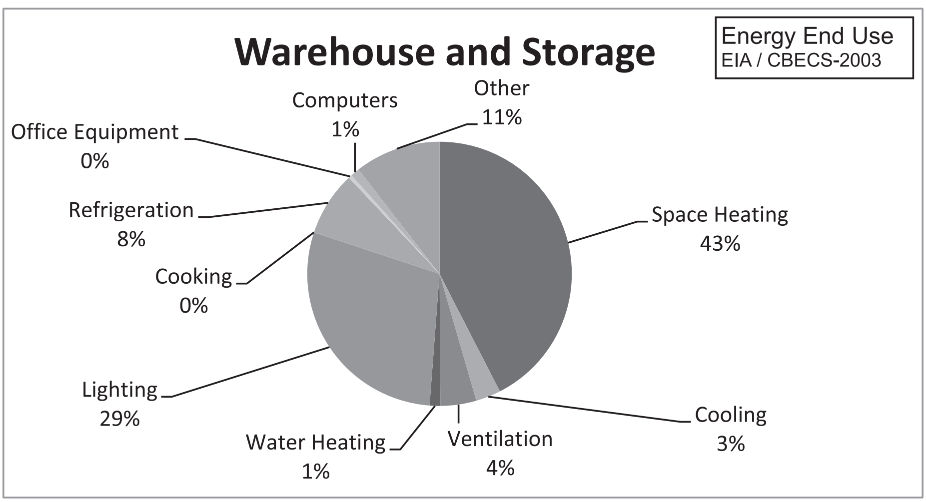

Mercantile - Retail (Other than Mall)
Energy End Use
EIA / CBECS-2003
Computers 1%
Other 7%
Office Equipment 1%
Refrigeration 7%
Space Heating 33%
Cooking 1%
Lighting 35%
Cooling 8%
Water Heating 2%
Ventilation 5%


Office
Energy End Use
EIA / CBECS-2003
Other 10%
Computers 7%
Space Heating 34%
Office Equipment 3%
Refrigeration 3%
Cooling 10%
Cooking 0%
Lighting 25%
Water Heating 2%
Ventilation 6%

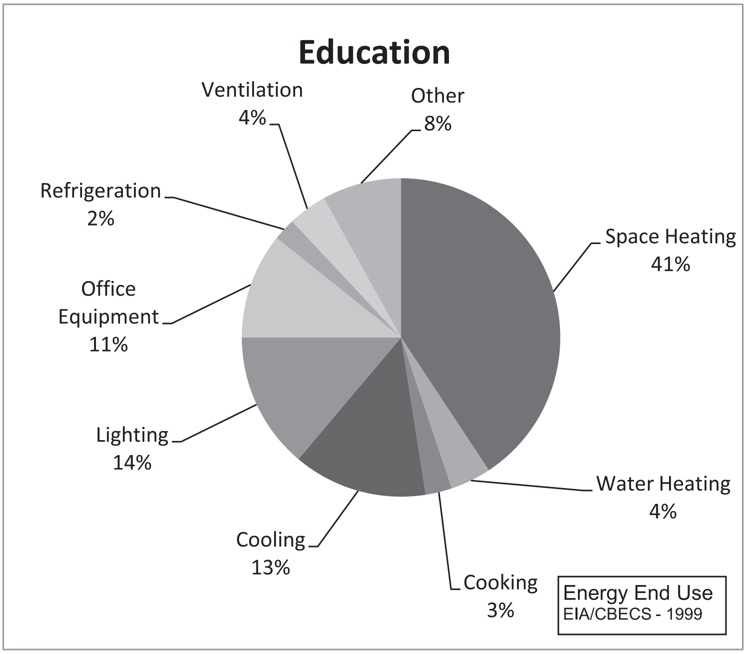

Public Assembly
Energy End Use
EIA / CBECS-2003
Office Equipment
0%
Computers
0%
Other
7%
Refrigeration
3%
Cooking
1%
Lighting
7%
Water Heating
1%
Space Heating
54%
Ventilation
17%
Cooling
10%

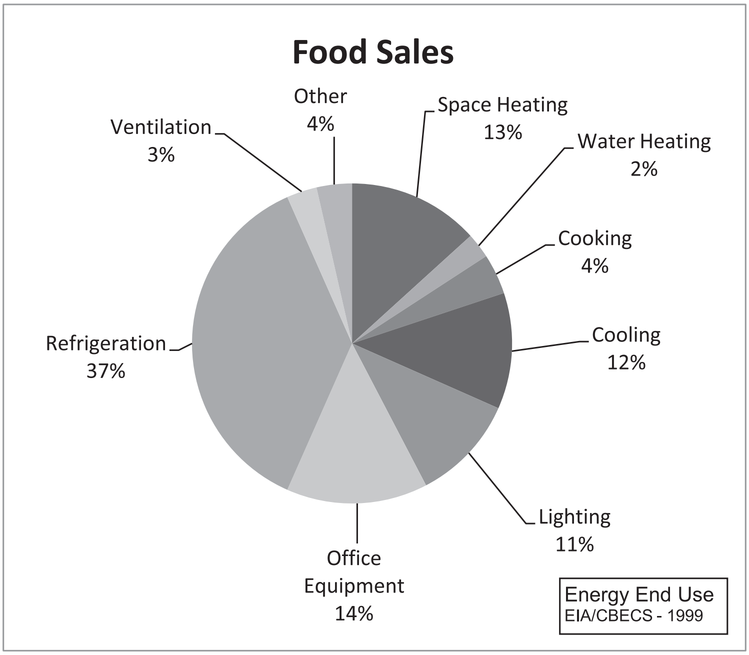

Religious Worship
Energy End Use
EIA / CBECS-2003
Computers
0%
Other
12%
Office Equipment
0%
Refrigeration
4%
Cooking
2%
Lighting
10%
Space Heating
60%
Water Heating
2%
Ventilation
3%
Cooling
7%

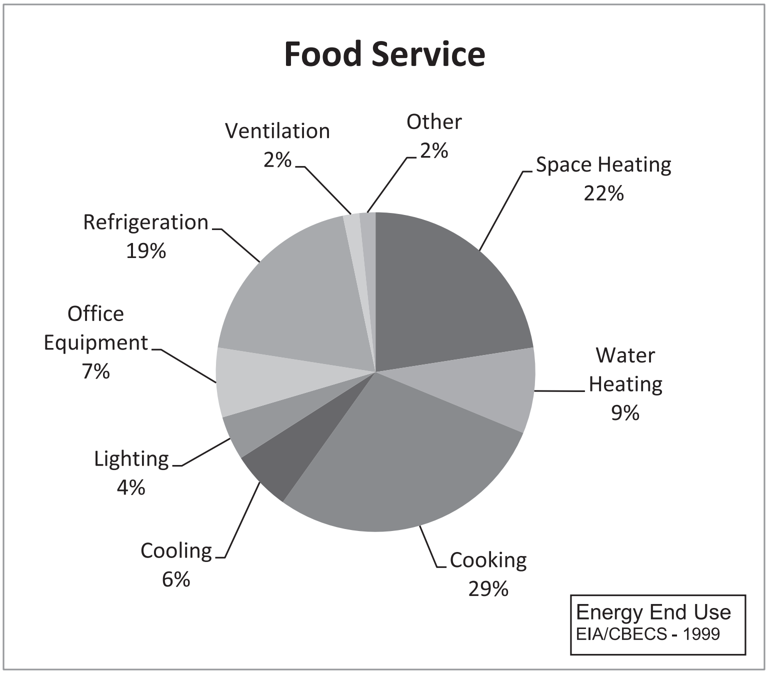

Service
Energy End Use
EIA / CBECS-2003
Computers 1%
Other 15%
Office Equipment 0%
Refrigeration 3%
Cooking 0%
Lighting 20%
Water Heating 1%
Ventilation 8%
Cooling 5%
Space Heating 47%

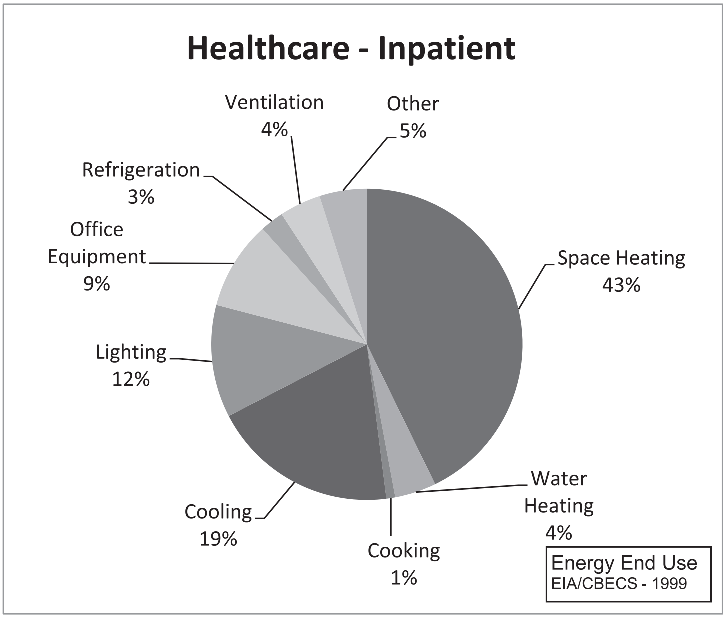

Warehouse and Storage
Energy End Use
EIA / CBECS-2003
Computers 1%
Other 11%
Office Equipment 0%
Refrigeration 8%
Cooking 0%
Lighting 29%
Water Heating 1%
Ventilation 4%
Cooling 3%
Space Heating 43%

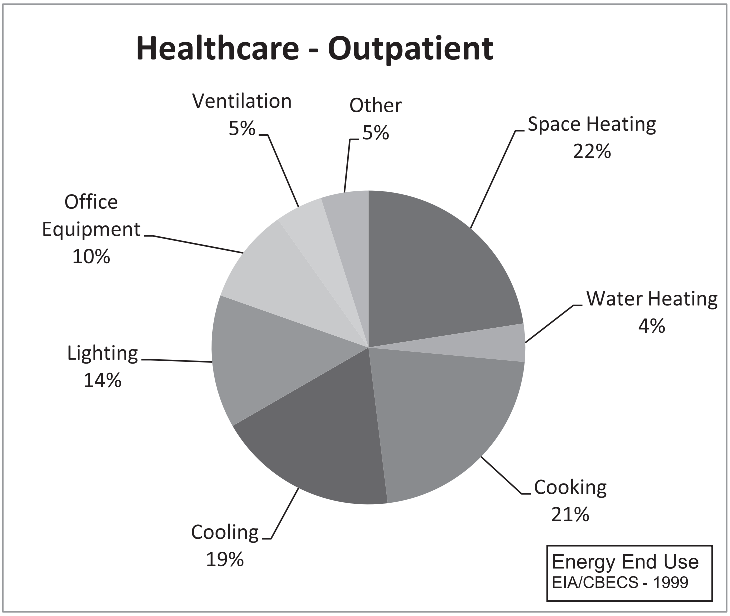
1999 End Use Data

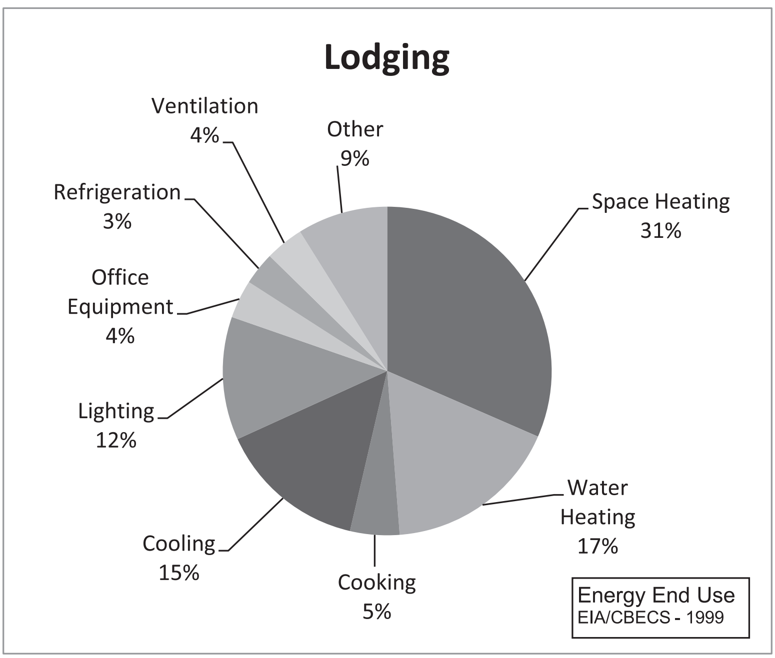


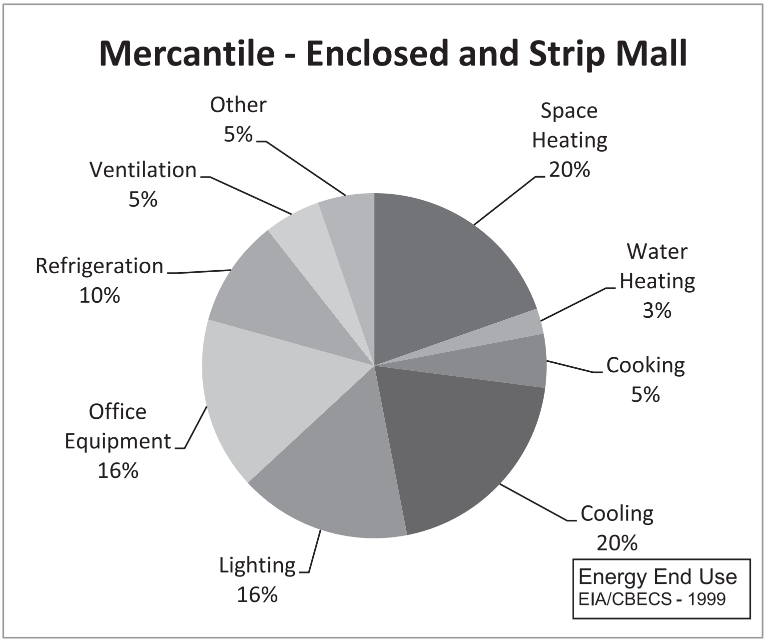


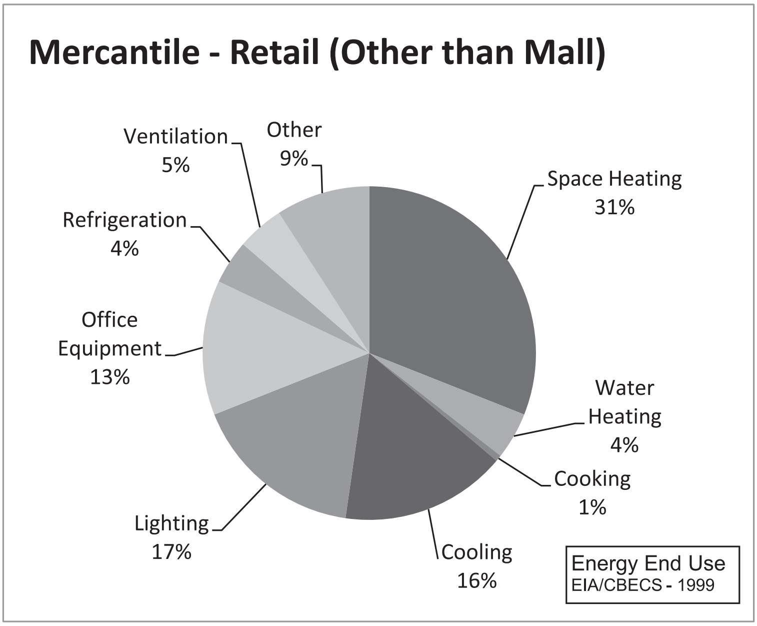

Food Service
Ventilation 2%
Other 2%
Space Heating 22%
Refrigeration 19%
Office Equipment 7%
Water Heating 9%
Lighting 4%
Cooling 6%
Cooking 29%
Energy End Use
EIA/CBECS - 1999

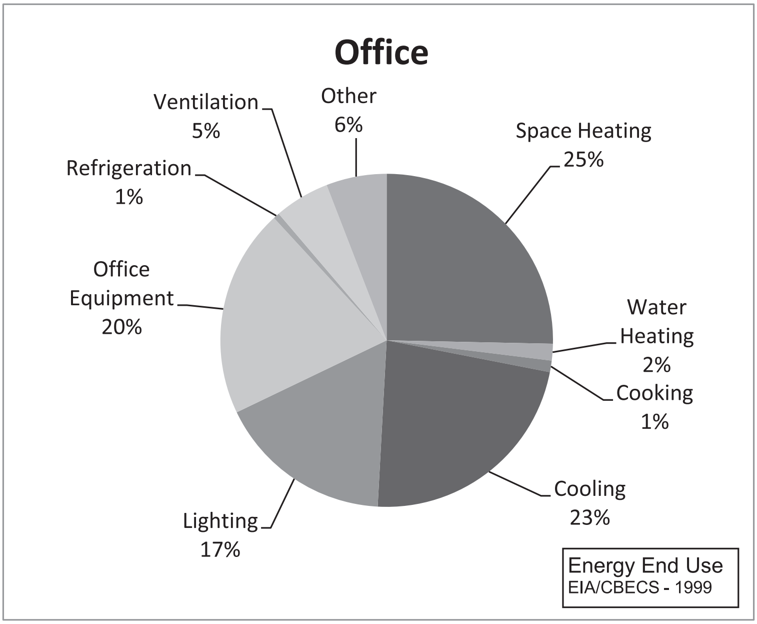

Healthcare - Inpatient
Ventilation 4%
Other 5%
Refrigeration 3%
Office Equipment 9%
Space Heating 43%
Lighting 12%
Water Heating 4%
Cooling 19%
Cooking 1%
Energy End Use
EIA/CBECS - 1999

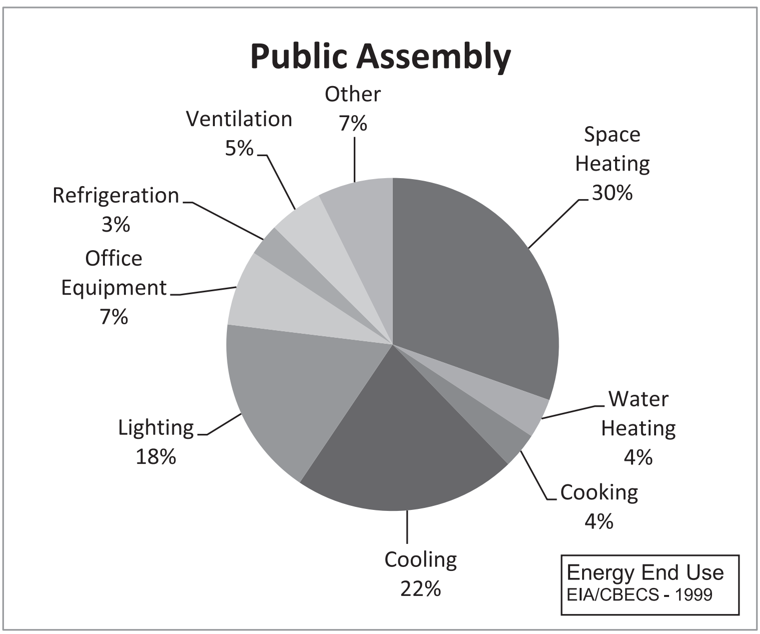

Healthcare - Outpatient
Ventilation 5%
Other 5%
Space Heating 22%
Office Equipment 10%
Water Heating 4%
Lighting 14%
Cooking 21%
Cooling 19%
Energy End Use
EIA/CBECS - 1999

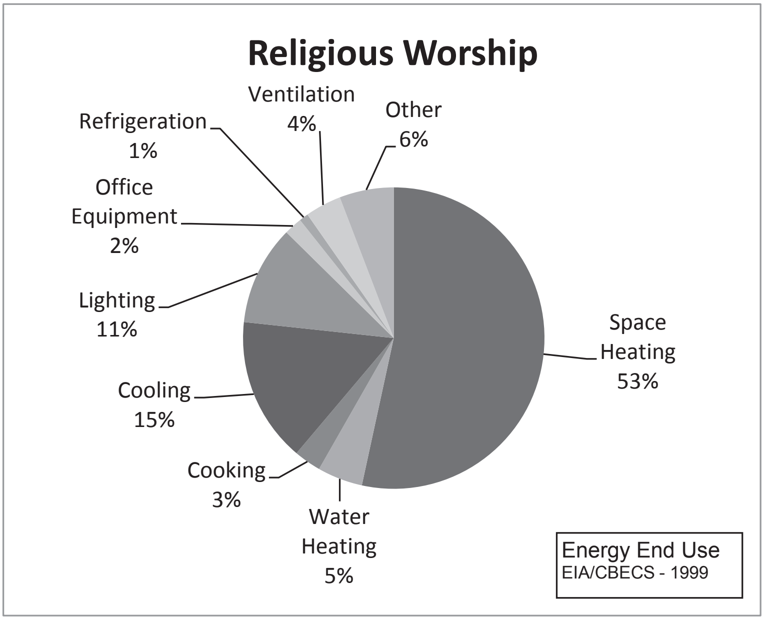

Lodging
Ventilation 4%
Other 9%
Refrigeration 3%
Space Heating 31%
Office Equipment 4%
Lighting 12%
Water Heating 17%
Cooling 15%
Cooking 5%
Energy End Use
EIA/CBECS - 1999

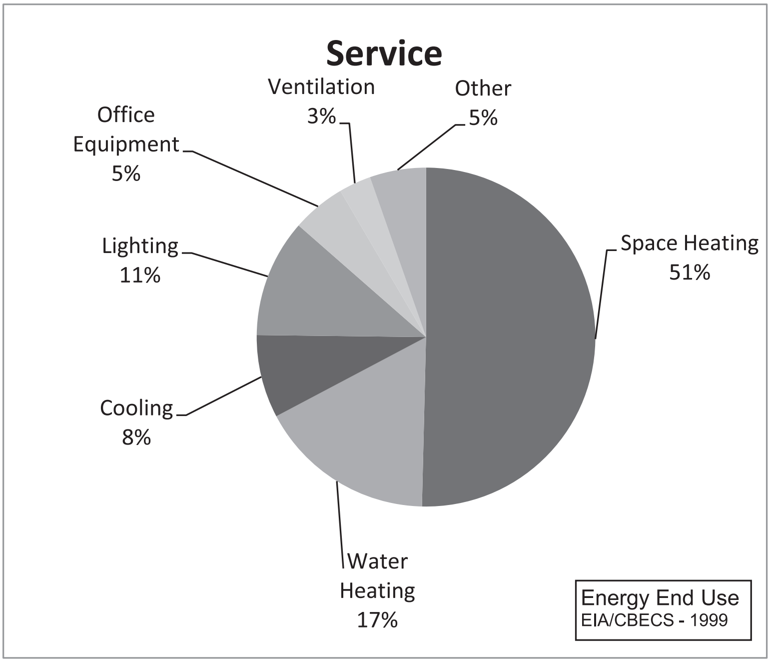

Mercantile - Enclosed and Strip Mall
Space Heating 20%
Water Heating 3%
Cooking 5%
Cooling 20%
Lighting 16%
Office Equipment 16%
Refrigeration 10%
Ventilation 5%
Other 5%
Energy End Use
EIA/CBECS - 1999

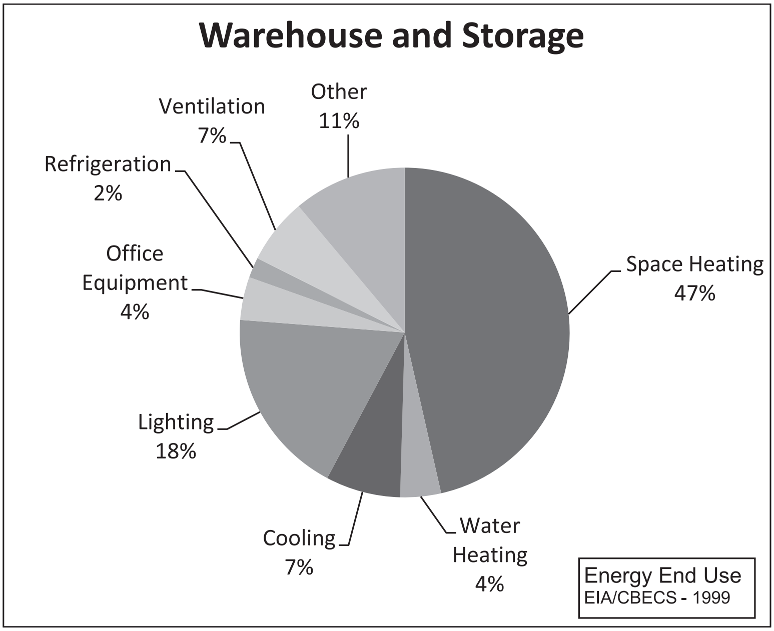

Mercantile - Retail (Other than Mall)
Space Heating 31%
Water Heating 4%
Cooking 1%
Cooling 16%
Lighting 17%
Office Equipment 13%
Refrigeration 4%
Ventilation 5%
Other 9%
Energy End Use
EIA/CBECS - 1999


Office
Ventilation 5%
Other 6%
Space Heating 25%
Refrigeration 1%
Office Equipment 20%
Water Heating 2%
Cooking 1%
Cooling 23%
Lighting 17%
Energy End Use
EIA/CBECS - 1999


Public Assembly
Other 7%
Ventilation 5%
Space Heating 30%
Refrigeration 3%
Office Equipment 7%
Water Heating 4%
Lighting 18%
Cooking 4%
Cooling 22%
Energy End Use
EIA/CBECS - 1999


Religious Worship
Ventilation 4%
Other 6%
Refrigeration 1%
Office Equipment 2%
Lighting 11%
Cooling 15%
Cooking 3%
Water Heating 5%
Space Heating 53%
Energy End Use
EIA/CBECS - 1999


Service
Ventilation 3%
Other 5%
Office Equipment 5%
Lighting 11%
Cooling 8%
Water Heating 17%
Space Heating 51%
Energy End Use
EIA/CBECS - 1999



APARTMENT BUILDINGS/MULTI-FAMILY/ DORMITORIES

Primary Energy Use Sources

- Heating and cooling
- Water heating
- Lighting

Controls

- Controls to lock out central cooling below 50 deg F.
- Controls to lock out central heating above 65 deg F.
- Individual room control in lieu of group zones such as "all north exposure," "all south exposure," etc.
- Reset boiler hot water temperature based on outside air temp.
- Lower domestic hot water temperature to 120 deg F.
- Turn off domestic hot water re-circulation pumps at night.
- Occupancy sensors for lighting in amenity areas (fitness room, etc.).
- Self-contained automatic control valves to replace manual heating valves.

Maintenance

- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes, cleaning apartment refrigerator coils.

- Look for open windows in cold weather as signs of defective controls.
- Repair or replace defective zone control valves (that don't work or don't close tight).
- Keep exterior doors tight fitting—repair or replace seals and sweeps.

Low Cost/No Cost

- Thermostats with a minimum of 4 degrees deadband between heating and cooling.
- Low flow faucets and shower heads.
- Insulate bare hot piping.
- Flyers to encourage energy conservation. Possible shared savings or picnic, etc.

Retrofit—Or Upgrade at Normal Replacement

Heating and Cooling

- Higher efficiency heating and cooling equipment.
- Increase roof insulation thickness to current energy code level as part of roof replacement.
- Window replacement.

Water Heating

- Condensing domestic water heater.
- Stack damper for gas-fired domestic water heater.

Lighting

- Higher efficiency lighting.

Other

- Low water volume/high speed spin wash machines (E-Star).
- Energy Star Appliances on replacement.
- Replace manual heating valves with self-contained automatic control valves.
- Replace manual heating valves with automatic control valves that have set-back capability (remote sensor, not self contained).

May Only Be Viable During New Construction

- Envelope construction to match local residential energy code for insulation.
- Overhangs or other exterior shading at windows.
- Toilet exhaust on demand (light switch) instead of continuous.
- Low volume bathtubs.
- High performance glass on all windows.

Pool: See "Pools" this chapter.
If a boiler is used: See "Boilers" this chapter.
If a chiller is used: See "Chillers" this chapter.
If the building is over four stories in height: See "High Rise" this chapter.

CHURCHES/WORSHIP

Primary Energy Use Sources

- Heating and cooling
- Lighting

Controls

- Scheduled start-stop of HVAC equipment and lighting.
- Control to occupied temperatures during services, allowing temperatures to float higher and lower in other times.
- Morning warm-up before services with outside air closed.
- Demand controlled ventilation for variable occupancy.
- For all roof openings to hoods and equipment not active in winter, dampers should be tightly closed during heating operation and when roof equipment is off.

Maintenance

- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes.
- Verify air economizer controls are functional—annually.

Low Cost/No Cost

- Programmable thermostat.

Retrofit—Or Upgrade At Normal Replacement

Heating and Cooling

- Higher efficiency heating and cooling equipment.
- Anti-stratification fans for high bay, heated areas to move warm air to the floor.
- Reduce outside air to proper quantities if excessive.
- Increase roof insulation thickness to current energy code level as part of roof replacement.

Lighting

- Higher efficiency lighting

May Only Be Viable During New Construction:

- Passive shading elements for large glass areas.
- High performance low E glazing
- Exhaust heat recovery for high ventilation times.
- Vestibule or revolving door.

If a boiler is used: See "Boilers" this chapter.

DATA CENTERS

Primary Energy Use Sources

- Computer equipment
- Cooling
- Lighting (distant third)

See Chapter 24A—Data Center Energy Efficiency

EDUCATION—COLLEGES AND UNIVERSITIES

Primary Energy Use Sources

- Heating and cooling

- Lighting
- Ventilation

Controls
- Scheduled start-stop of HVAC equipment and lighting.
- Compress schedules using occupant 2-hr override button on zone sensors.
- 5 degree deadband between heating and cooling operations.
- Eliminate simultaneous heating and cooling.
- Set space temperatures during unoccupied times up to 85/set back to 60 at night.
- Lock out cooling below 55 degrees and use only economizer.
- Optimum start and stop of primary cooling and heating equipment.
- Demand controlled ventilation for variable occupancy areas such as classrooms and lecture halls.
- For all roof openings to hoods and equipment not active in winter, dampers should be tightly closed during heating operation and when roof equipment is off.
- Turn off domestic hot water circulating pump at night.

Maintenance
- Calibrate controls every five years.
- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes.
- Verify air economizer controls are functional—annually.

Low Cost/No Cost
- Programmable thermostat.
- Occupancy sensors for lighting in classrooms, lecture hall, and labs.
- Photo cell control of parking lot and exterior lighting.
- Global control for "computer monitors off" after 15 minutes of inactivity, instead of screen savers.

Retrofit—Or Upgrade At Normal Replacement

Heating and Cooling
- Higher efficiency heating and cooling equipment.
- Convert constant volume HVAC to VAV.
- Synchronous belts and toothed pulleys for larger motors.

- Anti-stratification fans for high bay, heated areas to move warm air to the floor.
- Dedicated cooling system for 24x7 needs in small areas, to allow the main building HVAC system to shut off at night.
- Reduce outside air to proper quantities if excessive.
- Increase roof insulation thickness to current energy code level as part of roof replacement.

Lighting
- Higher efficiency lighting.
- Reduce excessive light levels by de-lamping.
- Combine de-lamping and reflectors to maintain light levels.
- Photo sensor to turn off lights in areas with substantial glass and ample sunlight.

May Only Be Viable During New Construction
- Light color exterior walls.
- Passive shading elements for large glass areas.
- High performance low-E glazing.
- Exhaust heat recovery for high ventilation areas such as labs.
- Replace flat filters with angled filters.
- Circuiting of lights to allow first 10 feet inboard from the perimeter to be turned off during bright outdoor hours.
- Daylight lighting design.

Dormitories: See "Apartments" this chapter
Cafeteria: See "Food Service" this chapter
Boiler: See "Boilers" this chapter
Chiller: See "Chillers" this chapter
VAV: see "Variable Air Volume Systems" this chapter
If the building is over four stories in height: See "High Rise" this chapter

EDUCATION—SCHOOLS K-12

Primary Energy Use Sources
- Heating and cooling
- Lighting
- Ventilation

Controls

- Scheduled start-stop of HVAC equipment and lighting.
- Compress schedules using occupant 2-hr override button on zone sensors.
- Set space temperatures during unoccupied times up to 85/set back to 60 at night.
- Lock out cooling below 55 degrees and use only economizer.
- Lock out heating above 65 deg F.
- 5 degree deadband between heating and cooling operations.
- Eliminate simultaneous heating and cooling.
- Unit ventilators use ASHRAE Control Cycle 2.
- For all roof openings to hoods and equipment not active in winter, dampers should be tightly closed during heating operation and when roof equipment is off.
- Demand controlled ventilation in assembly areas and other large controllable variable occupancy areas.

Maintenance

- Calibrate controls every five years.
- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes.
- Verify roof hood dampers close tightly.
- Verify air economizer controls are functional—annually.

Low Cost/No Cost

- Programmable Thermostat.
- Occupancy sensors for lighting in classrooms and labs.
- Photo cell control of exterior lighting.
- Global control for "computer monitors off" after 15 minutes of inactivity, instead of screen savers.

Retrofit—Or Upgrade At Normal Replacement

Heating and Cooling

- Higher efficiency heating and cooling equipment.
- Reduce outside air to proper quantities if excessive.
- Synchronous belts and toothed pulleys for larger motors.
- Vestibule at main entry points. Air curtain if vestibule not possible.
- Increase roof insulation thickness to current energy code level as part of roof replacement.

Lighting
- Higher efficiency lighting.
- Reduce excessive light levels by de-lamping.
- Combine de-lamping and reflectors to maintain light levels.
- Photo sensor to turn off lights in areas with substantial glass and ample sunlight.

May Only Be Viable During New Construction
- Light color exterior walls.
- Passive shading elements for large glass areas.
- Exhaust heat recovery for high ventilation areas such as labs.
- Replace flat filters with angled filters.
- Circuiting of lights to allow first 10 feet inboard from the perimeter to be turned off during bright outdoor hours.
- Daylight lighting design.
- Ground source heat pumps.
- Pre-treatment of ventilation air with heat recovery. Ventilation loads are very high in schools.

Cafeteria: See "Food Service" this chapter
Boiler: See "Boilers" this chapter
Chiller: See "Chillers" this chapter
VAV: see "Variable Air Volume Systems" this chapter

FOOD SALES—GROCERY STORES

Primary Energy Use Sources
- Refrigeration
- Lighting
- Heating and cooling

Controls
- Demand controlled ventilation for variable occupancy
- Set back temperatures if not open 24 hours.

Maintenance
- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes.

- Calibrate refrigeration controls every year.
- Verify air economizer controls are functional—annually.

Low Cost/No Cost
- Air curtains at customer entrance and shipping docks.

Retrofit—Or Upgrade At Normal Replacement
Refrigeration
- Higher efficiency refrigeration equipment.
- Convert air-cooled to water-cooled refrigeration equipment.

Heating and Cooling
- Higher efficiency heating and cooling equipment
- Synchronous belts and toothed pulleys for larger motors.
- Increase roof insulation thickness to current energy code level as part of roof replacement.
- Air curtain for entry door.

Lighting
- Higher efficiency lighting.

Other
- Power factor correction for groups of large motors.

May Only Be Viable During New Construction
- Heat recovery from refrigeration equipment for space heat or dehumidification.
- Daylight lighting design.

FOOD SERVICE/RESTAURANTS

Primary Energy Use Sources
- Cooking
- Heating and cooling
- Refrigeration
- Water heating

Kitchen and Equipment

- Organizational change to have food service management accountable for energy cost.
- Run hoods only when needed.
- Control multiple hoods independently and provide variable demand-based make-up air.
- Turn down cooking equipment to standby temperature when not in use.
- Provide controls so the food warming lamps only operate when there is food there to be warmed, instead of always on.
- Limit kitchen cooling to 75 degrees.
- Limit freezer temperature to zero deg F—put ice cream in a separate small chest freezer.
- Limit cooler temperature to 37 deg F.
- Limit hood make-up air tempering to 50 degrees heating.
- Disable the heating for the HVAC units that serve the kitchen area, or lower the setting to only come on below 60 degrees. This is a "cooling only" application and should not require heating.
- Install interlock for the dishwasher hood exhauster so that it only runs when the dishwasher runs.
- Gaskets and door sweeps on walk-in cooler and freezer doors.
- Hanging plastic strips at openings into walk-in coolers and freezers.
- Locate condensing units to reject heat outside, for coolers, freezers, ice makers.
- Gas-fired booster heater.
- Use chemical sanitizer rinse instead of 180 deg F booster heater.
- Variable flow heat hoods (Class 2).
- UL listed hoods for reduced air flows.
- Direct fired hood make-up heating.
- Separate hot and cold food storage equipment.
- Air balance to assure proper hood capture of hot cooking fumes.
- Walk-in freezer and cooler combined so entrance to freezer is from the walk-in cooler.
- Schedule cooking equipment use such that energy use closely tracks production, and will allow proportionally reduced energy cost during slow times.
- Make-up air for hoods: 80% from a dedicated make-up air unit in-

stead of taking air from the building. Air tempering mid-way between outside air and inside conditions.

Controls
- Scheduled start-stop of HVAC equipment and lighting.
- Set space temperatures during unoccupied times up to 85/set back to 60 at night.
- Lock out cooling below 55 degrees and use only economizer.
- Lock out heating above 65 deg F.
- Lower domestic hot water temperature to 120 deg F for hand-washing, and 140 deg F for dish washing, if possible.
- Turn off domestic hot water re-circulation pumps at night.
- (Dining areas) demand controlled ventilation unless tied to kitchen exhaust/make-up system.

Maintenance
- 4 x per year: maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes. Include refrigeration equipment, coolers, and freezers. Clear ice build-up on any freezers without automatic defrost.
- Verify air economizer controls are functional—annually.

Low Cost/No Cost
- Insulate bare hot piping for domestic water.

Retrofit—Or Upgrade At Normal Replacement

Heating and Cooling
- Add hood make-up air system, if none exists.
- Add A/C economizer for kitchen area, if none exists.
- Higher efficiency cooling equipment.
- Down-size HVAC cooling equipment if oversized.
- Evaporative cooling for kitchen and hood make-up.

Water Heating
- Stack damper for gas-fired domestic water heater.
- Condensing domestic water heater.
- Gas-fired booster heater for dishwasher final rinse.
- High efficiency, low water use dishwasher—e.g., counter-flow water recycling type (larger dishwasher).

May Only Be Viable During New Construction

- Variable flow grease hoods (type 1).

HEALTH CARE—HOSPITAL

Primary Energy Use Sources

- Heating and cooling
- Lighting
- Water heating
- Ventilation

Controls

- Set space temperatures during unoccupied times up to 85/set back to 60 at night, for staff areas not continuously occupied.
- 5 degree deadband between heating and cooling operations.
- Eliminate simultaneous heating and cooling. Often many opportunities since cooling and heating probably run year round simultaneously.
- Use outdoor economizer instead of mechanical cooling below 55 deg F, if prescribed space pressurization can be maintained.
- Constant volume HVAC—supply air reset from demand to reduce reheat.
- Avoid humidification if possible.
- Lower domestic hot water temperature to 120 deg F for hand washing and bathing.
- Turn off domestic hot water re-circulation pumps at night to unoccupied areas of the building.

Maintenance

- Calibrate controls every five years.
- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes.
- Verify air economizer controls are functional—annually.

Low Cost/No Cost

- Occupancy sensors for lighting in meeting rooms and day-time office areas.

- Insulate bare hot piping for domestic water.
- Low flow faucets.

Retrofit—Or Upgrade At Normal Replacement

Heating and Cooling

- Higher efficiency heating and cooling equipment.
- Heat recovery from exhaust air, where exhaust is continuous.
- Synchronous belts and toothed pulleys for larger motors.

Water Heating

- Low flow shower heads.
- Condensing domestic water heater.
- Stack damper for gas-fired domestic water heater.

Lighting

- Higher efficiency lighting.
- Reduce excessive light levels by de-lamping.
- Combine de-lamping and reflectors to maintain light levels.

Other

- Power factor correction for groups of large motors.

May Only Be Viable During New Construction

- Replace flat filters with angled filters.
- Water-side economizer to make chilled water with just cooling towers, if air economizers are not used.
- VAV—supply and return box tracking instead of constant volume.
- Separate the water heating systems for 120, 140, and 160 degree hot water.

If a boiler is used: See "Boilers" this chapter

- Boiler stack gas heat recovery.

VAV: see "Variable Air Volume Systems" this chapter

If the building is over four stories in height: See "High Rise" this chapter.

HEALTH CARE—NON HOSPITAL

Primary Energy Use Sources

- Heating and cooling
- Cooking (live-in facilities)
- Lighting

Controls

- Scheduled start-stop of HVAC equipment and lighting.
- Set space temperatures during unoccupied times up to 85/set back to 60 at night.
- 5 degree deadband between heating and cooling operations.
- Eliminate simultaneous heating and cooling.
- Lock out cooling below 55 degrees and use only economizer.
- Lock out heating above 65 deg F.
- Optimum start and stop of primary cooling and heating equipment.
- Constant Volume HVAC—supply air reset from demand to reduce reheat.
- Avoid humidification if possible.
- Eliminate simultaneous humidification/dehumidification.
- Demand controlled ventilation in office areas.

Maintenance

- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes.
- Verify air economizer controls are functional—annually.

Low Cost/No Cost

- Programmable thermostat—unless serving sensitive equipment.
- Occupancy sensors for lighting in meeting rooms and restrooms.

Retrofit—Or Upgrade At Normal Replacement

Heating and Cooling

- Higher efficiency heating and cooling equipment.
- Dedicated small cooling system for 24x7 needs in small areas, to allow the main building HVAC system to shut off at night.
- Reduce outside air to proper quantities if excessive.
- Increase roof insulation thickness to current energy code level as part of roof replacement.

Lighting
- Higher efficiency lighting.
- Reduce excessive light levels by de-lamping.
- Combine de-lamping and reflectors to maintain light levels.

May Only Be Viable During New Construction
- Replace flat filters with angled filters.

Kitchen: See "Food Service" this chapter.
VAV: see "Variable Air Volume Systems" this chapter.

LAUNDRIES—COMMERCIAL

Primary Energy Use Sources
- Water heating
- Heat for tumble drying
- Steam for pressing

Controls
- Turn off domestic hot water re-circulation pumps at night.
- Extend "spin" cycle to reduce remaining moisture in wet clothes. (See **Chapter 5 ECM Descriptions—Laundry/"Remaining Moisture Content" [RMC])**

Maintenance
- Annual cleaning of heat transfer surfaces, including water heaters, heat recovery equipment.

Low Cost/No Cost
- Insulate bare hot piping.
- Reduce water temperature if possible.

Retrofit—Or Upgrade At Normal Replacement
Water Heating
- Heat recovery for waste water, to preheat cold water.
- Heat recovery for hot exhaust from dryers and roller irons, to pre-heat cold water.

- Condensing water heater or direct-contact heater.
- Stack damper for water heater.

Cooling

- Evaporative cooling, spot cooling, or plain exhaust.

May Only Be Viable During New Construction

- Washers that use less water.

Boiler: See "Boilers" this chapter.

LIBRARIES/MUSEUMS

Primary Energy Use Sources

- Heating and cooling
- Lighting
- Humidity Control

Controls

- Scheduled start-stop of lighting even if HVAC must run continuously.
- 5 degree deadband between heating and cooling operations if possible.
- Eliminate simultaneous heating and cooling.
- If air economizer is not allowed due to humidity control requirements, use water economizer or dry cooler below 55 degrees outside temperature.
- Lock out cooling below 50 degrees F.
- Lock out heating above 65 deg F unless dehumidification cycle is needed.
- Constant volume HVAC—supply air reset from demand to reduce reheat.
- Avoid humidification if possible.
- Eliminate simultaneous humidification/dehumidification.
- Demand controlled ventilation for unoccupied periods and variable occupancy.

Maintenance
- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes.
- Calibrate HVAC controls every five years.

Low Cost/No Cost
- Light sensors to harvest daylight near skylights and reading areas near glass.

Retrofit—Or Upgrade At Normal Replacement

Heating and Cooling
- Higher efficiency heating and cooling equipment.
- Install solar screening over large skylights and sun rooms.
- Synchronous belts and toothed pulleys for larger motors.
- Vapor barrier isolation and dedicated HVAC for any areas that are humidified.

Lighting
- Higher efficiency lighting.
- Reduce excessive light levels by de-lamping.

Boiler: See "Boilers" this chapter
Chiller: See "Chillers" this chapter
VAV: see "Variable Air Volume Systems" this chapter

LODGING/HOTELS/MOTELS

Primary Energy Use Sources
- Heating and cooling
- Lighting
- Water heating

Controls
- Scheduled start-stop of HVAC equipment and lighting in staff areas and other areas without continuous guest access.
- Set space temperatures during unoccupied times up to 85/set back

to 60 at night, for staff areas not continuously occupied and amenity areas without continuous guest access.

- Lock out cooling below 55 degrees and use only economizer (common and amenity areas).
- Lock out heating above 65 deg F (common and amenity areas).
- Eliminate overlapping heating and cooling in common area HVAC equipment by adjusting control settings to have a "deadband."
- Lower domestic hot water temperature to 120 deg F for hand washing and bathing, and 140 deg F for dish washing, if possible.
- Turn off domestic hot water re-circulation pumps at night.
- Guest room occupancy controls to shut off lights and set back HVAC.

Larger Facilities

- Demand controlled ventilation for variable occupancy areas such as meeting rooms, ballrooms, and common areas.
- Avoid over-cooling large meeting rooms and ballrooms. Limit these space temperatures to 70 degrees.
- Close down whole buildings, wings, or floors in slow seasons.

Maintenance

- Calibrate controls every five years.
- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes.
- Repair or replace defective zone control valves.
- Verify air economizer controls are functional—annually.

Low Cost/No Cost

- Insulate bare hot piping for domestic water.
- Use cog belts instead of standard V-belts.
- Install solar screening over large skylights and sun rooms.
- Occupancy sensors for lighting in amenity areas and meeting rooms.
- Close PTAC "vent" function if corridor ventilation is adequate.
- Low flow shower heads.

Retrofit—Or Upgrade At Normal Replacement

Heating and Cooling

- Higher efficiency heating and cooling equipment.
- Higher efficiency guest room package terminal unit air conditioners (PTACs).
- In mild climates, heat pump PTACs instead of electric heat.

- Increase roof insulation thickness to current energy code level as part of roof replacement.

Water Heating
- Stack damper for gas-fired domestic water heater.
- Condensing domestic water heater.

Lighting
- Higher efficiency lighting.
- Reduce excessive light levels by de-lamping.
- Combine de-lamping and reflectors to maintain light levels.

Larger Facilities
- Install solar screening over large skylights and sun rooms.
- Convert constant volume HVAC to VAV.
- Synchronous belts and toothed pulleys for larger motors.
- Power factor correction for groups of large motors.
- Dedicated cooling system for 24x7 needs in small areas, to allow the main building HVAC system to shut off at night.
- Reduce outside air to proper quantities if excessive.
- Low water volume/high speed spin wash machines (Energy Star).

May Only Be Viable During New Construction
- Envelope construction in guest areas to match local residential energy code for insulation.
- Overhangs or other exterior shading at windows.
- Toilet exhaust on demand (light switch) instead of continuous.
- Low volume bathtubs.
- Separate the water heating systems for 120, 140, and 160 degree hot water.
- Gas-fired PTACs instead of electric resistance.
- Ground source heat pumps.

Kitchen: See “Food Service” this chapter
Pool: See “Pools” this chapter
Boiler: See “Boilers” this chapter
Chiller: See “Chillers” this chapter
VAV: see “Variable Air Volume Systems” this chapter
If the building is over four stories in height: See “High Rise” this chapter

OFFICE BUILDINGS

Primary Energy Use Sources
- Heating and cooling
- Lighting
- Office equipment

Controls
- Scheduled start-stop of HVAC equipment and lighting.
- Set space temperatures during unoccupied times up to 85/set back to 60 at night.
- 5 degree deadband between heating and cooling operations.
- Eliminate simultaneous heating and cooling.
- Lock out cooling below 55 degrees and use only economizer.
- Lock out heating above 65 deg F.

Larger Facilities
- Optimum start and stop of primary cooling and heating equipment.
- Stage electric heating to prevent setting seasonal demand and invoking ratchet charges.
- Demand controlled ventilation for variable occupancy areas such as conference rooms and open plan office areas.

Maintenance
- Calibrate controls every five years.
- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes.
- Verify air economizer controls are functional—annually.
- Seal return air plenums against any leakage. Verify plenum temperature is within 2 degrees F of room temperature in summer and winter design days.

Low Cost/No Cost
- Occupancy sensors for lighting in conference rooms and meeting rooms.
- Global control for "computer monitors off" after 15 minutes of inactivity, instead of screen savers.

Retrofit—Or Upgrade At Normal Replacement
Heating and Cooling
- Higher efficiency heating and cooling equipment.

- Convert constant volume HVAC to VAV.
- Reduce outside air to proper quantities if excessive.
- Increase roof insulation thickness to current energy code level as part of roof replacement.

Lighting
- Higher efficiency lighting.
- Reduce excessive light levels by de-lamping.
- Combine de-lamping and reflectors to maintain light levels.

Larger Facilities
- Synchronous belts and toothed pulleys for larger motors.
- Power factor correction for groups of large motors.
- Dedicated cooling system for 24x7 needs in small areas, to allow the main building HVAC system to shut off at night.
- Light colored interior shades for buildings with large amounts of glass.

May Only Be Viable During New Construction
- Overhangs or exterior shading for glazing.
- Light color exterior walls.
- Circuiting of lights to allow first 10 feet inboard from the perimeter to be turned off during bright outdoor hours.

Data center: See **Chapter 24, "Data Center Energy Efficiency"**
Boiler: See "Boilers" this chapter
Chiller: see "Chillers" this chapter
VAV: see "Variable Air Volume Systems" this chapter
If the building is over four stories in height: See "High Rise" this chapter

RETAIL/SALES

Primary Energy Use Sources
- Heating and cooling
- Lighting

Controls
- Scheduled start-stop of HVAC equipment and lighting.

- Set space temperatures during unoccupied times up to 85/set back to 60 at night.
- Lock out cooling below 55 degrees and use only economizer.
- Lock out heating above 65 deg F.
- Maintain a 5 deg F deadband between heating and cooling for all HVAC equipment.
- Turn off display lighting except during customer times.

Larger Facilities
- Optimum start and stop of primary cooling and heating equipment.
- Demand controlled ventilation for variable occupancy times.
- Photo cell control of parking lot lighting.
- For all roof openings to hoods and equipment not active in winter, dampers should be tightly closed during heating operation and when roof equipment is off.

Maintenance
- Calibrate controls every five years.
- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes.
- Verify air economizer controls are functional—annually.

Low Cost/No Cost
- Occupancy sensors for lighting in break rooms.
- Install solar screening over large skylights.

Retrofit—Or Upgrade At Normal Replacement
Heating and Cooling
- Higher efficiency heating and cooling equipment.
- HVAC air economizer, lock out mechanical cooling below 55 deg F.
- Increase roof insulation thickness to current energy code level as part of roof replacement.

Lighting
- Higher efficiency lighting.
- Reduce excessive light levels by de-lamping.
- Combine de-lamping and reflectors to maintain light levels.

Larger Facilities

- High bay lighting retrofit, HID to fluorescent.
- Convert constant volume HVAC to VAV.
- Synchronous belts and toothed pulleys for larger motors.
- Reduce outside air to proper quantities if excessive.

May Only Be Viable During New Construction

- Overhang over storefront window.
- HVAC variable volume air systems.
- Daylight lighting design.

Boiler: see “Boilers” this chapter
Chiller: see “Chillers” this chapter
VAV: see “Variable Air Volume Systems” this chapter
If the building is over four stories in height: see “High Rise” this chapter

WAREHOUSES

Primary Energy Use Sources

- Heating
- Lighting

Controls

- Occupancy sensors to turn off lights in unmanned areas and during unoccupied times.
- Set space temperatures to 80 cooling/60 heating unless product storage requires closer temperature control.
- Set space temperatures during unoccupied times up to 90/set back to 50 at night unless product storage requires closer temperature control.
- For all roof openings to hoods and equipment not active in winter, dampers should be tightly closed during heating operation and when roof equipment is off.

Maintenance

- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils.

Low Cost/No Cost
- Interlock heating and cooling at loading docks to stop when roll-up doors are opened.

Refrigerated Warehouse
- Hanging plastic strips at openings into refrigerated areas.
- Air curtains at openings into refrigerated areas.

Retrofit—Or Upgrade At Normal Replacement:
Heating and Cooling
- Higher efficiency heating equipment.
- Gas-fired radiant heating in lieu of space heating.
- Anti-stratification fans for high bay, heated areas to move warm air to the floor.
- Increase roof insulation thickness to current energy code level as part of roof replacement, for buildings that are heated or cooled.
- Air exchange fans or swamp coolers instead of air conditioning.

Lighting
- High bay lighting retrofit HID to fluorescent.
- Higher efficiency lighting.
- Motion sensors for overhead lights, except re-strike time makes this impractical for HID lighting.
- Reduce excessive light levels by de-lamping.
- Combine de-lamping and reflectors to maintain light levels.

Refrigerated Warehouse
- Higher efficiency lighting to reduce refrigeration load.
- Convert air-cooled to water-cooled refrigeration.
- Increase roof insulation thickness over refrigerated areas.
- Air curtains, hanging plastic strips, or quick moving barriers to keep cold air from escaping.

May Only Be Viable During New Construction
- Improved insulation for buildings intended to be heated or cooled

Refrigerated Warehouse
- Improve envelope insulation
- Light color exterior walls for refrigerated areas.
- Cool roof for refrigerated areas.
- Daylight lighting design.

POOLS

Primary Energy Use Sources

- Pool water heating
- Pool air heating and dehumidifying
- Shower water heating

Controls

- Maintain air temperature within 2 degrees above water temperature.
- Maintain air humidity above between 50-60%, do not over dry the air through excessive ventilation or other means.
- Except in very humid climates, do not dehumidify with simultaneous heating and cooling
- Reduce pool water temperature overall.
- Relax pool water temperature, air temperature, and humidity requirements during unoccupied times. Allow water to cool a few degrees at night if possible.
- Reduce air flow rates and air exchange rates in unoccupied periods.
- For all roof openings to hoods and equipment not active in winter, dampers should be tightly closed during heating operation and when roof equipment is off.

Maintenance

- Calibrate controls every two years.
- Annual efficiency checks for gas fired heaters.
- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes.
- Look for condensation on interior walls during winter that indicate poor insulation and high heat loss.

Low Cost/No Cost

- Drain outdoor pools in winter instead of heating.

Retrofit—Or Upgrade at Normal Replacement

Pool Water Heating

- Pool covers for indoor and outdoor pools.
- Condensing water heater.

- Solar water heater

Pool Air Heating and Dehumidifying
- Reduce outside air to proper quantities if excessive.
- Heat recovery from exhaust to make-up air.

Shower Water Heating
- Low flow shower heads.

May Only Be Viable During New Construction
- Replace mechanical cooling system with ventilation system in climates with acceptable summer humidity levels.
- For indoor pools, add a heating system for the surrounding air if there isn't one, to eliminate the pool from heating it.

ICE RINKS

Primary Energy Use Sources
- Refrigeration
- Space Heating
- Lighting

Controls
- Stage brine pumps proportionally with load. Avoid excessive pumping which is, itself, a significant heat load to the ice making equipment.
- Control ice sheet temperature no lower than necessary.
- Control brine temperature no lower than necessary.
- Control condenser water temperature as low as possible to reduce head pressure.
- Turn off heaters in ice areas in unoccupied times.

Maintenance
- Calibrate controls every two years.
- Annual efficiency checks for gas fired heaters.
- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes.

Low Cost/No Cost
- Schedule ice-building times to use off-peak utility rates.

- Hanging barriers to keep warmed air for spectators from heating the ceiling over the ice slab.
- Window shades for glazing at ice slabs.
- Re-direct warm air heating supply to spectators but not at or over the ice sheet.

Retrofit—Or Upgrade at Normal Replacement

Refrigeration

- Replace conventional expansion valve with electronic expansion valve to allow colder condenser water, lower head pressures.
- Higher efficiency refrigeration equipment.
- Synchronous belts and toothed pulleys for larger motors.
- Higher efficiency cooling tower with ample capacity (smaller fan motor)
- Variable speed brine pumping.
- Heat recovery from refrigeration system for space heating, domestic water heating, or ice melting.
- Light colored roof over ice sheet.
- Increased wall and roof insulation around ice sheet.
- Increased insulation under ice sheet.
- Low-emissivity paint in interior roof over ice sheet.
- Heavy tint or frittered glazing near ice sheet.
- Switch to ethylene glycol if propylene glycol is being used.
- High efficiency lighting over ice sheet.
- Barriers to keep heated air over spectators from heating the ice sheet.
- Power factor correction.

Space Heating

- Higher efficiency heating equipment
- Radiant heat for spectators instead of air heating.

Lighting

- Higher efficiency lighting has double effect by reduced load on the ice sheet.

SPECIFIC SUB-SYSTEMS

BOILERS

- Higher efficiency equipment upon replacement.
- Boiler construction and control that allows low automatic reset to 140 degF or less supply water temperature. This may include air system heating coil replacement to allow deep reset.
- Controls to lock out boiler above 60 deg F.
- For multiple boilers, sequencing to maintain >50% load at all times.
- Reset boiler temperature from outside air.
- Ventilation louver/damper controls for when boiler is off.
- Direct venting of burner combustion air inlet instead of large louver openings into the room.
- Annual verification that hydronic heating and cooling automatic control valves close tightly and prevent any internal leak-by.
- Modular boilers to reduce standby losses, upon replacement.
- Jockey boiler for unavoidable summer reheat loads, e.g. 100 degF water.
- Stack dampers for single boilers that cycle frequently during part load.
- VFDs for FD and ID fans, feed pumps, and other auxiliaries so this energy is kept in proportion to boiler load.
- Annual efficiency checking. Take corrective action if efficiency is found to be less than 95% of new equipment values.
- Automatic boiler isolation valves if piping allows hot water through an "off" standby boiler.
- Insulate bare heating water piping.
- Maintain proper water treatment and ensure that normal make-up does not cause dilution and scaling.
- Convert constant flow heating water to variable flow for larger systems with high annual run hours.
- Separate domestic water heating equipment, instead of using heating boiler during summer for this purpose.
- Insulate boiler surface areas and access panels that have no casing insulation.
- Reduce excess air for burner to 30% or less at all loads. Ideally, separate the control of air and fuel and O_2 trim.
- Convert steam heat to hot water heat to allow part load hot water reset.

CHILLERS

- Higher efficiency equipment upon replacement.
- Variable chilled water pumping instead of constant flow, so this auxiliary energy is kept in proportion to cooling load.
- Increase chilled water system differential to reduce variable pumping costs. May require modification of air system cooling coils and controls.
- Lower condenser water temperature for water-cooled chiller.
- Higher efficiency/capacity cooling tower upon replacement. Suggested criteria for replacement cooling tower is a maximum design approach of 7 degrees to design wet bulb conditions and a fan power budget of no more than 0.05 kW/ton.
- VFDs for cooling tower fans.
- Control multiple cooling towers as one tower to reduce fan energy.
- Annual condenser tube cleaning for water-cooled chiller.
- Controls to lock out the chiller below 50 deg F.
- For multiple chillers, sequencing to maintain >50% load at all times.
- Raise chilled water temperature if possible, but use caution when dehumidification is needed (loss of dehumidification effect).
- Convert constant flow chilled water to variable flow for larger systems with high annual run hours.
- Insulate bare chilled water piping.
- Correct air recirculation for cooling towers or air-cooled chillers within enclosures.

HIGH RISE

Envelope
- Entry door vestibule or revolving door.
- Seal around windows.

Stack Effect
- Seal vertical shafts air tight.
- Seal air plenums air tight.

- Motorized damper to close elevator shaft except in fire mode, if allowed by local building regulations.

VARIABLE AIR VOLUME SYSTEMS

Controls
- Supply air reset from outside air instead of space or return temperature. Allow warmer supply air in heating season, to reduce the reheat penalty, but no reset during cooling season.
- Reduce supply duct static pressure.
- Reset supply duct static pressure from zone demand or outside air, using less fan power in winter instead of constant.
- Verify VAV box minimums are appropriately low and that the "heating minimum CFM" is no higher than the cooling minimum CFM, e.g. air flows stay at minimum during heating mode other than morning warm-up.
- Intermittent use areas, such as meeting rooms, conference rooms, classrooms, and lecture halls with dedicated VAV boxes: Occupancy sensor to reset VAV box minimum setting closed when unoccupied for over 15 minutes.

Maintenance
- Seal return air plenums against any leakage to outside.
- Verify plenum temperature is within 2 degrees of room temperature in summer and winter design days. Summer plenum temperature equal to or lower than the space temperature indicates duct leakage.

Retrofit—Or Upgrade At Normal Replacement
- VFD instead of other modulation methods.
- Balance return air drops at each floor to prevent induced air movements through building cavities.
- Parallel style VAV fan boxes so the fans only operate in heating mode.

Chapter 4

Manufacturing and Unit Operations

INTRODUCTION

Manufacturing is often called a business sector, but is really a collection of dozens of other sectors. This text does not attempt to discuss each manufacturing sub-sector in detail, however some generalities are provided that may be useful.

Some manufacturing processes are very energy intensive, while others are not. Therefore, the effect of weather can vary from substantial to insignificant. More often, the energy use of a manufacturing facility is driven by the process and the monthly energy use variations are strongly dependent on production volumes.

Manufacturing makes things, from other things. The things can be seen as ingredients, and energy is one of them. In this light, the energy cost should go up and down with production and is a controllable cost, not an overhead cost. It follows then that there should be enough but just enough of this ingredient, and its cost should be managed. Further, it should be understood that excess energy expense beyond what is needed either drives up the cost of the finished product or erodes the profit from sales.

This section presumes the manufacturing company is interested in reducing energy use to save money and improve profit margins, but this may not always be the case. The combination of *profit margin* and *percent of total operating cost that is from energy expense* provides an indicator of how motivated the business will likely be for pursuing energy reduction. See **Appendix: Operating Expenses: Percent that are from Utility Costs (Manufacturing)**. Many manufacturing endeavors report utility costs at 2 or 3% of total operating costs. When energy cost is small portion compared to total operating cost the interest in reducing it may also be small. But there is another aspect: profit margin. See **Appendix: Equating Energy Savings to Profit Increase**. Note that when energy cost is 3% of operating cost and profit margin is <u>10%</u>, a 30% reduction in energy use improves profitability from 10% to 11% (10% more profit); however if the sales profit margin is only <u>1%</u>, the profitability would improve from 1% to 1.9% (90% more profit). Thus, a proposal to reduce energy cost can have different value to different businesses, and the lower the current profit margin the greater the profit margin improvement for a given operating cost reduction.

When pursuing cost reductions from energy or any other expense, it is helpful to separate expenses that are necessary from expenses that are from waste.

COMMON THEMES IN APPROACHING PROCESS EFFICIENCY

- The level of effort for achieving energy savings makes sense to be proportional to the expense. If energy cost is trivial, savings potential is trivial and the amount of expense applied to create savings should be trivial.
- Energy is a controllable expense, not an uncontrolled overhead expense.
- Energy use occurs in process, process-related shared systems, and building services.
- For unit operations, the goal is to have actual energy use very close to ideal.
- For all energy uses, the goal is to provide enough, but just enough of what is needed.
- Energy use applied as an ingredient will ideally reduce proportionally with capacity (so 50% of output equates to 50% of energy requirement); throttling and standby losses should be minimized.
- Energy improvements can focus on equipment efficiency or process efficiency.
- Measures that reduce productivity or impact quality should be excluded. Measures that promote productivity have additional value.

Overarching Approaches to Reducing Manufacturing Energy Cost

(These can be combined)

- **Lower cost energy supply.** This treats the process and support services as untouchable—so the focus is finding lower cost energy supplies. De-regulation is servant to this approach along with wheeling and transportation rates (tolls for transporting energy from a remote source through a territory's pipes or wires). This also includes mining utility bills for errors and negotiating lower rates.

This is a valid approach, but stopping here leaves untapped savings potential in process and support service energy use.

- **Review non-process areas only.** This presumes the process is untouchable, limiting the review to process-related support services and building services. Example measures could be lighting, boilers and process heat, compressed air, process cooling, motor and drive efficiency, compressed air leaks, steam leaks, oven doors, maintenance practices, etc. This is a very common energy audit approach that can bring good results while not risking impact to the process. The weakness is lack of optimization in the process itself; i.e. measures that result in needing less energy per unit of production to begin with.
- **Process review.** This evaluates the process itself, as the end use point for all of the process-related support services, building services, and utility expenses. Where measures are found to use less to begin with, energy use and cost reduce proportionally.

METRICS: MACRO VIEW

Macro views identify the overall energy use and cost per unit of production:

Manufacturing efficacy = total energy use (cost)/units of production **(eq. 1)**

Comparison metrics may be "what it was last year," sister plants doing the same kinds of things, or reported usage for peers in the same industry. Reference **Figure 4-1 and Process Efficiency Example #1.** Macro view looks at the aggregate performance as a whole—the collective effect of all individual processes and all energy use associated with the plant combined. It is the view held by upper management—dollars in and dollars out—and an appropriate view. Metrics like Btu per pound, kWh per million gallons, dollars per ton, are all viable. For numerical analysis, the entire plant is inside a box with expenses being injected while product comes out the other end. Energy input data comes from the bulk meters, typically the utility billing meters.

Energy per Unit of Production = Total Energy Used/Total Units Produced **(eq. 2)**

Advantages: This metric is easy to implement and makes it easy to gage yearly, monthly, daily plant efficiency. Quantification of macro performance is a simple ratio of production vs. utility inputs. This method has the advantage of capturing all energy uses and makes a good 'needle' metric from which to manage. Low production inefficiencies are easily spotted - where energy use per unit production becomes disproportionate when output is low. Allows comparison to similar processes on an overall basis.

Disadvantages: The macro view by itself cannot determine how much waste there is. This method does not provide the magnitude of potential savings for individual or overall process (how much more efficient could we be?).

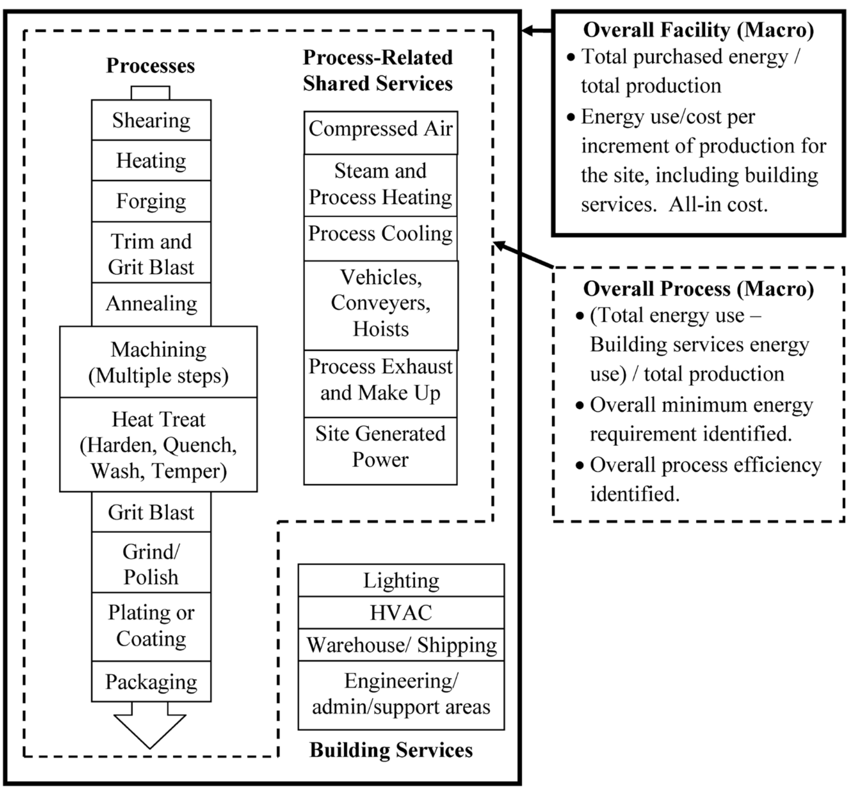


Figure 4-1. Macro View of Energy Use in Manufacturing

<u>Additional remarks</u>: The macro view approach can also be applied to individual plant areas (e.g. heat treatment area vs. plating) to provide energy management feedback. The same concepts apply (energy use per unit of production) but the energy data may require private sub metering.

Process Efficiency Example #1: Macro View (Overall Performance)
Values used to illustrate the method are not an actual case study.

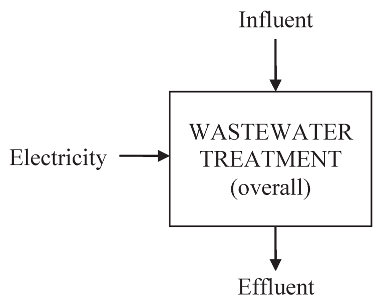
A facility processes waste water on a continuous basis. Some of the individual process steps are more energy intensive than others, but none of the steps are measured for energy input. The blowers are assumed to be the largest single user of energy. There are three equal size blowers that constantly add air to the process and they are sequenced in 1-2-3 fashion based on need. The blowers are not sub metered but a load profile has been estimated from operational patterns. Electricity is used by a multitude of equipment items as well as facility lighting and staff areas, and total electric usage is known from monthly utility bills. The monthly throughput of waste water is known. Analyze the <u>macro</u> view (overall performance) of this process in kWh/Mgal (million gallons) of finished product and compare to available industry data for similar plants that is 1500 kWh per MGal. Estimate the portion of total energy use from the blowers.

<u>Additional Data:</u>
- Period is one month, 720 hours
- Electrical use for the period was 946,000 kWh
- Influent volume for the period that was treated was 500 Mgal
- Blower load profile for the period was
  - — 1500 kW 5% of the time
  - — 1000 kW 40% of the time
  - — 500 kW 55% of the time

METRICS: MICRO VIEW

Mass and Energy Balancing for Unit Operations (Micro)

Conservation of mass and conservation of energy state simply that what goes into a process must come out of it unless there is some accumulation. The analytical tool for balancing energy and mass is to draw a control volume 'box' around the process. **Figure 4-2** is from a study done



Overall Plant Performance

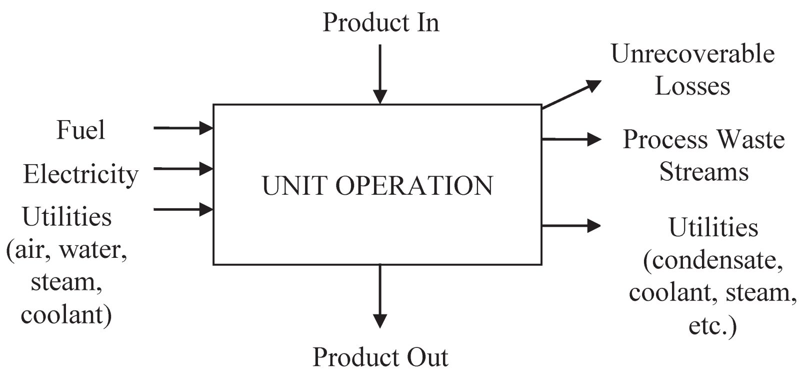
kWh	946,000	
Mgal	500	
kWh/Mgal	**1,892**	
kWh/Mgal	1,500	benchmark

hours/ month	720	**Blowers**

% time	hours	kW	kWh	% from blowers
5%	36	1,500	54,000	
40%	288	1,000	288,000	
55%	396	500	198,000	
100%	720		540,000	**57%**
check	check			

Process Efficiency Example 1: Macro View (Solution)

in the United States for prominent manufacturing processes; **Figure 4-3** is a more generalized diagram for unit operations. This concept can be used to analyze both batch and continuous operations; continuous operation requires a defined time period. Material balances are important for processing, accounting for input ingredients in the 'recipe', and final output yield. Similarly, energy balances account for the energy streams involved in the process. Mass and energy balances can be combined into the same control volume and it is often convenient to do so. See **Suggested References** at the front of this book for a good source of supplemental information on process diagramming and mass-energy balance methods.



Unit Operation Data	Stream Data
- Process temperature	- Temperature
- Process pressure	- Pressure
- Endothermic or exothermic	- Mass flow
- Hourly profiles	- Energy
- Thermal efficiency	- Specific heat
- Fuel Requirements	- Containment Data (waste streams only)

Figure 4-2. Generalized Unit Operation Data

Source: *Energy Analysis of 108 Industrial Processes*, Brown, H.L., 1980, The Fairmont Press

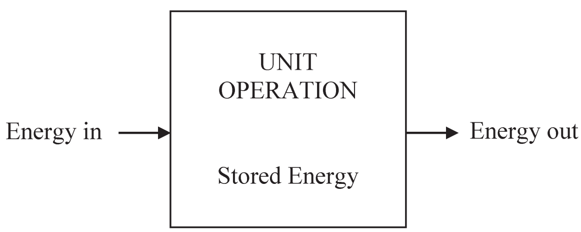


Figure 4-3. Generalized Energy Balance

For every control volume, the input energy must equal the output energy plus any residual internal energy; the internal energy can be heat remaining in the product, heat absorbed with a chemical or physical reaction, or other work done to the product that required energy input and did not leave the control volume. Energy balancing can sometimes be as simple as relating mass (flow) and specific heat to Btus, but can also be complex with changes of state, coefficient of performance, and chemical reactions (endothermic, exothermic). Product flows can be split, such as one inlet and two outlets. Energy input/output flows can be include direct energy inputs, cooling water, heat lost to ambient surroundings, ventilation losses, etc.

The control volume 'micro view' energy accounting method can be helpful in energy management by identifying sources and magnitudes of losses, ideal process energy requirements, and process efficiency. Reference **Process Efficiency Example #2 and #3**. Knowing the best case minimum (ideal) energy use for a given process identifies the magnitude of waste and the process efficiency.

We know that input - output = work done to the product or losses. So:

Ideal Minimum Energy Use = Input Energy Use—Output Losses **(eq. 3)**

Input Energy Use—Ideal Minimum Energy Use = Output Losses **(eq. 4)**

Output Losses = Energy Waste = Maximum Possible Savings **(eq. 5)**

Process Efficiency = Energy Needed/Energy Used **(eq. 6)**

Note: The concept of identifying the minimum needed energy use is only intended to give a good idea of optimum performance for that process at that site. It also does not preclude additional savings by using a different process.

There are different ways of identifying the ideal minimum energy use for a process:

- **Calculation**: This requires a thorough understanding of the process, so that the energy required for the process is identified accurately. Some processes are simple (i.e. heating, lifting, evaporating), but many include chemical and physical changes to the product that are not easily quantified.
- **Mass and Energy Balance**: The difference between input energy and loss energy represents the work done to the product inside the control volume.

  Note: Using mass and energy balance to identify minimum energy needs for a process requires separating leaving energy streams that are true losses from energy streams required for the work done to the product. This can be seen in Process Efficiency Example #2, where the water driven off the dough in baking is not considered a loss to the process.

Advantages:
Knowing what the ideal needed process energy requirements are simultaneously identifies waste, and the magnitude of potential savings. I.e. does a proposed improvement address 1% of the known waste or 30% of the known waste?

Disadvantages: Micro view analysis is time consuming and costly which explains why it is seldom done.

Additional remarks:

1. Knowing where all the energy goes is desirable but the cost of quantification for *all* process steps may be prohibitive. Where it is certain the bulk of the energy use occurs in a few particular process steps, it may make sense to limit the energy analysis to those steps, i.e. invest the time and effort on the few places where the 80% cost is occurring.
2. When certain forms of energy within a given process dominate, it may be acceptable to neglect some or to lump them together as 'other' that are accounted for within a gross measured input or output, or stored/internal energy remainder.

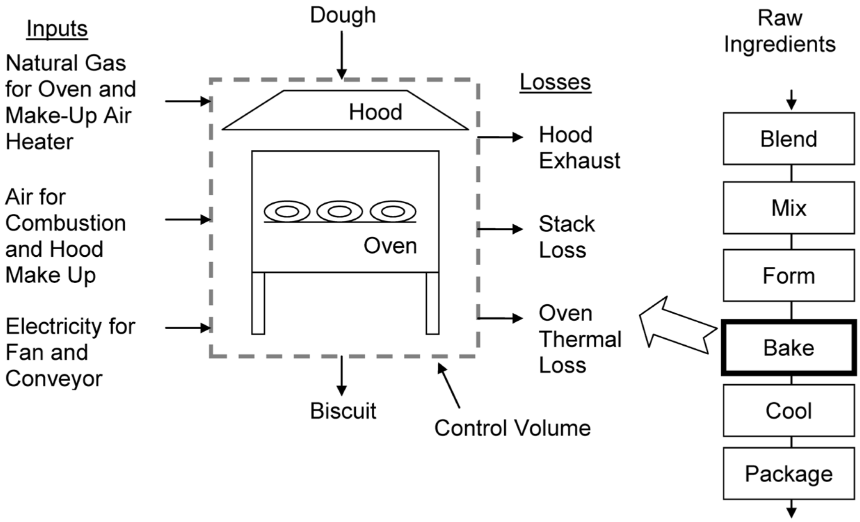
Process Efficiency Example 2:
Micro View (Review an Individual Process)

Values used to illustrate the method are not an actual case study.

The baking process removes some of the water from the biscuits. Additional heat is absorbed by the dough during chemical and physical changes during baking. There is not sufficient data on the reactions during cooking to quantify them or to know if they can are significant or can be neglected. Mass and energy balance will determine the heat involved in baking the biscuits. Referencing the control volume diagram, the difference between entering heat and leaving heat represents the heat absorbed in the biscuit and the minimum energy required for this process. Note that using this method removes the need to quantify the heat burden of the various changes that occur within the dough during baking. The natural gas use to the oven is measured with a sub meter and usage is recorded for one hour. In that time 1,000 lbs of dough is baked and 14.5 therms of gas are used.

<u>Additional Data:</u>

- Natural gas oven with 80% combustion efficiency
- Oven thermal loss estimated at 20% of product heat load (conveyor oven, open door)
- Use 0.3 cfm combustion air per 1000 Btuh of forced draft burner output. Source: *Combustion Air Requirements for Boilers*, Cleaver Brooks, Tip Sheet November 2010
- 5000 cfm of exhaust air from the oven hood; annual average outside air temperature for replacement air is 40F and air is tempered to 60F with natural gas direct fired at 100% efficiency at sea level. The fan is 2 kW
- Electric conveyor system for the oven is 5 kW
- Cost of natural gas is $0.80 per therm; cost of electricity is $0.07 per kWh
- Ignore building services loads of lighting and HVAC other than hood make-up air
- In determining the minimum energy required for the process, assume the following are required per the baking process and not a 'loss' in the context of process efficiency:
  - — Removal of water from the biscuit
  - — Heat in finished biscuit lost to surroundings after baking



Solution (Process Efficiency Example 2 - Micro)

- Fuel input for Oven: 1,450,000 Btu = 14.5 therms <u>measured</u>
- Electric input: 2 kWh for exhaust fan, and 5 kWh for the conveyor, 7 kWh total; 7 * 3.413 kBtu/kWh=<u>23.9 kBtu</u>
- Stack loss
  — Oven combustion efficiency given at 80%, so the stack loss is 0.20 * 1,450,000 = 290,000 Btu = <u>290 kBtu</u>.
- Oven thermal loss (casing/door heat loss)
  — Remaining heat after stack loss is 1,450,000 - 290,000 Btu = 1,160,000 Btu.
  — Casing and door loss given at 20% of product load; 0.20 * 1,160,000 = 232,000 Btu = <u>232 kBtu</u>
- Heating requirement for replacement air (loss, and additional fuel input)
  — Air flow is 5000 cfm hood exhaust plus combustion air. Combustion air is given as 0.3 cfm per 1,000 Btuh burner output = 1,160,000/1000 * 0.3 = 348 cfm. Total air flow 5,348 cfm
  — Mass of air at sea level = 5348 cfm * 60 min/hr * 0.075 lbs/ft^3 = 24,066 lbs air.
  — Q=mcΔT, 24,066 lbs air * 0.24 Btu/lb-degF * (60-40) = <u>115,517</u> Btu~1.16 therms

See tabulation: **Results (Process Efficiency Example 2 - Micro)**

Process Efficiency Example 3: Micro View
(Simplified Review an Individual Process)

Values used to illustrate the method are not an actual case study.

Industrial laundry drying operation takes 500 lbs. of 100F laundered cloth at 40% remaining moisture content (RMC) and dries it in a natural gas dryer to 5% RMC. Each dryer processes up to 3500 loads per year, however only a single drier is evaluated, with the results to be scaled to the whole plant. Consider only the removal of moisture from the cloth, since the latent heat of water is the dominant energy requirement.

<u>Additional Data:</u>

- Latent heat of evaporation for water is 970 Btu/lb. for evaporation and (212-100) * 1.0 Cp = 112 Btu/lb. to raise the water temperature to 212F; total 1082 Btu/lb.

Step No.	Step Descrip.	Fuel input qty	Elect Input qty	Inlet	Description	Mass (lb)	Energy In (kBtu)	Loss	Description	Mass (lb)	Energy Out (kBtu)
	Baking	therm	kWh	prod. in	Dough	1000		prod. out	Biscuit	444	
									Hood exh.		116
		14.5			Gas, oven		1,450		Stack		290
		1.2			Gas,air heater		116		Oven casing		232
			7.003		Electric		23.9				

1,589 Total input energy kBtu

Total losses **638** kBtu

Input - Loss	Minimum Energy Required kBtu	**952**
Min. / Input	Process Energy Efficiency	**60%**
	2144 Btu/lb finished product	

	Fuel	Elect	
total	16 therm	7 kWh	0.029 $/lb finished product
unit cost	0.80	0.070	
cost	12.52 $ gas	0.49 $ elec	13.01 $ total energy cost

Results (Process Efficiency Example 2 - Micro)

<u>Calculation</u>

- Initial water content: 500 lbs. * 0.4 = 200 lbs. of water; thus the dry cloth weight is 300 lbs.
- Final water content: (10%/40%) * 200 lbs. = 50 lbs. of water.
- Water removed 200-50 = 150 lbs.
- Heat needed to remove the water: 150 lbs. * 1082 Btu/lb. = 162,300 Btu heat needed.
- Actual fuel used is measured at 3.0 therms.
- Heat used = 3.5 * 100,000 Btu = 300,000 Btu input.

<u>Results</u> Process efficiency = heat needed/heat used

162,300 Btu needed/300,000 Btu used = 54% efficient.

- In addition to the water removed from the cloth as steam (the needed heat), additional heat loss is found from input - output 300,000 Btu input—162,300 output = 137,700 Btu loss per load, per drier.

This information can be used in various ways

- Value of loss equals the maximum savings possible for this process
- Enables heat recovery options to be explored, using magnitude, timing, and quality of available waste heat.
- If measures are implemented to reduce dryer fuel usage, this will form the baseline from which to gage savings achieved.
- Note: In addition to improving this process, different processes can be considered. For example, additional extraction would reduce the amount of water in the cloth prior to drying and the cost of extraction can be weighed against savings in drying fuel.

Different Approaches

Compare process micro examples #2 and #3. Note that for example #2, unknowns in the unit operation (reactions) were solved by calculating losses and determining the process energy needs by remainder. For example #3 the process needs were readily, and the losses were determined by remainder. The approach to quantifying other processes will vary depending on complexity and what can be measured, and there will some processes that defy quantifying. The examples given do not suggest that all processes can be quantified or that they should, but that there is energy management benefit when it can be done for the larger consumers of energy.

PROCESS DIAGRAMMING

As part of the process energy audit, it helps to understand the process. A process flow chart is a good way to visualize it. There are different forms of a flow diagram, but each with the goal of using visualized data to add understanding and prompt improvements for improvement. See **Fig 4-4** and **Fig 4-5**. See **Suggested References** at the front of this book for a good source of supplemental information on process diagramming and mass-energy balance methods.

Some individual processes will be more energy intensive than others and individual processes are not necessarily equated to specific machinery items. **Fig 4-4** gives the energy use for each process in Btu/lb. of finished product is very helpful since it draws attention to the higher energy use areas. Note in **Fig 4-4** 84% of the total process energy use comes from three locations. In most plants, 10-15% of the equipment in use accounts for the majority of a facility's energy consumption. (Source: "Save Energy Now," Industrial Technologies Program (ITP), 2006, US DOE Office of Energy Efficiency and Renewable Energy.)

Process flow diagrams also indicate process dependencies (what depends on what, goes before or after, etc). Some facilities are single purposed and others have branching lines of product flow.

The diagramming step can sometimes reveal processes with potential to be linked together for energy benefit; e.g. waste from one process is a useful byproduct for another process. Examples of this are shown in **Fig 4-5**.

STANDBY LOSSES

Manufacturing tool energy use measurements (amps) are useful to determine how well the tool energy use will turn down along with the process; i.e., what fraction of the full load production energy use is spent while idling. Many modern tools turn down to 10% or less in a "sleep mode," while many older tools consume half of the full load energy just sitting there. The higher energy use in idle mode has a pronounced effect on plant profitability during periods of reduced output. Idling loads can be identified on a gross scale by reviewing interval data inactive periods, followed by tours to identify where the usage is coming from and if it is necessary or can be turned off/turned down.

Standby loss considerations can become fuel switching considerations. For example, heating will be more expensive with electric resis-

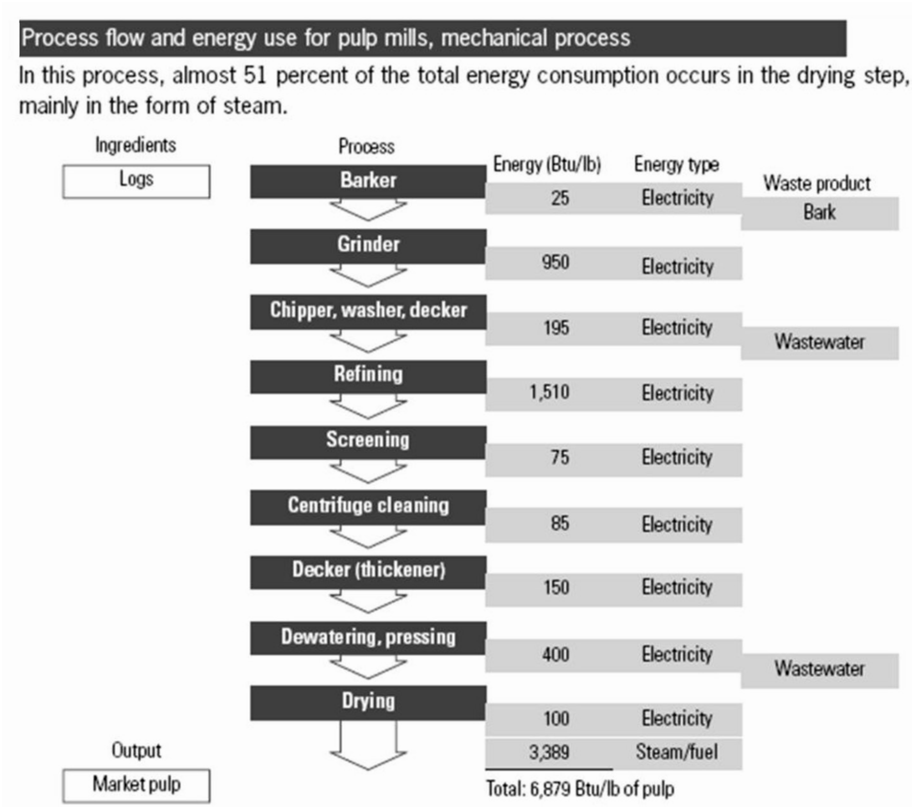


Figure 4-4. Manufacturing Process Flow Chart with Energy Use Identified
Source: E Source, 2004; data from H.L. Brown

tance; however when a heating process is used only occasionally, the standby losses for a fuel fired heater may exceed the electric cost differential when the electric option can be turned off/on at will, avoiding standby losses.

PROCESS-RELATED SHARED SYSTEMS

There are usually ancillary systems that are shared by different processes. The energy use for the shared systems is not easily attributed or apportioned by individual process. For example, in the case of a burner dedicated to a process, the energy use associated with the process would include heating, stack losses, blow down, flash losses, standby losses, and thermal losses. But if the burner is on a boiler that provides steam

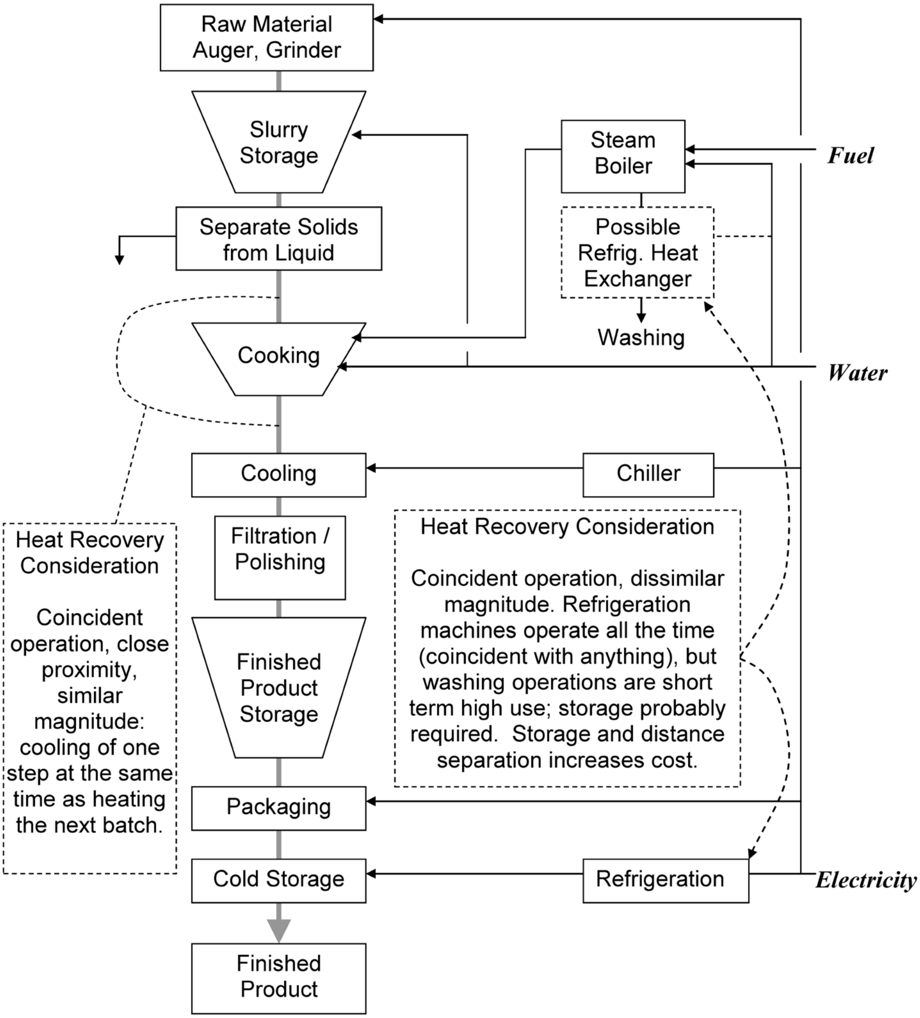


Figure 4-5. Example Manufacturing Process Flow Diagram with Utility Inputs Added, Showing Coincidence Impact to Heat Recovery

to multiple processes, the output heat is associated with the process as an input, but the losses are separate. Additional uses of the boiler, such as clean up are also a shared system energy expense. A common shared electrical expense is compressed air; in this case the input energy for compressed air to a given process is rarely measured and then the entire electrical use becomes a shared energy expense. See **Figure 4-1** for some common process-related shared systems. These energy expenses are automatically captured in the macro method.

BUILDING SERVICES ENERGY USE

Whether non-process building services are included in the mix for process energy use is a matter of philosophy. It can be said that the office areas of the plant, lunch room, lighting, etc. are not process related; but it can also be said that those things would not exist, nor the building, except as an appendage to the process and necessary for the sake of selling the product. In the macro view, the default measurements for energy are utility meters which serve both building energy and process use and so in the absence of additional sub metering, calculations or estimating the building use will be lumped in with process use. From a process optimization standpoint, building services "non-process" loads should be separated so that the same process has the same results in any location or building.

For this text, the micro view of process analysis will not include the energy of building services, however it can be identified if the entire process energy use is identified, from:

**Building Services Energy = Total Energy Use—
Process Energy Use (eq. 7)**

The macro view of process energy is assumed to be driven solely from utility meter data and will include building services. Example building service items are given in **Figure 4-1.**

REVIEWING COINCIDENCE OF ACTIVITIES

Coincidence looks at *timing* of processes; those that run at the same time and whether coincident activities are random or due to dependencies. Some coincident activities are not important such as taking fuel oil from a tank (what matters is when it's empty) or natural gas (sold by the therm); while other coincident activities may have associated costs such as on peak power rates, and are desirable to avoid. Some things inherently must run together to make the product, others are sequential, and others are random. One section of the plant may operate independently of another section and the combined coincident load may be adjustable.

When about electric demand costs, the charge is the cumulative effect of the multitudes of loads running at the same time, and the question is whether there are ways to save money by spreading them out, if possible. One tell-tale sign that encourages looking at coincident electric loads is a low load factor.

Load factor = average kW/max kW **(eq. 8)**

When coincident electric loads create high utility cost from low load factor (demand charges are a large portion of total cost from short term high electric demands), the focus will be to de-couple coincident loads if possible. In some cases, such as a job shop, there may be no reliable pattern of electric loads because what they work on is related to phone calls that day and delays are not tolerated by the customer.
Note: Just In Time manufacturing (JIT) has cost benefits to manufacturers, but it is at odds with Time of Use (TOU) utility pricing which subtracts from those benefits.

The coincident load is a composite pile of individual loads. Short duration high demands are common in manufacturing and are a source of aggravated electric charges stemming from low load factor.

When load factor is low, demand charges become a larger portion of the utility bill, raising the overall cost of electricity (an ingredient cost increase).

- Base loads include lighting, circulating fans and pumps, power conditioning, and computers.
- Near-steady loads that vary with use but have high persistent minimum loads include refrigeration and compressed air.
- Intermittently used machines contribute to the coincident load with the range of standby (idling) load to full demand (e.g. equipment idling, and flywheel driven devices, induction heating).
- Continuous processes (e.g. wire drawing + tempering + coating + coiling, extruding, conveyor ovens).
- Long batch processes (e.g. plating, semiconductor crystal growing, smelting).
- Coincident batch equipment (e.g. baking, heat treating, vulcanizing, pasteurizing, electric paint drying).
- Start-up loads (e.g. initial pull up heating for ovens and hot equipment)
- Misc. random loads

Some thoughts on approaching problematic coincident loads

- The coincident demand question may be observable directly, may be best observed with data logging, and may have no pattern at all.
- Individual loads within a given batch process will defy change with regards to coincidence unless the recipe itself is changed, but parallel or sequential batches are a possibility to modify.
- Parallel product lines that do not connect except at the shipping dock have coincidence that is random can sometimes be adjusted to reduce concurrent load. One method is having certain machines—large loads - operate mutually exclusively of each other by interlock (prevented from operating at the same time). Production questions must be addressed.
- In some cases coincidence can be addressed by storing material at some intermediate step (until other machines are finished with their task), thereby making two large demands mutually exclusive. Production questions must be addressed, as well as how to 'catch up', and what other downstream effects could occur.
- Sometimes an automated or long term low maintenance process can be moved to another shift.
- Activities that fight or counteract each other are problematic, with a false-loading effect that sustains the magnitude of coincident energy use (example: pressurize, only to de-pressurize at the end of the same pipe). There are often ways to mitigate this parasitic loss through control automation that makes adjustments aimed at providing enough, but just enough, of the commodity. Examples in **Table 4-1.**

In some cases, coincidence is a good thing and can be leveraged. When two processes are linked by time and one needs heat while the other is expelling waste heat, heat recovery potential exists. Examples in **Table 4-2 and Fig. 4-5.**

REVIEWING THE PROCESS ITSELF

This approach includes repetitive use of the words "why?" and "how about?" and challenges the processes themselves. Is there a better way? Why is that step needed? Some examples in **Table 4-3** and **Table 4-4**.

Table 4-1. Improvement from Reducing Coincidence

Process	Improvement
Simultaneous electric loads for short durations of time create electric demand charges, disproportionate to average usage (low load factor / high demand charges)	• Schedule work that includes large electric loads to run exclusive of other large electric loads, spreading out the demands. • Schedule work that includes large electric loads to utility off-peak times. • Move work processes to another shift to distribute load. Stagger-sequence electric resistance loads during daily "warm up" period to avoid setting the monthly demand. • Fuel switch electric heating loads to natural gas • Thermal storage; heating or cooling
Simultaneous coincident actions are occurring with no benefit to the product • Heating / cooling • Humidifying / dehumidifying • Pressurize / head loss (pumps, fans, compressors) • Capacity control by throttling • Capacity control by blending • Other false loading	• Dead band control that inserts a zero energy gap between the opposing forces, reducing the fighting and back-and-forth cycle • Anticipation of need / adaptive control. Example is embedded temperature sensors in a plastic injection mold controlling actual product cure process (time/temperature) rather than controlling water jacket temperature within a half degree. • Polling software to connect end use demand to supply demand, with the goal of providing enough, but just enough (of whatever) and reducing throttling losses. See also <u>Special Topics:</u> "**Coordinating Upstream and Downstream Set Points"**

Table 4-2. Improvement from Leveraging Coincidence

Process	Improvement
Batches of milk or beer are sequentially heated and cooled as they are processed	First step of cooling the exiting liquid is preheating the next batch
Air compressor waste heat is ducted outdoors	Diverter dampers use the heat in the adjacent area in cold weather
Process cooling load is steady and rejects heat to a cooling tower; operations building areas are heated with electric resistance heaters	Water source heat pumps provide heating at COP=4+ using warm cooling tower water as a heat source.
Evaporator unit uses natural gas to create disposable solids from chemical solution	Flue gas stream from adjacent ovens provides heat for this process using waste heat.
Large deionizing water load in one part of the plant; large process boiler in another part of the plant, concurrent operation.	Switch from DI water to distilled water, using stack gas as source of heat to make distilled water.

Table 4-3. Example Improvement to the Process Itself

Process and Issue	Improvement
One process involves wetting the product, and is followed by a process that dries it.	Air-dry parts and eliminate the heater.
One process applies paint, and is followed by a process that cures it in an oven.	Switch to a paint that is designed to air-dry and eliminate the oven step entirely.
Replacement annealing oven is presumed to be electric, like the one being replaced.	Opt for natural gas oven, for reduced fuel cost.
Oxidizer oven used to remove solvent from exhaust air which includes solvent and non-solvent exhaust.	Segregate the solvent exhaust and downsize the oxidizer; use a solvent scrubber.

Table 4-4. Example O/M Improvements Related to Process

Process and Issue	Improvement
Compressed air used for blow off for chips, dust, etc.	Air eductors (amplifiers) entrain room air and use a fraction of the compressed air.
Product rinsing waste has high sewer charges for suspended solids. Significant cooling tower use for refrigerated storage.	Separate the solids and reuse water in a cooling tower or for irrigation.
Plant start-up includes multiple coincident electric loads from initial warm up of tanks, ovens, etc., setting monthly demand charges.	Stagger large electric heat loads to only let them be coincident once at temperature and cycling.
Fume exhaust flow always on.	Variable capacity fan responds to end use changes which open and close depending on use.
Plating agitator pumps always on.	Adjust PLC controls to turn pumps on for batches that require it, otherwise off; and off when plant is closed.
Area exhausters for metal machining expelled from the building, creating large load for makeup air equipment.	Filter and re-introduce the air into the building, monitor for pollutant of interest.

(Continued)

Table 4-4. Example O/M Improvements Related to Process

Process cooling water at 60F provided by chilled water, using refrigeration machinery.	Evaporative cooling with a fluid cooler maintains 70F in most annual hours, with the mechanical cooling unit kept as standby. Increased flow or higher leaving water temperature required.
Factory is conditioned for comfort of workers in fixed work positions.	Heat/cool the workers only (spot), not the entire building.
Process HVAC air is cooled and reheated to worst day conditions of a summer rain storm to assure constant conditions at all times.	Faster reacting controls to achieve cooling coil dew point control, combined with weather anticipation.
Lab/clean area air changes maintained at all times.	Control based on particle count or clean room objective.
Work occurs at floor level; overhead bulk lighting is used throughout to provide strong light at work points. To achieve a given level of work level light intensity (lumens), wattage increases as the square of the distance above it.	Add task lighting at each workstation; reduce overhead lighting wattage to general ambient levels. Lower the lighting fixtures and reduce lamp wattage. Note: usually also requires additional fixtures for even area coverage.

Process review is very touchy because the profitability of manufacturing depends on repeatedly producing a salable product; changes to a working process are a business risk and will be met with resistance, for good reasons. Products certified as ISO-9000 earn that label by having a documented and repeatable process, so a proposed change will see additional resistance for the re-certification costs.

Some processes are also proprietary. And, of course, if the process is so unique and lucrative energy cost may be trivial, although this enviable condition usually doesn't last.

For effective communication, it is important to understand that the manufacturing customer is often more in tune with increasing production rate in their facility than reducing energy use. In general, the higher the percentage of operating cost that comes from utilities and the lower the net profit margin, the greater the interest in reducing energy cost. See **Appendix "Equating Energy Savings to Profit Increase."**

DESIRE FOR ENERGY USE TO FOLLOW PRODUCTION RATES

A very important first step is to equate percent energy use to percent production for the same period. Graphs are very helpful for this. If energy use changes do not closely follow production rate changes, this probably indicates high standby losses such as ovens left on or machinery that will not fully unload while idling. **See Chapter 2—Analyzing Energy Use Graphs "Business Volume (Production Rates**)"

PRIMARY ENERGY USE SOURCES

- Most energy use is directly proportional to production.
- End use breakdown depends entirely on the process.
- Choice of measures depends upon the process.

PRODUCTION SCHEDULING

- For companies running multiple shifts, evaluate operations that are on-peak and off-peak. It may be possible to modify the current production sequence to use high energy consuming equipment during off-peak times. This may require intermediate storage of materials.

- Where shift pay differentials exist, labor costs will usually outweigh energy savings from adding another shift or moving to night shift to leverage off peak rates.

MAINTENANCE

- Annual maintenance on all heat transfer surfaces, including good quality filters, cleaning coils, cleaning tubes. Increase frequency of cleaning where process causes rapid fouling of heat exchange surfaces.
- Locate and repair leaking oven door seals.
- Use cog belts instead of standard V-belts for large motors.
- Survey and log fired heat exchanger process flue temperatures and other indicators of fouling, for predictive maintenance. Use heat exchange "approach" as a benchmark and indicator of waste.
- Maintain proper boiler water treatment and ensure that normal make-up does not cause dilution and scaling.

CONTROLS

- Monitor and track energy use as a function of production, to establish benchmark data and cost influence on final product. Identify energy use rate in terms of Btu per pound, ton, or cubic foot of product, Btu per quantity of product, etc. Use this benchmark to measure the consistency of energy use "per product" and to measure results of process changes.

  Note: the approach of manufacturing ECMs may be to reduce the energy cost per product output; or to increase production output without a proportional increase in energy input.
- Control process equipment to make energy use track production use; increasing and decreasing proportionally with production rates.
- Turn off unused equipment.
- Determine the cost impact of just-in-time manufacturing.
- Eliminate simultaneous heating and cooling.
- Use controls to optimize processes. Provide enough, but just enough, to do the job properly.
- Lower settings for high bay space heaters.
- For precision temperature control processes using both heating and cooling medium, widen control deadband if possible to reduce the

overlap.

- For all roof openings to hoods and equipment not active in winter, automatic dampers should be tightly closed during heating operation and when roof equipment is off.
- Provide process ventilation and make-up only when needed, instead of continuously.
- Utilize optimized control settings that allow adaptation for production rates, and seasonal effects, instead of one-size-fits-all fixed set points. Fixed set points almost always translate into energy use increase.

SOME COMMON ECMS FOR MANUFACTURING

Low Cost/No Cost

- Turn off equipment that is not in use for over a half hour.
- Lower standby temperatures for ovens, mold machines, etc.
- Schedule work such that electrical load is leveled, to avoid high utility costs associated with poor load factor (average demand divided by maximum demand, in percent).

Retrofit—Or Upgrade at Normal Replacement

- Switch from air conditioning to evaporative cooling.
- Switch from air conditioning to spot cooling.
- Switch from mechanically cooled process cooling water to a fluid cooler and evaporative cooling.
- Heat recovery for adjacent and coincident heating and cooling processes. Waste heat can serve for process use or for space heating. Large sources of make-up air are candidates for preheating.
- When heating and cooling operations are weather-independent, heat recovery opportunities are improved because available hours per year are increased.
- Evaporative pre-cooling on large air cooled chillers and air package rooftop units.
- Boiler stack gas heat recovery.
- Rinse-water or wash-water effluent heat recovery to the concurrent make-up water stream.
- High bay lighting retrofit from HID to fluorescent.
- Roof insulation for heated buildings, as part of roof replacement, for heated or cooled buildings.

- Anti-stratification fans for high bay, heated areas to move warm air to the floor.
- Lids, floating balls, or other means to reduce evaporation from heated tanks.
- Power factor correction for groups of large motors, inverters, and welders.
- Higher efficiency motors.
- Higher efficiency cooling equipment (HVAC or process cooling).
- Insulate bare hot piping for process, including hot water, steam and condensate, vats, heat exchangers, tanks, furnace casings, boiler casings, etc.
- Process cooling equipment condensers vent to outdoors instead of inside the facility.
- Process ovens retrofitted with automatic doors or other means to minimize heat release.
- Reduce excessive light levels by de-lamping.
- Combine de-lamping and reflectors to maintain light levels.
- Variable flow heat hood exhaust. If the exhaust captures just heat, vary the exhaust flow rate to maintain a leaving temperature 30 degrees higher than surrounding indoor ambient air temperature so the system always acts like a spot exhauster and reduces the volume make-up air.
- Large steam uses after pressure reducing valves may be an opportunity for a separate low pressure steam system.
- Very large steam uses after pressure reducing valves may be an opportunity for a steam work extraction element (turbine driven device or cogeneration).
- Combine processes where possible, to reduce "new energy" input. Examples are using rejected heat from an air compressor for heating, rejected from one process to regenerate a desiccant bed, rejected heat from a cooling operation to a concurrent heating need.

If There are Office Areas

See "Office Buildings" this chapter.

Process Boilers, Furnaces, Ovens, and Other Combustion Heating Systems

- Monitor stack gas temperature by routine measurement. High or increasing heat exchange approach temperatures often indicate

fouling and need for cleaning.
- Monitor combustion efficiency by routine measurement. Verify air-fuel ratio is accurately controlled at all loads (100%, 50%, 25% firing rate). Reduce excess air.
- Turn equipment off if there are long periods of idle time between batches
- Reduce operating temperature if possible. Provide just enough heating. Optimal temperature setting may be lower at part load than full load.
- Install high turndown burners if frequent on-off cycling is happening.
- Reduce air leaks into the furnace or hot gas leaks from the furnace. These can be from door seals, and openings for conveyors.
- Insulate bare casings.
- Insulate bare piping.
- Preheat combustion air or make-up water from waste heat source or stack gas.
- Preheat the product.
- Automatic boiler isolation valves if piping allows hot water through an "off" standby boiler.

Process Chillers
- Higher efficiency chiller.
- Raise chilled water temperature if possible.
- Lower condenser water temperature for water-cooled chiller.
- Higher efficiency/capacity cooling tower.
- Annual condenser tube cleaning for water-cooled chiller.
- Controls to lock out the chiller below 50 deg F.
- Dry coolers or fluid coolers for process chilled water in cool/dry weather instead of chiller.
- Evaporative pre-cooling for air-cooled chillers.

Process Cooling Water
- Use cooling water from a cooling tower or fluid cooler instead of chilled water.

Steam Systems
- Lower steam pressure if possible.
- Implement a steam trap repair program to regularly visit these devices and prevent steam leakage.

Compressed Air Systems

- Locate and repair compressed air leaks.
- Reduce compressed air pressure.
- If source must be maintained more than 20 psi above the end of line point of use, storage or pipe size increase may be appropriate.
- Separate low pressure high volume compressed air uses from higher pressure air, and supply with an air blower instead of an air compressor.
- Pipe compressor air inlet to outdoors, with provisions for low temperature operation.
- Heat reclaim from compressed air heat exchanger (screw), such as ducting to an adjacent area for free heating.
- Where multiple pressure levels are maintained through pressure reduction and a single source, provide separate system for the higher source if it is a lower volume demand than the lower pressure zones.

Humidifiers

- Lower the humidity set point if possible to reduce humidification load.
- Use adiabatic humidifiers (evaporative pads, spray nozzles, atomizers, ultrasonic) if the process combines cooling and humidification.
- Look for inadvertent dehumidification, such as a cooling coil downstream of a humidifier.
- Raise chilled water temperature 5 degrees to allow sensible cooling without the side effect of dehumidification.
- Separate humidified and un-humidified areas to limit the humidification load, using a wall and doorway to adjacent areas, to reduce the humidity load.
- Ensure a complete vapor barrier surrounding the space that is humidified, including walls, roof, floor, and penetrations.
- Reduce exhaust if possible, to reduce the humidification load for make-up air.
- Humidity recovery between exhaust and make-up air via 'total heat' wheel with both heat and moisture capture.

Plating Tank Covers

Basis of Savings: Evaporation losses dominate heated tank thermal losses.

- Note: push-pull air curtain systems are for fume control not ener-

gy savings. The increased air flow across the tank or vat increases evaporation.
- Even partial covers will reduce evaporation losses.
- Floating balls can reduce evaporation rate by up to half if compatible with parts and quality.
- **See Appendix "Evaporation Loss from Water in Heated Tanks."**

Process Oven Door Seals

Basis of Savings: Reduced loss from leakage.
- Automated conveyer part baking requires doors be left open. For these, reducing the size of the opening and reducing any ambient air currents that would 'sweep' hot air out of the oven are about all you can do.

SPECIFIC LIGHT MANUFACTURING ECMS BY PROCESS

Semi-Conductor Fab Multi-Stage HVAC Tempering
See Chapter 24—Special Topics

Hot Asphalt Mix Plants
- Energy use benchmark is in kBtu/ton. A 2008 value of this heat burden is 308 kBtu/ton, but varies regionally and is strongly dependent on local moisture content.

Energy consumption in asphalt pavement production is focused in three main areas:
- Drying the aggregate so the liquid asphalt cement can adequately adhere to the stone
- Keeping the virgin asphalt cement stored in a heated liquid state
- Operating the facility
- Each 10% change in moisture content of the aggregate has a corresponding 10% change in fuel use. (This gives incentive to keep the aggregate covered from rain.)
- Insulating the mixing drum can reduce fuel use by 5% for the drying operation.

Source: Young, TJ, Energy Conservation at the Plant Makes Sense and Saves Dollars

Aggregate and oil are mixed together in a large drum and heated/tumbled, then kept in heated silos for truck filling. Process is in batches.

Some considerations for energy:

- Electric load factors are poor, and cost per kWh is correspondingly high
- 80+ pct of total energy is spent in the drying drum
- Could be a Combined Heat and Power match.
- Hot product can be stored for up to two days, and is then re-cycled.
- Variable speed drives in lieu of discharge dampers, for dryer drum exhaust that goes to a bag house for particulates.
- Waste heat from the drum and bag house is 150-200 degF and would be suitable for pre-drying the aggregate if an economical method of doing this were devised.
- Seals at each end of the drum are a wear item and will leak considerable amounts of heated air when worn.

Pre-Mix Concrete Plants

- Usually located by a quarry, the aggregate is mined, ground, and sifted/sorted into grades. These machines can jam, and so are usually belt driven. Cog belts can offer minor improvements (1-2%).
- Power factor will be variable and low, due to large motors sized for torque and intermittently unloaded.
- Electric load factors are poor, and cost per kWh is correspondingly high
- If cement is made on site, or when concrete pre-mix is made in cold weather, large amounts of heat are needed. Waste oil burners for crankcase oil can supplement the heat needs.

Waste Water Treatment Plants

- Energy use benchmark is in kWh per Million Gallons.
- A variety of pumps are used, many that are belt driven for jam protection. Cogged belts can offer minor improvements (1-2%) .
- The single largest energy use point is the air blower which is used to aerate the process. Controlling to lowest workable pressure using most-open-valve control is appropriate.
- Blower capacity modulation range is limited to around 40% unless variable speed drives are used. Load is reduced at night and equipment sizing must accommodate the load swing within its throttling range to remain efficient at all loads—a jockey air pump may be needed.
- Low resistance air filters for the blowers will reduce the head lift and horsepower.

Forging, Stamping, Machining

- Energy use benchmark can be kBtu/ton or per part
- Preheating for forging is from induction units, so power factor correction is viable.
- Shifting induction +forging operations to off peak period can leverage utility rates.
- Oil/dust control exhausters can utilize bag or cyclone filters and return the air to the plant and reduce make up air tempering costs significantly.
- Painting and other batch work with large exhausters can use occupancy sensors to activate the ventilation system on demand.
- Large compressed air loads—certain shifts may do well with 10-20 psi less air pressure and this is easily scheduled with a plant PLC controller.
- Turn off idling machines when not in use, including conveyors.

Heat Treating/Tempering/Reducing Ovens

- Gas heat in lieu of electric heat, if temperature control can be achieved.
- Door seals.
- Schedule batch work to avoid multiple electric ovens operating concurrently
- Utilize waste heat for winter space heating in adjacent areas
- Schedule batch operation of reducing ovens instead of 'always on'

Brewing

- Energy use benchmark is Btu/lb or Btu/gallon
- For storage tanks at different temperatures, use different refrigeration units at needed temperature, in lieu of common refrigeration brine at lowest temperature and blending the others.
- Heat recovery from hot waste water between batches.

Clean Rooms and other Year-round Cooling Loads

- Use cooling equipment that is water cooled in to gain winter efficiencies from water economizers or dry coolers.
- Where humidification is used, seal the enclosure and entrances from adjacent non-humidified areas and to outside.

Plating, Coating

- Hot plating tanks have highest losses from evaporation. Increased humidity and temperature of surround air will reduce evaporation.

Covers are a direct approach, but are often not compatible with dipping racks.

- Rectifiers that operate at reduced load can have very poor efficiency (60%)
- Powder coat curing ovens lose a great deal of heat through the conveyor openings, which is normally exhausted locally. This 100 degF air can be transferred to a neighboring area needing heated make-up air, e.g. a plating line.
- Heated wash basins become fouled with debris and have low thermal efficiency—can easily be spotted with high flue temperatures for a simple 120 degF bath.
- For plating lines with electric immersion heaters, the habit of starting them all at the same time on Monday morning can easily set the peak demand for the month. These can be staged and improve load factor.
- Air blowers can reduce compressed air energy use for blow off stations.
- Process in batches or shifts if possible, in lieu of continuous conveyor and curing oven operation.
- Switch to air-dry paint if possible, in lieu of heat cured paint to inherently require less energy.

Light Manufacturing/Assembly/Warehousing

With low interior heat loads, space heating dominates. Considerations:

- Evaporative or spot cooling instead of air conditioning.
- Dampers on all exhausters to prevent stack effect siphoning warm air out of the building.
- High bays can benefit from anti-stratification fans if 5 degF of stratification is present.
- More efficient lighting, including task lighting for fixed workstations.
- Occupancy sensor control for warehouses with intermittent occupancy.
- Turn off idling machines when not in use, including conveyors.

Dairy Processing/Pasteurizing

- Energy Benchmark is Btu/lb
- Very large sequential/simultaneous heating and cooling loads for pasteurizing.

- Heated water for cleaning and rinsing is single pass.
- Very good opportunity for heat recovery from refrigeration systems to domestic hot water
- 24x7x365 refrigeration loads allow optimization for reduced head pressure in cold weather.

Printing
- Operating printing presses at night leverages off peak utility rates.
- Large compressed air loads and warehouse areas allow heat recovery from air compressor waste heat.
- Lighting and heating measures for sorting/assembly/shipping areas consistent with warehousing.
- Where humidification is used, seal the enclosure and entrances from adjacent non-humidified areas and to outside.
- Air-drying ink in lieu of heated rollers and hot air ink drying.

Injection Molding
- Process cooling for injection mold cool/release jackets can be accommodated with evaporative fluid coolers in lieu of process chillers in drier climates.
- Bead driers can be operated with natural gas in lieu of electric resistance.
- Single pass steam generators are good candidates for feed water preheating.

Refrigerated Food Processing
- Envelope insulation and sealing.
- Exterior color and emissivity properties reduce solar gains.
- Dock door sealing.
- Cold wash water can become cooling tower make-up.
- High efficiency lighting minimizes additional refrigeration load caused by lighting.
- Plastic curtains, speed-roll-up doors, etc. to control air losses in fork lift paths or to non-refrigerated areas.

Refrigerated Warehouse
- Envelope insulation and sealing.
- Exterior color and emissivity properties reduce solar gains.
- Dock door sealing.

- Refrigeration systems can be turned off in winter and simply exchange the air, when stored items are not critical for food cleanliness—e.g. kegs of beer and other sealed products.
- High efficiency lighting minimizes additional refrigeration load caused by lighting.
- Plastic curtains, speed-roll-up doors, etc. to control air losses in fork lift paths or to non-refrigerated areas.

Combination Office/Sales/Warehouse Facilities

- Separate and seal warehouse from other areas.
- Looser climate control in warehouse— lower winter temperatures/higher summer temperatures.
- Avoid mixing air conditioned volumes with warehouse volumes, especially if evaporative cooling is used in summer. Keep doors closed.

Chapter 5

ECM Descriptions

ECM DESCRIPTIONS—ENVELOPE

Envelope Leaks—Infiltration

Basis of savings: Heating and cooling unwanted outside air

- Infiltration from construction cracks usually manifests itself as a comfort complaint, but always increases energy use. In extreme cases it can result in frozen piping. A building pressure test is the perfect solution, but usually not practical. One easy way to check for infiltration leaks is to check return air plenum temperatures during very cold weather. The return plenum is a negatively pressurized area. If there is leakage, it can be found with a hand held infrared thermometer while scanning the perimeter above the ceiling tiles.
- Usually hard to quantify.
- In extreme cases, the leakage around old operable windows can represent more of an energy improvement opportunity than replacing the single pane windows with double pane windows.
- Return air ceiling and shaft plenums are intended to be as air tight as any other duct but are usually far from that. This is especially problematic at building perimeters when the lack of proper construction sealing couples the return air plenum to the building envelope. Since the return air plenum is slightly negative by design, this almost assures infiltration through the envelope. User complaints that point to this are comfort issues at perimeter and cold temperatures above the ceilings at the perimeter. In extreme cases, water pipes can freeze because of this. In humid climates, severe mold damage and sick buildings can be traced to this.

- Infiltration is responsible for 13% of heating cost and 3% of cooling cost in U.S. office buildings (1998). For newer buildings the percentage is higher: 25% of heating and 4% of cooling, due to higher insulation levels. Losses are higher for pressurized buildings than non-pressurized buildings.

Source: Energy Impacts of Infiltration and Ventilation In U.S. Office Buildings Using Multizone Airflow Simulation , Emmerich,S.J., Persily, A.K., 1998.

See Infiltration in Chapter 17: Building Envelope Information

Exterior Color

Basis of savings: Reflecting, instead of absorbing heat. Savings are for cooling.

- **See Chapter 16—Lighting "Reflective Values of Common Colors."**
- If the solar heat gain of the wall or roof can be identified, the difference in reflectivity represents the savings potential for cooling.
- The greater the wall insulation value, the less heat is transferred through the wall and the less savings the color of the wall will have.
- For heating, lighter colors work in reverse and will reduce some passive solar heating effect. The savings has overall merit when the cooling savings outweigh the heating benefit.
- For roofs, using a 'pure white' reflectance factor for long-term energy savings is not recommended, since the color will naturally darken with rain and dirt.

Cool Roof

Basis of savings: Reflecting, instead of absorbing heat. Savings are for cooling.

- The cool roof technology uses materials with high solar reflectivity and high infrared emissivity properties, the combination of which reduces surface temperatures of the roof system and the heat transfer through the roof system that results in additional cooling load.
- The greater the roof insulation value, the less heat is transferred through the roof and the less savings the color of the roof will have.
- For buildings such as warehouses with only air-change ventilation for cooling, savings are nil although it will likely be more comfortable inside.
- For climates where heating loads dominate cooling loads seasonally, the trade-off between cooling savings and heating increase should be investigated, since the technology works in reverse during cold weather.
- For buildings with cooling loads dominated by envelope loads, cool roofs can be an attractive investment. For buildings with envelope loads that are minor compared to total load, such as those with large lighting or process loads inside, the payback period will be longer.
- Roofs, especially flat ones, get dirty and discolored. Thus, a degrada-

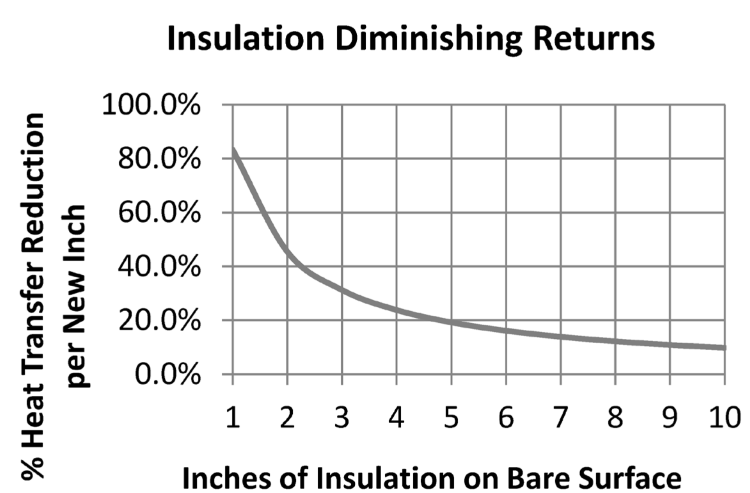
tion factor must be included to account for normal weathering, such that the persistent savings are realistic. For example, a reflectance factor of 0.8 for a new white roof may deteriorate in a few years to 0.55 from normal weathering.

- Single ply roofs have a life span equal to any other single ply roof. Mop-on cool roof coating products have a shorter lifespan.

Insulation

Basis of savings: Reduced thermal transmission

- A rule of thumb is that 'The first inch of insulation captures 80% of the savings.'
- The biggest opportunities for retrofit are hot or cold surfaces with no insulation at all.
- Increasing insulation thickness during new construction is often cost effective since labor is similar and the material is not expensive.
- Adding insulation to a roof during a roof replacement that will occur anyway is usually cost effective.



Heat transfer is proportional to U, Btu/SF-degF, which is 1/R.
Curve is percent change in U for each inch of added insulation.
Initial value of R (bare surface) is 0.2.
Plotted values are (U2-U1) / U1. In percentage values, the curve is the same regardless of insulation R-value or delta T.

Window Upgrade

Basis of savings: Reduced thermal transmission

- Glass is not a very good insulator, but a trapped air space does offer some insulation.
- In all but the most moderate of climates, single pane glazing is a poor choice from an energy perspective. In general, where heating or cooling is needed, single pane units should be avoided and are candidates for replacement if energy use reduction is desired.
- In addition to the glazing, some old window frames are a thermal short circuit. Evidence of this can often be found as water stains on the metal frames from winter condensation. New glazing should always have thermal breaks.
- For replacement of operable windows, poor seals and associated infiltration can pose as much of an energy loss as the windows themselves. New windows should have tight fitting seals.
- Windows are expensive and paybacks are commonly 15-20 years if only energy savings are used to justify the expense. However, window upgrades (the incremental upgrade cost) are easier to justify when windows are being installed anyway.

Window Shading

Basis of savings: Reduced solar load

- Exterior shades are best so that the heat never gets inside.
- Trees, awnings, overhangs, screen covers over skylights.
- Interior shades, if used, should be light colored and highly reflective.
- If existing glazing is clear, coatings or shades can reduce cooling load substantially and can reduce A/C equipment size requirements.
- **See Chapter 17—Envelope Information "Glazing Properties"** for more information on glass and shading.

Light Harvesting

Basis of savings: Reduced interior lighting

- Portions of the solar load are deliberately allowed into the building to provide day lighting without the use of artificial light. Often this is done near the top of the room enclosure along the wall and reflected off the ceiling. Skylights and clerestories are used to harvest light from the roof to the room below.
- In all cases, the solar gain and added air conditioning load subtract from the lighting savings, although there is usually a net gain.
- For skylights and clerestories, transparent element acts like a hole

in the insulation and increases heat loss at that point substantially, especially at night. The envelope heat loss adds heating load and subtracts from the lighting savings. Covers can mitigate this, but are cumbersome and expensive.

ECM DESCRIPTIONS—LIGHTING

Without evaluating the lighting as a system, it is possible to perpetuate existing problems and overlook potential savings.

Begin each lighting project by evaluating light levels (over lit, under lit), reported issues (glare, noise, headaches, slow start, flicker), and controllability (zoning, occupancy/vacancy control, daylight harvesting, task lighting, dimming requirements, and any areas of no control). For recessed ceiling fixtures ("can lights"), determine if they are vented or if they can relieve trapped heat to avoid high temperatures. Try to assess the condition of the existing fixtures; if condition is poor, a new fixture may be a better choice than a retrofit.

Demand Side

- Avoid excessive lighting levels
- 'Tune' light levels with options for lamp wattage and ballast factor
- Controls as enablers for occupants to choose less light, such as 1-2-3 circuiting and task lights
- Occupancy/vacancy sensors, photo sensors and circuiting in areas near perimeter glass and skylights to automatically reduce lighting demand when not needed
- Controls or procedures to prevent lights from being left on for extended periods, such as for cleaning crews

Supply Side

- More efficient light sources (lumens per watt)
- More efficient light fixtures (less loss within the fixture)
- Cleaning fixtures
- Reflectors
- Task lighting combined with bulk overhead lighting for reduced overall lighting power

- Lowering fixtures closer to the work surface
- Lighter and more reflective surfaces, to reduce absorption
- Direct vs. indirect lighting. If indirect lighting, use light colored, clean reflective ceiling or cove surfaces and maintain these surfaces to be as-new for lighting performance

See Chapter 16 - Lighting

Reflectors

Basis of savings: De-lamping with equivalent light

- All light fixtures result in some of the light being "trapped" in the fixture. Optical reflectors are available as retrofits. Made from highly reflective materials, these push more of the light out of the fixture. In many cases one of the tubes of an existing fixture can be eliminated by virtue of the reflectors. For example, a 3-lamp fixture, retrofitted with reflectors, may provide adequate light with the center tube removed.
- Be certain that the reflector unit is UL listed for a retrofit application.

De-Lamping

Basis of savings: Reduced number of lamps reduces wattage in over-lit areas.

- If it is determined that light levels can be reduced, de-lamping is the simplest of all lighting measures.
- Magnetic ballast may or may not save the amount of energy implied by removing a tube. For example removing 1 of 3 tubes may not reduce energy use by 1/3.
- Most electronic ballasts are capable of operating with one less tube with no harm. If removing more than one tube, consult the ballast manufacturer to be sure. Power reduction with electronic ballast is roughly equivalent to the number of tubes removed.

ECM DESCRIPTIONS—HVAC

Seal Air Duct Leaks

Basis of savings: Reduced fan energy for a given air flow to the space. For leaks outside the conditioned space, savings are also from reduced unwanted outside air that is heated and cooled.

- Never overlook the obvious.
- In cooling season on a warm day the plenum temperature should be slightly warmer than the space below the ceiling. If it is found to be cooler in the return plenum than in the space on a warm day, then there is almost surely a duct leak.
- Duct sealing is especially important in unconditioned spaces (attics, basements) since air leaked at these points is truly lost. Duct leaks within the insulated envelope are not as critical since the heating or cooling energy is still there providing some benefit during the season.

Correct Control Valves Leaking By Internally

Basis of savings: Eliminating unwanted heating and cooling. In most cases, leaking control valves result in overlapping heating and cooling which doubles the energy waste.

- Testing consists of creating a "full close" command to the device and seeing if the downstream piping or coil returns to ambient or has measurable heating. Sometimes the first row of the coil will be found to be warmer or cooler than the rest of the coil, indicating leaks.
- Electronically actuated valves are especially prone to this due to adjustable travel stops that do not always have good residual close off seating pressure.
- For small piping, 1 inch and less, quarter-turn ball valves can be cost effective and have improved seating quality compared to conventional metal seated globe valves.

Insulate Piping and Valves

Basis of savings: Reduced thermal losses.

- Applies to both hot and chilled water systems, although hot systems represent the higher delta-T and thus the higher heat loss.
- For heating piping, safety is an added justification since many of these represent scald hazards.
- For chilled piping, corrosion protection is an added justification since the condensation that accompanies the cold surface temperature accelerates rust damage.

Lower Chilled Water Condensing Temperature

Basis of savings: Reduced refrigeration cycle "lift"

- 1-1.5% reduction in kW per degree lowered.

Raise Chilled Water Evaporating Temperature

Basis of savings: Reduced refrigeration cycle "lift"

- 1-1.5% reduction in kW per degree raised.

Air-side Economizer

Basis of savings: Avoided refrigeration compressor run time.

- Savings benefit varies by location (available hours of cool air) and by internal load characteristics (hours when cooling is needed while it is coincidentally cool outside).
- The outside air damper and relief dampers should be tightly closed whenever the equipment is off, and relief damper should be closed tightly in winter when minimum outside air is used, to avoid infiltration in cold weather through these large dampers.
- For economizer mixing boxes that include a relief damper, automatically control it to be tightly closed whenever the outside air damper is at its minimum position, and only begin opening once the outside air damper position moves beyond minimum, lagging behind the outside air damper travel.
- There should be a setting for the economizer operation (usually 55-60 degrees F), below which outside air is deemed sufficient for any needed cooling. Below this point, there should be a positive "cooling lockout" function that prevents compressor operation or forces the chilled water valve fully closed.

Water-side Economizer

Basis of savings: Cooling loads and hours concurrent with low ambient wet bulb temperature can be met with the evaporative effect of the cooling tower directly, without running the chiller.

- The best use for these is if there are steady cooling loads in winter that cannot be served with air-side economizers, often due to the seasonal outside air humidity swings.
- These are inherently not as efficient as an air-side economizer since their use depends on pumps and cooling towers as well as the air handler. However, the chiller does get to shut off so there are substantial savings.

The trouble with most flat plate heat exchanger applications is that

- They are expensive.
- Their capacity is highest when indoor cooling load are usually lowest.

- They are arranged as all-or-nothing, so they are switched off even when they could provide most of the load.
- They share the chilled water system with "non-critical" chilled water loads. Without proper controls (namely, below 55 degrees, no chilled water to these units), the system reaches the cutoff point sooner than it would need to.

Water Economizer vs. Air Economizer

When an air-economizer can be used, it can provide cooling for most/all of the hours of the flat plate unit, with less cost. This is shown in **Figure 5-1**. The determining factors are:

- The dry bulb temperature below which all cooling can be provided by the air economizer.
- The wet bulb temperature below which all cooling can be provided by the water economizer.

The example is for 35 degF wet bulb (to produce 45 degF chilled water) and 55 degF air economizer (to produce 57 degF supply air). If the chilled water or supply air temperatures can be elevated somewhat, then different crossing points will result.

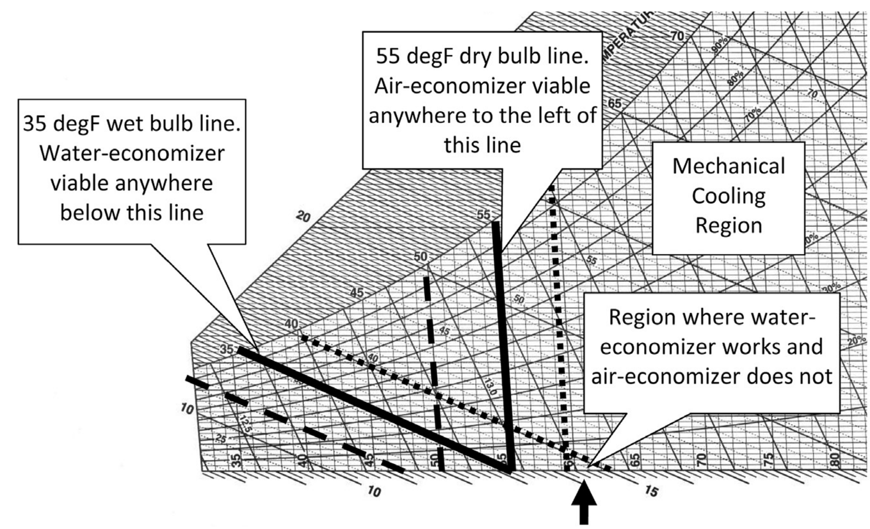


Figure 5-1. Water Economizer vs. Air-Economizer Psychrometric Crossing Point

Water Economizer Hours

To accurately determine the potential for this ECM, the load profile (at what temperatures are what loads required) must be overlaid with weather data—this establishes the coincident values of free cooling that match up with needed cooling load. The simple case is a data center which characteristically has a continuous cooling load and cannot tolerate low humidity—not a good candidate for air-economizers. The data center opportunity is then all of the hours when wet bulb temperature is low enough to achieve necessary chilled water temperature. The hours can be taken directly from bin weather tables. Note that not all of the hours are useful, so be conservative. For example, if there are two hours in a particular day when the wet bulb is low enough, it is unreasonable to expect the entire chiller plant to switch over to water-economizer operation for such a short period.

For representative maximum water economizer hours, determine the required wet bulb temperature which is:

Chilled water supply temperature
+ heat exchanger approach
+ cooling tower approach
+ pump heat allowance (usually 1 degF)

Then, find the annual hours that are below that temperature, from bin weather data. See **Appendix: Hours per Year Below Outside Dry Bulb and Wet Bulb Temperatures.**

Example: Location is climate zone 3B, 48F chilled water temperature in winter, 2F heat exchanger approach, 10F cooling tower approach,1F pump heat, and the system operates continuously. Combined approach = 13F and maximum wet bulb temperature is 48-13=35F. From the **Appendix** for 8760 hours, there are approximately 1042 hours below 35F outside wet bulb.

A partner activity to this ECM, which aims to get as many 'free' cooling hours as possible, is chilled water reset. The hours of operation are determined by when the combination of cooling tower and heat exchanger can meet the chilled water temperature requirement; when they cannot make the set point, the control option is to give up economizer mode and revert to mechanical cooling. The longer the mechanical cooling stays off the more hours in economizer mode and the more annual savings. In design, this benefit can be maximized by low approach values for the cooling tower and heat exchanger. For example, when the combined approach of the cooling tower plus heat exchanger is 15F and a chilled water temperature of 45F is needed, the maximum outdoor temperature for full capacity from the economizer system is 45-15=30F wet bulb; but if the tower/heat exchanger combination is selected for 10F approach, the system can operate at 35F wet bulb which is more hours per year. Once the equipment exists, this fundamental limit is what it is, however the same increase in hours can be achieved by raising the chilled water temperature.

- The key to longer hours is the operating wet bulb temperature which is:
  Chilled water supply temperature
  + cooling tower approach (7-15F)
  + heat exchanger approach (2-4F)
  + pump heat (probably 1F).
- When in water-side economizer mode, reset chilled water temperature upward based on zone demand or outside air, simultaneously raising the outside air wet bulb temperature at which the system reverts back to mechanical cooling.
- An enabler for raising chilled water temperature is to also raise supply air temperature leaving cooling coils.
- This cooling is in cold weather so there should not be dehumidification concerns from vapor drive or ventilation, but if there are the reset limits will be tempered as needed.

The benefit from increasing the OA wet bulb switch point can be seen directly in bin weather. If the limiting temperature used to be 35F but is now 38F, the total of hours below that temperature increase which are the hours the economizer will be able to supplant the compressor operation.

Extended Water Economizer Operation

There are some piping and pumping arrangements that will allow the flat plate system to run concurrently with the chiller, normally to pre-cool the chilled water return. This is a complex design solution but may be effective if the load profile and wet bulb temperature profile shows potential by stretching out the flat plate cutoff point. Sharing a single cooling tower, this requires some form of compensation for the chiller, since it cannot run on excessively cold condenser water. The accommodation may be throttling the condenser water flow to maintain head pressure on the chiller, or with recirculating / blending pump at the chiller condenser water inlet to temper the cold water. Where additional pumps are used to accommodate dual operation, the benefit of the partial flat plate operation must be compared with any additional pumping energy expense.

Estimating the hours of benefit involves several factors that change simultaneously, and the solution involves knowing the cooling load at various values of wet bulb, and several flat plate heat exchanger selection iterations. For maximum benefit, the flat plate heat exchanger surface area may need to be increased. A simplified example for a constant cooling load is shown in **Figure 5-2.**

- As the wet bulb temperature rises, the coincident dry bulb temperature usually rises and the magnitude of building load goes up (except for a data center or some other constant load independent of weather).
- Controlling to meet CHW supply temperatures directly , the capacity of the flat plate goes down because the approach becomes less.
- Controlling to pre-cool CHW return keeps the approach temperature high and the capacity of the flat plate high until the wet bulb temperature is within the heat exchanger approach value (within 10 degrees of CHW return temperature in **Figure 5-2**), and then will taper off.

EXTENDED WATER ECONOMIZER OPERATION FOR ONE CITY

Mode	Performance	Approx. pct Load From Flat Plate	6a-6p Annual Hours	24x7 Annual Hours	Outside Air Wb	Lowest CHW temp from flat plate (Wb + 10)
EXTENDED (add hrs)	0-5 deg pre-cool	0-50%	547	953	40-45	50-55
EXTENDED (add hrs)	5-7.5 deg pre-cool	50-75%	587	1072	35-40	45-50
CONVENTIONAL	Chillers OFF	100%	1684	3590	<=35	45

Temperatures are degrees F.
All possible water economizer hours, based on 10 degF approach.
Conventional hours assumes 45 degF chilled water temperature.
Pre-cool hours assume 55 degF return water temperature.
Wet bulb temperature data, 5 degF increments, Colorado Springs.

Figure 5-2. Sample Extended Water Economizer Hours

Chart indicates hours when the climate conditions allow economizing. Actual benefit depends on the cooling load coincident with the available economizing hours—e.g. if there is no cooling load at a particular low temperature there is no savings.

Areas with chronic high humidity climates are not good candidates for air economizers or water economizers unless there are high internal loads requiring cooling at low temperatures. The few number of hours of free cooling availability will usually not justify the expense of the equipment necessary to harvest it. This is illustrated in **Figure 5-3.**

Percent of Cooling Load Removed by Economizer

	6a-6p 4380 hours	**Standard**						**Extended**					
Zone	Balance temp, F ->	65	60	55	50	45	40	65	60	55	50	45	40
1A	Miami, FL	0	0	0	0	1	1	0	0	1	2	2	3
2A	Houston, TX	0	0	0	1	2	4	0	1	3	5	7	9
2B	Phoenix, AZ	0	0	0	1	4	6	0	1	3	7	10	12
3A	Memphis, TN	0	0	0	2	4	7	0	2	4	8	12	15
3B	El Paso, TX	0	0	0	2	4	7	0	1	4	8	12	15
3C	San Franscisco,CA	0	0	0	11	20	26	0	13	27	42	50	55
4A	Baltimore, MD	0	0	0	2	6	10	0	2	6	10	15	20
4B	Albuquerque, NM	0	0	0	3	6	11	0	2	5	10	15	20
4C	Salem, OR	0	0	0	7	15	23	0	5	14	24	34	42
5A	Chicago, IL	0	0	0	3	6	10	0	2	6	11	16	20
5B	Boise, ID	0	0	0	4	10	17	0	3	7	14	22	28
5B	Colorado Spgs, CO	0	0	0	3	8	13	0	3	8	15	21	27
6A	Burlington, VT	0	0	0	4	9	14	0	4	10	18	24	29
6B	Helena, MT	0	0	0	4	10	17	0	4	11	20	27	34
7	Duluth, MN	0	0	0	6	14	22	0	7	15	25	34	41
8	Fairbanks, AK	0	0	0	11	21	30	0	10	21	36	46	53
	8760 hours	**Standard**						**Extended**					
Zone	Balance temp, F ->	65	60	55	50	45	40	65	60	55	50	45	40
1A	Miami, FL	0	0	0	0	1	1	0	1	1	2	3	4
2A	Houston, TX	0	0	0	1	3	5	0	2	4	7	9	12
2B	Phoenix, AZ	0	0	0	2	5	7	0	1	4	8	12	16
3A	Memphis, TN	0	0	0	2	5	9	0	2	5	10	14	18
3B	El Paso, TX	0	0	0	2	6	9	0	2	5	9	14	18
3C	San Franscisco,CA	0	0	0	20	34	41	0	15	33	54	64	69
4A	Baltimore, MD	0	0	0	3	7	11	0	3	8	13	19	24
4B	Albuquerque, NM	0	0	0	3	8	12	0	3	8	14	20	25
4C	Salem, OR	0	0	0	10	22	32	0	6	17	32	43	52
5A	Chicago, IL	0	0	0	4	8	13	0	4	9	16	22	27
5B	Boise, ID	0	0	0	5	12	19	0	4	9	17	26	33
5B	Colorado Spgs, CO	0	0	0	5	12	19	0	5	12	22	30	37
6A	Burlington, VT	0	0	0	5	12	18	0	6	13	22	30	36
6B	Helena, MT	0	0	0	6	14	22	0	5	14	25	34	41
7	Duluth, MN	0	0	0	9	19	28	0	8	18	31	42	50
8	Fairbanks, AK	0	0	0	13	24	34	0	11	23	39	49	57

Notes for table:

1 "Standard economizer" cools with mechanical cooling or economizer, but not both

2 "Extended economizer" operates above the compressor cut out point, with the compressor as supplement in addition to full outside air, when conditions are favorable

3 Load is max at max dry bulb and zero at balance temperature

4 Economizer is available between set points of "cut in and cut out". For this table:
For standard economizer, cut in is 55F and cut-out is 0F
For extended economizer, cut in is 65F and cut-out is 0F

5 Compressor off below 55F for standard and extended economizer

6 if extended, economizer operation above the compressor cut in value is pro rated (shared) Pro-ration assumes 75F return air, 55F supply air. Model assumes that when outside air is 65F, outside air does 1/2 of the work, and the compressor does 1/2

Figure 5-3. Approximate Cooling Benefit from Air Economizer

Angled Filters Instead of Flat Filters

Basis of savings: Reduced average pressure drop reduces air horsepower in VAV systems and in CAV systems if re-balanced.

- Can be combined with early change out for increased benefit.

Bag Filters instead of Cartridge Filters

Basis of savings: Reduced average pressure drop reduces air horsepower in VAV systems and in CAV systems if re-balanced.

- Can be combined with early change out for increased benefit.
- Filter manufacturers caution on "dirt release" from bag filters when stopped and started regularly, therefore this measure is best suited for fans with extended run times.

Multi-zone Conversion to VAV

Basis of savings: Reduced overlapping heating and cooling and reduced energy transport work since air flow is proportional to load instead of constant volume.

- Hot deck is blanked off and zone mixing dampers sealed in the "cold deck" position. VAV boxes added for each zone, usually near the air handler and re-use the zone ductwork.
- A sketch of this modification is in **Chapter 24 "HVAC Retrofits of the Three Worst Systems."**

Multi-zone: VAV Conversion Using Existing Zone Dampers

Basis of Savings: Reduced heating-cooling overlap and reduced fan energy.

- Main fan is controlled on static pressure in the hot/cold deck plenum box.
- Existing zone dampers act as pressure dependent VAV dampers. Uses existing ducts for economy.
- Linkage between hot and cold dampers must be split and actuators provided for independent hot and cold damper control.
- Unless the zone mixing dampers are in good condition and tight sealing, it may be more cost effective to convert to VAV.
- Care must be used to provide minimums and ventilation air.

Multi-zone Conversion to Texas Multi-zone

Basis of savings: Reduced overlapping heating and cooling penalty due to the introduction of the "bypass" air path.

- This conversion makes the hot deck of a multi-zone unit into a bypass deck.

- Reheat coils are added for each zone downstream of the mixing dampers.
- The air stream delivered to the zones can be either cooled, warmed, or bypass (re-circulated), compared to either warmed or cooled.
- System remains constant volume.

Dual Duct Conversion to Separate Hot Deck and Cold Deck Fans

Basis of savings: Reduced heating burden in the hot deck.

- Independent control of hot duct and cold duct in the zone mixing boxes reduces the inherent overlap in these systems.
- Independent hot and cold duct systems via separate fans allow seasonal optimization for hot and cold decks. For example, mixing outside air to reduce cooling cost in winter adds to heating cost since both ducts share the same mixed air stream.
- Introduce the ventilation air into only one of the ducts—normally the cold duct.
- The hot deck simply re-circulates and does not see the cold ventilation air as a load.
- Additional savings are possible by converting from constant volume to variable volume in both ducts.
- A sketch of this modification is in **Chapter 24 "HVAC Retrofits of the Three Worst Systems."**

Spot Cooling

Basis of savings: Heat only the worker or process, not the whole factory.

- Concepts of "air changes" and "cfm per SF" do not apply.
- Duct size, cfm, and heat/cool energy delivered is much smaller.
- Can be very beneficial for un-insulated buildings, high bay buildings, and factories with a lot of stationary heat producing equipment (ovens, kilns, laundry, etc).
- Special design considerations for this system to be effective.

De-Stratification

Basis of savings: Heat rises so, with the temperature control near the bottom, the average temperature of a heated building will be higher for a stratified building, increasing envelope loss at upper levels. Systems that push heat down to the floor where the people are and reduce the average indoor temperature reduce the thermal drive and envelope losses.

- Savings potential is proportional to the amount of stratification—if it is the same temperature at the ceiling as the floor, there are no savings.

- Common opportunity in high spaces, such as auditoriums, theatres, gymnasiums, factories, hangars, shopping malls.
- The maximum benefit of an anti-stratification measures (e.g. fans) occurs when the temperature at the ceiling is equal to the floor. Source of savings from de-stratification :
    1. Initially, more heat is required of the HVAC system to "fill" the stratified space and turn off. This is shown graphically in **Figure 5-4.**
    2. Higher temperatures in the upper section of the envelope increase envelope losses proportionally per SF and proportionally again for the total SF since that layer includes the roof. The higher losses result in proportionally longer heating cycles.

NOTE: Although the stratification pattern may be nearly linear, the affected surface areas and U-values are not, and so attempts to estimate savings from overall U-value and overall average inside temperature reduction will understate savings considerably. Evaluation by layer is recommended, accounting for relative weighting in areas and insulation.

Example:

A warehouse with a 30-foot-high bay has a floor temperature of 65 degF and a ceiling level air temperature of 70 degF. The average winter de-

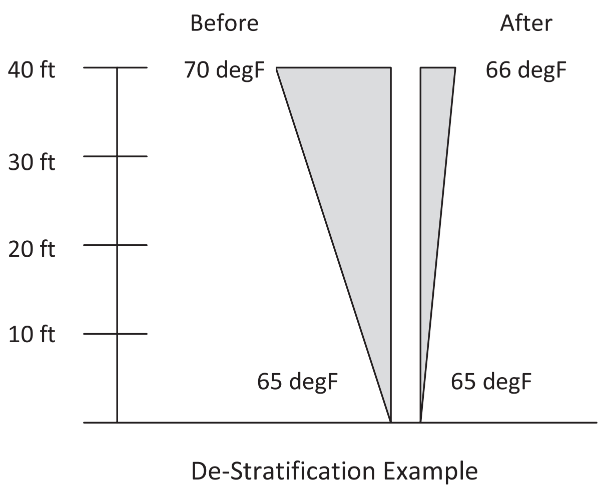


Figure 5-4. De-stratification Concept Diagram

sign temperature for this area is 20 degF. Fans are proposed to push warm air down to the floor to create a nearly even temperature at all levels. If the air stratification is reduced to degF, what is the percent of heating energy expected?

Solution:
Evaluate by layers and not average temperature.
Walls R-10, no glass, overall U-0.10, area of walls = 12,000 SF for each 10-foot rise in elevation. Roof R-20 with 5% skylights at U-0.30, overall roof U-0.0625, area of roof = 90,000 SF.

Note that the highest level accounted for most of the savings in the example, which is expected in most cases because of the dominant roof area weighting.

De-stratification Sample Calculation

Using Q=UAdT for each layer, with different dT, instead of average indoor temperature reduction and percent of change assuming a consistent U-value.

Table 5-4.

COMPONENT LEVEL METHOD - ESTIMATE BENEFIT OF DE-STRATIFICATION
Evaluate at T0=20 degF

Layer / elev ft.	Element	U-Value	SF	% area	T1 degF	T2 degF	T0 degF	Q1, MBH	Q2, MBH	BtuH saved	pct less	% energy saved by layer
40	Wall + Roof	0.0625	102000	74%	70.00	66.00	20	319	293	26	8.0%	78%
30	Wall	0.100	12000	9%	68.75	65.75	20	59	55	4	6.2%	11%
20	Wall	0.100	12000	9%	67.50	65.50	20	57	55	2	4.2%	7%
10	Wall	0.100	12000	9%	66.25	65.25	20	56	54	1	2.2%	4%
0					65.00	65.00						
			138000					490	457	33	6.7%	

Evaporative Pre-cooling for Air-cooled Condensers

Basis of savings: Reduction of dry bulb temperature from the evaporative cooling process lowers the condensing temperature and reduces refrigeration cycle lift. The air-cooled equipment "thinks" it is cooler outside and behaves accordingly.

- Approximately 1-1.5% kW reduction per degree reduction.
- Economic break even point starts around 50 tons.
- Water and waste costs compete with energy savings. Drain, fill, and freezing are design considerations.

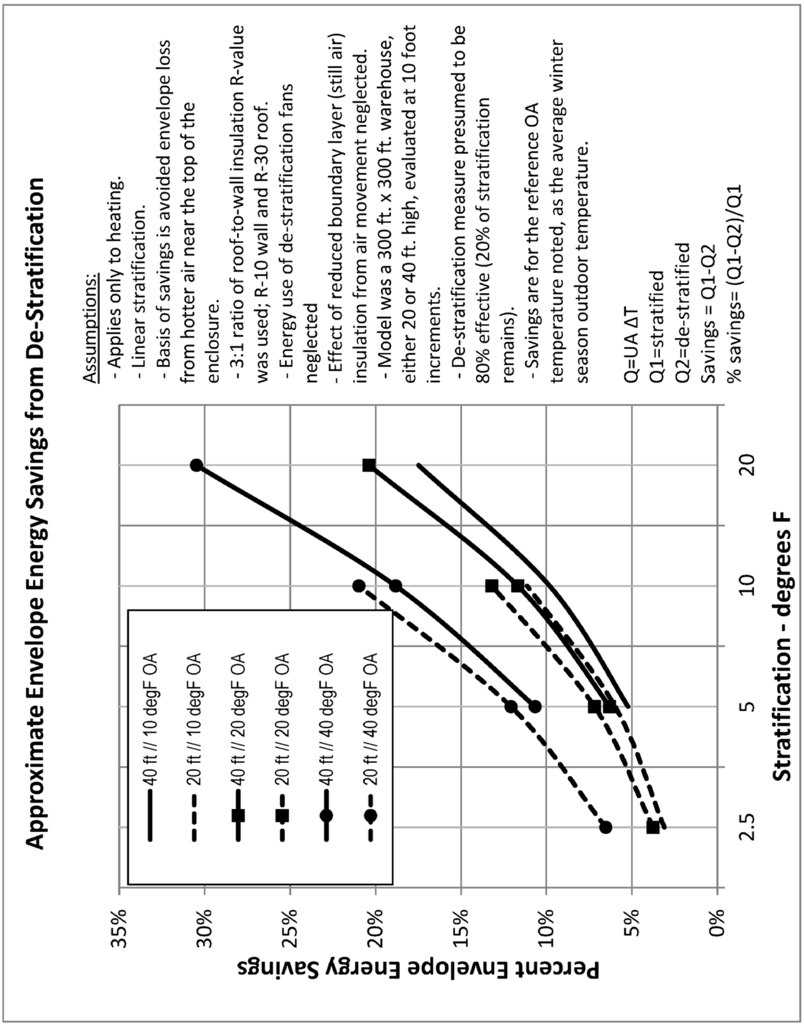


Figure 5-5. Approximate Envelope Energy Savings Potential from De-Stratification

	degF-S	40 ft-H	20 ft-H
10 degF	2.5		3.1%
Outside	5	5.2%	6.0%
Air	10	9.8%	11.1%
	20	17.5%	
20 degF	2.5		3.8%
Outside	5	6.3%	7.2%
Air	10	11.7%	13.2%
	20	20.4%	
40 degF	2.5		6.5%
Outside	5	10.7%	12.1%
Air	10	18.8%	21.0%
	20	30.5%	

Data used for Graph

ft-H Inside Building Height

degF-S Stratification Differential, degrees F

Data Used in Figure 5-5 Graph

Note: The taller the building, the higher the upper level temperature from stratification and the higher the inside-outside temperature difference. The top layer of the building includes roof area as well as wall area, i.e. the envelope surface area for heat loss is greatest at the top layer. Thus, most of the heat loss benefit from de-stratification occurs at the top of the building.

Two buildings with equal foot print areas and different heights would not have equal stratification naturally, but the upper and lower temperatures could be the same if, for example, the taller building had a partially effective de-stratification measure already in place. If the temperature at the top level were to be the same, the magnitude of loss for both buildings would be the same at the upper level. However the taller building has additional loss from additional wall area below the top level and the overall U-values of the two buildings are not the same since the proportions of wall and roof are different. Thus, for the special case of different height buildings with the same footprint and stratification values, the magnitude of savings for the taller building will be higher while the percent savings for the shorter building will be higher.

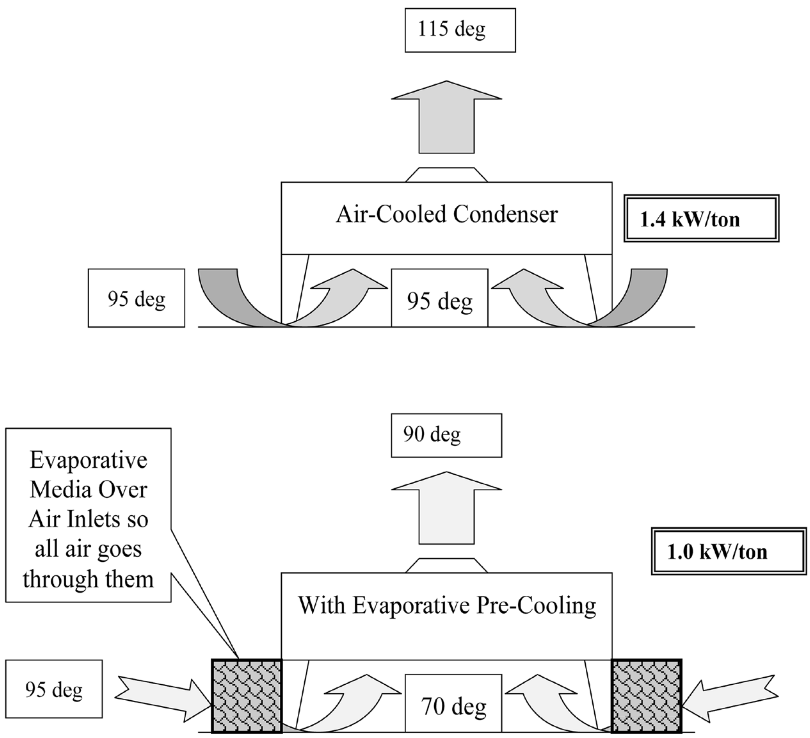


Figure 5-6. Evaporative Pre-cooling

This diagram shows the effect for air cooled equipment in a climate with design temperatures of 95 deg F dry bulb and 58 deg F wet bulb. Similar benefits are possible in different climates provided the wet bulb depression value is similar.

Adiabatic Humidification

Basis of savings: Evaporation without energy input to cause boiling.

- Includes evaporative pads, spray nozzles, atomization, and ultrasonic methods. Good results if combined with a process that simultaneously needs cooling. No energy advantage if the cooling effect is not beneficial and must be counteracted with new-energy heating.
- Psychrometric process follows the constant enthalpy line and increases moisture content as temperature drops.
- Compressed air and ultrasonic technologies each use approximately $1/10^{th}$ of the energy compared to boiling the same amount of water.

Note: all of the adiabatic evaporative methods require air at a reasonable temperature to humidify and generally will not work well in air temperatures lower than 50 deg F, therefore application is best suited to the return air stream or other tempered air stream and not in the outside air stream in cold weather.

Adjacent Air-cooled Equipment Too Closely Spaced

Basis of Savings: Correcting re-circulated discharge air condition lowers inlet air conditions so the air-cooled equipment energy use is lowered as if it were a cooler day.

- Warm air re-entrainment can occur from improper equipment spacing. Elevated intake cooling air directly raises refrigeration head pressure and compressor power by 1-1.5% per degree.
- A good rule of thumb for proper spacing is the air inlet or vertical finned coil height projected horizontally.

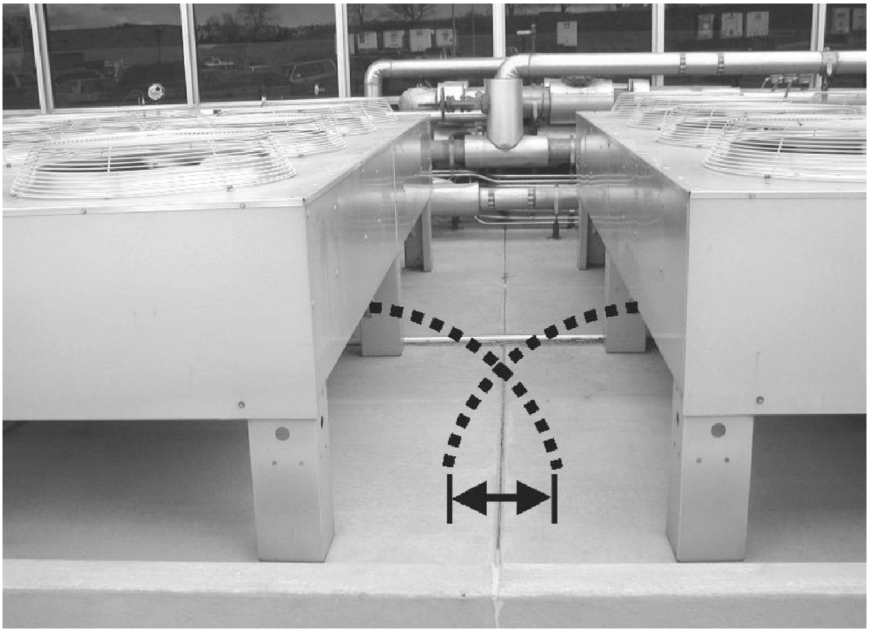


Figure 5-7. Improper Air-cooled Condenser Spacing

Constant Speed to Variable Speed Pumping Conversion

Basis of Savings: Pump energy use profile matching the heating/cooling load profile will use less energy than "constant volume" constant energy pumping.

- The key is to provide the fluid flow upon demand, but not all the time. For comfort systems, the system will be controlled to track the load profile. Applies to HVAC applications including chilled water pumping, condenser water pumping, and heating water pumping. Requires a variable speed controller for the pump and a load following signal.
- For example, if chiller load (tons) is known, the adjusted flow rate and speed for pumps can be derived automatically to proportionally follow the load change. The most substantial changes will occur at hours when load is between 50% and 100% load (flow between 50-100%).
- Additional flow savings can be obtained by increasing system delta T (dT), which allows the thermal energy transport to occur with less mass flow and pump work.
- The savings of pump energy should be weighed against any change in compressor efficiency to be sure there is a net gain, especially from condenser water.
- Chilled water flows below 2 feet per second often encroach on laminar flow and can result in compressor energy penalties that negate pump savings. The prudent approach is to verify for the specific chiller the compressor kW/ton at the proposed changed flows and temperatures to assure there is a system benefit and not just a pump benefit.

Constant Volume to Variable Air Volume Conversion

Basis of Savings: Fan energy use profile matching the heating/cooling load profile will use less energy than "constant volume" constant energy air moving.

- Applies to any air moving system, including exhaust, supply, and make-up. Applies equally to process air movement as comfort systems. The key is to provide the air flow movement upon demand, but not all the time.
- The energy savings is from having fan energy track the load profile. Applies to comfort air conditioning and heating or any other air moving task with a variable load that can be served by varying the volume of air. Requires a variable speed controller for the fan and a load following signal, commonly a downstream pressure sufficiently high to allow VAV boxes to operate.
- For a given heat load (heating or cooling) and differential temperature, the required air flow can be easily calculated and is seen to directly follow the changing load. The most substantial changes will

occur at hours when load is between 50% and 100% load (flow between 50-100%)

- Additional flow savings can be obtained by increasing system differential temperature (delta T), which allows the thermal energy transport to occur with less mass flow and fan work.

Constant Volume Terminal Reheat to VAV Reheat Conversion

Basis of Savings: Reduced overlapping heating/cooling and reduced fan horsepower.

- Each zone reheat coil is replaced with a VAV box. Most will require a heating coil unless serving only an interior area.
- Careful evaluation of the upstream duct system is required to be certain it will be suitable for the higher pressures involved.
- Heat is available only after the air flow has been reduced to "minimum cooling flow," therefore less reheating of supply air occurs.
- A sketch of this modification is in **Chapter 24 "HVAC Retrofits of the Three Worst Systems."**

Testing Adjusting and Balancing (TAB)

Basis of Savings: Reduce system pressure by excessive damper throttling, thereby reducing transmission energy requirements.

- This measure requires knowledge of existing positions of dampers or valves. In some cases, if operations staff or maintenance procedures over the years have spoiled the original balancing effort, the balancing status maybe a large unknown quantity and this measure may be as much about restoring confidence and performance as anything else. But, while it is being balanced, do so in a way that encourages low fan/pump energy input. Also, be sure to have locking provisions and permanent marks for the balancing devices, to help sustain the work.
- Impeller trimming for oversized pumps can be a big energy saver, equal to the amount of energy dissipated at the balancing valve.

Proportional Balancing Method

Described for air, but is applicable to both water and air balancing.

- The system is first measured with all dampers fully open and the fan at full output.
- With the initial iteration, each outlet is measured and given a per-

centage of the design intended flow. Some may be above and some may be below the intended flow rates.

- The outlets are numbered in order of increasing percentage of design airflow. The outlet with the lowest percent remains open and is not adjusted. Its percentage is designated as B and is the basis for other branch adjustments.
- The flow rate on the branch with the next lowest percentage is adjusted so that it has the same percentage as B. All other branches are adjusted to this same value of B.
- When that initial step is complete, then adjust the main fan capacity (by dampers, sheaves, motor speed, motor change, etc.) to achieve full design capacity.
- Then return to the individual outlets and spot check at least 20% of them to assure that they are within the stated tolerance. If they are not, then repeat this process iteratively until they are. At the conclusion, there is still at least one damper that is fully open, to minimize overall system loss.

Remove Inlet Vanes or Discharge Dampers

Basis of Savings: Reduced System Pressure and Fan Hp.

After converting to VFDs, these devices create unwanted air pressure drop even in the wide open position.

The wide-open loss depends on the free area (FA) of the damper and the air velocity. Inlet vanes and control dampers are typically around 80% FA.

Figure 5-8. Inlet Vane and Outlet Damper Pressure Drop Loss

Source: *American Warming & Ventilating*

Based on the following relationship and Free Area Factors.

$$\textbf{(FA Factor)(Velocity/4005)}^2$$

<u>FA Factors</u>

70% - (0.624)

75% - (0.434)

80% - (0.306)

85% - (0.224)

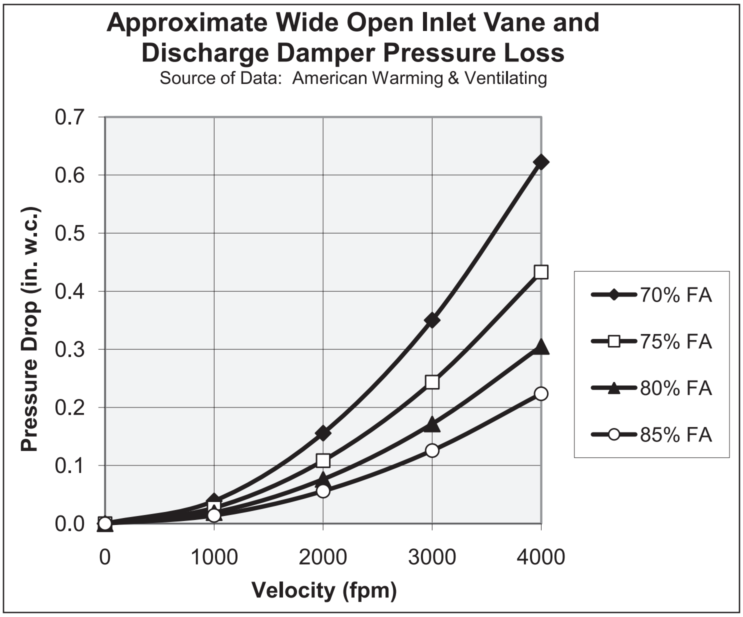


Figure 5-8. Inlet Vane Pressure Drop

Lower VAV Duct Static Pressure Control Setting
Basis of Savings: Power is reduced exponentially with a reduction in pressure.

See Chapter 15—Savings Impact When Controlling to a Constant Downstream Pressure—VAV and Variable Pumping

See Chapter 15—Savings From Lowering Downstream Maintained Pressure Setting

Reduce Resistance in Distribution Ducts and Pipes
Basis of Savings: Power is reduced with a reduction in resistance.

This model considers removing an obstruction or otherwise 'lowering the bar' of the whole system for pressure requirement and applies to both constant and variable flow.

Before considering changes to equipment and controls, consider the source and the distribution system. Begin by using less if possible: less air, less water. Do you need all that air or water circulating?

Possible ways to reduce system resistance:

Water

- Remove balancing valves after conversion to VFD
- Re-balance so at least one balancing valve is fully open
- Clean strainers earlier
- Replace restrictive main line strainers
- Replace restrictive main line flow meters (orifice plates)

Air

- Remove old inlet vanes or discharge dampers after conversion to VFD
- Re-balance so that at least one balancing damper is fully open
- Clean coils
- Change filters earlier
- Extended surface or angled filters
- Eliminate unnecessary duct appurtenances
- Smooth out 'bad' duct fittings

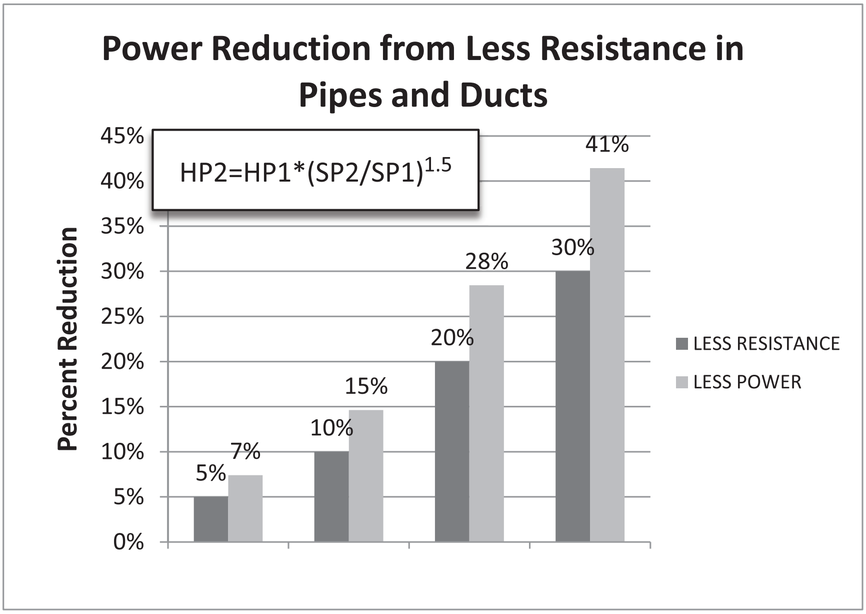


Figure 5-9. Power Reduction from Reduced Duct Friction and Pipe Friction
For systems with friction as the dominant load. Not for systems with a significant portion of total work from lift or static head.

Correcting Bad Duct Fittings, Entrance Losses, Exit Losses

Basis of Savings: Reducing system pressure reduces fan horsepower requirements.

The basic relationship for air horsepower is

$$\mathbf{HP(air) = CFM * {}^{TSP}/_{6356}}$$

Where CFM is cubic feet per minute air flow and TSP is the total static pressure, in. w.c.

Motor Hp is higher than air horsepower, because losses from fan efficiency, fan drive, and motor efficiency must be incorporated.

See **Chapter 11—Mechanical Systems "Fan/Pump Motor Work Equation"** and **"Fan and Pump Efficiencies, and Belt Drive Efficiencies"**

See **Chapter 12—Motors and Electrical Information "Motor Efficiencies"**

For a given air flow, reducing TSP directly reduces horsepower. In addition to the friction losses, coil and filter pressure drops etc., part of the work of the fan is fittings and entrance/exit losses. When one of these is especially bad, the pressure drop can be unusually high and an opportunity for improvement.

Some Rules of Thumb:

- Transitions should be no more than 30 degrees, and 15 degrees per side is better.
- Abrupt (blunt) changes in duct geometry are usually bad, unless velocity is very low (<1000 fpm)
- Square elbows should always have turning vanes
- Velocity is usually highest at the fan outlet
- Fan discharge conditions are high loss areas unless they have smooth transitions
- Losses increase exponentially with velocity
- Many bad duct fittings are there because of lack of room and don't have good solutions

- HVAC air velocity over 2500 fpm requires special care in design to avoid high losses

Losses are normally calculated in terms of "number of velocity heads" (Hv), where tables provide the "C" factor based on testing.

$$\mathbf{Hv\ (air) = (^{V}/4005)^2}$$

where Hv is in. w.c., and V is velocity in feet per minute

Example:

A poor duct fitting has a C-factor of 0.9 and can be replaced with one having a C-factor of 0.20. What are the savings in hp if the air flow is 20,000 cfm and the duct velocity is 2000 fpm at that point?

Ans:

Hv = $(2000/4005)^2$ = 0.25 in. w.c.
Savings is the differences of "C," which is 0.9-0.20 = 0.7
Reduced pressure is 0.7 velocity heads, or 0.7 * 0.25 = 0.175 in. w.c.
HP = (20,000 * 0.175)/6356 = 0.55 hp

Often bad duct fittings are accompanied by noise, either from the fitting or from the fan laboring to move air through the turbulence.

See **Appendix "Duct Fitting Loss Coefficients"** for a list of some bad duct fittings and their "C" factors.

Indoor Cooling Tower Sump

Basis of savings: Eliminates basin heater freeze protection load.

- For roof-mounted cooling towers, the conversion is relatively simple. For ground-mounted cooling towers, the tower will usually require elevating onto a platform.
- See **Appendix: "Cooling Tower Cold Water Basin Heat Loss."**

ECM DESCRIPTIONS—BOILERS AND DISTRICT HEATING

Note on Condensing Hot Water Boilers

Very high efficiencies are possible with condensing boilers, but only when inlet and average heat exchange temperatures are sufficiently low to achieve condensing.

Example 1: operating at 180 degF for HVAC heating hot water will not result in condensing and efficiencies of a boiler labeled at "95%" will be in the 85% range.

Savings from these boilers depend upon the connected system and whether it will operate at lower temperatures.

Example 2: HVAC heating coils designed to provide necessary heat at 200 degF will not adequately heat the building if provided with 140 degF water, thus limiting the amount of reset and energy savings.

To get full benefit from a condensing boiler may likely take the project past the boiler room, and can require system changes such as 4-row heating coils sized for 140 degF water. Deep resets that lower the supply water temperature—equipment that cannot accept cold return or supply hot water and must operate on minimum temperatures for the sake of the materials can give up considerable savings.

Distributed Heating Instead of District Heating

Basis of Savings: Reduced thermal losses at part load.

There are some good reasons to use district heating.

- Centralized maintenance
- Aesthetics
- Possibility to couple waste heat from another process (cogen) and distribute.

However, energy of this system is usually higher than for a distributed heating system, where heating equipment is located closer to the point of use. The main reason for this is the thermal line losses. While these may be kept to a tolerable level at design loads, unless the heating fluid temperature is reset in mild weather, these losses are constant and form a larger and larger fraction of the total energy use at part load. Historically, larger boilers were more efficient than smaller ones, but this is no longer the case.

Options:

1. Mitigate standby losses by de-centralizing heat uses during summer months, allowing the plant to fully shut down for that period. Savings will be the standby losses for those months.

2. Reduce system temperature (hot water systems) during summer months, to proportionally reduce the standby losses.
3. Reduce distribution losses through improved piping insulation.
4. Eliminate the distribution losses entirely by using distributed heating units in lieu of district heating.

Figures 5-10A, 5-10B show overall system loss at part load, given full load thermal losses. These losses (2%, 5%, 10%, 20%) represent total distribution thermal losses (piping losses).

Figure 5-11 Example Energy Use for One District Heating System This system was kept running in summer for domestic hot water loads. Pct system load is actual, standby losses estimated. This underscores the poor thermal efficiency of these systems at low loads.

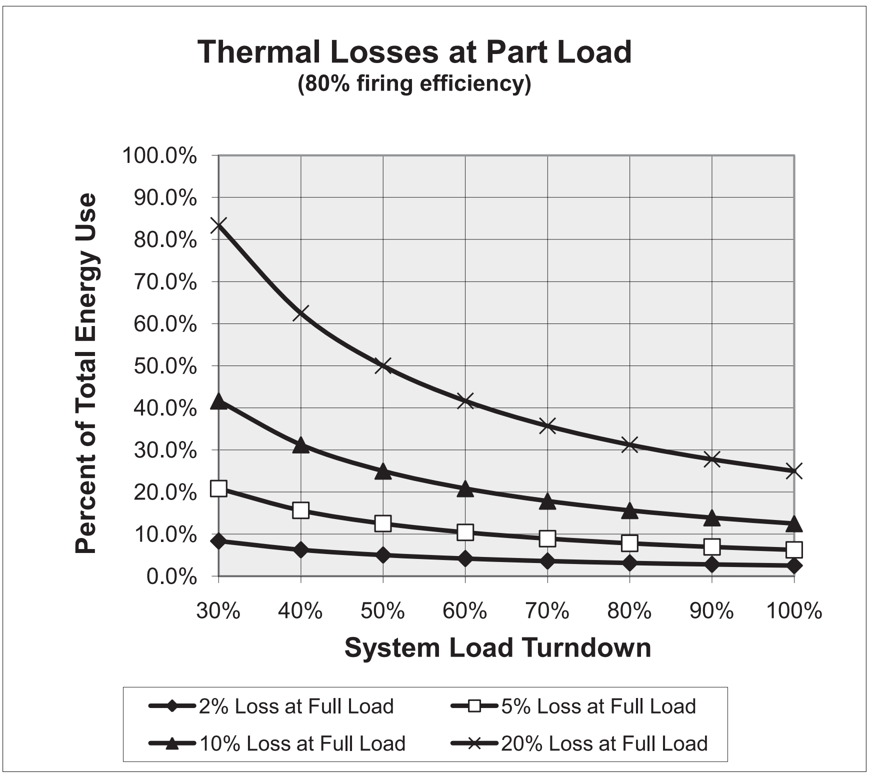


Figure 5-10A. Thermal Losses at Part Load

Jockey Boilers
Basis of Savings: Reduced standby and cycling losses.

Discussion is for heating. Concept applies to cooling, pumping, air compressors, etc.

Heating systems are designed for a designated maximum load, and with the ability to throttle to reduced loads—within reason. Hot pipelines, pumps, valves and equipment shells all surrender some loss to the environment, even when insulated. These losses are normally small in proportion to the design load, but become a greater and greater portion of the total energy at reduced load. A heating or cooling system operated at 10% of design load may have half of the energy expense given to these 'overhead' losses, making the overall system efficiency very poor at those loads. Ideally, the load would not be so low without the system shutting

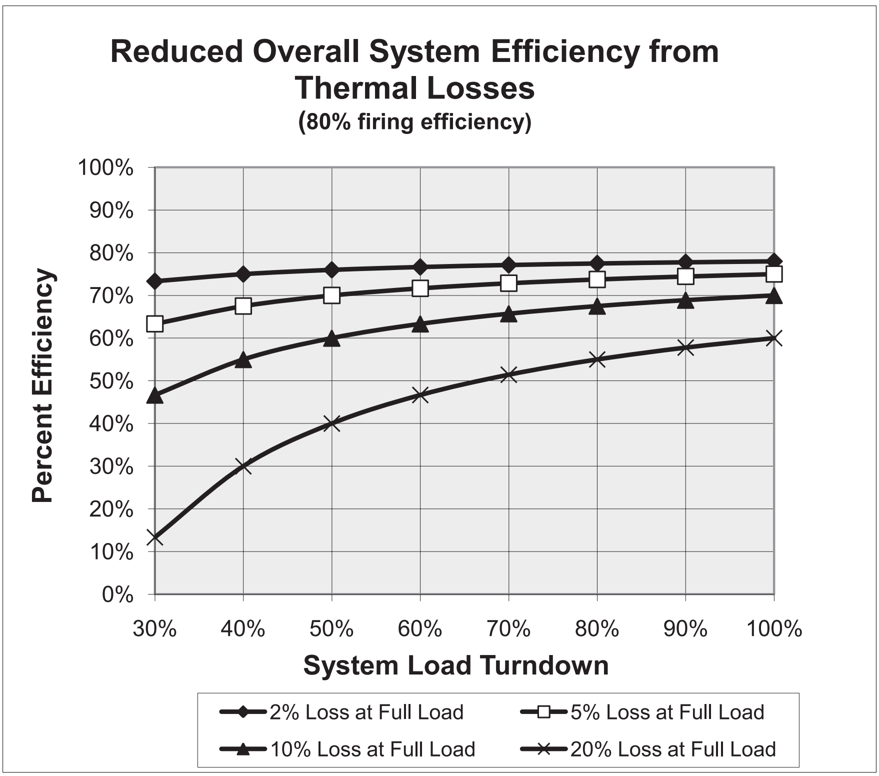


Figure 5-10B. Overall System Efficiency Loss at Part Load

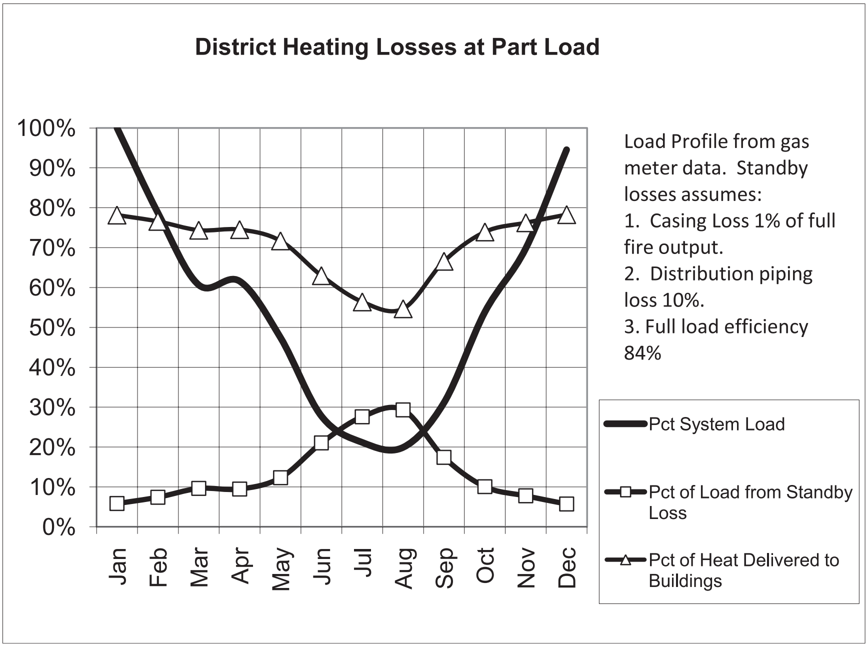


Figure 5-11. Example Energy Use for One District Heating System

off for the season, but when large equipment stays running with very low load the energy efficiency suffers.

There is some level of 'turndown' that all heating systems will experience. If the primary equipment, operating multiple units in sequence, can run at 50% capacity or greater while serving the minimum load, the central plant equipment will not be a large contributor to part load losses. But if not, then a jockey (small) seasonal boiler may be cost effective. Wear and tear costs are usually gauged by run hours, so reducing run hours on a large machine is an additional savings, because the jockey boiler allows it to rest.

The jockey boiler concept is the basis of savings for "modular" (incremental) boiler systems.

Stack Dampers

Basis of Savings: Reduced thermal losses at part load.

All combustion equipment has a tail pipe, and most are arranged for vertical discharge. Some systems use all or part of the buoyancy effect of the hot gases, but in all cases the vertical arrangement forms a chimney that can expel building air as easily as combustion products. When boilers cycle, the stacks allow natural convection to siphon heat away from the boiler, making the next cycle of firing come sooner and using more energy. A stack damper is a device that closes the stack when the equipment is off, breaking the natural chimney effect and holding the heat inside the equipment. Stack dampers are readily available for small gas-fired appliances, and can be field built for any size boiler stack. Careful attention is required for the safety interlock, to prevent the equipment from firing when the damper is closed.

See also **Chapter 13: Savings from Various Boiler Improvements**

Boiler Isolation Valves

Basis of Savings: Reduced thermal loss during off cycle, improved thermal efficiency

When hot water is pumped through a boiler that is off, the boiler heat exchanger acts in reverse and the boiler loses heat—becomes a heater. How effective the 'heater' is depends on several factors including the amount of surface area for conduction to the room, the temperature of the room, and how well the chimney effect will be to sweep the heat out of the boiler through the flue. When the loss is high, it can be mitigated with stack dampers and/or isolation valves.

In some cases, the heat loss is great and there is a measurable difference in water flow in and out, but this is not a consistent observation so attempts to arrive at a rule of thumb for all hot water boilers has not been successful. In general:

- Stack dampers stop chimney losses for all fired appliances
- Gravity flue systems without stack dampers are worse than forced draft systems.
- Forced draft systems with burner air dampers that close tight effectively stop the chimney action.
- Multiple boilers/appliances sharing a common flue will have a chimney effect applied any time at least one unit is firing.

In the case of a single pump delivering only half of the design water flow through a firing boiler, there are additional savings from adding isolation valves, because the other half is going through the 'off' boiler—giving the boiler only half of the water flow it is expecting. The reduced water flow will result in a higher temperature rise through the running boiler (mixing with the other one) and notably higher flue gas temperature as a result. Efficiency loss is approximately 1 percent for each 40 degF rise in stack temperature.

See also **Chapter 13: Savings from Various Boiler Improvements**

Boiler Combustion Fan Control

Basis of Savings: Reduced auxiliary energy use at part load.

A common control of combustion air in a boiler is a constant speed blower that is throttled with a discharge damper; this is true for packaged boilers and larger boilers alike. Like with heating water pumps, a similar argument can be made that the wasted fan energy results in beneficial heating of the air, and so the analysis of the value of correcting the "waste" is one of fuel cost: electric heating vs. combustion heating. Some packaged boilers have very large fan motors— e.g. 50 Hp—although most are smaller. While it is easy to provide variable speed to a motor, in this case it must be done very carefully to assure that air-fuel mixture control and boiler safety are preserved. One of the conveniences of the discharge damper control method is that it can be easily mechanically linked to the gas valve throttling mechanism, such that they modulate together; this arrangement has been used successfully for years and boiler manufacturer's will stray from such success tentatively.

Providing variable speed control of the forced draft (FD) and induced draft (ID) fans severs the mechanical link and delegates it to automatic controls. For example, increasing and decreasing firing rates requires the two quantities to move in unison, especially when excess air is low, to prevent unsafe fuel-rich mixtures. With a mechanical linkage, the manufacturer is assured this will happen in lock step, but once air and fuel are separated the assurance must lie elsewhere. These ECMs are worth pursuing, but need to be done with close participation of the boiler or burner manufacturer.

Boiler Economizers

Basis of Savings: Improved thermal efficiency from capturing waste heat.

Boiler economizers are heat recovery devices that pre-heat the incoming water, feed water, or incoming combustion air, thereby increasing the thermal efficiency of the boiler (useful heat / heat input). Like all heat exchangers, the cost goes up as the approach goes down, so these are more cost effective on boilers with high stack gas temperatures. For practical purposes, high end condensing boilers already have an economizer built in via larger heat exchangers and the capability to operate on lower return water temperatures to leverage it. It follows that economizers are probably not viable on condensing boilers used for building heating, swimming pool heating, etc. Process boilers that operate at 500 degrees F stack gas to heat a manufactured product would be a good candidate for an economizer. Facilities that use steam for heating year round (hospital) operate at stack temperatures around 350 degrees F and, with the long hours of operation, can be viable for an economizer ECM.

The usual application of a boiler economizer is to pre-heat its own air or water, although this is not a requirement. If a coincident need for lower grade heat exists and the proximity and quantities are a match, heat recovery is always possible, subject to the economics. Interactions with these ECMs include condensation, corrosion and loss of natural draft, both a result of cooling the stack gas.

Preheat Combustion Air (Economizer)

Basis of savings: Increases combustion temperature and heat transfer.

- Approximate savings are 1% efficiency increase for each 40 degF of preheat.

Preheat Feed Water (Economizer)

Basis of savings: Reduces the cooling effect of the feed water on the boiler water, reducing heating load.

- Approximate savings are 1% efficiency increase for each 10 degF of preheat.

Lower Steam Pressure

Basis of savings: Several Factors:

- Lower steam pressure means lower steam temperature, and heat equal heat transfer occurs at reduced combustion temperatures. Thus, reduced pressure reduces stack temperature and casing radiation losses.
- Reduces feed water pressure and pumping requirements
- Reduces losses from leaks
- Reduces condensate temperature and attendant losses
- See also **Chapter 13—Savings from Steam System Improvements**.

Lower Boiler Excess Air

Basis of savings: Reduces the cooling effect of the extra air, increasing combustion temperature and heat transfer. Excess air 'sweeps' heat out of the boiler with no benefit.

- Approximately 0.4-1.4% efficiency gain for each 10% reduction in excess air, depending on the stack temperature and fuel.

See also **Chapter 13— Savings from Reducing Excess Air**

ECM DESCRIPTIONS—SWIMMING POOLS

Pool Covers

Basis of savings: Isolates the water surface from air, preventing evaporation.

Reduce Pool Evaporation

Basis of savings: Water lost through evaporation absorbs heat, 1000 Btu per pound, cooling the remaining water. Additionally, the water lost is replaced and must be heated to pool temperature. Reducing evaporation reduces these losses directly.

Pool energy use is mostly due to water heating, and heating burden (other than initial heating) is largely due to evaporation, so controlling evaporation and other water losses is an important consideration in an energy program. Evaporation is strongly affected by two things: *wind speed* at the surface and the *differential vapor pressure* between the water and air, so energy use from pool evaporation can be reduced by:

- Lowering the pool temperature
- Raising the surrounding air temperature
- Raising the surrounding air humidity
- Lowering the wind speed at the air-water surface interface

Note: HVAC systems that exchange air to dehumidify should do so based on relative humidity controls, and excessive drying by using outside air will increase evaporation.

Recommended Pool Water and Air Temperatures

Surrounding air temperature should generally be higher than and within 2 degrees of the water temperature to reduce evaporation, although air temperatures over 85 deg F are not recommended for comfort reasons. Higher humidity lowers evaporation rates, although humidity over 60% is not recommended to reduce risk of biological growth.

Table 5-5. Recommended Pool Air and Water Temperature

Type of Pool	Air Temperature, deg F	Water Temperature, deg F	Relative Humidity, pct rH
Recreational	75-85	75-85	50-60
Therapeutic	80-85	85-95	50-60
Competition	78-85	76-82	50-60
Diving	80-85	80-90	50-60
Whirlpool/Spa	80-85	97-104	50-60

Source: ASHRAE Applications Handbook, 1999, © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org.

Pool Evaporation Facts

Actual values will vary depending upon individual conditions, but the basic relationships can be illustrated as follows:

- At 50% rH, increasing wind speed over the water increases evaporation rate 30% for each 1-mph of wind speed increase.
- At 50% rH, reducing surrounding air temperature increases evaporation by 4% for each 1-degree lowered.
- At 60% rH, reducing air relative humidity over the water increases evaporation 2-3% for each 1-pct rH lowered.
- Mechanically heating the pool air over the water is 15% more effi-

Simplified Calc for Pool Evaporation
ASHRAE 1999 Applications Handbook

wp=0.1A * (pw-pa) * Fa

wp = evaporation of water, lb/hr
A = area of pool surface, ft^2
pa = saturation pressure at room air dew point, in. Hg
pw = saturation vapor pressure taken at surface water temperature, in. Hg
Fa = Activity Factor:

Residential pool:	0.5
Condominium	0.65
Therapy	0.65
Hotel	0.8
Public, schools	1.0
Whirlpools, spas	1.0
Wave pools, water slides	1.5

Figure 5-12. Simplified Pool Evaporation Formula

cient than using the pool to heat the surrounding air, due to reduced evaporation.

ECM DESCRIPTIONS—HEAT RECOVERY

General Criteria

The term "heat recovery" is the commonly used term for a variety of systems that recovery a portion energy from a waste stream and put it to use, displacing new energy consumption that would otherwise occur. "Energy recovery" would be a more descriptive term since the concept applies to more than just heat. The most common application in industry is taking waste heat and using it to heat or pre-heat another mass that needs heating anyway, thereby saving fuel. The same principles apply for air or water flows needing to be cooled anyway—cooling or pre-cooling from waste streams that are cooler. Humidity recovery is also viable for large air streams by pre-humidifying from a more moist exhaust air stream, or dehumidifying make-up air with a drier exhaust air stream. In all cases, heat recovery serves to reduce new energy input by re-using waste products. While the goal is admirable, there are a number of practical barriers.

Heat Recovery Viability Test

For heat recovery to viable, it must meet these requirements:

1. *Same Time*. Generally, waste heat and the need for heat must occur concurrently. Thermal storage could assist this, although this is rare. To illustrate, drain waste pre-heating domestic hot water makes sense for a shower, but not for a bath.

2. *Right Proportions*. The only amount of recovered heat that counts is where the available heat and needed heat are equal. When there is an abundance of either waste heat or needed heat, it is the smaller of the two that determines the actual energy recovery effect. To completely eliminate the new-energy heating and supplant it entirely with waste heat, the waste heat quantity must exceed the need for heat recovered heat. Best economics occur when the two are in similar proportion.

3. *Right Temperatures*. Waste heat must be at a sufficiently higher temperature than the recovered heat sink to allow heat transfer. The greater the differential temperature available, the more economical

the apparatus due to the higher "approach" temperature there is to work with.

4. *Enough Recovered Heat to Make it Pay*. There must be sufficient heat transfer throughout the year to provide economic justification for implementation. One rule of thumb is 5000 CFM of outside air intake with equal exhaust nearby. Single pass systems (outside air make-up, raw water heating) are good candidates since they are energy intensive. Facilities with a large number of operating hours per year, especially continuous operations, provide quicker returns than others.

Heat Recovery Application Notes

- New construction can benefit the most from heat recovery proposals since the cost of the recovery equipment is subsidized by the reduced size and cost of the primary heating equipment that would otherwise be provided. The down side to this is that the facility now becomes dependent upon the heat recovery system, and so its proper sizing, application and maintenance for sustained operation become more critical.
- For retrofits the heating equipment is backup and the recovery benefit simply reduces its load. Savings in energy alone must pay for the recovery equipment and so returns are longer.
- Recovered heat energy is penalized from additional air horsepower from pressure drop through coils, pumping energy, standby losses, and from capital investment requirements.
- When either the waste heat or recovered heat temperatures vary over time, there can be times when heat transfer is marginal, or when heat recovery is detrimental. Whenever the differential temperature is near zero, the actual heat transfer will be very low and the energy used by active system components (fans, pumps) may not be justified. Controls monitoring the differential temperatures should be used to determine when to shut down the system.
- For *refrigeration* system heat recovery, a good rule of thumb is 4000 Btuh per ton of refrigeration capacity available for hot water heat recovery.

Exhaust–to-Make-up Air

Basis of savings: Reduced heating and cooling energy for make-up air, compared to 'raw' outside air.

Examples:
- Gas clothes dryer vent used to pre-heat incoming combustion air.
- Paint booth exhaust pre-heating make-up air.
- Building exhaust pre-heating make-up air.

Rejected Heat-to-Make-up Water

Basis of savings: Reduced heating energy for make-up water, compared to 'raw' make-up water.

Examples:
- Boiler economizer, pre-heating make-up water.
- Refrigeration hot gas (before going to the condenser), pre-heating make-up water.

Rejected Heat-to-Space Heat

Basis of savings: Reduced heating demand on the space heating system.

Examples:
- Waste heat off an air compressor that can warm a section of a factory.
- Refrigeration system rejected heat used as heat in an air handler or as reheat in a dehumidification cycle.

Wastewater-to-Make-up Water

Basis of savings: Reduced heating energy for make-up water, compared to 'raw' make-up water.

Examples:
- Commercial laundry waste water heat exchange to pre-heat wash water.
- Injection mold cooling jacket water to warm process water or boiler feed water.

Combined Heat and Cool: The Water-to-Water Heat Pump

Basis of savings: Compound savings: cooling energy saved and heating energy saved. Usually expressed as cheap heating and free cooling.

- Waste refrigeration heat can be used as a primary heat source, provided there is a concurrent need for both heating and cooling. The chiller pre-cools the chilled water return, shedding load on the main chillers, and the condenser becomes a water heater.
- Special equipment is needed to achieve refrigerant hot gas temperatures sufficient to make space heating water temperatures of 140 deg F or higher. Coefficient of Performance values (COPs) go down at these temperatures but are still around COP=2.0 which is not great

for a cooling machine but is very respectable for a heating machine.

Recovery of Humidified or De-humidified Air
Basis of savings: Where humidified or dehumidified exhaust occurs, the energy normally used for humidifying or dehumidifying the make-up air can be recovered.

- Special heat wheels with desiccant or other moisture holding material are used.

Examples:

- Building exhaust pre-heats and pre-humidifies outside air intake in winter.
- Building exhaust pre-cools and pre-dehumidifies outside air intake in summer.

Table 5-6.
Air-side Heat Recovery Equipment Efficiency Guidelines

Type	*Efficiency (note 1)*
Heat Wheel	60-80%
Heat Pipe	60-70%
Plate-Box	60-80%
Run-Around Coil	40-60%

Note 1: thermal heat recovery potential. Does not include parasitic losses of fan or pumps, or added resistance of the recovery equipment in the fluid path. There are a number of variables that determine application efficiency such as relative temperatures and proportions of flow rates. Values vary by manufacturer—for example some low end commercial heat recovery wheels may be less than 50% efficient.

Double Use of Process Air and Water in Heat Recovery
Basis of savings: Both the energy and fluid itself are re-used directly, without heat exchange apparatus. The second point of use is seen as 'free'. Note that consideration of contaminants as well as implications of failure and shutdown of individual equipment is required.

- General exhaust from a theater used as make-up and cooling for a projector

- General exhaust from a building used as make-up for toilet exhaust or kitchen exhaust
- Steam condensate not being returned added directly to wash water
- Waste heat exhaust from an air compressor intercooler used for boiler make-up air in winter
- Single pass refrigeration cooler water used for cooling tower make-up

ECM DESCRIPTIONS—THERMAL STORAGE (TES)

Basis of savings: Savings for all thermal energy storage cooling systems is the ability, via storage, to use energy during off peak times when it costs less.

TES Pros and Cons

- Full storage systems are capable of keeping the chiller off the entire on-peak time. The storage systems cost more since the number of ton-hours is higher. However, conversions of existing conventional chiller plants to TES may be sufficiently sized and are candidates for full storage if the refrigeration equipment is in good condition with life remaining in the equipment.
- Partial storage TES systems serve to defray part of the on-peak demand and flatten the electrical load profile. They run concurrently with the refrigeration system during the day and run at night to re-charge the storage. This means the refrigeration equipment runs almost continuously. Still, these systems, especially on new construction, are less expensive to install.
- Flexibility is a key detriment to most of these systems. Even a properly sized system can be rendered obsolete if the rates change, and the chances of rates changing during a 20-year equipment life cycle are very good.
- Cool storage and warm storage systems are all plagued with stratification losses. Various attempts have been made to deal with it, and it remains an engineering challenge.
- Systems with cyclic freeze-thaw are often plagued with expansion damage. Design must include ample provision to accommodate these forces.
- For warm or cool storage, the container volume is an order of magnitude larger with correspondingly greater surface area (compared

to ice), and standby losses become increasingly important. Standby losses are probably on the order of 20% for warm or cool storage.

- Minimum of 25% energy penalty in ice making mode, even with lower condensing temperatures at night.
- TES normally only makes economic sense when there is a large rate incentive for off-peak use, and never makes sense if energy conservation is important. If the rates are there, these systems can save utility bill money, but almost always use more energy, and almost always cost more to install.
- Significant off-peak utility rate discounts for energy and demand are usually required to make such systems attractive. However, other considerations may make thermal storage a good choice, such as ride-through back up for critical cooling applications, allowing hours of cooling in the event of a power loss while using generator power for circulating pumps only.

Rules of Thumb for TES Systems

- Cool storage: 100 gal/ton-hr (15 deg dT), 150 gal/ton-hr (10 deg dT)
- Encapsulated ice: 17-22 gal/ton-hr
- Ice on coil: 18-26 gal/ton-hr
- Installed TES system (ice), all types: approx $100 per ton-hour.
  Source: Cryogel, 2007.

Conditions Favoring Thermal Energy Storage

- Average cooling loads are much less than the peak cooling load.
- Large differential between on-peak and off-peak energy and demand charges.
- Low off-peak demand charges.
- High number of seasonal cooling (or heating) hours and ton-hours load.
- Utility incentives to defray first cost.
- Available space for storage containers.
- Higher-than-average operational staff technical expertise.

Cool Storage

- Chilled water is created during off peak times when power costs are less. The chilled water is stored in a large tank and used during the day allowing the chillers to be turned off.

Ice Storage

- Ice is created with low temperature brine during off peak times when power costs are less. The chilled water is stored in a large tank and used during the day allowing the chillers to be turned off.

Phase Change Material (PCM) Storage

- Same as ice storage, except PCMs can be selected for phase change at temperatures closer to utilization temperature (45 deg F) instead of ice (32 deg F). This technology has the potential to leverage the compactness advantage of ice storage equipment without the 25% inherent energy penalty.

- PCM barriers include cost, as well as corrosive properties for some. When encapsulated, heat transfer is slowed and approach temperatures can be high, as much as 5-9 degF.

- An enabler for savings using PCMs with chilled water is locating the TES unit in the return water rather than the supply for partial storage systems. In practice, a portion of the cooling load is negated by the TES and chiller load is reduced. By choosing return water as the location, the PCM can be selected at a higher phase change temperature (e.g. 50F) which then allows the chillers to 'freeze' the PCM at a higher temperature with less lift and power than is required to make ice or cold water storage.

Cool Storage—Evaporative Cooling

- In dry climates, it is possible to use evaporative cooling at night in conjunction with cool storage, to reduce or eliminate mechanical refrigeration. This is a very good way to save energy in refrigeration—by turning it off completely—however pumping costs will be higher and will offset some of the savings. Cooling season wet bulb temperatures in the mid-40s and low 30s are needed to drive this. In dry climates, this variation has strong promise.

TES option comparison	Conventioal mechanical refrigeration	FULL STORAGE chilled brine ice storage	PARTIAL STORAGE chilled brine ice storage	FULL STORAGE chilled water cool storage	FULL storage evaporative cooling with cool storage
Max cooling load, tons	500	500	500	500	500
Annual load, ton-hrs/yr	720,000	720,000	720,000	720,000	720,000
Storage stand-by loss %	0%	5%	5%	20%	20%
% system storage	0%	100%	50%	100%	100%
Storage stand by losses	0	36,000	18,000	144,000	144,000
Total cooling load, ton-hours	720,000	756,000	738,000	864,000	864,000
Tons capacity	500	350	175	350	350
hours of storage	0	10	10	10	10
Design daily storage load, ton-hrs (tons*hrs*storage factor, incl storage loss)	0	5,250	2,625	6,000	6,000
Storage tank size, gal	N/A	95,000	47,500	600,000	600,000
Chjiller kW/ton	0.60	0.75	0.75	0.60	0.00
Auxiliary kW/ton	0.20	0.30	0.30	0.30	0.30
Total kW/ton	0.80	1.05	1.05	0.90	0.30
Total kW demand (kW/ton * installed tons)	400	368	184	315	105
Annual energy use, kWh (ton-hrs * kW/ton)	576,000	793,800	774,900	777,600	259,200
kWh on-peak	200,000	0	150,000	0	0
kWh off-peak	376,000	793,800	624,900	777,600	259,200
kW demand on-peak	400	0	184	0	0
kW demand off-peak	400	368	184	315	105
kW demand off peak above on-peak	0	368	0	315	105
$/kWh on peak utility cost	0.06	0.06	0.06	0.06	0.06
$/kWh on peak utility cost	0.03	0.03	0.03	0.03	0.03
$/kW-yr on-peak utility cost	120	120	120	120	120
$/kW-yr off-peak utility cost	80	20	80	20	20
$ for kWh on-peak	12,000	0	9,000	0	0
$ for kWh off-peak	11,280	23,814	18,747	23,328	7,776
$ for demand on-peak	48,000	0	22,050	0	0
$ for demand off-peak	0	7,350	0	6,300	2,100
$ Total elec cost	71,280	31,164	49,797	29,628	9,876
M$ Total 20-yr elec cost	1.43	0.62	1.00	0.59	0.20
$ Maintenance cost per year	50,000	50,000	40,000	30,000	20,000
M$ 20-yr maintenance cost	1.00	1.00	0.80	0.60	0.40
M$ Installed cost premium	0.0	0.4	0.0	0.6	0.1
M$ Total 20-yr cost	**2.43**	**2.02**	**1.80**	**1.79**	**0.70**

Figure 5-13. Sample TES Cost Comparison

Notes:

1. Parameters will vary by locale. This is intended to show how the factors to compare.
2. No first cost incentives considered.
3. For partial storage, any number of fractions of storage are possible. 50-50 was used for this example
4. Cost of water considered equal for each option and not shown.
5. For full storage systems, a special thermal storage electric rate is assumed.

ECM DESCRIPTIONS—ELECTRICAL

Power Factor Correction

Basis of savings: Electric utility charges for power factor.

Note: *The additional current in cables from reactive power does create I^2R loss like any current. However, in all but extreme cases, the energy savings from reduced heating of cables is trivial and is ignored here.*

- Utility fees alone can often justify power factor correction projects.
- Example: For some utilities, the power factor charge is a 1-for-1 increase in demand charge. If the power factor is habitually 80 percent and the utility charges for anything lower than 95 percent, then the cost of poor power factor in this case would be (95-80) = 15 percent increased demand charges.
- Note: Simple capacitor installation may bring good results for facilities with mostly motor loads, welding, etc. But in facilities with high levels of harmonics should be studied very carefully and special and costly power factor correction equipment may be required.
- See **Chapter 12—Motors and Electrical Information "Power Factor Correction Capacity Quick Reference Chart."**

Load Balancing

Basis of savings: Motor performance and efficiency presumes equal voltage on each phase. If the voltage and current are different, then one phase will pull harder than the rest, and the motor windings fight, with ensuing energy loss and motor heating.

- A 2 percent voltage imbalance on a polyphase motor can reduce efficiency by 5 percent. For example, a motor with an 85 pct eff. nameplate could be 0.85 * 0.95 = 80.75 percent efficient.
- See **Chapter 12—Motors and Electrical Information "Voltage Imbalance Effect on Motor Efficiency"** for motor efficiency losses at other degrees of imbalance.

ECM DESCRIPTIONS—COMPRESSED AIR

Demand Side

- Use less to begin with
- Turn off compressors when plant is inactive
- Reduce leaks and monitor leakage as an ongoing management process

- Eliminate inappropriate uses of compressed air
- Add automatic shutoff to continuous air use sources, to use air only when needed
- Minimize desiccant dryer air losses
- Air amplifiers (eductors) for blow-off operations
- Reduce supply air pressure
- Use cooler inlet temperature

Distribution
- Correct restrictive piping
- Segregate areas of higher and lower pressure and serve separately
- Add load-side storage if large short term demands require elevated supply pressure

Supply Side
- Use compressor unloading options that reduce part load losses
- Sequence compressors for highest cfm per kW output
- Recover waste heat for a useful purpose
- Sub meter compressed air electric use and air flow

See **Chapter 14 – Compressed Air**

ECM DESCRIPTIONS – LAUNDRY

Longer spin cycles and/or higher spin speeds remove more water from wet clothes which means less heat is required to drive off the remaining moisture. The trade-off between added electrical energy and reduced heat energy strongly favors the saved heat. In some cases, there is more washing equipment installed than drying equipment and the additional time required for the spin (even an extra five minutes) impacts production, in which case the measure involves capital expense for additional wash equipment to eliminate the bottleneck.

Most commercial and industrial washers are programmable, for various 'recipes' to wash and sanitize different types of cloth, and the controls include duration of spin cycle. So, for most facilities, simply adjusting the cycle time implements the measure, with no cost.

The additional time does incur additional electric use, so it needs to be verified that there is actually a problem to be fixed (if already spinning enough, more spinning will add cost but not produce savings). The metric is remaining moisture content (RMC) and is found by weighing the clothes after extraction (before they go into the dryer), and then again when they come out of the dryer, using the same cart.

Roller drying/folding machines require some residual moisture to operate and, in fact, if the wet laundry is too dry it may require addition of water prior to using the machine. This places a practical limit on extraction where such machines are used, on order of 17% RMC.

Range of RMC for washer extractors and presses is 20-40% (Source 2)

There is an approximate 1:1 ratio between reducing RMC and reducing drying energy. As drying energy is the single largest energy using task for laundry, reducing RMC is a recommended strategic approach. For modern commercial machines, RMC of <20% is attainable.

Commercial Laundry Benchmarks (per pound)

<u>Water use</u>:

2.2-3.2 gal/lb normal horizontal axis (Source 1)

1.0 +/- gal/lb tunnel washer

<u>Energy use</u>:

2.0 kBtu/lb normal system water use (Source 1)

5.2 kBtu/lb, 25% RMC (1) combined with (Source 2)

Tunnel washer less due to reduced water heating

Sources:

(1) Commercial Laundry Facilities, Riesenberger, J., Koeller, J, 2005.

(2) Source: Boston Washer Study, 2001, Oak Ridge National Laboratory, for U.S. DOE

RMC = remaining moisture content

RMC = (wt before drying - wt after drying) / w after drying

Where wt=weight (lbs.)

Example:

<u>Wt (lbs) - now</u>	
334	after extraction (before drying)
201	after drying
133	water weight
66.2%	RMC=(334-201)/201

Rearranging:

wt after drying = wt before drying / (1+RMC)

Rearranging and substituting initial soiled weight for dry weight

wt before drying = initial soiled dry weight * (1+RMC)

Water weight (the water the dryer must deal with)

Water wt. removed=wt before drying- wt after drying
Substituting:
Water reduction from reducing RMC
=Initial soiled dry weight before * (RMC before – RMC after)

Example of using RMC to estimate heating savings

A facility processes 400,000 lbs of laundry per year (soiled, dry laundry). Sample weighing of laundry indicates 50% RMC using current spin cycle times. Dryers use natural gas and are 90% efficient. Natural gas cost is $1.00 per therm and electricity cost is $0.08 per kWh.

Proposal: reduce RMC from 50% to 30%
Initial soiled dry weight: 400,000 lbs
Existing:

Wt before drying =	400,000 * 1.50 = 600,000 lbs water
Wt after drying =	400,000 lbs
Water weight =	50,000 lbs
Proposed	
Wt before drying =	400,000 * 1.30 = 520,000 lbs water
Wt after drying =	400,000 lbs
Water weight =	30,000 lbs
Water reduction =	80,000 lbs water
Btu/lb water =	approx 1000 Btu/lb to heat and evaporate the residual water
Btu/year saved =	80,000lb * 1000 Btu/lb * 1/0.9eff * 1 therm/100,000 Btu
=	889 therms per year (ans)

Alternate calculation for water evaporation reduction

Water reduction =	wt of soiled dry laundry * (RMC before – RMC after)
Water reduction =	400,000 * (0.5-0.3) = 80,000 lbs water

From **figure 5-14** for RMC 0.5 before and 0.3 after, a value of 200 therms/100,000 lbs of soiled dry laundry is given. This is at 100% heating efficiency, so the value to use is 400,000 lbs * 200 therms/100,000 lb * 1/0.9 = 889 therms

Electrical input is needed for the additional drying time. Understanding the give and take between electrical cost and heat savings is important. Electrical energy per machine will vary.

It is unknown what portion of the electric usage in a washer cycle is devoted to spinning and how the figure varies according to a change in ratio between RMC before and RMC after.

Clearly, a small reduction in RMC will require less spin time and less spin electrical energy than a large reduction in RMC. See figure 5-14. This chart includes assumptions made to reasonably proportion the electrical burden according to a change in RMC. These values may be used in the absence of definitive values from equipment manufacturers, to answer the relevant question of approximately how much of the heating savings are negated by the extra electrical usage.

Continuing the prior example to include the offsetting electrical burden:

Additional electric energy for the longer spin cycle (**figure 5-14**)

For RMC 0.5 before and 0.3 after, a value of 353 kWh per 100,000 lbs of soiled dry laundry is given

Converting to actual laundry weight, 400,000 lbs * 353 kWh/1000 lb = 1412 kWh

Fuel reduction already calculated at or taken from the chart = 889 therms

Fuel savings: 889 therms * $1.00/therm = $889

Electric cost: 1412 kWh * $0.08/kWh = $113

Net savings: 889-113 = $776

Electricity cost erosion of heat savings: 113/889 = 13%

Had natural gas cost been $0.50/therm and electric cost $0.15/kWh

Savings from reduced heat would have been $445, and added electrical cost would have been $212. Electricity cost erosion of heat savings: 48%

Had the dryer been electrically heated, at $0.08/kWh

Heat saved (note: 100% eff) is 800 therms or 800*100,000 / 3413 = 23,440 kWh

Savings from reduced heat would have been $1875, and added electrical cost would have been $113. Electricity cost erosion of heat savings: 6%

Therms Saved per 100,000 lbs of Laundry

Assumes 1000 Btu/lb water removed,and 100% heat source efficiency

		RMC After					
		0.70	**0.60**	**0.50**	**0.40**	**0.30**	**0.20**
RMC	**0.80**	100	200	300	400	500	600
Before	**0.70**		100	200	300	400	500
	0.60			100	200	300	400
	0.50				100	200	300
	0.40					100	200
	0.30						100

Approximate kWh Added per 100,000 lbs of Laundry

Added electrical usage for spin work to lower RMC

		RMC After					
		0.70	**0.60**	**0.50**	**0.40**	**0.30**	**0.20**
RMC	**0.80**	230	275	337	431	587	900
Before	**0.70**		236	290	372	509	783
	0.60			243	314	431	666
	0.50				255	353	548
	0.40					275	431
	0.30						314

Assumptions used for this chart.

1. Do not extrapolate beyond the chart limits
2. Laundry weight is dry. Initial (soiled) and final (clean) dry weight presumed equal
3. One survey measured washer electrical usage per pound of dry soiled laundry in the range of 0.02-0.05 kWh per pound, (0.035 average). Source: Boston Washer Study, 2001, ORNL, for DOE. Values read from chart in the report
4. Assume 50% of washer electrical use goes to spinning
5. Assume the additional spin time for a large reduction in RMC increases that by 50%. Then, the maximum electrical burden that is exchanged for lowering RMC is 0.035*0.5*0.5 = 0.00875kWh per lb. +/- rounded to ~ 0.009 kWh/lb additional electric energy use for lowering the RMC
6. The reduction rate for electrical burden for extra spin work is assumed to be linear to the change in ratio between RMC before and RMC after
7. 0.009 kWh/lb is associated with an RMC reduction from 80% before to 20% after (4:1)
8. A minimum value of added electrical use for extra spinning is assumed to be 25% of the maximum electrical penalty (0.009 * 0.25 = 0.0023 kW/lb) and is associated with an RMC reduction from 80% before to 70% after (1.14:1)
9. Other table values are proportioned linearly according to the ratios of RMC before / RMC after

Figure 5-14. Heat Savings and Approximate Added Electricity Usage for Changes in Laundry Remaining Moisture Content (RMC)

Chapter 6

Utility Rate Components

Note: For heating fuel, only natural gas is discussed here.

ELECTRIC

Common utility rates are summarized here. Infrastructure fees, administrative fees, special customer rates, extremely large customer rates, cogeneration, net metering, wheeling, and de-regulated direct purchase rates and surcharges to fund conservation programs are not included.

Energy

Typical Units: $ per kWh or $ per MWh

Sliding Scale (Block Rates)

Price either increases or decreases as consumption increases, depending on the intended price signal.

- For volume pricing, as in for large industrial use, the price would decrease as usage increases. This is the usual case (declining blocks).
- For conservation, the price would increase as usage increases (inclining blocks).

Market Adjustment

Typical units: $ per kWh

This is used for uncontrolled costs the utility incurs, such as:

- Fuel cost adjustments for generation, and
- Costs for operating peak generating equipment
- Spot purchases due to peaks or unscheduled outings.
- Other unplanned expenses passed on

Time of Use

Pricing signal adjustment for leveling load for the utility.

- Applies to kW and kWh fees.
- On-peak pricing is applied during high demand periods.
- Off-peak pricing is applied in other times.

Demand Charges

Typical Units: $/kW per month or $/kW per day

- Measurement of power draw, not energy use.
- This charge reflects the infrastructure burden on the utility since the instantaneous demand must be met with appropriate generation, transmission, and distribution equipment.

Load Factor

Defined as "average/maximum electric demand."

- Usually not billed directly, but customers with low load factors will end up getting high demand charges.
- Maximum demand is metered and appears on the utility bill. Average demand is calculated:

Avg. Demand = Total kWh/total hours in the period

For example, if a billing period is 30 days long and records 100,000 kWh and 350 kW, find load factor:

Max demand = 350
Avg demand = 100,000/30/24 = 139 kW
Load factor = 139/350 = 40%

Business Type	Average Load Factor
Grocery	75-80%
Health Care	55-65%
Multi-family	50-65%
Retail	50-60%
Lodging	45-60%
Education	45-55%
Restaurant	50-55%
Office	45-55%
Manufacturing—general	15-40%
Manufacturing—semi conductor	85-95%

Figure 6-1. Load Factors by Business Sector

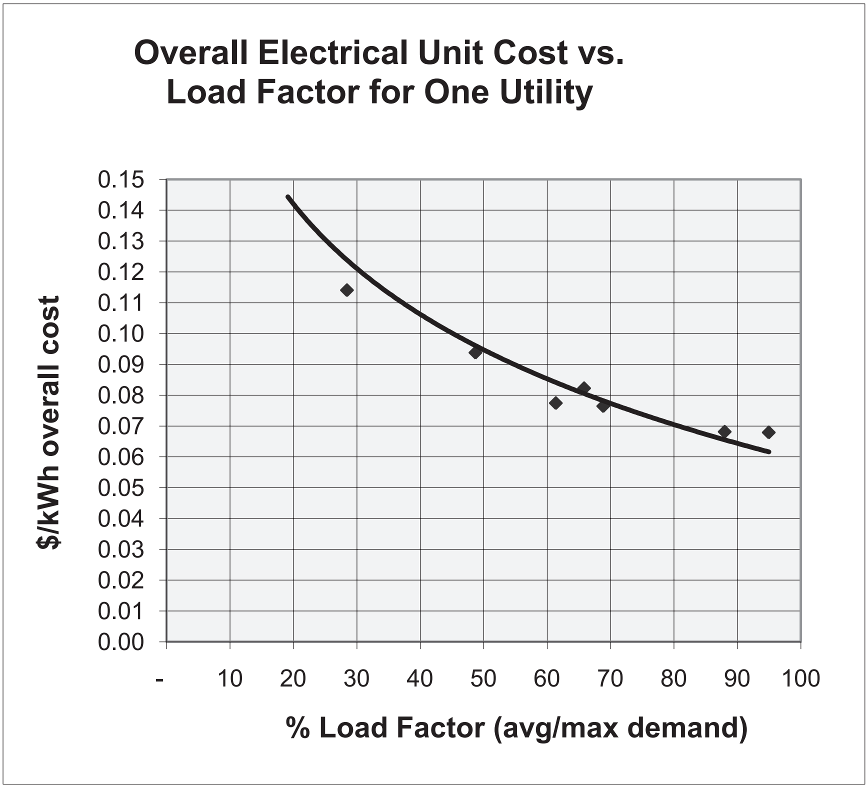


Figure 6-2. Load Factor Relationship to Electric Cost for One Utility

- When a customer uses a high rate of power for a short time, but much less power the rest of the time, the installed utility equipment is underutilized much of the time and represents a stranded cost without sufficient revenue to pay for it. The rate design recovers costs for low load factors to counter the underutilization effect on their revenue stream.
- Load factor is not billed for directly. Rates are designed to increase demand charges for low load factors to recover costs for under-utilized equipment.

Power Factor

The difference between "real" and "apparent" power, this factor addresses the magnetizing currents required by electromagnetic devices

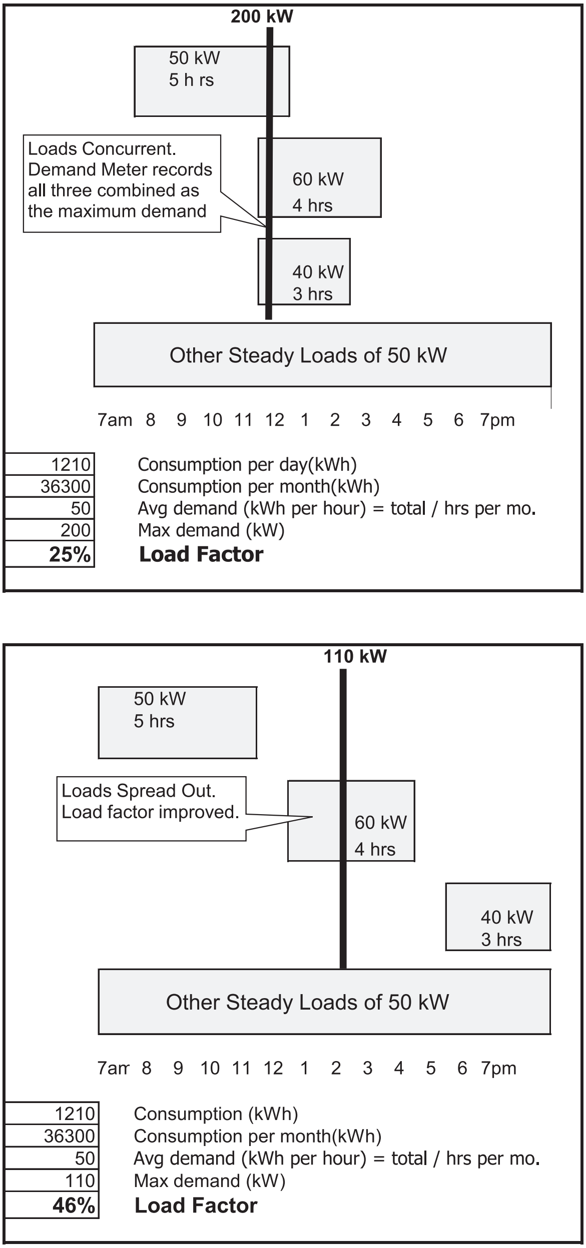


Figure 6-3. Example Re-Scheduling Loads to Improve Load Factor

such as motors, transformers, inverters, and welders, especially when oversized or idling.

- Utility meters record watts which is "real power" and the additional current from power factor less than 1.0 are not recorded on watt meters. Since the added currents are felt by the generators, transmission, and distribution equipment they represent real costs.
- The power factor fee is used to recover these costs. The power factor measurement is usually determined by a gross measurement of kW vs. kVA.

Ratchet Clause
(Minimum Demand)

Typical units are 65-75% of the highest demand during the prior 12 months.

- Utility rates are set based on assumed utilization of installed equipment. If the customers demand for power decreases suddenly, such as when a large tenant moves out, the installed power supply equipment becomes a stranded investment with insufficient revenue to pay for it.
- The ratchet clause is a common feature in utility rates to recover these costs. The ratchet serves to bill at a minimum demand even when the actual demand is lower.

Interruptible

Utilities that are resource-limited may utilize the interruptible rate as part of their resource supply mix. By turning off a customer's power, the system capacity is available for other customers without building new generator or distribution capacity.

- In exchange for the ability to interrupt at will, the customer receives lower rates.
- The utility will "call" an interruptible event, at which time the customer agrees to turn off load.
- If the customer fails to turn off load, significant penalties are applied.

GAS

Common utility rate types are summarized here. Infrastructure fees, administrative fees, special customer rates, extremely large customer rates, seasonal, transmission, and de-regulated direct purchase rates are not included.

Energy

Units: therms or dekatherms.
Therm = 10^5 Btu
Dekatherm = 10^6 Btu

In some areas, gas is billed in cubic feet, usually hundreds (ccf) or thousands (mcf). When billed on volume, pressure and temperature corrections apply.

Sliding Scale (Block Rates)

Price either increases or decreases as consumption increases, depending on the intended price signal.

- For volume pricing, as in for large industrial use, the price would decrease as usage increases. This is the usual case (declining blocks).
- For conservation, the price would increase as usage increases (inclining blocks).

Market Adjustment

Typical Units: $ per therm or $ per ccf.

- This is used for uncontrolled costs the utility incurs, such as changes in purchase price for gas.

Interruptible

Utilities that are resource-limited may utilize the interruptible rate as part of their resource supply mix. By turning off a customer's gas supply, the system capacity is available for other customers without building new distribution capacity.

- In exchange for the ability to interrupt at will, the customer receives lower rates.
- The utility will "call" an interruptible event, at which time the customer agrees to turn off load.
- If the customer fails to turn off load, significant penalties are applied.

Table 6-1. Common Utility Bill Items to Extract (Part 1).

Common bill elements and what they can imply. Not a complete list.

Interruptible, super peak, net metering, ratchet charges, and district heating bill items not included.

Fuel utility units shown for natural gas apply similarly to other fuels.

Metric	How to Calculate	How Used
Overall $/kWh (blended rate)	Total electric cost / Total kWh used	• Measures defined in terms of kWh savings are converted to $ savings with this factor. When high compared to other facilities in a given reason, there is likely a low load factor. • The use of 'blended rates' for equating kWh reduction to dollar savings is valid if the ECM affects demand and energy use in on/off peak times in similar proportion to average use. Blended rates can introduce errors when the ECM focuses mostly on one area, such as only saving in off peak times, only saving demand charges, etc.
Overall $/therm for gas	Total gas cost / Total therms used	Measures defined in terms of therm savings are converted to $ savings with this factor.
Block rate cost differentials		• Identifies thresholds above or below which a step change in unit cost occurs. • Declining block rates can provide incentive for aggregating meters. • For example, there can be additional incentive for an ECM that keeps usage safely away from a high priced tier.
Overall $/1000 gallons for water and waste	Total water+waste cost / total water gallons used (thousands)	• Measures defined in terms of water savings are converted to $ savings with this factor. • These costs also subtract from measures that use water to save energy such as cooling towers and evaporative cooling.
Differential between on-off peak kWh charges	$/kWh on peak - $/kWh off peak	• For measures that produce savings predominantly in off-peak hours, 'blended rates' will overstate dollar savings. • For time of use rates, is possible that savings only in off peak times will not reduce demand charges and dollar savings for these measures will be at a fraction of the overall blended rate ($/kWh).
Differential between on-off peak demand charges	$/kW on peak - $/kW off peak	• The higher the differential, the greater the incentive to shift loads to off peak. • This is a key parameter for economic viability of Thermal Energy Storage (TES) systems.

Table 6-1. Common Utility Bill Items to Extract (Part 2).

Metric	How to Calculate	How Used
Overall $/therm for electric heating, compared to gas	• $/therm electric heating = $/kWh * (100,000/3413) • $/therm for gas, from above, divide by firing efficiency (e.g. 0.80)	• Provides relative benefit of fuel switching options. • For example, if gas heating costs half per delivered therm compared to electric heating, then replacing an electric boiler with a gas-fired boiler would reduce fuel cost by half for the same heating duty.
Fraction of electric bill that is demand	Demand charges / total electric charges	• Establishes relative importance of demand charges. • For example, if demand charges are two thirds of the bill, demand will get more focus than if it is 25% of the bill.
Magnitude of power factor charges	Usually denoted on the bill.	• Establishes a budget for power factor correction measures. • For example, with a 3 year payback hurdle, power factor charges of $10,000 per year mean that up to $30,000 in corrective measures would constitute a viable alternative for the customer.
Energy Use Index (EUI) in kBtu/SF-yr	• Convert electric energy use to kBtu with kWh * 3.413 • Convert gas energy use to kBtu with Therms/100 • Other fuels, to kBtu • Total energy kBtu divided by total building SF	• EUI values are benchmarked for common building uses. • Similar strategy for manufacturing, where the EUI is in terms of Btu/lb of milk, Btu per ton of concrete, Btu/gallon of beer, etc. • Where benchmarks are available, this simple comparison of the customer to their peers establishes whether existing energy use, in general, is high, low, or average and suggests reasonable targets for improvement.
Energy Cost Index (ECI) in $/SF-yr	Total energy cost divided by total building SF	• In manufacturing, this can also be unitized by production units, as an 'ingredient', $/unit.

Chapter 7

Automatic Control Strategies

GENERAL

Implementing, expanding, or modernizing an automatic control system, especially when migrating from pneumatic controls, provides a wide array of opportunities for optimization and energy savings.

The cost of a new control system is high. Cost justification for these are usually based on avoided cost of the existing controls if in poor repair, or energy savings opportunities, although often the need for control *at all* is not factored in. This is like trying to find economic justification for having a steering wheel in a car—obviously you need one anyway. The better approach to economic justification is to evaluate the *upgrade* incremental cost between a basic "temperature control only" system and the more high powered optional system or features.

- For a building with an aging pneumatic control system, the baseline could be a new pneumatic control system, actuators and all, compared to DDC, then examining the differential cost and challenging the new technology to justify the upgrade.

Additional benefits can be realized from improved indoor comfort and reduced response time for service calls; however these are often hard to quantify. The greatest savings from optimization will occur when the customer is willing to embrace new and complex control strategies, however this must be tempered with skill level and training level of operational staff.

The cost of installing a modern control system is considerable, but improvements made with modifications and adjustments to a control system in place can offer among the best returns on investment. It is safe to say that *once in place*, a modern digital control system can do a lot for energy savings.

COST/BENEFIT RATIO FOR CONTROL SYSTEM EMCS

See **Figure 7-1, Automatic Control System Cost/Benefit Ratios.**

CONTROL SYSTEM APPLICATION NOTES

Enablers and Disablers

Success of all control measures depends on cooperation by that which is being controlled, and control optimization is limited to the higher priorities of maintaining the process or comfort. Measures applied to shared systems find limits based on a 'weak link'. Shared system examples include:

- A chilled water system that serves multiple zones
- A heating water system that serves multiple zones
- A fan or duct that serves multiple air flow points of use

The goal of saving energy without losing comfort or process will be improved by incorporating zone level feedback to assure each zone gets what it needs. Additionally, if comfort issues surface immediately upon trying to implement an optimizing routine or even a reset schedule, it suggests a weak link exists that may be singularly limiting valuable savings from the rest of the 'chain'. When this happens, the easy answer is to give up the routine, but it may also be possible to identify and remedy a weak link. This involves some supplemental investigating and measuring, and can end with mechanical work such as cleaning coils, new coils, or air balance. For example, if there are 20 coils and 19 perform fine at 10F deltaT in July, but one loses control above 4F, remedial work to the one becomes an enabler for all 20. This logic also applies to the "one VAV box that can't get enough air," the "one heating coil that is too small and needs extra hot water," etc. If the entire building is designed and built with such a constraint it may be the fate of the building and its energy use, but if the limit is an anomaly it will be worth the effort to intervene at the system level to enable the controls to achieve deeper savings.

Some examples of disablers (and correcting the weak spot becomes the enabler):

- One coil undersized for the cooling load drives the system to low temperature
- One zone requiring very cold air drives the system to low temperature

- One coil undersized for the heating load drives the system to high temperature
- One zone served by a restrictive duct or pipe drives the system to high pressure
- Pipe fittings that leak upon any thermal movement drives the system to constant temperature
- One zone requiring constant HVAC service drives the system to constant operation

In applying controls, system knowledge is very important and results can be limited without it.

Some examples of things that enable for best results with control optimization are shown and also indicate how valuable design influence can be for energy efficiency. Controls can make a good design sing but inherit fundamental limits of the design.

- Communication from source to end use for the same system. Example is air handler and zone level control. This allows control as a system and enables optimization by supplying what is actually needed.
- Buildings sectioned into areas or floors enable logical energy reduction when small area uses partial vacancies exist.
- Equipment with efficient part load provisions enables energy use turndown to approximate ideal. Designs that use variable speed motors and multiple increments of capacity instead of one large one are enablers to optimal sequencing and energy use that turns down with load.
- Free-flowing ducts and pipes without constricted sections are enablers for optimizing pressure at part load.
- Terminal units that require minimal pressure drop to function enable low residual pressures for main fans and pumps
- Piping and building envelopes that do not have restrictions on temperature swings enable deep resets and setbacks.
- Areas that have unique environmental and operational schedules served by independent systems enable optimizing each without compromise.
- Granular zoning of lighting areas and HVAC control areas are enablers for schedule and occupancy controls to provide services where needed, but only where needed.

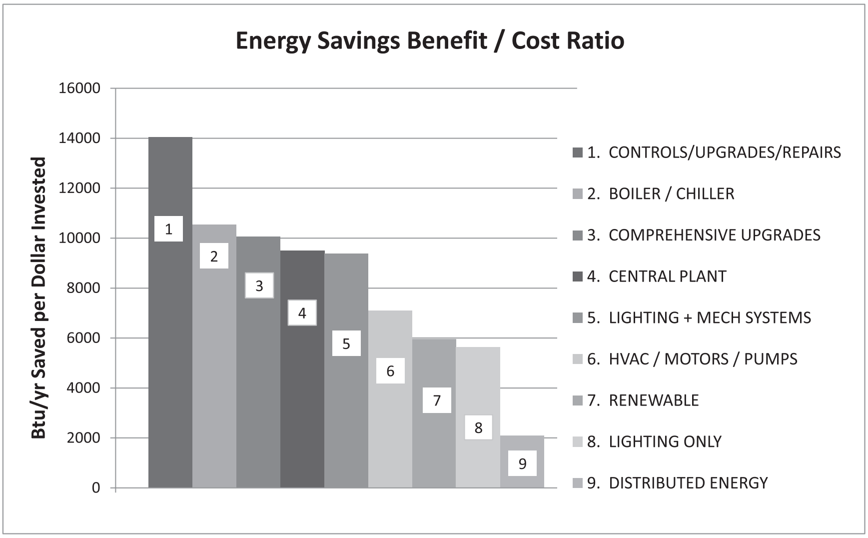


Figure 7-1. Automatic Control System Cost/Benefit Ratios

<u>Graph of Linear Data Values</u>
Note: Standard Deviation Ranges are not shown. This is for general comparison only.
Source: "Analysis of Energy Savings in the Federal Sector through Utilities Service Programs," *Strategic Planning for Energy and the Environment*, Vol. 25, No. 4, 2006.

ECM	Btu/yr per dollar	Std. dev
1. CONTROLS/UPGRADES/REPAIRS	14051	761
2. BOILER / CHILLER	10539	1091
3. COMPREHENSIVE UPGRADES	10072	2787
4. CENTRAL PLANT	9502	1403
5. LIGHTING + MECH SYSTEMS	9383	511
6. HVAC / MOTORS / PUMPS	7097	948
7. RENEWABLE	5964	1241
8. LIGHTING ONLY	5643	270
9. DISTRIBUTED ENERGY	2103	766

Figure 7-2. Automatic Control System Cost/Benefit Ratios

Control System Application Notes

- A caution in the application of any control system is to assure that the systems under control are "controllable" and that controls are not used in an attempt to solve mechanical design, installation or misapplication issues; always fix those issues concurrently with the new control system.
- Almost any control loop with a fixed set point is an opportunity for energy savings.

LIGHTING CONTROL STRATEGIES—BASIC

Scheduled Start-Stop

Basis of Savings: Reduced run time.

- Timed coincident with occupant schedules.
- Two features are important for acceptance of these systems:
  - — A warning that the lights are about to go out
  - — Local overrides that allow people to keep the lights on
- Combining with zone level overrides, control schedules can be compressed to reduce hours of operation, including hours use by cleaning crews.

Photo Cell Control of Outdoor Lights

Basis of Savings: Reduced run time.

- Outdoor lights off during day-lit hours:
  ON-Dusk
- OFF-Dawn
- Time switch to shut off some site lighting after dark, but security lighting stays on.
- Improved savings are possible by combining a photocell with a time switch, set to a late hour, e.g. 1am:
  ON-Dusk
  OFF-1am

Occupancy Sensor Lighting Control

Basis of Savings: Reduced run time.

- Use for individual offices, meeting rooms, classrooms, multi-purpose rooms, warehouses, etc., that are only occasionally occupied. For warehouses, it is good practice to leave a few lights on continuously for safety and security. Not practical for restrooms, since they can leave people in the dark, unless at least one light is left on.

Timed Tenant Override

Basis of Savings: Reduced run time.

- When after-hours tenants invoke the override, limit the amount of time this will remain active before the system reverts to unoccupied mode again (e.g. 2 hours), and what actually gets overridden.

HVAC CONTROL STRATEGIES—BASIC

Scheduled Start-Stop

Basis of Savings: Reduced run time.

- Any equipment left to run continuously should be controlled for automatic start-stop based on occupancy schedules. This can be energy management system (EMS) control or a simple time switch. Outside air dampers are usually closed and exhaust fans off during unoccupied times.
- Settings should closely match actual use. Occupancy settings for worst case may be easiest to implement, but represent lost savings opportunities. For example, if a classroom is normally vacated at 5pm but occasionally occupied until 10pm, setting the occupied controls to always leave it in "occupied" mode until 10pm is wasteful.

Zone level 2-hour overrides are a partner to scheduling and allow schedule compression.

Standardize Indoor Comfort Settings

Basis of Savings:

1. Reduced envelope losses by reducing the temperature difference between indoors and outdoors.
2. Standardized temperatures prevent simultaneous heating and cooling from adjacent zones operating at different temperatures.
3. A partner measure is limiting the local adjustment. For example, if standardized values are 74F cooling and 70F heating, combined with a +/– 2F limit on local adjustment, adjacent zones will not

fight: a zone in heating will be worst case 70 + 2 = 72F, and a neighboring zone in cooling will be worst case 74 – 2 = 72F (no overlap).

For **Figure 7-4**, initial/typical settings are noted but will vary. The ideal setting is where comfort or process requirements are met when just enough air/water flow is provided, just enough pressure is provided, just enough cooling is provided, just enough heating is provided, and just enough ventilation is provided—without incurring losses from throttling, false loading, heat/cool overlap, pressure/dissipation overlap, or other inefficiencies, Energy efficiency is improved to the extent that actual delivered values approach ideal.

Setting	Description	Basis of Savings
74 degF **(Note 1)**	Space Cooling – Occupied	Compared to lower temperatures: Reduced temperature difference (dT) between inside and outside, lowering envelope transmission. Reduced cooling system lift by raising the evaporator temperature. 1-1.5% per degree efficiency gain in the compressor. **(Note 4)**
85 degF	Space Cooling – Unoccupied	Reduced dT between inside and outside, lowering envelope transmission during unoccupied times. Can be raised further to assure mechanical cooling stays off entirely, subject to equipment or furnishings that may suffer from elevated temperature and longer pull down time.
70 degF	Space Heating – Occupied	Compared to higher temperatures: Reduced temperature difference (dT) between inside and outside, lowering envelope transmission.
60 degF	Space Heating – Unoccupied	Reduced dT between inside and outside, lowering envelope transmission during unoccupied times. Can be lowered further, subject to freeze concerns and longer pickup time.
4-5 degF	Minimum dead band between heat and cool	Reduces likeliness of overlapping heating and cooling. Adjacent heating and cooling operations sequenced on a common temperature are very likely to overlap.
1.0 in. w.c.	Duct Static Pressure	Compared to higher pressures: Reduces fan energy. A 10% reduction in duct static reduces fan power by 10% for VAV systems and 15% for constant volume systems. Can be reset seasonally or continuously optimized. See **Chapter 15: *Savings From Lowering Downstream Maintained Pressure Setting.***

Figure 7-4. Energy-Saving HVAC Control Settings

Continued

Setting	**Description**	**Basis of Savings**
45 degF (dehumidifying) 50 degF (sensible cooling)	Chilled Water Temperature	Compared to lower temperatures: Reduced cooling system lift by raising the evaporator temperature. 1-1.5% per degree efficiency gain in the compressor **(Note 5)**. Note: increasing chilled water temperature above this point will reduce dehumidification potential of air coils, so this should be used with caution in areas where humidity removal is important.
(note 2)	Condenser Water Temperature	Compared to higher temperatures: Reduced cooling system lift by lowering the condenser temperature. 1-1.5% per degree efficiency gain in the compressor. Note: there is a balance between compressor savings and added cooling tower horsepower added, as well as the sizing of the cooling tower in "approach" capability. See Chapter 7 - Automatic Control Strategies, HVAC Controls Strategies (advanced) **"Optimum Condenser Water Set Point"**
(note 3)	VAV box cooling minimum air flow	Compared to higher settings: Cooling mode minimum flow: Reduced heating and cooling overlap in areas where demand for air flow is less than allowed by the minimum stop. These areas require supplemental heating or result in over-cooling when internal loads do not match the minimum air flows. Energy use from space heaters in over-cooled areas is part of the HVAC energy use. Note: ventilation air in a standard VAV design is mixed in with the supply air, which is the reason for the minimum setting. Reducing this should be done with caution to assure proper ventilation is provided for occupants. The virtue of the Dedicated Outside Air System (DOAS) design is allowing VAV minimums for primary air to be zero. Heating mode minimum flow: Reduced reheat penalty. During heating mode, the supply air must be heated to room temperature before any heat added results in heating of the room. The amount of reheat required depends on the temperature of the supply air and the amount of it, and VAV supply air reset in heating mode curbs the reheat burden. Raising minimum air flows in heating increases reheat burden.

Figure 7-4. Energy-Saving HVAC Control Settings (*Continued*)

Notes for Figure 7-4

1. Cooling comfort settings will vary by climate zone. Drier climates will generally allow higher summer temperatures. Equally important to the settings are the separating "dead band" between heating and cooling. This provides a good measure of prevention against simultaneous heating and cooling.
2. Cooling tower performance is almost entirely driven by wet bulb temperature. Since "design" wet bulb temperatures vary by locale, the available condenser water temperature varies as well. A rule of thumb for efficient cooling towers is wet bulb +7 degF as the condenser water supply temperature, but this varies based on cooling tower specifications (a range 7-15F approach is common), and also varies according to load (approach improves at reduced load).
3. Cooling VAV zone air flow should be as low as possible while still meeting general ventilation requirements, and the heating VAV zone air flow should not exceed the cooling air flow. The lower the minimum setting, the lower the over-cooling potential in summer and the lower the reheat penalty in winter.
4. For savings to occur in the refrigeration cycle, the rise in space temperature must result in a temperature/pressure rise at the refrigeration machine input. Examples are single zone DX where the air temperature rise is felt directly by the machine, or when a there is a corresponding increase in chilled water temperature. Note that raising space or supply air temperature by throttling water flow and without a change in water temperature does not result in refrigeration cycle savings.
5. Raising supply water temperature, combined with a fixed return water temperature, lowers the differential and requires higher flows for a given load. This does not affect constant flow water systems (flows are almost always in excess of requirements and the surplus water is shunted to the return by a 3-way valve), but does affect variable flow water systems. In the case of chilled water systems there is usually a net gain by raising the water temperature despite the additional flow. In the case of air cooling systems, the fan power increase in cooling mode offsets the cooling savings. This give-take arrangement, and the comparison between water and air as the medium, is shown in a calculated example in **Chapter 9-Quantifying Savings**.

Restrict Tenant Adjustment Limits

Basis of Savings:

- Prevent simultaneous heating and cooling from adjacent zones operating at different temperatures.
- Where "zone of greatest demand" optimization routines are used, imposing this limit prevents a rogue zone setting from driving the routine.
- Remote lock-out of tenant adjustment for space temperature, or limiting the adjustment to +/- 2 degrees F at most.
- People, as thermostats, can detect a 2 degF change, but not much less. So, a plus/minus 2 degF setting will give a sense of control, alleviating a common complaint of DDC systems where no local adjustment is provided at all.

Match Equipment Capacity to Changing Loads

Basis of Savings: Provide "just enough" cooling or heating or air flow or water flow.

- Resetting primary cooling temperatures upward reduces power consumption by 1-1.5% per degree for refrigeration equipment.
- Resetting primary heating temperatures downward, if at the boiler, reduces flue temperature and fuel use by 1% per 40 degrees.
- Resetting primary heating temperatures, if only the distribution system from a mixing valve, reduces standby losses from distribution piping by reducing the differential temperature. Whatever the circulation thermal losses are, they will be reduced proportionally.
- Reducing circulating mass flow rates for air and water reduces circulating fan and pump horsepower proportionally as the reduction of flow (half the flow = half the power).

Vary fluid temperatures, pressures, supply air flows, water flows, outdoor air intake rates, based on demand and not worst case. The objective is always to provide enough, but just enough, of the item.

Optimum Start

Basis of Savings: Reduced equipment run time.

- Use to delay HVAC system start-up as long as possible. During the warm-up mode, the building is normally unoccupied so the outside air damper can remain closed and exhaust fans off until occupied.

Occupied/Unoccupied Mode (Set Up/Set Back)

Basis of Savings: Reduced envelope losses by reducing the temperature difference between indoors and outdoors.

- Use to set up and set back space temperatures, and to reduce or stop outside air intake during unoccupied times. Typical unoccupied temperatures are 55 deg F heating and 85 deg F cooling.
- The need for rapid return to temperature (hotel) or sensitive equipment (computers) may temper the suggested settings.

This model identifies the sequential operational modes. Heating is used for the example; cooling is similar. **See Figure 7-5.**

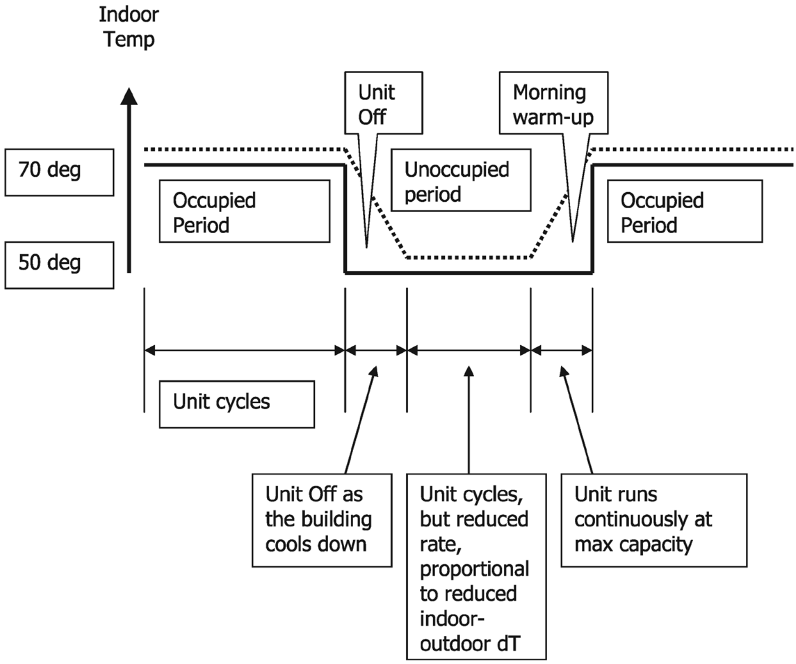


Figure 7-5. Setback Temperature Savings (Heating)

Notes for Figure 7-5

A. Unoccupied mode begins: unit off, temperature of indoors drifts downward as heat is lost, but no heat is added until the building temperature drops sufficiently to reach the "unoccupied" lowest allowed setting.

B. At unoccupied temperature: At this low temperature, the heating will be allowed to cycle off and on. The hours spent at this low temperature represent the savings, since the reduced inside-to-outside differential temperature (dT) reduces heat transfer and load.

C. Nearing the new occupied period: The heating system must be started in advance of the onset of the new occupied period to allow time for the building to warm up. The heating unit will be on full capacity at this time and may do so for several hours.

Energy use of the ECM is found by piecing together the sequences A,B,C. Variables that may be hard to estimate are:

A. The time, upon entering unoccupied mode, the unit will be completely off.
B. The duration of on and off cycles during the unoccupied period.
C. The time, in advance of the new occupied period, the unit will need to start again—to reach the "occupied" temperature.

Each of these are related and depend on the outdoor temperature: the colder outdoors the quicker the building will drop in temperature to the setback temperature and begin cycling and the sooner it will have to re-start to warm up the building. So the benefit of this measure is dynamic and will vary by building envelope type and climate zone.

Anecdotal estimates of savings vary and actual test data could not be found. A common rule of thumb for setback heating thermostats is:

- One (1) percent for each degree of setback that is kept there for at least 8 hours.
- Two (2) percent for each degree of permanent change.

A spreadsheet with bin weather for different cities and some assumptions was used to validate the rule of thumb for setback thermostats in heating mode. See **Figure 7-6.**

APPROXIMATE HEATING SET BACK SAVINGS IN VARIOUS CITIES

Bin Weather Data, M-F, 6am-6pm
Saturday 10a-4p, Sunday off all day
Savings are percent of envelope heating loss, compared to all hours at occupied temperature

Assumptions:
- 70F occupied, 50F unoccupied, 10 hours occupied, 14 hours unoccupied
- When OA = indoor temp, heater output is zero.
- When OA = design temp, heater output is 100%
- During warm-up period, heater is at 100% capacity.
- Actual occupied period is the nominal time less the initial off-time, less the warm-up time
- Heater assumed to be completely off at the onset of unoccupied period until the building cools off

								10 degrees F Setback			**20** degrees F Setback		
	Heating design temp F	hours btwn 45-55F OA	hours btwn 35-45F OA	hours btwn 25-35F OA	hours btwn 15-25F OA	hours btwn 5-15F OA	hours btwn [-5] - 5F OA	heating savings unoccd hours	heating savings overall hours	pct per degF (overall hours)	heating savings unoccd hours	heating savings overall hours	pct per degF (overall hours)
Colorado Springs, CO	7.1	859	890	732	409	134	61	36%	21%	**2.1%**	55%	32%	**1.6%**
Chicago, Ill	5.4	702	805	814	378	118	51	35%	21%	**2.1%**	54%	31%	**1.6%**
Oklahoma City, OK	18.2	765	650	402	131	71	13	43%	25%	**2.5%**	62%	36%	**1.8%**
Tampa, FL	42.9	393	121	16	0	0	0	70%	41%	**4.1%**	85%	50%	**2.5%**
Portland, OR	27	1572	1115	183	43	0	0	54%	32%	**3.2%**	72%	42%	**2.1%**
Phoenix, AZ	41.6	678	242	18	0	0	0	68%	40%	**4.0%**	84%	49%	**2.4%**
Anchorage, AK	-4.8	1044	853	937	748	390	152	30%	18%	**1.8%**	49%	28%	**1.4%**

Figure 7-6. Set Back Thermostat Savings (Heating)

Savings are approximated for a 20 deg F setback. Smaller setbacks would reduce savings proportionally

Savings for permanently lowered heating settings are higher. See **Chapter 7—*Percent Heating Change Per Degree of Indoor Temperature Change***

Comfort Heating Savings from Changing Indoor Temperatures

Limits for use of tables in **Figure 7-7**:

1. Not applicable to cooling season due to the strong variability in heat gain from solar load.
2. Applies where envelope and ventilation (combined) loads are a high percentage of heating energy use.
3. If internal loads are high and highly variable, then the error in using this approach will increase and should not be used.
4. If thermal stratification is high, e.g. tall areas, this method will understate savings and evaluation by layer is recommended instead. See **Chapter 5, De-Stratification**.
5. Does not apply to "self heating" buildings with very high internal gains, e.g. buildings whose heating loads are mostly weather-independent.
6. Does not apply to central portions of large buildings where envelope loads cannot be felt.
7. Increase/decrease values are for permanent temperature setting changes; set-up/set-back has additional dynamics discussed separately.
8. For the method shown, the value "T0" is the reference value of outdoor air temperature associated with the heating load at indoor temperature "T1" or the alternate indoor temperature "T2." For evaluating heating energy percent change at a given outdoor temperature, choose the value of "T0" directly. For percent change in annual heating energy, choose the value of "T0" equal to the seasonal average value of outdoor air temperature when the unit is running. For example, if a building has an "occupied" or "on" schedule from 8am-5pm, "T0" will be the average outside air temperature for the heating season for those hours. For percent change in maximum heating load, "T0" will be the coldest temperature anticipated during the operating hours.

In buildings where there are no internal heat gains, all heating load is from envelope and ventilation, and can be easily estimated from differential temperatures. Obviously few buildings fit this description. A more flexible approach can be used for many buildings as long as the internal gains follow a steady pattern of use throughout the year. For each building there is a normal indoor temperature, a balance point (below which heating is needed), and a maximum heating load that occurs at the coldest day of the season. All these work together to form the heat load 'signature' of that building. For steady internal loads, the heating energy used is proportional between zero (when internal loads equal envelope losses) and maximum (on the coldest day). With the heat turned completely off, the building temperature will stabilize at some value which is the balance temperature, but this can be ignored as long as the internal loads are consistent. Whatever the balance temperature is, it is, and raising or lowering the indoor temperature will alter the differential temperature and heating energy use proportionally. The "percent change" argument can be used for ECMs such as:

- Lowering winter heating set point, through standardized control settings.
- Radiant heating, when it can be shown that comparable comfort is achieved with reduced indoor air temperature.
- De-stratification measures, where it can be shown that the average indoor temperature is reduced.

The following terms are shared:

T1 = Original indoor temperature

T2 = New indoor temperature

T0 = Reference outdoor temperature relating indoor temperature T1 and base load Q1

Q1 = Heat load baseline for envelope and ventilation at original indoor temperature T1

Q2 = Heat load change for envelope and ventilation at new indoor temperature T1

Q1=f(T1-T0) and Q2=f(T2-T0).

"f" is the building envelope heat loss signature, the same for both cases, so is dropped.

Percent change is:

(Q1-Q2)/Q1, and substituting:

[(T1-T0)-(T2-T0)]/(T1-T0), and, factoring

Percent change is $1-\left[\frac{(\mathrm{T2}-\mathrm{T0})}{(\mathrm{T1}-\mathrm{T0})}\right]$

Note: Negative values indicate an increase in heating energy use.

Example:

Existing building is maintained at 70 degF and has a heating season average outside temperature of 10 degF. Changing to 69 degF (one degree F lower) has what effect on heating use?

T1=70	
T2=69	% change = (1- [(69-10)/(70-10)])
T0=10	=1.4% reduction

$$1 - \left[\frac{(T2 - T0)}{(T1 - T0)}\right]$$

T0: **40**

Approximate Heating Savings for Space Temperature Change

40 degrees F Base Winter Temperature

Units are percent change from energy use at T1. Negative values are energy increase.

T1 \ T2	75	74	73	72	71	70	69	68	67	66	65	64	63	62	61	60
75	**0.0**	2.9	5.7	8.6	11.4	14.3	17.1	20.0	22.9	25.7	28.6	31.4	34.3	37.1	40.0	42.9
74	(2.9)	**0.0**	2.9	5.9	8.8	11.8	14.7	17.6	20.6	23.5	26.5	29.4	32.4	35.3	38.2	41.2
73	(6.1)	(3.0)	**0.0**	3.0	6.1	9.1	12.1	15.2	18.2	21.2	24.2	27.3	30.3	33.3	36.4	39.4
72	(9.4)	(6.3)	(3.1)	**0.0**	3.1	6.3	9.4	12.5	15.6	18.8	21.9	25.0	28.1	31.3	34.4	37.5
71	(12.9)	(9.7)	(6.5)	(3.2)	**0.0**	3.2	6.5	9.7	12.9	16.1	19.4	22.6	25.8	29.0	32.3	35.5
70	(16.7)	(13.3)	(10.0)	(6.7)	(3.3)	**0.0**	3.3	6.7	10.0	13.3	16.7	20.0	23.3	26.7	30.0	33.3
69	(20.7)	(17.2)	(13.8)	(10.3)	(6.9)	(3.4)	**0.0**	3.4	6.9	10.3	13.8	17.2	20.7	24.1	27.6	31.0
68	(25.0)	(21.4)	(17.9)	(14.3)	(10.7)	(7.1)	(3.6)	**0.0**	3.6	7.1	10.7	14.3	17.9	21.4	25.0	28.6
67	(29.6)	(25.9)	(22.2)	(18.5)	(14.8)	(11.1)	(7.4)	(3.7)	**0.0**	3.7	7.4	11.1	14.8	18.5	22.2	25.9
66	(34.6)	(30.8)	(26.9)	(23.1)	(19.2)	(15.4)	(11.5)	(7.7)	(3.8)	**0.0**	3.8	7.7	11.5	15.4	19.2	23.1
65	(40.0)	(36.0)	(32.0)	(28.0)	(24.0)	(20.0)	(16.0)	(12.0)	(8.0)	(4.0)	**0.0**	4.0	8.0	12.0	16.0	20.0
64	(45.8)	(41.7)	(37.5)	(33.3)	(29.2)	(25.0)	(20.8)	(16.7)	(12.5)	(8.3)	(4.2)	**0.0**	4.2	8.3	12.5	16.7
63	(52.2)	(47.8)	(43.5)	(39.1)	(34.8)	(30.4)	(26.1)	(21.7)	(17.4)	(13.0)	(8.7)	(4.3)	**0.0**	4.3	8.7	13.0
62	(59.1)	(54.5)	(50.0)	(45.5)	(40.9)	(36.4)	(31.8)	(27.3)	(22.7)	(18.2)	(13.6)	(9.1)	(4.5)	**0.0**	4.5	9.1
61	(66.7)	(61.9)	(57.1)	(52.4)	(47.6)	(42.9)	(38.1)	(33.3)	(28.6)	(23.8)	(19.0)	(14.3)	(9.5)	(4.8)	**0.0**	4.8
60	(75.0)	(70.0)	(65.0)	(60.0)	(55.0)	(50.0)	(45.0)	(40.0)	(35.0)	(30.0)	(25.0)	(20.0)	(15.0)	(10.0)	(5.0)	**0.0**

$$1 - \left[\frac{(T2 - T0)}{(T1 - T0)}\right]$$

T0: **30**

Approximate Heating Savings for Space Temperature Change

30 degrees F Base Winter Temperature

Units are percent change from energy use at T1. Negative values are energy increase.

T1 \ T2	75	74	73	72	71	70	69	68	67	66	65	64	63	62	61	60
75	**0.0**	2.2	4.4	6.7	8.9	11.1	13.3	15.6	17.8	20.0	22.2	24.4	26.7	28.9	31.1	33.3
74	(2.3)	**0.0**	2.3	4.5	6.8	9.1	11.4	13.6	15.9	18.2	20.5	22.7	25.0	27.3	29.5	31.8
73	(4.7)	(2.3)	**0.0**	2.3	4.7	7.0	9.3	11.6	14.0	16.3	18.6	20.9	23.3	25.6	27.9	30.2
72	(7.1)	(4.8)	(2.4)	**0.0**	2.4	4.8	7.1	9.5	11.9	14.3	16.7	19.0	21.4	23.8	26.2	28.6
71	(9.8)	(7.3)	(4.9)	(2.4)	**0.0**	2.4	4.9	7.3	9.8	12.2	14.6	17.1	19.5	22.0	24.4	26.8
70	(12.5)	(10.0)	(7.5)	(5.0)	(2.5)	**0.0**	2.5	5.0	7.5	10.0	12.5	15.0	17.5	20.0	22.5	25.0
69	(15.4)	(12.8)	(10.3)	(7.7)	(5.1)	(2.6)	**0.0**	2.6	5.1	7.7	10.3	12.8	15.4	17.9	20.5	23.1
68	(18.4)	(15.8)	(13.2)	(10.5)	(7.9)	(5.3)	(2.6)	**0.0**	2.6	5.3	7.9	10.5	13.2	15.8	18.4	21.1
67	(21.6)	(18.9)	(16.2)	(13.5)	(10.8)	(8.1)	(5.4)	(2.7)	**0.0**	2.7	5.4	8.1	10.8	13.5	16.2	18.9
66	(25.0)	(22.2)	(19.4)	(16.7)	(13.9)	(11.1)	(8.3)	(5.6)	(2.8)	**0.0**	2.8	5.6	8.3	11.1	13.9	16.7
65	(28.6)	(25.7)	(22.9)	(20.0)	(17.1)	(14.3)	(11.4)	(8.6)	(5.7)	(2.9)	**0.0**	2.9	5.7	8.6	11.4	14.3
64	(32.4)	(29.4)	(26.5)	(23.5)	(20.6)	(17.6)	(14.7)	(11.8)	(8.8)	(5.9)	(2.9)	**0.0**	2.9	5.9	8.8	11.8
63	(36.4)	(33.3)	(30.3)	(27.3)	(24.2)	(21.2)	(18.2)	(15.2)	(12.1)	(9.1)	(6.1)	(3.0)	**0.0**	3.0	6.1	9.1
62	(40.6)	(37.5)	(34.4)	(31.3)	(28.1)	(25.0)	(21.9)	(18.8)	(15.6)	(12.5)	(9.4)	(6.3)	(3.1)	**0.0**	3.1	6.3
61	(45.2)	(41.9)	(38.7)	(35.5)	(32.3)	(29.0)	(25.8)	(22.6)	(19.4)	(16.1)	(12.9)	(9.7)	(6.5)	(3.2)	**0.0**	3.2
60	(50.0)	(46.7)	(43.3)	(40.0)	(36.7)	(33.3)	(30.0)	(26.7)	(23.3)	(20.0)	(16.7)	(13.3)	(10.0)	(6.7)	(3.3)	**0.0**

Figure 7-7. Tables of Percent Heating Change Per Degree of Indoor Temperature Change

$$1 - [\frac{(T2 - T0)}{(T1 - T0)}]$$

T0: **20**

Approximate Heating Savings for Space Temperature Change

20 degrees F Base Winter Temperature

Units are percent change from energy use at T1. Negative values are energy increase.

T1 \ T2	75	74	73	72	71	70	69	68	67	66	65	64	63	62	61	60
75	**0.0**	1.8	3.6	5.5	7.3	9.1	10.9	12.7	14.5	16.4	18.2	20.0	21.8	23.6	25.5	27.3
74	(1.9)	**0.0**	1.9	3.7	5.6	7.4	9.3	11.1	13.0	14.8	16.7	18.5	20.4	22.2	24.1	25.9
73	(3.8)	(1.9)	**0.0**	1.9	3.8	5.7	7.5	9.4	11.3	13.2	15.1	17.0	18.9	20.8	22.6	24.5
72	(5.8)	(3.8)	(1.9)	**0.0**	1.9	3.8	5.8	7.7	9.6	11.5	13.5	15.4	17.3	19.2	21.2	23.1
71	(7.8)	(5.9)	(3.9)	(2.0)	**0.0**	2.0	3.9	5.9	7.8	9.8	11.8	13.7	15.7	17.6	19.6	21.6
70	(10.0)	(8.0)	(6.0)	(4.0)	(2.0)	**0.0**	2.0	4.0	6.0	8.0	10.0	12.0	14.0	16.0	18.0	20.0
69	(12.2)	(10.2)	(8.2)	(6.1)	(4.1)	(2.0)	**0.0**	2.0	4.1	6.1	8.2	10.2	12.2	14.3	16.3	18.4
68	(14.6)	(12.5)	(10.4)	(8.3)	(6.3)	(4.2)	(2.1)	**0.0**	2.1	4.2	6.3	8.3	10.4	12.5	14.6	16.7
67	(17.0)	(14.9)	(12.8)	(10.6)	(8.5)	(6.4)	(4.3)	(2.1)	**0.0**	2.1	4.3	6.4	8.5	10.6	12.8	14.9
66	(19.6)	(17.4)	(15.2)	(13.0)	(10.9)	(8.7)	(6.5)	(4.3)	(2.2)	**0.0**	2.2	4.3	6.5	8.7	10.9	13.0
65	(22.2)	(20.0)	(17.8)	(15.6)	(13.3)	(11.1)	(8.9)	(6.7)	(4.4)	(2.2)	**0.0**	2.2	4.4	6.7	8.9	11.1
64	(25.0)	(22.7)	(20.5)	(18.2)	(15.9)	(13.6)	(11.4)	(9.1)	(6.8)	(4.5)	(2.3)	**0.0**	2.3	4.5	6.8	9.1
63	(27.9)	(25.6)	(23.3)	(20.9)	(18.6)	(16.3)	(14.0)	(11.6)	(9.3)	(7.0)	(4.7)	(2.3)	**0.0**	2.3	4.7	7.0
62	(31.0)	(28.6)	(26.2)	(23.8)	(21.4)	(19.0)	(16.7)	(14.3)	(11.9)	(9.5)	(7.1)	(4.8)	(2.4)	**0.0**	2.4	4.8
61	(34.1)	(31.7)	(29.3)	(26.8)	(24.4)	(22.0)	(19.5)	(17.1)	(14.6)	(12.2)	(9.8)	(7.3)	(4.9)	(2.4)	**0.0**	2.4
60	(37.5)	(35.0)	(32.5)	(30.0)	(27.5)	(25.0)	(22.5)	(20.0)	(17.5)	(15.0)	(12.5)	(10.0)	(7.5)	(5.0)	(2.5)	**0.0**

$$1 - [\frac{(T2 - T0)}{(T1 - T0)}]$$

T0: **10**

Approximate Heating Savings for Space Temperature Change

10 degrees F Base Winter Temperature

Units are percent change from energy use at T1. Negative values are energy increase.

T1 \ T2	75	74	73	72	71	70	69	68	67	66	65	64	63	62	61	60
75	**0.0**	1.5	3.1	4.6	6.2	7.7	9.2	10.8	12.3	13.8	15.4	16.9	18.5	20.0	21.5	23.1
74	(1.6)	**0.0**	1.6	3.1	4.7	6.3	7.8	9.4	10.9	12.5	14.1	15.6	17.2	18.8	20.3	21.9
73	(3.2)	(1.6)	**0.0**	1.6	3.2	4.8	6.3	7.9	9.5	11.1	12.7	14.3	15.9	17.5	19.0	20.6
72	(4.8)	(3.2)	(1.6)	**0.0**	1.6	3.2	4.8	6.5	8.1	9.7	11.3	12.9	14.5	16.1	17.7	19.4
71	(6.6)	(4.9)	(3.3)	(1.6)	**0.0**	1.6	3.3	4.9	6.6	8.2	9.8	11.5	13.1	14.8	16.4	18.0
70	(8.3)	(6.7)	(5.0)	(3.3)	(1.7)	**0.0**	1.7	3.3	5.0	6.7	8.3	10.0	11.7	13.3	15.0	16.7
69	(10.2)	(8.5)	(6.8)	(5.1)	(3.4)	(1.7)	**0.0**	1.7	3.4	5.1	6.8	8.5	10.2	11.9	13.6	15.3
68	(12.1)	(10.3)	(8.6)	(6.9)	(5.2)	(3.4)	(1.7)	**0.0**	1.7	3.4	5.2	6.9	8.6	10.3	12.1	13.8
67	(14.0)	(12.3)	(10.5)	(8.8)	(7.0)	(5.3)	(3.5)	(1.8)	**0.0**	1.8	3.5	5.3	7.0	8.8	10.5	12.3
66	(16.1)	(14.3)	(12.5)	(10.7)	(8.9)	(7.1)	(5.4)	(3.6)	(1.8)	**0.0**	1.8	3.6	5.4	7.1	8.9	10.7
65	(18.2)	(16.4)	(14.5)	(12.7)	(10.9)	(9.1)	(7.3)	(5.5)	(3.6)	(1.8)	**0.0**	1.8	3.6	5.5	7.3	9.1
64	(20.4)	(18.5)	(16.7)	(14.8)	(13.0)	(11.1)	(9.3)	(7.4)	(5.6)	(3.7)	(1.9)	**0.0**	1.9	3.7	5.6	7.4
63	(22.6)	(20.8)	(18.9)	(17.0)	(15.1)	(13.2)	(11.3)	(9.4)	(7.5)	(5.7)	(3.8)	(1.9)	**0.0**	1.9	3.8	5.7
62	(25.0)	(23.1)	(21.2)	(19.2)	(17.3)	(15.4)	(13.5)	(11.5)	(9.6)	(7.7)	(5.8)	(3.8)	(1.9)	**0.0**	1.9	3.8
61	(27.5)	(25.5)	(23.5)	(21.6)	(19.6)	(17.6)	(15.7)	(13.7)	(11.8)	(9.8)	(7.8)	(5.9)	(3.9)	(2.0)	**0.0**	2.0
60	(30.0)	(28.0)	(26.0)	(24.0)	(22.0)	(20.0)	(18.0)	(16.0)	(14.0)	(12.0)	(10.0)	(8.0)	(6.0)	(4.0)	(2.0)	**0.0**

Figure 7-7. (*Continued*. Tables of Percent Heating Change Per Degree of Indoor Temperature Change

$$1-[\frac{(T2-T0)}{(T1-T0)}]$$

T0: **0**

Approximate Heating Savings for Space Temperature Change

0 degrees F Base Winter Temperature

Units are percent change from energy use at T1. Negative values are energy increase.

T1 \ T2	75	74	73	72	71	70	69	68	67	66	65	64	63	62	61	60
75	**0.0**	1.3	2.7	4.0	5.3	6.7	8.0	9.3	10.7	12.0	13.3	14.7	16.0	17.3	18.7	20.0
74	(1.4)	**0.0**	1.4	2.7	4.1	5.4	6.8	8.1	9.5	10.8	12.2	13.5	14.9	16.2	17.6	18.9
73	(2.7)	(1.4)	**0.0**	1.4	2.7	4.1	5.5	6.8	8.2	9.6	11.0	12.3	13.7	15.1	16.4	17.8
72	(4.2)	(2.8)	(1.4)	**0.0**	1.4	2.8	4.2	5.6	6.9	8.3	9.7	11.1	12.5	13.9	15.3	16.7
71	(5.6)	(4.2)	(2.8)	(1.4)	**0.0**	1.4	2.8	4.2	5.6	7.0	8.5	9.9	11.3	12.7	14.1	15.5
70	(7.1)	(5.7)	(4.3)	(2.9)	(1.4)	**0.0**	1.4	2.9	4.3	5.7	7.1	8.6	10.0	11.4	12.9	14.3
69	(8.7)	(7.2)	(5.8)	(4.3)	(2.9)	(1.4)	**0.0**	1.4	2.9	4.3	5.8	7.2	8.7	10.1	11.6	13.0
68	(10.3)	(8.8)	(7.4)	(5.9)	(4.4)	(2.9)	(1.5)	**0.0**	1.5	2.9	4.4	5.9	7.4	8.8	10.3	11.8
67	(11.9)	(10.4)	(9.0)	(7.5)	(6.0)	(4.5)	(3.0)	(1.5)	**0.0**	1.5	3.0	4.5	6.0	7.5	9.0	10.4
66	(13.6)	(12.1)	(10.6)	(9.1)	(7.6)	(6.1)	(4.5)	(3.0)	(1.5)	**0.0**	1.5	3.0	4.5	6.1	7.6	9.1
65	(15.4)	(13.8)	(12.3)	(10.8)	(9.2)	(7.7)	(6.2)	(4.6)	(3.1)	(1.5)	**0.0**	1.5	3.1	4.6	6.2	7.7
64	(17.2)	(15.6)	(14.1)	(12.5)	(10.9)	(9.4)	(7.8)	(6.3)	(4.7)	(3.1)	(1.6)	**0.0**	1.6	3.1	4.7	6.3
63	(19.0)	(17.5)	(15.9)	(14.3)	(12.7)	(11.1)	(9.5)	(7.9)	(6.3)	(4.8)	(3.2)	(1.6)	**0.0**	1.6	3.2	4.8
62	(21.0)	(19.4)	(17.7)	(16.1)	(14.5)	(12.9)	(11.3)	(9.7)	(8.1)	(6.5)	(4.8)	(3.2)	(1.6)	**0.0**	1.6	3.2
61	(23.0)	(21.3)	(19.7)	(18.0)	(16.4)	(14.8)	(13.1)	(11.5)	(9.8)	(8.2)	(6.6)	(4.9)	(3.3)	(1.6)	**0.0**	1.6
60	(25.0)	(23.3)	(21.7)	(20.0)	(18.3)	(16.7)	(15.0)	(13.3)	(11.7)	(10.0)	(8.3)	(6.7)	(5.0)	(3.3)	(1.7)	**0.0**

$$1-[\frac{(T2-T0)}{(T1-T0)}]$$

T0: **-10**

Approximate Heating Savings for Space Temperature Change

-10 degrees F Base Winter Temperature

Units are percent change from energy use at T1. Negative values are energy increase.

T1 \ T2	75	74	73	72	71	70	69	68	67	66	65	64	63	62	61	60
75	**0.0**	1.2	2.4	3.5	4.7	5.9	7.1	8.2	9.4	10.6	11.8	12.9	14.1	15.3	16.5	17.6
74	(1.2)	**0.0**	1.2	2.4	3.6	4.8	6.0	7.1	8.3	9.5	10.7	11.9	13.1	14.3	15.5	16.7
73	(2.4)	(1.2)	**0.0**	1.2	2.4	3.6	4.8	6.0	7.2	8.4	9.6	10.8	12.0	13.3	14.5	15.7
72	(3.7)	(2.4)	(1.2)	**0.0**	1.2	2.4	3.7	4.9	6.1	7.3	8.5	9.8	11.0	12.2	13.4	14.6
71	(4.9)	(3.7)	(2.5)	(1.2)	**0.0**	1.2	2.5	3.7	4.9	6.2	7.4	8.6	9.9	11.1	12.3	13.6
70	(6.3)	(5.0)	(3.8)	(2.5)	(1.3)	**0.0**	1.3	2.5	3.8	5.0	6.3	7.5	8.8	10.0	11.3	12.5
69	(7.6)	(6.3)	(5.1)	(3.8)	(2.5)	(1.3)	**0.0**	1.3	2.5	3.8	5.1	6.3	7.6	8.9	10.1	11.4
68	(9.0)	(7.7)	(6.4)	(5.1)	(3.8)	(2.6)	(1.3)	**0.0**	1.3	2.6	3.8	5.1	6.4	7.7	9.0	10.3
67	(10.4)	(9.1)	(7.8)	(6.5)	(5.2)	(3.9)	(2.6)	(1.3)	**0.0**	1.3	2.6	3.9	5.2	6.5	7.8	9.1
66	(11.8)	(10.5)	(9.2)	(7.9)	(6.6)	(5.3)	(3.9)	(2.6)	(1.3)	**0.0**	1.3	2.6	3.9	5.3	6.6	7.9
65	(13.3)	(12.0)	(10.7)	(9.3)	(8.0)	(6.7)	(5.3)	(4.0)	(2.7)	(1.3)	**0.0**	1.3	2.7	4.0	5.3	6.7
64	(14.9)	(13.5)	(12.2)	(10.8)	(9.5)	(8.1)	(6.8)	(5.4)	(4.1)	(2.7)	(1.4)	**0.0**	1.4	2.7	4.1	5.4
63	(16.4)	(15.1)	(13.7)	(12.3)	(11.0)	(9.6)	(8.2)	(6.8)	(5.5)	(4.1)	(2.7)	(1.4)	**0.0**	1.4	2.7	4.1
62	(18.1)	(16.7)	(15.3)	(13.9)	(12.5)	(11.1)	(9.7)	(8.3)	(6.9)	(5.6)	(4.2)	(2.8)	(1.4)	**0.0**	1.4	2.8
61	(19.7)	(18.3)	(16.9)	(15.5)	(14.1)	(12.7)	(11.3)	(9.9)	(8.5)	(7.0)	(5.6)	(4.2)	(2.8)	(1.4)	**0.0**	1.4
60	(21.4)	(20.0)	(18.6)	(17.1)	(15.7)	(14.3)	(12.9)	(11.4)	(10.0)	(8.6)	(7.1)	(5.7)	(4.3)	(2.9)	(1.4)	**0.0**

Figure 7-7 (*Continued*. Tables of Percent Heating Change Per Degree of Indoor Temperature Change

Example:
Same building in a different city with a heating season average outdoor temperature of 35 degF.

T1=70
T2=69 % change = (1- [(69-35)/(70-35)])
T0=35 =2.9% reduction

Percent Heating Change Per Degree of Indoor Temperature Change

It is common to express the benefits of changing indoor temperature settings in terms of percent change per degree. The percentage number will float depending on the indoor and outdoor temperatures.
Note: Percentages must be kept in context. For example, a large percentage reduction of heating usage in Miami may be meaningless when there is practically no heating needed.

Timed Tenant Override

Basis of Savings: Reduced run time.

- When after-hours tenants invoke the override for comfort, limit the amount of time this will remain active before the system reverts to unoccupied mode again (e.g. 2 hours), and what equipment needs to operate.
  Zone sensor override button allows compressing schedules to the bulk average occupancy times for the building, further reducing run time; i.e., the scheduled on/off times do not need to accommodate occasional outlier events—those can be added as needed from the local override button.

Deadband Thermostats

Basis of Savings: Prevent simultaneous heating and cooling.

- Use or adjust thermostats to allow for a zero-energy deadband, a range where no action is taken by the HVAC system. A simple, but effective routine that should be part of every non-critical comfort control application. The deadband separation of heating and cooling controlled devices provides a positive separation to prevent the energy waste associated with heat/cool overlap.
- A minimum of four (4) deg F deadband for space temperatures is suggested.
- A minimum of five (5) deg F deadband for sequential air handler heating/cooling equipment is suggested.

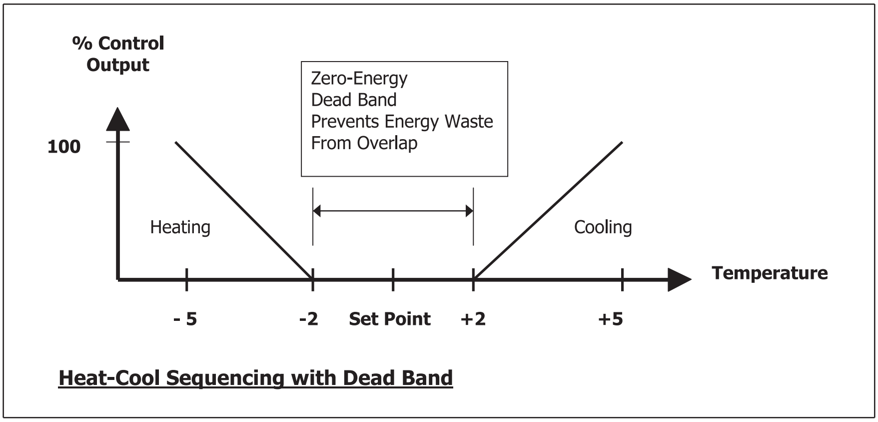


Figure 7-8. Diagram of Deadband Control
Source: *Energy Management Handbook*, 6th Ed, Chap 22.

VAV Box Control Deadband

Basis of Savings: Prevent simultaneous heating and cooling.

- Independent VAV heating and cooling set points with a minimum 4 degree zero-energy deadband in between, in lieu of a single setting with a proportional band. This will increase the zero-energy deadband and reduce simultaneous heating and cooling.

NOTE: For normal VAV systems where the ventilation air comes mixed with the supply air, there is a minimum stop setting, for minimum ventilation, that creates some heating-cooling overlap. The deadband application for VAV boxes minimizes, but does not eliminate this overlap.

Air Handler Control Deadband

Basis of Savings: Prevent simultaneous heating and cooling.

- Independent preheat, mixed air, and cooling coil control set points, with a minimum five (5) degree zero-energy deadband in between, in lieu of a single setting with a proportional band. This will reduce simultaneous heating and cooling.

Shed Non-essential Equipment at Peak Times

Basis of Savings: Savings are from reduction in utility demand charges as well as energy savings.

- Schedule things like fire pump testing and battery charging to off peak times.

- Any load shifting to off peak serves this measure.
- Demand charges are usually determined from the greatest demand recorded any time in the month, so this must be done faithfully to achieve savings, i.e. through automation.

Outside Air for Morning Building Cool-down (Air-purge)
Basis of Savings: Reduced heating and cooling energy related to tempering outside air.

- Use outside air for pre-occupancy cool-down cycle in the summer, in conjunction with scheduled off time and/or optimal start routines. This routine takes advantage of cool nights and flushes the warm air out of the building that has accumulated and avoids or reduces demand on the primary cooling equipment.

NOTE: Normally limited to dry climates. In humid climates or whenever the outside air humidity is higher than building humidity, this has the potential to add load if subsequent de-humidification (and energy expenditure) occurs as a result.

Extended Full Economizer Range of Operation
Basis of Savings: Reduced run hours of cooling compressor equipment.

- Adjust settings for economizer operation based on outdoor dew point levels to keep cooling equipment off longer during mild and dry weather days.
- If outside air dew point levels are below 47 degrees and outside air dry bulb temperature is below 65 degrees, it should be possible in most cases to circulate the available air instead of 55 degree air and maintain comfort in many buildings. Doing this will delay the use of mechanical cooling and create savings.
- Much of the savings in mechanical cooling energy is taken by increased air flow rates in VAV systems, however there is still a net gain. Rule of thumb is that half of the savings from keeping the compressors off is spent in added fan energy.
- Some form of humidity watch-dog control is suggested except in very dry climates.

Extended Partial Economizer Operation
Basis of Savings: Reduced mechanical cooling energy use in periods where full economizer is not practical but outside air in combination with mechanical cooling is more economical than plain mechanical cooling.

- Extend air-side economizer operation as far as practical, with the use of enthalpy comparison sensors, in lieu of the standard practice of limiting the economizer operation to 55 degrees F or less. With this method, whenever the outside air has less energy in it than the return air, cooling costs will be reduced by using outside air instead of return air. On days when the outside air is significantly drier than inside air, the free cooling range can be extended and outside air temperatures above return air temperatures can be utilized. When the outside air is humid, 55 degrees F is a smart cutoff point.
- Note: Use this method with caution. Enthalpy sensors are difficult to calibrate and so accuracy of the readings and resulting control decisions have a measure of uncertainty with them. Actual results may vary from predicted results depending on accuracy and drift. Comfort complaints will be the feedback to operations if they drift in one direction, but lost savings will be the result of drifting in the other direction, with no feedback.
- An alternate to enthalpy sensors or enthalpy calculations, with the same results and less accuracy problems, is dew point control. Measuring dew point directly is expensive and high maintenance (e.g. a chilled mirror), but the calculation is straight forward and will perform well if high quality and accurate inputs for dry bulb temperature and relative humidity are used.
- The limiting factor on when the extended economizer has reached its limit is when it costs more to process the outside air than the inside air. Were it not for humidity, the measurement would simply be return air temperature vs. outside air temperature, and which one to introduce into the cooling coil. On a psychrometric chart, the dew point lines are horizontal and flat and can be visualized as a "water level" for the building. Once the water level is too high, the HVAC system will have additional work to do (energy expenditure) to remove it. Thus, by monitoring the "level" of outdoor moisture, the DDC system would end the use of economizer during humid days. The indoor moisture "level" may be omitted in dry climates and may be used if desired in more humid climates, depending upon the degree of control desired.
- For a comparison of dry-bulb, enthalpy, and dew point economizer strategies, **see Figure 7-9.**

Boiler Lockout from Outside Air Temperature

Basis of Savings: Reduced run time of the boiler.

- Generally, prohibit boiler operation above 60 degrees F. Suggested control would be to cut-in at 55 and cut-out at 60 degrees F outside air. This is open loop control, but saves energy compared to "start based on demand" routines, since all it takes is one strong demand point to run the large machine for extended periods, or to start it unnecessarily in hot weather.
- Where outside air reset schedules are used for hot water temperatures (hot water boilers only), it is preferable to include the lockout in the same schedule, so that the two functions stay properly sequenced over time. If separated, the independent adjustments made can easily create unwanted sequencing.

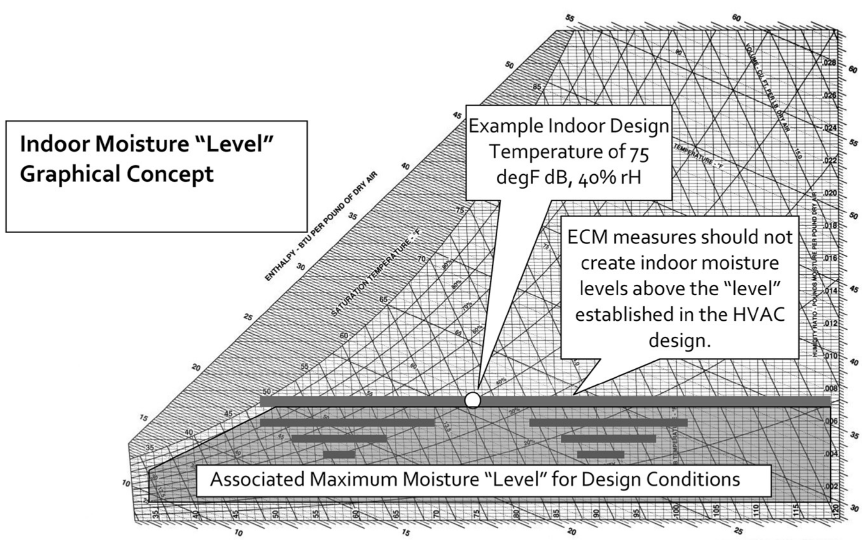


Figure 7-9. Indoor Moisture "Level" Concept Relationship to Economizer Optimization.

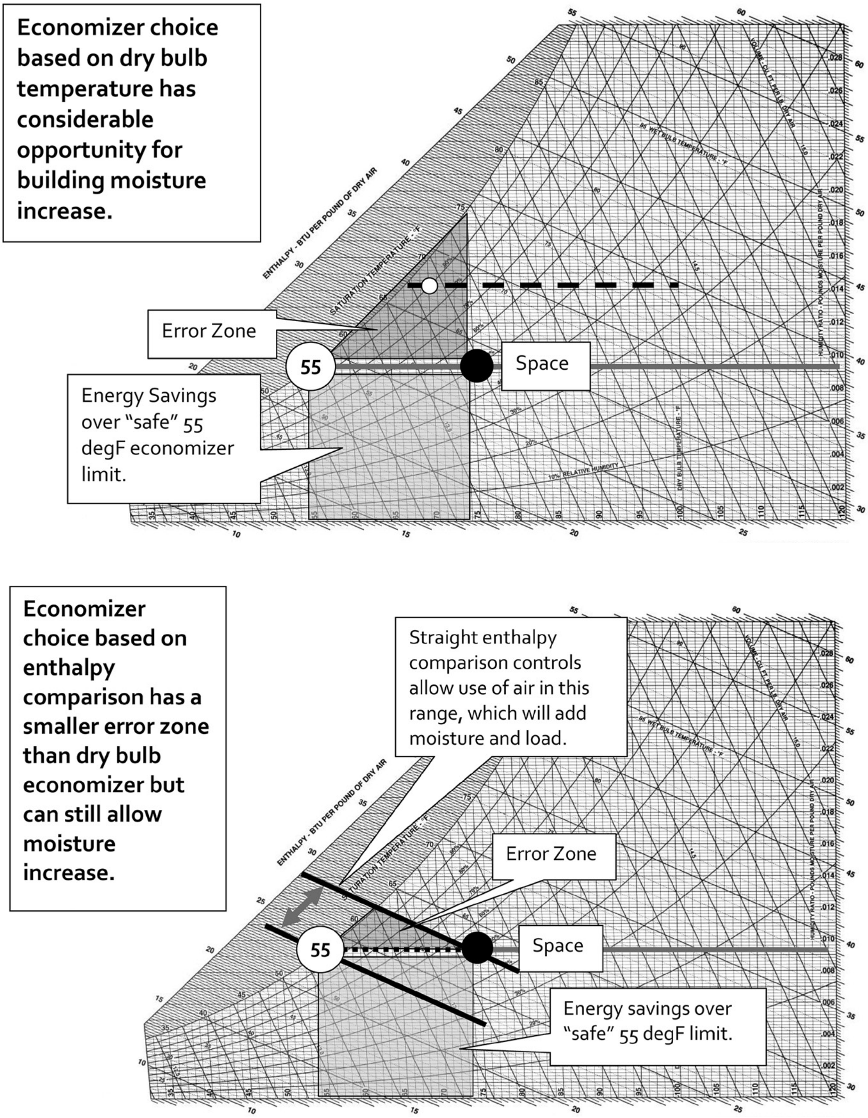


Figure 7-9. (*Continued*). Indoor Moisture "Level" Concept Relationship to Economizer Optimization.

Chiller Lockout from Outside Air Temperature

Basis of Savings: Reduced run time of the chiller.

- Generally, prohibit chiller operation below 50 degrees F, and coordinate with air and water economizer settings. Suggested control would be to 'cut-in' at 55 and 'cut-out' at 50 degrees F outside air. This is open loop control, but saves energy compared to "start based on demand" routines, since all it takes is one strong demand point to run the large machine for extended periods, or to start it unnecessarily in cold weather.
- Where outside air reset schedules are used for chilled water temperatures, it is preferable to include the lockout in the same schedule, so that the two functions stay properly sequenced over time. If separated, the independent adjustments made can easily create unwanted sequencing.

**Chilled Water Reset from Return Temperature—
Constant Flow Pumping**

Basis of Savings: Improved refrigeration cycle.

- Approximately 1-1.5% power reduction for each degree raised.
- Reset is from return temperature. As cooling load is reduced, return water temperature drops and reflects the reduced load.
- Chilled water supply temperature is raised according to a reset schedule. Example reset schedule:

CHW Return	CHW Supply
55 degF	45 degF
50 degF	48 degF

- Limits must be placed on the reset schedule. Chilled water temperatures above 50 degrees will do very little comfort cooling and can significantly reduce dehumidification action in cooling coils.

Chilled Water Reset—Variable Flow Pumping

Basis of Savings: Improved refrigeration cycle.

- Approximately 1-1.5% power reduction for each degree raised.
- Raising chilled water temperature increases flow rate requirements per ton and increases pump energy, eroding savings; usually eroding half to 2/3 of the savings. Still a net gain.
- Reset cannot be from return temperature since return is relatively constant with this system.
- Cooling load reduction is either measured directly (tons) or implied by outside air temperature.
- Chilled water supply temperature is raised according to a reset schedule. Example reset schedules below. Note the outdoor air reset schedule includes the chiller lockout function.

Cooling Load	CHW Supply
100%	45 degF
50%	48 degF

Outside Air	CHW Supply
80 degF	45 degF
60 degF	48 degF
Below 60 degF	Chiller Off

- Limits must be placed on the reset schedule. Chilled water temperatures above 50 degrees will do very little comfort cooling and can significantly reduce dehumidification action in cooling coils.

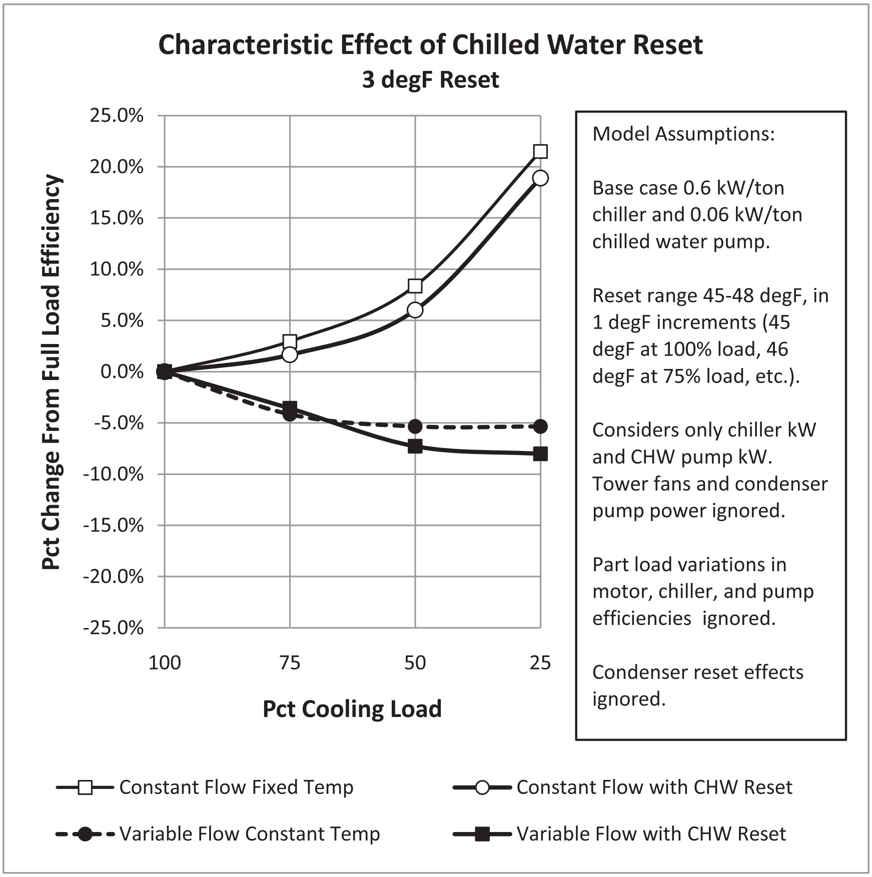


Figure 7-10. Characteristic Effect of Chilled Water Reset

The tradeoff between chiller efficiency and pump energy.

For variable flow systems there is a slight gain if the reset is narrow. Compare the two graphs with 4 degF and 8 degF reset; with 8 degF the increased water flow is greater and pump energy overtakes chiller savings

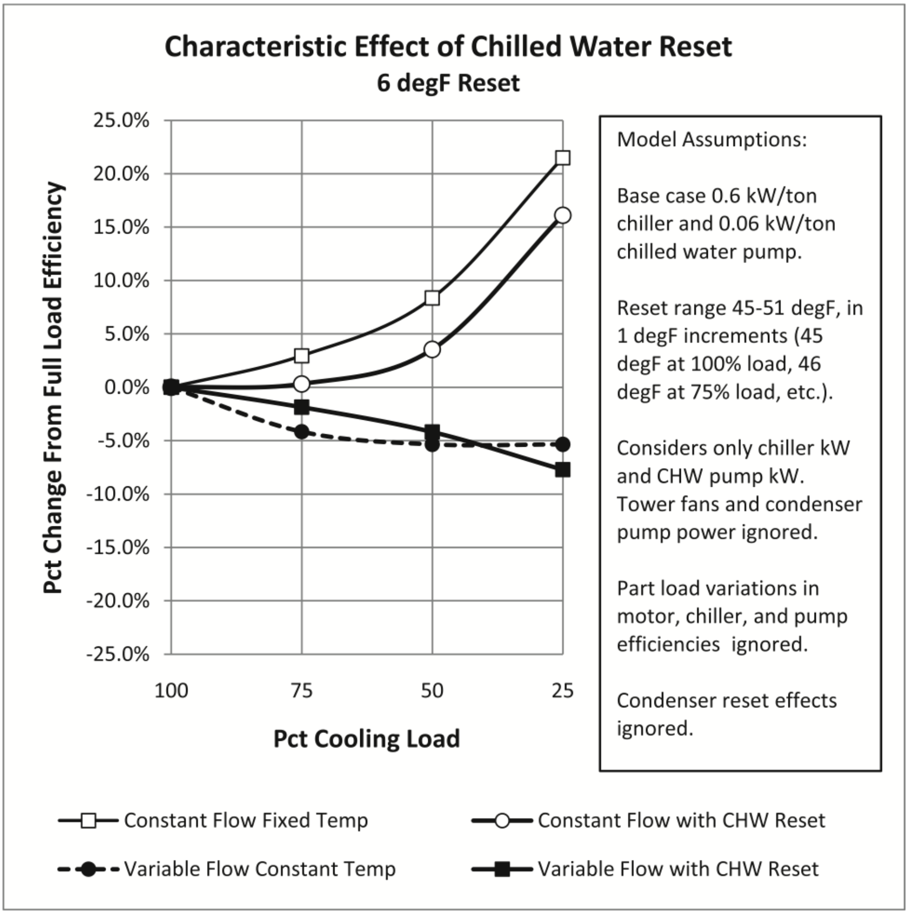


Figure 7-11
Characteristic Effect of Chilled Water Reset
The tradeoff between chiller efficiency and pump energy.
For variable flow systems there is a slight gain if the reset is narrow. Compare the two graphs with 4 degF and 8 degF reset; with 8 degF the increased water flow is greater and pump energy overtakes chiller savings.

Supply Air Temperature Reset for VAV Air Systems from Supply Fan Speed

- This method uses the supply fan VAV output percent command along with outside air temperature to estimate when most of the VAV boxes are at minimum and in the heating mode, signaling the time to begin resetting supply air temperature. It was originally used with pneumatic controls, measuring the fan control output which was a very rough estimate of fan flow rate. The end result is to imply when the VAV boxes are in heating mode and minimum air flow, to activate the reset. Fan volume is proportional to fan speed, so x% speed implies x% flow. The logic is valid for the overall group of VAV boxes, when they behave similarly, but is blind to any zone of high demand. An important feature of this reset schedule is that NO reset occurs during summer months, preserving the basis of VAV savings. A simple reset schedule is used for this method as follows:

SA Fan % Capacity	Outside Air Temp (degF)	Supply Air Temp Setpoint (degF)
100%	Any	55
40%*	40**	55
20%	10	62

*The percent of the supply fan capacity at which the reset begins should correspond to the aggregate VAV box minimum settings. For example, if the aggregate (weighed average) VAV box setting is 40%, then a supply fan value of 40% indicates that most, but not all, VAV boxes are at minimum.

**The temperature at which the reset begins should correspond to the building thermal balance point, which is the point below which the 'self heating' nature of the building is overcome by envelope losses and ventilation loads and heating use begin to dominate.

Note: this method will require some experimentation to get right. Percent fan capacity (VFD output, inlet vane position, etc.) is close, but not equal, to percent of air flow.

Supply Air Temperature Reset for VAV Air Systems from Outside Air Temperature

- Outside air can be used for supply air reset of VAV systems, with the goal of knowing when heating loads will begin to appear. This is easy to implement, but is an open loop control system, i.e. no amount of change in supply temperature can alter outside temperature. Like all open loop control routines, the settings will need to be adjusted through a couple of seasons to get it reasonably close. Note also that setting a supply air temperature too low has no impact on the process of comfort, since the VAV reheat coils will simply pick up the added load. Thus, this type of control provides only partial feedback for optimization and will usually result in missed savings opportunities.

- A reset schedule for this method would look like the following (note it is not linear):

Outside Air Temp (degF)	Supply Air Temp Set point (degF)
Above 70	55
60 *	60
40	65**

*The values of the lower reset parameter will vary depending on the balance temperature of the building.
**The values of the upper reset parameter will vary depending on the response from the zones (complaints) if any happen to need cooling in winter.

Supply Air Temperature Reset for Constant Volume Air Systems from Return Air Temperature

Basis of Savings: Reduced unnecessary reheat load from duct heating coils.

- For constant volume systems, the return air temperature will vary with space temperature and is a good indicator of space temperature. Therefore, resetting from return air temperature is akin to resetting from (average) space temperature.
- Variable flow systems are not effectively reset from return air/water temperatures.
- Most commonly linear, the reset schedules can be more effective with multiple lines; one for summer/one for winter.
- Sample reset schedule for a constant volume air system:

Return Air Temp (degF)	Supply Air Temp (degF)
77	55
71	61
65	67

Reset Hot Water Converters from Outside Air

Basis of Savings: Reduced standby losses from heating fluid distribution piping.

- Compared to a constant temperature setting, this will reduce standby losses. In the cooling season, these heat losses are also unnecessary cooling loads.
- If the reset is for a boiler instead of a single converter, coordinating the reset schedule with boiler lockout is preferred over a separate lockout setting since, over time, they will tend to become different—this way they are forced to stay in proper sequence.

Outside Air Temp (degF)	Leaving HW temp (degF)
Above 65	Boiler Off
65	120 (or minimum temperature)
10	180 (or maximum temperature)

Reset Fan Static Pressure from Outside Air

Basis of Savings: Reduced fan Hp.

- VAV system design air flows occur in hot weather. In winter, air flows are less, and can be achieved with less maintained duct pressure.

Outside Air Temp (degF)	Static Pressure (in. w.c.)
70	1.0
40	0.8

Water Loop Heat Pump Loop Water Reset and Sequencing
Basis of Savings: *Reduced refrigeration lift in summer, increased heating COP in winter, and reduced reliance on either one when heat can be moved between zones with only circulation energy.*

This system uses an array of water-source heat pumps served by a continuously circulating loop that includes a primary heater (e.g. hot water boiler) and a primary cooler (e.g. fluid cooler or cooling tower with heat exchanger). The most significant savings for this system come not from controls, but the good fortune of having and good proportion of zones needing heat while the others need cooling, because in this case the heat is literally moved from room to room. Controls merely seek to minimize the 'new energy' from the primary heat and cool equipment. Key to optimizing is knowing when most heat pumps are running in heating mode, most heat pumps are running in cooling mode, or heating and cooling are in near equal proportion—a different best loop temperature is identifiable for each condition.

- In cooling mode, the heat rejection is evaporative and primary energy input is small. The main concern is not overlapping cooling and heating control.
- In heating mode, the heat is merely 'moved' by the heat pump but the heat comes from the boiler, so a tradeoff exists between the cost of raising the water temperature vs. the cost of operating at a lower COP; settings shown below presume a natural gas boiler and favorable comparison between gas and electric costs.
- When in swing mode there is a mix of zones calling for heating and cooling; this mode is very efficient, since heat pumps literally move heat from room to room without any input from the primary equipment.
- A constant loop temperature is undesirable for this system since it risks 'fighting' (cooler and boiler operating together).

Determine mode as heat, cool, or swing:

- Set predominant cooling mode when outside air is >75F
- Set predominant heating mode when outside air is <40F.
- Set swing mode when not in either heat or cool mode (default mode)

When in predominant **cooling mode**:

Cooling cut-in >70F and heating cut in <50F, boiler is locked out

When in predominant **heating mode**:

Heating cut-in <75F and cooling cut in >90F, cooler is locked out

When in **swing mode**:

Cooling cut in >80F and heating cut in <50F, neither the cooler or heater operate

Mid-Range Vestibule Temperature

Basis of Savings: Reduces temperature difference at the envelope opening means less loss.

- For vestibule spaces, temper the space to a mid-range temperature, half way between indoor and outdoor temperatures.

Operable Window Interlock

Basis of Savings: Reduce loss of conditioned air whenever windows are opened. Lost air results in more make-up air that must be tempered.

- Where operable windows are used, provide automatic control interlock to disable the HVAC serving that room, to avoid heating and cooling the outdoors. This can also be used as a freeze alarm.

Roll-up Door Interlock

Basis of Savings: Reduce loss of conditioned air whenever windows are opened. Lost air results in more make-up air that must be tempered.

- Provide automatic control interlock to disable the HVAC serving interior area, to avoid heating and cooling losses when the door is up. This will encourage people to keep the door closed.

Staggered Heating and Cooling System Start-Up
(Buildings with demand charges)

Basis of Savings: Reduced demand charges by avoiding setting a peak event.

- After extended 'off' periods, such as night set back, normal automatic control response will be to drive the heating and cooling equipment in order to reach the occupied set point. To avoid setting the utility maximum electrical demand value and subsequent higher demand charges coming out of unoccupied periods, bring large electrically driven heating and cooling equipment loads on in segments for 30 minutes or until the process has "caught up" and is no longer at full load. Once at setpoint, electric heat will cycle and overall demand will reduce to half or less compared to "all on." Once the first group of heaters is caught up and cycling, the next group of heaters can be activated.

NOTE: This is not in reference to inrush currents for starting motors.

For example:

Consider a building heated by electric resistance heat. By default, coming out of unoccupied periods during winter will result in 100% of the heat being energized simultaneously. Controlling in 2 or more zones and allowing ample pull-up time for zone 1 to get to temperature and begin cycling normally before starting zone 2 will reduce the maximum demand for that day and, if done diligently through automation, will reduce the overall seasonal demand and demand charges.

- Electric resistance heat (winter)
- Packaged HVAC cooling equipment (summer)

LIGHTING CONTROL STRATEGIES—ADVANCED

Daylight Harvesting

Basis of Savings: Reduced run time of electric lighting.

- Control perimeter lighting on/off or modulate, in response to daytime sunlight entering the building.
- In-board/out-board switching for interior lighting.

Programmable Lighting Ballast

Basis of Savings: Reduced run time with additional flexibility in scheduled operation.

- Special "addressable" lighting ballast are available that can be controlled individually, without conventional relays. Remote control, custom scheduling by room, by area, and by light fixture are possible with this system to minimize lighting use. This allows greater granularity of control and providing light only where/when needed.

HVAC CONTROL STRATEGIES—ADVANCED

NOTE: Many optimization routines rely on end-use polling of demand or valve/damper positions such as "most open valve" routines. This can be done cost effectively without actual measurement of position—by polling the individual "percent commanded output." This is referred to as "implied position" and is acceptable in most cases in lieu of actual position.

Note also that routines that use polling have the potential to be inefficiently operated if one errant measurement exists. For polling space temperatures, for example, limiting the user adjustment is strongly recommended in conjunction with demand polling of space controls. Additionally, it may make sense to 'discard' the high and low values from such polling to prevent errant operation. Some polling techniques wait to react until several "calls" exist; this reduces the chance of an errant signal driving the entire heat/cool plant, but also introduces dissatisfaction if a single and legitimate call exists, since the control system would ignore it.

Occupancy Sensor Control of HVAC
(where a dedicated VAV box or terminal unit exists)

Basis of Savings: Reduced heating and cooling load. Reduced VAV system reheat penalty. Reduced ventilation load.

- When the room is sensed as unoccupied, the space temperatures revert to unoccupied values.
- For VAV boxes, minimum flow settings are adjusted to zero.
- For unit ventilators, temperature and ventilation settings revert to unoccupied values.
- For fan coils, fans turn off.

Note: Perimeter areas strongly influenced by envelope loads may need specially defined "day time unoccupied" settings to avoid comfort complaints and to allow quicker return to occupied temperatures when the room is re-occupied.

Optimized Supply Air Static Pressure Reset—VAV Systems

Basis of Savings: Reduced fan horsepower via affinity laws.

- This requires polling of individual VAV boxes for air valve position and reduces system pressure until at least one box is 90% open, thereby providing the optimal system duct pressure (just enough pressure). This reduces fan horsepower.

Optimized Supply Water Pressure Reset—Variable Pumping Systems from Zone Demand

Basis of Savings: Reduced pump horsepower via affinity laws.

- This requires polling individual air handlers for control valve position and reduces system pressure until at least one control valve is 90% open, thereby providing the optimal system water pressure (just enough pressure). This reduces pump horsepower.

Optimized Condenser Water Reset

Basis of Savings:

1. Cooling towers: refrigeration compressor power savings of 1-1.5% per degree lowered. Optimizing allows this to occur without excessive tower fan energy penalty.
2. Evaporative cooling sequenced with mechanical cooling: reduced run hours of mechanical cooling equipment, while avoiding moisture issues from excessive indoor humidity.

- This uses outdoor air wet bulb temperature, which can be measured directly, but is usually calculated from temperature and humidity. The evaporative process can get close to, but never reach or exceed, the wet bulb temperature. By knowing the wet bulb temperature, the control system will know its boundaries and won't try to achieve something it cannot. For water-cooled refrigeration equipment, the low limit for the reset is normally around 55 or 65 degrees F and the chiller manufacturer needs to be consulted to confirm. Resetting the cooling water temperature down in this way, in lieu of a constant temperature setting, will reduce kW/ton energy use and demand.

Note: achieving colder condenser water from a cooling tower is a tradeoff between improved chiller kW/ton and increased cooling tower fan kW/ton, so evaluation for diminishing returns is required.

- See **Chapter 9—*Quantifying Savings*** for expanded example of this ECM

Optimized Supply Air Temperature Reset for VAV Air Systems from Zone Demand

Basis of Savings: Reduced VAV reheat penalty in winter without losing cooling system benefit of constant cold air temperature.

- This requires polling individual VAV box air valve position and reheat valve position. This utilizes a fixed temperature for cooling (e.g. 55 degrees F) with no reset at all until polling of individual boxes indicates that most of the boxes are in heating—only then is the SA temperature allowed to gradually be reset upward to its maximum limit (e.g. 62 degrees F). The reset is accomplished by polling VAV reheat valve positions and the air temperature is gradually reset upwards until at least one VAV box reheat valve is 90% open, thereby providing optimal air temperature (just warm enough). Simultaneously, VAV damper positions are polled to be sure enough cooling is being provided for any zone still calling for cooling, and cooling will usually prevail if the two are at odds.

CO_2 Demand Controlled Ventilation (DCV)

Basis of Savings: Reduced heating and cooling energy for tempering outside air. Energy savings will be proportional to reduction in outside air intake.

- Also described as ventilation reset from CO_2
- Has application for large open assembly areas, characterized by a single point of CO_2 control in the occupied space of the single zone unit serving that area. Applications include a ballroom, theatre or open plan office area. A separate CO_2 point of control would be required for each dividable meeting area, each assembly area, and each group area, so that areas of great ventilation demand are served appropriately and do not get overlooked by sensor averaging.
- In essence, this control method uses a CO_2 sensor(s) as a people counter, thereby optimizing the use of outside air, and the energy required to condition it.

Note: When varying outside air intake, exhaust will normally also need to be varied to maintain building pressure/air balance.

Optimized Water-Side Economizer Pumping

Basis of Savings: Reduced Pump Energy.

When mechanical cooling is active, pumping control includes consideration of chillers. When in flat plate mode, chillers are off and so there is freedom for other control options. Using "water economizer mode active" as a yes/no software interlock, the non chiller mode of control can be different than the chiller mode of control. In water-side economizer mode (chillers off):

- When comfortably within the heat exchanger capacity (e.g. below 45F), modulate the chilled water pump and condenser pump to maintain 8-10 degF deltaT. This will avoid over-pumping with the symptom of 1- 2 degF delta T. Note that changing from 2F deltaT to 8F deltaT reduces flow by a factor of 4:1.
- When nearing the limit of economizer operation (above 45F OA temp), delta T control will give way to capacity control and the "over pumping" will be needed. This should only be needed above 45F or within 5F of whatever the ultimate tipping point is.
- This sequence requires variable speed drives for the pumps.

Optimized Sequencing of Multiple Chillers/Boilers

Basis of Savings: Efficiency differential of better choice equipment vs. default choice equipment.

- Strategically selecting cut-in and cut-out points to keep the primary equipment operating in its most efficient range. Typically, maintaining this equipment between 50-90% load achieves good efficiency, but verifying the actual best-efficiency points or range for each system, from manufacturer's load profile data, can provide additional savings.
- For most burners, higher levels of excess air are needed at low loads, which lowers combustion efficiency; however, thermal efficiency improves somewhat at low load as heat exchangers become effectively oversized and approach values improved. So the actual sweet spot takes some effort to identify.

Optimized Hot Water Reset from Zone Demand

Basis of Savings: Reduced standby loss of circulating heated fluid.

- This requires polling of individual hot water points of use, for control valve position, such that at least one valve is open 90%. By resetting based on demand, this routine will provide the optimal water temperature (just hot enough).

Optimized Multi-zone Hot Deck/Cold Deck Reset from Zone Demand

Basis of Savings: Reduced simultaneous heating and cooling, which is inherent in this HVAC system type.

- See also **Chapter 24-6, HVAC Retrofits for the Three Worst Systems.**
- This requires polling of individual multi-zone (MZ) mixing damp-

ers to determine the greatest cooling and heating demands, ideally so that at least one zone damper is 90 percent open to cooling and another zone damper is 90 percent open to heating. By resetting hot and cold deck from space demand, optimal heating and cooling (just enough of each) will be provided in the hot and cold decks. Since these systems inherently mix cooled and warmed air, this reduces simultaneous heating and cooling.

- Turning off the boiler or heat source in cooling assists this control measure, but will overlook any zone that legitimately needs heat, leading to comfort complaints.

Dual Duct Terminal Unit: Split Damper and Add Deadband

Basis of Savings: Reduced simultaneous heating and cooling, which is inherent in this HVAC system type.

- Unless constant volume is critical to the air system design, splitting the two dampers for independent control allows optimization.
- With the dampers split (separate actuators), and ventilation air in one or the other air streams (usually the cold duct), control like a VAV box with a deadband. During a call for cooling, the hot duct damper would remain closed while the cold duct damper modulates to maintain temperature. As cooling demand decreases, the cold duct damper throttles toward closed and finally reaches its minimum position (for ventilation). Here it will float within the deadband. If space temperature falls sufficiently to require heating, the hot duct damper would begin to throttle open with the cold duct damper at minimum.
- This sequence minimizes heating-cooling overlap and significantly reduces the energy use of the dual duct system that otherwise has simultaneous heating and cooling waste built into it.
- This control sequence requires converting the supply fans to variable speed control to maintain downstream pressure.
- Turning off the boiler or heat source in cooling assists this control measure, but will overlook any zone that legitimately needs heat, leading to comfort complaints.

Optimized Constant Volume Terminal
Reheat Reset from Zone Demand

Basis of Savings: *Reduced heat-cool overlap (reheat penalty)*

- See also **Chapter 24-6, HVAC Retrofits for the Three Worst Systems.**

This system is characterized as cooling only at the air handler, with all heating coming from the zone reheat coils. Without some form of control to combat it, the inherent energy of the system is high because air volume is constant. In summer, to the extent that the cooling is too much for a given zone, the reheat energy is equal to that over-cooling amount (to negate it). In winter, to the extent that supply air is colder room temperature, a reheat penalty occurs before any heating of the room occurs, equal to heating supply air to room temperature.

- This requires polling of each reheat coil control valve to gage the 0-100% call for heat. The air handler supply air temperature is gradually increased until at least one zone heating valve is no more than 10% open, signaling that a further rise in supply air will risk comfort.
- Turning off the boiler or heat source in cooling season assists this control measure, but will overlook any zone that legitimately needs heat, leading to comfort complaints.

District Heating and District Cooling Delta-T Control

Basis of Savings: Reduced pump horsepower from reduced flow that comes from high differential temperatures. Savings via affinity laws.

- By actively controlling differential temperatures through buildings points of use to be as high has practical, flow can be reduced and pumping costs minimized. This can be touchy and, if too aggressive, can lead to comfort issues.
- Many, if not all, of these distribution systems pump more water than they need to. There are a variety of reasons for this, which are beyond the scope of this report, but many of them can be mitigated by imposing a control limit of minimum delta T (differential temperature) across the coil, building, or segment of the distribution system. This has the effect of requiring the point of use to extract all the available energy out of the circulated fluid before returning it and thus reduce pumping volumes and pumping energy.
- Control is implemented by some manner of throttling device, either a control valve or a pump speed, and supervised by differential temperature measurement and/or flow measurement.
- Applications vary, but the common theme is to wring the heat out of the water and reduce flow when possible, while still maintaining comfort.
- The same action that saves pump energy (mandating a certain del-

ta-T) is also the source of potential comfort issues. For example, if one air handler in the building is marginally sized compared to its load, it may be getting by with some combination of additional flow, low water temperature, or low delta T (aka flooding the coil). If this condition exists, the conservative nature of the bridal loop control will deny the air handler the means to control temperature, leading to comfort complaints. See also "**Enablers**" in this section.

OTHER WAYS TO LEVERAGE DDC CONTROLS

- Utility load tracking and real-time feedback to proposed efficiency changes; early warning of high demand.
- Predictive maintenance for heat exchangers (fouling) from measuring approach temperatures, filter changing (pressure drop), etc.
- Global point sharing that justifies higher quality instrumentation, such as outdoor air temperature, humidity, dew point, wet bulb, etc.

CONTROL SYSTEM CALIBRATION

- For the control system to make good choices, it must have reliable input data. Spot checking of control systems in the field shows a consistent disregard for calibration and large errors in instrument readings. Errors in space temperature of 2-4 degF are common, valves are found at full stroke not quite closed, etc. The notion that once installed these systems can run with no care is false.
- For sustained savings, all analog input and output sensors, transducers should be verified initially and re-calibrated on a 2-year cycle.
- Actuators of valves and dampers should be verified for travel and tight close-off initially and re-calibrated on a 2-year cycle.
- Control set points and occupancy schedules should be reviewed on a 2-year cycle.
- The cycle duration may be extended depending upon the quality of the instrumentation used. If the 2-year calibration shows no issues, then the cycle can be extended to 3-5 years.

Chapter 8

Building Operations and Maintenance

VACANCY

Energy Impact from Vacancy

Vacancy can occur in any facility, but is a common cycle for hotels, schools, office buildings. The effect of vacancy on energy use is elusive, difficult to quantify and will vary by specific case.

For an overall weighted average of all building types, the effect of vacancy can be roughly approximated from the CBECs survey data "vacant" category. Better than nothing, but the most significant message in the number showing for this category is that *energy use is not zero for vacant buildings*.

Considerations for estimating the effect of vacancy:

- What actual effect occurs when a person is/isn't there?
- Is the vacancy uniformly spread out? Or are there defined areas (buildings/wings/floors) that are mothballed during the period of the vacancy?
- When vacancy is in pockets of an otherwise occupied building, what energy use persists from shared systems? HVAC systems designed to service an entire building may or may not have effective provisions for segregating floors or areas from the balance of the building; return air plenums commonly 'bridge' occupied and unoccupied areas where they are used, tending to equalize spaces or create midway values.
- In the case of an office building or hotel, an unoccupied room usually has the lights turned out, plug loads removed or turned off, and the HVAC turned off other than basic freeze protection. In this case, lighting and envelope loads are proportionally less. However, envelope loads only apply to areas with a perimeter location and, furthermore, there is a measure of loss between cold or hot interior spaces and adjacent interior spaces, that will subtract from the predicted proportional savings. One approximation is to use half of the predicted envelope savings as being lost to interior space equalization.
- In some cases, each person has a predictable attribute of equipment

loads, e.g. computers and task lighting, which can be subtracted directly in proportion to the number of people 'vacant'.

- Is the ventilation reduced proportionally to changes in occupancy? Only if yes is this portion of load is proportional. HVAC ventilation controlled by demand-controlled ventilation (DCV) will detect the reduction in people and adjust, but conventional automatic controls will not unless actively changed.
- The energy consumed by the associated appliances and the heat released by people is an internal HVAC load that adds to summer cooling and subtracts from winter heat burden. So, having less occupancy is a give and take affair. Note that a change in internal loads and a change in envelope loads from cordoning off areas and lowering temperatures changes the building balance temperature. When vacancy (less people) is distributed in a building and no areas are cordoned off, *the balance temperature will rise. In this case it is common to see a reduction in electric use from partial vacancy, and simultaneously a rise in heating energy, from the reduction in internal loads.*
- In some cases, such as common amenities, there is a substantial base load energy use whether people use the amenity or not; e.g. swimming pool, restaurant (cooking equipment on, meals or not). Here, vacancy has little or no impact on energy use.

Improving the Correlation between Vacancy and Energy Use

Vacancy is one respect where buildings and processes have the most favorable ratio of energy expense per unit of 'machine' production when they are near maximum. All things equal, energy use will proportion with output or occupancy, but unfortunately all things are not equal. When downsizing, or in cyclical times of temporary vacancy, businesses would like to see a reduction in energy use and cost. Some general concepts can assist in business choices, such as whether to consolidate 3 buildings into 2. Exact quantities require case-by-case analysis. A rough estimate can be made using proportions of the building that are vacant with the CBECs 'vacant' energy use intensity category. The facility can also be modeled with an hourly analysis program to find the effects of internal loads and balance temperature.

- Vacating entire buildings has advantage over vacant floors, and vacant floors has advantage over vacant wings. The differences are less when HVAC systems are distinctly separated, no return air plenums are used and thorough physical separation of areas exists.

Differences are more when there is a large building surface area-to-floor area ratio, such as low rise and convoluted shape perimeters, and where envelope insulation levels are low.

Note that when a perimeter area is vacated, it attempts to become a 'vestibule' with a mid-range temperature, but the interior walls of the 'vestibule' are not insulated to envelope levels. So, while the skin loss for the vacated room has become less, the heat transfer to the adjacent occupied non-vacated area has increased. There is some insulating property of an interior wall, so the effect is not fully negated.

- Distributed vacancy (e.g. every other cubicle) has the least impact on energy use because common area energy such as overhead lighting and ventilation does not change.
- Long term vacancy has advantage over short term vacancy for heating and cooling. This varies according to building mass, but all buildings will have a flywheel effect. When a building is unoccupied and HVAC is turned off, the temperature does not change instantly.
- Thermally, isolating the vacancy to an entire building and for a long time has the greatest reduction and are seen to closely match the levels identified by CBECs 'vacant' energy use category. The true vacant building will probably have a minimal amount of heating (freeze protection, say 50F), emergency lights, and an occasional plug load (like a fire alarm panel). To the extent that the building is not fully isolated, the energy use will be a mid-range value between CBECs 'vacant' and normal occupied values.

Action items for encouraging energy savings from vacancy

- General: Creative marketing. Curb standby losses, idling losses associated with underutilized equipment.
- Offices: Sub-lease vacant areas.
- Hotels: For patrons, eliminate buildings or floors from availability if possible. For individual rooms, turn off all lights, turn off or set back HVAC
- Restaurants: Reduce inventory levels. Consider seasonal menu offerings. Use portions of the equipment (one stove instead of two, one oven instead of two, etc.), use a reach-in cooler or freezer instead of walk-in. Only run dishwasher loads when full.
- Schools (between semesters, K-12 traditional 9-month schedules): Charge for operational cost of special events, re-locate summer fac-

ulty area to avoid a large number of classrooms occupied by a single person, lower ventilation rates (these are normally very high in schools)
- Recreation facilities: Alternate uses for a facility (ice rink in slow season), close outdoor pools in winter, allowing warm-up time for features like saunas vs. always on 'in case someone comes'.

CLOSING A FACILITY FOR PART OF A WEEK

Changing operation from 5 to 4 days per week is sometimes employed to reduce operating costs. Savings are not proportional to the closed days because most buildings are not fully 'un plugged' from the utilities and uses energy even without people.

One government agency tried this and measured savings over several months in the Denver area and reported a 6% savings for all utilities by changing from (5) 8-hour days to (4) 10-hour days and closing the facilities on Friday. If fully linear, the savings for a building "off" day would be 1/7th of operating cost or 14% reduction.

Factors influencing how much will be saved include the following. These are the 'phantom loads' for a building that is unoccupied.

- Total hours of occupancy (in this example it was the same)
- If the extra 'off' day connects to a weekend or does not (longer set-back period helps)
- Amount of night set-up/set-back allowed by automatic controls (more is better, but freeze concerns limit it)
- Persistent operation of equipment such as HVAC zone terminals, circulators, block heaters, basin heaters, plug loads, water heaters, appliances, domestic booster and drainage pumps, and idling computer equipment even when no people are present.
- Exterior lights
- Interior emergency lights

CLOSING A PORTION OF A FACILITY FOR EXTENDED PERIODS

Barring freeze protection, savings nearly proportional to square footage are possible when sections are well sealed off, all lights are off, and all plug loads removed. Interior walls and doors are not insulated like exterior walls and doors, and so there will be some heat movement through these partitions. HVAC systems with zoning matching the closed-off area can simply be turned off, however larger HVAC systems require intervention, such as blocking off supply and return vents, return plenum openings, and setting VAV box minimums to zero air flow. Freeze protection is always a concern, either for plumbing pipes or sprinkler pipes, so some minimum temperature needs to be maintained, e.g. 50 degF.

For heating/cooling loads, a quick gage of the effectiveness is to measure the vacant space temperature as it relates to outdoor and indoor temperatures; the closer the vacant space temperature is to outdoors—for 100% of potential savings the vacant space temperature would be equal to outdoor temperature after a period of time. If areas are not sealed off, natural air movement will occur through available openings and savings may be half or less of what they could be.

FACILITY REPAIR COSTS

Setting aside a capital reserve of 2% to 3% of the total cost of the building will usually be adequate to replace major systems (equipment, roof) at the end of their life. Without such planning, many building owners will experience large unplanned expenses around 20-25 years.

MAINTENANCE VALUE

Improved HVAC preventive maintenance resulting directly from technician training has a significant energy savings potential. From a study of nine community colleges in California, savings potential estimates ranged from 6% to 19% of total annual campus energy costs, or $0.09-0.26/SF-yr. [1999 dollars]

Source: "Quantifying The Energy Benefits of HVAC Maintenance Training and Preventive Maintenance," *Energy Engineering*; Vol. 96; Issue 2, 1999.

POOR INDOOR COMFORT AND INDOOR AIR QUALITY COSTS

Comfort Productivity Increase

"Impaired Air and Thermal Quality": 1.5%

Source: NEMI, National Energy Management Institute, Productivity Benefits Due to Improved Indoor Air Quality, August 1995, pp. 4-9.

Indoor Air Quality Productivity Increase

Unhealthy Building: 3.5%

("Conditions similar to an SBS building …but with a lower percentage of employees affected.")

Source: NEMI, National Energy Management Institute, Productivity Benefits Due to Improved Indoor Air Quality, August 1995, pp 4-9.

Indoor Air Quality: 6% for SBS/BRI
[Sick Building Syndrome/Building-related Illness]

Source: NEMI, National Energy Management Institute, Productivity Benefits Due to Improved Indoor Air Quality, August 1995, pp-4-9.

To identify a dollar value of improvements to comfort or IAQ, use the appropriate percentage productivity increase with the total productivity benchmark value.

PRODUCTIVITY VALUE

$/SF-yr	Building Use
$13	Assembly
$45	Education
$19	Food Service
$110	Healthcare
$19	Lodging
$23	Mercantile and Services
$97	Office

Figure 8-1. Worker productivity costs $/SF [1995 dollars]

Source: Table data derived from gathered data in the following document: NEMI, National Energy Management Institute, Productivity Benefits Due to Improved Indoor Air Quality, August 1995, pp 4-8, 4-11.

For example, if a 1% productivity increase (from improved comfort or IAQ) is expected for a 25,000 SF office building with a total productivity rate of $97 per SF-yr, the savings would be 0.01*$97*25,000 or approximately $24,250 per year.

NOTE: Claims of productivity increases equated to dollars are often not received well since they are difficult to measure. It is suggested that attempts to quantify these benefits only be used in cases of extremely poor comfort or indoor air quality, when the owner has specifically asked for improvements in these areas. It is also suggested to de-rate the results (e.g. multiply by 0.7 or 0.5) to add further credibility to the claim.

MAINTENANCE ENERGY BENEFITS

Cleaning Evaporator Air Coils

- Up to 20% energy penalty in hp per ton, if badly fouled.
- Retarded heat transfer requires greater differential temperature, and lower suction pressures, with corresponding loss of efficiency and extended run times.

Cleaning Condenser Coils

- Up to 20% energy penalty in hp per ton, if badly fouled.
- Retarded heat transfer requires greater differential temperature, and higher head pressures, with corresponding loss of efficiency and extended run times.

Cleaning Condenser Coils and Evaporator Coils

- Individual efficiency losses of condenser and evaporator are additive if both coils are badly fouled.

	Reduction in Capacity	**Pct. Increase in Energy Use, HP Per Ton**
Dirty Condenser	8.2	20
Dirty Evaporator	18.9	18
Dirty Condenser and Evaporator	25.4	39

Figure 8-2. Effect of Dirty Coils on Energy Use

Data are for a 15-ton reciprocating compressor. Source: *Handbook of Energy Engineering*, 5th Ed, Thumann/Mehta.

Early Filter Change Out

- The average between initial and final pressure drops equal the amount of average fan resistance and horsepower associated with filters. Some customers delay filter changes to the maximum differential pressure (dP) allowed by the filter manufacturer. While this reduces filter costs, it increases fan energy cost. Lowering the change-out point lowers the average dP and reduces fan energy.

NOTE: Results are predictable for VAV systems, where air flow requirements are determined from thermal demand. For constant volume systems, the effect is negligible unless the fans are re-balanced because greater cfm will result from reduced friction.

Proper Refrigerant Charge

- Normal efficiency is maintained within a range of ~ ± 10% of ideal charge.
- Excess refrigerant fills part of the condenser, effectively reducing the heat exchanger size, increasing lift and power.
- Insufficient refrigerant reduces subcooling, lowering efficiency, and can create extended run time from loss of capacity.

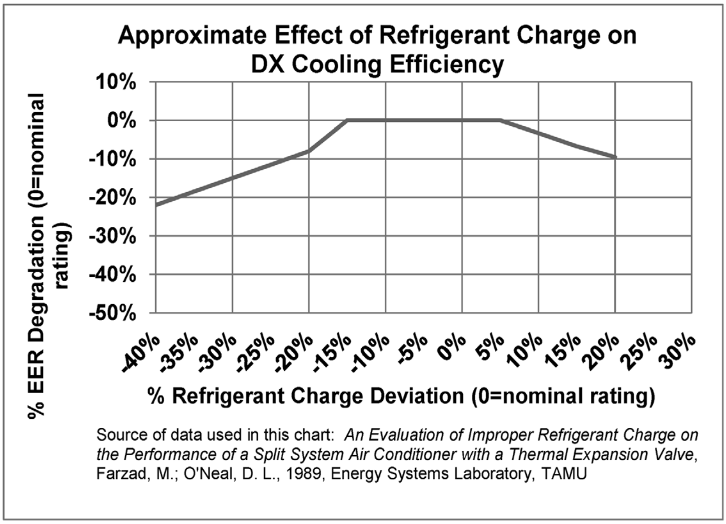


Figure 8-3. Energy Penalty from Refrigerant Charge (over-charge/undercharge)

Cleaning Chiller Condenser Tubes

- Typical savings are 1-1.5% hp per ton improvement for each degree the refrigerant condensing temperature is lowered.
- Retarded heat transfer requires greater differential temperature, and higher head pressures, with corresponding loss of efficiency.

Cleaning Chiller Evaporator Tubes

- Typical savings are 1-1.5% hp per ton improvement for each degree

the refrigerant suction temperature is raised.
- Retarded heat transfer requires greater differential temperature, and lower suction pressures, with corresponding loss of efficiency.
- Most chilled water systems are "sealed" and fouling is minimal. This measure applies mostly to un-sealed systems or possibly to a sealed system with chronic leaks and long term use of make-up water.

Cleaning Chiller Condenser Tubes and Also Evaporator Tubes
- Individual efficiency losses of evaporator and condenser tubes are additive if both tubes are badly fouled.

Cleaning Boiler Fire Tubes
- Savings is a function of soot thickness.
- Retarded heat transfer requires greater differential temperature, and higher combustion temperature, resulting in higher stack temperatures, with corresponding loss of efficiency.

Cleaning Boiler Water Tubes
- Savings is a function of mineral build-up thickness and type of mineral.
- Retarded heat transfer requires greater differential temperature, and higher combustion temperature, resulting in higher stack temperatures, with corresponding loss of efficiency.

Cleaning Boiler Fire Tubes and Water Tubes
- Individual efficiency losses of boiler tubes and fire tubes are additive if both tubes are badly fouled.
- Monitor heat exchange equipment for "approach temperatures."
- By knowing the baseline approach performance, the actual measured approach will infer heat exchange fouling and provide the prompt for predictive maintenance. As the approach temperatures creep up, this usually indicates fouling.
- The very best indication of "clean, new condition" approach is from start-up testing data, or immediately following a thorough cleaning event. As a general starting point, the following are typical values of approach temperatures.
- Note: In some cases, reduced air or water flow through the heat exchanger can cause readings that look like fouling but are not.
- Note: For equipment with variable loads, the approach temperature

will vary somewhat with load. Knowing baseline approach values for 100%, 75%, and 50% load will allow good predictions even at part load.

Losses Due to Soot Build-up in a Boiler

Soot Layer on Heating Surfaces, inches	Increase in Fuel Consumption (%)
1/32	2.5
1/16	4.4
1/8	8.5

Source: American Boiler Manufacturers Association.

Losses Due to Water-side Fouling in a Boiler
(% Increase in Fuel Consumption

	Scale Type		
Scale Thickness, inches	**"Normal"**	**High Iron**	**Iron Plus Silica**
1/64	1.0	1.6	3.5
1/32	2.0	3.1	7.0
3/64	3.0	4.7	—
1/16	3.9	6.2	—

Note: "Normal" scale is usually encountered in low-pressure applications. The high iron and iron plus silica scale composition results from high-pressure service conditions.
*Extracted from National Institute of Standards and Technology, Handbook 115, Supplement 1.
Source: "Clean Boiler Water-side Heat Transfer Surfaces," Industrial Technologies Program (ITP) Steam Tip Sheet #7, December 1999, US DOE Office of Energy Efficiency and Renewable Energy.

Figure 8-4. Boiler Fouling Losses

Notes on Heat Exchangers

The measure of heat exchanger performance is "approach" which generally means the temperature difference that can be achieved by the device to heat or cool some gas or fluid with a supply or ambient gas or fluid. The approach is strongly influenced by heat transfer surface area and turbulence at the boundary layers. For the infinitely sized heat exchanger, the leaving fluid can be an exact match for the ambient fluid supplied, but in practical terms it can only *approach* it. There is some inconsistency in how approach is measured. For this text, the logic is generally **"What you are trying to make" and "with what."**

High Approach Values

Knowing what the approach 'should be' is necessary to evaluate 'what it is' and decide if that is OK or not OK. Approach values are controllable by design and sustained by maintenance. But things change. Load on a heat exchanger may become more or less than it initially was; flows may change; the heat exchanger may be rebuilt and have different properties. A good practice is to identify the value of what approach 'should be' when at its best, e.g. with the system flows balanced and verified and the heat exchanger in new condition. With this piece of information, the goal in operations is to monitor actual approach against the proper value and, when performing maintenance to clean a heat exchanger, return the heat exchanger to the state of performance as when it was new. Monitoring can be as sophisticated as temperature sensors and trend logs with alarms, or as basic as thermometers with marks indicating normal / clean values and additional marks that say 'time to clean'. The practice of deferring heat exchanger maintenance until the equipment cannot perform may incur excess energy usage for extended periods of time.

If the approach temperature appears higher than expected, it can be from any of these or a combination:

- **Fouling**. When one side is fouled, small temperature change will be detected because the exchanger is insulated – here the heat exchanger is doing very little and achieving intended performance will require higher differential temperatures to move the heat across the insulating boundary. In the case of a boiler, a fouled heat exchanger is accompanied by high stack temperature and very heavy fouling is accompanied by a loss of capacity.
- **Insufficient fluid flow**. If half of the water is flowing through a hot water boiler, the stack temperature will be twice its normal value even with a clean heat exchanger. The high stack temperature in this case will be accompanied by a high temperature rise in the water, with the very high leaving water temperature appropriately approaching the exhaust temperature. This is no fault of the heat exchanger.
- **Increased load**. The heat exchanger is asked to do more than it was

designed for. The heat exchanger surface area is too low for the duty and heat transfer can only stabilize at higher differential temperature.

- **Undersized heat exchanger**. When there is not enough surface area, the symptoms are the same as increased load; stable heat exchange can only occur at higher differential temperatures. Marginally sized heat exchangers are often found in lower end equipment where first cost is exchanged for operating expense.

Without specific knowledge of what the approach 'should be', sometimes the only criteria is what is commonly seen in other facilities. This can be reasonable but also can be misleading. For example, an undersized heat exchanger and a fouled heat exchanger behave similarly.

The **shell and tube** heat exchanger in general has a fundamental limitation such that the two leaving fluid temperatures can only approach each other. For most of these types of heat exchangers, the two temperatures can only be economically brought to within 5 or 10 degrees of each other, and 10 degrees F is typical. For special applications of the shell and tube heat exchanger, such as the refrigerant condenser, special baffles can put the liquid refrigerant intimately in contact with the entering fluid for sub-cooling of the liquid after the bulk of the heat transfer has occurred in the main body of the shell.

The **plate-frame** heat exchanger, by its nature, is able to achieve much closer approach temperatures and can actually 'cross' temperatures such that the leaving fluid of one stream approaches the entering (not leaving) temperature of the other stream. This is a special application of the normal concept of "approach" temperatures when discussing heat exchangers, and is also fundamental efficiency advantage that plate frame units have that should be pointed out. Plate frame heat exchangers are also very compact. However there are other considerations of plate frame heat exchangers such as cleaning and first cost. For example, high pressure variations of these units include brazing of the plates to increase pressure ratings (e.g. for refrigeration condensers) which renders them not cleanable at all other than with circulating caustics; for such applications, successful long term operation may require special water filtration which adds cost.

Note: More often, long term operation will experience heat exchanger performance degradation from fouling as a result of a non-cleanable design.

The **fin-tube** heat exchanger is common in air heating and cooling coils (water-to-air) and for some heat recovery devices.

Other heat exchange methods exist that are optimized for the heat recovery media in use, such as heat wheels and drums, and heat pipes.

Heat Exchange Arrangement	Between Where and Where	Typ. Value degF	Diagram
Counter flow Exchanger, General Case, Water or Air	Heating: Hot side in minus cold side out Cooling: Hot side out minus cold side in	---	**Heating: A** **Cooling: B**
Parallel Flow Exchanger, General Case, Water or Air	Heating: Hot side out minus cold side out Cooling: Hot side out minus cold side out	---	**Heating: C** **Cooling: D**
Water-Cooled Condenser (shell and tube)	Saturated condensing temp minus entering condenser water temp. This can be approximated by the liquid temperature	0.5-5	**E**
Water-Chiller Evaporator (shell and tube)	Leaving chilled water temp minus saturated evaporator (suction) temp	0.5-5	**F**
Air-Cooled Condenser	Saturated condensing temp minus entering ambient air temp. This can be approximated by the liquid temperature	20-30	**G**
DX Cooling Coil	Leaving air temp minus saturated evaporator (suction) temp	10-20	**H**
Dry Cooler	Fluid out temp minus entering air temp (ambient)	20-30	**I**
Hot Water Boiler	Flue gas out temp minus leaving hot water temp	75-150	**J**
Fired Water Heater	Flue gas out temp minus leaving hot water temp	20-100	**J**
Fired Steam Boiler	Flue gas out temp minus saturated steam outlet temp	75-150	**K**

Figure 8-5. Heat Exchanger Approach Values
Typical clean values. See diagrams (*Cont'd*)

Heat Exchange Arrangement	Between Where and Where	Typ. Value degF	Diagram
Fired Air Heating Furnace	Flue gas out temp minus leaving air temp	20-100	**L**
Steam Heater	Saturated steam temp minus leaving hot water temp	10-30	**M**
Cooling Tower	Leaving (sump) water temp minus ambient wet bulb temp	7-15	**N**
Fluid Cooler (coil pack)	Leaving process fluid temp minus sump water temp sprayed onto the coil pack	10-20	**P**
Chilled Water Coil – Air Cooling (counterblow multi-row coil, coldest air in contact with coldest water)	Leaving air temp. minus chilled water inlet temp	7-10	**Q**
Hot Water Coil – Air Heating (counterblow multi-row coil, hottest air in contact with hottest water. 50 degF approach is for single row coils)	Hot water supply inlet temp minus leaving air temp	10-50	**R**
Shell and Tube – Heating, Water-to-Water, Hottest Water in the Shell	Shell water inlet temp minus tube water outlet temp	10-20	**S**
Shell and Tube – Heating, Water-to-Water, Hottest Water in the Tubes	Tube water inlet temp minus shell water outlet temp	10-20	**T**
Hot Water Boiler with Return Water Economizer	Flue gas out temp minus exit temperature of the economizer temp	50-100	**U**

Figure 8-5. Heat Exchanger Approach Values
Typical clean values. See diagrams

HEAT EXCHANGER APPROACH DIAGRAMS

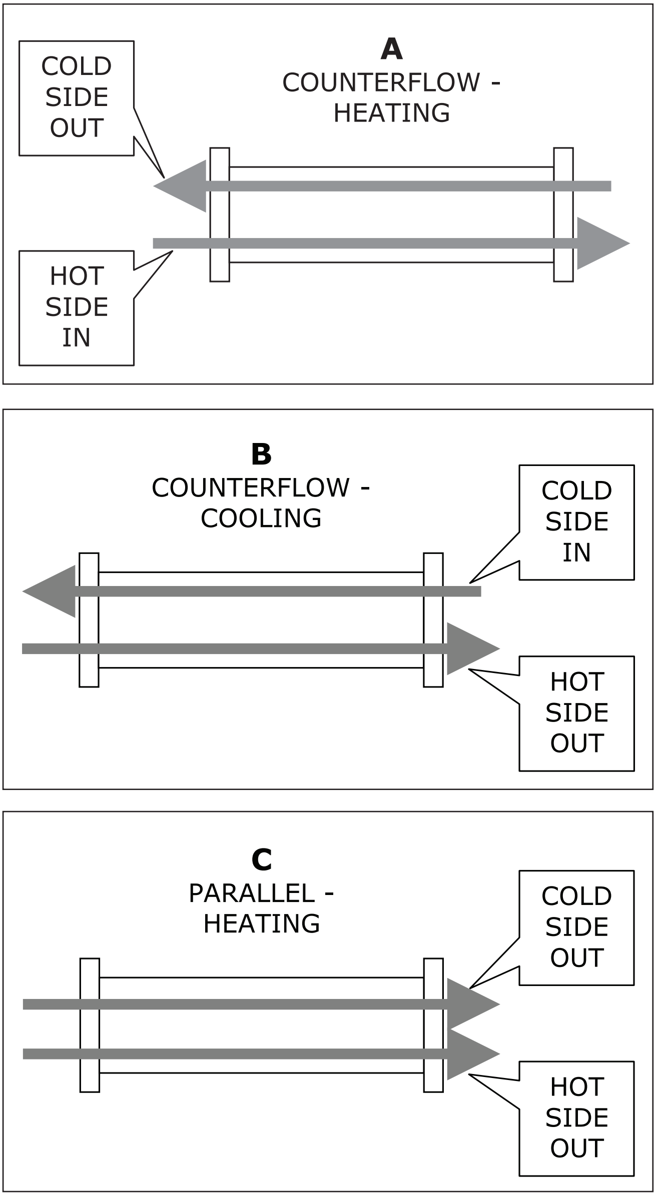


Figure 8-6. Heat Exchanger Approach Diagrams

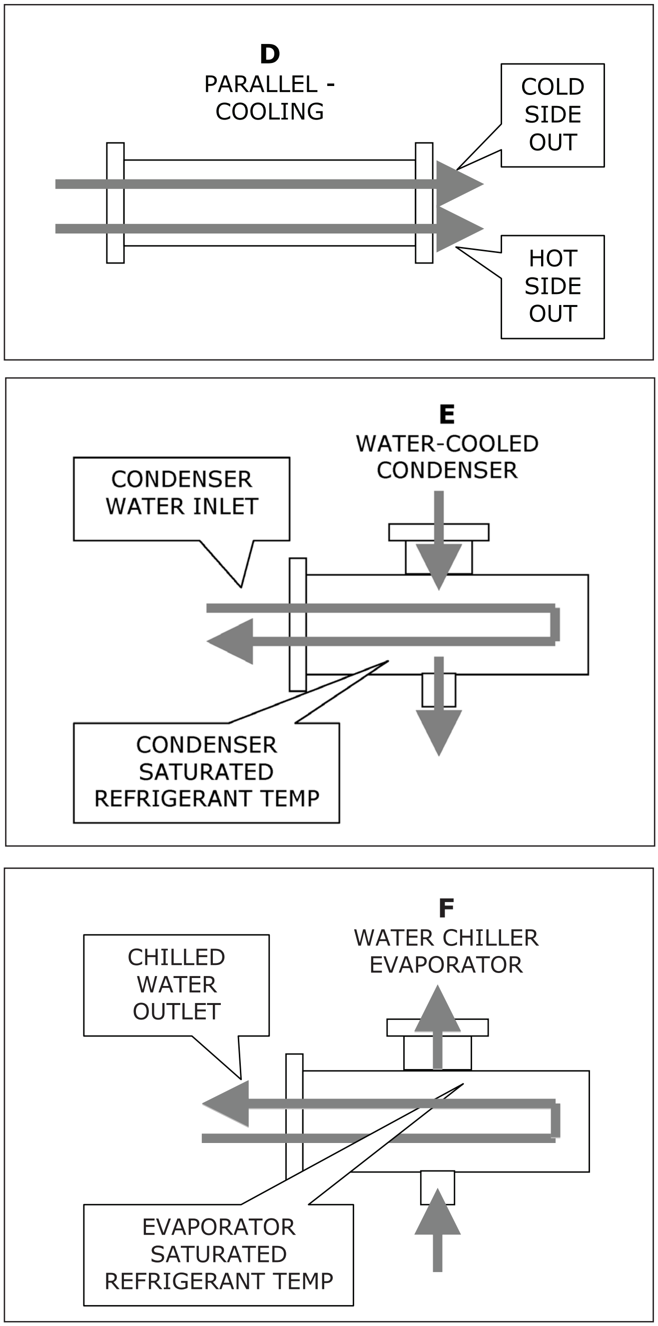


Figure 8-6. Heat Exchanger Approach Diagrams (*Continued*)

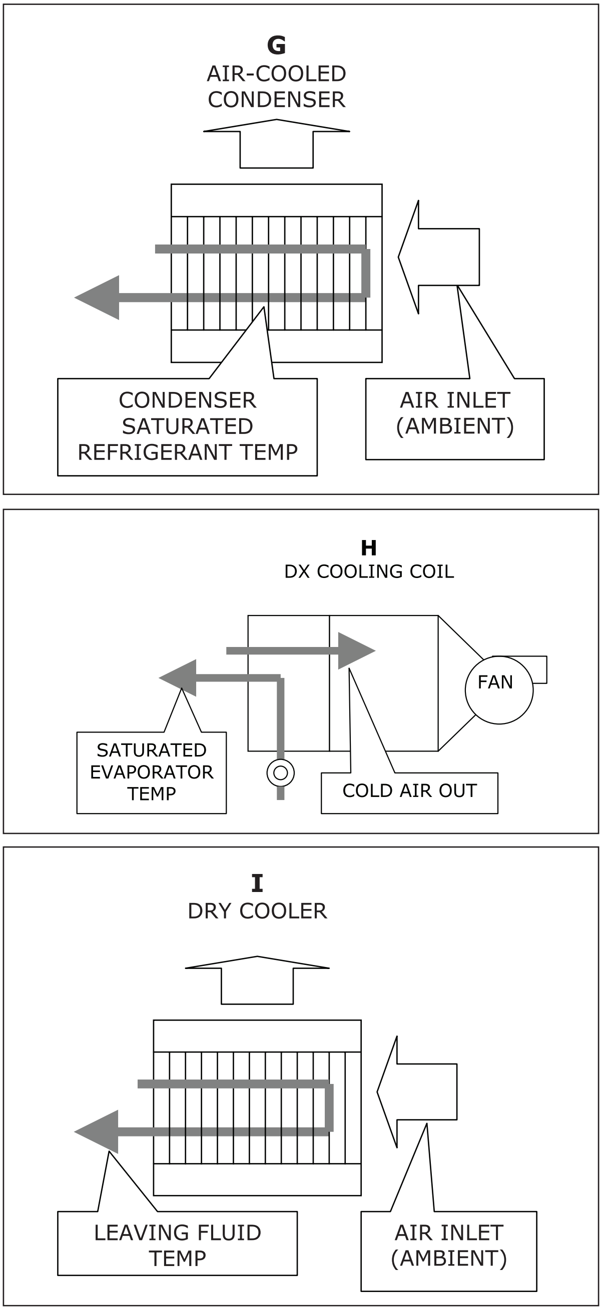


Figure 8-6. Heat Exchanger Approach Diagrams (*Continued*)

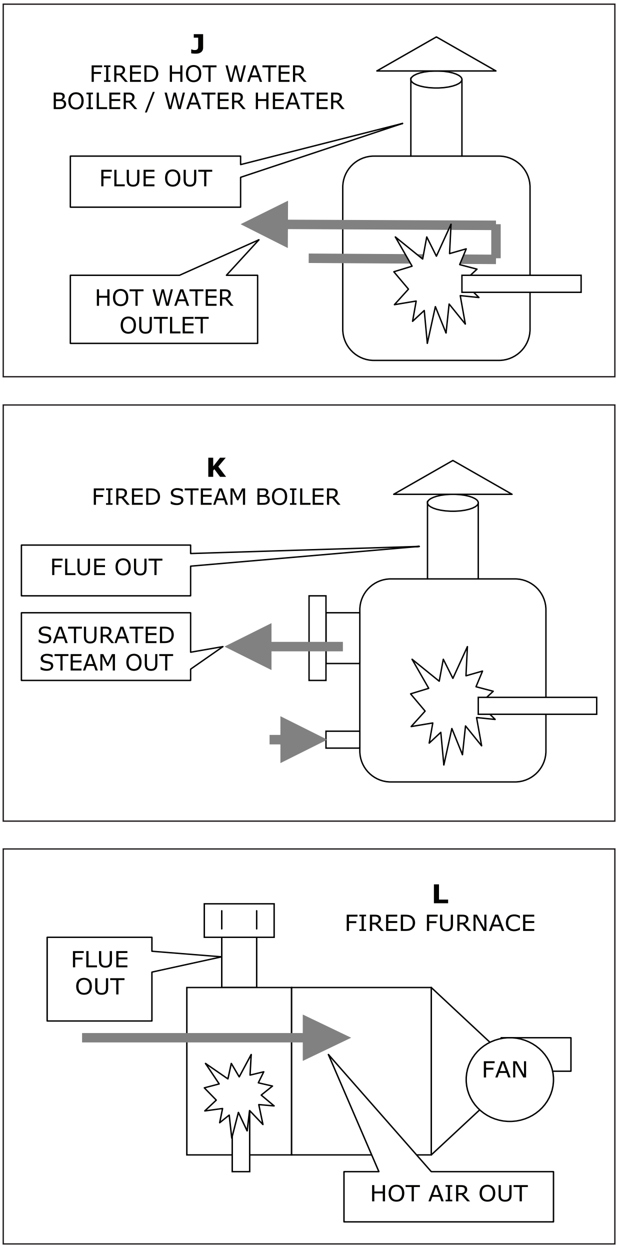


Figure 8-6. Heat Exchanger Approach Diagrams (*Continued*)

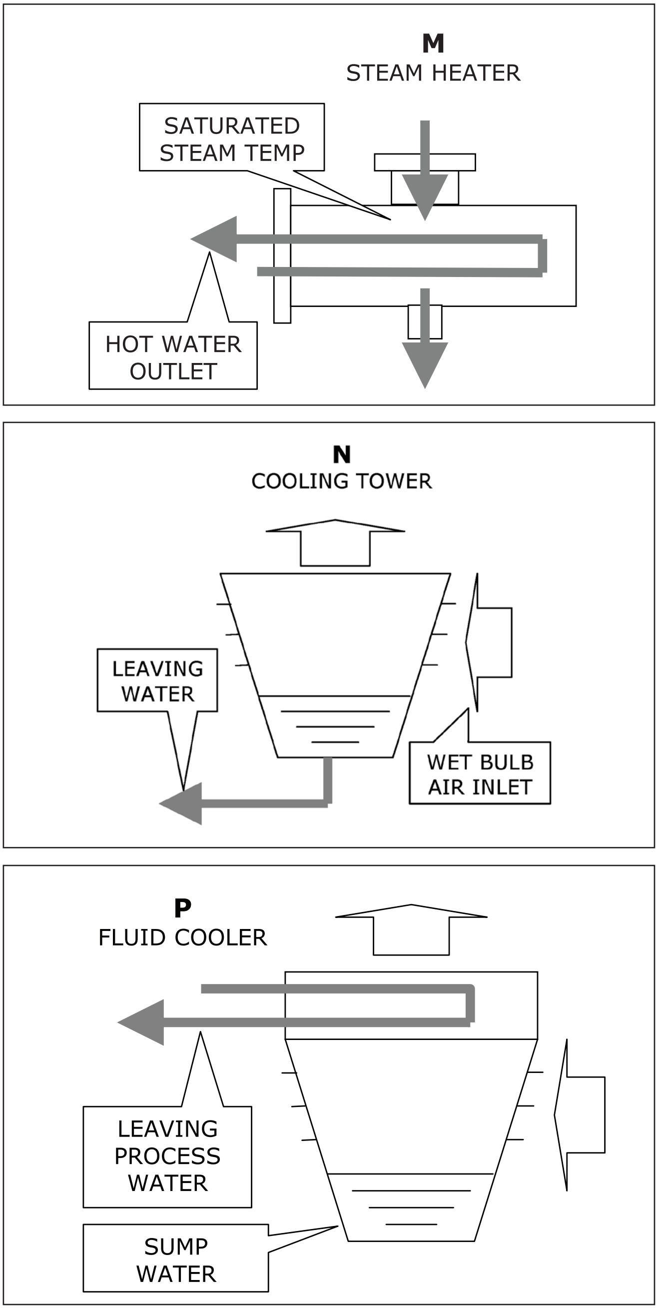


Figure 8-6. Heat Exchanger Approach Diagrams (*Continued*)

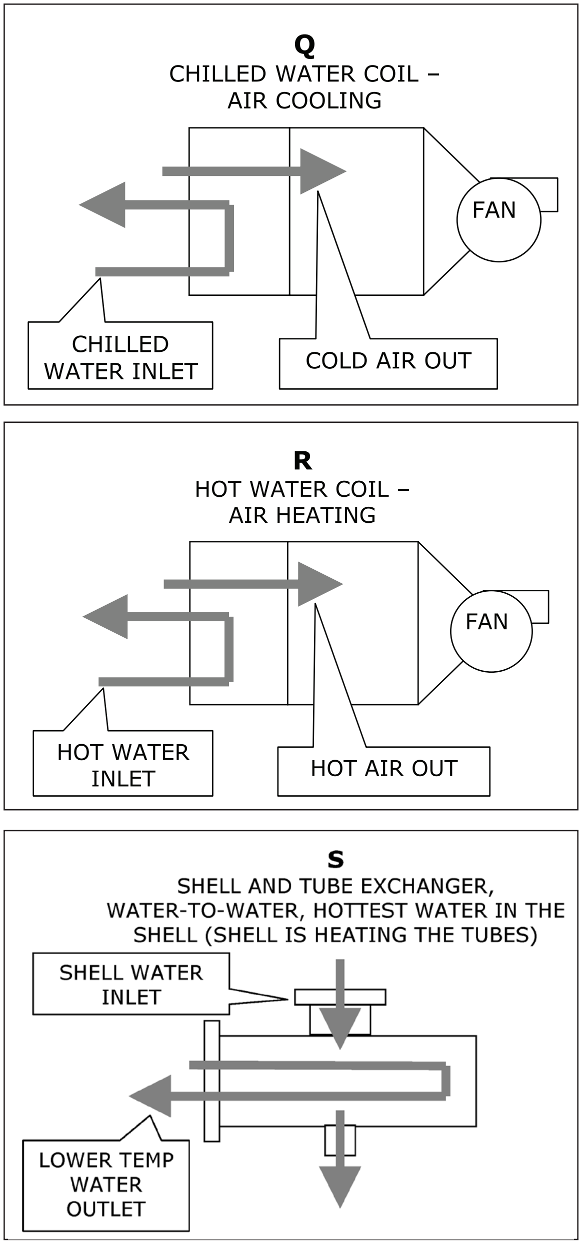
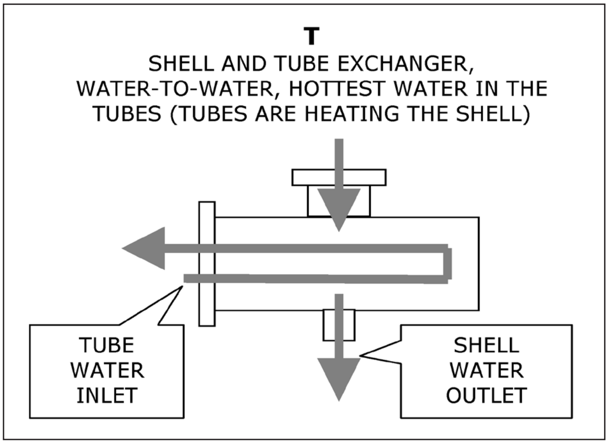


Figure 8-6. Heat Exchanger Approach Diagrams (*Continued*)



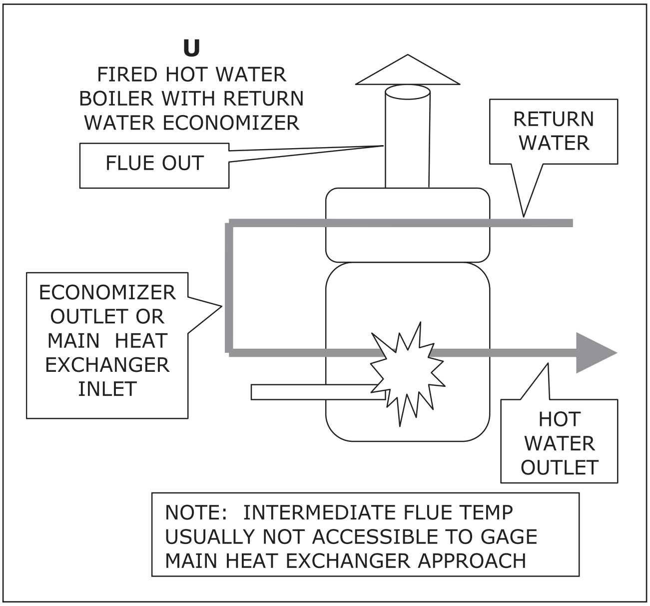


Figure 8-6. Heat Exchanger Approach Diagrams (*Continued*)

Chapter 9

Quantifying Savings

GENERAL

Business decisions to go or not go with energy improvement projects are normally based on simple payback or internal rate of return (IRR). Many other projects compete for available funds and so are ranked by need or by merit. The success of transitioning from the report phase to actual constructive change depends largely on the ability to accurately predict cost and savings.

Computer modeling is commonly provided when guaranteed savings are used or when the dynamics involved make it impractical to estimate any other way. This method has the potential to be the most accurate and flexible for 'what if' scenarios, but is also time-intensive to implement.

In some cases, manual calculations or spreadsheets will do a satisfactory job, and **examples** of those are provided in this chapter.

In other cases, modeling or calculations are not effective, and these may rely on anecdotal Rules of Thumb or be ignored altogether. The more dynamic the process, the harder it is to estimate.

Of course, a barrier 'versus' all of these is cost, and determining the cost of the ECMs is beyond the scope of this text. The driver for most business decisions regarding capital improvements is cost vs. benefit. This section gives some **examples** of how to arrive at the first half of the equation.

Most calculations, including computer models, will include assumptions to simplify the work and consequently will have a plus/minus tolerance that should be understood to be there, regardless of the number of decimal places. On a good day, the calculations will yield results that are close to reality. Few energy estimates are better than +/– 30% and are limited to those with very definable loads and hours (e.g. lighting) and a minimum of uncertainties such as dynamics, heat/cool overlap, measure overlap, and human behavior. Considering performance contracts, the concept of guaranteeing savings where such uncertainties exist gives rise to de-rating savings—who wouldn't?

Understanding the processes at work sufficiently to apply correlations and equations in spreadsheets is necessary before attempting manual calculations. Established estimating modeling software has an advantage here because the research has been done in advance, leaving only application and parameter input to the user.

One source of equations for estimating savings of standard DSM measures is *Vol. 2: Fundamental Equations of Residential and Commercial End Uses*, EPRI, 1993.

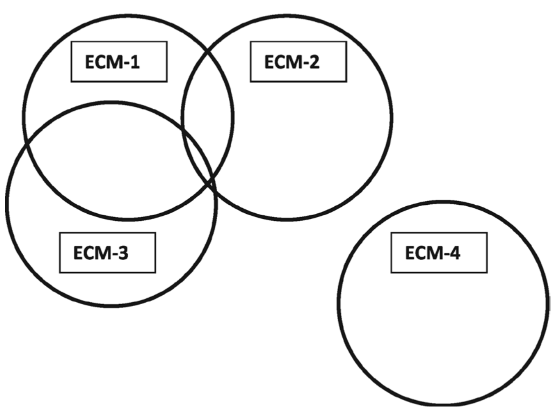
Finally, there are some things that defy quantifying. How these items are treated is up to the reader. Some people ignore them entirely if they can't be quantified, while others rely on anecdotal rules of thumb to claim some savings.

This chapter will provide solved **examples** to illustrate some of the common concepts responsible for ECM savings.

MEASURE INTERACTIONS

Where a list of ECMs is being considered, it is common for customers to be presented with a 'shopping list' of items and the choice of which to pursue. When there are interactions, the first iteration usually presumes each measure could be chosen as the only one. In this case, the model is adjusted so the interactions are isolated and the output represents a stand-alone ECM.

- If multiple ECMs are chosen that include overlaps, additional calculation runs are required to identify the probable energy use of



each scenario or bundle of ECMs. For computer calculations, this takes the form of re-named and "saved-as" scenarios.

- Where interdependencies are large, the measures either need to be presented "with and without" the partner measure, or be discussed in advance with the customer to decide which way to present it.

It is easy to over-state savings when interactions exist. In some cases interacting measures amplify savings, but more often the overlap subtracts.

General Approach to Interacting Measures

A common case of interacting measures is where one measure is conservation-based, using less to begin with, and another is a more efficient source.

If both measures are implemented, the savings are over-stated because:

For the more efficient source measure, the percent improvement [(usage/eff1)—(usage/eff2)] applies to a smaller number when the conservation measure—using less to begin with - is applied. *The conservation measure reduces the magnitude of the usage which in turn reduces the magnitude of savings from the more efficient source.*

When two measures are implemented, one reducing end use and one improving source efficiency:

1. Calculate the reduced usage for the end use first, then

2. Calculate the savings from the higher efficiency source

Example:

Measure 1: More efficient lighting
Measure 2: Reduce lighting
Measure 3: More efficient cooling

Measures are presented in menu form, pending a customer decision on which ones to fund. Independently (if only one is chosen), these may be the correct values.

Measure 1: $10,000 savings
Measure 2: $3,000 savings
Measure 3: $4,000 savings

In reality, measure 1 will have less savings if measure 2 is implemented, since the quantity of light to be made more efficient is now less. Likewise, the cooling benefit of less light (measures 1 or 2) will be overstated if more efficient cooling equipment is installed. Measure 3 will also be proportionally less if either measure 1 or 2 are implemented. If all three were implemented, savings might be:

Measure 1: $8,000 savings
Measure 2: $2,500 savings
Measure 3: $3,000 savings

The most accurate approach is an iterative one, or to present measures in bundles. The very worst approach is one that is unaware of the interactions. Some common interacting ECMs are noted in the **Appendix—Conflicting ECMs and 'Watch Outs'** but this is not a complete list by any means which underscores the need for system knowledge and awareness for the successful energy auditing professional.

In all cases, including the use of computer models, the single most important skill in quantifying energy projects is the understanding of how systems operate alone and with other systems, what things influence energy use and what things can be leveraged to reduce the energy use.

BIN WEATHER USED TO ESTIMATE LOAD PROFILE AND ECM SAVINGS

HVAC load estimating is dynamic. The actual heating and cooling load is an overlay of the varying influences of weather, including moisture content, thermal time constants of materials, solar patterns, occupancy patterns, and internal loads all stirred together to form a rhythm of energy needs unique to that building. This **load profile is then overlaid with the equipment efficiency profile** to determine the energy usage. To arrive at the most accurate value of energy use possible, each building would be evaluated using a detailed computer simulation that was pre-built to consider all these facts. Even with this level of rigor, the day-to-day variables introduce uncertainties.

The bin weather approach you will see in this chapter acknowledges that there are uncertainties anyway, and proposes a short-cut method to produce good approximations in short order.

While convenient and quick, this method has limitations. The assumption shown in many of the solved examples, presumes that the HVAC loads are significantly impacted by outside air conditions; thus, the highest air conditioning demand will occur the hottest day of the year, and the highest heating demand will occur on the coldest day of the year. In the majority of commercial buildings this author has surveyed, this is the case. For thermal transmission and sensible ventilation loads the driver in heat transfer is differential temperature, and so presuming the load varies proportionally with outside air temperature has basis. If the internal loads are consistent, then the only real unknowns are (a) thermal lags, (b) balance temperatures, (c) the influence of humidification and dehumidification requirements, and (d) the occurrence of overlap amongst systems. Other uncertainties, such as user behaviors, are the same for bin weather or any other modeling attempt.

The bin weather method can be used when both load and efficiency can be correlated on weather. Reasonable to use when:

- Both the load profile and system efficiencies are weather-dependent. When this is true, the load and equipment efficiency both vary directly with outdoor dry bulb temperature; e.g., for the hours of 100% load it will be hot outside (with efficiency -x), and partial loads will occur in milder weather (at efficiency -y). This is a similar assumption model for the seasonal efficiency rating of some air-cooled HVAC systems (SEER).
- When the load a function of outside air temperature but the equipment efficiency is not, the method can be used by entering equipment efficiencies manually.

The bin weather method cannot be used unless the load can be correlated on weather. Do not use for:

- Buildings with weather independent load profiles such as a data center.
- Thermally heavy buildings, e.g. those with internal loads that dominate skin loads such that changes in outdoor temperature are poor indicators of heating/cooling loads.
- Interior spaces fully shielded from building envelope effects
- Any other time that outside air changes are poor indicators of heating/cooling loads such as basements or underground facilities.

EQUIPMENT EFFICIENCY PROFILES

The general pattern of energy quantification in this chapter is identifying a load, the hours that load occurs, and the efficiency or efficacy of the equipment serving that load at those times. To apply the equipment efficiency values properly something must be known about the systems and equipment involved. Examples:

- Motor efficiency varies by load, especially at low loads
- A variable cooling load served by a single compressor will require different percentages of power at a given load than an array of smaller chillers that are sequenced and kept, individually, at higher percent machine load.
- A boiler requires more excess air at low output capacity and has an efficiency disadvantage compared to multiple smaller boilers that are sequenced and kept, individually, at higher loads.
- A variable fan load served by an inlet vane will require different percentages of power at a given percent air flow than a VSD, variable pitch propeller, or other modulating methods.
- A variable compressed air load served by an on-off compressor with storage will require different percentages of power for a given percent air flow than a VSD, inlet modulation, load-unload control, or other modulating methods.
- Air-cooled refrigeration cycle equipment is affected by ambient temperatures and will have pronounced changes in efficiency seasonally, with higher efficiencies at lower outside air condensing temperatures. However, this is limited and below some value (40-60F is common) equipment that runs will have 'false load' design provisions to keep the head pressure from falling further; thus the kW/ton at 50F might be the same as 0F.
- A cooling tower's capacity is driven mostly from wet bulb temperature. For a given outlet temperature, there is an approach value between ambient entering wet bulb air temperature and leaving water temperature. The greater the difference, the greater the capacity and, when fan energy is being accounted for, the kW/ton value for the fan varies with changes in the actual approach temperature.

ROUGH ESTIMATING ENVELOPE IMPROVEMENT SAVINGS
See also Chapter 17—BLC Heat Loss Method

A detailed model of the building will estimate this very closely. But it is also possible to make a reasonable estimate to use as a starting spot, when a few things are known:

- Approximate portion of total load that belongs to heating and cooling
- Approximate portion of the heating and cooling energy that is affected by insulation.
- The issue is radiant heat. It travels through glass. A large portion of most cooling loads is from solar. Some portion of the heat loss is through glazing, but a big chunk of heating is infiltration and ventilation.
- The U-value of the roof, walls, and glazing before and after.
- The relative areas of roof, wall, and glazing.

Then, using simple proportions, the order of magnitude benefit can be arrived at. This can allow quick determination if envelope improvements have sufficient merit to evaluate more closely.

From CBECS end use pies……………………	40%	Pct of total energy from heating
Subjective (the other part is infiltration)……	50%	Pct of heating energy affected by insulation
From CBECS end use pies……………………	15%	Pct of total energy from cooling
Subjective (the other part is solar)…………	50%	Pct of cooling energy affected by insulation

EXISTING

	Roof	Wall	Glass	OVERALL WALL	OVERALL ENVELOPE
Area	50%	25%	25%		
R	5.00	5.00	0.92		
U	0.200	0.200	1.087	0.322	0.422

Areas are proportion of total surface area

Overall envelope U
= (Uroof * %roof)
+ (Uwall * %wall)
+(Uglass * %glass)

PROPOSED

	Roof	Wall	Glass	OVERALL WALL	OVERALL ENVELOPE
Area	50%	25%	25%		
R	15.00	5.00	2.50		
U	0.067	0.200	0.400	0.150	0.183

Areas are proportion of total surface area

Overall wall U
=(Uwall * %wall
+Uglass * %glass

Energy Improvement of Glass
=(GlassU Improve)/Old GlassU
=(1.087-0.4)/1.087 = 63% less.
(typ for glass, wall, roof, and overall envelope)

SAVINGS

63%	glass
53%	wall
67%	roof
57%	overall envelope
11%	heating energy
4%	cooling energy
16%	overall building

Heating Energy Savings
=Overall Envelope Savings
% of Energy Used for Heating % of Heat Loss from Insulation
=0.57 * 0.4 * 0.5 = 11% less.
(typ for heating, cooling, and overall envelope)
These chosen envelope improvements will save roughly 11% of building heating energy.

Figure 9-1. Example Method of Rough Estimating Envelope Improvements—putting things into proportion.

ESTABLISHING THE HVAC LOAD PROFILE

This is priority #1 for estimating HVAC energy use. The more accurate the profile, the more accurate the predictions. To exactly determine the profile requires a well calibrated computer model. It is possible to arrive at a reasonably close profile in many cases using a few assumptions. Sample **Figure 9-2** and **Figure 9-3** show a simplified model that assumes:

1. Cooling load is greatest at the highest outdoor temperature
2. Heating load is greatest at the lowest outdoor temperature
3. Cooling load will be zero at some value of outdoor temperature
4. Heating load will be zero at some value of outdoor temperature
5. Heating/Cooling loads will vary linearly between zero and maximum according to outside air temperature.

This simplified model works well when

- Thermal balance temperatures are known (when the chiller is off, when the boiler is off)
- Heating and cooling loads follow outside air dry bulb temperature

Step 1

Establish max heating and cooling loads. This can be done with load calculations or, if you are fortunate enough to be in the facility on the coldest and hottest days, use observed actual values.

Use caution if attempting to calculate heating and cooling loads with a spreadsheet, especially cooling loads. If you do this, validate the output against actual loads. Unlike residential loads where skin loads (envelope) represent up to 70% of heating/cooling energy demand, commercial buildings have varying degrees of internal loads which complicate the estimation of cooling and heating load profile.

See **"Other Sanity Checks"** in this chapter for estimating maximum loads from **check numbers** and **field observations.**

Step 2

Establish balance temperature, which is where the internal heating is equal to the envelope loss; at outside temperatures above this cooling will be needed and at outside temperatures below this heating will be needed. This is not easy and depends upon thermal insulation, thermal mass, percent of glazing, and internal loads. One way to get a general idea is to review control settings and practices to see when boilers and chillers are turned on. Refer to **Chapter 11—Mechanical Systems**, Air-Side Economizer, **"Representative Balance Temperatures for Different Building Activities."**

Step 3

Establish a deadband. Plus/minus (5) degrees F is suggested. There is uncertainty of whether heating or cooling (or neither or both) is going on in this region, so it is common practice to ignore any savings in this range of temperatures. In some buildings, both the heating and cooling operate in this range and so ignoring this area may not capture all of the energy use.

Step 4

Establish minimum heating and cooling temperatures. Minimum cooling (zero cooling) is assumed to be (5) degrees above the balance temperature. Minimum heating (zero heating) is assumed to be (5) degrees below the balance temperature.

*Step 5**

Between minimum and maximum cooling outdoor temperatures, the cooling load can be presumed to vary directly with the outside air temperature. This is a simplified model and ignores swings in humidity and internal loads.

*Step 6**

Between minimum and maximum heating outdoor temperatures, the heating load can be assumed to vary directly with outdoor temperature.

Step 7

Normalize the heating and cooling loads to 0-100%. This is the load curve, with coincident hours. See the sample spreadsheet in **Figure 9-2** and pictorial representation of the concept in **Figure 9-3.**

*In a spreadsheet, the formula to proportion the load between minimum and maximum can take on different forms. Here is one method that uses a value of "gain," calculated separately, to simplify the cell formulas. In this example, the "gain" is (450-0) / (92.5-57.5) = 12.86 tons per degF, the overall tons change for each degree of change in outside air temperature. This defines the slope of the line. The additional logic shown caps the minimum and maximum values (at highest values of OA, load is "max," at lowest values of OA, load is "min."

Logic: If(Toa>=Tmax,LoadMax,elseif(Toa<=Tmin,LoadMin,(((Toa-Tmin)*gain)+Qmin)))

In words:

If OA is above the high limit, use the highest listed load
If OA is below the low limit, use the lowest listed load
If OA is between the two, proportion

LoadMax	450	tons	Max cooling load	
Tmax	92.5	degF	at this OA temp	
				Example Gain for Cooling Load Profile
LoadMin	0	tons	Min cooling load	
Tmin	57.5	degF	at this OA temp	
12.86		tons/degOA	gain	

LINEARIZING METHOD

To linearize Cooling load between max and min loads using bin temperatures.

Cell formula:
((B-C) / (A-C)) * Max Load

This example:
(82.5-57.5) / (92.5-57.5) * 250
= 179

To linearize Heating load between max and min loads using bin temperatures.

Cell formula:
((Z-Y) / (Z-X)) * Max Load

This example:
(52.5-27.5) / (52.5-[-7.5])) *
2000 = 833

	Mid-pts	DB (F)	24x7 bins Total Bin Hours for each value of percent load	Load	Pct Load. This is the load curve	**5 DegF Bins**
A	**92.5**	90 to 95	37	**250**	100%	Max
	87.5	85 to 90	100	214	86%	Clg Load
B	**82.5**	80 to 85	285	**179**	71%	**250**
	77.5	75 to 80	369	143	57%	tons
	72.5	70 to 75	461	107	43%	
	67.5	65 to 70	539	71	29%	
	62.5	60 to 65	865	36	14%	
C	**57.5**	55 to 60	813	**0**	0%	Zero Clg
Z	**52.5**	50 to 55	744	**0**	0%	Zero Htg
	47.5	45 to 50	729	167	8%	
	42.5	40 to 45	657	333	17%	
	37.5	35 to 40	869	500	25%	
	32.5	30 to 35	693	667	33%	
Y	**27.5**	25 to 30	561	**833**	42%	
	22.5	20 to 25	399	1000	50%	Max
	17.5	15 to 20	302	1167	58%	Htg Load
	12.5	10 to 15	134	1333	67%	**2000**
	7.5	5 to 10	95	1500	75%	Mbh
	2.5	0 to 5	81	1667	83%	
	-2.5	-5 to 0	24	1833	92%	
X	**-7.5**	-10 to -5	3	**2000**	100%	

Figure 9-2. Linear HVAC Load Profile based on Temperature
(55 degF Balance Temperature) Bins are for Colorado Springs

Note: Examples in this text use bins in five (5) degree increments to conserve space. It is more customary to use 2 degree interval bins for detailed calculations.

24 X 7 **5 Degree Bins**			Total Bin Hours for each value of percent load	Load	Pct Load. This is the load curve
	Mid-pts	DB (F)			
Observed Maximum	92.5	90 to 95	37	**250**	100%
Cooling Load	87.5	85 to 90	100	214	86%
250 tons	82.5	80 to 85	285	179	71%
	77.5	75 to 80	369	143	57%
	72.5	70 to 75	461	107	43%
	67.5	65 to 70	539	71	29%
	62.5	60 to 65	865	36	14%
CROSS OVER ZONE	57.5	55 to 60	813	**0**	0%
ASSUME NO SAVINGS	52.5	50 to 55	744	**0**	0%
	47.5	45 to 50	729	167	8%
	42.5	40 to 45	657	333	17%
	37.5	35 to 40	869	500	25%
	32.5	30 to 35	693	667	33%
	27.5	25 to 30	561	833	42%
Observed Maximum	22.5	20 to 25	399	1000	50%
Heating Load	17.5	15 to 20	302	1167	58%
2000 Mbh	12.5	10 to 15	134	1333	67%
	7.5	5 to 10	95	1500	75%
	2.5	0 to 5	81	1667	83%
	-2.5	-5 to 0	24	1833	92%
	-7.5	-10 to -5	3	**2000**	100%

Original Spreadsheet shown in Figure 9-2 without the Annotations

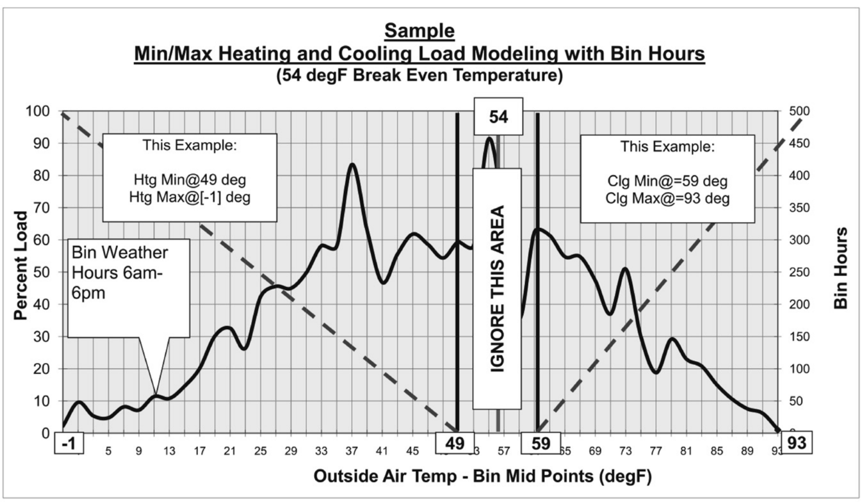


Figure 9-3. Pictorial Representation of Linear HVAC Load Profile
Dashed line shows assumed linear heating and cooling load variation

ADJUSTING THE HVAC LOAD PROFILE FOR HUMID CLIMATES

The basic assumption that temperature drives most of the cooling load is valid in many climates and will produce satisfactory results in predicting a load profile and energy use pattern. Even in humid climates, the highest load will almost always be on the hottest day. But with dehumidification, the load will not drop proportionally as dry bulb temperature drops, but will be more persistent. Thus, a simple dry bulb proportional load profile will understate the cooling load when dehumidification occurs on a significant scale.

However, humid climates will require modifications to this simplified model:

- Adjust the load profile for cooling, if it is sufficiently humid to cause a persistent refrigeration load at moderate temperatures.
- Add air reheating if a reheat dehumidification cycle is used. (Note 1)

The cooling load profile can be modified to accommodate humid climates by including dew point temperature. Beginning with a nominal value of dew point, e.g. 45-50 degF, it can be visualized that the building indoor humidity 'level' will be 'full' at that point and it can be reasoned that any outside air humidity that is higher than this will require some dehumidification. The "level concept" is shown in **Figure 9-4A**.

Step 1. Identify the indoor target dew point temperature.

Step 2. Identify the maximum dew point temperature for the climate.

Step 3. The range of dew point between the indoor target and maximum dew point for the climate represents the range of dehumidification work from 0-100%.

Step 4. Identify the portion of total refrigeration capacity set aside for dehumidification, e.g. 1/3 of the total. This value is termed "sensible heat ratio" (SHR) in HVAC design practice. Example, for a cooling load with a SHR of 0.7, 30% of the total load is applied to dehumidification or "latent" work. This adds to the total cooling load.

The dehumidification load is calculated separately and <u>added</u> to the "sensible" load (the load related simply to temperature). When this is done, the load profile skews toward higher load for the hours of higher dew point. An example of this is shown in **Figure 9-4B** for a humid climate. Compared to a load profile assumed to be linear from dry bulb temperature alone, adjusting for dehumidification in humid climates has the effect of changing the load for each bin, usually in the mid range—see example in **Figure 9-4C though Figure 9-4E**.

The two methods were applied to representative cities in five climate zones to determine the error by not considering the dehumidification effect on the load shape. By using the resulting load profile with before-and-after equipment efficiencies, it was found that errors of up to 20% for the actual magnitude of energy use, but less than 10% error in the differential (savings) results. As always, when extreme accuracy is needed, detailed modeling is required.

Climate Zone	City	Avg. Error in Magnitude	Avg. Error in Differential (savings)
1	Anchorage, AK	1%	1%
2	Colorado Springs, CO	NA	NA
3	Portland, OR	-10%	-7%
4	Atlanta, GA	-12%	-4%
5	Tampa, FL	-14%	-2%

Note 1:

Reheating is sometimes used, sometimes not. When dehumidification loads are light and thermal loads are high (a hot humid day), reheat can occur naturally. When refrigeration-based dehumidification has trouble is mild weather when it is not hot outside but is uncomfortably humid; this is where reheat is needed and without it the psychrometric loop will not close. However, in many cases where psychrometrics indicates reheat is needed, it is not used – either because designs do not acknowledge the need or there is an aversion to the cost of reheat. (Also, there are restrictions to using reheat in energy codes). In any case, on mild temperature / high humidity days when buildings are over-cooled and *not* reheated, space temperatures will become cold and clammy. Based on the normal space temperature, the *absolute* humidity level has been lowered, but at the lower-than-normal space temperature the *relative* humidity remains high. The example calculation provided includes the extra cooling load for dehumidification and applies it when needed, but does not show reheat energy burden. When reheat is to be used, additional provisions are needed in the spreadsheet to understand when it is needed and to quantify the additional energy. Basic reasoning for the added logic:

- For each day with dehumidification, find total load assuming it runs on call for humidity and the sensible load. If the required sensible capacity from dehumidification action is greater than the sensible demand, then over-cooling occurs. When this happens, account for the over-cooling as waste, plus an equal amount of reheat to balance it; i.e. the amount of over-cooling is countered with reheat.

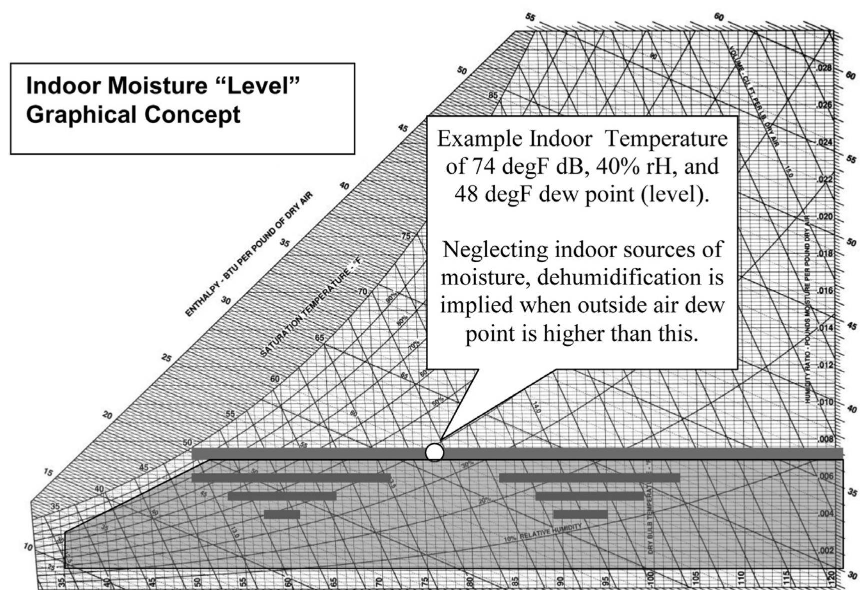


Figure 9-4A. Humidity "Level" Concept for Predicting Humidification and Dehumidification

Dew point range = 69.8-48.0 = 21.8 degF

This example has a maximum of 30 tons available for dehumidification (SHR=0.7)

At 66 degF dew point, dehumidification tons is approximated as:
((66-48)/(69.8/48)) * 30
= (18/21.8) * 30

= **25** tons of dehum.

	Sensible Capacity TONS	Dehum Capacity TONS	Max load TONS
	70	**30**	**100**
DEHUM	69.8	55.0	**48.0**
	max dp for this weather set	Db lockout for cooling operation	DP limit for dehum

Coincident Dew Point (calculated from db/wb)	Total Bin Hours for each value of percent load	Apparent Cooling Load from DB temp	Pct of sensible influence	Pct of dehum influence	xtra tons for dehum	Cooling Load (tons)	Pct Load.
69.8	9	70	100%	100%	30	100	100%
68.4	56	62	88%	94%	28	90	90%
68.2	196	54	76%	93%	28	81	81%
66	758	45	65%	83%	**25**	70	70%
64.9	768	37	53%	78%	23	60	60%
64	1314	29	41%	73%	22	51	51%
59.5	885	21	29%	53%	16	36	36%
53.8	1027	12	18%	27%	8	20	20%
48.7	790	4	6%	3%	1	5	5%

Figure 9-4B. Modified HVAC Load Profile—Humid Climate

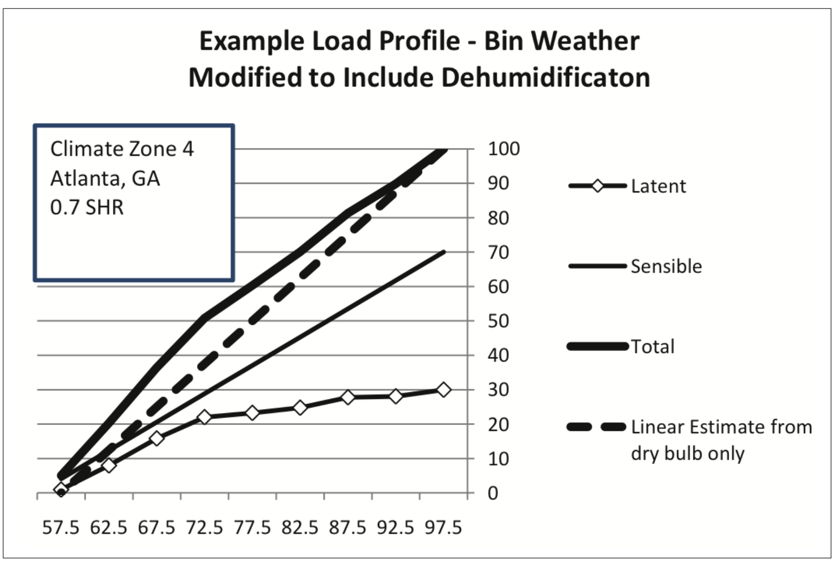


Figure 9-4C. Modified HVAC Load Profile—Humid Climate (Graphical)

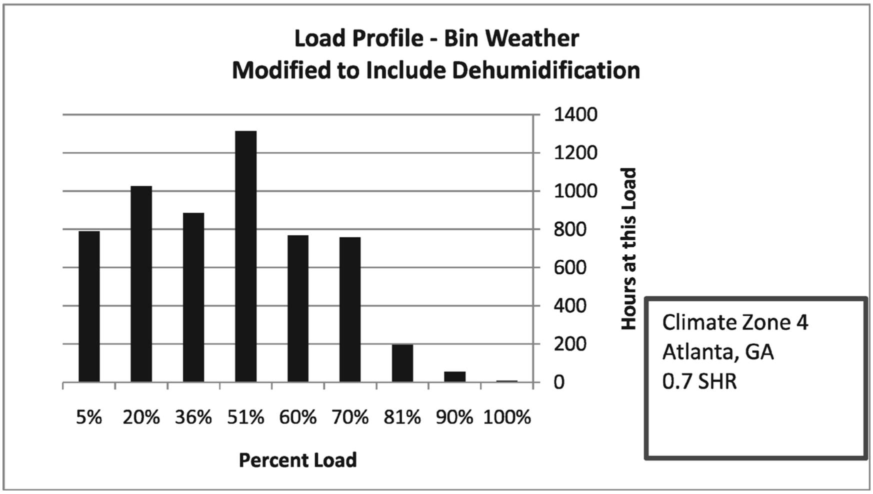


Figure 9-4D. Example Load Profile—Humid Climate with Combined Effect of Sensible and Latent Loads.

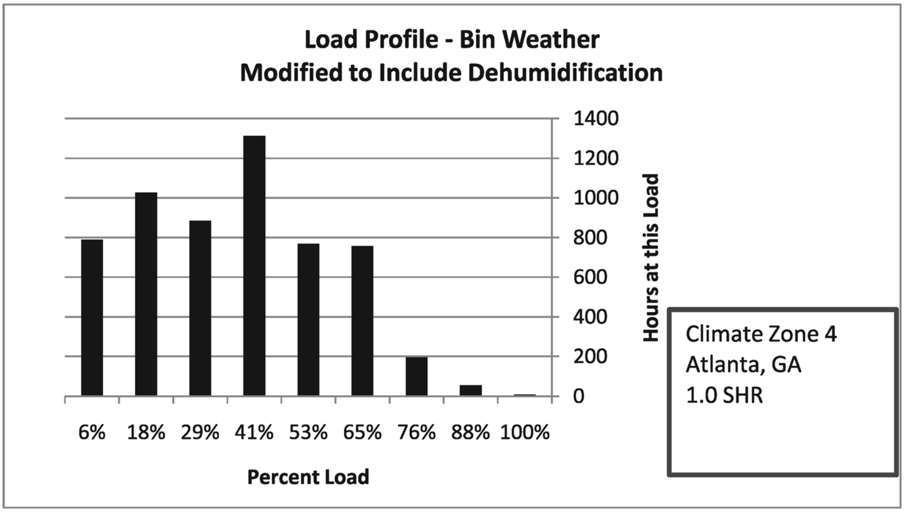


Figure 9-4E. Example Load Profile—Humid Climate Assumed Linear from Dry Bulb Temperature.

Bin hours are the same, but the loads at those hours are different.

ADJUSTING THE HVAC LOAD PROFILE FOR OVERLAPPING HEATING AND COOLING

The load profile zone between heating and cooling is usually ignored due to uncertainty of what exactly is going on. Air economizers and temperature resets are active in this area, mild temperatures may have windows open and equipment off, etc. *If you are sure* there is consistent overlap (simultaneous heating and cooling), the load profile can be adjusted to show it. Linearizing from temperature in the overlap zone will overstate the amount of overlap, so a subjective factor for minimum persistent load is added to quantify the heat/cool loads even when weather does not dictate. **Figure 9-6** shows this method and the results are graphed in **Figure 9-5**.

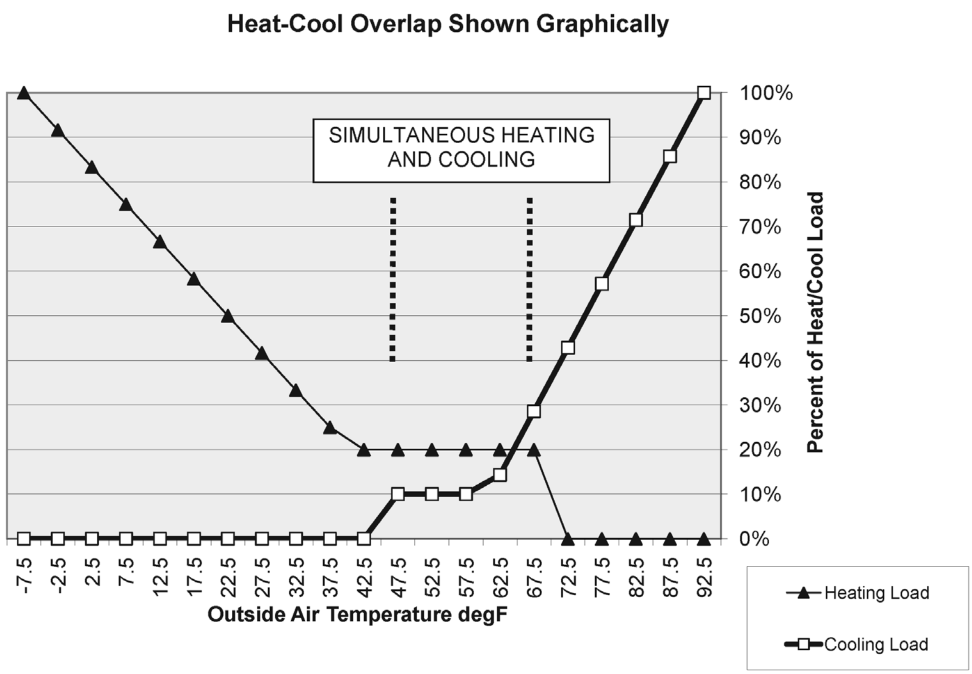


Figure 9-5. Modified HVAC Load Profile—Heat-Cool Overlap

OVERLAPPING HEATING AND COOLING EXAMPLE

24x7 bins

		Mid-pts	DB (F)	Total Bin Hours for each value of percent load	Cooling Load from Weather	Min Clg load during ovelap period	Clg Load	Pct Clg Load. This is the cooling load curve	Heating Load from Weather	Min Htg load during ovelap period	Htg Load	Pct Htg Load. This is the heating load curve	
Min	Max	92.5	90 to 95	37	**250**	25	250	100%	0	0	0	0%	
Clg Load	Clg Load	87.5	85 to 90	100	214	25	214	86%	0	0	0	0%	
10%	**250**	82.5	80 to 85	285	179	25	179	71%	0	0	0	0%	
in overlap	tons	77.5	75 to 80	369	143	25	143	57%	0	0	0	0%	
zone		72.5	70 to 75	461	107	25	107	43%	0	**0**	0	0%	OVERLAP
	Zero Htg	67.5	65 to 70	539	71	25	71	29%	0	400	400	***20%***	
		62.5	60 to 65	865	36	25	36	14%	0	400	400	***20%***	
	Weather zero for Clg	**57.5**	55 to 60	813	**0**	25	25	***10%***	0	400	400	***20%***	
	Weather zero for Htg	**52.5**	50 to 55	744	0	25	25	***10%***	**0**	400	400	***20%***	
		47.5	45 to 50	729	0	25	25	***10%***	167	400	400	***20%***	
	Zero Clg	42.5	40 to 45	657	0	**0**	0	0%	333	400	400	***20%***	
		37.5	35 to 40	869	0	0	0	0%	500	400	500	25%	
		32.5	30 to 35	693	0	0	0	0%	667	400	667	33%	
		27.5	25 to 30	561	0	0	0	0%	833	400	833	42%	
Min	Max	22.5	20 to 25	399	0	0	0	0%	1000	400	1000	50%	
Htg Load	Htg Load	17.5	15 to 20	302	0	0	0	0%	1167	400	1167	58%	
20%	**2000**	12.5	10 to 15	134	0	0	0	0%	1333	400	1333	67%	
in overlap	Mbh	7.5	5 to 10	95	0	0	0	0%	1500	400	1500	75%	
zone		2.5	0 to 5	81	0	0	0	0%	1667	400	1667	83%	
		-2.5	-5 to 0	24	0	0	0	0%	1833	400	1833	92%	
		-7.5	-10 to -5	3	0	0	0	0%	**2000**	400	2000	100%	

Figure 9-6. Modified HVAC Load Profile—Heat/Cool Overlap

SAMPLE SAVINGS CALCULATIONS

LOAD-FOLLOWING AIR AND WATER FLOWS VS. CONSTANT FLOW (VSD BENEFIT)

SUPPLY AIR RESET VS. REHEAT – CONSTANT VOLUME

SUPPLY AIR RESET WITH VAV VS. INCREASED FAN ENERGY

CONDENSER WATER RESET VS. CONSTANT TEMPERATURE

CHILLED WATER RESET FOR VARIABLE PUMPING VS. INCREASED PUMP ENERGY

WATER-SIDE ECONOMIZER VS. CHILLER COOLING

HIGHER EFFICIENCY LIGHTING VS. EXISTING LIGHTING

HIGHER EFFICIENCY MOTORS VS. EXISTING MOTORS

HIGHER EFFICIENCY CHILLER VS. EXISTING CHILLER

HIGHER EFFICIENCY BOILER VS. EXISTING BOILER

HOT WATER RESET FROM OUTSIDE AIR VS. CONSTANT TEMPERATURE

REDUCE AIR SYSTEM FRICTION LOSSES – CONSTANT VOLUME

LOAD-FOLLOWING AIR AND WATER FLOWS VS. CONSTANT FLOW (VSD BENEFIT)

Basis of Savings: Reduced energy transport power (fan and pump power)

Assume the flow rate tracks the load for 50-100% load. Below 50% load it may not track directly since factors like minimum air flows and laminar coil conditions may begin to dominate.

This can be for any load following scenario. For illustration, HVAC load following will be used, i.e. as the heating or cooling load decreases, the air flow and/or water flow decreases proportionally.

Refer to **Figure 9-3**. The 100% cooling load condition represents the constant volume energy. The reduced percent load dashed line represents the fan or pump load-following potential improvement and energy reduction from using variable flow technology.

With load following, power consumption for any bin is the air [or water] Hp transport energy:

Fan power:

= CFM * TSP * Fa/(6356 * fan eff * drive eff * motor eff)

Where CFM= cubic feet per minute air flow and

TSP = total static pressure, in. w.c.

Fa = density correction factor from altitude or elevated temperature

Pump Power:

= GPM * HEAD * SG/(3960 * pump eff * drive eff * motor eff)

Where GPM = gallons per minute water flow

HEAD = total resisting pressure, ft. w.c.

SG = specific gravity (density correction). Water is generally taken as SG = 1.0.

Refer to **Figure 9-7A, B, C**. First calculate the water/air horsepower for the 100% case which is the constant volume baseline. Then in a parallel column calculate the water/air horsepower for the modulating case. It is suggested to limit the minimum load to 30%.

Reduction in flow for centrifugal devices are matched with reductions in rpm, within a reasonable range, such as down to 30%. Reductions in energy use generally follow the affinity laws, but a suggested conservative method is to modify the affinity law and use the square instead of the cube, as follows:

$$\mathbf{HP2 = HP1 \times (^{N2}/_{N1})^{2.0}} \leftarrow \textbf{use square instead of cube}$$

For most commercial variable flow systems, there is a constant downstream pressure that is maintained, ~1.0 in. w.c. for air or ~50 ft. w.c. for water. In this case, the affinity law only applies to the dynamic (friction) portion of the work which rides on top of the constant pressure work. Here, the system curve is defined as

$$\mathbf{SP_2 = \left[SPx * (^{CFM2}/_{CFM1})^{2.0}\right] + CSP}$$

Where:

CSP = the constant downstream pressure (lift)

SPx = static pressure related to friction = (max pressure – CSP)

See **Chapter 15: *Savings Impact When Controlling to a Constant Downstream Pressure—VAV and Variable Pumping***. This adjustment more closely predicts the actual fan horsepower and will avoid overstating savings.

For each bin, there will be a horsepower (baseline and ECM) and a number of hours, and the remainder of the calculation is straight forward:

Savings = (HP-hrs saved) * kW/Hp * $/kWh

See **Figures 9-7A, B, C.**

Air flow pct set equal to load profile pct load, but only turns down to a pre-defined minimum.

VAV CONVERSION
CONSTANT VOLUME vs. VARIABLE VOLUME

AHU Min air flow	**30%**
Max Air Flow	**30,000**
Zone minimum air flow	**25%**
Fan efficiency	**65%**
Altitude Factor Fa	**1.0**
Motor efficiency	**85%**
Minimum Motor Load	**20%**
CAV TSP at full air flow, in.w.c.	**3.5**
VAV Min SP in.w.c.	**1.0**
VAV TSP at full air flow, in.w.c.	**4.5**
Observed Maximum Cooling Load	**250** tons
Max Heating Load	**2000** mbh

	Mid-pts	DB (F)	Total Bin Hours for each value of percent load (work day bins)	Load	Pct Air Flow Load.
	92.5	90 to 95	36	**250**	100%
	87.5	85 to 90	99	214	86%
	82.5	80 to 85	270	179	71%
	77.5	75 to 80	335	143	57%
	72.5	70 to 75	360	107	43%
	67.5	65 to 70	353	71	30%
	62.5	60 to 65	460	36	30%
CROSS-OVER	57.5	55 to 60	384	**0**	30%
ZONE	52.5	50 to 55	372	**0**	30%
	47.5	45 to 50	380	167	30%
	42.5	40 to 45	325	333	30%
	37.5	35 to 40	398	500	30%
	32.5	30 to 35	295	667	30%
	27.5	25 to 30	245	833	30%
	22.5	20 to 25	153	1000	30%
	17.5	15 to 20	101	1167	30%
	12.5	10 to 15	83	1333	30%
	7.5	5 to 10	51	1500	30%
	2.5	0 to 5	30	1667	30%
	-2.5	-5 to 0	8	1833	30%
	-7.5	-10 to -5	2	**2000**	30%

Cooling set to cut out below 60 degF
Heating set to cut out above 50 degF

Note that the reduced motor load results in an additional benefit in cooling season and and a penalty in the heating season.,
The heating mode increase comes from the excess motor work that was partially heating the building.
In all-heating applications, the savings will be adjusted by the ratio of the cost of heating to the cost of electricity.
In all-cooling applications, there is no heating penalty.

Figure 9-7A. VAV Conversion

Load Following feature of VAV

Using SPx,
the friction portion follows fan laws with reduced flow, but the control pressure does not.

SP=[SPx(cfm2/cfm1)^2] +Csp

HP(air) = CFM * TSP *Fa/ (6356 * fan eff)

for CAV and VAV

Sanity check for existing motor size

SPx 3.5

		CAV Existing	VAV	VAV		CAV Existing	VAV	VAV	VAV
Mid-pts	Air flow CFM	Air System Static Pressure (in.w.c.)	New Full Load Air System Static Pressure	Air System Static Pressure (in.w.c.)	Total Air system SP decrease (in. w.c.)	Calcu-lated Fan HP	Calcu-lated Fan HP	BHP	MOTOR SIZE
92.5	30,000	3.5	4.5	4.5	-1.0	25.4	32.7	38.4	**40**
87.5	25,714	3.5		3.6	-0.1	25.4	22.2		
82.5	21,429	3.5		2.8	0.7	25.4	14.4		
77.5	17,143	3.5		2.1	1.4	25.4	8.9		
72.5	12,857	3.5		1.6	1.9	25.4	8.0		
67.5	9,000	3.5		1.3	2.2	25.4	8.0		
62.5	9,000	3.5		1.3	2.2	25.4	8.0		
57.5	9,000	3.5		1.3	2.2	25.4	8.0		
52.5	9,000	3.5		1.3	2.2	25.4	8.0		
47.5	9,000	3.5		1.3	2.2	25.4	8.0		
42.5	9,000	3.5		1.3	2.2	25.4	8.0		
37.5	9,000	3.5		1.3	2.2	25.4	8.0		
32.5	9,000	3.5		1.3	2.2	25.4	8.0		
27.5	9,000	3.5		1.3	2.2	25.4	8.0		
22.5	9,000	3.5		1.3	2.2	25.4	8.0		
17.5	9,000	3.5		1.3	2.2	25.4	8.0		
12.5	9,000	3.5		1.3	2.2	25.4	8.0		
7.5	9,000	3.5		1.3	2.2	25.4	8.0		
2.5	9,000	3.5		1.3	2.2	25.4	8.0		
-2.5	9,000	3.5		1.3	2.2	25.4	8.0		
-7.5	9,000	3.5		1.3	2.2	25.4	8.0		

Figure 9-7B. VAV Conversion (cont'd)

Energy = power X hours

Power savings = Fan Hp saved X 0.746/fan eff

Fan kWh saved X 3413 /12000 = ton-hours of cooling energy (cooling of the fan heat)

Fan kWh saved X 3413 /100,000 = therms of heating energy now added to the building heating system.

Note: At max load, Hp is higher with VAV

Mid-pts	Fan BHp Reduction	Fan Input Energy Saved, kWh	Added cooling load from fan energy ton-hours	Cooling system efficiency, kW/ton	Net Savings for Cooling, kWh	Total Elec Savings	Heating system eff	Lost heat produced by fan heat, therms	Additional heating burden added to building system
		kWh savings from fan alone 68641			Added Savings: kWh saved 4340	72982		Penalty: Added heat load (therms) 1350	
92.5	-7.3	-229	-65	**0.6**	(39)				
87.5	3.2	277	79	0.6	47				
82.5	11.0	2599	739	0.6	443				
77.5	16.5	4858	1382	0.6	829				
72.5	17.4	5502	1565	0.6	939				
67.5	17.4	5395	1535	0.6	921				
62.5	17.4	7031	2000	0.6	1,200				
57.5	17.4	5869	1669	0.6	1,002				
52.5	17.4	5686	1617	0.6	970				
47.5	17.4	5808	1652	0.6	991		**0.8**	247.8	
42.5	17.4	4967	1413	0.6	848		0.8	211.9	
37.5	17.4	6083	1730	0.6	1,038		0.8	259.5	
32.5	17.4	4509	1282	0.6	769		0.8	192.4	
27.5	17.4	3745	1065	0.6	639		0.8	159.8	
22.5	17.4	2339	665	0.6	399		0.8	99.8	
17.5	17.4	1544	439	0.6	263		0.8	65.9	
12.5	17.4	1269	361	0.6	216		0.8	54.1	
7.5	17.4	780	222	0.6	133		0.8	33.3	
2.5	17.4	459	130	0.6	78		0.8	19.6	
-2.5	17.4	122	35	0.6	21		0.8	5.2	
-7.5	17.4	31	9	0.6	5		0.8	1.3	

Ton hours of fan-cooling energy X kW/ton cooling eff = kWh. Reduced fan heat is a bonus in summer

Figure 9-7C. VAV Conversion (cont'd)

SUPPLY AIR RESET VS. REHEAT—CONSTANT VOLUME

Basis of Savings:

1. Refrigeration system savings in cooling mode
   1-1.5 percent power reduction per deg F raised.

2. Reduced reheat energy penalty in heating mode.
   1.08 * altitude factor * CFM * dT
   where:
   CFM is the heating mode cfm
   dT is the difference between supply air temperature and room temperature (degF)—i.e. the heating penalty incurred before any heating of the room begins.

See **Chapter 21—*Formulas and Conversions, "Air Density Ratios (Altitude Correction Factors*)"**

NOTES for constant volume systems:

- Air flow is constant
- Cooling load follows the cooling load profile
- Heating load follows the heating load profile, but includes additional reheat work from having to warm the supply temperature that is colder than room temperature.
- Heating energy savings dominate cooling savings by an order of magnitude and so the cooling energy savings can generally be neglected.
- Supply air reset in heating mode directly saves heating energy by reducing the reheat penalty. Whenever supply air is delivered below room temperature and heating is required in the room, the heating load for that room is not the total heat required. The supply air must be heated as well.
- However, supply air reset only saves cooling energy if the source cooling unit is reset. Simply resetting the temperature of an air handler while using the same temperature of chilled water does not save cooling energy.
- Cooling below 55 degrees F outside air temperature is normally from air economizer and no refrigeration costs or supply air reset benefits occur.
- Ref **Figures 9-8A, B, C.**

SUPPLY AIR RESET FOR CONSTANT VOLUME REHEAT

Load modulates, but air flow is constant

	Mid-pts	DB (F)	Total Bin Hours for each value of percent load	Load	Heating and Cooling pct load	Air Flow (cfm)
	92.5	90 to 95	37	**250**	100%	**50000**
	87.5	85 to 90	100	214	86%	50000
	82.5	80 to 85	285	153	61%	50000
	77.5	75 to 80	369	87	35%	50000
	72.5	70 to 75	461	37	15%	50000
	67.5	65 to 70	539	11	4%	50000
	62.5	60 to 65	865	2	1%	50000
CROSS OVER	57.5	55 to 60	813	**0**	0%	50000
ZONE	52.5	50 to 55	744	**0**	0%	50000
	47.5	45 to 50	729	92	8%	50000
	42.5	40 to 45	657	183	17%	50000
	37.5	35 to 40	869	275	25%	50000
Altitude Factor 1.0	32.5	30 to 35	693	367	33%	50000
	27.5	25 to 30	561	458	42%	50000
	22.5	20 to 25	399	550	50%	50000
Heating Eff 0.8	17.5	15 to 20	302	642	58%	50000
	12.5	10 to 15	134	733	67%	50000
	7.5	5 to 10	95	825	75%	50000
	2.5	0 to 5	81	917	83%	50000
	-2.5	-5 to 0	24	1008	92%	50000
	-7.5	-10 to -5	3	**1100**	100%	50000

Cooling set to cut out below 60 degF in this example
Boiler set to cut out above 50 degF in this example

Figure 9-8A. Supply Air Reset for Constant Volume Reheat

Without reset

Space temp – supply air temp during heating mode

With reset

Space temp – supply air temp during heating mode

Total Therms Saved 27,623

1.08*CFM * altitude correction*dT/ heating eff/100,000*bin hours

Mid-pts	SA Temp (degF)	SA Temp Reset (degF)	Space Temp (degF)	Htg dT Base (degF)	Htg dT Reset (degF)	Htg dT improvement	Therms saved
92.5	**55**	**55**	**74**				
87.5	55	**55**	74				
82.5	55	**56**	74				
77.5	55	**58**	74				
72.5	55	**60**	74				
67.5	55	**62**	74				
62.5	**55**	**64**	**74**				
57.5	55	64					
52.5	55	64					
47.5	55	64	**70**	15	6	9	4,429
42.5	55	64	70	15	6	9	3,991
37.5	55	64	70	15	6	9	5,279
32.5	55	64	70	15	6	9	4,210
27.5	55	64	70	15	6	9	3,408
22.5	55	64	70	15	6	9	2,424
17.5	55	64	70	15	6	9	1,835
12.5	55	64	70	15	6	9	814
7.5	55	64	70	15	6	9	577
2.5	55	64	70	15	6	9	492
-2.5	55	64	70	15	6	9	146
-7.5	55	64	70	15	6	9	18

Figure 9-8B. Supply Air Reset for Constant Volume Reheat (cont'd)

(SA temp with reset) – (SA temp without reset)

Refrigeration savings are minor. The big savings are from avoided reheat.

1-1.5 pct per degree power reduction by raising refrigeration low side temp. Note that if this is a chilled water air handler and the CHW temp is not raised in conjunction with this ECM there are no cooling savings.

			Total kWh saved 2094
Cooling dT improvement	kW/ton chiller power (just the chiller)	Refrigeration effect per degree	kWh saved
0	**0.6**	**1.25%**	0
0	0.6	1.25%	0
1	0.6	1.25%	327
3	0.6	1.25%	726
5	0.6	1.25%	648
7	0.6	1.25%	303
9	0.6	1.25%	89

Figure 9-8C. Supply Air Reset for Constant Volume Reheat (cont'd)

SUPPLY AIR RESET WITH VAV VS. INCREASED FAN ENERGY

Basis of Savings:

1. Refrigeration system savings in cooling mode. (see NOTES for VAV *Systems)*

   1-1.5 percent power reduction per deg F raised.

2. Reduced reheat energy penalty in heating mode.
   1.08 * altitude factor * CFM * dT
   where:
   CFM is the heating mode cfm
   dT is the difference between supply air temperature and room temperature (deg F)—i.e. the VAV heating penalty incurred before any VAV reheat penalty of the room begins.

NOTES for Variable Air Volume (VAV) systems:

- Cooling load and air flow follows the cooling load profile
- Heating airflow is constant at a minimum setting (fraction) of the cooling cfm
- Heating load follows the heating load profile, but includes a penalty for heating the supply air temperature up to room temperature.
- Heating energy savings dominate, and cooling system savings are negligible. Cooling mode reset benefits are minor, due to the competing effects of increased fan energy.
- The higher the VAV box minimum air flows, the higher the savings.
- **VAV reheat penalty**. Whenever supply air is delivered below room temperature, and heating is required in the room, the heating load for that room is not the total heat required. The supply air must be heated as well. This is the VAV reheat penalty that is built into all single path VAV systems.
- **See Chapter 21—Formulas and Conversions, "Air Density Ratios (Altitude Correction Factors)"**
- Supply air reset in heating mode directly saves heating energy by reducing the reheat penalty.
- Supply air reset only saves cooling energy if the source cooling unit is reset. Simply resetting the temperature of an air handler while using the same temperature of chilled water does not save cooling energy.
- Cooling below 55 degrees F outside air temperature is normally from air economizer and no refrigeration costs or supply air reset benefits occur.

VAV cooling mode reset competing effects:
Air horsepower traded for refrigeration horsepower—VAV systems in cooling mode.

As supply air in cooling mode is reset upward, the chilled water temperature at the central plant can be reset upward and that creates savings from the refrigeration cycle:

For variable volume air systems, the higher supply air temperature results in a need for greater air flow for a given load and so fan horsepower increases and refrigeration work increases (from added fan work). For most reset schedules, especially those based on return air temperature the air penalties are usually higher than the refrigeration savings. Even for a very modest reset schedule based on outside air or percent cooling load, only a small portion of the apparent savings remains after the penalties are subtracted. **For this reason, unless M&V testing shows otherwise, supply air reset for VAV systems should be limited to heating mode operation.**

Air Flow Increase Factor:

The difference between space temperature and supply air temperature determines the amount of air required for a given load, so the factor for VAV air cfm increase is:

VAV Cooling Mode Reset Air Flow Increase Factor

$$= \left[\text{RM} - \text{SA NO RESET} / \text{RM} - \text{SA RESET} \right]$$

where:

RM = room temperature
SA NO RESET = supply air temperature with no reset
SA RESET = supply air temperature with reset

The additional air flow causes additional fan work

VAV cooling mode fan kW increase from reset

$$= 0.746^{*}\ \text{HP1} \times \left(\text{CFM2} / \text{CFM1} \right)^{2.0}$$

where:

HP1 = fan hp with no reset
CFM2 = supply air flow with reset
CFM1 = supply air flow with no reset

The additional fan work ends up as heat in the air stream, increasing refrigeration load. (Any added energy transport loss—more fan heat—in cooling mode compounds itself by increasing refrigeration work.)

VAV cooling mode refrigeration load kW increase from fan kW increase

$$= \textbf{kW increase} \times \textbf{(CH kW/ton)(3413/12,000)}$$

where:

kW increase = fan kW increase from reset

CH kW/ton = chiller efficiency kW/ton

See **Figures 9-9A, B, C.**

(SA air flow without reset) X Air Flow Reset Factor shown on final page of spreadsheet

Fixed SA temp and reset values

Cooling air flow follows load profile

Min air handler turn down in cooling	Max Cooling Load
30%	**250** tons

Min zone turn down in heating	Min system SP in. w.c.
20%	**1.5**

S.P. in. w.c. at full cooling air flow	
4.5	CROSS OVER ZONE

Fan eff
65%

Motor eff	Heating eff
80%	**80%**

Altitude Factor	Max Heating Load
1.0	**2000** mbh

SUPPLY AIR RESET FOR VARIABLE VOLUME REHEAT

Mid-pts	DB (F)	Total Bin Hours for each value of percent load	Load	Pct Heating and Cooling Load.	Air Handler Cooling Air Flow (cfm)	Reset Air Handler Cooling Air Flow (cfm)	Zone Heating Air Flow (cfm)	SA Temp (degF)	SA Temp Reset (degF)
92.5	90 to 95	**37**	**250**	100%	**87000**	87000		**55**	**55**
87.5	85 to 90	**100**	214	86%	74,571	74,571		55	**55**
82.5	80 to 85	**285**	179	71%	62,143	65,595		55	**56**
77.5	75 to 80	**369**	143	57%	49,714	59,036		55	**58**
72.5	70 to 75	**461**	107	43%	37,286	54,495		55	**61**
67.5	65 to 70	**539**	71	29%	26,100	26,100		55	**63**
62.5	60 to 65	**865**	36	14%	26,100	26,100		55	**65**
57.5	55 to 60	**813**	**0**	0%	26,100	26,100	17,400	55	65
52.5	50 to 55	**744**	**0**	0%	26,100	26,100	17,400	55	65
47.5	45 to 50	**729**	167	8%			17,400	55	65
42.5	40 to 45	**657**	333	17%			17,400	55	65
37.5	35 to 40	**869**	500	25%			17,400	55	65
32.5	30 to 35	**693**	667	33%			17,400	55	65
27.5	25 to 30	**561**	833	42%			17,400	55	65
22.5	20 to 25	**399**	1000	50%			17,400	55	65
17.5	15 to 20	**302**	1167	58%			17,400	55	65
12.5	10 to 15	**134**	1333	67%			17,400	55	65
7.5	5 to 10	**95**	1500	75%			17,400	55	65
2.5	0 to 5	**81**	1667	83%			17,400	55	65
-2.5	-5 to 0	**24**	1833	92%			17,400	55	65
-7.5	-10 to -5	**3**	**2000**	100%			17,400	55	65

Cooling set to cut out below 60 degF in this example
Boiler set to cut out above 50 degF in this example
Very low fan loads ignored, and minimum motor load not included in this example

Figure 9-9A. Supply Air Reset for VAV Reheat

(SA temp with reset) – (SA temp without reset)

1-1.5 pct per degree power reduction by raising refrigeration low side temp. Note that if this is a chilled water air handler and the CHW temp is not raised in conjunction with this ECM there are no cooling savings.

1.08*CFM * altitude correction*dT/heating eff/100,000*bin hours

(space temp-supply temp) with/ without reset

(dT) X (pct savings per degree) X (kW/ton) X (tons) bin hours

					Heating: Total Therms Saved 10,681					Refrigeration: Total kWh saved 8417
Mid-pts	Space Temp (degF)	Htg dT Base (degF)	Htg dT Reset (degF)	Htg dT improvement	Therms saved (80%e)		Clg dT improvement	kW/ton chiller power (just the chiller)	Refrigeration effect per degree	Refrigeration kWh saved
92.5	**74**						0	**0.6**	**1.25%**	0
87.5	74						0	0.6	1.25%	0
82.5	74						1	0.6	1.25%	382
77.5	74						3	0.6	1.25%	1186
72.5	74						6	0.6	1.25%	2223
67.5	74						8	0.6	1.25%	2310
62.5	74						10	0.6	1.25%	2317
57.5	74									
52.5	70									
47.5	**70**	15	5	10	1,712					
42.5	70	15	5	10	1,543					
37.5	70	15	5	10	2,041					
32.5	70	15	5	10	1,628					
27.5	70	15	5	10	1,318					
22.5	70	15	5	10	937					
17.5	70	15	5	10	709					
12.5	70	15	5	10	315					
7.5	70	15	5	10	223					
2.5	70	15	5	10	190					
-2.5	70	15	5	10	56					
-7.5	70	15	5	10	7					

Figure 9-9B. Supply Air Reset for VAV Reheat (cont'd)

Air Flow Reset factor = (RM-SA NO RESET) / (RM-SA with RESET)

Initial SP x (CFM2/CFM1) 2 + maintained SP setting CFM1 is the max air flow. CFM 2 is the air flow with/ without reset

((CFM * TSP) / (6356 *Fan eff *motor eff)) X 0.746 kW/hp X bin hours

With/without reset

Refrigeration savings less fan penalty.

Note this is a net loss

COOLING AIR RESET FACTOR = (RM-SA NO RESET) / (RM-SA RESET)					Total kWh baseline fan energy 80569	Total kWh reset fan energy 97277	Fan Penalty Total kWh penalty 16708	Total kWh penalty 2851	Total kWh saved **-11142**
Mid-pts	SA Reset Air Flow Factor (tends to increase air flow and negate VAV savings)	No Reset Air system SP (in. w.c.)	Reset Air system SP (in. w.c.)	Reset Air system SP increase (in. w.c.)	No Reset VAV fan energy	Reset VAV fan energy	Reset Air system kWh fan penalty	Added cooling load from fan energy	Net Savings for Cooling
92.5	1.00	4.5	4.5	0.0	3270	3270	0	0	0
87.5	1.00	4.5	4.5	0.0	7574	7574	0	0	0
82.5	1.06	3.8	4.1	0.3	15174	17123	1949	333	-1900
77.5	1.19	3.0	3.6	0.6	12295	17564	5269	899	-4982
72.5	1.46	2.3	3.3	0.9	9026	18517	9490	1620	-8887
67.5	1.73	1.9	1.9	0.0	6049	6049	0	0	2310
62.5	2.11	1.9	1.9	0.0	9707	9707	0	0	2317
57.5	[illegible]	9	1.9	0.0	9124	9124	0	0	0
52.5	[illegible]	9	1.9	0.0	8349	8349	0	0	0
47.5	[illegible]	5	[illegible]	[illegible]	0	0	0	0	0
42.5	[illegible]	5	[illegible]	[illegible]	0	0	0	[illegible]	[illegible]
37.5	[illegible]	5	[illegible]	[illegible]	0	0	0	[illegible]	[illegible]
32.5	[illegible]	1.5	[illegible]	[illegible]	0	0	0	[illegible]	[illegible]
27.5	3.00	1.5	[illegible]	[illegible]	0	0	0	[illegible]	[illegible]
22.5	3.00	1.5	[illegible]	[illegible]	0	0	0	0	0
17.5	3.00	1.5	1.5	0.0	0	0	0	0	0
12.5	3.00	1.5	1.5	0.0	0	0	0	0	0
7.5	3.00	1.5	1.5	0.0	0	0	0	0	0
2.5	3.00	1.5	1.5	0.0	0	0	0	0	0
-2.5	3.00	1.5	1.5	0.0	0	0	0	0	0
-2.5	3.00	1.5	1.5	0.0	0	0	0	0	0

More air flow, more static pressure.

Consequently, fan energy increase tends to negate any refrigeration savings

Fan energy difference from reset.

(with reset) – (without reset)

Extra fan energy is also an extra cooling load

Figure 9-9C. Supply Air Reset for VAV Reheat (cont'd)

CONDENSER WATER RESET VS. CONSTANT TEMPERATURE

Basis of Savings: Reduced chiller power.

This is a balance between refrigeration power savings and added cooling tower fan power.

The key parameters are refrigeration load and coincident dry bulb and wet bulb temperature. Any apparent savings below 55 deg F dry bulb should be disregarded since the air economizer should be active and chillers locked out for most buildings. Within the occupied hours that are NOT lower than 55 deg F, identifying the refrigeration load and the coincident wet bulb temperature will support the necessary math to identify savings.

Step 1

Using a bin weather program, make the bins of occupied times, with dry bulb and coincident wet bulb temperatures.

Step 2

Block out all bin hours lower than 55 deg F dry bulb temperature.

Step 3

Apply zero cooling load to 55 degrees F and full design load (not nameplate capacity) to the highest bin temperature, and interpolate linearly between to establish the cooling load line.

Step 4

Determine the economical "approach" temperature to operate the cooling tower at. This is normally between 7-12 degrees, and depends upon the cooling tower fan energy per ton. This is a complex process but is made easier by the introduction of a term "Relative Cooling Tower Factor." An **example** is provided.

Step 5

Determine the lowest allowable condenser temperature for the chiller. This varies by manufacturer, but usually ranges from 60-70. Some chillers can operate on colder water, but most cannot.

Step 6

For each row of bin hours, determine the optimal condenser temperature from the relationship "wet bulb + approach, but not less than the equipment low limit." Compare the optimal setting to the base design fixed setting. For hours that the optimizing routine would have condenser temperatures lower than the base design, these are hours where savings will occur.

- For hours when the wet bulb temperature is much lower than the dry bulb temperature, savings for this measure are pronounced and approach the theoretical 1-1.5 percent per degree power reduction. But when the wet bulb depression is close to the economical approach value, the savings are reduced because power is added from cooling tower horsepower to achieve the gains. Lower condenser water reduces chiller energy, but requires additional cooling tower fan horsepower. The lower the temperature, the higher the incremental cost of lowering the water temperature, so there is a balance point below which the overall energy consumption (chiller plus tower) no longer improves and may actually get worse. This dynamic can be approximated by using a "Relative Tower Capacity Factor."

  The relative cooling tower capacity factor concept quantifies the fact that at drier conditions the cooling tower can achieve set point much easier, and that the condenser reset routine will work the cooling towers harder, spending more tower fan energy. Only if the gains in chiller power are greater than the expense of extra tower energy is this routine beneficial.

Relative Cooling Tower Capacity Factor

$$= \left(\text{AVAILABLE APPROACH} / \text{DESIGN APPROACH}\right)$$

where:

AVAILABLE APPROACH = cooling tower set pt—actual wet bulb temperature

DESIGN APPROACH = design cooling tower set pt—design wet bulb temperature

And, cooling tower kW/ton at various conditions

= Tower kW/ton @ design conditions/Cooling Tower Capacity Factor

This will identify the cooling tower fan energy used. Combined with the chiller energy used, this provides the basis of evaluating savings.

Overall kW/ton for each bin of hours
=Chiller kW/ton + cooling tower kW/ton

See **Figure 9-10A, B.**

- While the energy savings from condenser water reset near design conditions may be marginal, most of the chiller hours will be at wet bulb conditions that are more favorable, and this is where the energy savings are attained. The control strategy capitalizes on this by continuously adjusting the operating set point based on ambient wet bulb. For those chiller operating hours when the wet bulb is significantly lower than design and the cooling tower can easily produce colder water with little tower energy penalty, the savings will be much more pronounced and closer to the theoretical 1-1.5% per degree. Of course, the chiller load during these shoulder seasons or overnight periods are usually lower than maximum, so the chiller load profile and coincident wet bulb temperature is key to quantifying savings.

- Refer to the table in **Figure 9-11**. For a chiller with 0.5kW/ton efficiency, paired with a 0.07 kW/ton cooling tower, a proposed 5 degree reduction (from 12 to 7 degrees F approach), would yield *worse* energy consumption than leaving it at 12 degrees F, due to the high cooling tower fan energy penalty. This same example, with all things equal except a 0.04 kW/ton cooling tower, would save 0.35% per degree lower overall cooling energy. This example shows that while operating the cooling tower **at or near design wet bulb conditions, most or all of the theoretical chiller savings from condenser water reset will usually be negated by the added cooling tower energy use, unless the cooling tower is very efficient (0.04 kW/ton or less).**

Condenser Water Reset:
Effect of Cooling Tower Energy Use on Overall Energy Savings

- Remember, without the cooling tower fan energy expense, the savings from the chiller would be 1-1.5% per degree lowered. Note that without an efficient cooling tower, condenser water reset can make overall efficiency worse due to high tower fan penalty.

- The "approach" value is a parameter of the optimized condenser temperature calculation, and should be selected based on the

CONDENSER WATER RESET

This is the balance between chiller savings and fan power cost, and will vary.

Chiller set to cutout below 55 degF in this example

Max Load **400** tons

	Mid- pts	DB (F)	Bin Hrs	co-incident WB (F)	load (tons)	econo-mical approach	Lowest allowed temp
	92.5	90 to 95	**36**	**57.7**	**400**	**9**	**62**
	87.5	85 to 90	**99**	**57.3**	350	9	62
	82.5	80 to 85	**270**	**56.9**	300	9	62
	77.5	75 to 80	**335**	**56**	250	9	62
	72.5	70 to 75	**360**	**54**	200	9	62
	67.5	65 to 70	**353**	**51.4**	150	9	62
	62.5	60 to 65	**460**	**48.5**	100	9	62
	57.5	55 to 60	**384**	**45**	50	9	62
No	52.5	50 to 55	**372**	**40.8**	**0**		
Cooling	47.5	45 to 50	**380**	**37.3**			
Savings	42.5	40 to 45	**325**	**34**			
In	37.5	35 to 40	**398**	**30.8**			
Shaded	32.5	30 to 35	**295**	**27.5**			
Area	27.5	25 to 30	**245**	**23.9**			
(Chiller	22.5	20 to 25	**153**	**20.4**			
OFF)	17.5	15 to 20	**101**	**15.3**			
	12.5	10 to 15	**83**	**10.7**			
	7.5	5 to 10	**51**	**6**			
	2.5	0 to 5	**30**	**1.3**			
	-2.5	-5 to 0	**8**	**-3.5**			
	-7.5	-10 to -5	**2**	**-8**			

Chiller limitations. Check for each chiller manufacturer

Figure 9-10A. Condenser Water Reset

cooling tower in use, for best economy. Suggested values of cooling tower approach for best overall cooling efficiency (kW/ton) are shown in **Figure 9-12**. By using the "ratio" instead of specific combinations, this information can be applied to any combination of chiller and cooling tower. This is the value inserted in the sequence "…optimum cooling tower set point shall be equal to the calculated wet bulb temperature plus approach…," and provides further evidence of the importance of amply sized cooling towers with low fan Hp.

Wet bulb temp + approach, but not less than the chiller's minimum allowable temp

Set point – wet bulb temp, with and without reset

Ratio of actual/design approach with and without reset

Chiller savings from 1-1.5% per degree lowered condenser water with and without reset

Mid- pts	optimum cond. temp	base case fixed cond. temp	cond. temp reduction	No Reset available approach	Reset available approach	design approach	No Reset tower relative capacity factor	Reset tower relative capacity factor	No Reset chiller kW per ton	Reset chiller kW per ton
92.5	66.7	**75**	8.3	17.3	9	**10**	1.73	0.9	**0.60**	0.54
87.5	66.3	75	8.7	17.7	9	10	1.77	0.9	0.60	0.53
82.5	65.9	75	9.1	18.1	9	10	1.81	0.9	0.60	0.53
77.5	65	75	10	19	9	10	1.9	0.9	0.60	0.53
72.5	63	75	12	21	9	10	2.1	0.9	0.60	0.51
67.5	62	75	13	23.6	10.6	10	2.36	1.06	0.60	0.50
62.5	62	75	13	26.5	13.5	10	2.65	1.35	0.60	0.50
57.5	62	75	13	30	17	10	3	1.7	0.60	0.50

Design tower kW/ton divided by the tower capacity factor, with and without reset

Overall effect. Chiller savings vs. extra tower fan power.

								kWh saved 22,183
Mid- pts	tower kW per ton at design	No Reset actual tower kW per ton at load	Reset actual tower kW per ton at load	refrigeration savings per degree	No Reset chiller/tower combined kW/ton	Reset chiller/tower combined kW/ton	savings kW/ton	savings kWh
92.5	**0.05**	0.029	0.056	**1.25%**	0.629	0.593	0.036	513
87.5	0.05	0.028	0.056	1.25%	0.628	0.590	0.038	1315
82.5	0.05	0.028	0.056	1.25%	0.628	0.587	0.040	3266
77.5	0.05	0.026	0.056	1.25%	0.626	0.581	0.046	3832
72.5	0.05	0.024	0.056	1.25%	0.624	0.566	0.058	4194
67.5	0.05	0.021	0.047	1.25%	0.621	0.550	0.072	3787
62.5	0.05	0.019	0.037	1.25%	0.619	0.540	0.079	3649
57.5	0.05	0.017	0.029	1.25%	0.617	0.532	0.085	1627

Figure 9-10B. Condenser Water Reset (cont'd)

Cooling Tower kW/ton	Ratio of kW/ton Chiller to Tower (chiller 0.5 kW/ton)	Overall Cooling Energy Percent Savings per degF Lowered (chiller + cooling tower energy)	
0.03	16.7	0.63%	
0.04	12.5	0.35%	
0.05	10.0	0.08%	better
0.06	8.3	-0.18%	worse
0.07	7.1	-0.42%	
0.08	6.3	-0.66%	
0.09	5.6	-0.88%	
0.10	5.0	-1.10%	

Condenser Water Temperature Lowered from 12 to 7 deg F Approach

Figure 9-11. Cooling Tower Fan Energy vs. Lower Condenser Water Temperature

Chiller kW/ton	Tower kW/ton	Ratio of Chiller/Tower kW/ton	Typical Lowest Economical Tower Approach For Condenser Water Reset, degF
0.5	0.1	5:1	**15.0**
0.5	0.085	6:1	**13.5**
0.5	0.07	7:1	**12.0**
0.5	0.06	8:1	**11.0**
0.5	0.05	10:1	**10.0**
0.5	0.04	12.5:1	**8.5**
0.5	0.03	17:1	**7.0**

Figure 9-12. Cooling Tower Sizing Effect on Optimum Condenser Water Temperature
Amply sized cooling towers are required for achieving chiller savings without high cooling tower fan energy penalties.

CHILLED WATER RESET FOR VARIABLE PUMPING VS. INCREASED PUMP ENERGY

Basis of Savings: Reduced chiller power.

The savings are offset partly by added pump energy. As the load decreases, the water temperature is raised and the variable flow pumping compensates with increased flow.

Refrigeration system savings in cooling mode.

1-1.5 percent power reduction per deg F raised.

Water Flow Rate when load and dT are known

$$\mathbf{GPM = {}^{Load}/_{(500\ *dT)}}$$

Where GPM = gallons per minute water flow

Load = Btuh (design load Btuh * percent capacity)

dT = temperature difference between return and supply temperatures, assuming the return temperature stays the same. i.e. half the dT will require twice the water flow.

Pump Power:

$$\mathbf{Pump\ Power = GPM * HEAD * \left(\frac{1}{3960 * pump\ eff * drive\ eff * motor\ eff}\right)}$$

Where GPM, is gallons per minute water flow and

HEAD is the total resisting pressure, ft. w.c.

What changes in this ECM is the GPM.

Assume the system head pressure remains the same.

The amount of pump penalty depends on the amount of reset, but losing about half of the benefit is typical. This is because as chilled water temperature is raised, additional flow is required for a given load.

$$\mathbf{GPM = \left({}^{TONS\ *\ 12000}/_{500\ *\ dT}\right)}$$

where:

dT = differential temperature between return and supply, degrees F

The increase in chilled water flow from increasing supply temperature, with a fixed return temperature, is therefore directly proportional to the decrease in differential temperature.

Iterations with a spreadsheet can increase the ECM benefit by optimizing the two competing factors.

This acknowledges that flow reduction will track load downwards only to a point. In this case, 40% is assumed.
Flow = (tons x 12,000) / (500 x dT)

Max flow = (450 x 12,000)/(500 x (55-45)) = 1080 gpm

Min flow = 40% x 1080 = 432

VARIABLE FLOW CHILLED WATER WITH TEMPERATURE RESET

Chiller set to cutout below 55 degF in this example

	Mid-Points	Range	Bin Hrs	Load (tons)	Pct Load	No Reset CHW temp	Reset CHW temp	Return Temp	Minimum CHW flow for analysis
	92.5	90 to 95	**36**	**450**	100%	**45**	**45.0**	**55**	432
Max Load	87.5	85 to 90	**99**	394	88%	45	**45.0**	55	432
450	82.5	80 to 85	**270**	338	75%	45	**46.0**	55	432
tons	77.5	75 to 80	**335**	281	63%	45	**47.0**	55	432
	72.5	70 to 75	**360**	225	50%	45	**48.0**	55	432
Min. Linear	67.5	65 to 70	**353**	169	38%	45	**49.0**	55	432
Load	62.5	60 to 65	**460**	113	25%	45	**50.0**	55	432
40%	57.5	55 to 60	**384**	56	13%	45	**51.0**	55	432
No	52.5	50 to 55	372	**0**					
savings	47.5	45 to 50	380						
in the	42.5	40 to 45	325						
shaded	37.5	35 to 40	398						
area	32.5	30 to 35	295						
	27.5	25 to 30	245						
Chiller	22.5	20 to 25	153						
Off	17.5	15 to 20	101						
	12.5	10 to 15	83						
	7.5	5 to 10	51						
	2.5	0 to 5	30						
	-2.5	-5 to 0	8						
	-7.5	-10 to -5	2						

The reset schedule for this example is:

OA	CHWS
85	45
55	51

Figure 9-13A. Variable Chilled Water Flow with Temperature Reset

(Tons x 12,000) / (500 * dT) with and without reset

System pressure turndown is limited, so there is pressure available to areas needing flow.

(Variable flow/Constant flow)2 x design head pressure, with and without reset, but not less than the established minimum pressure. Note that CHW reset adds flow and adds pressure.

Mid-Points	No Reset CHW Flow	Reset CHW Flow	if constant volume	Design max head pressure (ft. w.c.)	minimum head pressure (ft. w.c.)	No Reset head pressure (ft. w.c.)	Reset head pressure (ft. w.c.)	Pump eff	Motor eff
92.5	1080	1080	1080	**80**	**40**	80	80	**75%**	**85%**
87.5	945	945	1080	80	40	61	61	75%	85%
82.5	810	900	1080	80	40	45	56	75%	85%
77.5	675	844	1080	80	40	40	49	75%	85%
72.5	540	771	1080	80	40	40	41	75%	85%
67.5	432	675	1080	80	40	40	40	75%	85%
62.5	432	540	1080	80	40	40	40	75%	85%
57.5	432	432	1080	80	40	40	40	75%	85%

(gpm x head) /

(3960 x e-pump x e-motor)

with/without reset.
this is pump input power

Pump penalty for CHW reset

Chiller savings from CHW reset

Net savings. In this example 3 steps forward and 2 back

	No Reset	Reset	Pump Penalty kWh		No Reset		Reset	Chiller Savings kWh	kWh Saved
			5,140					8,619	3,479
Mid-Points	Pump power	Pump power	Pump energy penalty from reset	Refrig-eration benefit percent per degree	Chiller kW/ton	degF of reset	Chiller kW/ton	Chiller savings kWh	
92.5	34	34	0	**1.25%**	**0.6**	0.0	0.60	0	-
87.5	23	23	0	1.25%	0.6	0.0	0.60	0	-
82.5	14	20	1081	1.25%	0.6	1.0	0.59	683	(398)
77.5	11	16	1406	1.25%	0.6	2.0	0.59	1413	8
72.5	9	12	1052	1.25%	0.6	3.0	0.58	1823	771
67.5	7	11	1014	1.25%	0.6	4.0	0.57	1787	773
62.5	7	9	587	1.25%	0.6	5.0	0.56	1941	1,353
57.5	7	7	0	1.25%	0.6	6.0	0.56	972	972

Figure 9-13B. Variable Chilled Water Flow with Temperature Reset (cont'd)

WATER-SIDE ECONOMIZER VS. CHILLER COOLING

Basis of Savings: Reduced chiller run time.

If air-side economizer is available, it should be used in lieu of water-side economizer for best energy efficiency. This is because the air-side uses only an air handler fan, while the water-side uses a cooling tower fan and condenser pumps and chilled water pumps—everything in the chiller plant other than the chiller itself. Water-side economizer is useful when no air-economizer exists or where the use of winter outside air is problematic, such as in museums and data centers.

The water-side economizer allows the chiller to be turned off and, instead, uses the cooling tower water indirectly through a heat exchanger. For this to work, the ambient wet bulb temperature must be sufficiently low to achieve the desired chilled water temperature—i.e. for 45 deg F chilled water a wet bulb temperature of 35 degrees or lower is usually required.

The following presumes air-side economizer is ruled out. Thus, cooling hours at all outside temperatures are viable.

Analysis compares cooling costs with a chiller to cooling costs with the water-side economizer.

The limitation of the flat plate is wet bulb temperature since that drives the cooling tower; however the A/C load on a building is influenced by dry bulb temperature. In most cases, the capacity of the flat plate in a building is greatest when the need for the free cooling is the least and so economic justification of these systems usually depends on there being some persistent winter cooling load that cannot be served by a standard air economizer.

Step 1

Identify the hours of operation the flat plate system could be used, the coincident load in tons, and the number of hours.

For envelope loads, use dry bulb bin weather data with coincident wet bulb temperatures, estimate the balance temperature, and interpolate between that point (zero cooling) and maximum cooling on the warmest bin temperature. Then rule out those bins that have coincident wet bulb temperatures too high to make the flat plate system work.

For high internal loads that exist all year long, such as a data center, the load is independent of weather and constant. This analysis only requires the number of hours of sufficiently low wet bulb that are concurrent with the cooling load.

Step 2

Base case, with the chiller. Identify the chiller and cooling tower kW; pump energy is the same in both cases and can be neglected. Note that for low ambient cooling tower operation, most cooling towers have 8-10% of their rated nominal capacity with no fan running, and at low wet bulb temperatures this effect will be amplified. Cooling tower energy use will be markedly different for base case and water-side economizer option.

Step 3

Alternate case, with water-side economizer. Chiller kW = 0, and cooling tower fan run time will be more. Identifying the capacity (in HVAC tons) of the cooling tower at low ambient wet bulb is key to this analysis. This can be approximated using the relative cooling tower capacity factor concept, which quantifies the fact that at drier conditions the cooling tower can achieve set point much easier. During water-side economizer operation, the tower will be driving for a low water temperature with a low approach this will work the cooling towers harder, spending more tower fan energy.

Relative Cooling Tower Capacity Factor

$$= \left(\text{AVAILABLE APPROACH}/\text{DESIGN APPROACH}\right)$$

where:

AVAILABLE APPROACH = cooling tower set pt—actual wet bulb temperature

DESIGN APPROACH = design cooling tower set pt—design wet bulb temperature

And, cooling tower kW/ton at various conditions

$$= \text{Tower kW/ton @ design conditions}/\text{Cooling Tower Capacity Factor}$$

Using units of kW/ton for chiller and tower allows direct conversion from ton-hours to kWh.

Example: For a 35 degF wet bulb ambient, compare the work of the cooling tower for making 70 degree condenser water for a chiller vs. making 42 degree condenser water for the water-side economizer, ultimately to make chilled water for the building with the chiller off. The cooling tower is selected nominally for a 9 degree approach while making 70 degree leaving water and is rated at 0.08 kW/ton as paired to the chiller at design conditions.

Ans:
For the chiller, available approach is 70-35=35 degrees, and the factor is 35/9 = 3.8
Adjusted tower kW/ton is 0.08/3.8 = 0.02

For the water-side economizer, the available approach is 7, and the factor is 7/9 = 0.8
Adjusted tower kW/ton is 0.08/0.8 = 0.1 kW/ton

For each ton of cooling provided by the cooling tower in water-side economizer service at these conditions the tower will use five times as much energy than it would if serving the chiller at normal condenser temperatures.

This is still much less energy than running the chiller, but it does show the give and take of the water-side economizer. Note the advantage the simple air economizer has over the water-side economizer at most temperatures, since none of the chilled water system components are needed.

See **Figures 9-14 A, B, C** (weather dependent load)
See **Figures 9-15 A, B, C** (weather independent load)

The dry bulb bins establish the load profile in tons as usual. The coincident wet bulb bins establish the performance of the plate heat exchanger for each point in the load profile. This will tell when the heat exchanger can or can't provide the cooling.

WATER-SIDE ECONOMIZER SERVING BUILDING
WITH MODERATE INTERNAL LOADS

This example assumes a 40 degree break even temperature, and a water-side economizer that will function at 37 degrees wet bulb

Max Load **400** tons

	Mid- pts	DB (F)	DB Bin Hrs	co-incident WB (F)	load (tons)	Chiller kW/ ton	Nom-inal Tower kW/ton
No	92.5	90 to 95	**36**	**57.7**	**400**	**0.6**	**0.08**
Savings	87.5	85 to 90	**99**	**57.3**	364		
In Shaded	82.5	80 to 85	**270**	**56.9**	327		
Area	77.5	75 to 80	**335**	**56**	291		
	72.5	70 to 75	**360**	**54**	255		
Above 37 degF	67.5	65 to 70	**353**	**51.4**	218		
Wet Bulb Tempera	62.5	60 to 65	**460**	**48.5**	182		
	57.5	55 to 60	**384**	**45**	145		
	52.5	50 to 55	**372**	**40.8**	109		
	47.5	45 to 50	**380**	**37.3**	73	0.6	0.08
	42.5	40 to 45	**325**	**34**	36	0.6	0.08
No	37.5	35 to 40	**398**	**30.8**	**0**		
Savings	32.5	30 to 35	**295**	**27.5**			
In Shaded	27.5	25 to 30	**245**	**23.9**			
Area	22.5	20 to 25	**153**	**20.4**			
	17.5	15 to 20	**101**	**15.3**			
Below	12.5	10 to 15	**83**	**10.7**			
Building	7.5	5 to 10	**51**	**6**			
Break Even	2.5	0 to 5	**30**	**1.3**			
Temperature	-2.5	-5 to 0	**8**	**-3.5**			
	-7.5	-10 to -5	**2**	**-8**			

NOTE there are good savings per hour but very few hours per year that are viable for this example. Without a very dry climate or a very low break even temperature, this system has limited value in serving ordinary building loads

Figure 9-14A. Water-side Economizer with Weather-dependent Load

Available approach = condenser water temp set point – wet bulb temp

Relative Tower Capacity Factor = ratio of

available / design approach

Actual tower kW/ton = design kW/ton divided by the Relative Tower Capacity factor

Chiller + Tower kW/ton

BASE CASE - CHILLER

Mid-pts	tower design appr-oach	cond water temp	Avail-able appr-oach	Rela-tive tower capacity factor	Actual tower kW/ton	Total kW/ton
92.5						
87.5						
82.5						
77.5						
72.5						
67.5						
62.5						
57.5						
52.5						
47.5	**9**	**70**	32.7	3.63	0.02	0.62
42.5	9	70	36	4.00	0.02	0.62
37.5						
32.5						
27.5						
22.5						
17.5						
12.5						
7.5						
2.5						
-2.5						
-7.5						

Figure 9-14B. Water-side Economizer with Weather-dependent Load (cont'd)

Cooling towers are run harder to achieve very cold water in water-side economizer mode

Condenser water temp set point – wet bulb temp

Ratio of available / design approach

kWh saved
21,160

WATER-SIDE ECONOMIZER

Mid-pts	tower design appr-oach	cond water temp	Avail-able appr-oach	Rela-tive tower capacity factor	Actual tower kW/ton	Total kW /ton	Reduc-tion in kW/ ton	kWh savings
92.5								
87.5								
82.5								
77.5								
72.5								
67.5								
62.5								
57.5								
52.5								
47.5	9	45	7.7	0.86	0.09	0.09	0.53	14,606
42.5	9	45	11	1.22	0.07	0.07	0.55	6,554
37.5								
32.5								
27.5								
22.5								
17.5								
12.5								
7.5								
2.5								
-2.5								
-7.5								

Tower hp per ton is high, but there is no chiller so the combined power is low

Efficiency gain kW/ton x tons x hours = kWh

Figure 9-14C. Water-side Economizer with Weather-dependent Load (cont'd)

WATER-SIDE ECONOMIZER SERVING BUILDING
WITH HIGH AND WEATHER INDEPENDENT INTERNAL LOADS

This example assumes NO break even temperature, and
a water-side economizer that will function at 37 degrees wet bulb

Max Load **400** tons

No Savings In Shaded Area

Above 37 degF Wet Bulb Tempera

Mid- pts	DB (F)	DB Bin Hrs	co-incident WB (F)	load (tons)	Chiller kW/ ton	Nom-inal Tower kW/ton
92.5	90 to 95	**36**	**57.7**	**400**	**0.6**	**0.08**
87.5	85 to 90	**99**	**57.3**	396		
82.5	80 to 85	**270**	**56.9**	392		
77.5	75 to 80	**335**	**56**	388		
72.5	70 to 75	**360**	**54**	384		
67.5	65 to 70	**353**	**51.4**	380		
62.5	60 to 65	**460**	**48.5**	376		
57.5	55 to 60	**384**	**45**	372		
52.5	50 to 55	**372**	**40.8**	368		
47.5	45 to 50	**380**	**37.3**	364	0.6	0.08
42.5	40 to 45	**325**	**34**	360	0.6	0.08
37.5	35 to 40	**398**	**30.8**	356	0.6	0.08
32.5	30 to 35	**295**	**27.5**	352	0.6	0.08
27.5	25 to 30	**245**	**23.9**	348	0.6	0.08
22.5	20 to 25	**153**	**20.4**	344	0.6	0.08
17.5	15 to 20	**101**	**15.3**	340	0.6	0.08
12.5	10 to 15	**83**	**10.7**	336	0.6	0.08
7.5	5 to 10	**51**	**6**	332	0.6	0.08
2.5	0 to 5	**30**	**1.3**	328	0.6	0.08
-2.5	-5 to 0	**8**	**-3.5**	324	0.6	0.08
-7.5	-10 to -5	**2**	**-8**	**320**	0.6	0.08

Same analysis with load profile nearly independent of weather. Here, the flat plate is providing value anytime the wet bulb temperature is low enough, which includes all winter. Savings are much greater due to increased hours and persistent cooling load.

Figure 9-15A. Water-side Economizer with Weather-independent Load

BASE CASE - CHILLER

Mid-pts	tower design appr-oach	cond water temp	Avail-able appr-oach	Rela-tive tower capacity factor	Actual tower kW/ton	Total kW /ton
92.5						
87.5						
82.5						
77.5						
72.5						
67.5						
62.5						
57.5						
52.5						
47.5	**9**	**70**	32.7	3.63	0.02	0.62
42.5	9	70	36	4.00	0.02	0.62
37.5	9	70	39.2	4.36	0.02	0.62
32.5	9	70	42.5	4.72	0.02	0.62
27.5	9	70	46.1	5.12	0.02	0.62
22.5	9	70	49.6	5.51	0.01	0.61
17.5	9	70	54.7	6.08	0.01	0.61
12.5	9	70	59.3	6.59	0.01	0.61
7.5	9	70	64	7.11	0.01	0.61
2.5	9	70	68.7	7.63	0.01	0.61
-2.5	9	70	73.5	8.17	0.01	0.61
-7.5	9	70	78	8.67	0.01	0.61

Figure 9-15B. Water-side Economizer with Weather-independent Load (cont'd)

20x savings for this example for process load because heat exchanger capacity is coincident with load

kWh saved
413,104

WATER-SIDE ECONOMIZER

Mid-pts	tower design appr-oach	cond water temp	Avail-able appr-oach	Rela-tive tower capacity factor	Actual tower kW/ton	Total kW /ton	Reduc-tion in kW/ ton	kWh savings
92.5								
87.5								
82.5								
77.5								
72.5								
67.5								
62.5								
57.5								
52.5								
47.5	**9**	**45**	8	0.86	0.09	0.09	0.53	73,104
42.5	9	45	11	1.22	0.07	0.07	0.55	64,882
37.5	9	45	14	1.58	0.05	0.05	0.57	80,431
32.5	9	45	18	1.94	0.04	0.04	0.58	59,791
27.5	9	45	21	2.34	0.03	0.03	0.58	49,578
22.5	9	45	25	2.73	0.03	0.03	0.59	30,803
17.5	9	45	30	3.30	0.02	0.02	0.59	20,224
12.5	9	45	34	3.81	0.02	0.02	0.59	16,486
7.5	9	45	39	4.33	0.02	0.02	0.59	10,037
2.5	9	45	44	4.86	0.02	0.02	0.59	5,845
-2.5	9	45	49	5.39	0.01	0.01	0.59	1,542
-7.5	9	45	53	5.89	0.01	0.01	0.60	381

Figure 9-15C. Water-side Economizer with Weather-independent Load (cont'd)

HIGHER EFFICIENCY LIGHTING VS. EXISTING LIGHTING

Basis of Savings: Incremental source efficiency improvement

Hours of operation * differential kW of lighting = kWh savings.

Break this down by area and don't over-estimate the hours of operation. (8760 hours per year is usually unrealistic). See **Chapter 16—Lighting "Average Lighting Hours by Building Type"** for typical hours of operation for lighting in different sectors.

See **Figures 9-16A, B.**

Note: This is a very simplified example. Additional granularity is needed whenever options have large differences in longevity and operational maintenance cycles. In all cases, the difference in power should be taken for equivalent light delivered to the work surface at mid-life.

This helps quantify the ancillary benefit in cooling and the ancillary penalty in heating mode from having less watts of light releasing heat. This can also be estimated with bin weather if a break even temperature exists.

Be careful not to over-state the run hours of the lights

LIGHTING EFFICIENCY IMPROVEMENT

Area	Use	ECM descrip.	Hours of operation per week	Weeks of operation per year	Pct of hours in cooling mode	Pct of hours in heating mode	No. of fixtures
1	Office	T12 to T5 direct / indirect	**70**	**50**	**25%**	**75%**	**65**
2	Data Center	T12 to T8	**168**	**50**	**100%**	**0%**	**15**
3	Break Room	T12 to T8	**50**	**50**	**25%**	**75%**	**10**
4	Meeting Room	Incandes-cent to CFL	**30**	**50**	**25%**	**75%**	**20**
5	Hallway	T12 to T8	**80**	**52**	**25%**	**75%**	**35**

Figure 9-16A. Higher Efficiency Lighting—Quantified Savings

kWh = (Differential watts) / 1000 x (% of fixtures lit) x (no. of fixtures) x hours

Often overlooked. Usually not every bulb is lit.

From catalog data, for the complete fixture including bulb(s) and

kWh savings from lighting alone 14,644

Convert kWh saved to ton-hours cooling avoided:

kWh x (3413/12000) x

% of hours in cooling

Area	Pct of fixtures lit	Exist Light watts per fixture	New Light watts per fixture	Differential watts per fixture	kWh per year saved	Added cooling load from lighting energy ton-hours	Cooling system efficien-cy, kW/ton
1	**90%**	**130**	**105**	25	5,119	364	0.6
2	**95%**	**130**	**92**	38	4,549	1,294	1.0
3	**100%**	**97**	**65**	32	800	57	0.6
4	**100%**	**75**	**28**	47	1,410	100	0.6
5	**95%**	**65**	**45**	20	2,766	197	0.6

Figure 9-16B. Higher Efficiency Lighting—Quantified Savings

Ton-hours x kW/ton

Lighting + cooling kWh savings

Convert kWh saved to therms to be picked up by building heating system:

kWh x (3413/100,000) x

% of hours in heating mode

	Added Savings: kWh saved 1724	Total Elec Savings 16368			Penalty: Therms heat load added to building 323
Area	Net Savings for Cooling, kWh	kWh per year saved total	Lost heat produced by lighting heat, therms	Heating system eff	Additional heating burden added to building system
1	218	5,337	131	**0.8**	164
2	1294	5,842	-	0.8	-
3	34	834	20	0.8	26
4	60	1,470	36	0.8	45
5	118	2,884	71	0.8	89

Note that the reduced lighting load results in an additional benefit in cooling season and and a penalty in the heating season.
The heating mode increase comes from the excess lighting heat that was partially heating the building.
In all-heating applications, the savings will be adjusted by the ratio of the cost of heating to the cost of electricity.
In all-cooling applications, there is no heating penalty.

Figure 9-16C. Higher Efficiency Lighting—Quantified Savings

HIGHER EFFICIENCY MOTORS VS. EXISTING MOTORS

Basis of Savings: Incremental source efficiency improvement

Load profile is required to properly analyze this and to not over-estimate savings.

Equipment efficiency changes at different loads can usually be neglected since both the existing and competing motor will have a similar pattern.

For a motor that is load following for HVAC, the same method of estimating pct load with bin weather can be used. For motors that are constant load, the energy impact is simply the motor size * percent load. Note that motors are seldom operated at full nameplate load, so the "percent load" factor applies to all motor calculations. Using nameplate motor hp alone will almost always over-state energy use and savings.

A reasonable estimate of percent motor load can be obtained with a simple amp reading, noting the ratio of measured amps to full load nameplate amps. This will yield a good approximation provided motor loads are >50%. For low motor loads, watt measurement is needed.

$$\textbf{Approx Motor pct load} = {}^{\text{(Measured Amps)}}/_{\text{(Full Load nameplate Amps)}}$$

$$\textbf{Motor kW input} = {}^{\text{(HP) * (\%Load) * (0.746)}}/_{\text{motor eff}}$$

NOTE: Motors, like all machines, have internal losses. These are minor compared to full load output, but become more pronounced at very low loads. **A motor with nothing connected to the shaft will not consume zero work.** For scenarios that produce very low estimated percentage of load (turndown), a factor for "minimum motor load" is appropriate.

See **Figures 9-17, 9-18A, and 9-18B.**

Constant efficiency is shown. In fact, efficiency drops at reduced load, so results are conservative

kW input = Hp x 0.746 / efficiency

Full motor load x % load

Nom. Hp	Pct Motor Load	Hrs	Output Hp	motor eff, exist	motor eff, new	Input kW exist	Input kW new	kW savings	kWh savings
									Total kWh 14,193
125	100%	20	125	90.0%	95.4%	104	98	6	117
125	90%	100	113	90.0%	95.4%	93	88	5	528
125	80%	600	100	90.0%	95.4%	83	78	5	2815
125	70%	1000	88	90.0%	95.4%	73	68	4	4105
125	60%	1000	75	90.0%	95.4%	62	59	4	3519
125	50%	900	63	90.0%	95.4%	52	49	3	2639
125	40%	200	50	90.0%	95.4%	41	39	2	469
	63% weighted avg load	3820 total hours							

Figure 9-17. Higher Efficiency Motor with known Motor Load Profile

VARIABLE AIR FLOW FAN MOTOR EFFICIENCY UPGRADE

Air flow reduction follows load reduction, and fan Hp will drop exponentially

Note the minimum motor load factor

Max Cooling Load - tons	Max Heating Load - mbh
150	**1,175**

Maximum Air Flow cfm.	Minimum fan load
47,000	**20%**

Minimum Motor Load	Air Handler Flow Minimum
10%	**35%**

Fan Eff	Zone Air Flow Minimum
65%	**20%**

Air Pressure at Max Load in. w.c.	Minimum Pressure in.w.c.
5.0	**1.5**

Mid-pts	DB (F)	Total Bin Hours for each value of percent load	Clg Load (tons) / Htg Load (Mbh)	Pct Load.	Air Flow (cfm)	Pct Air Flow
92.5	90 to 95	**37**	**150**	100%	**47,000**	100%
87.5	85 to 90	**100**	129	86%	40,286	86%
82.5	80 to 85	**285**	107	71%	33,571	71%
77.5	75 to 80	**369**	86	57%	26,857	57%
72.5	70 to 75	**461**	64	43%	20,143	43%
67.5	65 to 70	**539**	43	29%	16,450	35%
62.5	60 to 65	**865**	21	14%	16,450	35%
57.5	55 to 60	**813**	**0**	0%	16,450	35%
52.5	50 to 55	**744**	**0**	0%	16,450	35%
47.5	45 to 50	**729**	98	8%	9,400	20%
42.5	40 to 45	**657**	196	17%	9,400	20%
37.5	35 to 40	**869**	294	25%	9,400	20%
32.5	30 to 35	**693**	392	33%	9,400	20%
27.5	25 to 30	**561**	490	42%	9,400	20%
22.5	20 to 25	**399**	588	50%	9,400	20%
17.5	15 to 20	**302**	685	58%	9,400	20%
12.5	10 to 15	**134**	783	67%	9,400	20%
7.5	5 to 10	**95**	881	75%	9,400	20%
2.5	0 to 5	**81**	979	83%	9,400	20%
-2.5	-5 to 0	**24**	1077	92%	9,400	20%
-7.5	-10 to -5	**3**	**1175**	100%	9,400	20%

Cooling set to cut out below 55 degF in this example
Boiler set to cut out above 50 degF in this example

Figure 9-18A. Higher Efficiency Motor with BIN Weather Load Profile

Pressure reduction with VAV and constant downstream pressure:

Max SP X (CFM2/CFM1) 2.0 +downstream SP

HP(air) = CFM * TSP *Fa / (6356 * fan eff)

but not less than the minimum

kW input =

Hp x 0.746 / efficiency

Total kWh 3,824

Sanity Check

Mid-pts	Pressure in.w.c.	Fan power (motor output), bhp (square instead of cube law)	Existing Motor Size	motor eff, exist	motor eff, new	kW exist	kW new	kW saved	kWh savings
92.5	**5.0**	57	**60**	**0.90**	**0.95**	47	44	3	99
87.5	5.0	49		0.90	0.95	40	38	2	229
82.5	4.1	33		0.90	0.95	27	26	2	440
77.5	3.1	20		0.90	0.95	17	16	1	353
72.5	2.4	12		0.90	0.95	10	9	1	255
67.5	2.1	8		0.90	0.95	7	7	0	213
62.5	2.1	8		0.90	0.95	7	7	0	341
57.5	2.1	8		0.90	0.95	7	7	0	321
52.5	2.1	8		0.90	0.95	7	7	0	294
47.5	1.7	6		0.90	0.95	5	5	0	205
42.5	1.7	6		0.90	0.95	5	5	0	185
37.5	1.7	6		0.90	0.95	5	5	0	245
32.5	1.7	6		0.90	0.95	5	5	0	195
27.5	1.7	6		0.90	0.95	5	5	0	158
22.5	1.7	6		0.90	0.95	5	5	0	112
17.5	1.7	6		0.90	0.95	5	5	0	85
12.5	1.7	6		0.90	0.95	5	5	0	38
7.5	1.7	6		0.90	0.95	5	5	0	27
2.5	1.7	6		0.90	0.95	5	5	0	23
-2.5	1.7	6		0.90	0.95	5	5	0	7
-7.5	1.7	6		0.90	0.95	5	5	0	1

Figure 9-18B. Higher Efficiency Motor with BIN Weather Load Profile (cont'd)

HIGHER EFFICIENCY CHILLER VS. EXISTING CHILLER

Basis of Savings: Incremental source efficiency improvement

Load profile is required for proper analysis.

Ton-hours * kW/ton = kWh

For comfort cooling chillers, weather bins are a reasonable way to estimate the load profile. Establish maximum load at the highest bin, and then estimate the breakeven point below which the chiller will be off.

For process chillers (other than HVAC cooling), load profile will be determined by other than weather.

NOTE: This example is for a single chiller and illustrates the efficiency loss at part load. Where multiple chillers exist and can be staged, much of this efficiency loss can be averted.

See **Figures 9-19, 9-20.**

This method assumes constant condenser water temperature. It can also use chiller data with 'condenser relief' (condenser temperature drops as load drops) if consistent for existing and new, if you are sure the condenser temperature controls will actually do this.

For process chiller, hours and loads are more related to production and must be estimated with other than weather data.

kW/ton x tons kW, differential kW x hours/yr = kWh/yr

HIGHER EFFICIENCY CHILLER UPGRADE

NOTE: assumes constant condenser temperature.

Effect of condenser reset evaluated separately

Total kWh 220,371

Design Capacity (tons)	Pct Load	Hours	Load, Tons	eff, exist kW/ton	eff, new kW/ton	kW exist	kW new	kW saved	kWh savings
745	80%	**1000**	596	**0.550**	**0.460**	328	274	53.6	53640
745	70%	**1200**	522	**0.570**	**0.490**	297	256	41.7	50064
745	60%	**1400**	447	**0.600**	**0.510**	268	228	40.2	56322
745	50%	**1000**	373	**0.650**	**0.550**	242	205	37.3	37250
745	40%	**600**	298	**0.700**	**0.600**	209	179	29.8	17880
745	30%	**200**	224	**0.800**	**0.700**	179	156	22.4	4470
745	20%	**50**	149	**0.900**	**0.800**	134	119	14.9	745

Design capacity is max load, not nameplate unit capacity

kW/ton efficiency evaluated at different values of pct load

Figure 9-19. Higher Efficiency Chiller—Process Chiller

Same method, but hours and loads are estimated with bin weather data.

HIGHER EFFICIENCY WATER COOLED CHILLER UPGRADE

NOTE: assumes constant condenser temperature.
Calculate effect of condenser reset separately

Total kWh 10,762

Mid-pts	DB (F)	Total Bin Hours for each value of percent load	Mechanical Cooling Load (tons)	Pct Load	eff, exist kW/ton	eff, new kW/ton	kW exist	kW new	kW saved	kWh savings
92.5	90 to 95	**37**	**144**	100%	0.60	0.55	86	79	7	266
87.5	85 to 90	**100**	126	88%	0.56	0.52	71	65	6	591
82.5	80 to 85	**285**	108	75%	0.57	0.52	61	56	5	1451
77.5	75 to 80	**369**	90	63%	0.58	0.53	52	48	4	1605
72.5	70 to 75	**461**	72	50%	0.65	0.59	46	43	4	1784
67.5	65 to 70	**539**	54	38%	0.72	0.66	39	36	3	1746
62.5	60 to 65	**865**	36	25%	0.87	0.80	31	29	3	2258
57.5	55 to 60	**813**	18	13%	0.87	0.80	16	14	1	1061
52.5	50 to 55	**744**	**0**	0%	0.87	0.80	0	0	0	0
47.5	45 to 50	**0**		0%	0.87	0.80	0	0	0	0
42.5	40 to 45	**0**		0%	0.87	0.80	0	0	0	0

Chiller set to cut out below 55 degF in this example

Figure 9-20. Higher Efficiency Chiller—Comfort Cooling

HIGHER EFFICIENCY BOILER VS. EXISTING BOILER

Basis of Savings: Incremental source efficiency improvement

Load profile is required to properly analyze this and to not over-estimate savings.

For comfort heating boilers, weather bins are a reasonable way to estimate the load profile. Establish maximum load at the lowest bin, and then estimate the breakeven point above which the boiler will be off.

For process boilers (other than HVAC heating), load profile will be determined by other than weather.

Design capacity is max load, not nameplate unit

For process boiler, hours and loads are more related to production and must be estimated with other than weather data.

Input = output / eff

1 therm = 100,000 Btu

Therms saved 4856

Design Capacity (MBh)	Pct Load	Hours	Load, MBh	eff, exist	eff, new	input, exist, therm	input, new	therm savings
2000	100%	25	2000	80%	88%	625	568	57
2000	90%	200	1800	80%	88%	4,500	4,091	409
2000	80%	600	1600	80%	88%	12,000	10,909	1091
2000	70%	800	1400	80%	88%	14,000	12,727	1273
2000	60%	750	1200	80%	88%	11,250	10,227	1023
2000	50%	600	1000	80%	88%	7,500	6,818	682
2000	40%	300	800	75%	82%	3,200	2,927	273
2000	30%	50	600	70%	79%	429	380	49
2000	20%	0	400	65%	76%	-	-	0
2000	10%	0	200	55%	72%	-	-	0

64% weighted avg load	3325 total hours

firing efficiency evaluated at different values of pct load

Figure 9-21. Higher Efficiency Boiler—Process Boiler

Same method, but hours and loads are estimated with bin weather data.

HIGH EFFICIENCY BOILER UPGRADE

Total Therms Saved: 4,887

Mid-pts	DB (F)	Total Bin Hours for each value of percent load	Load Profile (mbh)	Pct Load	eff, exist	eff, new	input, exist, therm	input, new	therm savings
62.5	60 to 65	**865**	**0**	0%	0.75	0.92	-	-	0
57.5	55 to 60	**813**	68	7%	0.75	0.92	737	601	136
52.5	50 to 55	**744**	136	14%	0.75	0.92	1,349	1,100	249
47.5	45 to 50	**729**	204	21%	0.75	0.92	1,983	1,616	366
42.5	40 to 45	**657**	272	29%	0.75	0.92	2,383	1,942	440
37.5	35 to 40	**869**	340	36%	0.75	0.92	3,939	3,212	728
32.5	30 to 35	**693**	408	43%	0.75	0.92	3,770	3,073	697
27.5	25 to 30	**561**	476	50%	0.75	0.92	3,560	2,903	658
22.5	20 to 25	**399**	544	57%	0.75	0.92	2,894	2,359	535
17.5	15 to 20	**302**	612	64%	0.75	0.92	2,464	2,009	455
12.5	10 to 15	**134**	680	71%	0.75	0.92	1,215	990	224
7.5	5 to 10	**95**	748	79%	0.75	0.92	947	772	175
2.5	0 to 5	**81**	816	86%	0.75	0.92	881	718	163
-2.5	-5 to 0	**24**	884	93%	0.75	0.92	283	231	52
-7.5	-10 to -5	**3**	**952**	100%	0.75	0.92	38	31	7
-12.5	-10 to -15	**0**		0%	0.75	0.92	-	-	0
-17.5	-15 to -20	**0**		0%	0.75	0.92	-	-	0

Boiler set to cut out above 60 degF in this example

Figure 9-22. Higher Efficiency Boiler—Comfort Heating

Note: When condensing boilers are evaluated, it is necessary to evaluate as-connected to the heating distribution and end uses, to determine if/when the boiler will condense (i.e., when the building can be heated at low enough water temperatures for condensing operation to occur).

HOT WATER RESET FROM OUTSIDE AIR VS. CONSTANT TEMPERATURE

Basis of Savings: *Reduced standby losses (thermal loss through piping)*

Assume a constant indoor temperature to simplify the analysis. Standard tables of insulated pipe heat loss can be used for different temperatures. These estimates will usually be conservative, since seldom are heating water piping systems 100% insulated and bare pipe and fitting losses are much higher.

Heat loss during winter is not a complete loss. The heat isn't in the place you normally will want it, but if it is inside the insulated envelope at least part of it is beneficial.

The method shown provides an approximation of savings, but has several subjective parameters that should be acknowledged, such as the beneficial heat factor.

Step 1

Determine the number of hours of operation for the associated outside air temperatures.

Step 2

Estimate the length of pipe involved. 2 inch pipe is shown in this example, but other sizes would be considered as well in practice.

Step 3

Estimate the fraction of the piping system that is insulated. Assuming it is 100% insulated is usually not realistic.

Step 4

Determine the desired reset schedule.

Step 5

Determine the heat loss for bare pipes and insulated pipes at fixed and reset temperatures.

See **Figures 9-23A, B, and C**.

HOT WATER RESET

Bin hours at "60" degF is the sum of bin hours for temperatures between 55-65 degF. This is done only because thermal loss tables are not more specific than 10 degF increments.

Mid Point OA Temp (degF)	Bin Hours in this Range of nominal temp	Mid-pts	DB (F)	Total Bin Hours for each value of percent load
		65	64 to 66	**274**
		63	62 to 64	**307**
60	1862	61	60 to 62	**312**
		59	58 to 60	**178**
		57	56 to 58	**334**
		55	54 to 56	**457**
		53	52 to 54	**291**
		51	50 to 52	**297**
50	1462	49	48 to 50	**272**
		47	46 to 48	**293**
		45	44 to 46	**309**
		43	42 to 44	**278**
		41	40 to 42	**234**
40	1534	39	38 to 40	**315**
		37	36 to 38	**417**
		35	34 to 36	**290**
		33	32 to 34	**291**
		31	30 to 32	**249**
30	1205	29	28 to 30	**225**
		27	26 to 28	**228**
		25	24 to 26	**212**
		23	22 to 24	**132**
		21	20 to 22	**163**
20	620	19	18 to 20	**151**
		17	16 to 18	**101**
		15	14 to 16	**73**
		13	12 to 14	**54**
		11	10 to 12	**57**
10	212	9	8 to 10	**36**
		7	6 to 8	**41**
		5	4 to 6	**24**
		3	2 to 4	**27**
		1	0 to 2	**48**
0	86	-1	-2 to 0	**11**

Figure 9-23A. Hot Water Reset from Outside Air

Beneficial Heat Factor (BHF): Heat lost through the piping is not a complete waste during heating season, as long as the heat loss is within the thermal envelope. It's not all beneficial in cold weather either, since the heat is released above ceilings, in pipe chases, etc.

Resetting the temperature during cooling season is the source of most of the savings

NOTES:

1 At OA temperatures below 40 degrees half of the heat loss is considered beneficial (subjective)

2 Boiler off above 65 degrees

3 Only one size of pipe is shown, to reduce the size of the table.

4 80% efficiency of heating supply is assumed.

OA Range	Mid Point OA Temp (degF)	Bin Hours	Beneficial Heat Factor (Note 1, 2)	Base HW Temp (degF)	Reset HW Temp (degF)	Hours the heating system is circulating	Fraction of pipe and valve system that is insulated
55 - 65	60	1862	1.0	**180**	**140**	1862	**0.8**
45 - 55	50	1462	1.0	180	**150**	1462	0.8
35 - 45	40	1534	1.0	180	**160**	1534	0.8
25 - 35	30	1205	0.5	180	**170**	1205	0.8
15 - 25	20	620	0.5	180	**180**	620	0.8
5 - 15	10	212	0.5	180	**180**	212	0.8
[-5] - 5	0	86	0.5	180	**180**	86	0.8

157 weight-ed avg temp	6981 total hours

Reset Schedule

OA	HW
60	**140**
20	**180**

Subjective Factor for Beneficial Heat Loss

	degF OA	factor
at or above	**40**	**1**
less than	**40**	**0.5**

Figure 9-23B. Hot Water Reset from Outside Air (cont'd)

There is also a small efficiency improvement by running boilers at lower temperatures. This has been ignored since it is likely that the hot water is simply reset from a mixing valve and the boiler water is not reduced.

Repeat for different sizes of pipe as required.

Total Therms 2,196

Mid Point OA Temp (degF)	Length of pipe, ft (2 inch)	Heat loss of bare pipe at max temp Btuh/ft (No Reset, 2 inch)	Heat loss of bare pipe at reset temp Btuh/ft (Reset, 2inch)	Insulat-ed heat loss at max temp Btuh/ft (No Reset, 2 inch)	Insulat-ed heat loss at reset temp Btuh/ft (Reset, 2 inch)	Overall heat loss, constant temp (therm/ yr) (No Reset, 2 inch)	Overall heat loss, reset (therm/ yr) (Reset, 2 inch)	firing eff	therm savings
60	**2500**	**152**	**91**	**19**	**12**	2,123	1,294	80%	1,036
50	2500	152	**106**	**19**	**14**	1,667	1,170	80%	621
40	2500	152	**121**	**19**	**15**	1,749	1,388	80%	451
30	2500	152	**137**	**19**	**17**	687	616	80%	88
20	2500	152	**152**	**19**	**19**	353	353	80%	-
10	2500	152	**152**	**19**	**19**	121	121	80%	-
0	2500	152	**152**	**19**	**19**	49	49	80%	-

Heat loss of bare pipe, with and without reset

Heat loss of insulated pipe, with and without reset

Feet of bare pipe x heat loss rate x BHF / 100,000 = therms for bare pipe

Feet of insulated pipe x heat loss rate x BHF / 100,000 = therms for insulated pipe

This evaluation with and without reset shows the approximate savings

Figure 9-23C. Hot Water Reset from Outside Air (cont'd)

REDUCE AIR SYSTEM FRICTION LOSSES—CONSTANT VOLUME

Basis of Savings: Reduced fan horsepower.

Since airflow is constant, Hp is constant. What is needed are the total hours and maximum air flow.

The same calculation applies to constant flow water systems.

Note: This same calculation applies to variable volume systems, with the flow profile added. With variable air/water flow, Hp is variable roughly as the square of the flow reduction. Savings will depend on the number of hours with high flow rates (e.g. cooling mode), since air horsepower is only significant during these times. For climates with a low number of cooling hours, savings will be minimal compared to cities with a large number of cooling hours.

DUCT AIR LOSS REDUCTION

						Existing	Improved angled filters	Improved discharge transition	Improved other
	Hours	Air flow CFM	Altitude factor	Fan eff	Motor eff	Air system SP (in. w.c.)	Air System SP improvement (in.w.c.)	Air System SP improvement (in.w.c.)	Air System SP improvement (in.w.c.)
Total	4500	25000	1.0	0.7	0.82	4.5	0.3	0.25	0.4
Cooling	1500								
Heating	3000								

Air friction loss reduction measures that reduce the system horsepower requirements

Figure 9-24A. Reducing Air System Friction Losses—Constant Volume (cont'd)

Bhp = CFM x TSP * Fa / (6356 x fan eff)

For constant volume systems, these savings are present whenever the

(cooling hrs / total hrs) x fan energy kWh savings x (3413/12000) = ton hours of cooling benefit

			Sanity Check		kWh savings from fan alone 21,853	
	Total Air system SP decrease (in. w.c.)	Calcu-lated current motor HP input	Existing motor size	Air Hp Reduction	Fan Input Energy Saved	Added cooling load from fan energy ton-hours
Total	0.95	30.8	**40**	**5.3**	21,853	
Cooling						2072
Heating						

Figure 9-24B. Reducing Air System Friction Losses—Constant Volume

(heating hrs / total hrs) x fan energy kWh savings x (3413/100,000) = therms of output that must be replaced by the heating system.

Input = output / efficiency

		Added Savings: kWh saved 1243	Total Elec Savings 23,097			Penalty: Therms heat load added to building 622
	Cooling system efficiency, kW/ton	Net Savings for Cooling, kWh		Heating system eff	Lost heat produced by fan heat, therms	Additional heating burden added to building system
Total						
Cooling	0.6	1243				
Heating				0.8	497	622

Ton-hours x kW/ton = kWh cooling savings

Note that the reduced motor load results in an additional benefit in cooling season and and a penalty in the heating season.,
The heating mode increase comes from the excess motor work that was partially heating the building.
In all-heating applications, the savings will be Adjusted by the ratio of the cost of heating to the cost of electricity.
In all-cooling applications, there is no heating penalty.

Figure 9-24C. Reducing Air System Friction Losses—Constant Volume (cont'd)

AUTOMATIC CONTROL SAVINGS EXAMPLES

While this is by no means a complete listing, it will hopefully convey the general method of energy accounting for control-related measures. Some of the examples show straight forward benefits, and others show system interactions (parasitic losses or competing processes). These examples underscore the need for system knowledge because savings are from the systems being orchestrated by the controls, and why producing a number for control measure savings is not always easy. The logic steps outlined can be used to frame spreadsheets for repeated application.

Several examples share the same fundamental calculation and these are shown as "section notes" which are collected at the end of this section.

Scheduled Start-Stop vs. Continuous Run

Basis of Savings: Reduced run time of equipment.

Interactions: If turned off for load shedding or demand reduction, work is deferred but not eliminated. Indirect savings and penalty exist as a result of reduced power use inside the building envelope.

Principles:

- **Savings = Power * time reduction**
- Indirect cooling savings and heating penalty from reduced internal load (Section note 12)

Remarks:

- Applies to lighting or other discrete loads that have an identifiable usage whenever "on."
- When start-stop is really "enable-disable" of a device with thermostatic or other control, savings only occur when the device is on.
- The value of power for lighting includes any ballast.
- Lighting is usually de-rated to acknowledge that some lights are burned out and off, before you turn them off. For an example of lighting de-rate factors see (**Section note 5**.)
- For motor-driven, heat-cool, compressed air, and process equipment, actual equipment load or load profile must be known for the proposed off-hours to avoid over-stating savings.
- Do not apply this approach to duty cycling, which defers load to a later time and is aimed at demand savings (dollars) rather than energy savings.

Set-Up/Set-Back vs. Constant Temperature in Off Hours

Basis of Savings: Reduced envelope loss

Interactions: If turned off, set-up/set-back savings are null. If constant temperature operation also included constant ventilation, additional savings apply. Ventilation is only needed when occupied.

Principles: **Savings =~ 1% per degF set-up/set-back if kept there for at least 8 hours.**

Remarks:

- If thermally heavy, this measure will have minimal benefit since envelope loads would be minor. (**Section note 10**).
- A detailed explanation of how to model set-back savings has been shown. See **Chapter 7 Automatic Control Strategies "Occupied/ Unoccupied Mode (Set Up/Set Back**)."

Condenser Water Reset vs. Constant Supply Temperature

Basis of Savings: Reduced refrigeration cycle lift

Interactions:

Condenser water reset for cooling towers with low kW/ton fan burden are better than those with high kW/ton fan burden and estimating net savings for this measure requires knowing the specific power requirement of both the chiller and the cooling tower fan.

- Higher specific power for heat rejection (kW/ton for the cooling tower fan) requires additional fan energy which subtracts from measure savings.
- High specific power for heat rejection (kW/ton for the cooling tower fan) is a disabler for condenser water reset savings.
- Depending upon the pairing of chiller and tower, increased tower fan energy can consume 40% to 100% of the chiller theoretical savings from condenser water reset.

Principles: 1-1.5% chiller power reduction per degree lowered (**Section note 1**).

- Cooling tower fan energy increase can be reasonably estimated using the term "Z":
  Z= $({}^{approach}/_{design\ approach})$ (Section note 2).
- Change in cooling tower fan **kWh: new kWh = old kWh * $({}^{1}/_{Z})$**
- Change in cooling tower fan kW **: **new kW = old kW * $({}^{1}/_{Z})$**
- It is convenient to express both chiller power and cooling tower power in kW/ton, to see one go up as the other goes down, or

how the overall combination is affected, so

New cooling tower kW/ton = design clg twr kW/ton * $\left({}^{\text{design approach}}/_{\text{approach}}\right)$

New cooling tower kW/ton = design clg twr kW/ton * $\left({}^{1}/_{Z}\right)$

***Note: The cooling tower motor has not changed sizes and kW does not increase past full load kW. The cooling tower kW and kW/ton estimated in this way is the part load value at those conditions; for cooling towers controlled on-off, it represents the equivalent continuous kW with a duty factor.*

Examples using "Z" are in **Section Note 2**

<u>Remarks</u>:

- This reset routing floats both supply and return temperatures and does not change flow rate or pumping power.
- Ignoring cooling tower fan horsepower penalty will over-state savings.
- Knowing the chiller load profile and coincident wet bulb is essential to quantifying savings.
- Design approach temperature of heat rejection equipment limits this measure and an extremely undersized cooling tower (high kW/ton tower fan power requirement) may eliminate it completely.
- This measure is separate from variable or reduced condenser water flow pumping.

Chilled Water Reset vs. Constant Supply Temperature

<u>Basis of Savings</u>: Reduced refrigeration cycle lift

<u>Interactions</u>:

- Increased pumping energy
- Decreased dehumidification

Principles: 1-1.5% chiller power reduction per degree raised (Section note 1).

- Chiller savings is best determined by applying the factor to the annual load profile, but can be applied to the annual aggregate load (ton-hours) with little error.
- Transport energy for water is low compared to air (less pounds of it to circulate for a given amount of thermal energy to transfer); because of this, the chiller savings will exceed the added pump

energy burden unless chiller is extremely efficient and pump circuit is extremely restrictive.

- Unless the building or process return water temperature is able to float, raising chilled water temperature will decrease the system differential temperature (delta T) because return water temperature is a fixed reflection of building temperature. So, chilled water reset will increase the flow for a given load; this increases pumping energy will erode some of the chiller savings. Pump energy change can be determined with affinity laws, with the new vs. baseline flow:

  $Hp2 = Hp1\ (Q2/Q1)^2$ (Section note 3).

  Hp=horsepower, Q=flow, gpm

- New chilled water flow rate is determined by:

  New flow = old flow * (dT_{old}/dT_{new}) (Section note 4).

  dT=differential temperature

- Evaluate added pump energy using:

  Pump Hp = ($gpm * head * SG/3960$) * [$1/(pump\ eff * drive\ eff * motor\ eff)$]

  gpm=gallons per minute, head=ft of fluid, SG=specific gravity

- Calculate indirect penalty from increased pump energy **(Section note 12).**

<u>Remarks</u>:

- Chilled water reset will cause a reduction in dehumidification at the HVAC air coil. Loss of humidity control in humid climates risks mold damage, unless the reset control is layered with indoor and ambient moisture levels (dew point) that restrict the reset activity to times of acceptably low humidity.
- Loss of unintended dehumidification in dry climates, data centers, and process areas augments savings.
- This measure is separate from variable or reduced condenser water flow pumping.

Load Following Fan and Pump Circulation vs. Constant Flow and Bypass

<u>Basis of Savings</u>: Reduced distribution energy by delivering just enough of the fluid, avoiding bypass flows.

<u>Interactions</u>:

- Creates a new in-line control routine which is a maintained constant pressure for the common pipe/duct serving all the zones, to assure they get what they need.

- Flow is now by demand, but pressure in the system is higher than it was and fan/pump energy is a combination of both.

<u>Principles</u>:

- An approximation of the maintained downstream pressure for variable air flow and variable pumping divides the hydraulic pressure into two pieces:
  - — Friction (that responds to affinity laws)
  - — Residual control pressure which 'rides on top of it' as lift which adds to it **(Section note 11).**
- Apply a load profile which can be from process production (if related) or weather (if related). A method to approximate a load profile from weather is in **(Section note 8).**
- Existing system pressure can be taken from drawings or from field measurements. New system pressure is existing pressure plus control pressure.
- Fan/pump Hp at full flow will be higher with the variable flow option, but drop quickly at all other flows. Increase is from added overhead of control pressure, and the additional series loss of the VFD.

  Pump Hp = $(\text{gpm} * \text{head} * \text{SG}/_{3960}) * [1/_{(\text{pump eff} * \text{drive eff} * \text{motor eff})}]$
  gpm=gallons per minute, head=ft of fluid, SG=specific gravity
  Fan Hp = $(\text{cfm} * \text{TSP} * \text{FA}/_{6356}) * [1/_{(\text{fan eff} * \text{drive eff} * \text{motor eff})}]$
  cfm=cubic feet per minute, TSP = total static pressure in. w.c., FA=altitude factor for air density
- For part load flows and hours, Hp is reduced by affinity laws

  Hp2 = Hp1 $({}^{Q2}/_{Q1})^2$ **(Section note 3).**
  Hp=horsepower, Q=flow, gpm or cfm
- Indirect cooling savings (additional indirect savings, according to cooling system efficiency)

  Indirect cooling savings = fan kWh savings/COP of cooling system
- Heating burden from reduced fan power (fan savings represents additional heating burden)

  Convert fan kWh savings to heating load

  Combustion heating: $\text{kWh}^{*3413}/_{\text{heating efficiency}}$
  Electric resistance heating: kWh subtracts directly
- Calculate indirect cooling savings and heating penalty from fan energy changes if part of a heated and cooled volume **(Section note 12)**.

Remarks:
- In almost all cases cooling systems are easier to justify retrofits to variable flow than heating systems.
- Cooling flows are higher because differential temperatures are lower; and excess fan/pump energy creates additional cooling, roughly 1 ton of cooling for each 5 Hp.
- Any wasted heat from excess circulation motor energy is beneficial to a heating system
- Piping designed for constant flow can experience problems from air or sediment trapping when converted to variable flow.
- Old pump balance valves are not needed for variable flow water systems. Ideally remove them.
- Old inlet vanes are not needed for variable flow air systems. Ideally remove them.
- Include the effect of glycol if present.

Reset Control Pressure for Variable Fan or Pump Based on Demand
Basis of Savings: Reduced throttling loss by providing 'just enough' pressure.
Interactions: Anything that changes load and resulting flow, e.g. envelope or internal loads.
Principles:
- Model the existing flow/pressure signature as two pieces:
  - Friction (that responds to affinity laws)
  - Residual control pressure which "rides on top of it" as lift which adds to it **(Section note 11)**.
- Apply a load profile which can be from process production (if related) or weather (if related). A method to approximate a load profile from weather is in **(Section note 8)**.
- Created a parallel model using existing friction and new control pressure.
- Evaluate and subtract the two for savings.
  Pump Hp = $\left(\text{gpm} * \text{head} * \text{SG}/_{3960}\right) * \left[1/_{(\text{pump eff} * \text{drive eff} * \text{motor eff})}\right]$
  gpm=gallons per minute, head=ft of fluid, SG=specific gravity
  Fan Hp = $\left(\text{cfm} * \text{TSP} * \text{FA}/_{6356}\right) * \left[1/_{(\text{fan eff} * \text{drive eff} * \text{motor eff})}\right]$
  cfm=cubic feet per minute, TSP = total static pressure in. w.c., FA=altitude factor for air density
- For part load flows and hours, Hp is reduced by affinity laws
  Hp2 = Hp1 $\left(Q2/_{Q1}\right)^2$ **(Section note 3).**

Hp=horsepower, Q=flow, gpm or cfm

- Calculate indirect savings/penalty from changes in fan/pump energy if part of a heated and cooled volume (**Section note 12**).

Remarks: The control method that reduces the residual 'lift' (control pressure) can be

- Adjusted manual seasonally (we need less in shift 3, or less in winter)
- Adjusted automatically by outside temperature if that is representative of higher/low flows
- Optimized from automatic polling of end points and providing 'just enough' to satisfy the zone of greatest demand. This is the 'most open valve' control routine.

Air Handler – VAV Supply Air Reset vs. Constant Supply Temperature

Basis of Savings:

- Cooling: Reduced refrigeration cycle lift
- Heating: Reduced reheat burden

Interactions:

- Cooling
  - Hours when air-economizer is used do not count toward these savings.
  - Additional fan horsepower subtracts from refrigeration savings, according to affinity laws and may eliminate most or all savings.
  - Added fan horsepower is a parasitic load, creating additional cooling load
- Heating
  - Raising supply temperature benefits heating zones but adds minor flow and fan horsepower to cooling zones. Savings will be good when most zones are in heating mode together.

Principles:

- Cooling
  - Raising supply air temperature in cooling mode will decrease the system differential temperature (delta T) and increase the flow for a given load; this increases fan energy which will erode cooling savings. Fan energy change can be determined with affinity laws, with the new vs. baseline flow:

    $$\mathbf{Hp2 = Hp1\ ({}^{Q2}/_{Q1})^2}$$ **(Section note 3).**

Hp=horsepower, Q=flow in cfm, cfm = cubic feet per minute

— New supply air flow rate is determined by:

New flow = old flow * ($^{dT\ old}/_{dT\ new}$) (Section note 4).
dT=differential temperature

— Evaluate added fan energy using:

Fan Hp = ($^{cfm\ *\ TSP\ *\ FA}/_{6356}$) * [$^{1}/_{(fan\ eff\ *\ drive\ eff\ *\ motor\ eff)}$]
cfm=<u>added</u> cubic feet per minute, TSP = total static pressure in. w.c., FA=altitude factor for air density

- Heating
  — Elevating the supply air when most boxes are at minimum positions reduces the reheat coil burden. This is because the VAV reheat system must first heat the primary air to room temperature, before any building heating occurs. This is known as the inherent VAV reheat penalty. Primary savings are from reducing the difference between supply air room air temperatures in heating mode.
  — Calculate room to SA differential before/after
  - **A=(Rm –SA no reset)**
  - **B=(Rm - SA with reset)**

Savings=**1.08*altitude correction* heating cfm in heating mode*(A-B)**

- Calculate indirect cooling mode penalty and heating mode benefit from increased fan energy **(Section note 12)**.

<u>Remarks</u>:

- Cooling
  — Added fan power negates most/all of the cooling savings. For generously sized duct and low air friction losses, net gains have been measured.
  — For chilled water systems, savings will only occur if increases in supply air temperature cause an increase in chilled water temperature.
  — Supply air reset will cause a reduction in dehumidification at the HVAC air coil. Loss of humidity control in humid climates risks mold damage, unless the reset control is layered with indoor and ambient moisture levels (dew point) that restrict the reset activity to times of acceptably low humidity.
- Heating
  — Fan penalty occurs at times when load is between partial cool-

ing mode and 'all heating' mode, e.g. the areas still needing cooling will be receiving higher air flows.

— At minimum air flow conditions (winter), the extra fan Hp is a low magnitude penalty (If at 40% of maximum air flow, fan Hp will be 20% or less of full load.) Once the primary air flow is at minimum (with most VAV boxes at minimum), then raising supply air temperature produces savings that outweigh the added fan energy. The aggregate VAV box minimum air flow settings establish the winter supply air flow rate for the fan horsepower penalty calculation.

— Heating savings will dominate fan increase unless there is a large portion of total air flow that requires cooling year-round. Heat of any fan power increase is useful to zones needing heating, but delivered at a fuel cost equal to electric resistance heating.

Demand-Controlled Ventilation vs. Constant Ventilation

Basis of Savings: Reduced outside air tempering load. Reduced VAV reheat.

Interactions:

- Where existing ventilation is currently low, this measure will restore ventilation and increase energy use.
- Savings do not occur in economizer hours. For systems serving multiple zones (VAV or reheat systems) with 'cooling' supply air temperature control and an economizer cycle, ventilation reduction savings in cold weather may be nullified.
- Savings do not occur in setback hours or unoccupied hours.
- Changes to outside air intake must be balanced with need for building pressurization and exhaust make-up and actual reductions may be less than predicted by people count.
- Minimum flows for straight VAV are used for moving heating air into spaces and will limit how low the air flows can go in heating season.
- Changes to space temperature settings (for single zone systems) or supply air temperature settings (for multiple zone systems)

Principles:

- Create an occupancy profile with estimated actual people count, by hour, for occupied times. From this, estimate outside air cfm by hour. Compare to baseline ventilation values to determine the possible reduction of ventilation air.

- If people load is reduced from design but a constant value, bin weather data can be used. If people count varies by hour, weather data will need to be hourly to correlate.
- Two model types:
  — For single zone systems, supply air temperature is controlled directly by the space. Here, consider outside air (OA) load is as if it were delivered directly to the space and so the outside air must be brought to room conditions.
  — For systems serving multiple zones, control is cascading and supply temperature is based on the needs of one or more zones that always need cooling. Here, the load of the outside air is directly felt by supply air rather than the space, and so outside air must be brought to supply air conditions.
- Heating
  — Single zone:
  Savings = 1.08*FA*cfm * dT for each bin or hour, where dT is between <u>room</u> and <u>outside air</u>. No savings when OA temp is above room temperature.
  cfm=cubic feet per minute of outside air, FA=altitude factor for air density, dT=differential temperature
  — Multiple zone:
  Savings = 1.08*FA*cfm * dT for each bin or hour, where dT is between <u>supply</u> and <u>outside</u> air. No savings when OA temp is above supply air temperature.
- Cooling (excluding economizer hours)
  — Single zone
  Same as heating except additionally add dehumidification savings depending upon climate. Use the difference in moisture content (lbs/lb or grains/lb) between OA and space air, but savings only occur if it would otherwise have resulted in dehumidification work, so a threshold of space moisture is needed.
  — Multiple zone
  Same as heating except additionally add dehumidification savings depending upon climate. Use the difference in moisture content (lbs/lb or grains/lb) between OA and supply air, with these savings being determined by mixed air dry/wet bulb mix being taken to supply air delivery temperature (will it condense at 55F or won't it).

Remarks:

- Baseline of "constant ventilation" may not be constant at all for VAV systems. Unless controls exist to cause that to happen, it is common for outside air volumes to rise and fall as supply fan capacity rises and falls; in this case, OA quantity will be at design value only when max cooling load exists. This does not meet ventilation standards, but is by far the most common control application in use for VAV single path designs.
- Single zone unitary equipment that allows the fan to turn off unless calling for heating or cooling will also interrupt ventilation.
- Where the existing systems is not providing constant ventilation, or enough ventilation, the use of DCV may increase energy use as it restores proper ventilation.

Hot Water Reset for Heating System vs. Constant Temperature

Basis of Savings: Reduced piping heat loss. Reduced stack temperature. Reduce cooling false load if operating in warm weather.

Interactions:

- A portion of the heat lost is useful in cold weather, when piping is within the building envelope. If operated in warm weather, waste heat false loads the cooling system.

Principles:

- Assumes constant flow heating hot water, three way valves bypassing un-needed flow.
- Piping loss: estimate insulated pipe heat loss from charts for water temperatures and ambient temperatures, in 10F increments. Overlay with bin weather and hours of operation. A portion of this may be useful heat so is not savings; subjective: if above a ceiling, perhaps half of it is useful; if in a pipe tunnel, none of it.
- Reduced stack temperature: rule of thumb is **1% efficiency increase for a 40F reduction in stack gas temperature**.
- Cooling false load, if running in warm weather: In warm weather none of the heat is useful, plus whatever portion is felt by the cooling system (all of it if inside the envelope) is additional energy savings according to the cooling system kW/ton.

Remarks:

- Hottest water is needed for 'design day' load which is normally the coldest days of the year. Reset from outside air temperature approximates the reduced need. Reducing the fluid temperature reduces piping losses proportionally.

- If the heating water system is variable flow, then indirect heating benefit (**Section note 12**) can be calculated from the increased flow that occurs from lowering supply temperature and keeping return temperature fixed.

Multi-Zone or Double Duct Independent Actuator Control vs. Linked Dampers

<u>Basis of Savings</u>: Reduced heat/cool blending, reduced fan energy

<u>Interactions</u>: Controls that reset hot/cold duct temperatures, controls that turn off heat source in summer or cooling source in winter

<u>Principles</u>:

- Analyze as the traditional system; then adjust for variations.
- Traditional system is constant volume, constant hot and cold deck temperature, and minimum ventilation (ignored). Competing system will be variable volume (necessary when splitting the linkage) with sequenced heat/cool control rather than linked.
- View the array of zone blending boxes as 'one big blending box'. Improving this requires modeling every space and the building.
- Base system heat/cool profile:
    - — Determine or estimate thermal balance temperature for the building (**Section note 10**). Create heating and cooling load profiles in percentages, where both are zero at the balance temperature and 100% at their respective outside air temperatures. (heating is 100% at coldest temperature, cooling is 100% at highest outside air temperature)
    - — Presume that at the building thermal balance point, air flow is half cooling, half heating (**Section note 9**).
    - — With the proportion of cooling varying between maximum (entire fan air flow) and minimum flow according to maximum outside air temperature and thermal balance temperature; then the heating air flow= (1-cooling air flow).
    - — At temperatures above the balance point, cooling flow increases proportionally and heating decreases proportionally. However, heating does not stop being added to the building until the cooling damper is 100% open. This is the false loading blight of this system.
    - — At temperatures below the balance point, heating flow increases proportionally and cooling decreases proportionally.

- — Adjust the mixing assumptions when unusual loading exists such as a zone with significant internal heat gain that is always cooling.
- — Regardless of mixing damper position, mixing only occurs when there is a source of heating active (hot deck is hot) and a source of cooling active (cold deck is cold) at the same time. If the boiler is turned off in summer, the mixing is with cold air and return air; however if the chiller is turned off in winter, there will still be mixing when an economizer cycle is active, false loading the hot deck.

- Alternate heat/cool profile
  - — With dampers uncoupled (individual actuators), control like a VAV box
  - — Flow is proportional to cooling load between maximum flow at maximum OA temperature, and minimum flow at balance temperature. Heat source is closed except when cooling demand has reached minimum. Heating damper activity occurs only while the cooling damper is fixed at the low value. This provides near mutual exclusivity between heating and cooling.
- Contrast two systems
  - — Calculate fan energy for both systems and subtract the two.

Fan Hp = $\left(\text{cfm} * \text{TSP} * \text{FA}/_{6356}\right) * \left[1/_{(\text{fan eff} * \text{drive eff} * \text{motor eff})}\right]$

cfm=cubic feet per minute, TSP = total static pressure in. w.c., FA=altitude factor for air density

  - — Calculate heating energy for both systems and subtract the two **1.08*FA*cfm * dT** for each bin, where dT is between room and hot deck.

    cfm=cubic feet per minute of hot duct air, FA=altitude factor for air density, dT=differential temperature
  - — Calculate cooling energy for both systems and subtract the two

    1.08*FA*cfm * dT for each bin, where dT is between room and cold deck.

    cfm=cubic feet per minute of cold duct air, FA=altitude factor for air density, dT=differential temperature
- Determine indirect cooling savings and heating benefit from fan power changes. (**Section note 12**)
- Variations include economizer or not, hot/cold deck reset or not

Remarks:

- Multizone and double duct systems are schematically and functionally the same, so this method applies to both.
- The method of treating an array of terminal units as 'one big terminal unit' is a simplification that is not perfect but enables simpler modeling methods.

Air-Side Economizer

Basis of Savings: Avoided mechanical cooling

Interactions: Cooling efficiency improvements affect the savings of avoided cooling cost. Envelope improvements or changes in internal loads (lighting) move the balance temperature.

Principles:

- Approach 1
  - Determine balance temperature. (**Section note 10**)
  - Combine weather data, occupied hourly, and thermal balance temperature data to determine hours and cooling loads that would otherwise cause a compressor to run.
- Approach 2
  - Use tables of annual savings compared to annual ton-hours. Results will be quicker, but less accurate.
- A key principle for most commercial buildings is that the cooling loads are small during periods the economizer is available (when it's cool outside), so savings are usually not huge overall. Compare to available rough figures for sanity checks. (**Section Note 6**)
- Another key principle is that a building with low internal gains will have a high thermal balance temperature and little or no economizer benefit will occur. (**Section note 7**)

Remarks:

- Climate dependent, building use dependent (thermal balance dependent)
- For there to be savings, it must simultaneously be cool outside and here must be a need for cooling inside. Further limits exist, based on dew point, when outside air is above 55F and used for cooling, to avoid comfort or issues from increased humidity levels.

Section Notes (for Estimating Energy Savings from Automatic Controls)

These are expanded descriptions of topics repeated in preceding control savings examples.

Section Note 1. See **Chapter 24 Special Topics "Percent per Degree Rule of Thumb for Refrigeration Cycle Improvement"** for explanation of basis for this rule of thumb.

Section Note 2. Z=(approach/design approach) for a cooling tower. Approach is leaving condenser water temperature – ambient wet bulb temperature. New fan energy is roughly estimated as

New fan energy = old fan energy * (design approach/approach), or

New fan energy = old fan energy * ($^1/_Z$)

If designed at 10F approach (leaving water temp – ambient wet bulb), operating at half the design approach will create Z=10/5=2. Achieving the lower temperature narrows the approach and fan energy increases. The fan energy roughly doubles because it will run twice as long, or at higher speed if variable speed. The change can be expressed as higher kWh directly or higher specific cooling tower power (kW/ton).

This effect shows up in

- Cooling tower operation in winter (lower than nominal fan speed)
- Cooling tower reset (exchanging fan energy for chiller energy)
- Water-side economizer operation (chiller is off, but cooling tower is working very hard)

Examples using "Z"

Example A: Condenser Reset

Example A1: CT fan 0.1 kW/ton at 12F approach

250 ton cooling load

1000 kW chiller load is 25%, operating at 250 kW

100 kW CT fan tracking chiller heat rejection, 25% capacity

Current wet bulb temperature is 63F WB

Before:

Current condenser temp is 75F

Approach=75-63=12F

Nominal CT fan load is 100 kW*25%=25 kW (neglecting CT casing internal loss)

Z= ($^{approach}/_{design\ approach}$) = 12/12

Adjusted CT fan load is 25 kW * 1/Z = 25 kW

After:

Condenser water reset lowers water temperature from 75Fto 70F

Approach is now **70**-63=**7F**

Z= ($^{approach}/_{design\ approach}$) = **7/12**

Adjusted CT fan load is 25 kW * 1/Z = **42.9** kW

Chiller kW is reduced 1% per degree which is 5% of 250 kW=12.5 kW decrease

Cooling tower kW is increased by 42.9-25=17.9 kW increase

The increased fan energy consumed 143% of the chiller savings.

Example A2: CT fan **<u>0.05</u>** kW/ton at **<u>7F</u>** approach

250 ton cooling load

1000 kW chiller load is 25%, operating at 250 kW

<u>50 kW</u> CT fan tracking chiller heat rejection, 25% capacity

Current wet bulb temperature is 63F WB

Before:

Current condenser temp is 75F

Approach=75-63=12F

Nominal CT fan load is 50 kW*25%=12.5 kW (neglecting CT casing internal loss)

Z= ($^{approach}/_{design\ approach}$) = 12/7

Adjusted CT fan load is 12.5 kW * 1/Z = 7.3 kW

After:

Condenser water reset lowers water temperature from 75Fto 70F

Approach is now **70**-63=**7F**

Z= (approach/design approach) = **7/7**

Adjusted CT fan load is 12.5 kW * 1/Z = **12.5 kW**

Chiller kW is reduced 1% per degree which is 5% of 250 kW=12.5 kW decrease

Cooling tower kW is increased by **12.5-7.3=5.2 kW** increase

The increased fan energy consumed **42%** of the chiller savings.

<u>Example B: Water-Side Economizer</u>

Example B1: Mechanical cooling in winter CT fan 0.10 kW/ton at 10F approach

250 ton cooling load
1000 kW chiller load is 25%, operating at 250 kW
100 kW CT fan tracking chiller heat rejection, 25% capacity
Current wet bulb temperature is 30F WB
Current condenser temp is 70F
Approach=70-30=40F
Nominal CT fan load is 100 kW*25%=25 kW (neglecting CT casing internal loss)
Z= $(^{approach}/_{design\ approach})$ = 40/10
Adjusted CT fan load is 25 kW * 1/Z = 6.25 kW
Total chiller +cooling tower fan kW = 250+6.25=256.5 kW

Example B2: **Water Economizer Mode, Chiller Off** CT fan 0.1 kW/ton at 10F approach

250 ton cooling load
1000 kW chiller load is **0%,** operating at **0 kW**
100 kW CT fan tracking chiller heat rejection, 25% capacity
Current wet bulb temperature is 30F WB
Current condenser temp is **40F**
Approach=**40**-30=**10F**
Nominal CT fan load is 100 kW*25%=25 kW (neglecting CT casing internal loss)
Z= $(^{approach}/_{design\ approach})$ = **10**/10
Adjusted CT fan load is 25 kW * 1/Z = **25** kW
Total chiller +cooling tower fan kW = **0**+25=**25** kW

Cooling tower fan power increased by **25-6.25=18.75 kW**
Savings were presumed to be the chiller load = 250 kW savings in economizer
Actual savings = 250-18.75=231.25 kW
Increased cooling tower fan consumed 7.5% of the measure savings.

Section Note 3. The affinity law for change in power is a cube function as Hp2 = Hp1 $(Q2/Q1)^3$. However, it is common to use square instead of cube to account for changes in equipment efficiency at reduced load, to avoid overstating savings, thus Hp2 = Hp1 $(^{Q2}/_{Q1})^2$.
See **Chapter 21 Formulas and Conversions "Affinity Laws"** for other forms of these formulas.

Section Note 4. For any value of heat transfer load not undergoing a state change, the circulating flow (water or air) is a function of differential temperature (dT). Twice the dT, half the flow, half the dT, twice the flow.

Section Note 5. See **Chapter 16 Lighting, Table 16-7** for approximate fraction of installed lighting that can be expected to be operational at a given time, which is the same fraction that would be turned on or off by scheduled start-stop. Use good engineering judgment when actual data exists and for other categories such as manufacturing or parking lot lighting. It is unlikely that values over 0.9 are real.

Section Note 6. See **Chapter 5 ECM Descriptions, Figure 5-3** for check numbers for economizer savings and the impact of thermal balance temperature for representative locations.

Section Note 7. See **Chapter 11 Mechanical Systems "Thermal Balance Concept for Buildings" Figure 11-14**, air economizer concept diagram relating thermal balance temperature to savings potential.

Section Note 8. Equating outdoor conditions to a cooling and heating load profile is not exact, nor is it far from true in most cases. For retrofit work, the maximum cooling load is known empirically (equipment exists) and in the absence of high internal loads this load will occur at the hottest day of the year. Some work or good judgment is needed to establish the zero weather load point; however, once identified or estimate, the load in between can be represented loosely with straight proportion. The value of this approximation is the ease of overlaying with bin weather which then yields magnitude of load with hours at that load; most all that remains then is the specific energy use for each value of load and annual energy use is known. See **Chapter 9 "Establishing the HVAC Load Profile**." Example of creating a basic load profile from weather:

- Maximum cooling load occurs at maximum outside temperature.
- Minimum load or zero load occurs at the balance temperature. Non-zero load means there is some persistent process load that always requires a base amount of cooling
- Cooling load is proportional between minimum and maximum values according to OA temperature

- Maximum heating load occurs at coldest outside temperature
- Minimum heating load occurs at the balance temperature
- Heating load is proportional between minimum and maximum values according to change in OA temperature.

Section Note 9. Repeated field observations of traditional double duct and multizone systems show that when the thermostat is satisfied, the blending dampers are at half-half. The building itself, modeled as 'one big blending box' is inherently at that satisfied point at the thermal balance temperature. This approach approximates the building as a whole, not boxes individually; not exact, but not far from truth.

Section Note 10. Thermal balance temperature is an essential figure but difficult to determine precisely – and it can move around when internal loads vary substantially (e.g. church on Sunday). Regression of electric use vs. outside air temperature can show the point above which mechanical cooling is needed; regressing fuel use against outside air temperature can show the point below which heating is needed. Concept of balance temperature is akin to energy management terms 'thermally light' and 'thermally heavy' which refer to internal thermal loads; the higher the internal thermal loads, the lower the thermal balance point (temperature below which heating occurs).

Section Note 11. Accuracy of fan and pump power analysis is improved when dynamic and static work is separated. Energy use for dynamic losses is velocity dependent and follows affinity laws; energy use for static work (lift and control pressure) is independent of velocity and unaffected by affinity laws. Lift is a static, unchanging, pressure and is in contrast to friction loss that changes dynamically with velocity. The term 'lift' is exactly applied in water pumping where source and destination are at different elevation; the work per unit of flow to 'lift' the fluid is the same regardless of velocity and does not respond to affinity laws. In some pumping applications, almost all of the pump work is lift with almost none of it being friction. In sealed circulating systems, there is no lifting of fluid since any piping elevation change is recovered on the return path. However, when a static control pressure is added to any pipe or fan system, it behaves as lift in that it is constant regardless of velocity. See **Chapter 15: Savings Impact When Controlling to a Constant Downstream Pressure-VAV and Variable**

Pumping.

$$SP2 = [SPx * (CFM2/CFM1)^{2.0}] + CSP$$

Where:

CSP=the constant downstream pressure (lift)

SPx=the static pressure related to friction

Section Note 12. Indirect savings and penalties exist when energy is reduced within a control volume.

- Cooling:
  - — Reduced heat dissipation in the space or a working fluid (air/water distribution) requires less cooling since the waste heat was part of the cooling load. Indirect savings are equal to the measure's direct energy reduction divided by the coefficient of performance (COP). For example a 10 kW reduction in lighting load served by a cooling system with a 2.5 COP would have a direct savings of 10 kW and an indirect savings of 10/2.5 = 4 kW added savings <u>during cooling mode</u>.
  - — Increased heat dissipation creates an indirect penalty for the cooling system.
- Heating:
  - — Reduced heat dissipation in the space or a working fluid (air/water distribution) requires more heating since the heating was beneficial. Indirect savings are equal to the measure's direct energy reduction divided by the coefficient of performance (COP). For example a 10 kW reduction in lighting load served by a heating system with a 0.8 COP would have a direct savings of 10 kW and an indirect <u>penalty</u> of 10/0.8 = 12.5 kW penalty <u>during heating mode</u>. If the same source of energy (electricity) is used for heating and lighting, the lighting savings are more than negated in winter months, however most commercial systems use fuel other than electricity for heating and fuel cost per Btu are less than heating with electricity.
  - — Increased heat dissipation creates an indirect benefit for the cooling system
- Seasonally there is a give and take with the indirect savings and indirect penalty where a reduction in internal loads impacts heating and cooling system serving the common volume. In some cases the two cancel, but in most cases there is a noticeable impact

one way or the other (Anchorage vs. Miami, for instance).

- If no heating or cooling of the control volume occurs, there are no indirect savings or penalties from a change in energy in the control volume.

COMPUTER MODELING/SIMULATING ENERGY USE

General

There are a variety of computer software programs to use. Some are proprietary, some are public domain, and some are produced by equipment manufacturers. Some are an attachment feature to load calculation software (for sizing equipment and systems) and some were developed specifically for energy use.

The desire for an energy model is the ability to evaluate and what-if the systems as desired, with all the interactions and minutia carefully considered, and results accurate enough to take to the bank. The more dynamic and interactive systems are, the more one yearns for such a tool, but also the more one wonders if it is possible to build. There is comfort in knowing that people with strong backgrounds in systems were consulted, but there is also a worry to become complacent with blind trust in the elegant software screen. It is always advised to reality check all tools, from a volt meter to a spreadsheet calculation, to a computer model with some known quantity and do this repeatedly until a comfort level is established. This advice holds true for any software where engineering or financing is involved.

Common traits of energy modeling software products

- Hourly weather data
- Composite insulation and building material routines to 'assemble' the buildings, including heat transfer functions that acknowledge mass, specific heat, and time functions of materials. I.e. time-shifting the occurrence of indoor loads from external influences.
- Zoning capability
- Schedule profiles for lighting, occupants, and internal loads. These identify coincident loads
- Efficiency profiles for equipment, from part load, ambient temperature, etc
- What-if capability for evaluating measure options

Variants of energy modeling software products

- Graphic presentment, importing building shapes from documents, building rotation.
- HVAC system types
- Default and customization options
- Assumptions and sub-models of how complex HVAC systems perform under varying load, such as blending, mixing, and reheating
- Different types of HVAC systems and controls that can be chosen

Energy Simulation Notes

Source: IPMVP-2002/DOE

1. Simulation analysis needs to be conducted by trained personnel who are experienced with the particular software and calibration techniques.
2. Input data should represent the best available information including as much as possible of actual performance data from key components in the facility.
3. The simulation needs to be adjusted ("calibrated") so its results match both the demand and consumption data from monthly utility bills within acceptable tolerances. The use of actual weather data may be necessary in cases where the actual weather data varies significantly from the average year weather data used in the simulation. Close agreement between predicted and actual annual total energy use is usually insufficient demonstration that the simulation adequately predicts the energy behavior of the facility.
4. Simulation analyses need to be well documented with paper and electronic copies of input and output files as well as the survey and metering/monitoring data used to define and calibrate the model. The particular version number of the software should be declared if it is publicly available so that any other party can fully review the many computations within the simulation.

Calibrating the Computer Simulation Model for Energy Use

Source: Jayme Buresh, PE, CEM

These are *general guidelines* for working with monthly utility data. If working with data that has been metered separately or broken out then it makes the calibration process much easier because then you can compare end usage to end usage adjust those separately.

1. Verify total annual use matches actual utility bills

Try to achieve agreement between simulated and actual values within 10% for each month and within 10% on the year as well.

2. Create the baseline energy usage

Get a consistent baseline for comparison, with no anomalies. Try to have at least three years' worth of data or at least a data set that is representative of what is currently happening in the building. For example, if you have five years of data for a building but major equipment has been updated with more efficient pieces in the last three years, then really only the last three years is relevant. Also, if one of the five years had a summer with record heat, the amount of cooling needed that summer would be significantly more than the other four years; then that year would be an anomaly and would be discarded.

An important note for model calibration is that simulation software uses weather files based on historical data, designed to indicate probable norms, but which will not reflect warming trends found in some areas. Some software products may allow the creation of custom weather data which, for example, could be an average of five recent years with anomaly years removed. Whether to require this additional step is a judgment call by the simulation analyst, but remember the level of accuracy intended by the model is not 100% agreement. One rule of thumb would be if a particular year is consistently 15F different than the weather data norms, it would be an anomaly year; and if non-anomaly years being calibrated to are consistently 15F different than the simulation weather files, either create custom weather or anticipate a higher error tolerance due to strong weather.

3. Chart the baseline usage versus the simulated usage, and baseline kW versus simulated kW

- The visual comparison of predicted vs. actual plotted together is very helpful. Look at overall shapes, rises and falls, and peaks. The more breakdowns (separate meters or logged data for end uses or specific panels) between utility bills you have the better and easier the calibration will be. For example, if lighting panel usage is recorded, then a sub-calibration could be lighting usage simulated vs. actual.

  Note: if using permanently installed customer sub meters, it should be recognized that these meters are likely without regular calibration, and so integrating the additional data is a potential source of error. In contrast, utility meters are 'revenue grade' and normally calibrated on a scheduled basis. In the example of a lighting panel, it would not be difficult to spot-check the sub meter reading with a portable meter of known good accuracy and use the data conditionally on agreement.

- Note that utility data for a given month is not necessarily aligned with the calendar month; for example a bill from June 10 to July 10 may be identified as "July" usage.

4. Analyze the graph shapes

There are several things you can do here. Unfortunately this really comes down to having a feel for what is contributing to the differences between the two sets of numbers (experience). Sometimes the graphs will clearly indicate where the difference is, but sometimes it's just working on it until you feel comfortable with it which can include several changes. It can be helpful to keep notes for each major change and how it affects the simulation and the calibration.

- If the shapes of the two graphs are similar, but one is higher or lower, this is likely representative of a constant load (such as lighting or plug loads or any fans that run 24/7) being either too high or too low in magnitude. This is an easy adjustment to make by looking at the peak kW for each month and then adjusting the simulation accordingly. Plug loads are the most variable portion of simulations so that is always a good place to start.
- Look at the cooling season and heating season. Cooling season example: If the two graphs are very close throughout the year except the summer months are lower for the simulated values than the baseline values, it likely represents too high of an efficiency in the model (or too low of an efficiency if usage is higher than the simulated values). The same is true for heating usage in winter months.
- If the swing seasons (fall and spring) are not matching, it is often due to the simulation not correctly taking into account the tradeoff between heating and cooling during these seasons. Maybe there is too much heating and not enough cooling or vice versa.
- If there are vast differences between modeled and actual data, or the general graph shapes that aren't even close, this can mean there are major issues with the simulation. In some cases, it is easier to re-verify the inputs and re-create the simulation.

5. Try to never make changes to things that you know are accurate

- If you were given schedules or took a lighting survey or have equipment name plate data then you know that information is accurate so it shouldn't be changed. But there are always inputs that have to be estimated so that is a good place to start.
- Tempering thoughts with field data may include equipment that does not always run as assumed, lights that are not always on (or burned out lights), or anomalies with schedules provided.

Simulation Calibration Notes and Procedure for M&V Method D

Source: IPMVP-2002/DOE

Calibration is achieved by verifying that the simulation model reasonably predicts the energy use of the facility by comparing model results to a set of calibration data. This calibration data should at a minimum be measured energy consumption and demand data, for the portion of the facility being simulated. Calibration of building simulations is usually done with 12 monthly utility bills. The calibration data set should be documented along with a description of its source(s). Other operating data from the facility can be used as simulation input data as part of the calibration data set. These data might include operating characteristics and profiles of key variables such as use and occupancy, weather, known loads, equipment operating periods and efficiency. Some variables may be measured for short intervals, recorded for a day week or month, or extracted from existing operating logs. Accuracy of measurement equipment should be verified for critical measurements. If resources permit, actual building ventilation and infiltration should be measured since these quantities often vary widely from expectations. Snap-shot measurements will significantly improve simulation accuracy. Where resources are limited, on/off tests can be used to determine snap-shot end-use measurements of lighting, receptacle plug loads and motor control centers. These tests can be performed over a weekend using a data logger or EMCS to record whole-building electricity use, usually at one-minute intervals, and in some instances with inexpensive portable loggers that are synchronized to a common time stamp (Benton et al. 1996, Houcek et al. 1993, Soebarto 1996).

Following collection of as much calibration data as possible, the steps in calibrating the simulation are:

1 Assume other input parameters and document them

2 Verify that the simulation predicts reasonable operating results such as space or process temperature/humidity

3 Compare simulated energy and demand results with metered data, on an hourly or monthly basis

- Use actual weather data when conditions vary significantly from average year weather data.
- Assess patterns in the differences between simulation and calibration data. Bar charts, monthly percent difference time-series graphs and monthly x-y scatter plots give visual presentations which aid the identification of error patterns.

4 Revise assumed input data in step 1 and repeat steps 2 and 3 to bring predicted results reasonably close to actual energy use and demand

- More actual operating data from the facility may also be needed to improve the calibration.

Buildings types which may not be easily simulated include those with:

- large atriums
- a significant fraction of the space underground or ground coupled
- unusual exterior shapes
- complex shading configurations
- a large number of distinct zones of temperature control

Some building ECMs cannot be simulated without great difficulty, such as:

- addition of radiant barriers in an attic
- HVAC system changes not enabled by the fixed options within some whole building hourly simulation programs.

Accuracy Expectation for Energy Simulations

Source: IPMVP-2002/DOE

Calibrations based on monthly utility data can achieve ±20% compared to monthly energy use. Hourly calibrations can achieve ±10% to ±20% of hourly energy use, or ±1% to ±5% of the monthly utility bill.

Other Sanity Checks

1. Compare Actual to Modeled Maximum Loads

If the model produces maximum heating and cooling loads, compare these to actual equipment sizes for a sanity check. Note that many

HVAC systems include a measure of redundancy in the primary equipment, so establishing the 'actual' max heating and cooling load is needed. For example, asking the operators if both boilers run during the coldest days, and if so does the lag boiler run hard or cycle. Actually being on site during the hottest or coldest days is an excellent way to establish maximum loads, by noting the percent capacity of a modulating machine or percent run time for an on-off machine. For example, if the boiler runs 20 minutes and is off 10 minutes, it is running at 2/3 capacity and the actual load at that point is 2/3 of the boiler's rated capacity.

2. Check Numbers for Heating and Cooling Systems

Heating and cooling capacities in terms of "Btuh per SF" (heating) and "SF per ton" (cooling) are good sanity checks, although these numbers – especially heating - vary by locale.

The more important number for energy calculations is the annual energy use. For heating this is "Btu per year," and for cooling this is "Ton-Hours per year." Publishing check numbers for annual heating and cooling energy use, especially cooling, are risky since there are a great number of variables, including vintage, locale, user settings, hours of operation, equipment condition, outside air, and internal loads. Heating loads are easier to quantify than cooling since most buildings with natural gas or fuel oil use most of it for heating (the utility meter values represent the heating energy input).

3. Estimating Maximum Loads from Field Observations

Direct Measurement. Empirical data is best. Being onsite on the hottest day of the year and measuring flow and differential temperature of the main pipe leaving a chiller plant establishes maximum load nicely, presuming the buildings served by it are under control. Similarly, heating 'export' energy units on a design day (coldest day) will provide this information. An enabler for load profiles that is always recommended is long term recorded trends from an energy management system. Temporary data loggers are also good choices when left in place long enough to 'capture' the peaks and trends.

Sub Meters. Data from customer-owned gas or electric sub meters is sometimes available and attractive for estimating maximum loads, or load profiles, for individual buildings on a campus or individual areas within a building. This data may be in the form of electronic file, trend, or clip board.

Note:
A cautionary note with any sub meter is its accuracy, which may be very poor unless it has been maintained and calibrated; badly neglected sub meters can be useless or even harmful if their data is used, so an appropriate first step is to prove the sub meter before using the data. For example, if adding up all the natural gas sub meters produces a number that is markedly different than what the master utility meter reads, there is trouble. Another example is a sub meter that is measuring amps in a switchboard – these will include a "multiplier" for the CT ratio, e.g. a 400:5 CT ratio would have a multiplier of 80. Checking the multiplier value or a parallel measurement with an amp probe and comparing with the value on the computer are reasonable steps. Pulse meters or add-on boards have a constant that is interpreted by the computer as "one pulse is equal to xxxxxxxx," and can be checked for reasonableness when attached to a utility meter, by comparing demand and or consumption values over a period of time.

Pull Up Loads. For buildings with other than 24x7 occupancy or with set-back controls, there are periods of "warm up" or "catch up" heating and cooling; these are pull-up loads. Equipment sizing normally includes provisions for this, however any snapshot of maximum heating or cooling load that is taken during a pull-up event will overstate the maximum building load somewhat, but do provide a point of reference. Pull-up work usually occurs for 1-3 hours after an unoccupied period.

Redundancy. Many systems include redundant equipment, for obvious reasons. It is not a safe assumption to simply add the capacity of the equipment and declare this to be the maximum heating, cooling, or compressed air load. Some way to discount the redundant capacity is needed.

Example 1: Gas heating. The annual heating burden is evident from the gas heat, if all/most of the gas is used for heating.

Example 2: Electric heat. Determine the baseline use for electricity that is unrelated to heating or cooling (regression against degree days). Hourly interval data viewed against outside air temperature, less baseline usage, will point to electricity used for heating and maximum heat demand. The uncertainty in this method is the accuracy of the value used for 'electric use other than heating."

Example 3: **Operator reports**. To use this method requires an indication of heating capacity, from the operator, and *any redundancy must* *be removed*. i.e. "The 1,000,000 Btu boiler runs at about 75% capacity on the really cold days. Boiler 2 is redundant."

Example 4: Modulating boiler. Some boiler control panels provide a value for firing rate, e.g. "75%." The value can apply to a modulating burner or a collection of staged burners or boilers; or a visual inspection of the travel position on the fuel control valve, usually a quarter turn valve with position markings. In the case of modular boilers, the accuracy improves when several boilers are at full fire and the trim boiler is modulating; the accuracy improvement is due to the reduced magnitude of the estimated value. If control is proportional, the firing values will be steady and, on a very cold day, indicative of the maximum load. However, if the control is on/off (cut-in/cut-out), the firing rate represents design load plus some pick-up load and so "on" and "off" (zero) firing rates need to be time-weight averaged.

Example 5: Modulating chiller.

Where good quality export flow and temperature instruments are installed, these are ideal since flow and temperature accurately defines the value of load.

Note that temperature differentials for chilled water systems are small and so even a 1 degF error can mean a substantial difference in the conclusion of tons load. Use of field installed gages is risky.

If dedicated pumps are used, the pump tag may be used but if there is one pump balance valve that has been set properly and left alone there nine others that have been messed with, so use pump tags with caution. If flow stations are available, they can be read directly if they are functional and there is access to the "dP vs. flow" chart furnished with the circuit setter device. The chiller barrel itself can be used as a flow meter - pressure drops through water chillers are carefully measured by the manufacturer and are easily used as a "flow meter" – assuming the tubes are clean and not plugged.

With clean condenser tubes, the temperature rise across the condenser is a good quick estimate of machine output at that time. For example if the design is 10F rise across the condenser outlet vs. inlet water temperature, a 7F rise indicates the chiller is 70% loaded.

<u>Note</u>:

Motor amps are convenient values, but can be misleading because they are influenced strongly by condenser temperature and so caution is advised for using them to estimate cooling load. For example, it is very possible to have full output of the chiller with half of the rated power (amps or kW), if the condenser water is cold enough; or to show high amps that do not correspond to high load, due to a dirty condenser or malfunctioning cooling tower. Likewise, the control panel display that indicates 'percent load' or the inlet vane actuator position can be misleading.

Example 6: Air compressor. On-off controlled compressors require patience and a stop watch; time the minutes "on" and minutes "off" for several cycles. The total output capacity of the machine multiplied by the percent run time indicates average load. For modulating air compressors, a flow meter is ideal, and a data logger is a further improvement in establishing confidence of what the maximum load is.

<u>Note</u>:

Estimating compressed air load from motor kW is risky for compressors with part load controls. For example, the relationship between kW and cfm will be different if the compressor is operating as VSD, inlet throttling, load-unload, or on-off.

Example 7: Modular on-off boilers or air-cooled chillers. When assembled or controlled in groups, there may be a master control panel that indicates "percent load" for each of the stages. These can be used as a reasonable gage of maximum capacity, i.e. a 400-ton chiller with four equal size compressors showing 1st stage 100%, 2nd stage 100%, 3rd stage 100%, and 4th stage 30% would yield 330 tons, more or less, as the output.

Part load controls such as cylinder unloaders or hot gas bypass will impact the estimate

Example 8: Air source heat pumps. Difficult to assess in the field. Sizing, run time, or motor load are not a good indicators of maximum heat load capacity for the building due to the falling nature of heat pump capacity and the incremental sizes of backup resistance heat. The best indicator here may be average electrical demand in very cold weather when heat pump compressors have been turned off in favor of the resistance heat.

	Commercial
Cooling Max Load (SF/ton) This is cooling system output, not labeled equipment size.	Source: ASHRAE Pocket Guide, 2001, Lo-Avg-Hi. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org. **Figures in ( ) are Denver area estimates** Apartments, High Rise **450-400-350 (600)** Auditoriums, Churches, Theatres **400-250-90 (500)** Schools, Colleges, Universities **240-185-150 (350)** Hotels, Motels, Dormitories **350-300-220 (500)** Libraries and Museums **340-280-200 (450)** Office Buildings (general) **360-280-190 (550) (380 for all glass walls)** Restaurants **135-100-80 (200)** Department Stores **350-245-150 (400)** Malls **365-230-160 (475)**
Heating Max Load (kBtu/SF) This is heating system output, with redundant equipment factored out. Input is output divided by efficiency.	Source: Author experience, Denver area **30** (post 70's) (range 15-45) **45** (pre-70's) (range 22-67)
Air Flow Density (CFM/SF) Varies by building size, envelope, and internal load	Source: Author experience, Denver area **0.8-1.2** range 0.5-2.25

Figure 9-25. Check Numbers for Cooling and Heating Design Loads

MEASUREMENT AND VERIFICATION (M&V)

General

A full treatment of this topic is beyond the scope of this book. The de facto standard is Volume 1 of the International Performance Measurement & Verification Protocol (IPMVP).

When reviewing these, bear in mind the intended audience for the document is the Performance Contracting industry and Guaranteed Savings contracts. With a savings guarantee, it is essential to define rules of engagement and minimize disagreement on the outcome, which usually manifests itself in the form of 'who owes who' some money. The purpose of M&V for Performance Contracting is to provide a basis for a business arrangement that is fair to both parties. Risk is a core theme in performance contracts and it is reasonable to expect to have control over something that one is guaranteeing. For the M&V aspect of a guaranteed savings contract, the customer is free to do what they want, but the contractor has the right to adjust the baseline value when appropriate. From IPMVP Volume 1, the general equation for calculating savings with M&V is:

Energy Savings	=	Base Year Energy Use	
	-	Post-Retrofit Energy Use	
	+/-	Adjustments	(eq.1)

Some Things that Impact the Baseline

See also **Chapter 24 Special Topics, Section N-Regression**

Savings are calculated from before and after energy usage and presume nothing changes in the building except the ECMs. When other things change, and they do, the baseline requires adjustment. Some adjustments can be made using regression (base building signature and variables), while others require a manual adjustment.

- **Weather**
  There is more to weather patterns than temperature (more or less sun, rain, or humidity) but the usual method is to normalize on temperature. In a given city or region, degree-days are an accepted metric of comparing weather in different years.
- **Building Size**
  If the building has been added onto, with the same function but simply more of it, then the magnitude of energy use will follow the proportional change in square footage. Construction temporary

power may also create a sizeable impact. If the new area has not yet been occupied, or has a different function, then an all-new baseline may to be established.

- **Building Use**
  A change in business activity can fundamentally change the amount and times energy is used. In manufacturing, changing operations to a new product can make the space more or less energy intensive.
- **Internal Loads**
  If new equipment is added, energy use will rise. Accounting for this is done by establishing the load profile of the equipment and adding it directly to the baseline.
- **Behavior Changes**
  User adjustments of climate conditions, process temperatures, ventilation rates, equipment schedules will markedly affect changes. It is common for digital control system changes to be monitored for this reason. These are the hardest changes to manage and quantify since they are so easy to change.
- **Occupancy/Census**
  If one year's occupancy level is significantly different than another, energy use will be impacted. This requires a calculation to determine the portion of energy use that is attributed to people and their activities (e.g. computers). Effects would include equipment, ventilation, cooling, and heating and benefits may counter-balance in some cases.
  One example is an office building that experiences a large drop in occupancy; the electric use from less personal equipment will drop, the lighting load may or may not drop depending on if the vacancy is grouped to an area or 'sprinkled' throughout, and the heating energy for the building may increase from the reduced self-heating effect.
- **Building Usage Pattern**
  If occupants or workers utilize the facility more or less hours from year to year, energy use will be affected. Adding a shift, switching from (5) 8-hour days to (4) 10-hour days, furloughs, strikes, overtime, holiday shutdowns, etc. will either raise or lower the baseline. These may be one-time events or new continuous patterns. When a building is "closed" for a day, the energy use does not reach zero, due to minimum heating and lighting requirements and other standby losses of an unoccupied building.

- **Production**
  Manufacturing energy use is us usually tied to production output. Significant increases in production will increase baseline use. However, a significant decrease in production, especially when below half of plant capacity, may have non-proportional decreases due to part load losses.

M&V Cost

M&V costs add to project costs, which can also be seen as subtracting from project savings. For this reason, M&V methods and rigor should be balanced with actual need, and these parasitic costs minimized to the extent possible. The argument for M&V value is reducing uncertainty in savings by virtue of additional rigor in M&V, although the customer view is that the advertised savings are the expectation and paying additional money to get those savings is counterintuitive. Another viewpoint for recurring M&V is guarding against savings decline from system degradation over time from normal wear and tear and, especially, from lack of proper maintenance, adjustment, and monitoring. Guidelines for appropriate M&V cost vary.

- 2-5% of annual cost savings Source: M&V Guidelines 4.0, FEMP/DOE
- <10% of annual cost savings Source: IPMVP-2002

Scope for M&V also varies and is associated with the cost of M&V.

Performance contract provisions share risk between the contractor and the customer. In most model contracts there is an option to discontinue M&V after a period of time. If the customer finds that the ECM has performed to expectations and gives indications that it can be expected to go on saving throughout its life, the customer can elect to stop the M&V for the measure. This stops the recurring cost of the M&V task, but also stops the guaranteed savings for the ECM; the guarantee is normally linked to the customer paying the recurring M&V charges. The rationale for stopping M&V after a few years is the savings have been as expected and the customer is able and willing to provide all the ongoing care to keep the ECM operating as efficiently as it can.

Notes on the Use of Stipulated Values in M&V

Stipulations can be used in methods A and D.

The use of stipulations is a practical, cost-effective way to reduce M&V costs and allocate risks. Stipulating certain parameters in the M&V plan can align responsibilities, especially for the items no one controls. However, the use of stipulated values have the potential to shift risk to the customer (Source: M&V Guidelines 4.0, FEMP).

Stipulation may be based on historical data, such as recorded operating hours or load profiles from base year, or may simply be agreed-upon values.

Examples:

- Assume 4500 hours per year operation
- Assume operating schedule is the same this year as last year
- Assume measured load profile measured this year will be the same for prior year
- Assume energy management load profile records logs from last year will be the same this year
- Assume the load profile before and after is xxx hours at 100% load, yyy hours at 80% load, zzz hours at 60% load, etc.
- Assume manufacturer's data values for energy use at different loads

M&V Characteristics that Steer Choices between Method A,B,C,D

There are choices of which Option (A,B,C,D) to apply, but certain characteristics often make one option a better choice. Remember: Savings are not directly measured, but are the difference between Energy use before and Energy use after, so part of the M&V method choice is based the need to measure before as well as after. As an extreme case, it would probably not be popular to suggest the customer operate inefficiently for an additional year to establish a good baseline measurement.

Some of the differences between the four M&V methods are obvious and some are more subtle. Additional descriptions and uses of the four M&V methods are provided to improve clarity of when it is appropriate to use which one.

M&V Application Notes

Avoided cost general format:

Energy Savings = Energy Before – Energy After

Avoided energy use = Adjusted Baseline Energy Use
- Reporting Period Energy Use
+/- Non Routine Adjustments

Key parameters are the primary thing changing; the thing that is driving the savings. Catalog data cannot be used as measurement of a key parameter.

Each M&V objective has a measurement boundary around it that defines what needs to be known to quantify savings. Where energy streams crossing the measurement boundary are minor, they can be estimated or considered negligible if the effect on savings is small; estimates used for this purpose should be conservative so as to understate savings.

M&V activities need a detailed plan prior to implementation, agreed upon by all and/or reviewed by a third party. The M&V plan needs to include things like:

Chosen M&V option
Basis of savings determination
Calculation methods
Baseline determination methods
Key parameters and non-key parameters
Assumptions and estimated quantities
Duration of measurements
Sampling and basis
Static factors (things that would impact energy use substantially if they changed)
Baseline adjustment factors
Meter data
Expected accuracy/uncertainty

M&V Option Application (Source 3)

Source: In part and paraphrased, IMVP volume 1, 2002, examples and comments added.

Option A

A-Characteristics

- Partial Measurement (key measurement), with some of the parameters (not all) can be stipulated instead of measured.

M/V Option	Description	Examples
Option A Retrofit Isolation with Key Parameter Measurement	This option is based on a combination of measured and estimated factors. Measurements are short-term, periodic, or continuous and are taken at the component or system level, for both the baseline and the retrofit equipment. Measurements should include the key performance parameters that define the energy use of the energy conservation measure. Estimated factors are supported by historical or manufacturers' data. Savings are determined by means of engineering calculations of baseline and reporting period energy use based on measured and estimated values.	Lighting retrofit projects. The key parameters are the power draws of the baseline and retrofit light fixtures. The operating hours are estimated based on facility use and occupant behavior. Energy savings are calculated as the difference in power draw multiplied by the operating hours.
Option B Retrofit Isolation with All Parameter Measurement	This option is based on short-term, periodic, or continuous measurements of baseline and post-retrofit energy use (or proxies of energy use) taken at the component or system level. Savings are determined from analysis of baseline and reporting period energy use or proxies of energy use.	Installation of a variable speed drive and associated controls on an electric motor. Electric power is measured with a meter installed on the electrical supply to the motor. Power is measured during the baseline period to verify constant loading. The meter remains in place throughout the post-retrofit period to measure energy use. Energy savings are calculated as the pre-retrofit energy use (adjusted to correspond to the length of the reporting period) minus the measured energy use during the reporting period.

Figure 9-26. M&V Option Application (Source 1)

Source: IPMVP-2002/DOE

M/V Option	Description	Examples
Option C Whole Facility Measurement	This option is based on continuous measurement of energy use (such as utility billing data) at the whole facility or sub-facility level during the baseline and post retrofit periods. Savings are determined from analysis of baseline and reporting period energy data. Regression analysis correlates energy use with independent variables such as weather and occupancy. Requires a detailed inventory of all equipment included in the meter reading as well as knowledge of equipment use patterns, building occupancy, and other factors affecting energy use.	Gas boiler replacement. Using natural gas billing data for 12 months during the baseline period, a baseline regression model is developed of monthly natural gas use with heating degree days. The baseline model determines baseline gas use in a typical year. During the reporting period the baseline model is run with current reporting period heating degree days, and predicted natural gas usage is compared to actual usage. Savings are defined as the normalized baseline gas use minus the normalized reporting period gas use.
Option D Calibrated Computer Simulation	Computer simulation software models energy performance of a whole facility (or sub-facility). Models must be calibrated with actual hourly or monthly billing data from the facility. Inputs to the model may include facility characteristics; performance specifications of new and existing equipment or systems; engineering estimates; spot, short-term, or long-term measurements of energy use of system components; and long-term whole building utility meter data. After calibration, savings are determined by comparing a simulation of the baseline with either a simulation of the performance period or actual utility data.	Retrofit involving multiple interactive conservation measures in a large building. A simulation model of the building with baseline equipment is calibrated to a minimum of 12 months of utility billing data. The baseline model is used to determine baseline energy use in a typical year at the site. Retrofit measures are implemented in the simulation model, and the model is run to estimate the post-retrofit energy use in a typical year. Energy use is determined as baseline energy use minus reporting period energy use. Spot measurements of equipment are made during the performance period to ensure that equipment performance conforms to the parameters used in the model.

Figure 9-26. M&V Option Application (Source 1)

BASELINE METHOD	APPLICATION
Measure equipment full load efficiency before, and after, and then assume an annual load profile to calculate before and after usage	Large, identifiable points of energy use, such as chillers, boilers, large banks of lighting, etc. Note that this method disregards ancillary equipment and part-load performance.
Average building use for 10 days prior, and compare that average to the 'test day.' Possibly eliminate wild-card days such as Mondays, Fridays, pre/post-holidays, other special considerations.	Demand limiting
Normalize building energy usage for the test period against the same period one year prior, adjusting usage based on occupancy data and weather data	Building heating and cooling modifications where weather and occupancy have a strong influence on energy use
Normalize manufacturing energy use for the test period against a different period of same duration, adjusting usage based on manufacturing product throughput and days of production.	Processes dependent on manufactured product volume, and independent of weather
Computer modeling of alternate building designs, with annual energy estimating, to show differentials, either in magnitude or proportion.	This is often done to determine preferred system designs

Figure 9-27. M&V Option Application (Source 2)
"Maximizing Energy Savings with Energy Management Systems," *Strategic Planning for Energy and the Environment Journal*, Winter 2005.

A-Strength
- Stipulated schedules/load profile are immune to changes in these items
- Reduced cost when measurement is reduced

A-Weakness
- Costly if lots of measures
- Interactions can be hard to measure
- Doesn't do well with diffuse measures like behavior or things like "replace windows"

A-Good Fits
- Where the focus is on single item or group of like items (repetitive)
- Savings too small to be detected by Method C
- Interactions between measures are minor or easy to measure or account for
- Independent variables affecting energy use are not complex
- Uncertainty of stipulations is acceptable
- The continued effectiveness of the measure can be evaluated by routine inspection of stipulated parameters
- Where stipulation is less costly than measurement (Method B) or modeling (Method D)

Savings = (stipulated hours and load profile)
*** (delta kW/cfm between old/new compressor, efficiency map at various loads)**
+/- any adjustments

A-Good Fit Examples
- Lighting replacement, three types, various uses but the uses are consistent and can be defined and agreed upon (stipulated)
  Savings is ΔkW * stipulated hours for each type of light
- Air compressor replacement, varying load throughout the year. Load profile can be stipulated.
  Savings is ΔkW * stipulated hours for each value of load

Option B

B-Characteristics
- *Full* measurement
- Isolates the measure and measures it. "Just It." Can be one measurement or many.

- Key is fully measured, and separate from everything else. Measurement period can be short or long, depending upon how steady usage will be. For a load that was steady but now is variable, the measurement must be left in place for the full period.

B-Strength
- Good for single measure, e.g. "boiler" or "lights"
- Good certainty (not a lot of guessing)

B-Weakness
- Costly if lots of measures
- Interactions can be hard to measure
- Doesn't do well with diffuse measures like behavior or things like "replace windows"
- "Before" measurements often not convenient
- Difficult if usage pattern varies before/after

B-Good Fits
- Where measurement is convenient
- Where the focus is on single item or group of like items (repetitive) with usage pattern not likely to change (simple equipment replacement projects)
- Savings too small to be detected by Method C
- Interactions between measures are minor or easy to measure or account for
- Independent variables affecting energy use are not complex

B-Good Fit Example
- New chiller. Baseline is established by performance testing the old unit prior to removal for a map of efficiency vs. load for various levels of cooling load. Measurements are power input and cooling load output. Measurement continues with the new chiller (same instruments) for one full season or cycle. Measured load profile determines what the electrical usage would have been with the old unit.

Savings = electric usage (would have been)
- electric usage (new)
+/- any adjustments

Option C

C-Characteristics
- Measures the whole building
- or all of a given sub metered building

C-Strength
- “Before” readings are convenient (utility data)
- Captures interactions as bottom line result
- Captures diffuse measures

C-Weakness
- Unable to discern between individual ECMs
- Small savings get lost in noise of uncertainty (ECMs need to be 10% or more of annual usage)
- Limited to buildings that can be normalized for weather, occupancy, production, etc.
- Difficult with large facility changes

C-Good Fits
- Complex equipment replacement and controls projects
- Variety of different ECMs in one building
- Savings are large enough to overcome noise of the whole meter
- Diffuse ECMs that can’t be easily isolated
- Interaction is substantial making Methods A/B prohibitive
- Where a good association exists for predicting energy use from independent variables (weather, occupancy, production, etc.)
- Large changes in the facility won’t occur during the measurement period

C-Good Fit Example
- Multiple ECMs in a building: lights, HVAC, controls. Measures interact. Reasonable R^2 for weather dependence. Building use is steady.

Baseline period (season) regression model applied to performance period conditions identifies what energy use would have been with no ECMs.

Savings = energy use (would have been)
- energy use (in the reporting period)
+/- any adjustments

Option D

D-Characteristics
- Uses a computer model. Proof is from a computer output, not measured

D-Strength
- Uses common –stipulated-schedules, weather, occupancy so these influences are neutralized
- Stipulated conditions are immune to large facility changes

- Creates "before" energy use as well as after
- Able to isolate measures (save-as, run again)
- Captures most interactions and diffuse ECMs

D-Weakness

- It's not real data – accuracy depends on the modeler, assumptions, and thoroughness
- Costly to model whole building for small ECMs

D-Good Fits

- New construction projects
- "Before" data is not available or is problematic (no baseline data)
- Quantity of ECMs makes Methods A/B impractical
- Diffuse ECMs
- ECM interaction not easy to measure and overall savings too small for Option C
- Large number of ECMs needing to be identified individually, which Method C cannot do
- Large changes in the building or process are expected during the measurement period that will be hard to adjust for in Method C

D-Good Fit Example

- Multiple ECMs in a building: lights, HVAC, controls. Measures interact. Reasonable R^2 for weather dependence. Searching for specific results by individual ECM.

Simulation model during the performance period with ECMs installed, is calibrated to actual utility u sage data. Run the model with standardized weather and conditions. Repeat the model run with ECM items exchanged for prior equipment efficiency, to identify what energy use would have been.

Savings = energy use (would have been)
- energy use (with ECMs)
+/- any adjustments

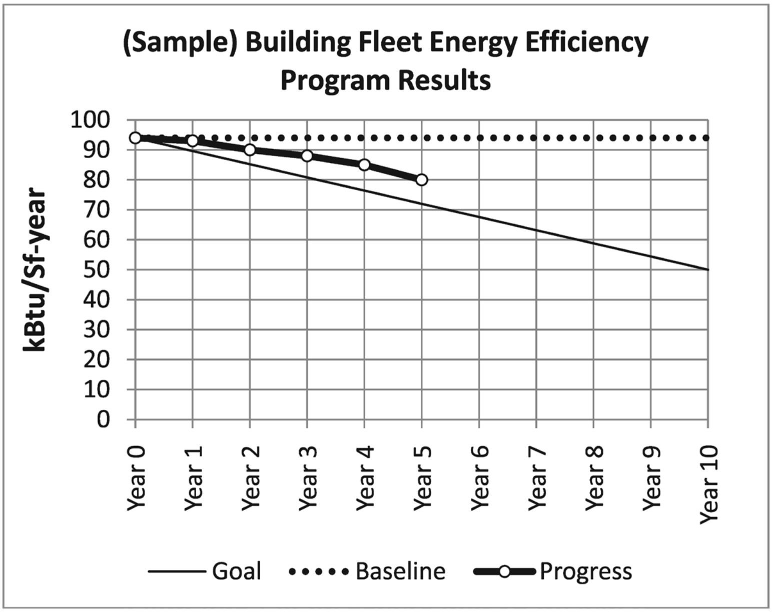


Figure 9-28. Sample Energy Performance vs. Baseline

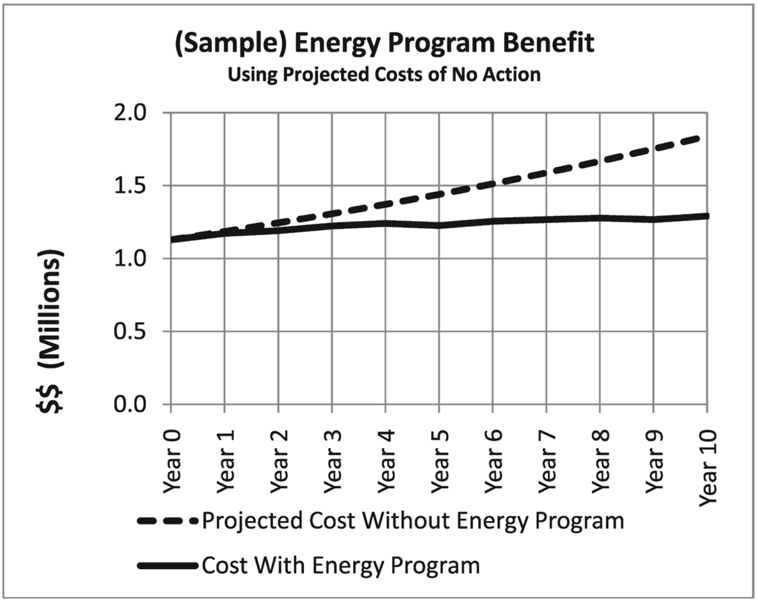


Figure 9-29. Sample Energy Program Cost Benefit
"What energy cost would have been"

Box Method

Evaluating the efficiency of a single equipment item does not explain the overall efficiency of the system to which it is connected. For example, an 80% efficient steam boiler connected to 1000 feet of bare piping may deliver 50% of its heat to the end source that it exists to serve. The 'box' method merely extends a control volume beyond one element to include others, to aid in the energy efficiency analysis and quest to find meaningful improvements.

The boundary can be wherever you want it to be. **A very useful application of the box method is to compare fuel input to the overall beneficial end use for overall process efficiency.**

Example 1:

- Instead of viewing only fuel input to a boiler vs. stack loss (combustion efficiency),
- Fuel input vs. heat output from the boiler (captures casing loss),
- Fuel input vs. plant export output (captures casing loss, and losses within the plant such as pipe and valve losses, pump casing heat, deaerator loss)

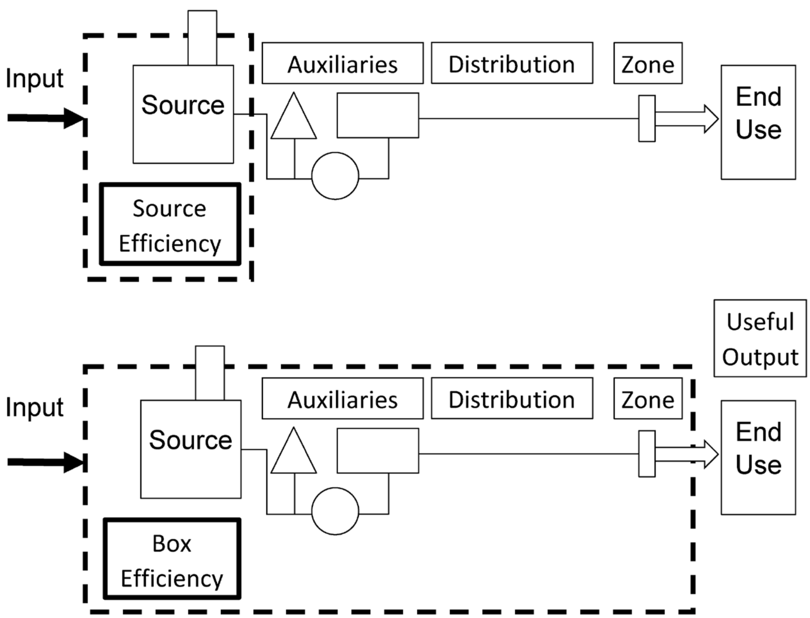


Figure 9-30. Box Method Used to Identify Overall Practical Efficiency

- Fuel input vs. delivered heat at the end use. (End-to-end efficiency. Captures casing loss, plant loss, distribution loss, condensate leaks, steam leaks, trap leaks)

    If the sum total of delivered heat from all heat exchangers and air coils is 1MMBtu/year and the sum total of fuel input to the boiler is 1.65MMBtu/year, the overall heating efficiency is output/input = 60%. In summer, idling boilers and constant distribution losses serving only minor heating loads, if heating output for summer is 0.15 MMBtu vs. 0.5MMBtu fuel used, or 30%. Meanwhile, for both cases, the combustion efficiency may be stable at 80%.

Example 2:

- Instead of viewing only electrical input to an air compressor vs. SCFM output,
- Electric input vs. compressed air to the plant (captures desiccant air loss)
- Electric input vs. compressed air used by the end devices (captures leaks)
- Electric input vs. work done at the end devices. (End-to-end efficiency)

    If the amount of work done by a pneumatic motor drill is 100 Hp and the air serving it comes from a 750 HP compressor, the conversion is 13% efficient. Another system serves air actuators in a factory; the 100 Hp compressor produces 400 scfm of air flow but 15% is lost to desiccant dryer purge and another 25% is lost to leaks – only 255 SCFM are being put to beneficial use, so the conversion is 2.55 cfm/hp. Meanwhile, for both cases, the compressor efficiency may be a stable 4 cfm/Hp.

There are many examples of using the box method. The point of each is to maintain a good overall view of a process and improves understanding of where energy goes and, then, what to do about it. Part load losses and distribution losses are quantified in this way.

- Example 1 might suggest distributed heating during summer and turning off the main plant, insulation of distribution piping, steam trap repair
- Example 2 might suggest an electric drill instead of a pneumatic drill, and remedial work to the dryer controls and leaks

Other examples where the box method can bring the big picture into view

- Looking at what motors are driving for efficiency gains rather than the motors themselves; i.e. including end uses and not just equipment
- Efficient lights left on too long
- Efficient boiler serving a building in a cold climate with single pane glass and no attic insulation
- Efficient chiller operating at unusually low temperatures for one zone that needs it
- Efficient circulating pump operating with negligible load (low delta-T)
- Ideal energy use for a manufacturing process unit operation vs. actual energy input

MACRO BASELINES—FOR GOALS AND PROJECTIONS

For a fleet of buildings, large campus properties, etc., it may be desirable to utilize a macro approach, measuring energy use on an overall basis against a baseline calendar year. While this is less accurate than the micro view, useful trends can be identified. Overall energy use per SF figures can be used to show progress toward established over-arching energy efficiency goals such as "30% less energy use intensity compared to the baseline year" (See **Figure 9-28**). With baseline year overall energy use per SF and utility cost escalation estimates, actual vs. projected energy use can be converted to actual vs. projected cost. This macro view shows avoided costs, e.g. what would have been, as the tangible value of having an energy management program. See **Figure 9-29**.

Chapter 10

Sustaining Savings

TENDENCY FOR INITIAL SAVINGS TO DETERIORATE

There is a significant body of evidence that initial energy savings from conservation projects tend to drift downward, eroding 10-20% of the initial savings within the first two years.

Source

1. Toole and Claridge, "Review on Persistence of Commissioning Benefits in New and Existing Buildings," International Conference for Enhanced Building Operations—Maximizing Building Energy Efficiency and Comfort, 2006.
2. "Is Commissioning Once Enough?," David Claridge et al, *Energy Engineering*, 2004.

Project Risk

Long-term payback assumptions are at risk from savings that are not persistent. It also represents risk for facilities that get funding for projects and subsequently get their operating budgets reduced by an amount equal to the initial savings. An awareness of this tendency for eroding savings is important, so that steps can be taken to mitigate it and sustain the benefits.

MAINTAINING INITIAL SAVINGS

Some ways to help prevent this backsliding are:

- Conservative calculations, however this merely 'hides' the savings.
- Measures that are immune to this; e.g. building shading elements.
- User buy-in and acceptance; measures that aren't too complicated for the people who will inherit them.
- Designs with ample provisions to encourage maintenance, especially the cleanability of heat transfer surfaces. This includes equipment that is easy to dismantle for cleaning, ready access and ample room for the work. Fouling factors in equipment selections add heat transfer surface area to the equipment and extend the service intervals.
- Clear and ample documentation so the project intent is not lost.
- Repeating training, testing, measurements and adjustments (Re-Commissioning).

- Management support is a key ingredient for success for any ECM related to a behavioral change. This facility director said it well: "To get energy standards to stick you will need support at the very highest level of the organization otherwise the doers have the responsibility without the authority and are doomed to failure."
- Once the building or process has been made as efficient as desired, identifying this as the new baseline usage and actively monitoring actual usage will provide prompts for corrective action if dysfunction or decay of efficiency occurs. In this way, the high efficiency can be maintained over time. Savings attributed to the ongoing watchdog process are equal to the avoided losses. See **Chapter 24o- Error Band: Using Energy Consumption Signatures as an Operational Control.**

CHECKLIST FOR SERVICE ACCESS AND OPERATIONS

The following was taken from a mechanical designer quality control (QC) checklist that serves to stress, from the beginning, the importance of service access for project success and sustained performance.

- User input received during the design development process, for buy-in?

- Drawings show service clearances shown with dotted lines to claim space?
  — filter and motor removal zones
  — coil pulls
  — tube pulls
  — compressor removal
  — automatic control panel access
  — above-ceiling item access
  — control valve access, etc.
  — motor controls, VFDs, etc.

- Can all equipment be reasonably and safely accessed for normal servicing? Including:
  — general inspection
  — measurement and verification
  — testing and balancing
  — filter replacement

— belt adjustment and replacement
— motor replacement
— lubrication
— coil cleaning
— fan cleaning
— control panel and control device access
— valve access
— pump seal repair
— nozzle cleaning, etc.

- Other access considerations:
  - — Access doors shown in rigid walls and ceilings for access, and are they large enough to work through ?
  - — Access doors upstream and downstream of each duct coil, duct humidifier, etc. for inspection and cleaning?
  - — Can equipment located over a lay-in ceiling be accessed from the floor by a step ladder?
  - — Permanent access to the roof for roof mounted equipment, preferably from indoors.
  - — All equipment, coils, and control valves have unions or flanges for removal.
  - — Control valves and dampers with position indication visible from floor.
  - — Thermometers, pressure gages, and indicating instruments visible from the floor.
  - — Shutoff valves and flanges/unions provided in a location to allow coil or equipment removal (e.g. not within the path of the coil pull).
  - — Shutoff valves provided to isolate all equipment.
  - — Drain, fill and vent provisions provided at all equipment.
  - — Drain, fill and vent provisions provided for piping systems.
  - — For piping that is racked (above ceilings, in chases, in tunnels, etc.) be sure each individual pipe system can be serviced or repaired. It should not be trapped by other pipes.
  - — Blank space between successive air handler coils sections for cleaning.
  - — Removable sections of insulation provided for items requiring routine servicing, such as pumps, strainers, control valves, etc.
  - — Service platforms provided for equipment on steep roofs.
  - — Service platforms provided for large or suspended equipment.

- Other owner-friendly design considerations:
  - Equipment drawn to scale? Will equipment fit in the rooms as shown ?
  - Can equipment be removed from the building after the building is built ?
  - For equipment with multiple acceptable sources of supply, have the clearances been designed to accommodate each of the substitutions ?
  - Equipment specified to be clearly labeled (tagged) to match the drawings.
  - Outside air intakes with insect screens that are removable and cleanable.
  - Water meters for system leak indication for hydronic make-up.
  - Hour meters for system leak indication for centrifugal chiller purge unit, fire protection dry pipe air compressor, etc.
  - Are schematic drawings included for each air and water system ?
  - Are schematic drawings included for each fire alarm, security, and other specialty electrical system?

- Control system acceptance and survivability features:
  - Identify the level of sophistication the operators will accept.
  - Make provisions for the system to be user-friendly.
  - Accommodate owner preferences for materials, system supplier, man-machine interface, etc.
  - Identify skill level and training needs.
  - Gages and thermometers by key instruments.
  - Test wells adjacent to pipe and duct sensors for calibration.

- Automatic control system user-friendly interface provisions:
  - Control drawings located by each panel.
  - Clear and detailed sequences of operation.
  - Means to view set point, input, and output for each control loop.
  - Means to shut off panel power/air supply.
  - Devices labeled to match the control drawing.
  - Means to temporarily override outputs for maintenance purposes.
  - DDC: Display-adjust feature or plug-in for portable interface.
  - Conventional: Gages/dials to read temperatures, pressures, etc.
  - Pneumatic: output gages marked to reflect device position.

Section II

General Information

Chapter 11

Mechanical Systems

GENERAL

In commercial buildings, on average, 25% of total energy used is HVAC. Of that, 20%-60% is fan/pump transport energy.

Source: E Source, 2004

"Right sizing" is important since over-sizing moves the operating point of percent load for the equipment to a lower value that may cause inherent inefficiency.

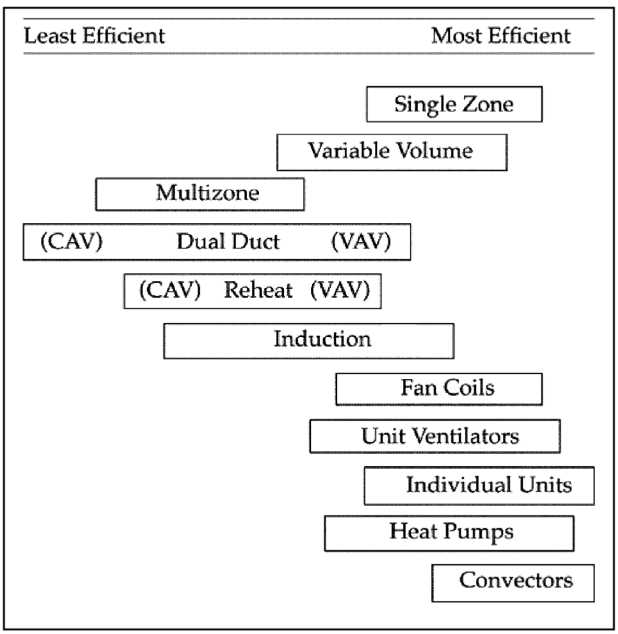


Figure 11-1. Relative Energy Efficiency of Air-conditioning Systems. Source: ***Energy Management Handbook*****, 7th Ed., Fairmont Press.**

See also **Chapter 24, Part Load HVAC Efficiency,** ***Relative Part Load Efficiency of HVAC Air System Types***

HVAC SYSTEM TYPES

Source: "Energy Conservation with Comfort," Honeywell, except where noted. Courtesy of Honeywell International Inc.

Table 11-1. Glossary of Basic HVAC System Types (*Cont'd*)

Source: "Energy Conservation with Comfort," Honeywell, except where noted. "Courtesy of Honeywell International Inc."

SYSTEM	DESCRIPTION
Single Zone System Constant Volume.	Single zone systems consist of a mixing, conditioning and fan section. The conditioning section may have heating, cooling, humidifying or a combination of capabilities. Single zone systems can be factory assembled rooftop units, or built up from individual components and may or may not have distributing duct work.
Terminal Reheat System Constant Volume.	Reheat systems are modifications of single zone systems. Fixed cold temperature air is supplied by the central conditioning system and reheated in the terminal units when the space cooling load is less than maximum. The reheat is controlled by thermostats located in each conditioned space.
Multi-zone System Constant Volume.	Multi-zone systems condition all air at the central system and mix heated and cooled air at the unit to satisfy various zone loads as sensed by zone thermostats. These systems may be packaged roof-top units or field-fabricated systems.
Dual Duct System Constant or Variable Volume.	Dual duct systems are similar to multizone systems except heated and cooled air is ducted to the conditioned spaces and mixed as required in terminal mixing boxes.
Variable Volume System	A variable air volume system delivers a varying amount of air as required by the conditioned spaces. The volume control may be by fan inlet (vortex) damper, discharge damper or fan speed control. Terminal sections may be single duct variable volume units with or without reheat, controlled by space thermostats.
Induction System Constant Volume.	Induction systems generally have units at the outside perimeter of conditioned spaces. Conditioned primary air is supplied to the units where it passes through nozzles or jets and by induction draws room air through the induction unit coil. Room temperature control is accomplished by modulating water flow through the unit coil.

HVAC System Types (cont'd)

These system descriptions are not from the referenced Honeywell publication.

Table 11-1. Glossary of Basic HVAC System Types (*Concluded*)

SYSTEM	DESCRIPTION
Fan Coil Unit Constant Volume.	A fan coil unit consists of a cabinet with heating and/or cooling coil, motor and fan and a filter. The unit may be floor or ceiling mounted and uses 100% return air to condition a space.
Unit Ventilator Constant Volume.	A unit ventilator consists of a cabinet with heating and/or cooling coil, motor and fan, a filter and return air-outside air mixing section. The unit may be floor or ceiling mounted and uses return and outside air as required by the space.
Unit Heater Constant Volume.	Unit heaters have a fan and heating coil which may be electric, hot water, or steam. They do not have distribution duct work but generally use adjustable air distribution vanes. Unit heaters may be mounted overhead for heating open areas or enclosed in cabinets for heating corridors and vestibules.
Perimeter Radiation Natural Convection	Perimeter radiation consists of electric resistance heaters or hot water radiators usually within an enclosure but without a fan. They are generally used around the conditioned perimeter of a building in conjunction with other interior systems to overcome heat losses through walls and windows.
Hot Water Converters	A hot water converter is a heat exchanger that uses steam or hot water to raise the temperature of eating system water. Converters consist of a shell and tubes with the water to be heated circulated through the tubes and the heating steam or hot water circulated in the shell around the tubes.

Table 11-2. HVAC System Types—Other (*Cont'd*)

SYSTEM	DESCRIPTION
Single Zone VAV	Low pressure cooling only system that acts like "one big VAV box" This system normally serves a single large space with constant temperature variable volume air flow in response to space temperature.
Dedicated Outside Air System (DOAS)	Outside air is treated and ducted separately from the main recirculating air system. The energy advantage to this method, when combined with a VAV system, is the ability to set VAV box minimums to zero and eliminate the built-in simultaneous heating / cooling penalty.
Spot Cooling / Spot Heating	Delivering "personal quantities" of cool or warm air to a workstation allows comfort of the occupant without the need to cool or heat the remainder of the space. Applicable to businesses with fixed workstations such as manufacturing, laundries, etc.; not applicable for businesses where workers or customers move around a lot. Cooling performance varies with air speed and is often used with high velocity diffusers.
Heat Pump	A system that recovers heat using a refrigeration cycle. Control valves in the heat pump refrigerant circuit are analogous to a package window unit that can be turned around in different seasons depending upon where the heat is needed to be added or extracted. Heat source/sink may be water, air, or earth. Heat pumps can deliver heated or cooled water or air. The term "Heat Pump Balance Point" applies to air-source heat pumps only. The outside air temperature at which the increasing heat load of the building served is just equal to the decreasing capacity of the heat pump unit. Below this value of outside air temperature, supplemental resistance heat is required.

Table 11-2. HVAC System Types—Other (*Cont'd*)

SYSTEM	DESCRIPTION
Ground Source Heat Pump (GSHP)	An array of water-source heat pumps serving individual zones are connected by a common circulating water system that is buried in the earth. The earth becomes both a heat sink (summer, heat rejection) and a heat source (winter, heat absorption). Deep burial of the loop piping creates efficiency in winter since the ground temperature at these depths is unaffected by surface weather temperatures. Care must be taken to evaluate the cyclic adding and subtracting of heat in the loop field so that long term creep of soil temperatures that surround the loop piping can be predicted and planned for. Some systems that have dominant cooling loads (rejecting heat to the earth) will rise in temperature over time and may warrant an auxiliary cooling tower to equalize the seasonal heat flux. The long term drift of soil temperature will affect loop water temperature and thus the efficiency of the heat pumps, and failing to recognize this can result in over-stated energy benefits over the system life.
Water Source Heat Pump Loop	An array of heat pumps serving individual zones are connected by a common circulating water system. During cold weather, the loop receives heat from an auxiliary heating source (boiler). During hot weather, the loop receives cooling from an auxiliary cooling source (cooling tower). The efficiency benefit of this system occurs during mild weather when some units call for heat and some call for cooling. In this range of temperatures, the heat from one zone is moved to another via the loop. To leverage this advantage, the loop temperature controls require a wide dead band to keep the auxiliary heating and cooling equipment off as much as possible.

Table 11-2. HVAC System Types—Other (*Cont'd*)

SYSTEM	DESCRIPTION
Radiant Heating	Heat transfer dominated by radiant rather than convective methods is accomplished with high surface temperatures, usually overhead. This system requires the source to "see" the object to be heated, and does not work well in the presence of obstacles that block the path between source and sink. Energy advantages: (1) Comfort is achieved for the person while the surrounding air temperature is lower than it would be for convective heating (where the system heats the air which in turn heats the person as well as everything else). The reduced indoor space temperature proportionally reduces envelope losses; (2) heat transfer occurs without distribution (fan/pump) energy.
Floor Heating	Dubbed 'radiant floor heating', this is primarily a convection and conduction heating system. The warm air buoyancy from the floor surface rises across the occupants which are presumably at floor level. Compared to overhead heating systems, more of the heat is used at the floor level than at the unoccupied higher spaces, and thus less heat is used overall. Comfort is also enhanced due to the warm vs. cold floor slabs, where people's feet touch. Part of the energy savings potential of this measure is given up to ground losses, unless ample provision is provided to insulate the slab.
Chilled Beams	The seasonal reverse of floor heating systems, chilled beams are a way to cool a space without parasitic losses from fan circulation. The cold air resulting from conduction from the beam surface creates a natural convection current where cool air falls to the space below. This has very attractive energy benefits since it eliminates fans and can operate at higher cooling temperatures, but there are limitations to consider: (1) heat densities in the space must be low , otherwise the natural convection effect will not reach the occupied level; (2) occupant density must be even within each zone and alterations to the system are not convenient; (3) comfort in cooling is influenced by air speed and so supplemental air movers (desk fans, ceiling fans) may be helpful; (4) careful attention to dew point is required to prevent condensation on the chilled beams, usually from temperature blending.

Table 11-2. HVAC System Types—Other (*Concluded*)

SYSTEM	DESCRIPTION
Under-Floor Air Distribu-tion	Rather than side-wall or ceiling air diffusers, this system provides air ducts below the floor and outlets in and amongst the occupied floor plate. By delivering the air at the floor, ventilation efficiency is greatly improved (the fraction of ventilation of air actually reaching the occupant), so less ventilation air is needed. The system also allows stratification in summer where warm air at the ceiling has no consequence and no cooling energy is needlessly spent cooling it as it is released overhead. Space planning is required to accommodate the floor cavity, but should proportionally reduce the need for overhead ceiling cavity space.
Variable Refrigerant Flow (VRF)	A direct expansion (DX) cooling system that utilizes diversity of load with an array of interconnected split systems, with 15 or more indoor units per outdoor condensing unit possible. With variable speed on one or more of the central compressors, capacity is sequenced and modulated to match load. Refrigerant and oil management is a key challenge, but will be made easier over time if oil-less technology is used. The large network of refrigerant piping also offers a large opportunity for leaks, so installation quality control and serviceability (access to pipes, valves, and fittings) is desirable. Usually ductless, these systems offer the immediate advantage of low energy transport costs from air and water distribution; the balance of the energy savings occurs during part load operation. They do not provide ventilation, so a separate system is needed for occupant ventilation. The system can be cooling-only, or heating-cooling with the addition of a third (discharge) pipe. The 3-pipe system can be used in swing weather where some zones require heating while others require cooling, with the refrigerant "recycling" the heat from one space to another. Like any split system there is no air economizer.

WATER-COOLED VS. AIR-COOLED—
See Chapter 22—Water Efficiency

THERMAL ENERGY TRANSPORT NOTES

- Rule of thumb for HVAC energy transport is that you can pump energy from point A to point B at about one fourth of the energy cost to blow it. Source: *Energy Engineering*, Vol. 102, No. 4, 2005. The basis for this is the fact that the specific heat (Cp) of water (1.0) is about four times that of air (0.24), so it takes four times as many pounds of air to carry the energy.
- The energy downfall of many district heating systems is the distribution piping thermal losses during reduced loads. These may account for 10%-15% of full load heating burden, but may account for 50% of total heating burden when in mild weather.
- Managing transport energy as a fraction of total heating and cooling energy is suggested. These include HVAC air systems, and central or district heating/cooling plants. Keeping the cost of transporting the thermal energy in proper proportion with the thermal energy itself is an obvious goal. Energy "budget" guidelines for matching these two are shown in **Chapter 24: " Facility Guide Specifications: Suggestions to Build-In Energy Efficiency**."

Note: Once constructed, the opportunities to improve on this are limited. For ***example***, if pipes or ducts are downsized to save initial cost, this is very hard to justify removing after the fact and are seldom changed.

- High distribution energy inputs (either fans or pumps) is a larger concern in cooling mode than heating mode, since almost all of the energy put into the circulator ends up in the water or air and adds directly to the cooling load, and these become parasitic losses. In heating systems, the pump/fan energy is beneficial heat.

SINGLE PASS MECHANICAL SYSTEMS

Single pass systems are among the most energy intense systems. Rather than a circulating/repeating process that sees loads change gradually, single pass systems are, in effect, always in "start-up" mode and can require very large energy inputs for extended periods. Exceptions to the high energy label are air economizers and direct evaporative cooling, both of which are 100% OA.

Approaches to reduce energy input for single pass mechanical systems:

1. **Use less**. Reduce the magnitude of the problem by reducing the amount of the flow stream. Usually the most cost effective approach.
2. **Heat recovery**. When expelled product cannot be re-used directly, salvage the heat content with heat exchange equipment.

OIL-LESS REFRIGERATION TECHNOLOGY

- A magnetic bearing itself may save 1%, but that's not the point. The big savings is that the magnetic bearing allows no oil to be used.
- The heat exchangers act "fouled" with oil on them, producing 15-25% efficiency loss over time from "oil logging." So, removing the oil can create savings equal to the avoided fouling.
- The lack of oil becomes an enabler for variable speed compressor applications, by removing the velocity constraint needed to carry oil through piping.

Oil in Evaporator	Performance Loss
1-2%	2-4%
3-4%	5-8%
5-6%	9-11%
7-8%	13-15%

Source: The High Cost of Ignoring Chiller Oil Buildup, Sine, J., RSES Journal, 2004

HERMETIC MOTOR ENERGY PENALTY

- Large motors are in the 93-95% range, so 5-7% of the input energy shows up as heat. For a compressor COP of 6, this adds ~1% energy use.
- But that one percent solves a lot of leak issues.

Single Pass System	Using Less	Energy Recovery
Make-up air for an exhaust process, e.g.: • Paint booth exhaust • Plating line • Process oven • Laboratory exhaust • Kitchen hood • Swimming pool • Locker/ shower • Boiler combustion • Batch oven heating	Use less exhaust, translating into less make-up air. Optimized /tested hoods, occupancy sensors in paint booths, kitchen hoods turned on when needed instead of all day.	Often the make-up system is paired and interlocked with the exhaust, so the air streams are in close proximity. Runaround coil, fixed or rotating air-to-air heat exchanger, heat pipe. Preheating incoming air or incoming water. Waste heat recovery for an adjacent process needing heat, although only viable if in similar proportion and occurring coincidentally.
100% outside air for HVAC indoor ventilation • Pre-conditioning units • Dedicated outside air system (DOAS)	Occupancy sensors or CO2 Demand Controlled Ventilation (DCV) to reduce the amount of outside air processed. Transfer general exhaust air to exhaust-only areas such as restroom exhaust, kitchen hoods, and projector rooms, thereby using the air and heat content twice.	Air-to-air heat exchange equipment listed above, or variants with coatings or chemical washes that can recover latent heat as well. Half or more of the energy waste can be recovered. Depends on the two air streams being in similar proportion and nearby – impractical for exhaust points that are scattered around the roof.
Domestic hot water for showers	Flow restricting shower heads to reduce demand on the water heater.	Drain-waste heat recovery
Heated or chilled wash water	Recycle and filter wash water, counter flow clothes washers and dishwashers, where the final rinse water is filtered and re-used in wash and eventually pre-wash operations, reusing much of the original water and it's on board energy.	Heat exchange to coincident make-up water stream to temper the need for new energy. Collect and transfer water to a second process such as cooling tower or scrubber make-up.

Figure 11-2. Single Pass Mechanical System Examples

CHILLERS

Chilled water systems are popular for a number of reasons:

- Water can transport a unit of cooling (or heating) more compactly in piping than with ducts.
- Suitable for remote location of central equipment, along with associated maintenance and noise.
- Cooling efficiency much higher than package rooftop equipment when water-cooled centrifugals are used.
- Scalable and flexible, to accommodate building additions and changes.

See also **Chapter 24: Chilled Water System Discussion and ECMs.**

Process	Compressor type	Compressor Sub-Type	Size range	Air-Cooled (95 degF)	Water-Cooled
Vapor compression	Positive displacement	Reciprocating	5-100	1.0 - 1.4 (COP 3.5-2.5)	0.8-0.9 (COP 4.5-4.0)
Vapor compression	Positive displacement	Screw / Scroll	50-400	1.0 - 1.4 (COP 3.5-2.5)	0.65-0.8 (COP 5.4-4.5)
Vapor compression	Kinetic	Centrifugal	200-4000		0.4-0.6 (COP 8.9-5.9)
Chemical absorption cycle.	N/A	N/A	200-2000	N/A	**COP (0.7-1.3)

Figure 11-3. Typical Chiller Efficiencies
Units are kW/ton (COP), Full Load

**Absorption units are rated in terms of COP from applied heat input. kW/ton rating does not apply to absorption units.

Centrifugal Chiller Notes

- Centrifugal chillers are popular due to compact size, high COP (high efficiency), robust nature, high reliability, and user friendly reputation.

- COP at 0.5 kW/ton (1 ton x 12,000 Btu)/(0.5 kW x 3412 Btu/kW) = 7.0

- Efficiencies fall off rapidly below 50%, unless condenser water reset is used, so staging control to keep operating load above 50% is very important.

- Chillers that can accept very cold condenser water (55 deg F) and have cooling needs coincident with reduced wet bulb temperatures such as in the Southwestern US, when it is easy to achieve colder condenser water temperatures, can achieve 10-15% annual overall cooling energy savings using condenser water reset, compared to a fixed temperature set point of 70 degrees F.

Absorption Chiller Notes

- Often very attractive if used with waste heat.

- Existing absorption chillers that use a 'new energy' heat source (not waste heat) are usually good candidates for removal, in favor of an electric chiller.

- Barriers to use are high first cost, large footprint, operational problems at part loads (best use is base load cooling), and high annual maintenance. Usually impractical if 'new' energy is used for the heat source (compare COP to mechanical refrigeration). Single effect units can run on low grade heat, and 15 psi steam. Double effect units require higher grade heat, or 100 psi steam.

- Single Effect:
  Steam Use 18-20 lbs per hour per ton
  COP, single effect (1 ton x 12,000 Btu)/18 lb x 1000 Btu per lb = 0.67

- Double effect:
  Steam Use 9-10 lbs per hour per ton
  COP, double effect (1 ton x 12,000 Btu)/9 lb x 1000 Btu per lb = 1.33

- Cooling tower water use with absorption cooling: 33% more cooling water required, 4 gpm per ton is typical.

Chilled Water System Auxiliaries

- Water-cooled chillers have auxiliary equipment that is required to make it work, such as cooling towers, condenser pumps and chilled water pumps. It is interesting to note that the efficiency rating of a chiller does not include the cooling tower fan energy or condenser pump energy. By contrast, air-cooled HVAC cooling equipment does include the energy spent in the condenser fans. Good practice will look at the entire chilled water system and not just the compressor, and will keep a proportional balance of the two.

Rule of Thumb for Auxiliaries

- Ideally, total auxiliary power would be limited to 1/10th of total power. With current design practice, a more reasonable energy budget for chiller auxiliaries is for each to have not more than 1/10th of the power requirement of the compressor. For example, if the chiller is 500 kW, the tower fan, condenser pump, and all chilled water pumping combined should not each use no more than 50 kW. Or, using cooling efficiencies and an example of 0.5 kW/ton:

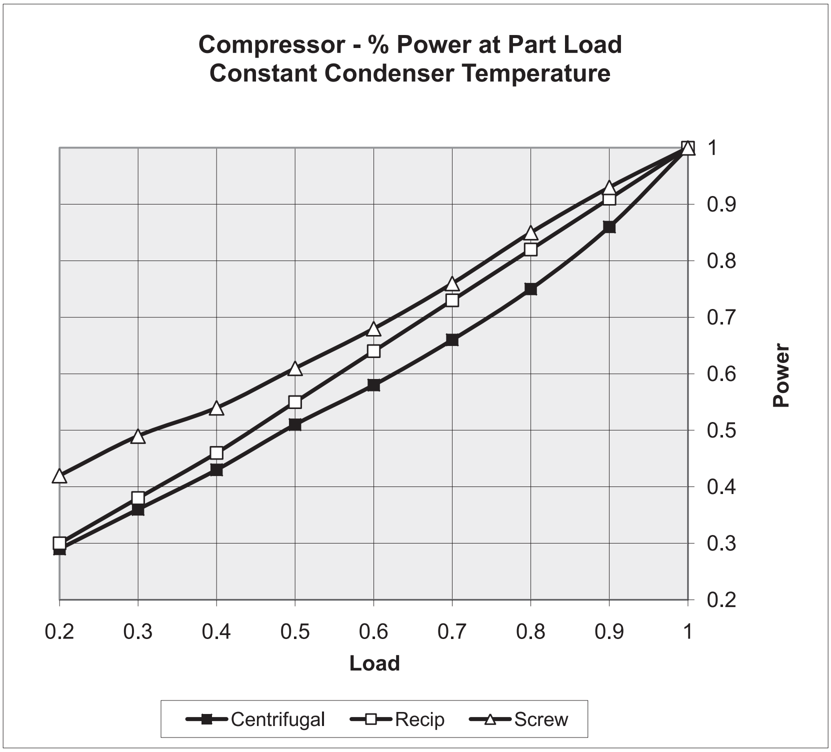
0.5 kW/ton	Compressor
0.05 kW/ton	Chilled Water Pumping (primary and secondary combined)
0.05 kW/ton	Condenser Pumping
0.05 kW/ton	Cooling Tower Fan

PART LOAD CHILLED WATER SYSTEM PERFORMANCE

- Chillers are rated at both full load and part load. The standard part load test is IPLV.

IPLV

- IPLV stands for "Integrated Part Load Value." Since chiller equipment seldom runs at full load and efficiency varies with load, a system of part-load evaluation was developed by ARI. The basis is the weighted average weather data of 29 cities across the U.S. The assumed run times at different loads for standard IPLV ratings are:



Percent Power at Part Load Refrigeration
Constant Condenser Temperature

pct load	Centrifugal	Recip	Screw
100%	100%	100%	100%
90%	86%	91%	93%
80%	75%	82%	85%
70%	66%	73%	76%
60%	58%	64%	68%
50%	51%	55%	61%
40%	43%	46%	54%
30%	36%	38%	49%
20%	29%	30%	42%

Figure 11-4. Refrigeration Compressor Percent Power at Part Load
Source of data for reciprocating and screw compressors: P.C. Koelet, "Industrial Refrigeration—Principles, Design and Applications," 1992, reproduced with permission of Palgrave Macmillan.

100% load	1%
75% load	42%
50% load	45%
25% load	12%

Source: Air-Conditioning and Refrigeration Institute (ARI) 550/590-2003

- The energy downfall of many central chilled water systems is part load performance. It is typical that the auxiliaries (pumps and cooling tower fans) make up 20-30% of the total power requirement at full load. Unless provisions are made for the auxiliary device input to throttle with the cooling load, the proportions will shift at lower loads so the auxiliaries form a much larger constituent energy user. The cooling tower is normally cycled on a thermostat or has a VFD so its energy use should track the chiller load profile. But where

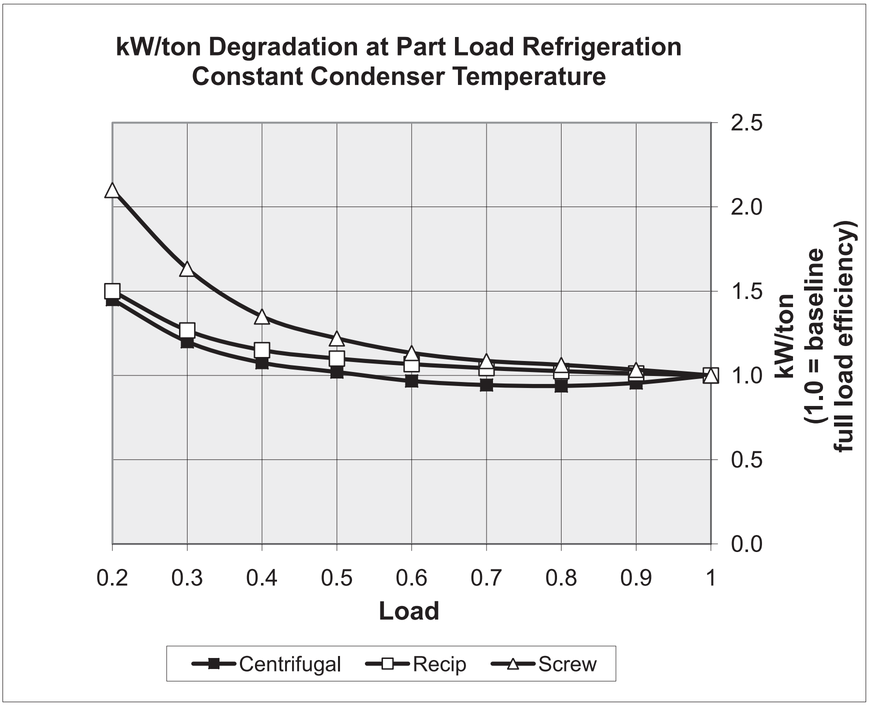


Figure 11-5. Refrigeration Compressor Efficiency Degradation at Part Load
Source of data for reciprocating and screw compressors: P.C. Koelet, "Industrial Refrigeration—Principles, Design and Applications," 1992, reproduced with permission of Palgrave Macmillan.

pumping is "constant volume" the pump energy will be constant even when the chiller load is low.

Variable Flow Pumping—Chillers

At full load, auxiliary components of a chilled water system are usually an order of magnitude less kW per ton of output cooling than the chillers. However, at part load cooling, constant flow pumps contribute a higher and higher percentage of the overall kW power consumption, and lower the overall cooling efficiency of the chilled water system.

By converting constant flow control to load-following variable flow, considerable improvements will be had during part load operation. For best economy, this will be standard practice for new designs. For retrofits,

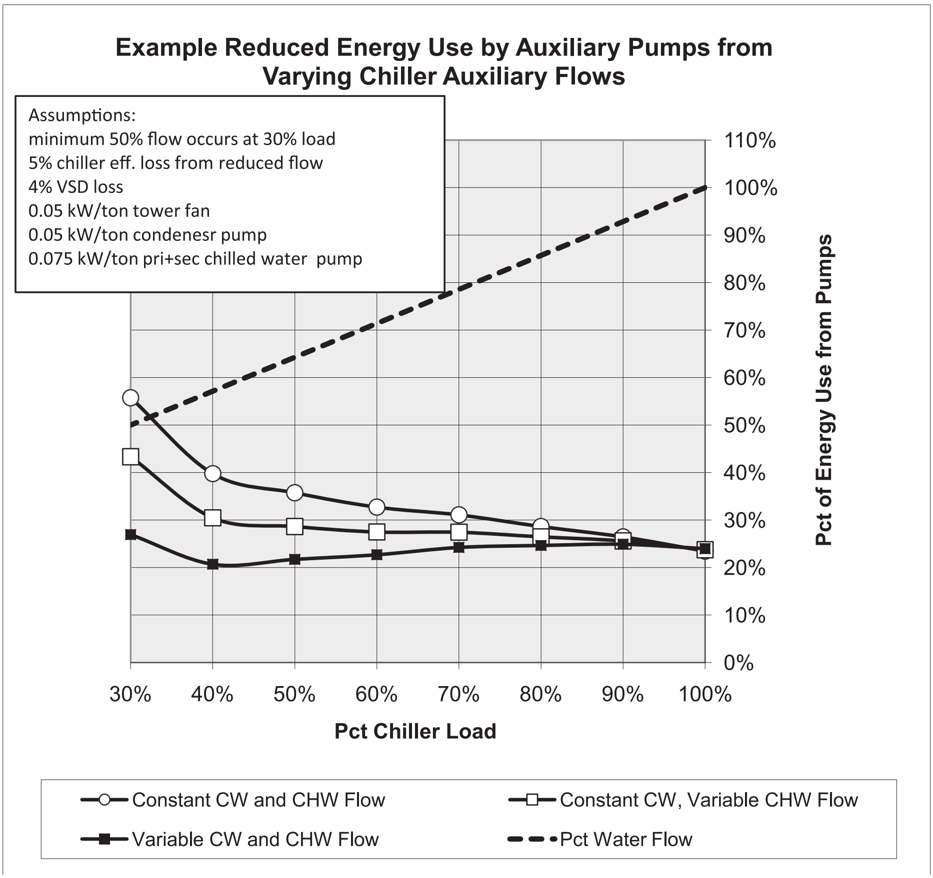


Figure 11-6. Chiller System Pump Energy Reduction from Variable Flow

there are operational considerations and this is an engineered change. For chilled water, flow control valves, air entrainment, and chiller tube velocity are considerations. For condenser water, control is usually from chiller condenser leaving temperature. In both cases, the excess flow keeps chiller heat transfer at its peak and reductions in velocity that accompany reductions in flow result in reductions in heat transfer coefficients and chiller efficiency. There is still a net gain, and the exact amount varies, but a rule of thumb is that half of the pump energy savings potential is given up as chiller efficiency loss.

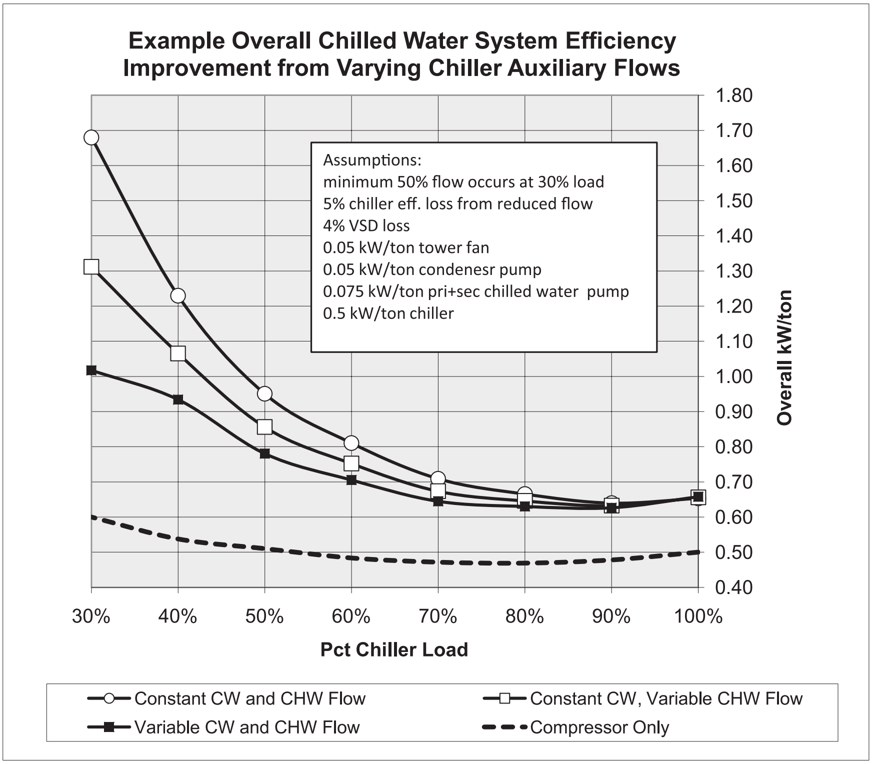
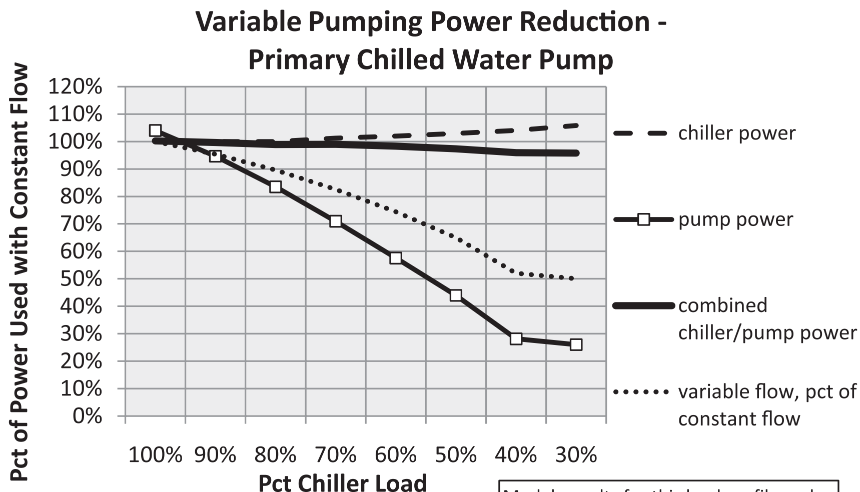


Figure 11-7. Overall Chilled Water System Efficiency Including Pumps
Assumes 0.6 kW/ton chiller at full load, and assumes half of the pump savings are given up to reduced chiller efficiency from reduced flow velocities.



Model Assumptions:
- Minimum 50% flow reduction.
- 2.4 gpm/ton full load, increasing to 3.1 gpm/ton at 50% load.
- Linear chiller efficiency penalty between 80% load(no penalty) and 30% load (5% penalty).
- 0.03 kW/ton chilled water pump full load power
- 0.54 kW/ton chiller full load power

Model results for this load profile and noted assumptions are 2% annual savings for chiller/pump combined.

Load	Hours
100%	50
90%	250
80%	600
70%	900
60%	1050
50%	1100
40%	600
30%	450

Figure 11-8. Variable Primary Chilled Water Flow—Overall Effect on Chiller and Pump Combined.

Chiller power data came from manufacturer's selection runs; each chiller is different. Chiller power goes up, and pump power goes down. By increasing the gpm/ton slightly at reduced flows, and not going below 50% chilled water flow ever, a slight overall gain is possible. In this example, 5% power reduction.

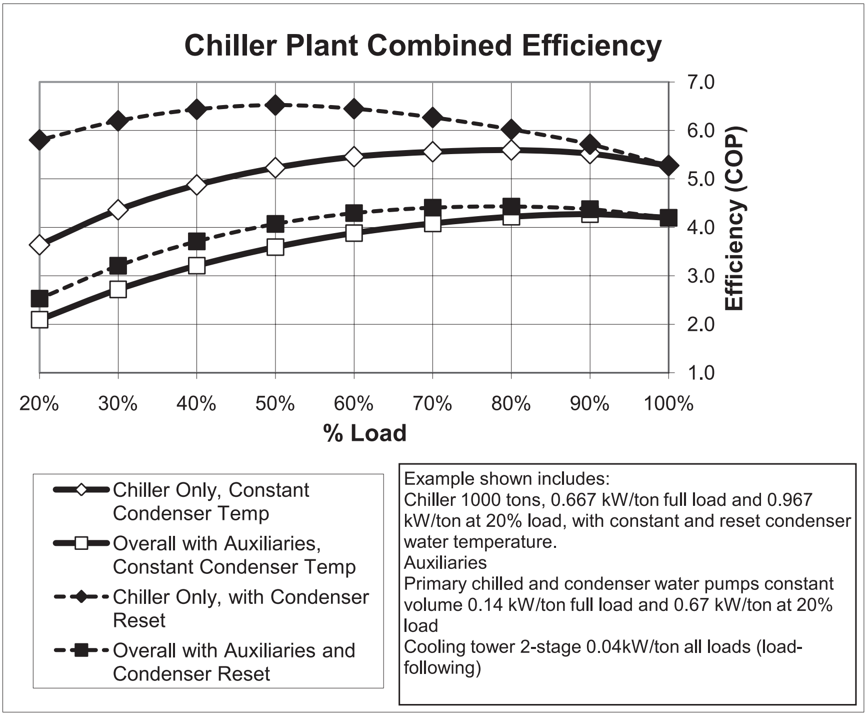


Figure 11-9. Chiller Plant Combined Efficiency with Auxiliaries Included

COOLING TOWERS AND EVAPORATIVE FLUID COOLERS

This equipment uses the cooling effect of evaporation to cool a body of water.

- Over 90% of the capacity of a cooling tower or fluid cooler is from wet bulb temperature, regardless of dry bulb temperature. In dry climates, this effect can create dramatic temperature drops in the water, but in high humidity climates the "wet bulb depression" can be small.
- A cooling tower evaporates a portion of the water to cool the remaining water.
- A fluid cooler cools indirectly through a heat exchanger, so the cooling and cooled fluids do not touch. The heat exchanger in a fluid cooler creates a barrier for heat transfer and raises the final 'cooled' temperature by 5-10 degrees, however are still widely used because the cooled

fluid can be a sealed system without distributing the mineral and corrosion properties of standard cooling tower water; heat transfer surface cleaning is limited to the fluid cooler heat exchanger.

Cooling Tower Approach

A measure of capacity of both cooling towers and fluid coolers is "approach," which is how close the leaving water temperature is to the theoretical value of wet bulb temperature. For ***example***, a cooling tower with a 10 degree approach at 60 degrees wet bulb will produce 70 degree leaving water.

- Lower approach temperatures are achieved either through increased water-air contact surface area or increased air flow.
- Increased air flow results in higher fan motor horsepower, so energy conservation would favor the use of larger body equipment to lower approach temperature instead of larger motors.
- Approach temperatures of 10 degrees are easy to achieve. Approach temperatures lower than 7 degrees result in exponential increases in cooling tower cost and are usually not warranted.
- The efficiency measure of a cooling tower is a combination of approach and fan motor horsepower. For ***example***, a cooling tower used for a centrifugal chiller will be matched in terms of tons capacity. Actually, the cooling tower "tons" capacity includes an extra measure of heat for the heat of compression and may use 15,000 Btuh/ton instead of 12,000 Btuh/ton.
- A rule of thumb for cooling towers serving chillers is to be an order of magnitude less in specific power consumption.
- Approach temperatures higher than 10 degrees or fan motors higher than 0.05 kW/ton represent lost opportunities in leveraging water cooled technology benefits.
- Using 0.5 kW/ton as a benchmark for a chiller, a performance specification for an energy efficient cooling tower would be **0.05 kW/ton, at 7 degrees F approach to local design wet bulb**.

DRY COOLERS

These are simple air coils. Normally filled with glycol for freeze protection, these units are simple and inexpensive. However, their performance is completely dependent on ambient dry bulb temperature.

A rule of thumb for leaving fluid temperature of a dry cooler is normally "Ambient Plus 30."

The energy implication of a dry cooler lies in the equipment it serves. For ***example,*** if a water-cooled refrigeration unit is connected to a dry cooler, it will see 110 degree condensing medium when it is 90 degrees outside. The energy penalty is significant compared to water that is cooled by evaporation in response to wet bulb temperature. The energy penalty for using 110 degree water instead of 70 degree water is 40-60%.

ELECTRONIC EXPANSION VALVES

Conventional expansion valves are normally sized using a 100 psi differential pressure to allow proper metering without hunting and over-feed that can damage compressors.

Electronic expansion valves can operate with less of a differential pressure across the valve without hunting and over-feeding. Eliminating this 'need' to have a high differential pressure allows the use of colder condenser water when it is available.

By replacing conventional expansion valves with electronic expansion valves, the lowest allowable operating head pressure requirement will be reduced. This measure can then be combined with reducing condenser water temperature to reduce condensing pressure and compressor power.

For example, the use of an electronic expansion valve may allow the use of 50 degree water for the condenser instead of 70 degree water, lowering compressor power 20-30 percent.

AIR AND WATER CIRCULATING SYSTEM RESISTANCE

Air Horsepower

The basic relationship for air horsepower is

$$\mathbf{HP(air) = CFM * TSP * Fa/_{6356}}$$

Where

CFM	=	cubic feet per minute air flow
TSP	=	total static pressure, in. w.c.
Fa	=	density correction factor from altitude or elevated temperature

Water Horsepower

The basic relationship for water horsepower is

$$\mathbf{GPM * HEAD * SG/_{3960}}$$

Where

GPM	=	gallons per minute
HEAD	=	total resisting pressure, ft. w.c.
SG	=	specific gravity
		water is generally taken as SG = 1.0

NOTE: Motor Hp is higher than air/water horsepower. Brake horsepower (Bhp) includes the losses from fan/pump inefficiency. System input power and energy includes the Bhp of the fan/pump, any drive losses (belt or coupling) and motor inefficiencies.

If the static pressure (TSP) or head loss (HEAD) reduction is known, savings can be estimated directly. For fittings and entrance/exit losses, dynamic factors are used and are referenced in tables published after empirical testing. These are in the form of "C" factors for air and "K" factors for water, and will vary depending upon the geometry of the fitting, the particulars of the entrance/exit condition, etc. In both cases, the dynamic losses are expressed in terms of "the number of velocity heads." The "velocity head" value multiplied by the "C" or "K" value is the anticipated loss of the item.

Velocity Head (Hv) Formulas

$$\mathbf{Hv\ (air) = (V/_{4005})^2}$$

where Hv is in. w.c.
and V is velocity in feet per minute

$$\mathbf{Hv\ (water) = (V^2/_{32.2})}$$

where Hv is ft. w.c.
and V is velocity in feet per second

The power required by the motor is greater than the fluid work because the pump or fan, drive connection and motor all incur losses. The fluid work must be divided by the product of the efficiencies of all losses along the way to determine the input requirement.

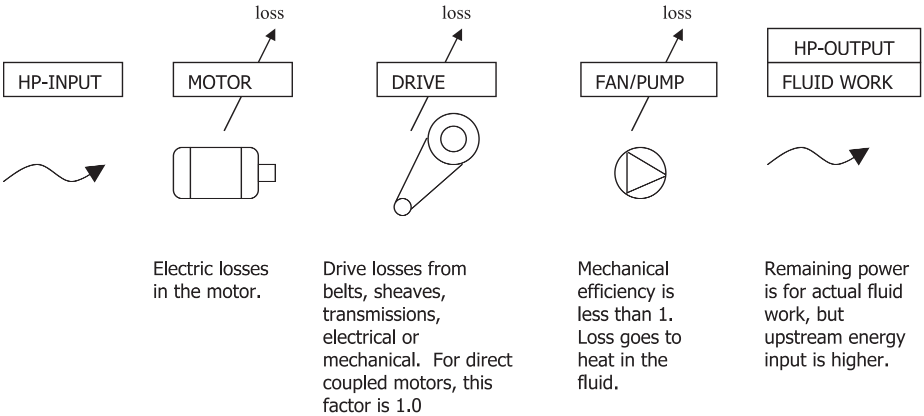


Figure 11-10. Fan/Pump Motor Work Diagram

Note: Hp-input values can be converted directly to kW and kWh.

FAN/PUMP MOTOR WORK EQUATION

$$\mathbf{WORK_{input} = (WORK\ output)/(Eff_{fan} \times Eff_{drive} \times Eff_{motor})}$$

Example: after determining the air horsepower is 0.5 Hp, find the input power given fan efficiency is 75%, drive efficiency is 96%, and motor efficiency is 91%.

Ans: Input = 0.5 / (0.75 * 0.96 * 0.91) = 0.76 Hp

NOTE: 0.76 Hp input * $^{0.746\ \text{kW}}/_{\text{Hp}}$ = 0.57 kW input

The answer is technically correct but horsepower is conventionally expressed as an <u>output</u>. Normally, the losses are combined at the motor output shaft as horsepower; then the motor efficiency loss is combined with a unit conversion for kW <u>input</u>.

Ans: Motor output = $^{0.5}/_{(0.75 * 0.96)}$

Motor input = $^{\text{output}}/_{\text{eff}}$ = 0.694 Hp

= 0.694 Hp * 0.746 $^{\text{kW}}/_{\text{Hp}}$ * $(^{1}/_{0.91})$ = 0.57 kW input.

Understanding the multiple factors that affect input energy will help yield more accurate calculations. But it is also a reminder that energy reduction gains can come from process improvements (less fluid flow, less resistance), a more efficient fan or pump, drive improvements, or driver improvements (motor).

FAN AND PUMP EFFICIENCIES

	Peak Eff. Range, %		
Device	**(1)**	**(2)**	**(3)**
Centrifugal Fan			
Airfoil	75-85	88	79-83
Backward Inclined	70-80	80	79-83
Forward Curved	60-65	70	60-65
Axial Fan			
Vane Axial	75-85	86	78-85
Tube Axial	65-75	75	67-72
Propeller	45-50	55	45-50
Centrifugal Pump			
End Suction	65-75		
Double Suction	75-85		

Figure 11-11. HVAC Fan and Pump Typical Efficiencies
Source: (1) author experience
(2) Selecting Efficient Fans, Murphy, J., 2010
(3) Bureau of Energy Efficiency (BEE), India, 2004

Related Information:
Motor Efficiencies**: Chapter 12—Motors and Electrical Information**
Drive Efficiencies: **Chapter 15—Fan and Pump Drives**
Throttling Method Efficiencies: **Chapter 15—Fan and Pump Drives**

THERMAL BALANCE CONCEPT FOR BUILDINGS

Buildings are like boxes. They have thermal losses and gains through the shell (envelope), and they also have appliances and activities inside that generate heat inside the box. There will be a point where the envelope loss just matches the internal heat gains and neither heating nor cooling is required; above this balance temperature cooling will be required, below this temperature heating will begin to be needed. This is generalized, and it is understood that there can be special areas requiring heating or cooling regardless of the bulk behavior of the building.

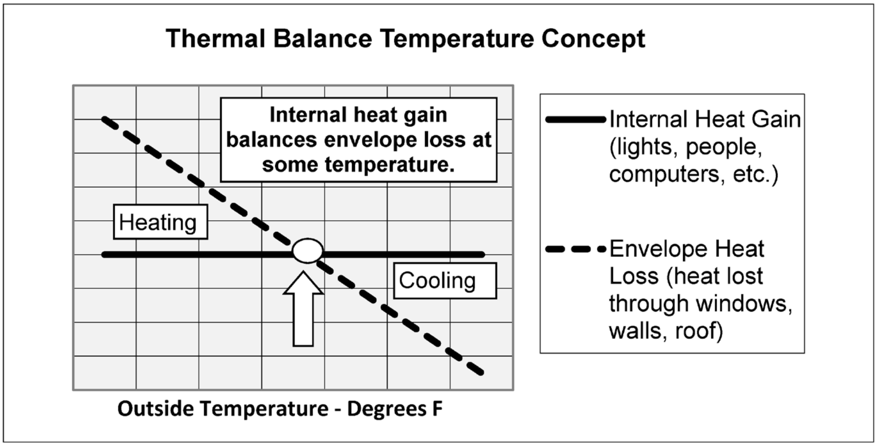


Figure 11-12. Thermal Balance Temperature Concept

Determining the Balance Temperature

Representative balance temperatures are shown in **Figure 11-15**. A method of estimating the balance temperature is located in **Chapter 17: BLC Heat Loss Method** .

Energy Implications of the Thermal Balance Temperature

Bin Weather Estimates—For most commercial buildings, the heating and cooling load is a mix of internal and envelope loads and, unless the internal loads dominate, the weather patterns are a good indicator of when heat and cool is used and to what degree. The switch from heat to cool depends upon the balance temperature, so to the extent that the break-even temperature is unknown there is unknown in any calculation of heating and cooling energy when based on weather.

Degree-Days

Degree-day calculation is based on the difference between outside air temperature and the building's balance temperature. The lower the balance temperature, the lower the heating degree-days and the higher the cooling degree-days.

Outdoor Air Economizer Savings

The lower the balance temperature, the more hours there are where cooling is needed indoors while cool air is available 'for free' outdoors at the same time. These are economizer hours.

Supply Air Reset for VAV Savings

The higher the balance temperature, the sooner VAV air flows will reach minimum and heating will become engaged for indoor comfort. Thus, for supply air reset based on outside air, the balance point determines when it will be economical to raise supply air temperature. Raising supply air temperature saves energy by reducing reheat coil loads (simultaneous heating and cooling).

Heating/Cooling Impact from Reduced Internal Loads

When there is a large amount of "self-heating," such as heavy lighting or heat-producing equipment, the space will need cooling even in cold weather-in this case the balance point between internal heat gain and envelope loss is at a low temperature. Building changes that affect the internal gains affect the balance point, and include removing heat-producing equipment (e.g. building usage change) or energy conservation measures such as more efficient lighting. Reducing waste heat from lighting reduces the "self heating" effect, moving the balance point upwards and increasing the HVAC heating system burden. When waste heat from lighting or equipment is actually heating the building, efficiency measures shifts the heating task to the HVAC system and the new energy spent on heating partially offsets the lighting savings.

In the specific case of a lighting efficiency retrofit:

(**See Figures 11-13-A,B,C.**) In climates where there is an equal amount of cooling weather as heating weather, the effects of a lighting retrofit negate. In climates where cooling dominates, reducing internal loads has a large benefit in reduced cooling that far exceeds any extra HVAC heating – and savings from the lighting replacement program are increased. In climates where heating dominates, cooling savings are minimal and extra HVAC heating is considerable—and savings from the lighting replacement program are decreased. In cold climates with marginal envelope insulation where HVAC heating is from electric resistance, the savings from a lighting retrofit can be largely erased.

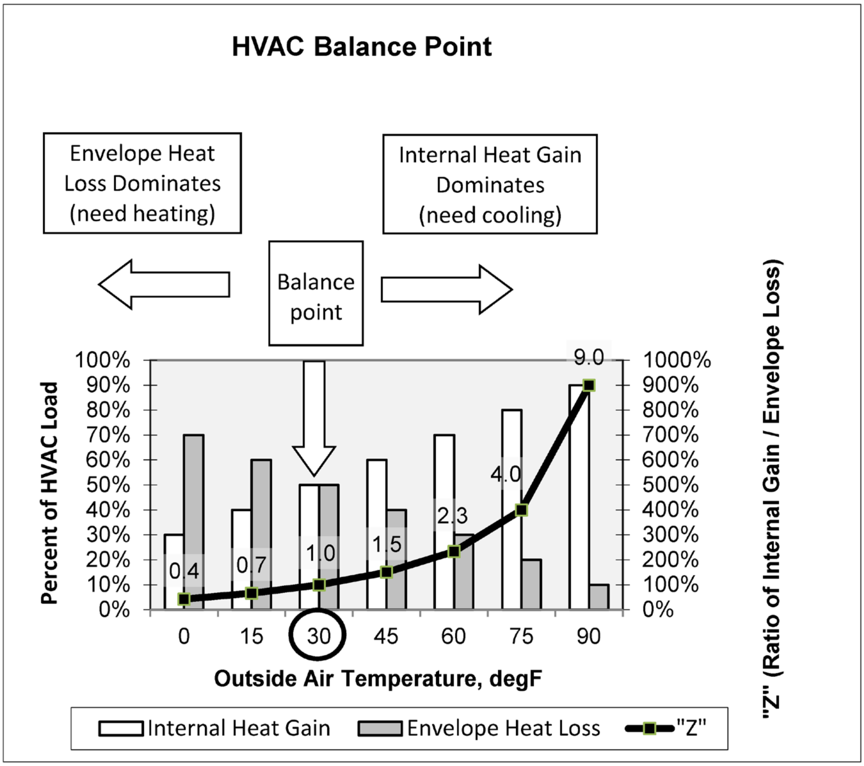


Figure 11-13A. Balance Temperature Shift from Ratio of Internal Gain/Envelope Loss @ 30F Balance

The term "Z" represents the ratio of internal loads to envelope heat loss. When this value rises above 1, cooling is needed.

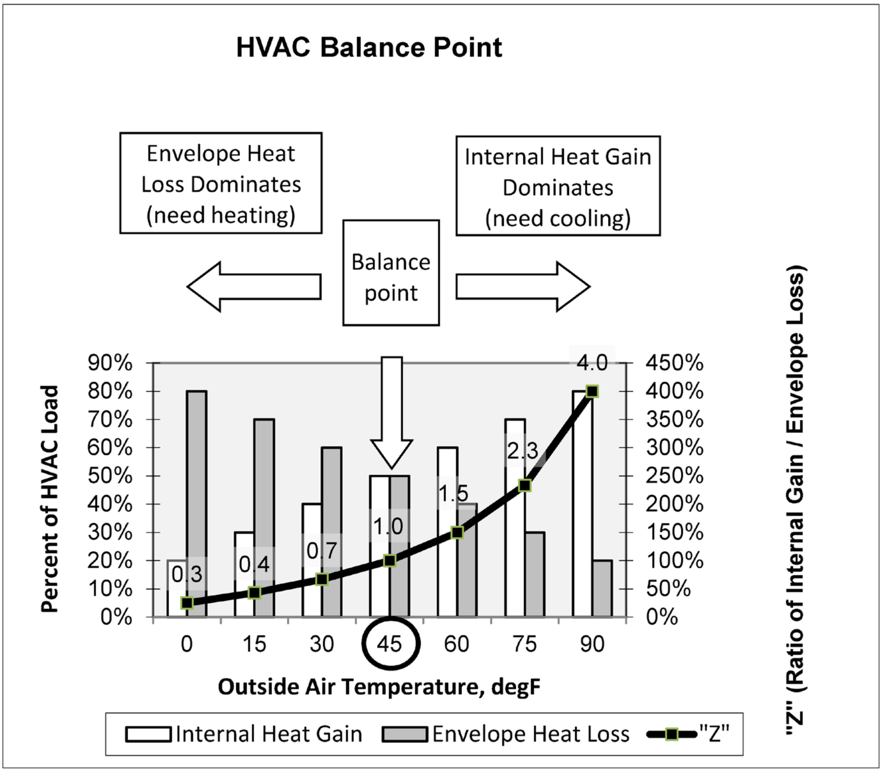


Figure 11-13B. Balance Temperature Shift from Ratio of Internal Gain/Envelope Loss @ 45F Balance

The term "Z" represents the ratio of internal loads to envelope heat loss. When this value rises above 1, cooling is needed.

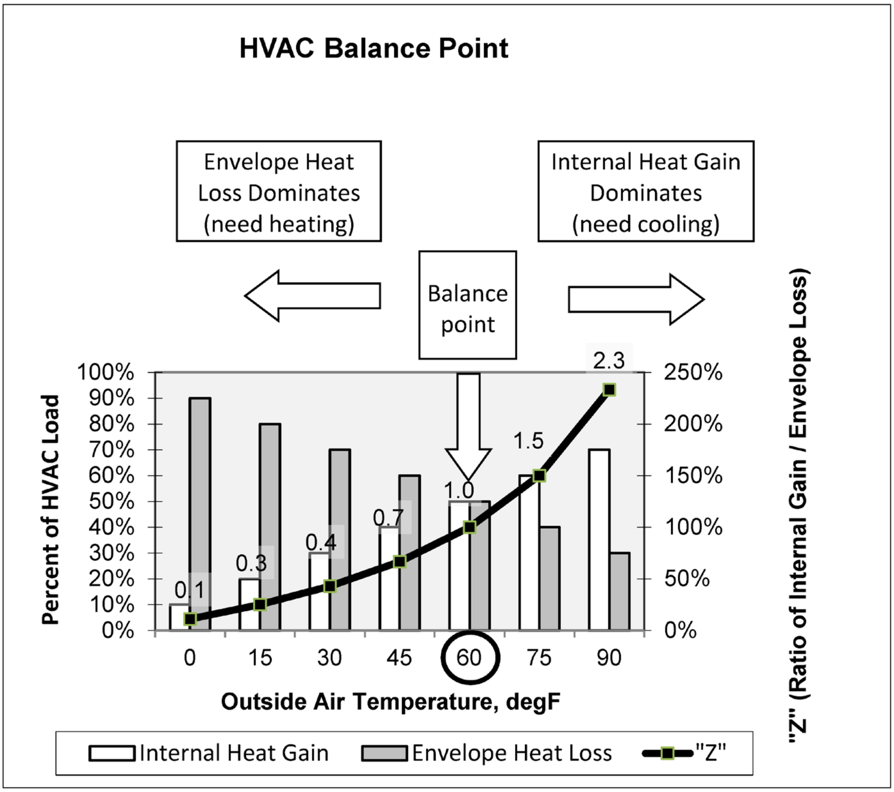


Figure 11-13C. Balance Temperature Shift from Ratio of Internal Gain/Envelope Loss @ 60F Balance

The term "Z" represents the ratio of internal loads to envelope heat loss. When this value rises above 1, cooling is needed.

AIR-SIDE ECONOMIZER

The economizer is most appropriate for thermally massive buildings which have high internal loads and require cooling in interior zones year round. **It is ineffective in thermally light buildings and buildings whose heating and cooling loads are dominated by thermal transmission through the envelope**. Economizer cycles will provide the greatest benefit in climates having more than 2000 (heating) degree days per year, since warmer climates will have few days cold enough to permit the use of outside air for cooling.

Source: ***Energy Management Handbook,* 7th Ed., The Fairmont Press**

The lower the balance point the greater the benefit of the air economizer. The key to savings in an air economizer is the number of hours that outside air is cold enough to provide cooling that occur simultaneously with the need for cooling indoors. Buildings with cooling loads dominated by envelope offer little opportunity for economizer benefit, simply because when it is cold outside it is also cold inside. But buildings with high internal loads that dominate envelope loads, the economizer is viable. Below the balance temperature no cooling is needed. A further limitation of the air economizer is the humidity level of the outside air used for cooling. For most comfort applications, any humidity in the outside air is acceptable if the air temperature is 55 degrees or lower. If the dew point of the incoming air is too high, indoor rH will increase with opportunities for discomfort and building detriment. So, there is an economizer "cutoff point" above which the system will revert to mechanical cooling.

The higher the cutoff point the more economizer hours and savings. However, raising this should be done with caution to prevent high humidity issues indoors.

The outdoor temperatures bounded by the building balance point and the economizer cut-off point are economizer hours. The HVAC load that occurs at these temperatures is usually low, but still represents avoided run time of the refrigeration equipment and makes the economizer worthwhile since it is very inexpensive to implement.

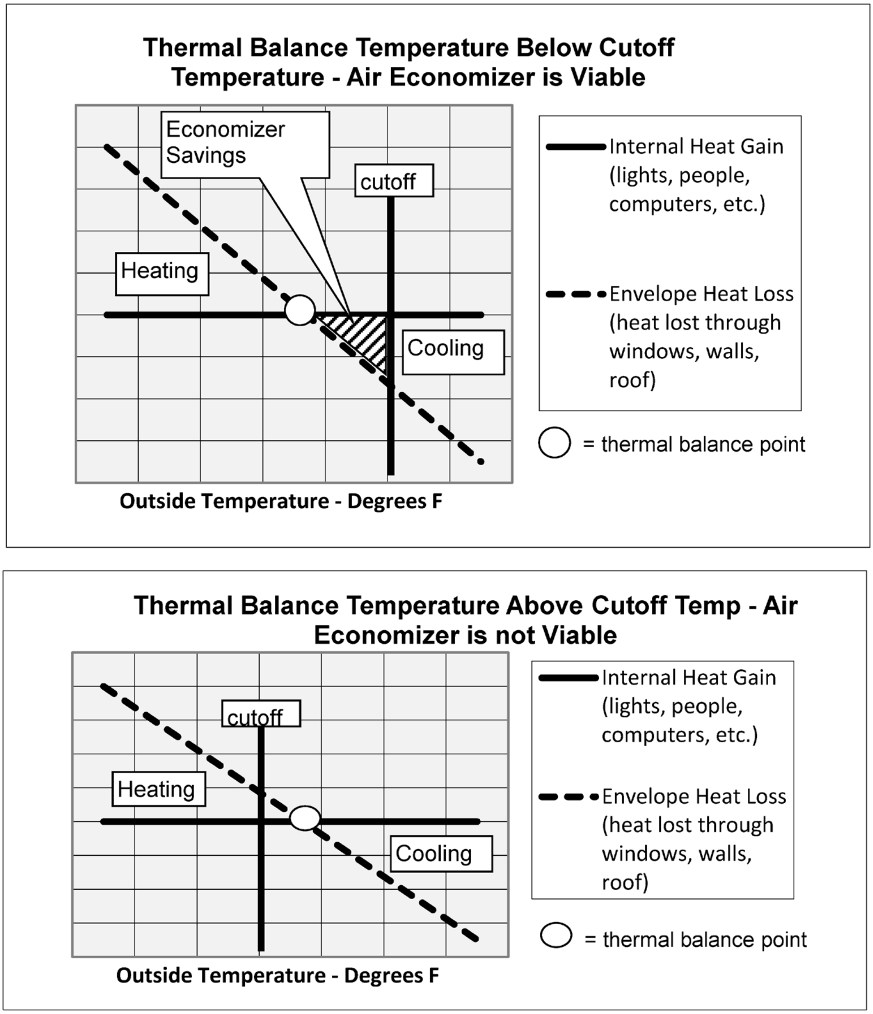


Figure 11-14. Effect of Thermal Balance Temperature on Economizer Savings

Building Type	Internal Gains or Envelope Dominates?	Approx. Balance Temp (est.)	Air-Economizer Viable?
General - Residential	Envelope	65	No
General – Buildings with a low percentage of perimeter surface area (e.g. cube shaped)	Internal Gains	Varies	Yes
General – Large building core areas	Internal Gains	Varies	Yes
General – Buildings with a large amount of perimeter envelope area compared to interior area.	Envelope	65	No
Apartments, Condominiums, Hotel Guest Rooms	Envelope	65	No
Assembly area with high people density (church, theatre, ballroom)	Internal Gains	20-40	Yes
Church building, other than busy times	Envelope	65	No
Office building interior areas	Internal Gains	40-50	Yes
Office building perimeter areas, near the glass, standard construction	Varies	50-60	Yes
Office building perimeter areas, near the glass, high performance glazing and well insulated.	Internal Gains	40-50	Yes

Building Type	Internal Gains or Envelope Dominates?	Approx. Balance Temp (est.)	Air-Economizer Viable?
Restaurant – Kitchen	Internal Gains	30-40	Yes
Restaurant – Dining Area with windows	Envelope	65	No
Warehouse that is heated and cooled, but used just for storage	Envelope	65	No
School class rooms, standard room filled with students	Internal Gains	40-50	Yes
Light manufacturing, low activity, minor interior equipment loads, sparse people loading.	Envelope	55-65	Marginal
Hot Process Manufacturing, Ovens, Baking, Cooking, etc.	Internal Gains	10-30	Yes
Computer data center, 20-100W/SF	Internal Gains	N/A	No. Most Computer Equipment is Humidity-Sensitive

Figure 11-15. Balance Temp Values

Representative Values Only

Figure 11-15. Representative Balance Temperatures for Different Building Activities. Representative values only.

See also: **Chapter 17, BLC Heat Loss Method** for a method to estimate balance temperature.

OA Temperature Degrees F	OA Dew point Degrees F	OA Relative Humidity
70 deg	47 deg	44%
69 deg	47 deg	45%
68 deg	47 deg	47%
67 deg	47 deg	49%
66 deg	47 deg	50%
65 deg	47 deg	52%
64 deg	47 deg	54%
63 deg	47 deg	56%
62 deg	47 deg	58%
61 deg	47 deg	60%
60 deg	47 deg	62%
59 deg	47 deg	64%
58 deg	47 deg	66%
57 deg	47 deg	69%
56 deg	47 deg	72%
55 deg	any	any

Figure 11-16. Maximum Relative Humidity for use in Air Economizer

Based on maintaining indoor air conditions at 47 degF dew point or less.

COOLING ENERGY BALANCE FOR HEAT PRODUCING EQUIPMENT

To balance, Q=Q. Once the Q of the equipment is known (the heat loss), the A/C cooling Q must be equal to that. The electrical input to the cooling system is then a function of the COP of the cooling system. COP = EER/3.413. The process heat load in Btu then converts to cooling input Btu, and finally to cooling electric input.

EER	kW/ton	COP	FACTOR
8	1.50	2.34	0.426
9	1.33	2.64	0.379
10	1.20	2.93	0.341
11	1.09	3.22	0.310
12	1.00	3.52	0.284
13	0.92	3.81	0.262
14	0.86	4.10	0.244
15	0.80	4.40	0.227
16	0.75	4.69	0.213
17	0.71	4.98	0.201
18	0.67	5.28	0.190
19	0.63	5.57	0.180
20	0.60	5.86	0.171
21	0.57	6.15	0.162
22	0.55	6.45	0.155
23	0.52	6.74	0.148
24	0.50	7.03	0.142

Figure 11-17. Factors for Cooling Energy Input When Paired to Known Heat Loads

- Identifies cooling loads driven by process, and also parasitic loads from cooling pump/fan delivery motor loads.
- Cooling only, not ancillary fan/pump impact unless those are included in the efficiency values of kW/ton or COP.

Process heat * Factor = cooling input (same units)
kW/ton = 12/EER
COP = 3.517/(kW/ton)
FACTOR = 1/COP

Example: 100 kW of computer heat, cooled by EER-10 equipment, 100*0.341 = 34.1 kW cooling input
Same load, water cooled equipment rated at COP 6.74, 100*0.148 = 14.8 kW cooling input

Example: 500,000 Btu of oven heat, cooled by EER-12 equipment, 500,000*0.284 = 142,000 Btu cooling input.

Using kW: 500,000/3413 = 146 kW heat load. 146*0.284 = 41.4 kW cooling input.

Example: 50 Hp fan motor serving a 100-ton packaged AC unit rated at EER-9, 50*0.746*0.379 = 14.1kW cooling load.
Note: This is 10.6 tons. Note the parasitic effect where 10% of the cooling load is consumed just in delivering the remaining cooling.

HUMIDIFIERS

Distribution method (pan, fog, atomizing, and spray) is irrelevant to energy use. The way the moisture is converted to vapor is what is important. When electric resistance or infrared light emission is used to make steam, the energy use is directly calculated at 1000 Btu/lb.; for gas heating, the calculation is the same other than flue losses and the difference in fuel cost. Adiabatic (evaporation) humidification uses an order of magnitude less energy (1/10th) for each unit of moisture compared to infrared or boiling.

Adiabatic humidifiers such as evaporative pads, atomizers, or ultrasonic get the heat of vaporization from the air and therefore cool and humidify the air simultaneously. Where simultaneous cooling and humidification are needed, such as in computer rooms; this can be leveraged for good savings, with the free cooling effect. However, if the cooling is not beneficial, the energy saved in the humidifier will be equal to the heat added to the air stream and the energy use will be equal.

Evaporative pad humidifiers are the least energy intensive of all, with only the air friction as a cost. Compressed air nozzle fogging humidifiers use considerable quantities of air and, with the compressor energy considered, are on par with Ultrasonic. Ultrasonic humidifiers and high quality fog systems require DI water and those costs must be considered.

See also **Chapter 24—Special Topics**, ***Data Center Efficiency.***

Medium	Technology	Energy Consumption	Remarks
Steam	Infrared Lamps	30+ kWh	
	Electric Resistance	30 kWh	
	Gas-Fired Steam	1.25 therms at 80%e	
Adiabatic	Compressed air atomization	3 kWh 12-15 cfm	Note 1
	Ultrasonic	2.5 kWh	Note 1
	Evaporative Pads	---	Note 1

Table Note 1: Adiabatic cooling can either help (if cooling is needed) or will require 1000 Btu of auxiliary heating per lb of evaporated moisture to compensate

Figure 11-18. Humidifier—Comparison of Energy Use per 100 lbs of Moisture

KITCHEN HOODS AND MAKE-UP AIR

Kitchen Hood Exhaust

Field constructed hoods are governed by building codes for air flow, such as 50 CFM/SF for wall-hoods and 75 CFM per SF for island hoods.

U.L. Listed "engineered" hoods use approximately 30-40% less air compared to field constructed hoods. So, for this example, a wall hood would use 30 CFM per SF and an Island hood would use 45 CFM per SF if they are an engineered hood. These hoods are an engineered package with specified air flows for exhaust and make-up stamped on the hood.

Kitchen Hood Make-up

The source of savings for kitchen hoods is to reduce the exhaust quantity if possible, since that also reduces make-up air, and to temper the make-up air as little as possible to achieve reasonable comfort, recognizing that any tempering of the air is thrown away.

- Turning off the hoods when not cooking saves energy. Often the hoods are started at 6am out of habit, or are used to cool the kitchen.
- Make-up as much as practical right at the hood so conditioned air from the space is spared. Make-up for the hood can be in front of the hood, at the lip of the hood, or short circuiting within the hood. Make-up percent can be up to 80%, with 60% being common.

HEAT PUMPS

These utilize the vapor compression cycle and equipment, but the "useful" portion of the cycle is generally the heat, which makes it reverse of standard refrigeration where the heat is rejected.

Like all refrigeration cycles, the work is determined by the lift of the cycle (difference between low and high pressures). For systems using air as a source of heat, this is an immediate disadvantage since you need the most heating when it is coldest outside and that is when the efficiency is least. Air systems also develop a coating of ice on the outdoor coils and must be periodically defrosted by a temporary reversal of the cycle and through auxiliary heating.

Heat pumps are normally used for cooling in summer, via a refrigerant "reversing valve." Using the same apparatus for both heating and cooling presents a formidable design challenge since the heating and cooling are almost always different. The result is sizing for the dominant load and having excess capacity in the opposite season. Short cycling can result and some form of capacity reduction (2-speed compressors, etc.) will usually be needed.

In cases where one half of the cycle is rejected (cooling or heating) efficiencies are the same as refrigeration equipment, plus or minus the effect of the ambient temperatures. However, in cases where the heating and cooling are BOTH used and neither is rejected the efficiencies are much higher.

Air-to-air Heat Pumps (Air-source Heat Pumps)

These are more efficient than gas heating in mild weather (40-65 degrees), but the efficiency drops rapidly with temperature and below 25 degrees are essentially electric resistance heaters below 25 degrees F. Viability is determined by annual hours of use and corresponding outdoor temperatures. Connected load is higher due to coincident compressor and electric heat operation in defrost mode.

An ideal, but probably impractical, system in moderate climates would utilize the heat pump down to 40 degrees and only use gas heating below that.

Using weather data of average temperatures and corresponding hours, savings benefit can be calculated. Assuming a 40 degree break-even point (where electric heating takes over) and a building where heating is required below 65 degrees, the following ***example*** chart shows the times of the year that an air source heat pump will have an efficiency benefit over straight electric heat. This example is for a hotel with standard PTACs that can consider the heat pump option upon replacement. Detailed analysis would be done in spreadsheet form using bin hours and consider efficiency changes at various temperatures. A similar analysis can be performed with heat pump vs. gas heat for standard furnaces.

Hybrid Air-Source Heat Pump (with Furnace)

Where envelope loads are large and single zone HVAC units are used, a combination air-source heat pump and gas (or other fuel) heat arrangement can leverage the two technologies nicely for energy savings. This can apply to small buildings, leased offices, or commercial split systems using furnaces. It works very well in a residential (lodging/dormitory) application.

The air source heat pump is extremely efficient at higher ambient temperatures (above 40 degF) but the efficiency drops off rapidly below that. If there are sufficient hours of heating above 40 degF and the cost relationship between heating fuel and electricity is appropriate, this

Figure 11-19. Example PTAC Heat Pump Benefit over Electric Heat.

Heat Pump Advantage over Electric Heat - Hotel PTAC Example (Denver Area)
Assumes Heat Pump Advantage between 40-65 degF

DRYBULB	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
0000	25.2	29.2	**40.6**	**50.6**	**59.6**	66.6	69.6	69.6	**63.6**	**53.6**	**43.2**	31.2
0100	23.9	27.9	39.3	**49.3**	**58.3**	65.3	68.3	68.3	**62.3**	**52.3**	**41.9**	29.9
0200	22.7	26.7	38.1	**48.1**	**57.1**	64.1	67.1	67.1	**61.1**	**51.1**	**40.7**	28.7
0300	21.7	25.7	37.1	**47.1**	**56.1**	63.1	66.1	66.1	**60.1**	**50.1**	39.7	27.7
0400	20.9	24.9	36.3	**46.3**	**55.3**	62.3	65.3	65.3	**59.3**	**49.3**	38.9	26.9
0500	20.7	24.7	36.1	**46.1**	**55.1**	62.1	65.1	65.1	**59.1**	**49.1**	38.7	26.7
0600	21.2	25.2	36.5	**46.6**	**55.6**	62.6	65.8	65.6	**59.6**	**49.6**	39.2	27.2
0700	22.4	26.4	37.8	**47.8**	**55.8**	63.8	66.8	66.8	**60.8**	**50.8**	**40.4**	28.4
0800	24.7	28.7	**40.1**	**50.1**	**59.1**	66.1	69.1	69.1	**63.1**	**53.1**	**42.7**	30.7
0900	27.9	31.9	**43.3**	**53.3**	**62.3**	69.3	72.3	72.3	66.3	**56.3**	**45.9**	33.9
1000	31.7	35.7	**47.1**	**57.1**	66.1	73.1	76.1	76.1	70.1	**60.1**	**49.7**	37.7
1100	35.9	39.9	**51.3**	**61.3**	70.3	77.3	80.3	80.3	74.3	**64.3**	**53.9**	**41.9**
1200	39.9	**43.9**	**55.3**	65.3	74.3	81.3	84.3	84.3	78.3	68.3	**57.9**	**45.9**
1300	**42.9**	**46.9**	**58.3**	68.3	77.3	84.3	87.3	87.3	81.3	71.3	**60.9**	**48.9**
1400	**44.9**	**48.9**	**60.3**	70.3	79.3	86.3	89.3	89.3	83.3	73.3	**62.9**	**50.9**
1500	**45.6**	**49.6**	**61.0**	71.0	80.0	87.0	90.0	90.0	84.0	74.0	**63.6**	**51.6**
1600	**44.9**	**48.9**	**60.3**	70.3	79.3	86.3	89.3	89.3	83.3	73.3	**62.9**	**50.9**
1700	**43.1**	**47.1**	**58.5**	68.5	77.5	84.5	87.5	87.5	81.5	71.5	**61.1**	**49.1**
1800	**40.4**	**44.4**	**55.8**	65.8	74.8	81.8	84.5	84.8	78.8	68.8	**58.4**	**46.4**
1900	37.1	**41.1**	**52.5**	**62.5**	71.5	78.5	81.5	81.5	75.5	65.5	**55.1**	**43.1**
2000	33.9	37.9	**49.3**	**59.3**	68.3	75.3	78.3	78.3	72.3	**62.3**	**51.9**	39.9
2100	31.2	35.2	**46.6**	**56.6**	65.6	72.5	75.8	75.6	69.6	**59.6**	**49.2**	37.2
2200	28.7	32.7	**44.1**	**54.1**	**63.1**	70.1	73.1	73.1	67.1	**57.1**	**46.7**	34.7
2300	26.7	30.7	**42.1**	**52.1**	**61.1**	68.1	71.1	71.1	65.1	**55.1**	**44.7**	32.7

PTAC heating electric heat
PTAC heat pump advantage

system can "fuel switch" with a simple outdoor thermostat; above 40 degF use heat pump, below 40 degF use fuel.

Evaluation requires three steps:

1. Energy efficiencies of the equipment choices
2. Cost of fuel choices
3. Hours and heating load for the outdoor temperatures above and below the fuel switch point

The efficiency comparison method is shown in **Figure 11-20A**.

Figure 11-20B and Figure 11-20C illustrate that fuel switching technologies depend entirely on the relative cost of the fuel choices.

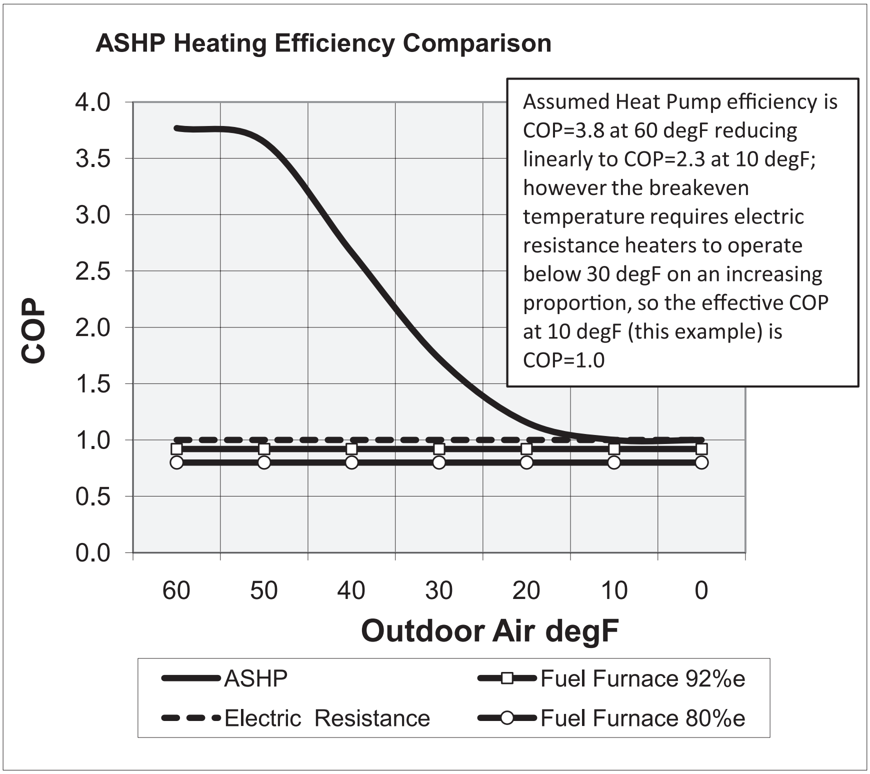


Figure 11-20A. Air-Source Heat Pump (ASHP) Efficiency Comparison.

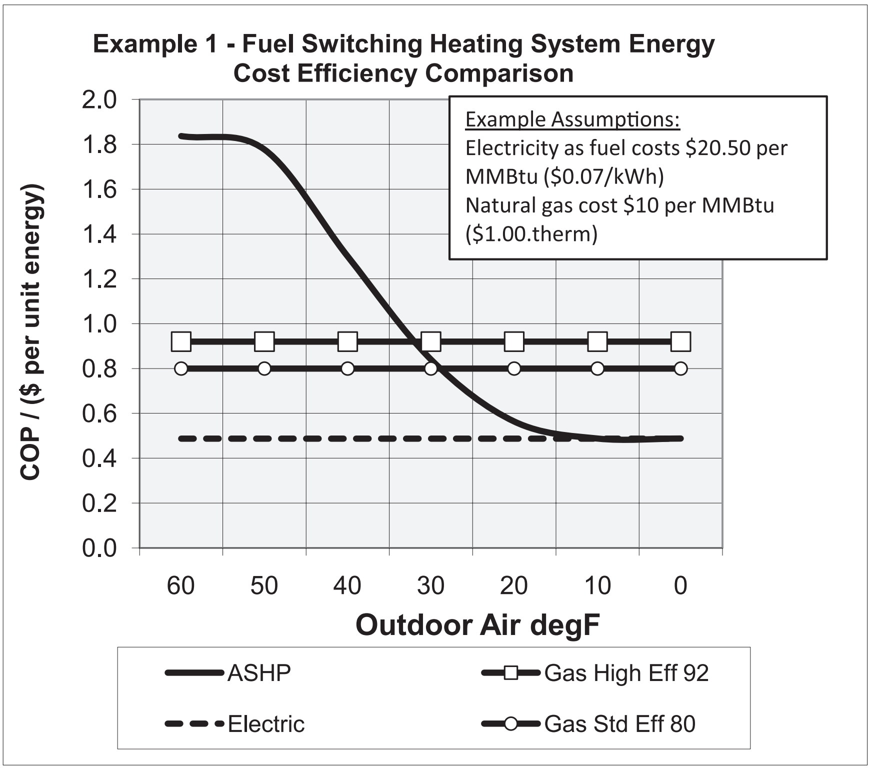


Figure 11-20B. ASHP Fuel Switching Example 1.

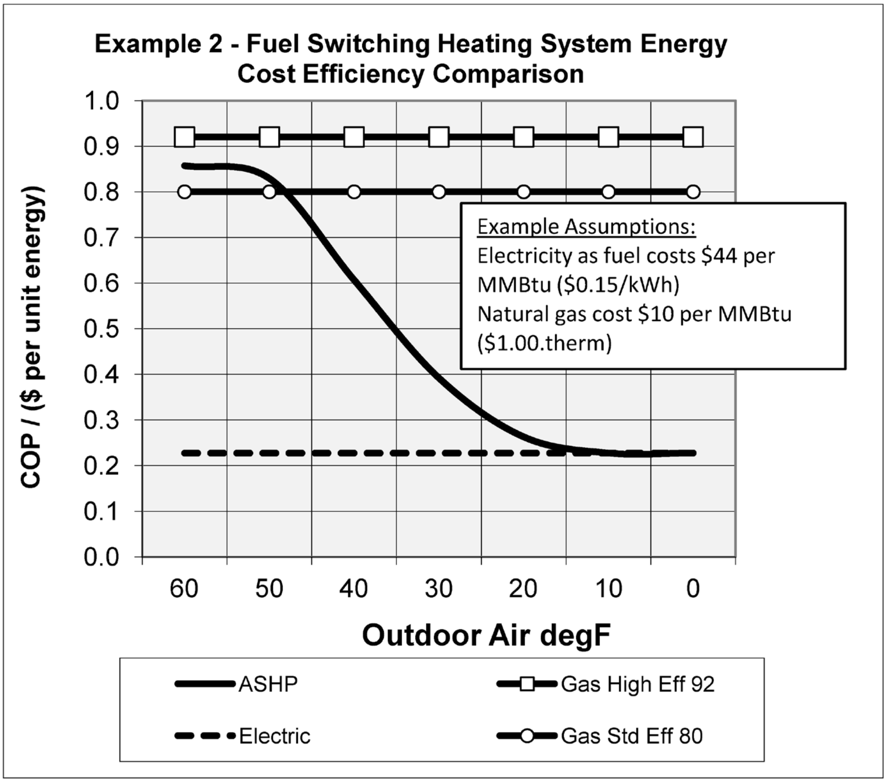


Figure 11-20C. ASHP Fuel Switching Example 2.

Water-to-air Heat Pumps (Water-source Heat Pumps)

Depending on the source of the water, these can be very useful. If, for example, a continuous source of tepid water exists with no energy input, the water supply would provide an excellent source and sink.

A common HVAC system utilizing water source heat pumps connects a number of heat pumps to a common circulating system. During moderate weather when some areas need cooling and some areas need heating this system is extremely efficient, because it is just moving the heat around the building. But during hot summer and cold winter months, this system is more energy intensive because when all the heat pumps are in one mode or the other, additional energy must be expended to stabilize the loop temperature.

With most/all zones calling for cooling, the heat pumps operate as any water-cooled air conditioner, with the heat being rejected by the fluid cooler by evaporative cooling.

With most/all zones calling for heating, the heat pumps operate in tandem with the supplemental heater, such as a gas-fired boiler. Here, the heating task is a combination of gas heating and refrigeration cycle. The delivered heat (heat pump output) is a combination of compressor power input and heat supplied by the boiler. The boiler output is less than it would be If simply heating with hot water fan coils. When compared on a Btu basis, the water source heat pump is slightly more efficient than 'plain heating'. However, cost of operation must acknowledge the cost differences between a Btu of electricity and a Btu of natural gas or other fuel. In the given example, if the ratio of electric cost to fuel cost is 4x, the system that is slightly more energy efficient could cost 15-50% more to operate.

An example is provided to illustrate the concept. **See Water-Source Heat Pump Energy and Operating Cost Example**. Actual values vary according to cost of energy sources and COPs.

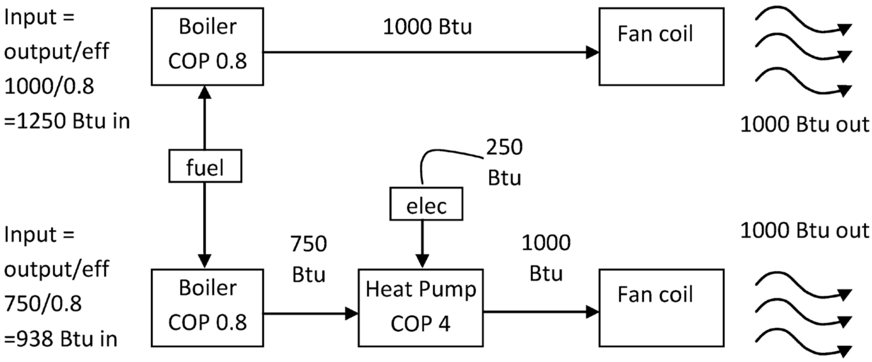
Water-to-water Heat Pumps

These have varied application. Applied to dedicated cooling areas using "house" condenser water, these have efficiency consistent with water-cooled equipment.

Excellent COPs where there exists a coincident and adjacent need for cooling and heating. In these cases, NONE of the heat is rejected on either side and the overall COP is very high as a result. An example is a need for heating hot water and cooling at the same time; using the heat pump to provide all the heating and a portion of the cooling can provide 140 degree hot water at COP=2 with free cooling and an overall COP=4.

Ground-source Heat Pumps (GSHP)

These are a variation of a Water-to-Air heat pump, using buried piping as a heat source/heat sink. Very high efficiencies are achievable with this system due to the large heat source/sink the earth provides. COP of 4.0 and EER of 17 are common with this system. Care in design and life cycle costing is recommended, to avoid over-stating long term savings, since the ground has a finite heat capacity and will "creep" up or down in temperature depending on whether the summer or winter heat flux dominates. For most commercial facilities, the cooling load will dominate and



Heat pump energy
input = output/COP = 1000/4
=250 Btu in from electricity. The rest is from the heat source, which is 750 Btu

Total input energy
HW: 1250 Btu WSHP: 1188 Btu

Total input cost for $EBtu/$Fuel Btu=2.0

HW: 1250*1 =1250 $ units
WSHP: (938*1) + (250*2) =1438 $ units

Water Source Heat Pump Operating Cost Example

For the example of COP=4 and heater eff=80%
Z=Ratio of Electric $/Btu to Heating Fuel $/Btu
HW=hot water; WSHP=water source heat pump

	HW	Water-Source Heat Pump			**Btu Basis** WSHP/HW
energy	fuel	fuel	elec	total	ratio
use Btu	1250	938	250	1188	0.95

Elec/fuel $/Btu ratio	HW	Water-Source Heat Pump			**$ Basis** WSHP/HW
Z	fuel	fuel	elec	total	ratio
1	1250	938	250	1188	0.95
2	1250	938	500	1438	1.15
3	1250	938	750	1688	1.35
4	1250	938	1000	1938	1.55

Water-Source Heat Pump Energy and Operating Cost Example

thus the ground will be expected to heat up over time, unless some form of intervention is applied to prevent this (adding cooling towers) or mitigate this (over-sizing the loop to delay the effect).

Heat Pump Coefficient of Performance

Once a COP is known at a set of conditions of lift, COP can be estimated for other lift values:

COP = C * (460 + Tcond)/(Tcond - Tevap)

C varies by machine, something less than 1, with 0.4-0.6 being common.

Tcond and Tevap vary by heat exchanger approach
Tcond is increased by the condenser hx approach;
Tevap is decreased by the evaporator hx approach

Example: A heat pump is rated at 4.0 with a high side temperature of 70F and a low side temperature of 40F. This is an air-source heat pump and a 20F heat exchanger approach is assumed for both source and sink (both indoor and outdoor coils). Find the value of COP for this machine at 10 degF

Step 1. Solve for Tcond and Tevap.
Tcond = 70+20 = 90F. Tevap = 40-20=20F

Step 2. Solve for C
43.0 = C*(460+90)/(90-20)
C=0.509

Step 3. Solve for other values of lift.
In this example, Tevap changed from 40 to 10F.
Tcond=90F, Tevap=[-10F]; lift = 100F
COP=0.509*(460+90)/ 100 = 2.8

Even this method won't be exact, since the machine efficiency can vary at higher and lower pressures and temperatures. But it will be very close within a reasonable range.

With the COP and heating output in Btuh, the heat pump input power and energy can be determined

Power, **kW = heating duty (Btu/hr) / (3,413Btu/kWh)* COP**
By bin, or annually if COP is steady, **kWh = Btu/(3413*COP)**

REFRIGERATION CYCLE

The vapor compression cycle enjoys widespread use in HVAC cooling, HVAC heating, refrigeration, process cooling, as well as many ancillary uses. From an energy standpoint, the goal is always the same. Lower the head pressure or raise the suction pressure, where possible, to reduce compression "lift" and motor kW. The typical effect of either measure is 1-1.5% improvement per degree F raised.

The challenge is to find economical ways to lower head pressures and raise suction pressures, while not creating any adverse effects on the customer's business process or equipment.

The work of the cycle is a function of the difference between the upper and lower horizontal lines in the Mollier Diagram.

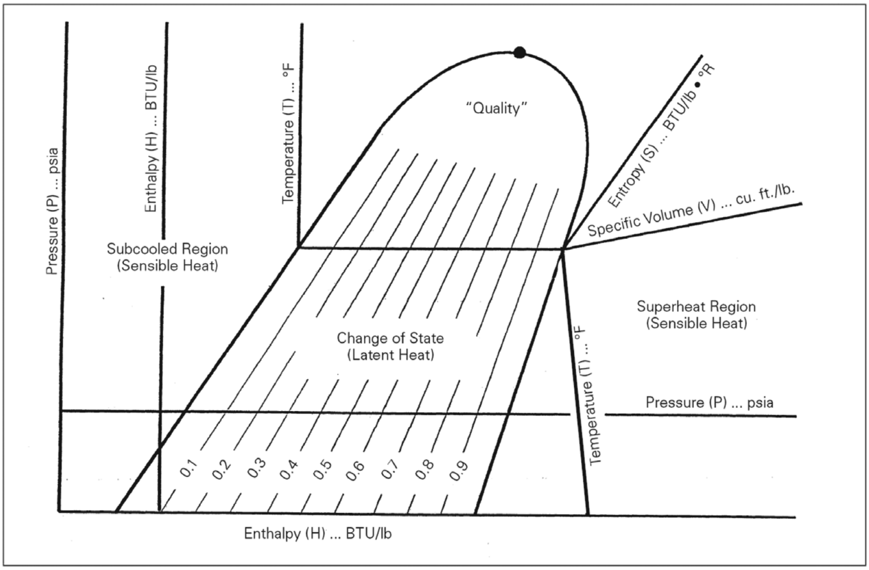


Figure 11-21. Basic Construction of the Pressure-Enthalpy (Mollier) Diagram. Source: DuPont

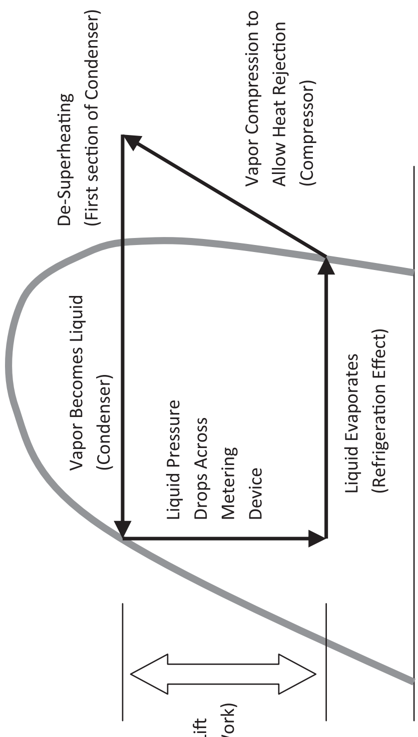


Figure 11-22. Basic Refrigeration Cycle

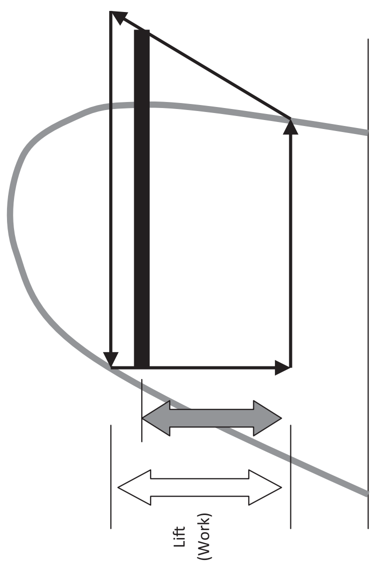


Figure 11-23. Reducing Energy by Lowering Head Pressure

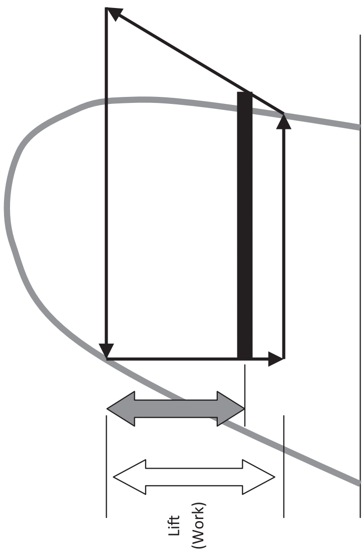


Figure 11-24. Reducing Energy by Raising Suction Pressure

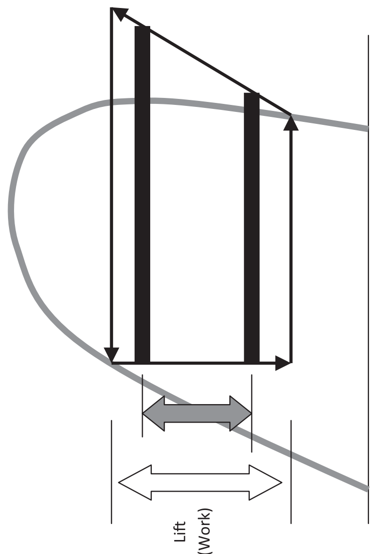


Figure 11-25. Lowering Head Pressure AND Raising Suction Pressure

EVAPORATIVE COOLING

Direct Evaporative Cooling

The air passes through the evaporative media and cools adiabatically, without a change in enthalpy. As the air absorbs moisture it cools proportionally so the total heat of the air remains unchanged. Since each subsequent pass adds moisture, this process is best suited for "single pass" applications, where the air is cooled/humidified and exhausted without recirculation. Attempts to recirculate air through a direct evaporation system risk high moisture issues in the space.

Applications for direct evaporation are kitchen hoods, factories, and other facilities where the air can be exhausted after a single pass.

Direct evaporation can also be used to advantage in air heat recovery equipment to lower the exhaust temperature and amplify the heat recovery effect without adding moisture.

The temperature of the air leaving the evaporative cooling equipment is a function of the ambient wet bulb temperature and thus the use of this technology is limited to climates where the ambient wet bulb temperature is sufficiently below the desired dry bulb supply air temperature.

Supply air for direct evaporative cooling systems is usually at somewhat higher temperature than achieved with mechanical refrigeration cooling, such as 60-62 degrees F. Consequently, increased air flow rates (cfm) are needed with correspondingly larger duct systems and fans.

The relationship for predicting direct evaporative air cooler performance is:

LATdb degrees F= EATdb- (eff*(EATdb—AMBwb))

Where:

LATdb = leaving air temperature, deg F dry bulb

EATdb = entering air temperature, deg F dry bulb

Eff = efficiency of the evaporative cooler. Range is 0.5 for felt pads to 0.9 for 12 inch thick media)

AMBwb = ambient wet bulb temperature, deg F.

Example, entering air is 65 degrees, ambient wet bulb temperature is 50 degrees, and the equipment uses 6 inch media at 80% efficiency.

Leaving air temperature = 65 – (0.8 * (65–50)) = 53 deg F leaving air temperature.

Indirect-direct Evaporative Cooling

Two sequential evaporative cooling processes provide improved cooling with less added moisture.

The first stage uses indirect cooling via a heat exchanger, sensibly cooling the air. Schematically the first stage is a cooling tower that produces water within 7-10 degrees of the ambient wet bulb temperature. The cooled water is circulated through a coil and cools it to a mid range temperature.

The second stage is a conventional direct evaporative cooling unit.

The advantage to adding the indirect step is to begin the adiabatic cooling at a lower initial temperature, thereby adding less moisture. By pairing this system with an appropriate fraction of constant building exhaust, the remainder of the air can be re-circulated without moisture damage indoors.

The use of this system is limited by ambient dew point temperatures—if operated during high dew point conditions the indoor relative humidity will rise and create discomfort for occupants.

Indirect-direct Evaporative Cooling with Supplemental Conventional Cooling

A variation of this system uses supplemental cooling to extend the effect of the first stage (indirect) cooling, thereby allowing operation in the indoor comfort zone throughout a greater range of outdoor wet bulb conditions.

Direct Evaporative Cooling—Post Cooling for Cooling Coils

Basis of savings: Evaporative cooling augments mechanical cooling.

- Effective in many climate regions, when outdoor wet bulb temperatures are sufficiently low (below 60 deg F wet bulb).

- Many HVAC cooling coils deliver 55 deg F air at or near saturation, so if the mechanical cooling is allowed to bring the air temperature to a certain point and stop, the evaporative step can do the remaining portion of cooling work with less energy and end up at the same psychrometric point.

- Can reduce mechanical cooling burden by 25%

Psychrometric Diagrams of Evaporative Cooling Processes

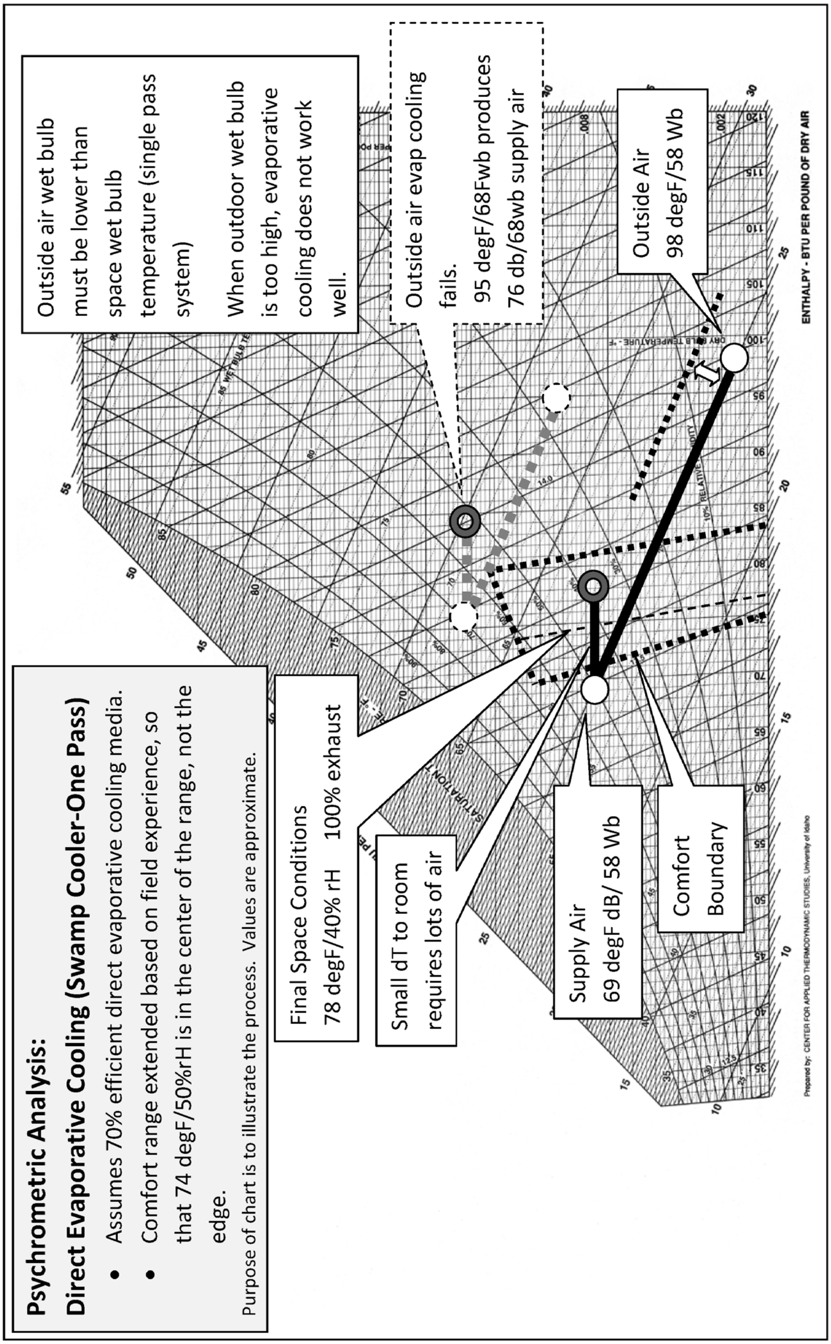


Figure 11-26. Direct Evaporative Cooling—Psychrometrics

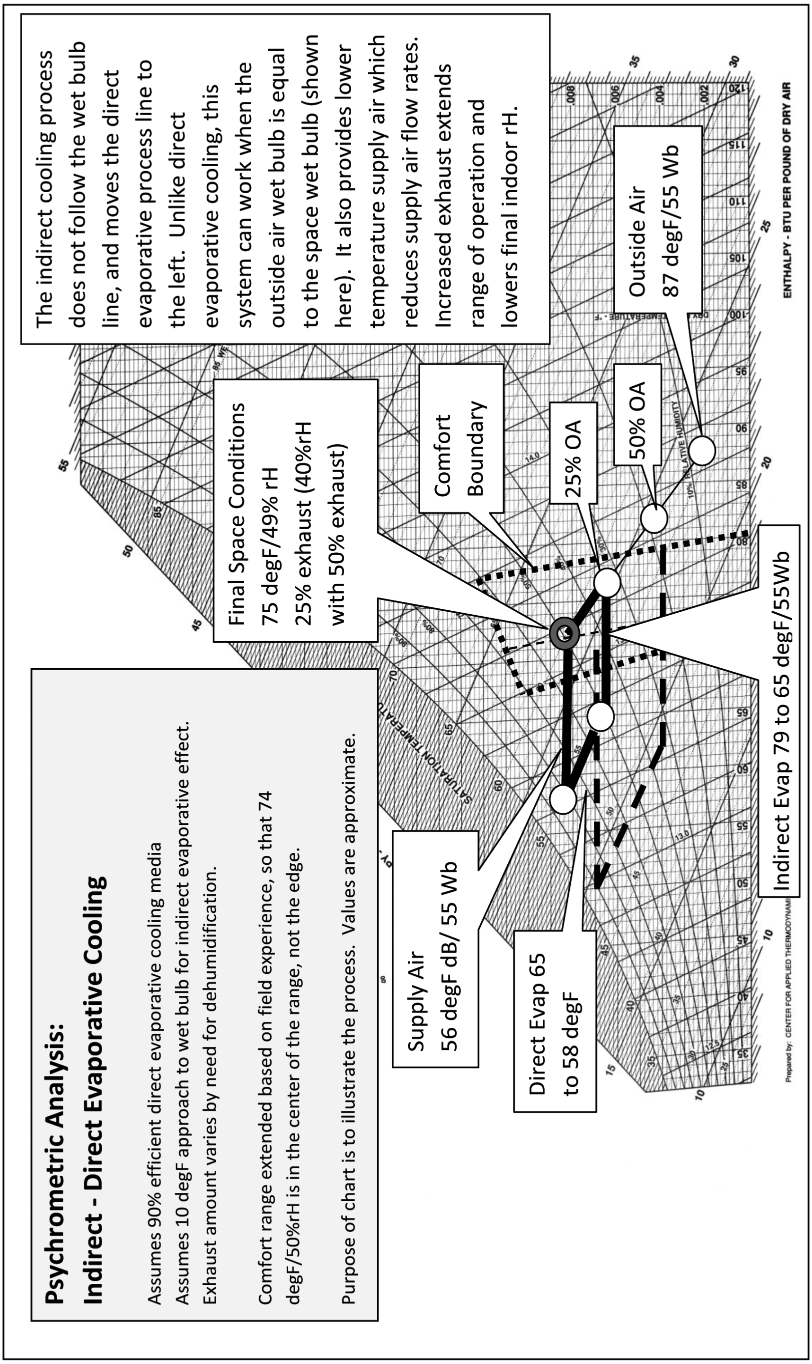


Figure 11-27. Indirect-direct Evaporative Cooling—Psychrometrics

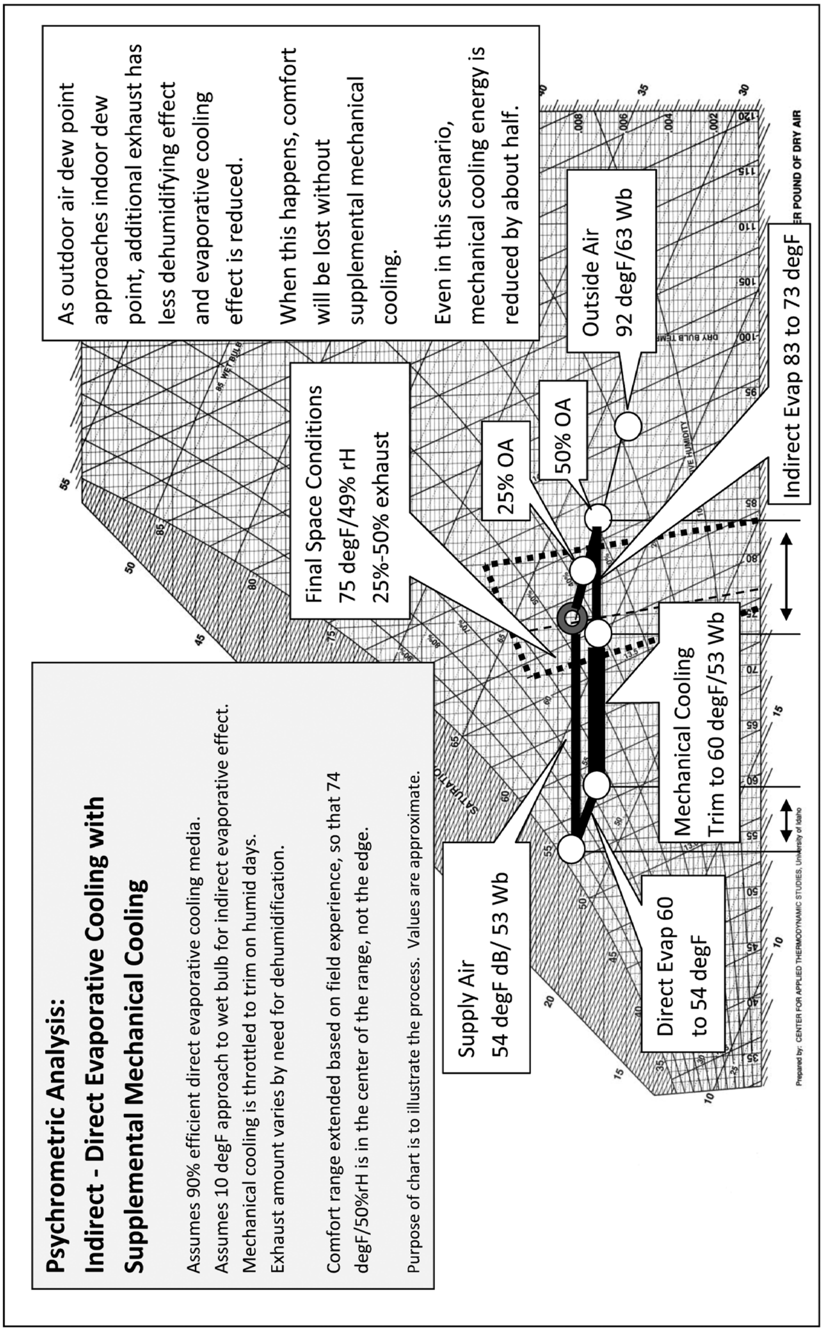


Figure 11-28. Indirect-direct Evap. Cooling with Supplemental Cooling–Psychrometrics

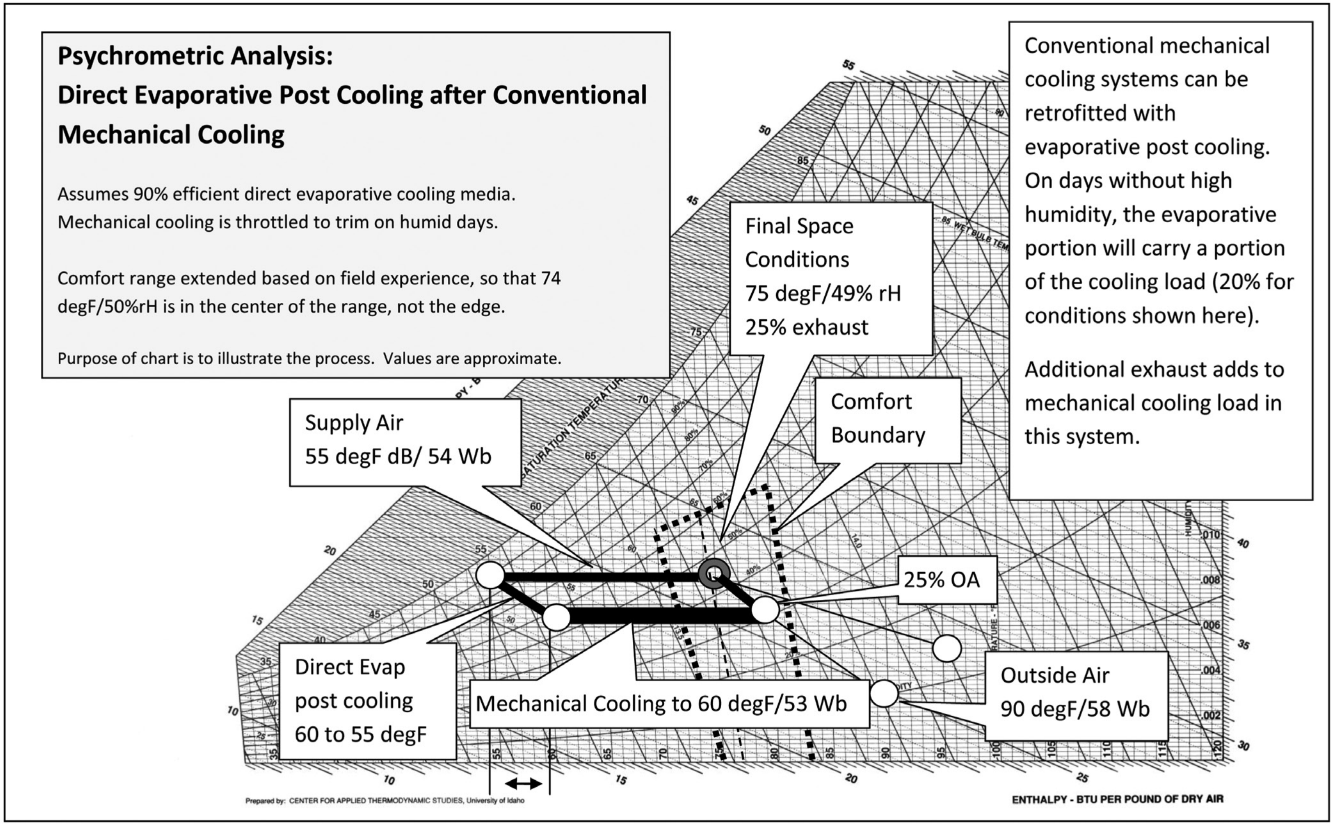


Figure 11-29. Direct Evaporative Cooling—Post Cooling—Psychrometrics

SPOT COOLING

If the dry bulb temperature of the air is below skin temperature, convection rather than evaporation cools workers. In these conditions, an 80 deg F air stream can provide comfort regardless of its relative humidity.

Source: ASHRAE Applications Handbook, 2003, © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org.

Comfort can be had at higher temperatures by using higher air flows, depending upon the relative humidity of the space. If the building temperature is raised 5 degrees from the benefit of strong air movement, the envelope losses will be reduced by 10% (2% per degree). NOTE: Cooling only the worker locations in a building, and not the building itself, will yield more savings.

Velocity, fpm	rH, %	Comfortable Temperature, deg F
0	20	76
0	40	75
0	50	74
0	60	73
300	20	81
300	40	80
300	50	79
300	60	78
600	20	85
600	40	83
600	50	81
600	60	80

Note: Consider air velocities in applications where papers could blow.

Figure 11-30. Air Velocity Effect on Comfort Zone

Data extracted from chart "Change in Human Comfort Zone as Air Movement Increases," ASHRAE Applications Handbook, 2003, © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org.

(Values are center of the range shown.)

VAV REHEAT PENALTY

Whenever supply air is delivered below room temperature, and heating is required in the room, the heating load for that room is not the total heat required. The supply air must be heated as well. This is the VAV reheat penalty that is built into all single path VAV systems.

This can be mitigated through

- Supply air reset in heating season, although if the VAV system serves zones needing cooling this can create comfort issues.
- Reduced air flow minimum settings, provided sufficient ventilation is maintained
- Separating the ventilation air into a separate system of ductwork, making the main VAV system a recirculating system and allowing zero VAV box settings. The separate system is called dedicated outside air system (DOAS).

GLYCOL VS. EFFICIENCY

Glycol Effect on Energy Use

Glycol is used in HVAC hydronic systems to reduce the chance of freezing. Higher concentrations are used when the pipe or equipment is exposed to colder weather. Not all systems require glycol; design provisions such as coil circulating pumps and air handler shutdown on a 'freeze stat' contribute to freeze resistance. Outdoor equipment that use sealed heat transfer fluids such as air cooled chillers or fluid coolers are prone to freezing if dormant in winter. Air handlers, especially 100% outside air tempering units, are also prone to freeze damage. Where freeze protection is needed, it is needed. The ideal freeze protection water additive would behave like water, with similar pumping requirements and specific heat.

Glycol increases viscosity which retards heat transfer. In turn, higher heat exchanger approaches are required.

- For heating applications, the 'hot side' (e.g. flue gas) must be hotter to cause equivalent heat exchange.
- For cooling applications, the 'cold side' (e.g. refrigerant) must be colder to cause equivalent heat exchange.

Glycol effect is more pronounced in chilled water applications, and less pronounced with higher temperatures found in heating systems.

Glycol adds pumping energy in several ways:

- Adds friction resistance for fluid in pipes, heat exchanger tubes, etc. with the effect being more pronounced in smaller pipes. Remember that friction in a pipe is a function of Reynolds number which is the ratio of intertial to viscous forces; increased viscosity lowers Reynolds number which raises friction factor and friction losses.
- Lower specific heat in glycol requires higher flow rates for the same amount of thermal energy transport, causing pressure drops to compound when used as a heat transfer fluid. If the specific heat is halved, the flow must double and pressure drop will quadruple.
- Pump efficiency is reduced with increased viscosity.

In addition to viscosity, glycol has other properties that impact energy use.

- Specific heat determines the 'carrying' capacity of the fluid and so for a given heat transfer rate the flow rate must change when the specific heat changes. Glycol specific heat values are less than water, and so more fluid flow is required compared to water.
- Weight density of the fluid affects pumping power directly.

Comparing ethylene to propylene glycol at a given temperature, ethylene glycol has a lower density and a lower specific heat although propylene glycol has higher viscosity. For pumping energy at equal ratios with water, ethylene glycol has a higher energy increase. For heat transfer, propylene has a higher energy penalty.

Toxicity: Ethylene glycol is toxic when ingested. Food-grade propylene glycol is not toxic, however all treated hydronic water should be considered toxic because they contain more than water and glycol. In addition to any glycol present, biocides and corrosion inhibitors are used and come from drums with hazard labels.

Viscosity units are varied and a source of frustration. **Table 11-3** provides several variations of viscosity units for convenience, in the chilled water range.

	Water to 30% EG	
Heat exchanger capacity	6.2 % decrease	Source: Sample tube and shell heat exchanger selection.

Figure 11-31. Glycol Effect on Water Heating Efficiency

	30% EG to water	30% PG to water	30% PG to 30% EG	Equipment Type
Chiller Power kW	3.6% decrease	6.8% decrease	3.4% decrease	Centrifugal Chiller - average
Chiller Power kW	1.6% decrease	2.5% decrease	0.9% decrease	Air Cooled Screw Chiller

Figure 11-32. Glycol Effect on Chiller Efficiency

All figures are at 45 degrees F chilled water.
Sources: Sample equipment selections from chiller equipment manufacturers
Note the differences between types of chillers; these have to do with whether the glycol is in the tubes or in the shell.

40 degF

Pumping Energy Change with Glycol as Heat Transfer Fluid

300 ft of 5 inch diameter steel pipe, 25 elbows.

Includes pump impact from viscosity change alone and additional pump impact from higher flows resulting from reduced specific gravity (for equal thermal energy transport).

Pump Hp is average of three different pumps.

				Base flow	Higher flow			Base flow	Higher flow
	gpm base flow	flow factor	gpm req'd flow	ft head	ft. head	Sp. Grav.	SSU Viscos.	Pump BHp	Pump BHp
Water	500	1	500	24	24	1	31	4.0	4.0
30% EG		1.16	581	25	33	1.05	37	4.4	6.3
40% EG		1.23	613	25	37	1.07	41	4.6	8.2
50% EG		1.30	649	25	42	1.08	46	4.6	9.9
30% PG		1.10	550	25	30	1.04	44	4.4	5.7
40% PG		1.15	573	26	34	1.05	56	4.7	6.7
50% PG		1.20	602	27	38	1.05	72	4.9	8.5

	Base flow		Higher flow	
	Hp saved Glycol to Water	Hp added Water to Glycol	Hp saved Glycol to Water	Hp added Water to Glycol
30% EG vs. Water	9%	10%	37%	60%
40% EG vs. Water	13%	15%	52%	108%
50% EG vs. Water	14%	16%	60%	149%
30% PG vs. Water	10%	11%	30%	43%
40% PG vs. Water	16%	19%	40%	68%
50% PG vs. Water	19%	24%	53%	114%

Figure 11-33. Glycol Impact on Pumping

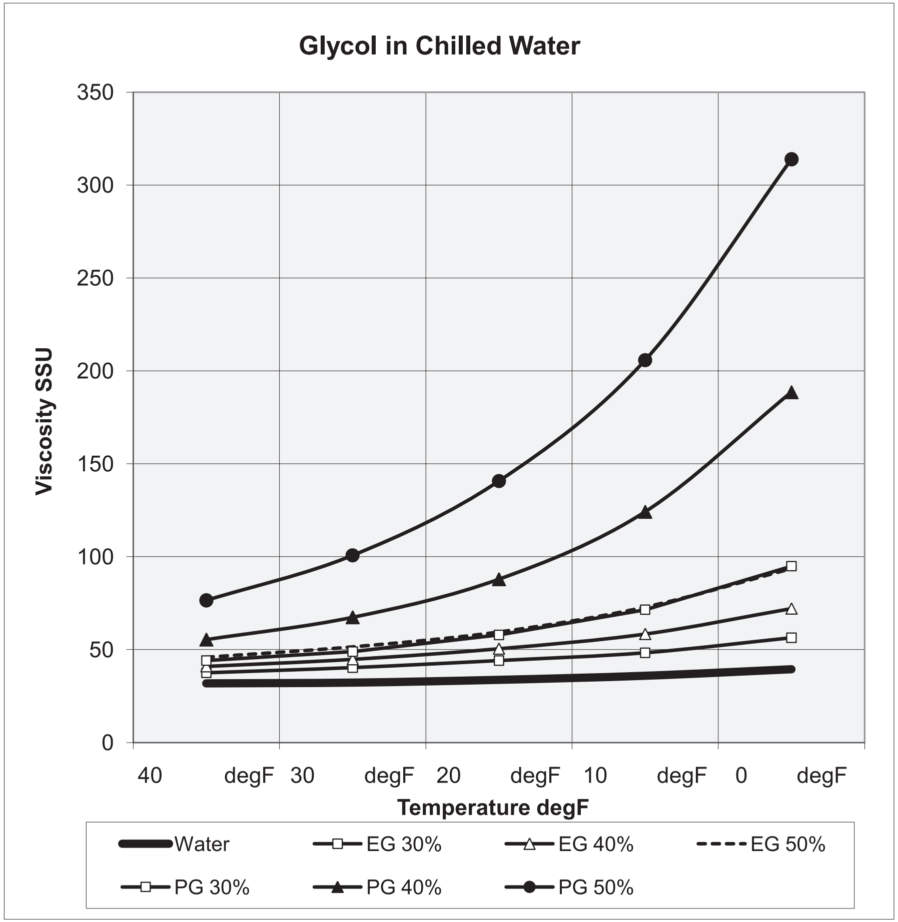


Figure 11-34. Viscosity Values for Glycol in Chilled Water

Units are SSU

Note: EG 40% and PG 30% lines are almost equal and are graphed on top of each other.

Data for graph is shown in **Table 11-3** with other units of viscosity

Table 11-3. Viscosity, Specific Heat, Specific Gravity of Glycol

			Density/ 62.4	Btu/ lb-degF	(1/sp. Ht)	Viscosity			
	Deg F	**Dens. lb/cu ft**	**Sp. Grav.**	**Sp. Heat**	**Flow factor for heat transfer**	**Centi poise**	**ft^2/ sec**	**Centi stoke**	**SSU**
Water	50	62.41	1	1.002	1.00	1.310	1.41	1.3	30
	40	62.43	1	1.004	1.00	1.546	1.66	1.5	31
(32)	30	62.42	1	1.007	0.99	1.794	1.93	1.8	32
	20								
	10								
	0								
EG 30%	50	65.6	1.05	0.864	1.16	2.95	3.02	2.81	35
	40	65.7	1.05	0.861	1.16	3.54	3.62	3.37	37
	30	65.8	1.05	0.857	1.17	4.33	4.42	4.11	39
	20	65.9	1.06	0.853	1.17	5.38	5.49	5.10	43
	10	65.9	1.06	0.849	1.18	6.83	6.96	6.47	47
	0								
EG 40%	50	66.5	1.07	0.821	1.22	4.04	4.08	3.79	38
	40	66.6	1.07	0.816	1.23	4.91	4.95	4.60	41
	30	66.7	1.07	0.812	1.23	6.09	6.14	5.70	44
	20	66.8	1.07	0.808	1.24	7.74	7.79	7.23	49
	10	66.9	1.07	0.803	1.25	10.1	10.2	9.45	57
	0	67.0	1.07	0.799	1.25	13.8	13.8	12.8	69
EG 50%	50	67.3	1.08	0.775	1.29	5.50	5.49	5.10	43
	40	67.5	1.08	0.770	1.30	6.77	6.74	6.26	46
	30	67.6	1.08	0.765	1.31	8.48	8.43	7.83	51
	20	67.7	1.08	0.759	1.32	10.9	10.8	10.0	59
	10	67.8	1.09	0.754	1.33	14.3	14.1	13.1	70
	0	67.9	1.09	0.744	1.34	19.3	19.1	17.8	88
PG 30%	50	64.5	1.03	0.913	1.10	4.52	4.71	4.37	40
	40	64.7	1.04	0.909	1.10	5.75	5.97	5.55	44
	30	64.8	1.04	0.906	1.10	7.46	7.74	7.19	49
	20	64.9	1.04	0.902	1.11	9.89	10.2	9.51	57
	10	65.0	1.04	0.898	1.11	13.4	13.9	12.9	69
	0								
PG 40%	50	65.1	1.04	0.877	1.14	7.21	7.45	6.92	48
	40	65.2	1.05	0.872	1.15	9.60	9.89	9.19	56
	30	65.4	1.05	0.868	1.15	13.1	13.5	12.5	68
	20	65.5	1.05	0.864	1.16	18.5	19.0	17.6	88
	10	65.6	1.05	0.859	1.16	27.0	27.6	25.7	122
	0	65.7	1.05	0.855	1.17	40.9	41.8	38.9	181
PG 50%	50	65.5	1.05	0.835	1.20	10.7	10.9	10.2	59
	40	65.7	1.05	0.830	1.20	14.3	14.6	13.6	72
	30	65.8	1.05	0.825	1.21	19.7	20.1	18.6	92
	20	66.0	1.06	0.820	1.22	27.8	28.3	26.3	125
	10	66.1	1.06	0.814	1.23	40.6	41.3	38.4	178
	0	66.2	1.06	0.809	1.24	61.3	62.2	57.8	267

COST OF VENTILATION

- Ventilation is necessary for a variety of reasons, and should not be reduced below what is necessary. However, excessive ventilation is costly and is a source of savings. Excess ventilation can be from:
  - — Ventilation for occupants that aren't there.
  - — Make-up air for exhaust fans that are left on for no reason
  - — Unwanted exfiltration or infiltration through envelope leakage

- The energy consequence of the extra ventilation lies in the tempering of the air. In humid climates, the cost of ventilation also includes dehumidification which can be considerable.

Energy Consumption for Heating Outside Air

Therms Per Hour for Heating Outside Air (per 1000 CFM)

Btu/H = 1.08*CFM*AF*dT

Altitude Correction Factor (AF): 1.00

Cubic Feet per Minute (CFM): 1000

Hours: 1

dt=differential Temp, degF

OA Temp degF	Bldg Space Temp, degF: 50	55	60	65	70	75
-10	0.648	0.702	0.756	0.810	0.864	0.918
-5	0.594	0.648	0.702	0.756	0.810	0.864
0	0.540	0.594	0.648	0.702	0.756	0.810
5	0.486	0.540	0.594	0.648	0.702	0.756
10	0.432	0.486	0.540	0.594	0.648	0.702
15	0.378	0.432	0.486	0.540	0.594	0.648
20	0.324	0.378	0.432	0.486	0.540	0.594
25	0.270	0.324	0.378	0.432	0.486	0.540
30	0.216	0.270	0.324	0.378	0.432	0.486
35	0.162	0.216	0.270	0.324	0.378	0.432
40	0.108	0.162	0.216	0.270	0.324	0.378
45	0.054	0.108	0.162	0.216	0.270	0.324
50		0.054	0.108	0.162	0.216	0.270
55			0.054	0.108	0.162	0.216
60				0.054	0.108	0.162

Figure 11-35. Energy Consumption for Heating Outside Air

Units = therms output per hour. Divide output by heating efficiency to obtain input.

Table is based on 1000 CFM of outside air at an average temperature being heated for per hour to final temperatures shown.

For table use, determine average outside temperature during the period of interest and multiply by hours. Formula can be used directly with bin weather data if desired.

SIMULTANEOUS HEATING AND COOLING

Overlapping heating and cooling occurs in many buildings. It can be deliberate or accidental. In all cases there is an energy penalty. A very constructive general action item is to identify and eliminate this waste. It is analogous to driving with the brakes on.

Larger HVAC systems that serve multiple zones tend to have more of this built into the buildings than smaller commercial HVAC, since smaller buildings are often served with unitary package HVAC equipment that is either heating OR cooling, but never both.

Some sources of this overlap are:

- Excessive minimum settings on VAV boxes
- Dehumidification cycle through over-cooling and reheating
- Unintentional overlap of cooling and heating set points so the heating and cooling are fighting
- Control sequenced heating and cooling coils that are in series
- Poorly adjusted electronic actuator open and close travel limits
- Valves that do not fully close off, either from system pressure, debris, or damaged metal seats
- Bare hot piping, including bypass legs of 3-way valves within the air stream of a cooled space
- Small area needing cooling at all times
- Certain occupants that who are always either too hot or too cold
- Space heaters
- Spot coolers
- Defective electric heat relays (stuck "on")
- Baseboard heat that is controlled independently of main HVAC systems
- Boilers left to idle in summer and circulate hot water through the building
- Chillers left to idle in winter and circulate chilled water through the building
- Comfort zoning problems that are fixed with duct heating coils.

See Also Chapter 24: Coordinating Upstream / Downstream Set Points

See Also Chapter 24: HVAC Overlapping Heating and Cooling

Comfort Envelope

Reference **Figure 11-36**. Comfort is very personal because it involves people and people are different. An attempt to identify comfort zones has been made in order to predict what will probably be comfortable. The data is empirical and represents some level of confidence such as 90% of people will be comfortable when inside the envelope. Influences on actual comfort depend on many factors:

- Activity level and metabolism
- Clothing
- Air speed
- Radiant temperature from surroundings

A common misconception in using a comfort chart is the concept of 'operative temperature' which includes dry bulb temperature and radiant temperature. For example, regardless of ambient air temperature, a person standing on a cold floor or close to a cold window will tend to be cold; similar effect when close to a sunlit window or hot object. Operative temperature is very difficult to determine exactly which makes the comfort chart very difficult to use.

Even when values of operative temperature are available, there must always be latitude with comfort conditions to accommodate the people that do not fit in the 90% mold.

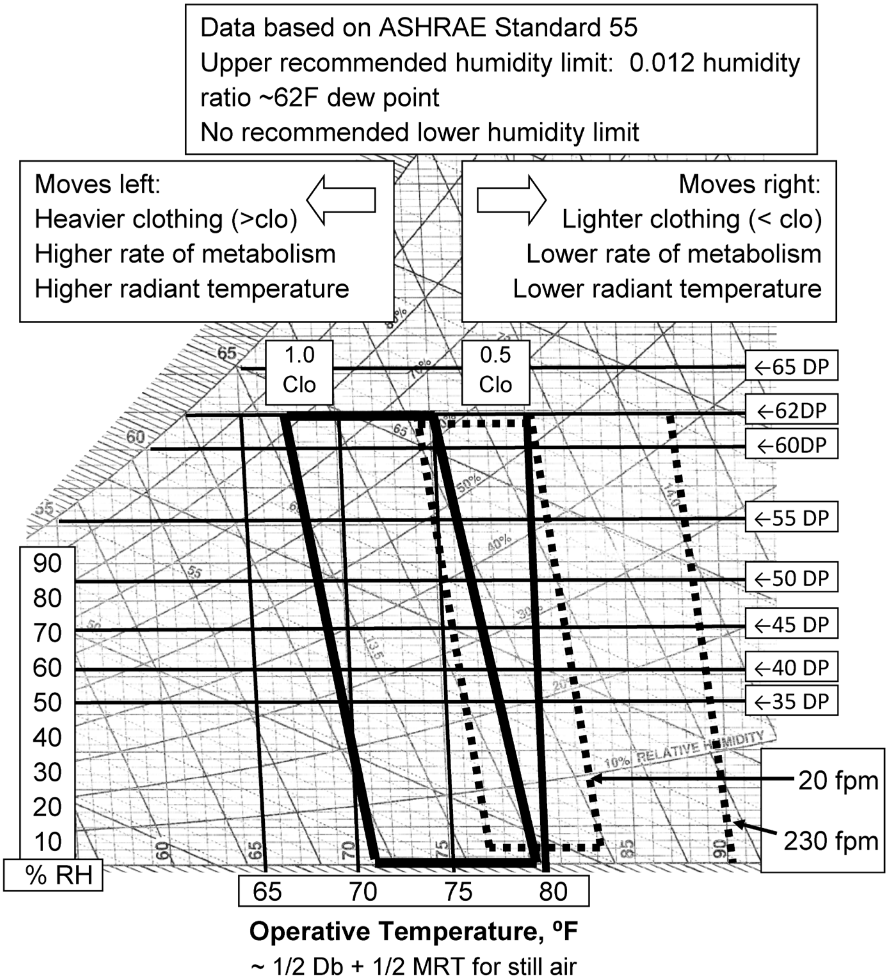


Figure 11-36. Comfort Envelope

Chapter 12

Motors and Electrical Information

FULL LOAD MOTOR EFFICIENCY

Nominal motor efficiency for NEMA Induction Motors Rated 600 Volts or less, polyphase, 1-200 Hp, full speed, full load.

HP	Open Motors			Enclosed Motors		
	3600 RPM (2-POLE)	1800 RPM (4-POLE)	1200 RPM (6-POLE)	3600 RPM (2-POLE)	1800 RPM (4-POLE)	1200 RPM (6-POLE)
1.0	---	82.5	80.0	75.5	82.5	80.0
1.5	82.5	84.0	84.0	82.5	84.0	85.5
2.0	84.0	84.0	85.5	84.0	84.0	86.5
3.0	84.0	86.5	86.5	85.5	87.5	87.5
5.0	85.5	87.5	87.5	87.5	87.5	87.5
7.5	87.5	88.5	88.5	88.5	89.5	89.5
10.0	88.5	89.5	90.2	89.5	89.5	89.5
15.0	89.5	91.0	90.2	90.2	91.0	90.2
20.0	90.2	91.0	91.0	90.2	91.0	90.2
25.0	91.0	91.7	91.7	91.0	92.4	91.7
30.0	91.0	92.4	92.4	91.0	92.4	91.7
40.0	91.7	93.0	93.0	91.7	93.0	93.0
50.0	92.4	93.0	93.0	92.4	93.0	93.0
60.0	93.0	93.6	93.6	93.0	93.6	93.6
75.0	93.0	94.1	93.6	93.0	94.1	93.6
100.0	93.0	94.1	94.1	93.6	94.5	94.1
125.0	93.6	94.5	94.1	94.5	94.5	94.1
150.0	93.6	95.0	94.5	94.5	95.0	95.0
200.0	94.5	95.0	94.5	95.0	95.0	95.0

Figure 12-1A. **Full-Load Nominal Efficiencies of EPAct Motors**

Source: U.S. DOE 10 CFR Part 431, October 1999

HP	Open Motors 3600 RPM (2-POLE)	Open Motors 1800 RPM (4-POLE)	Open Motors 1200 RPM (6-POLE)	Enclosed Motors 3600 RPM (2-POLE)	Enclosed Motors 1800 RPM (4-POLE)	Enclosed Motors 1200 RPM (6-POLE)
1.0	77.0	85.5	82.5	77.0	85.5	82.5
1.5	84.0	86.5	86.5	84.0	86.5	87.5
2.0	85.5	86.5	87.5	85.5	86.5	88.5
3.0	85.5	89.5	88.5	86.5	89.5	89.5
5.0	86.5	89.5	89.5	88.5	89.5	89.5
7.5	88.5	91.0	90.2	89.5	91.7	91.0
10.0	89.5	91.7	91.7	90.2	91.7	91.0
15.0	90.2	93.0	91.7	91.0	92.4	91.7
20.0	91.0	93.0	92.4	91.0	93.0	91.7
25.0	91.7	93.6	93.0	91.7	93.6	93.0
30.0	91.7	94.1	93.6	91.7	93.6	93.0
40.0	92.4	94.1	94.1	92.4	94.1	94.1
50.0	93.0	94.5	94.1	93.0	94.5	94.1
60.0	93.6	95.0	94.5	93.6	95.0	94.5
75.0	93.6	95.0	94.5	93.6	95.4	94.5
100.0	93.6	95.4	95.0	94.1	95.4	95.0
125.0	94.1	95.4	95.0	95.0	95.4	95.0
150.0	94.1	95.8	95.4	95.0	95.8	95.8
200.0	95.0	95.8	95.4	95.4	96.2	95.8

Figure 12-1B. Full-Load Nominal Efficiencies of NEMA Premium Motors
Source: NEMA Standard MG1 "Motors and Generators"

HP	Open Motors 3600 RPM (2-POLE)	Open Motors 1800 RPM (4-POLE)	Open Motors 1200 RPM (6-POLE)	Enclosed Motors 3600 RPM (2-POLE)	Enclosed Motors 1800 RPM (4-POLE)	Enclosed Motors 1200 RPM (6-POLE)
1.0	76.19	77.55	74.52	73.00	76.68	73.36
1.5	77.25	79.34	77.64	75.15	79.08	77.90
2.0	79.56	80.54	79.86	78.88	80.83	78.34
3.0	79.08	82.38	81.66	79.62	81.45	80.36
5.0	82.57	83.83	83.62	82.38	83.34	83.14
7.5	82.87	85.16	85.54	82.59	85.51	84.44
10.0	85.02	86.09	87.39	84.99	85.73	84.95
15.0	86.63	87.80	86.96	85.66	86.63	87.02
20.0	88.13	88.30	87.70	86.62	88.52	87.74
25.0	88.45	88.91	88.96	87.53	89.30	88.93
30.0	87.73	88.86	89.48	87.74	89.56	89.64
40.0	88.57	90.00	89.43	88.54	90.19	89.92
50.0	89.06	90.69	89.74	89.00	91.32	90.62
60.0	90.38	91.29	90.79	89.39	91.75	90.75
75.0	90.36	91.94	91.51	90.60	91.68	91.61
100.0	90.53	92.08	92.23	90.88	92.25	91.40
125.0	91.22	92.17	91.97	90.88	92.19	92.06
150.0	91.68	92.81	92.61	91.52	93.03	93.08
200.0	91.54	93.03	92.87	92.70	93.54	92.56

Figure 12-C. Pre-EPAct Default Motor Efficiency Table for Use When No Nameplate Efficiency is Available
Source: "The 1*2*3 Approach to Motor Management—Spreadsheet." Estimated efficiencies by Motor Master

PART LOAD MOTOR EFFICIENCY – CONSTANT SPEED

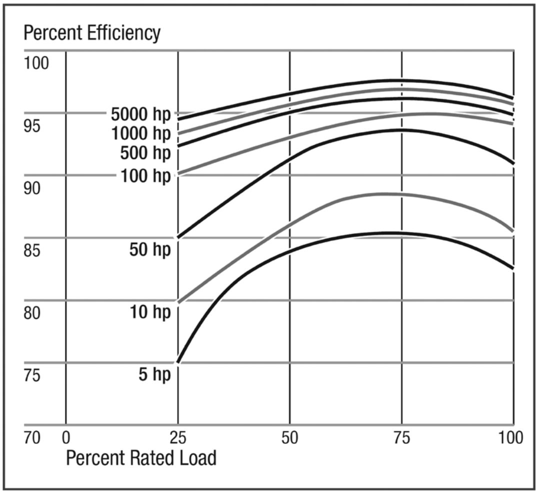


Figure 12-2. Motor Efficiency at Constant Speed
Source: "Improving Motor and Drive System Performance: A Sourcebook for Industry", U.S. DOE Industrial Technologies Program, 2008
Below 50% load, motor efficiency drops off sharply.

PART LOAD MOTOR EFFICIENCY—VARIABLE SPEED

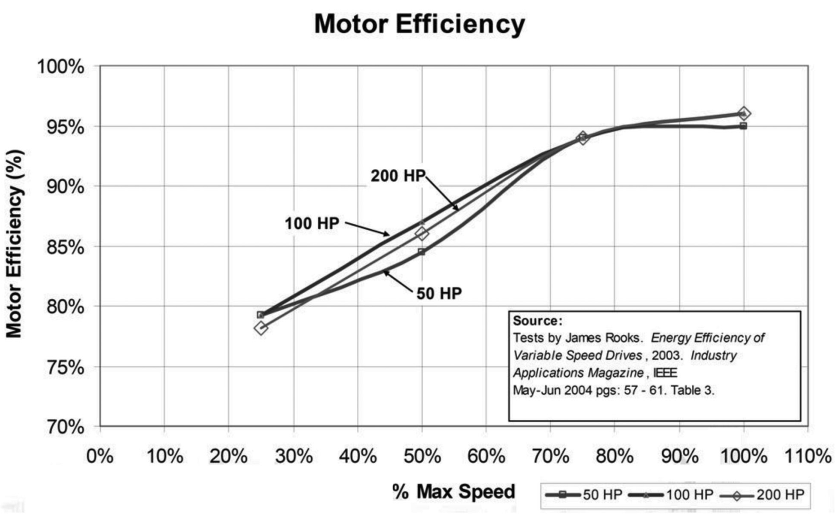


Figure 12-3. Motor Efficiency at Reduced Speed

Source: Stum and Koran, "Techniques and Tips for Retro Commissioning Energy Calculations", Presentation at the National Conference on Building Commissioning, 2007. Original Data Source as noted.

Motors lose efficiency at reduced speed, a fact often overlooked in estimating variable speed retrofit savings.

EFFECT OF VOLTAGE CHANGES ON INDUCTION MOTOR CHARACTERISTICS

Source: Engineering Cookbook: A Handbook for the Mechanical Designer. Loren Cook Co. Effects are generalized.

Characteristic	Voltage	
	110%	90%
Starting Torque	Up 21%	Down 19%
Maximum Torque	Up 21%	Down 19%
Percent Slip	Down 15-20%	Up 20-30%
Efficiency - Full Load	Down 0-3%	Down 0-2%
3/4 Load	0 - Down Slightly	Little Change
1/2 Load	Down 0-5%	Up 0-1%
Power Factor - Full Load	Down 5-15%	Up 1-7%
3/4 Load	Down 5-15%	Up 2-7%
1/2 Load	Down 10-20%	Up 3-10%
Full Load Current	Down Slightly to Up 5%	Up 5-10%
Starting Current	Up 10%	Down 10%
Full Load - Temperature Rise	Up 10%	Down 10-15%
Maximum Overload Capacity	Up 21%	Down 19%
Magnetic Noise	Up Slightly	Down Slightly

Figure 12-4.

VOLTAGE IMBALANCE

Voltage imbalance is undesirable for three phase motors since it causes a partial reverse rotation force which then acts like a brake. The extent of the imbalance determines the extent of the dragging and the efficiency loss. Rule of thumb is that imbalance should not exceed 1%.

It takes very little imbalance to de-rate a motor hp from nameplate, but it takes a lot of imbalance to do much to the efficiency.

$$\text{Voltage \% imbalance} = \frac{\text{(max voltage difference between any phase and the average voltage)}}{\text{Average voltage}}$$

Example:	
470V	Voltage A-B (phase A to phase B)
460V	Voltage B-C
475V	Voltage A-C
468V	**Average voltage**
2V	A-B differential (voltage – average)
8V	**B-C differential**
7V	A-C differential
1.7%	Voltage Imbalance, from 8/468

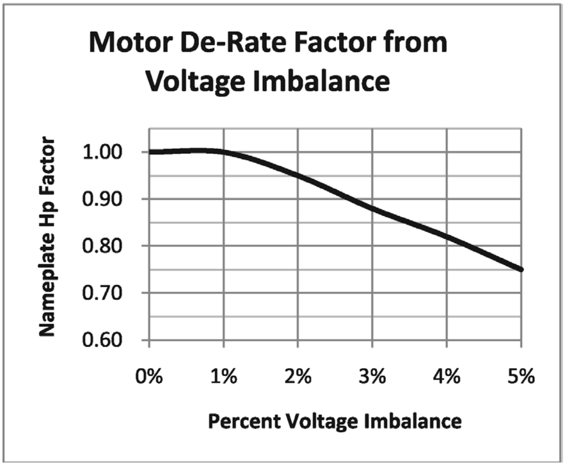
% Motor Load	Balanced	1% Imbalance	2.5% Imbalance
100	0.0%	0.0%	1.5%
75	0.0%	0.1%	1.4%
50	0.0%	0.6%	2.1%

Figure 12-5. Voltage Imbalance Effect on Motor Efficiency Rating

Data Source: "Electric Motor Voltage Quality Problems", Industrial Technologies Program (ITP), 2006, US DOE Office of Energy Efficiency and Renewable Energy.
Original Source NEMA Standards Publication MG-1-2003, Motors and Generators.

Due to motor heating, motors with unbalanced voltage are to be reduced from nameplate maximum horsepower, according to this chart.

Figure 12-6. Sources of Electric Motor Losses
Source: "Effect of Repair/Rewinding On Motor Efficiency," 2003, Electrical Apparatus Service Association (EASA)



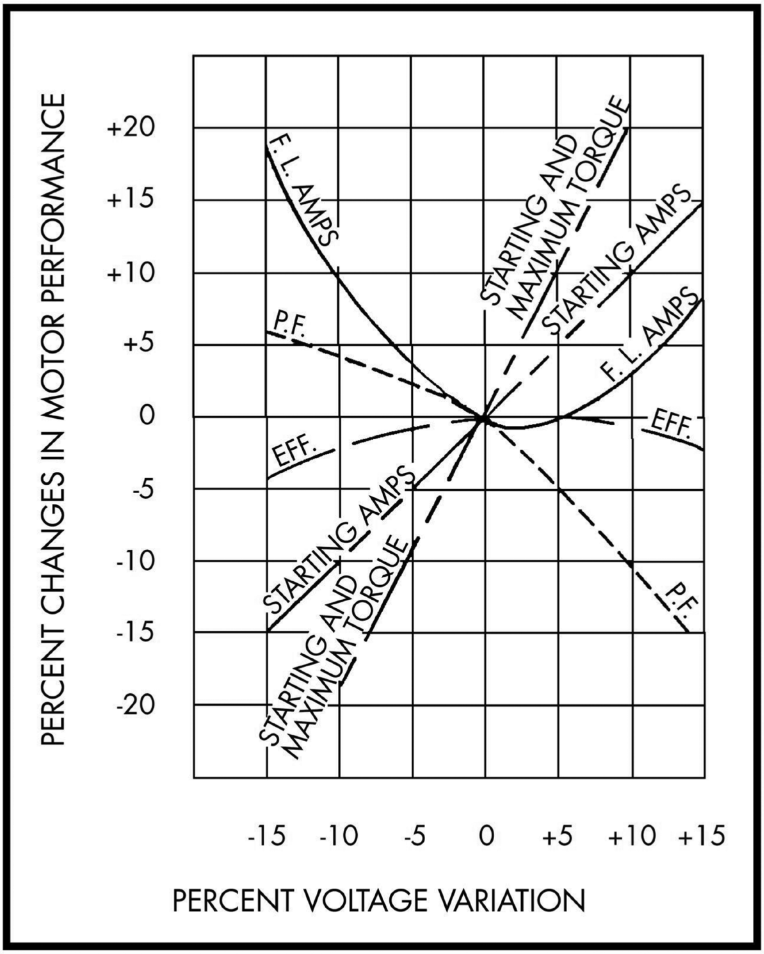


Figure 12-7. Voltage Imbalance Effect on Various Motor Properties

Source: Optimizing Your Motor-Driven System, US. Department of Energy, Motor Challenge Fact Sheet, 1996.

SOURCES OF MOTOR LOSSES

Losses	Fixed or variable loss	2-Pole average	4-Pole average	Factors affecting losses
Core losses	Fixed	19%	21%	Electrical steel, air gap, saturation
Friction and windage losses	Fixed	25%	10%	Fan efficiency, lubrication, bearings
Stator I^2R losses	Varies with load	26%	34%	Conductor area, mean length of turn, heat dissipation
Rotor I^2R losses	Varies with load	19%	21%	Bar and end ring area and material
Stray load losses	Varies with load	11%	14%	Manufacturing processes, slot design, air gap

Figure 12-8a. Sources of Electric Motor Losses

Source: "Effect of Repair/Rewinding On Motor Efficiency," 2003, Electrical Apparatus Service Association (EASA)

COMMON MOTOR DESIGN CHARACTERISTICS

Source: Engineering Cookbook: A Handbook for the Mechanical Designer. Loren Cook Co.

NEMA Design	Starting Current	Locked Rotor	Breakdown Torque	% Slip
B	Medium	Medium Torque	High	Max. 5%
C	Medium	High Torque	Medium	Max. 5%
D	Medium	Extra-High Torque	Low	5% or more

NEMA Design	Applications
B	Normal starting torque for fans, blowers, rotary pumps, compressors, conveyors, machine tools. Constant load speed.
C	High inertia starts - large centrifugal blowers, fly wheels, and crusher drums. Loaded starts such as piston pumps, compressors, and conveyers. Constant load speed.
D	Very high inertia and loaded starts. Also considerable variation in load speed. Punch presses, shears and forming machine tools. Cranes, hoists, elevators, and oil well pumping jacks.

Figure 12-8b. Common Motor Design Characteristics

PERMANENT MAGNET MOTORS

Permanent magnet motors have been used historically where space constraints require a compact sized motor, such as automobile accessory drive motors. A characteristic of permanent magnet motors is of significant interest in the energy field – the tendency to resist efficiency loss at reduced speed better than standard induction AC motors.

Permanent magnet motor technology uses rare earth magnets and is more expensive than conventional motor technology. These motors are seeing application in variable speed screw air compressors and refrigeration compressors where a high percentage of run hours are at part load.

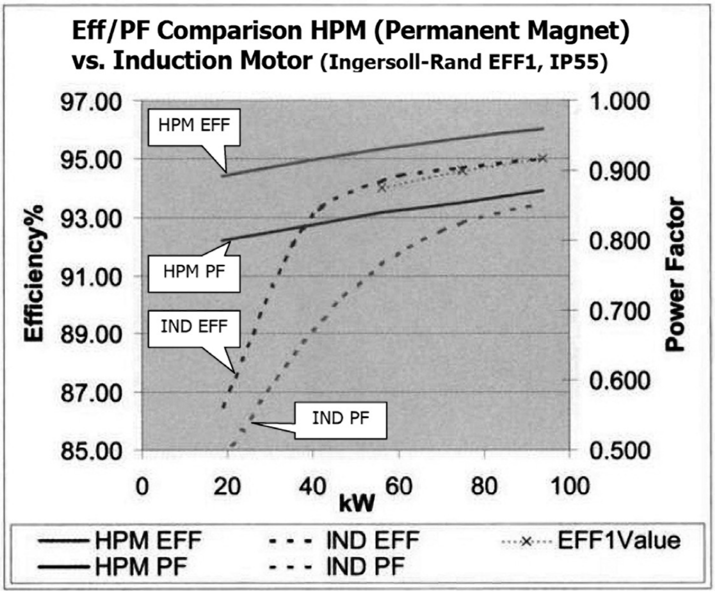


Figure 12-9. Permanent Magnet / Induction Motor Comparison
Source: Energy Efficiency in Motor Driven Systems, Parasalilti, F. and Bertoldi, P., eds., 2003, Springer.

FRACTIONAL HORSEPOWER MOTORS

Most motors encountered in commercial facilities are polyphase (three phase). Places where fractional horsepower motors are used in quantity that may have economic merit in changing:

- HVAC terminal units (VAV boxes, fan coils)
- Small HVAC furnaces, rooftop units, air-cooled condensers
- Walk-in cooler/freezer evaporator motors

The most economically beneficial applications for small motor efficiency upgrades are when the motors, and their heat, are integral to a cooling process, since the wasted energy has the parasitic effect of causing additional cooling work to remove it.

An energy conserving alternative to PSC and SP motors is the electrically commutated motor (ECM), which is a special adaptation of a squirrel cage motor with an on-board electronic controller to manipulate the electric signal and motor as though it were a DC motor with brushes and a commutator. Full load efficiency is notably better, with efficiencies of 70-80% common, and part load efficiency for throttling applications can be much more efficient.

Single Phase AC Motors (60hz)

Motor Type	HP Range	Efficiency	Slip	Poles/ RPM	Use
Shaded Pole	1/6 to 1/4 hp	low (30%)	high (14%)	4/1550 6/1050	small direct drive fans (low start torque)
Perm-split Cap.	Up to 1/3 hp	medium (50%)	medium (10%)	4/1625 6/1075	small direct drive fans (low start torque)
Split-phase	Up to 1/2 hp	medium-high (65%)	low (4%)	2/3450 4/1725 6/1140 8/850	small belt drive fans (good start torque)
Capacitor-start	1/2 to 34 hp	medium-high (65%)	low (4%)	2/3450 4/1725 6/1140 8/850	small belt drive fans (good start torque)

Figure 12-10. Fractional Hp Motor Efficiencies

Source: Engineering Cookbook: A Handbook for the Mechanical Designer. Loren Cook Co.

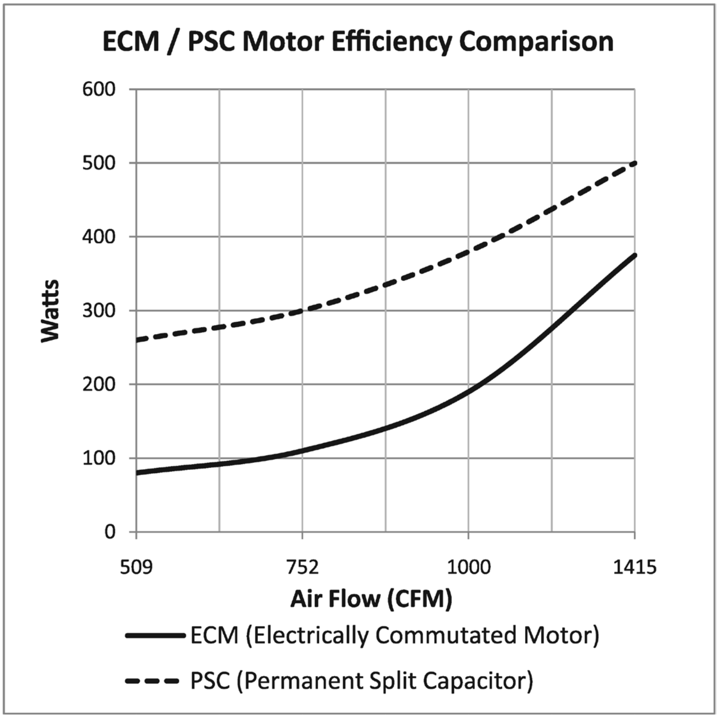


Figure 12-11. ECM Motor/ PSC Motor Efficiency Comparison
Source: Engineering Guidelines: Terminals, Controls and Accessories, Titus Inc.

VARIABLE SPEED DRIVES

Varying the speed of a pump, fan, conveyor, etc. saves energy and makes sense when there is a varying demand. Commercial examples are seasonal variations in heating/cooling loads which can be met with variable air flow/water flow. Common methods to proportion loads with changes in demand include:

- Start-stop cycling
- Variable speed motor and rigid connection to driven load
- Variable speed driven load coupling and constant speed motor

All variable drive technologies incur some loss, but these are generally small compared to the reduced power at part load compared to head loss throttling. Some lose efficiency faster at reduced speed.

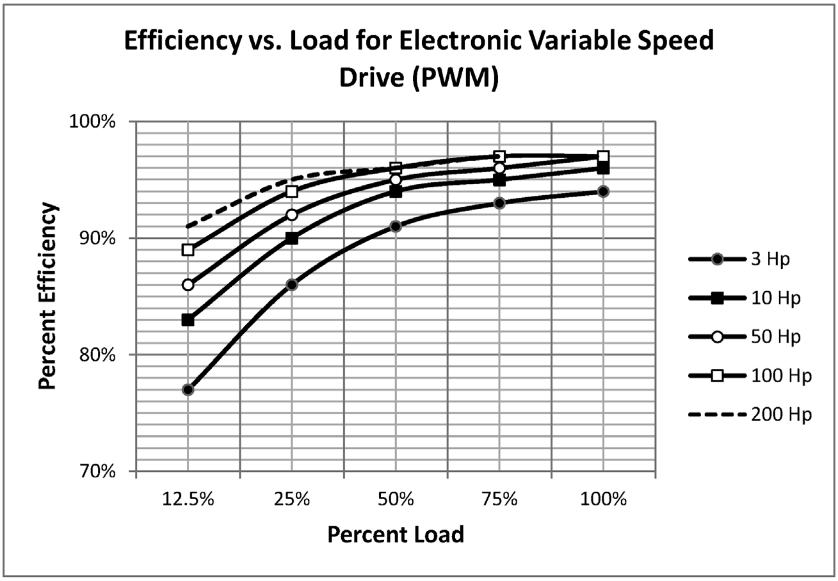


Figure 12-12. Part Load Efficiency of AC Electronic Variable Speed Drives (VFD)

Source: "Improving Motor and Drive System Performance: A Sourcebook for Industry," U.S. DOE Industrial Technologies Program, 2008

If efficiency profile information is not readily available from the manufacturer, these characteristic curves can be used.

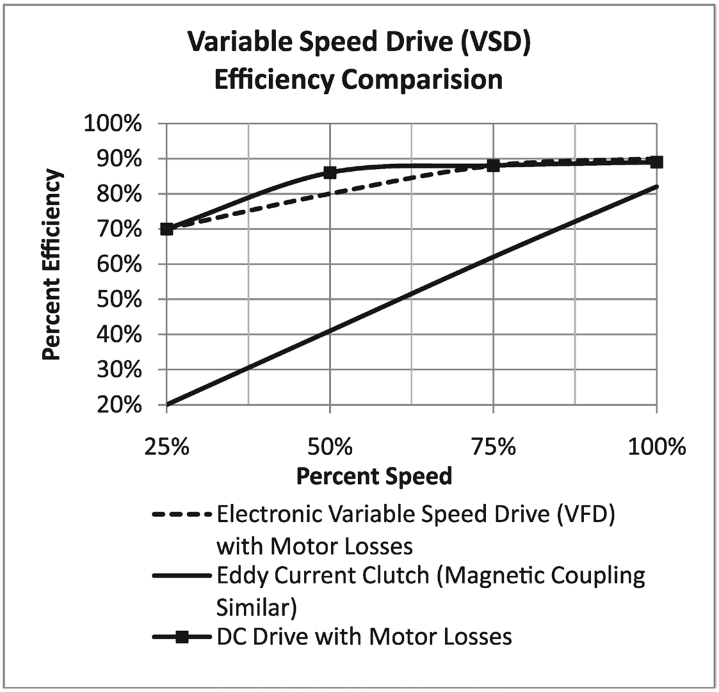


Figure 12-13. Efficiency Comparison of Electronic and Mechanical Variable Speed Drives

Source: "Power Quality & Utilisation Guide," 2007, European Copper Institute & Laborelec

POWER FACTOR

	KW Multipliers To Determine Capacitor KVAR Required															
	Corrected Power Factor															
Original Power Factor	0.80	0.81	0.82	0.83	0.84	0.85	0.86	0.87	0.88	0.89	0.90	0.91	0.92	0.93	0.94	0.95
0.60	0.583	0.609	0.635	0.661	0.687	0.714	0.740	0.767	0.794	0.821	0.849	0.878	0.907	0.938	0.970	1.005
0.61	0.549	0.575	0.601	0.627	0.653	0.679	0.706	0.732	0.759	0.787	0.815	0.843	0.873	0.904	0.936	0.970
0.62	0.515	0.541	0.567	0.593	0.620	0.646	0.672	0.699	0.726	0.753	0.781	0.810	0.839	0.870	0.903	0.937
0.63	0.483	0.509	0.535	0.561	0.587	0.613	0.639	0.666	0.693	0.720	0.748	0.777	0.807	0.837	0.870	0.904
0.64	0.451	0.477	0.503	0.529	0.555	0.581	0.607	0.634	0.661	0.688	0.716	0.745	0.775	0.805	0.838	0.872
0.65	0.419	0.445	0.471	0.497	0.523	0.549	0.576	0.602	0.629	0.657	0.685	0.714	0.743	0.774	0.806	0.840
0.66	0.388	0.414	0.440	0.466	0.492	0.519	0.545	0.572	0.599	0.626	0.654	0.683	0.712	0.743	0.775	0.810
0.67	0.358	0.384	0.410	0.436	0.462	0.488	0.515	0.541	0.568	0.596	0.624	0.652	0.682	0.713	0.745	0.779
0.68	0.328	0.354	0.380	0.406	0.432	0.459	0.485	0.512	0.539	0.566	0.594	0.623	0.652	0.683	0.715	0.750
0.69	0.299	0.325	0.351	0.377	0.403	0.429	0.456	0.482	0.509	0.537	0.565	0.593	0.623	0.654	0.686	0.720
0.70	0.270	0.296	0.322	0.348	0.374	0.400	0.427	0.453	0.480	0.508	0.536	0.565	0.594	0.625	0.657	0.692
0.71	0.242	0.268	0.294	0.320	0.346	0.372	0.398	0.425	0.452	0.480	0.508	0.536	0.566	0.597	0.629	0.663
0.72	0.214	0.240	0.266	0.292	0.318	0.344	0.370	0.397	0.424	0.452	0.480	0.508	0.538	0.569	0.601	0.635
0.73	0.186	0.212	0.238	0.264	0.290	0.316	0.343	0.370	0.396	0.424	0.452	0.481	0.510	0.541	0.573	0.608
0.74	0.159	0.185	0.211	0.237	0.263	0.289	0.316	0.342	0.369	0.397	0.425	0.453	0.483	0.514	0.546	0.580
0.75	0.132	0.158	0.184	0.210	0.236	0.262	0.289	0.315	0.342	0.370	0.398	0.426	0.456	0.487	0.519	0.553
0.76	0.105	0.131	0.157	0.183	0.209	0.235	0.262	0.288	0.315	0.343	0.371	0.400	0.429	0.460	0.492	0.526
0.77	0.079	0.105	0.131	0.157	0.183	0.209	0.235	0.262	0.289	0.316	0.344	0.373	0.403	0.433	0.466	0.500
0.78	0.052	0.078	0.104	0.130	0.156	0.183	0.209	0.236	0.263	0.290	0.318	0.347	0.376	0.407	0.439	0.474
0.79	0.026	0.052	0.078	0.104	0.130	0.156	0.183	0.209	0.236	0.264	0.292	0.320	0.350	0.381	0.413	0.447
0.80		0.026	0.052	0.078	0.104	0.130	0.157	0.183	0.210	0.238	0.266	0.294	0.324	0.355	0.387	0.421
0.81			0.026	0.052	0.078	0.104	0.131	0.157	0.184	0.212	0.240	0.268	0.298	0.329	0.361	0.395
0.82				0.026	0.052	0.078	0.105	0.131	0.158	0.186	0.214	0.242	0.272	0.303	0.335	0.369
0.83					0.026	0.052	0.079	0.105	0.132	0.160	0.188	0.216	0.246	0.277	0.309	0.343
0.84						0.026	0.053	0.079	0.106	0.134	0.162	0.190	0.220	0.251	0.283	0.317
0.85							0.026	0.053	0.080	0.107	0.135	0.164	0.194	0.225	0.257	0.291
0.86								0.027	0.054	0.081	0.109	0.138	0.167	0.198	0.230	0.265
0.87									0.027	0.054	0.082	0.111	0.141	0.172	0.204	0.238
0.88										0.027	0.055	0.084	0.114	0.145	0.177	0.211
0.89											0.028	0.057	0.086	0.117	0.149	0.184
0.90												0.029	0.058	0.089	0.121	0.156
0.91													0.030	0.060	0.093	0.127
0.92														0.031	0.063	0.097
0.93															0.032	0.067
0.94																0.034
0.95																

Figure 12-14. Power Factor Correction Capacity Quick Reference Chart

Source: US Motors

Multiply the motor kW by the factor to arrive at the capacity KVAR required.

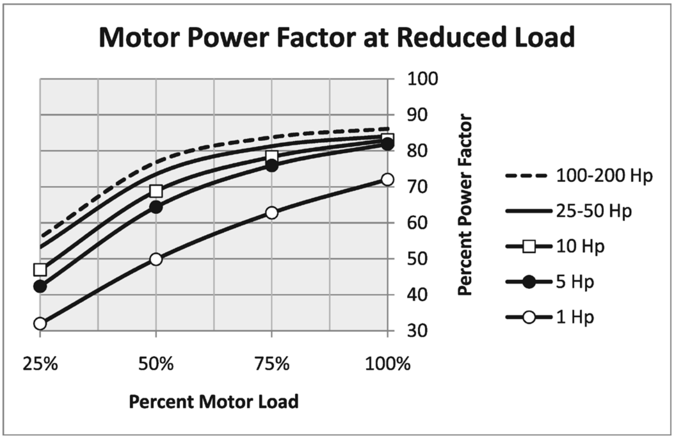


Figure 12-15. Motor Power Factor at Reduced Load

Data Source: Motor Master 4.0. Figures are averages of four similar motors in each category.

Equipment	PF	Remarks
Fluorescent Lighting (Magnetic Ballast)	0.4-0.6	
Fluorescent Lighting Ballast (Normal Electronic Ballast)	0.8-0.9	
Fluorescent Lighting Ballast (High PF Electronic Ballast)	0.95	
Compact Fluorescent Lighting (Normal Ballast)	0.6	
Compact Fluorescent Lighting (High PF Ballast)	0.95	
High Intensity Discharge Lighting (Magnetic Ballast)	0.4-0.8	
Solenoids, other Electro Magnets	0.2-0.5	
Induction Heating Equipment	0.6-0.9	
Small "dry" Transformers	0.3-0.9	Reduces with load
Welding – Transformer or Rectifier Type	0.2-0.4	
Welding – Inverter Type	0.9	
Rectifiers	0.8	Reduces with load
Induction Motors 3-100 hp	0.8-0.9	Reduces with load
Small Induction Motors	0.55-0.8	Reduces with load

Figure 12-16. Power Factor for Some Equipment

Chapter 13

Combustion Equipment and Systems

STEAM COST

Example Source: *Energy Management Handbook*, 7th Ed, The Fairmont Press.

Example calculation for cost per 1000 lbs of steam.

Assume 200 psig steam, 160 deg F feed water, 82 pct efficiency, fuel cost $2.20 per MMBtu
Enthalpy of steam: 1199 Btu/lb
Enthalpy of feed water: 130 Btu/lb
Heat added per lb of steam: 1199 – 130 = 1069 Btu/lb
Fuel Btu required to make steam: 1069/0.82 = 1304 Btu/lb
Cost of Steam: 1304/1,000,000 * 2.20 *1000 = $2.87 per 1000 lbs of steam

Heat loss for condensate leaks

- **Heat added to make up water is equal to the heat in the spent condensate**
- $mc\Delta T$, where c=1.0 and ΔT is ($T_{cond} - T_{make\ up}$)
- = m * (212F – $T_{make\ up}$)

Heat loss for steam leaks and one pass steam use (including defective steam traps)

- **Heat added to make up water is equal to the heat in the spent steam plus condensate**
- m * (212F – $T_{make\ up}$), plus
- m * (**heat of vaporization** at working pressure, from steam tables)

Boiler system electric losses usually neglected

COMBUSTION EFFICIENCY

Table 13-1. Combustion Efficiency for Some Equipment

Values are approximate, for new condition equipment.

In general, gravity flues operate with higher excess air and will have lower efficiency than forced draft equipment. Older equipment efficiencies are often less. Higher product temperatures result in lower efficiencies.

Equipment	Efficiency
Water heater- gravity flue	70-80%
Water heater – forced draft non-condensing	80-85%
Water heater or pool heater - condensing	90-95%
Packaged HVAC Rooftop Unit Furnace	80%
HVAC Furnace, standard	80%
HVAC Furnace, condensing	92-95%
Boiler, 'tray' type burner, gravity flue	70-80%
Boiler, forced draft non-condensing	80-85%
HW Boiler, condensing **	90-95%

**assumes compatible low temperature distribution temperatures that allow condensing.

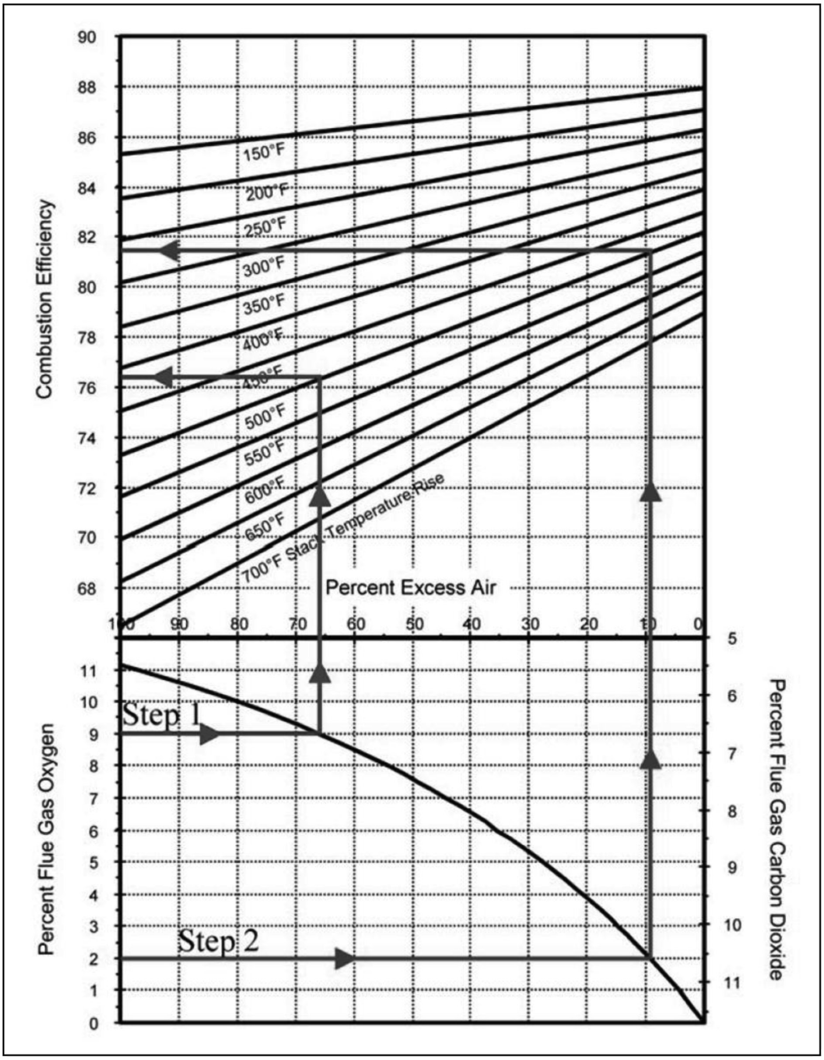


Figure 13-1A. Combustion Efficiency Nomograph
Source: "Actions You Can Take to Reduce Heating Costs," Fact Sheet PNNL-SA-43825, Jan 2005, US DOE Office of Energy Efficiency and Renewable Energy.

Step 1 and Step 2 are before and after examples.

Start with either pct O_2, pct CO_2 or pct excess air, and intersect the lower curve, then move directly up on the chart to intersect the stack temperature rise; read combustion efficiency to the left.

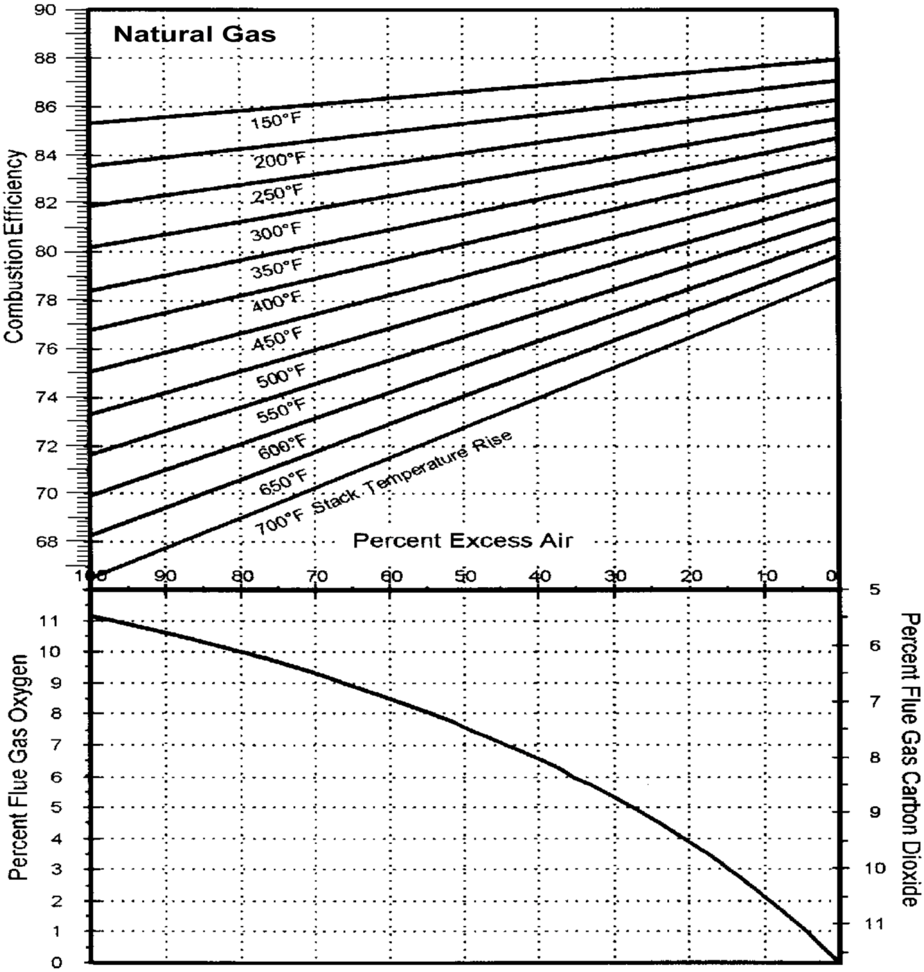


Figure 13-1B. Combustion Efficiency Chart for Natural Gas
Source: Boilers and Fired Systems, *Energy Management Handbook*, The Fairmont Press.

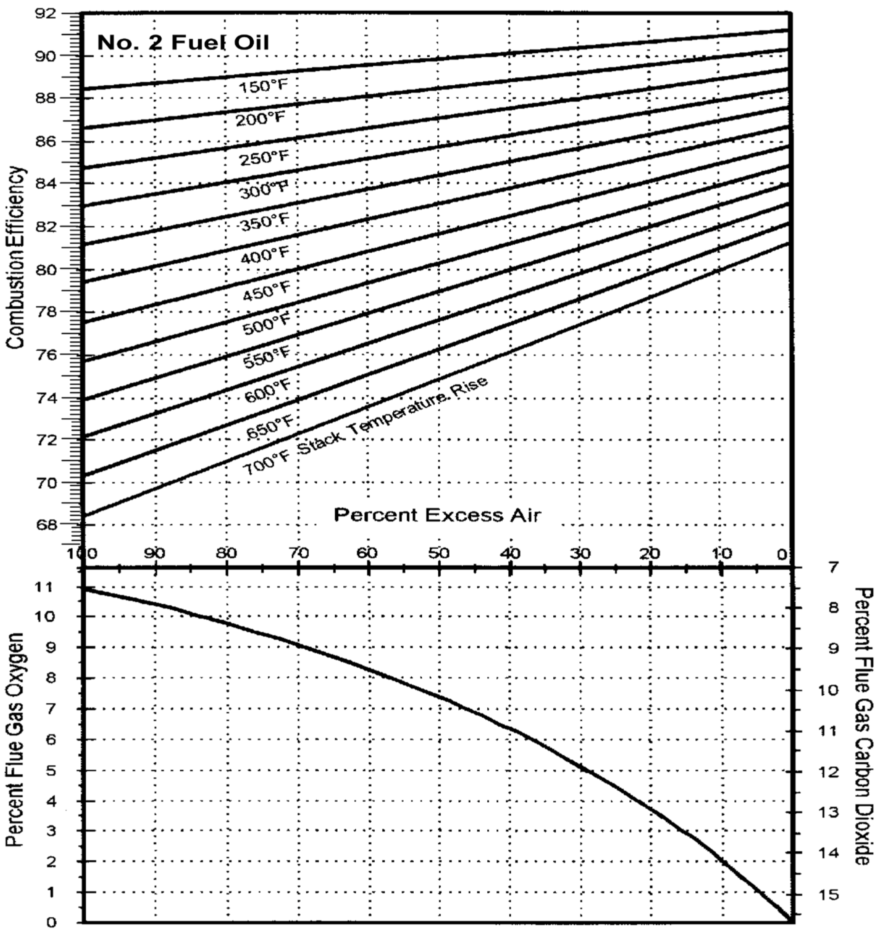


Figure 13-1C. Combustion Efficiency Chart for No. 2 Fuel Oil

Source: Boilers and Fired Systems, *Energy Management Handbook*, The Fairmont Press.

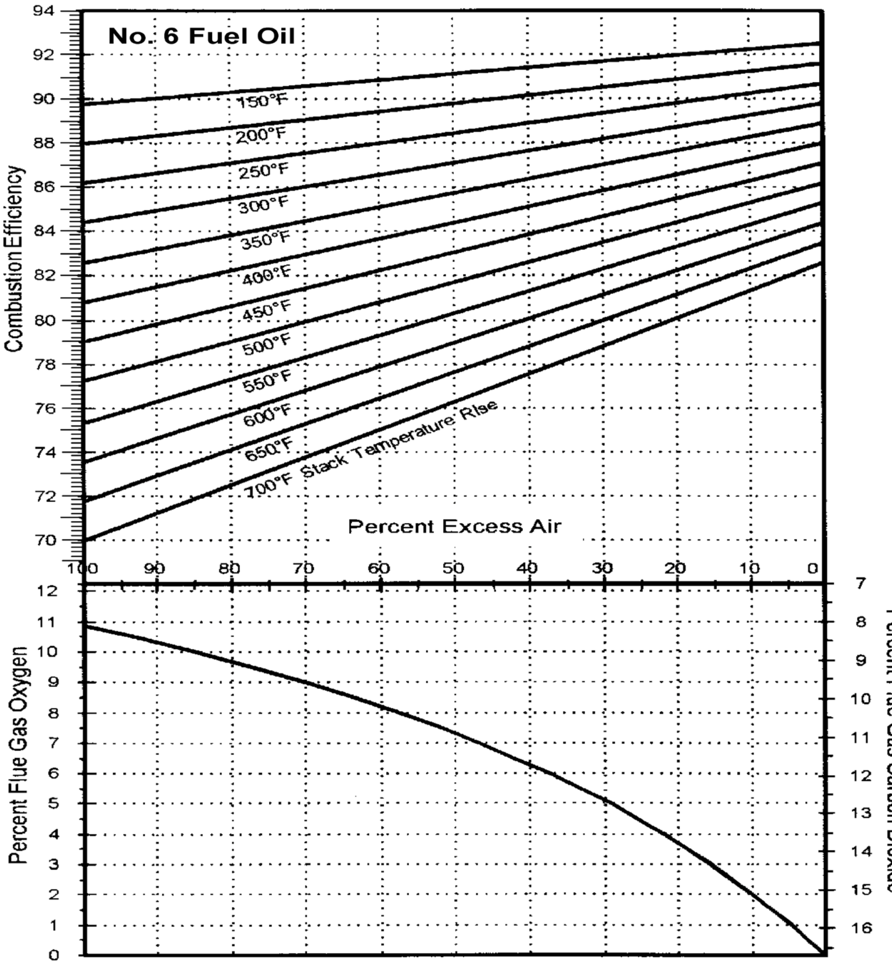


Figure 13-1D. Combustion Efficiency Chart for No. 6 Fuel Oil

Source: Boilers and Fired Systems, *Energy Management Handbook*, The Fairmont Press.

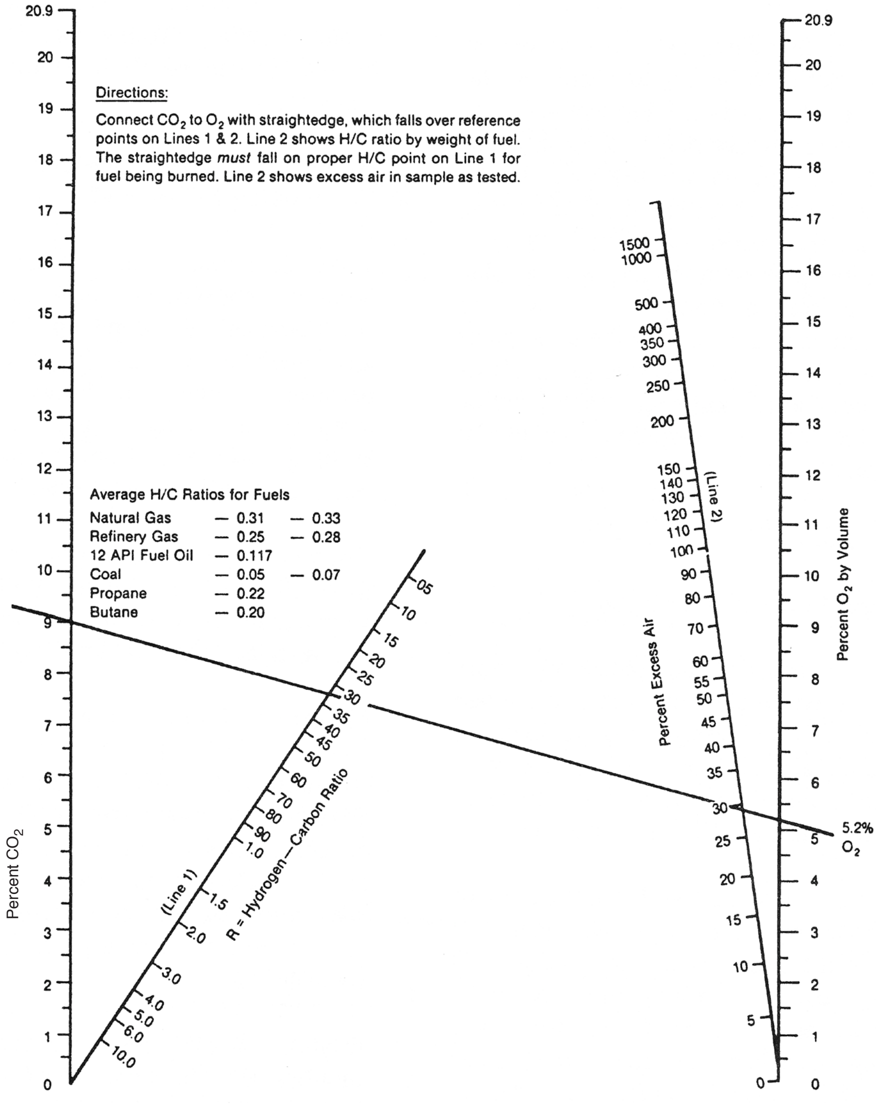


Figure 13-1E. Nomograph for Efficiency and Excess Air

Source: *Combustion Efficiency Tables*, Taplin, H., The Fairmont Press.

BOILER HEATING OUTPUT WHEN ONLY HEATING SURFACE AREA IS KNOWN

Approximate ratio is 5 SF of heating surface area per boiler hp, and 1 boiler hp = 33,500 Btuh, so Mbh Output = (Heating SF/5)*33.5.

Minimum Stack Gas Temperature vs. Corrosion

Table 13-2. Minimum stack Gas Temperatures to Avoid Corrosion Problems

Source: *Handbook of Energy Engineering*, 5th Ed, Thumann/Mehta

Fuel	Minimum Temp, deg F
Oil Fuel, >2.5%S	390
Oil Fuel, <1.0%S	330
Bituminous Coal, >3.5%S	290
Bituminous Coal, <1.5%S	230
Pulverized Anthracite	220
Natural Gas	220

BOILER STANDBY HEAT LOSS (BOILER SKIN LOSS)

About 1.5-2% of full load output.

Source: APOGEE Interactive, Inc.

These are for packaged boilers found in commercial buildings. For larger utility boilers the losses are less, in the range of 0.5% to 1% depending upon size. This is due to the larger units having a more favorable ratio of internal volume to surface area.

Losses for one manufacturer's 40MM hot water generator:

At reduced loads, the skin losses become a higher fraction of the total load on the boiler, effectively reducing its thermal efficiency. This is especially pronounced in mild weather or when using a winter heating boiler side-arm heater for summer domestic water heating.

Table 13-3.

Load (%)	Loss (%)
100	0.76
75	1.01
50	1.51
25	3.03

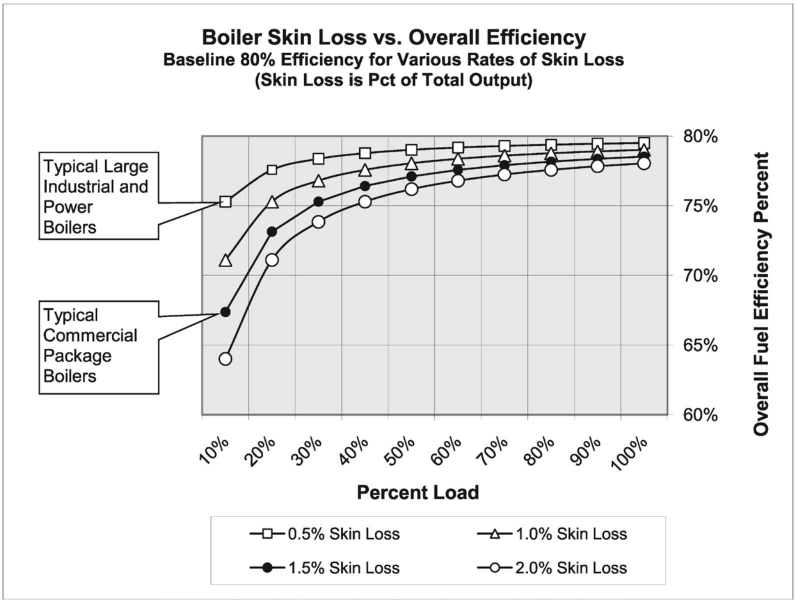


Figure 13-2a. Thermal Efficiency Reduction at Part Load Boiler Operation from Skin Loss

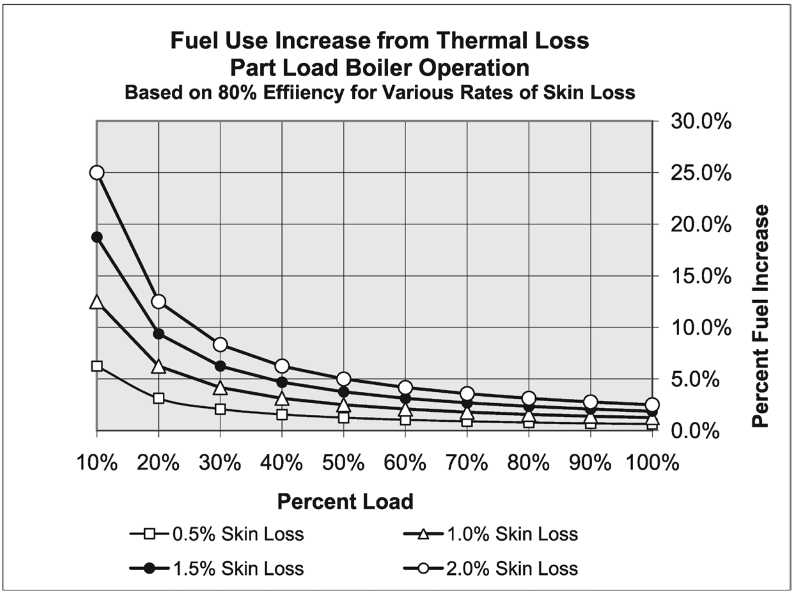


Figure 13-2b. Fuel Use Increase from Boiler Skin Loss at Part Load Operation

BOILER CYCLING LOSSES

Boilers without deep turndown burners must cycle on and off during part load operation. During periods when the boiler is off, the casing heat losses continue, creating a deferred load to pick up during the next firing cycle. The casing losses for a cycling boiler are essentially the same as a boiler idling at low fire; the difference being that the casing losses for the boiler on the off cycle gradually become less as the boiler cools off, while these losses remain constant if the boiler is left running continuously.

Other losses from cycling boilers come from pre- and post-purging air volumes that sweep heat out the chimney for safe firing.

For practical purposes, the stand-by (casing) losses define boiler part load losses, whether cycling or firing on very low fire. As with night setback, if the load is reduced for long enough, it pays to shut off the equipment. An extreme example of this is a high turn-down boiler with no load, idling at 2% of capacity. In this case, the boiler fuel use serves nothing but to maintain the boiler casing losses and efficiency is zero. If this condition occurs frequently, it makes the case for de-centralized heating or thermal storage.

Boiler Air Requirements

Stoichiometric air requirements can be calculated based on fuel, but additional air is needed for complete combustion and for ventilation. The ventilation rate may be less when appliances are direct vented.

For Forced draft burners, a rough estimate is 10 cfm per BHP (8 cfm for combustion air and 2 cfm for ventilation); plus an additional 3% for each 1000 feet above sea level.

Source: "Combustion Air Requirements for Boilers", Cleaver Brooks, Tip Sheet November 2010

This is approximately 0.3 CFM/kBtu output for boiler room air with a power burner (0.24 combustion air, 0.06 ventilation).

SAVINGS FROM VARIOUS BOILER IMPROVEMENTS (TABLE 13-4A)

Sources:
1. *Handbook of Energy Engineering*, Fifth Edition, Thumann/Mehta, 2001
2. Department of Energy
3. Natural Gas Consortium
4. Canada Energy Council
5. Minnesota Center for energy and Environment, Measured Savings from Integral Flue Dampers and thermal Vent Dampers on Domestic Hot Water Heaters in Multifamily Buildings, 1992.
6. Lutz, James D. (2009). Dampers for Natural Draft Heaters: Technical Report Lawrence Berkeley National Laboratory: Lawrence Berkeley National Laboratory. LBNL Paper LBNL-1963E.
7. S. Doty.

Table 13-4a. Estimated Savings from Boiler Improvements (*Cont'd*)

Measure	Basis of Savings	Approximate Savings	Source
Air Pre Heaters	Transfer energy from stack gases to incoming combustion air.	**1%** boiler efficiency increase for each 40 degF decrease in stack gas temperature.	1
Economizers	Transfer energy from stack gases to incoming feed water.	**1 %** boiler efficiency increase for each 10 degF increase in feed water temperature.	1
Fire Tube Turbulators	Increases turbulence in the secondary passes of fire tube units thereby increasing efficiency by increasing heat transfer.	**1 %** boiler efficiency increase for each 40 degF decrease in stack gas temperature.	1
O2 Trim for Oil and Gas Burners	Promote flame conditions that result in complete combustion at lower excess air levels	**1 %** boiler efficiency increase for each 4% decrease in excess O_2 or 20% decrease in excess air, depending upon the stack gas temperature.	1
Boiler tune-up for non condensing boilers	Adjust air-fuel mixture. Savings are from avoided degradation.	**Up to 5%** boiler efficiency increase. Savings depends on how bad it is to begin with. This estimate presumes a boiler far out of tune. One that is tuned regularly will see less improvement, e.g. **2%** if tuned up every two years, **0%** if tuned up annually.	2

Table 13-4a. Estimated Savings from Boiler Improvements (*Cont'd*)

Outdoor air reset on non condensing boilers	Reduced losses at part load, from lowering system temperatures, casing temperatures, stack temperatures.	**Up to 5%** annual fuel savings. Savings is less if the boiler itself is not reset, but only the loop temperature via a blending valve, e.g. **half of this**.	3
O2 Trim controls	Reduced excess air. Separate controls for air and fuel allow optimum mixture at all loads, which jackshaft systems cannot do.	**2 to 4%** combustion efficiency increase. Savings depends on how much excess air is currently used. Specific values are noted in **Source 1** of this table.	3
Wind box fan Throttling	Wind box fans jack-shafted to the fuel train often have the fan running at full speed and a damper varying the air flow proportionally with fuel. At low loads, the fan energy is constant. Savings come from removing the jackshaft and using variable speed fan control.	Savings follow part load profile, comparing discharge damper to VSD energy use. Note: For combustion air, the waste heat from the motor is beneficial, and the savings is from the differential cost of fuel.	7
Stack dampers	Reduced chimney losses, e.g. convective losses that pull heat out of the boiler during standby operation. For draft hood burners, savings effect is greater since the indirect draft hood connection adds about **40% dilution air (source 6)** and is a continuous source of air loss year round.	Up to **4%** annual fuel savings. Savings figure is for a natural draft burner with a draft hood. Forced draft burners with control dampers that close are much less, e.g. **1%.**	4,5

Table 13-4a. Estimated Savings from Boiler Improvements (*Concluded*)

Direct Venting (power burners)	Part 1. Heating for the combustion air is from input energy instead of output energy, and reduces the heat burden for the combustion air. "Dry Gas Loss" (DGL) is the sensible portion of total fuel used to heat the inlet air to flue gas temperature. One manufacturer of a 40MMBH burner reported DGL of 7%. Part 2. Reduces heat loss from the boiler room. Traditional hi/low louvers allow heat to escape. Direct venting reduces / eliminates room ventilation requirements, reduces auxiliary heaters to heat the space.	Part 1. ((1/e)-1)*RmHt% *DGL% Where: e = boiler nom. efficiency DGL% = Dry Gas Loss pct of fuel use RmHt% = Portion of combustion air heating that occurs prior to combustion without direct venting, e.g. 70 degF room, 30degF average winter outside air temp, 300 degF flue temp, RmHt% = (70-30)/(300-30) = 15%. **Example** for DGL=7%, RmHt%=15% for **80%**e: **0.26%** annual fuel savings For **90%e**: **0.12%** annual fuel savings Part 2. No rule of thumb for this, but if unit heaters are running in winter now, they will most likely not be running in winter if equipment is direct vented. Similar effect by closing wall louver air intake dampers on off cycle.	7

PILOT LIGHT FUEL CONSUMPTION

600 to 1500 Btu of gas per hour (residential fireplace).
Source: Natural Resources Canada, Office of Energy Efficiency, Chapter 3 - All About Gas Fireplaces

NATURAL DRAFT FLUE—DILUTION AIR

An additional 40% of air is drawn through the flue from the room in the natural draft flue arrangement.
Source: Dampers for Natural Draft Heaters: Technical Report, Lutz, James D., Lawrence Berkeley National Laboratory, 2009

The source was testing done for gas water heaters, but the principle applies to natural draft flues for heaters/boilers of all size, such as tray burners, with an indirect flue connection. This is known as "dilution air". The indirect flue collar is there to prevent excess draft at the burner. These systems create additional parasitic heating loss by removing air from the room, usually requiring additional energy expenditure to temper the replacement air.

There is a chimney to pull air out of the room at all times, even with the burner off.

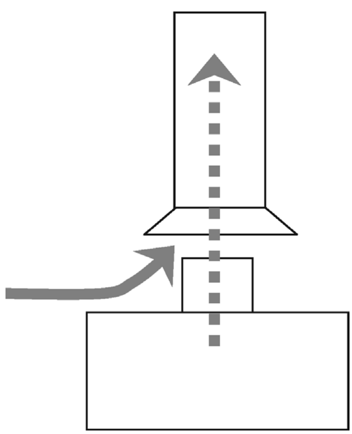


Table 13-5. Savings of Process Heating Equipment Improvements

Source: "Improving Process Heating System Performance", Industrial Technologies Program (ITP), 2006, US DOE Office of Energy Efficiency and Renewable Energy.
Savings are approximate

Heat Generation Opportunities	**Potential Savings**
Control air-to-fuel ratio at all loads	5 to 25%
Preheat combustion air	15 to 30%
Oxygen enriched combustion air	5 to 25%
Heat Transfer Opportunities	
Improve Heat Transfer with Advanced Burners and Controls	5 to 10%
Improving Heat Transfer within a Furnace	5 to 10%
Heat Containment Opportunities	
Reduce wall heat losses	2 to 5%
Furnace pressure control	5 to 10%
Maintain door and tube seals	up to 5%
Reduce cooling of internal parts	up to 5%
Reduce radiation heat losses	up to 5%
Heat Recovery Opportunities	
Combustion air preheating	10 to 30%
Fluid or load preheating	5 to 20%
Heat cascading	5 to 20%
Fluid heating or steam generation	5 to 20%
Absorption cooling	5 to 20%
Enabling Technologies Opportunities	
Install high turndown combustion systems	5 to 10%
Programmed heating temperature setting for part load operation	5 to 10%
Monitoring and control of exhaust gas oxygen and unburned hydrocarbon and carbon monoxide emissions	2 to 15%
Furnace pressure control	5 to 10%
Correct location of sensors	5 to 10%

SAVINGS FROM STEAM SYSTEM IMPROVEMENTS

Reduce Steam Pressure

One customer reported an 8% energy reduction by reducing steam header pressure from 125psig to 100 psig.

Some Rules of Thumb for Steam Boilers

Source: An Evaluation of the Losses and Costs Associated with Coal versus Natural Gas Firing at a North Carolina Textile Finishing Plant, Gibides, J. et al, *Energy Engineering*, Vol. 108, No. 5, 2011

- Radiation loss (gas or coal): 2% of annual fuel cost [assumes 1% of full fire loss, for a boiler operating at an average of 50% capacity]
- Blow down: ~ 2-3% of annual fuel use [depends on amount of make-up]
- CO (remaining energy in unburned fuel) (coal): 0-0.5% of fuel fired
- CO (remaining energy in unburned fuel) (gas): negligible
- Carbon in ash (coal): 0.5-1% of fuel fired [includes bottom ash and fly ash]
- Auxiliary electrical equipment (coal): 1.9% of annual total Btu energy use
- Auxiliary electrical equipment (gas): 0.2% of annual total Btu energy use
- 1% energy savings for each 10 psig steam pressure reduction

  Source: Optimizing Energy Saving Opportunities in Food and Dairy Plant Steam Boiler Systems, Becher, J, 2012, *Energy Engineering*, Vol 109, No. 2

Source of Data: "Reducing Steam Header Pressure Provides Attractive Operating Cost Savings," Office of Industrial Technology (OIT), 2000, US DOE Office of Energy Efficiency and Renewable Energy.

STEAM LEAKS

Table 13-6. Steam Leak Discharge Rate Table

	Steam Loss, lbs. per hour			
Trap Orifice Diameter, inches	**15 psig**	**100 psig**	**150 psig**	**300 psig**
1/32	0.85	3.3	4.8	—
1/16	3.4	13.2	18.9	36.2
1/8	13.7	52.8	75.8	145
3/16	30.7	119	170	326
1/4	54.7	211	303	579
3/8	123	475	682	1303

Source of Data: "Steam Tip Sheet #1," Industrial Technologies Program (ITP), 2006, US DOE Office of Energy Efficiency and Renewable Energy.
Original data from the Boiler Efficiency Institute. Steam is discharging to atmosphere through a re-entrant orifice with a coefficient of discharge equal to 0.72.

Figure 13-4. Steam Leak Chart

Source: "Actions You Can Take to Reduce Heating Costs," Fact Sheet PNNL-SA-43825, Jan 2005, US DOE Office of Energy Efficiency and Renewable Energy.

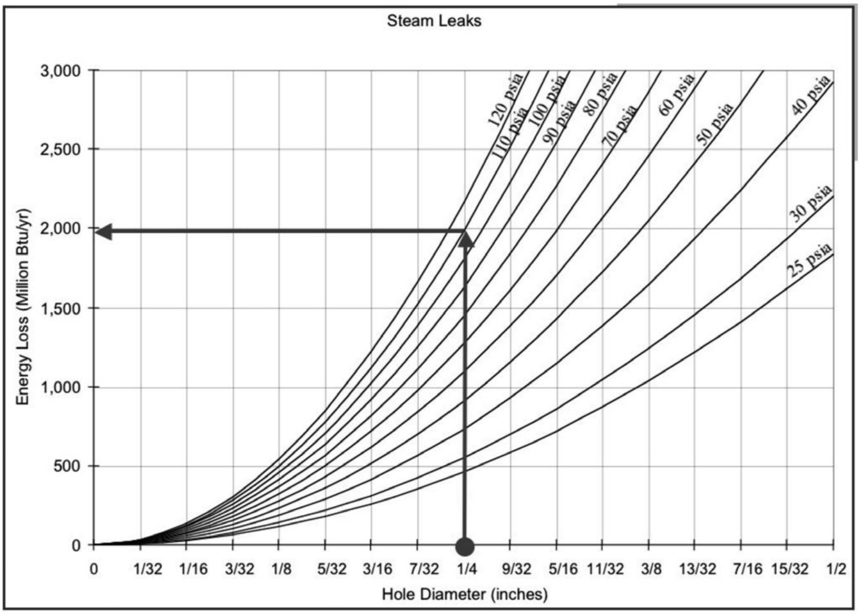


Table 13-7. Heat Loss for Un-insulated Steam Line

Distribution Line Diameter, inches	15 psig	150 psig	300 psig	600 psig
1	140	285	375	495
2	235	480	630	840
4	415	850	1120	1500
8	740	1540	2030	2725
12	1055	2200	2910	3920

Source of Data: "Steam Tip Sheet #2," Industrial Technologies Program (ITP), 2006, US DOE Office of Energy Efficiency and Renewable Energy.
Based on horizontal steel pipe, 75 deg F ambient air, no wind velocity, and 8760 operating hours per year.
Units are MMBtu/yr per 100 ft of uninsulated line

FLUE GAS RECOVERABLE HEAT

Table 13-8. Recoverable Heat from Flue Gas

Initial Stack Gas Temperature, deg F	25 MMBtu/hr Boiler Thermal Output	50 MMBtu/hr Boiler Thermal Output	100 MMBtu/hr Boiler Thermal Output	200 MMBtu/hr Boiler Thermal Output
400	1.3	2.6	5.3	10.6
500	2.3	4.6	9.2	18.4
600	3.3	6.5	13.0	26.1

Source of Data: "Steam Tip Sheet #3", Dept of Energy, Industrial Technologies Program (ITP), 2006, US DOE Office of Energy Efficiency and Renewable Energy.
Based on natural gas fuel, 15% excess air, and a final stack temperature of 250 degF.

Table 13-9. Recoverable Heat from Boiler Blowdown

Blowdown Rate, % Boiler Feedwater	50 psig	100 psig	150 psig	250 psig	300 psig
2	0.45	0.5	0.55	0.65	0.65
4	0.9	1.0	1.1	1.3	1.3
6	1.3	1.5	1.7	1.9	2.0
8	1.7	2.0	2.2	2.6	2.7
10	2.2	2.5	2.8	3.2	3.3
20	4.4	5.0	5.6	6.4	6.6

Source of Data: "Steam Tip Sheet #10", Industrial Technologies Program (ITP), 2006, US DOE Office of Energy Efficiency and Renewable Energy.

Table 13-10. Energy Savings from Installing Insulated Valve Covers

Operating Temperature, deg F	3 Inch Valve	4 Inch Valve	6 Inch Valve	8 Inch Valve	10 Inch Valve	12 Inch Valve
200	800	1,090	1,560	2,200	2,900	3,300
300	1,710	2,300	3,300	4,800	6,200	7,200
400	2,900	3,400	5,800	8,300	10,800	12,500
500	4,500	6,200	9,000	13,000	16,900	19,700
600	6,700	9,100	13,300	19,200	25,200	29,300

Source of Data: "Steam Tip Sheet #17", Industrial Technologies Program (ITP), 2006, US DOE Office of Energy Efficiency and Renewable Energy.
Based on installation of a 1-inch thick insulating pad on an ANSI 150 pound-class flanged valve with an ambient temperature of 65 degF and zero wind speed.

SAVINGS FROM REDUCING EXCESS AIR

Data for the chart was taken from Figures 13-1B, 13-1C, 13-1D (combustion efficiency charts for different fuels) according to excess air and stack temperature rise.

For each value of stack temperature rise, the stack losses were evaluated between 5.5 and 10.0 percent CO_2 and averaged on the basis of change in percent combustion efficiency change per 10% change in excess air, which are consistent for a given stack temperature.

Note that the natural gas and No. 6 fuel oil lines are on top of each other.

Example: a measure reduces excess air from 70% to 30% without changing the stack temperature which remains at 400F above air inlet temperature. Boiler is firing natural gas. Expected savings from the chart are 0.7% efficiency per 10% excess air reduction, or 2.8% efficiency improvement.

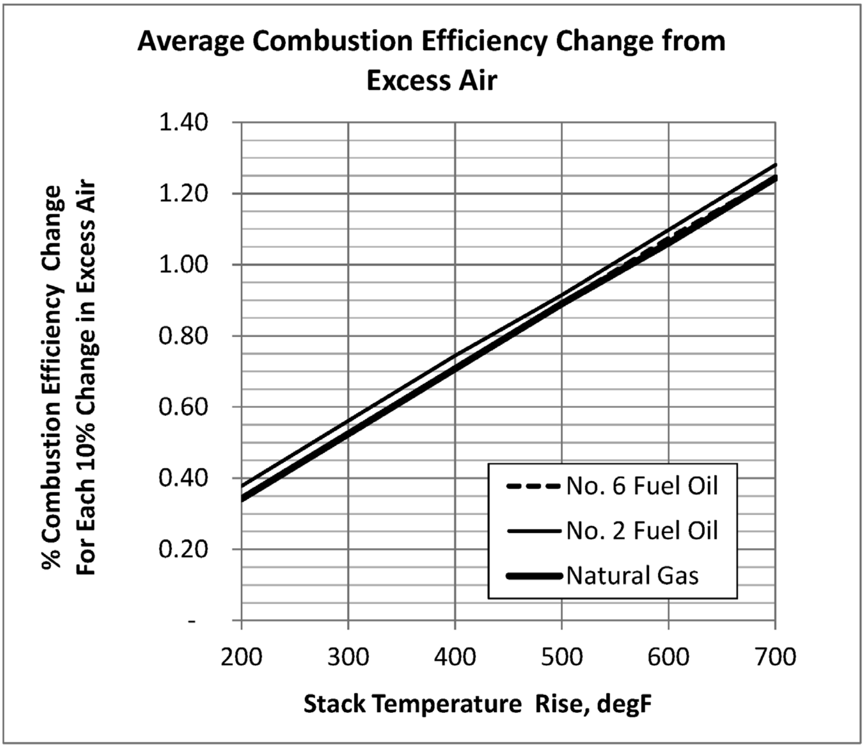


Figure 13-5. Savings from Reducing Excess Air

GENERATOR FUEL CONSUMPTION

Table 13-11. Generators: Approximate Specific Fuel Consumption

500 kW and larger.
Source: Mfg Literature
Note: Smaller motors may be 30% less efficient

	Per kW	**Per HP**
Diesel Reciprocating Engine	0.08 gph/kW	0.06 gph/Hp
Nat Gas Reciprocating Engine (1000 Btu/CF)	13 cfh/kW	9 cfh/Hp

HEAT RATE

Heat rate is a term used for electric generation, expressed in Btu per kWh. Btu heat input divided by the kWh output at the generator terminals or, on a real time basis, Btuh/kW. The more efficient the conversion, the lower the heat rate. Variations:

Gross turbine heat rate: Heat input between the feedwater heater divided by generator terminal output

Net turbine heat rate: Same as gross turbine heat rate, but the feedwater pump electric input is subtracted from the generator output as a parasitic load

Gross plant heat rate: Total heat from fuel added to the boiler divided by the generator terminal output

Net plant heat rate: Same as gross plant heat rate, but 'station power' is subtracted from generator output, e.g. this treats the power plant as a box with fuel heat input and electricity output (leaving the plant)

Chapter 14

Compressed Air

Acknowledgment: This chapter was co-authored by Kevin Carpenter, PE CEM, Clark Energy Group, Arlington, Virginia

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OVERALL EFFICIENCY OF COMPRESSED AIR

Compressed air is a friendly medium for manufacturing. Air-operated tools are inexpensive, simple, and durable; the negative aspect is energy efficiency. To run a 1 Hp air-operated tool requires about 7 Hp of air compressor motor power. Even with theoretical compressors free from friction and turbulence, the ratio would still be 4:1 or 5:1. Thermodynamics says we cannot make compressed air efficient, but we can do things to make it less inefficient.

Power for compressed air is a function of flow and pressure. Some compressors achieve part load flows without corresponding power reduction. Friction losses vary by the square of velocity for a given density. Knowing these and other concepts allows measures to be ranked by fundamental improvement potential. Opting for high efficiency equipment on purchase is obvious, so is not listed.

Compressed Air Efficiency Measures

General Measure	Examples
Use less air	• Leaks (piping, hose connections, float drains, timed drains). • Separate any 24x7 small continuous uses and serve separately at night so main system can be off instead of serving leaks all night • Automatic shut off control for air instead of constant use at machines, constant drain air flow, etc. • Air amplifiers (eductors) for blow-off end uses. These entrain room air and reduce compressed air use by 2/3 or more. • Electric motor in lieu of air motors • Electric cabinet cooling in lieu of compressed air cooling • Turn compressors off at night • Desiccant dryer improvements to reduce purge air • Relax dew point requirements if excessively low, to reduce desiccant purge air • Desiccant dryer interlock so purge losses stop when compressor is off, allowing it to stay off when plant is idle • Automatic-shutoff air solenoid valves
Lower the pressure	• Relieve restrictive piping • End use storage for large points of use removes the need to elevate overall pressure for occasional large demand • Lower pressure for separate shift if different needs • Blower for high-volume low-pressure uses like parts blow-off • Select filters and dryers with low rated pressure drop
Reduce throttling losses	• Base loading to avoid throttling • Variable speed for trim • Avoid inlet throttling for screw compressors • Avoid blow-off throttling for centrifugal compressors • Storage as an enabler for on/off and load/unload
Compressor efficiency	• Low resistance inlet piping • Cooling of inlet air temperature • Maintain cooling system for robust heat rejection.

STANDARD SCFM VS. ACTUAL ACFM

CFM is cubic feet per minute.

ACFM is actual cfm for a given site's air temperature, pressure and humidity conditions.

SCFM is a standardized set of conditions for compressor manufacturers.

SCFM defines the volumetric flow rate of an air compressor when compressing standard temperature and pressure (STP) air and therefore represents a standardized mass flow rate. The standardization of inlet conditions allows consistency in equipment ratings across different manufacturers. Note that the standardized conditions are not consistent in all industry sources. Standardized conditions in this text are per the Compressed Air and Gas Institute (CAGI), 2012.

Standard conditions for air compressors as of 2012 are 14.5 psia, 68F, 0% rH.

Source: CAGI, 2012

Author's note: Standard conditions are used in many fields and are close but not exactly the same. Differences are minor.

The general formula for converting SCFM to ACFM:

$$\mathbf{ACFM = SCFM * \left({}^{[Ps-(PVs*RHs)]}/_{[Pa-(PVa*RHa)]}\right) * (Ta/Ts) * \left({}^{Pa}/_{Pinlet}\right)} \qquad \textbf{(eq.1)}$$

Where:

Ps	=	Standard pressure
Ts	=	Standard Temperature, degR
PVs	=	Vapor pressure of water, psia (at standard temperature) (**Table 14-1**)
RHs	=	Standard relative humidity, %
Pa	=	Atmospheric pressure, psia (at site) (ref. **Table 14-2**)
Pinlet	=	Inlet pressure, psia (atmospheric – inlet pressure drop)
PVa	=	Vapor pressure of water, psia (atmospheric, at site) (**Table 14-1**)
RHa	=	Relative humidity, % (atmospheric, at site)
Ta	=	Atmospheric Temperature, degR (at site)

With relative humidity for standard conditions at 0 rH and allowing 0.2 psia for inlet piping pressure drop, the formula reduces to this.
Source: CAGI, 2012

$$\textbf{ACFM} = \textbf{SCFM} * \left(^{Ps}/_{[Pa-(PVa*RHa)]}\right) * \left[^{(Ta+460)}/_{(Ts+460)}\right] \qquad \textbf{(eq. 2)}$$

Where:

Ps = Standard pressure, **14.5** psia
Ts = Standard Temperature, **68** degF
Pa = Atmospheric pressure, psia (at site) (ref. **Table 14-2**)
PVa = Vapor pressure of water, psia (atmospheric, at site) (**Table 14-1**)
RHa = Relative humidity, % (atmospheric, at site)
Ta = Atmospheric Temperature, degF (at site)

Compressor air flows are described in terms of cfm, but remember this is referencing the inlet air conditions. So, a compressor indicating 50 cfm rated delivery at 100 psig should not be interpreted as the flow of air in its compressed state at 100 psig, but rather 50 cfm air flow rate at standard pressure and temperature, prior to being compressed. **Consider the compressor cfm rating as the air flow being drawn from the room or inlet pipe**. It is also conventional to define the usage rates of pneumatic end devices (cylinder actuators, pneumatic motors, etc.) in SCFM (inlet air conditions), which allows direct correlation between end use requirements and source capacity. However, piping evaluation requires knowing the actual density of air to understand its velocity, and storage evaluation requires knowing the pressure to understand the mass of air that fits within a given volume.

Summary of ACFM vs. SCFM.

- In general, cfm for compressors reference inlet air.
- When inlet conditions match standard conditions, ACFM and SCFM are the same. SCFM is cfm at standard conditions; ACFM is cfm at site conditions.
- The SCFM capacity of air compressors reduces with altitude, by the factor

  $\text{SCFM}_{\text{reduced}}/\text{SCFM}_{\text{standard}}$ which is equal to
  $\textbf{Pressure}_{\text{reduced (altitude)}}/\textbf{Pressure}_{\text{standard}}$ **(eq. 3)**

- Compared to standard conditions, if inlet pressure is lower or if inlet temperature is higher, air density is lower than at standard air conditions; air expands and it takes more volume for a given amount of mass.
- For compressors, the machine rating is in volume passed per unit of time (cubic feet per minute). When air is less dense than standard conditions, more input flow in ACFM is needed to get the proper weight of compressed air to equal the required SCFM value.
- Decreased density reduces the power requirement for a given ACFM air flow rate compressed to a given gage pressure.
- When inlet air density is lower than standard conditions, the compressor must be up-sized in order to deliver the required SCFM flow rate.

Some implications of ACFM vs. SCFM.

- The unit cost of air in kW/SCFM increases with altitude.
- Large inlet piping pressure drops and high inlet temperatures act in the same way as altitude: the air expands, the density reduces, the gap between ACFM and SCFM widens, machine capacity decreases, compression ratio increases, SCFM/Hp decreases, kW/SCFM increases.
- Use ACFM for compressor selection.
- Use ACFM for intake piping evaluation. Air at a lower density (high altitude) imposes higher volumetric flow for a given SCFM which in turn creates higher velocities and pressure drops which increases the compression ratio and compressor power requirements. These additional losses can be prevented with larger piping.
- Use SCFM when estimating the cost of air usage and leaks, i.e. for kW/cfm-hour and cfm/Hp, referencing SCFM.
- Use SCFM when estimating purge air flow for a desiccant drier. Standard desiccant dryer timed control would be 15% of compressor full load 'nameplate' SCFM; 'energy management' control (purging proportional to system air flow) would be 15% of actual SCFM system flow.

The usual use for the ACFM conversion is selecting an air compressor that is rated in the standard units. But there are other uses.

Example:

New compressor purchase. 500 SCFM is needed for a process. An allowance of 20% leaks is presumed and another 20% for future expansion, for a total of 720 SCFM. The plant is located at 6000 ft. elevation, and drawing air that is an average of 80F and 25% rH. Find the minimum required capacity for the compressor in SCFM units, and the compressor de-rate factor.

Solution:

At 6000 ft. elev. Atmospheric pressure is 11.8 psia (**Table 14-2**). At 80 degF, water vapor pressure is 0.5068 psia (**Table 14-1**).

SCFM required capacity =

$$ACFM = SCFM\ end\ use * \left({}^{Ps}/_{[Pa-(PVa*RHa)]}\right) * \left[{}^{(Ta+460)}/_{(Ts+460)}\right]$$

$$SCFM\ required\ capacity = ACFM = 720 * \left({}^{14.5}/_{[11.8-(0.5068*0.25)]}\right) * \left[{}^{(80+460)}/_{(65+460)}\right]$$

ACFM = <u>920 CFM</u> (required compressor capacity)

Compressor de-rate factor is 720/920 = <u>0.7826</u>

Table 14-1. Saturated Water Vapor Pressure by Temperature

Values from saturated steam-temperature table

Temp F	Vapor Pressure psia	Temp F	Vapor Pressure psia	Temp F	Vapor Pressure psia
32	0.8859	62	0.2749	92	0.7431
34	0.9600	64	0.2950	94	0.7906
36	0.1040	66	0.3163	96	0.8407
38	0.1125	68	0.3389	98	0.8936
40	0.1216	70	0.3629	100	0.9492
42	0.1314	72	0.3884	102	1.008
44	0.1419	74	0.4155	104	1.070
46	0.1531	76	0.4442	106	1.135
48	0.1651	78	0.4746	108	1.203
50	0.1780	80	0.5068	110	1.275
52	0.1917	82	0.5409	112	1.351
54	0.2063	84	0.5770	114	1.430
56	0.2218	86	0.6152	116	1.513
58	0.2384	88	0.6555	118	1.601
60	0.2561	90	0.6981	120	1.693

Table 14-2. Atmospheric Pressure by Altitude

Altitude (ft. above sea level)	Absolute Pressure psia
0	14.7
1000	14.1
2000	13.6
3000	13.1
4000	12.7
5000	12.0
6000	11.8
7000	11.3
8000	10.9
9000	10.5
10,000	10.1

After leaving the drier, moisture effects are nil and a generic version can be used to define the air flow at any point in the system; for example, to check velocity and pressure drop.

$$\mathbf{cfm2 = cfm1 * {}^{P1}/_{P2} * {}^{T2}/_{T1}} \qquad \textbf{(eq. 4)}$$

Where:

P1 = Initial pressure, psia
P2 = New pressure, psia
T1 = Initial temperature, degR
T2 = New temperature, degR
cfm = volumetric flow rate in consistent units, typically SCFM

This says if the new pressure (in absolute) is half, the new flow will be double (less compressed); and if the new temperature is double (in absolute), the new flow will be double (more expanded)

For compressed air distribution within a building, assuming temperatures of piping in different pressure zones equalize to room temperature, the formula simplifies further.

$$\mathbf{cfm2 = cfm1 * {}^{P1}/_{P2}} \qquad \textbf{(eq. 5)}$$

Where:

P1 = Initial pressure, psia
P2 = New pressure, psia
cfm = volumetric flow rate in consistent units, typically SCFM

COMPRESSOR EFFICIENCY

"The industry norm for comparison of compressor efficiency is given in terms of bhp/100 acfm (brake horse power per actual cubic feet per minute at a compressor discharge pressure of 100psig."
Source: Compressed Air Challenge, Fact Sheet #8, April 1998

Note the use of BHP, (brake horsepower), which represents actual horsepower required at the input shaft of the compressor. Calculating this metric using nameplate horsepower will incorrectly estimate the efficiency of the compressor unless it happens to be using 100% of the available power. It is common for air compressor manufacturers to burden motors beyond their nameplate capacity at maximum output conditions, i.e. allowing operation in the motor's service factor.

> Note: The service factor is a designation for a motor's capability to operate in excess of nominal power for short periods. When operated in this 'overload' state for prolonged periods, issues related to motor heating are anticipated.

Another common expression for air compressor efficiency is SCFM per hp, which is the reciprocal of hp/SCFM.

Compressor performance follows a polytropic process. Actual efficiency is somewhat less than predicted due to machine losses including friction and turbulence. There are various expressions of the polytropic equation. To express in units of required work (Btu/lbm):

$$W = \left[\frac{n}{n-1}\right] * R * T1 * \left[\left(\frac{P2}{P1}\right)^{\frac{n-1}{n}} - 1\right] \qquad \textbf{(eq. 6)}$$

Where:

W=work in, Btu/lbm
n = polytropic constant, value between 1.0 and 1.4 but can be assumed to be 1.12 for a compressor with a well-functioning heat rejection system (Note 1).
R = gas constant for air (Btu/lbm-R), 0.06855
T1 = inlet temperature, degR
P1 = inlet pressure, psia
P2 = discharge pressure, psia

Note 1: The polytropic constant will go up (closer to 1.4) with worse heat rejection, and will go down (closer to 1.0) with better heat rejection. The 1.0 would indicate purely isothermal compression, and 1.4 would indicate purely adiabatic compression.

Converting equation 6 output to kW input requires additional steps:

Example: For given conditions, **equation 6** indicates 83.2 Btu/lb. mass input work requirement.

The air flow rate is 1000 scfm and air density is 0.075 lbm/ft3, so Mass flow rate is 1000 ft3/min* 0.075lb/ft3 * 60min/hr. = 4500 lbs./hr. Then,

Power requirement is 83.2 Btu/lbm * 4500 lbm/hr. = 374,400 Btu/hr. or

374,400 Btuh/3413 Btuh/kW = 109.7 kW (ans)

This is the theoretical power, without machine losses. Actual input power will be increased by machine efficiency losses and motor efficiency losses. These calculations are easily handled with a spreadsheet.

Theoretical work and power from **equation 6** is less than actual work required by real machines due to friction, turbulence and other machine losses. Working with a particular class of compressors, there may be reliable test data from a manufacturer that gives machine input kW or brake Hp at given conditions, and a factor can be determined so equation 6 closely matches the actual power requirements of the real machines.

Example: a single stage screw air compressor is found to produce 4.50 SCFM/Hp at 100 psig discharge, 14.7 psia inlet pressure, and 68F inlet temperature. With this data point, **equation 6** can be entered into a spreadsheet along with an efficiency factor which is determined through iteration until a value of exactly 4.50 SCFM/Hp is produced. Using this example the values of **equation 6** were found to closely match the real machine power requirements with a machine efficiency factor

Actual work in = Polytropic work in * 1/eff$_{machine}$ (eq. 7)

In this example, the machine efficiency was found to be 0.66175.

Using **Equation 6**, polytropic work was determined to be 6.8 scfm/hp. Actual rated work input is 4.5 scfm/hp. The ratio of the two values equals 0.66175.

This value is not universal but will bring reasonable results for similar machines.

Completing the prior example:

Theoretical power was determined to be 109.7 kW, input to the compressor

Actual input power = theoretical power * 1/eff$_{\text{machine}}$ * 1/eff$_{\text{motor}}$ **(eq. 8)**

For a 94% efficient electric motor and a compressor efficiency of 66.175%:

Actual input power = 109.7 * 1/0.66175 * 1/0.94 = 176.4 kW (ans)

COMPRESSED AIR COST

The dollar value depends upon the cost of electricity. The kWh used in compressed air is:

kWh/SCFM-hour = [1/(SCFM/Hp)] * 0.746/motor eff **(eq. 9)**

The value of kWh/cfm-hour is used directly in energy calculations. For efficiency comparisons between compressors, it is common to use kW/100 SCFM for more comfortable numbers. (**Figure 14-1** and **Table 14-3**).

The full flow output rating of the compressor (cfm per Hp) varies by compressor type and discharge pressure. **Part load output performance (cfm per Hp) varies depending upon the capacity control methods used by the compressor**. Using full load compressor SCFM per Hp can understate the cost of compressed air if the compressor operates a lot of the time at a part loaded state.

For example, if a compressor is mechanically throttled to 50% capacity but is using 75% of motor horsepower due to throttling losses, a full load efficiency of 4.5/Hp is now 4.5*(0.5/0.75) = 3.0 SCFM per kW. And this same compressor operating at 25% flow with 50% power is now 4.5*(0.25/0.5) = 2.25 SCFM per Hp.

Thus the power requirements in kW/cfm for a given air compressor varies with the unloading characteristics, and the energy usage varies according to the power and time spent at each condition of SCFM. Determining the SCFM/Hp for each operating condition requires know-

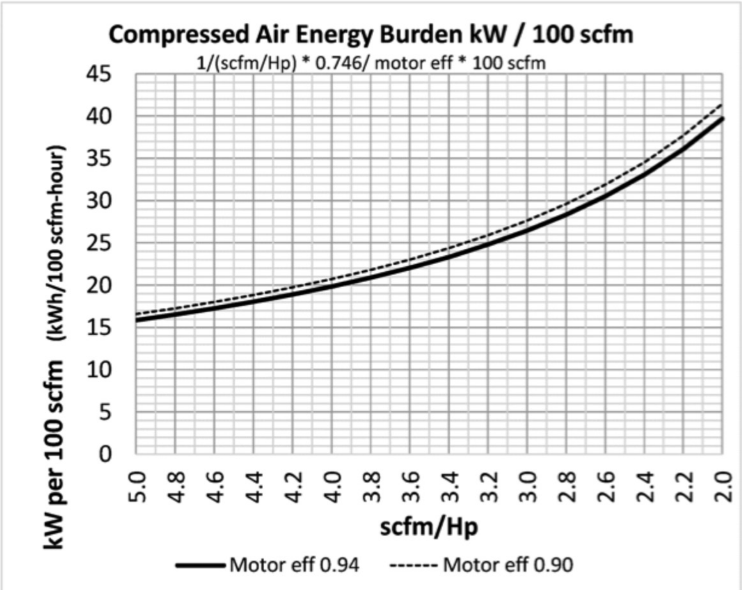


Figure 14-1. Energy Burden in Compressed Air

Table 14-3. Values for Figure 14-1

	Motor Efficiency 0.94		Motor Efficiency 0.92		Motor Efficiency 0.9	
scfm/ Hp	kW/ 100 scfm	kWh/ scfm-hr	kW/100 scfm	kWh/ scfm-hr	kW/100 scfm	kWh/ scfm-hr
5.0	15.9	0.159	16.2	0.162	16.6	0.166
4.8	16.5	0.165	16.9	0.169	17.3	0.173
4.6	17.3	0.173	17.6	0.176	18.0	0.180
4.4	18.0	0.180	18.4	0.184	18.8	0.188
4.2	18.9	0.189	19.3	0.193	19.7	0.197
4.0	19.8	0.198	20.3	0.203	20.7	0.207
3.8	20.9	0.209	21.3	0.213	21.8	0.218
3.6	22.0	0.220	22.5	0.225	23.0	0.230
3.4	23.3	0.233	23.8	0.238	24.4	0.244
3.2	24.8	0.248	25.3	0.253	25.9	0.259
3.0	26.5	0.265	27.0	0.270	27.6	0.276
2.8	28.3	0.283	29.0	0.290	29.6	0.296
2.6	30.5	0.305	31.2	0.312	31.9	0.319
2.4	33.1	0.331	33.8	0.338	34.5	0.345
2.2	36.1	0.361	36.9	0.369	37.7	0.377
2.0	39.7	0.397	40.5	0.405	41.4	0.414

ing the percent capacity and unloading methods. Understanding the potential part load energy impact draws attention to things such as:

Opportunity Item	Because
Systems with wide swings in compressed air flow	Part load controls will be in effect
Systems that spend significant time in low flow condition	Increases the magnitude of part load losses
Using multiple compressors sequenced to keep operating compressors at high percent flow	Reduces part load losses of individual compressors
Evaluating control options	Some control methods are better (or worse) than others at
	reducing input power with flow reduction. The reduction of SCFM/hp is more pronounced below 50% flow output
Using a variable speed drive compressor for the swings in load.	These compressors vary compressed air output by varying motor speed. Percent power follows percent flow and SCFM/hp is nearly constant at part load

Properly identifying the cost of a unit of compressed air enables accurate estimates for cost of operation, improvements, and leaks. Assuming a constant value of SCFM per Hp introduces errors in most cases. Examples where compressor SCFM per Hp will not vary significantly:

- On-off control
- Variable speed control
- Staging that allows operation without mechanical throttling

Normally the energy analysis is applied to known total flows vs. time, e.g. a load profile logged from a flow recorder.

<u>Example #1: Varying Values of Compressor SCFM per Hp</u>

Compressed air output profile is reviewed against the compressor part load power characteristics to determine the SCFM per Hp on a time-series basis. This allows improved accuracy of energy use estimates for the compressed air system at the different conditions, compared to assuming a constant value of SCFM per horsepower air delivery. **See Example 1 Results—Varying Values of Compressor SCFM per Hp**. In this example actual kWh is 33% higher than presuming full load efficiency at all loads. This example also points out the need to know unloading control characteristics along with an air flow profile to accurately determine energy use.

motor eff	**0.94**			Assuming				Actual, incorporating				
max SCFM	**750**			**Constant SCFM/Bhp**				**Varying SCFM/Bhp**				
SCFM/Bhp at max output	**4.5**											
									Base SCFM/ Hp * % flow / % full load power	1/ (SCFM/Hp) * 0.746 / motor eff	(kW/ 100 SCFM) / 100	kWh/ SCFM-hr * cfm * hours
Time	SCFM	hours	pct flow	SCFM / Bhp	kW/ 100 SCFM	kWh/ SCFM-hr	kWh	pct full load power	SCFM / Bhp	Kw /100 SCFM	kWh/ SCFM-hr	kWh
12:00	600	1.0	80%	4.5	17.6	0.176	106	85%	**4.24**	18.7	0.187	112
13:00	550	1.0	73%	4.5	17.6	0.176	97	80%	**4.13**	19.2	0.192	106
14:00	615	1.0	82%	4.5	17.6	0.176	108	87%	**4.24**	18.7	0.187	115
15:00	510	1.0	68%	4.5	17.6	0.176	90	76%	**4.03**	19.7	0.197	101
16:00	575	1.0	77%	4.5	17.6	0.176	101	83%	**4.16**	19.1	0.191	110
17:00	590	1.0	79%	4.5	17.6	0.176	104	84%	**4.21**	18.8	0.188	111
18:00	250	1.0	33%	4.5	17.6	0.176	44	58%	**2.59**	30.7	0.307	77
19:00	200	1.0	27%	4.5	17.6	0.176	35	52%	**2.31**	34.4	0.344	69
20:00	180	1.0	24%	4.5	17.6	0.176	32	50%	**2.16**	36.7	0.367	66
21:00	200	1.0	27%	4.5	17.6	0.176	35	52%	**2.31**	34.4	0.344	69
22:00	190	1.0	25%	4.5	17.6	0.176	34	50%	**2.28**	34.8	0.348	66
23:00	205	1.0	27%	4.5	17.6	0.176	36	52%	**2.37**	33.6	0.336	69
0:00	195	1.0	26%	4.5	17.6	0.176	34	50%	**2.34**	33.9	0.339	66
							total					total
							857					**1136**
						presumed	kWh				actual	kWh

Example 1 Results—Varying Values of Compressor SCFM per Hp

Notes for this example:

- Screw compressor with slide valve capacity control
- SCFM values are measured hourly
- The percent flow value is calculated from (flow/maximum flow) for the compressor.
- The percent of motor power (Bhp) is estimated from **Figure 14-2—Part Load Power Characteristics, Positive Displacement Air Compressors**
- The group 'constant SCFM/Hp' assumes the maximum compressor efficiency is available at all loads (the source of error)
- The group 'varying SCFM/Hp' is closer to actual values.

COMPRESSOR CAPACITY CONTROL

Compressors serving a variable demand must have some way to match the amount of air that is needed at its end use. Different methods of air flow capacity modulation have different characteristic power reduction signatures. Most methods do not turn down kW proportionally to air flow.

On-Off Control

Cut-in/cut-out pressures cause the compressor to start and stop. This is the most efficient control method since there are no false-loading losses and the compressor draws no power when it is not producing air. I.e. when half the air flow is needed, the compressor runs half of the time and the kWh is halved. Air storage is required with this control method to allow a suitable off-time period consistent with motor cooling. Pressure swings between cut-in/cut-out settings may be a 5-10 psi swing, or as needed for start/stop frequency to be compatible with motor life. Storage is needed for this control method as a 'flywheel', to allow reasonable time for alternating between cut-in and cut-out points. Greater storage will increase the time between start and stop, or greater storage can be leveraged for closer pressure control (reduced cut-in/cut-out span).

Technically, start-stop control is possible with any compressor type. However, frequent starting and stopping of large motors significantly reduces motor life, so modulation without stopping the motor is normal practice.

Screw Compressor Capacity Control (Positive Displacement)

Variable Speed Control

The output is varied by decreasing or increasing the speed of rotation. Minor energy losses are associated with variable speed drives, but overall part load energy use is nearly equal to on-off control. Pressure control is very tight (can control directly to a desired output vs. a control range above and below from start/stop or load/unload), and storage requirements are low.

Load/Unload Control

Cut-in/cut-out pressures initiate load or unload operation. When 'loading', the compressor works at full output, adding air to the system. When 'unloading', the inlet valve is mostly closed and remaining compressed air in the sump is vented to atmosphere. Load/unload control works best when coupled with adequate compressed air storage because the storage acts like a flywheel, lengthening load/unload cycles and dampening pressure swings. There is usually an 'automatic' shutoff option paired with this control method, turning the compressor off if it runs unloaded for a period of time, such as 5-10 minutes. Pressure control swings between cut-in/cut-out settings which may be a 5-10 psi swing.

Storage is needed for this control method to allow reasonable time for alternating between cut-in and cut-out points. Increasing storage capacity will increase the time between load and unload, or can be leveraged for closer pressure control (reduced cut-in/cut-out span).

Variable Capacity Control

Capacity is reduced by various mechanical means, with the effect of the air compression machinery behaving like a smaller unit, i.e. changing the effective rotor length. Slide valves and poppet/plug valves are two methods. The affected portion of the rotor only partly compresses the air which is then shunted back to the inlet, thus reducing machine output. Pressure control does not have the characteristic high/low pressure oscillation found with on/off and load/unload control.

Modulation Control

An inlet air valve is modulated toward closed to reduce capacity by false loading (raising the compressor's compression ratio). Efficiency is poor, but pressure control is very precise with this method.

Centrifugal Compressor Capacity Control

Inlet Vane and Blow-Off Control (Constant Pressure Mode)

Inlet vane control throttles and pre-rotates the incoming air and power reduction follows flow reduction. Inlet vane throttling is limited to approximately 70-100% of full load output to avoid low load surging. Below this limit, other throttling methods are needed. When capacity control by inlet valve/vane is no longer viable, compressed air is vented to atmosphere, false loading the machine. In 'constant pressure' mode pressure control is tight but part load efficiency is poor. For compressors using blow-off control, compressor power below about 70% flow is constant.

Auto-Dual Mode

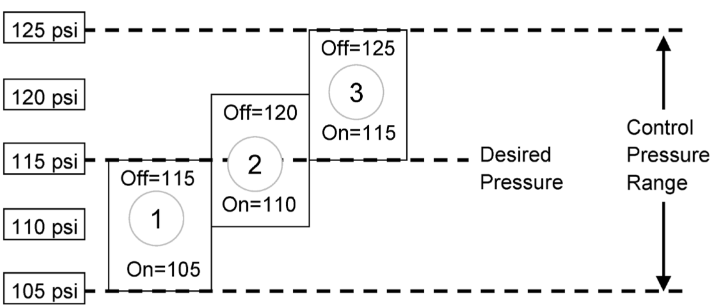
This combines constant pressure and load-unload control modes. Above the approximately 70% output range, inlet vanes throttle capacity with close pressure control and power reduction tracks air flow reduction. Below the limit of inlet vane control, the compressor operates in load-unload mode. When 'loaded', the compressor power capacity and power correspond to the minimum inlet vane position. When 'unloaded', compressor output is zero and power is low, but not zero. Values of 20-25% may be possible with some compressors but values of 30-45% have been observed in the field. This mode omits blow-off loss. Pressure swings exist while operating in load-unload mode. Sufficient storage is needed to enable load-unload operation.

Compressor Sequencing

Where multiple compressors exist, varying loads can be met by staging and base loading, such that only one compressor is experiencing part load losses. This approach is much more efficient than having multiple compressors each in a modulating mode. Sequencing choices come from a master controller with a single main pressure reading and inputs from the respective compressors. The control algorithm considers flow capacity, full load efficiency (cfm per kW), and part load power characteristics. Base load units will be those with the highest value of SCFM per kW and kept at or near full load, at a constant output; one compressor will 'swing' or 'trim' to fine tune the group output.

Compressor sequencing is accomplished by sensing the main pressure at the discharge receiver tank or main piping downstream of it. Basic sequencing is accomplished by staggering individual machine settings for output psi requirement using pressure switches. Example:

Trim	Compressor No. 1 = Cut in @ 105 psig/Cut out @ 115 psig
Trim	Compressor No. 2 = Cut in @ 110 psig/Cut out @ 120 psig
Base Load	Compressor No. 3 = Cut in @ 115 psig/Cut out @ 125 psig



Compressed Air Sequencing

In this example compressor 3 is base load and always runs and compressors 2 and 3 trim the load. As load increases and pressure drops, compressor will start and, if it cannot keep up, compressor 1 will start. This type of sequencing functions fine for plant air but creates a relatively wide overall control band. In the example, the control pressure range is 100-125 psi and power requirements are higher than necessary if, say 115 psi is the functional requirement for main air supply. Cut-in/cut-out are the actions of the pressure switches, which can either cause the compressors to start/stop or load/unload. Sequencing strategies must consider that frequent starting is hard on a compressor/motor and a compressor in 'unload' mode still sees considerable power requirements. Referring to the example, setting all three compressors for load/unload would result in all three operating in unload mode at low system flow, with significant part load energy waste. But part-load losses can be reduced by activating 'auto shutoff' so that compressors automatically shut off when unloaded for a user-defined period of time (usually 10-15 minutes). Alternatively, setting the trim compressors (#1 and #2) to on/off and the base load unit #3 to load/unload may make sense, provided sufficient storage exists so that the frequency of starts falls within NEMA guidelines.

'Master controllers' offer improvements over basic pressure switch controls. With PID controls and adjustable parameters for individual machines, the pressure range can be narrowed which avoids higher power associated with higher pressures (e.g. 110-120 psi range vs. 105-125 psig range in the pressure switch **example**). The additional control sophistication allows optimizing and reduced energy consumption, mostly from avoided part load losses.

Energy is valuable, but so is production up time. Compressor operating choices may be based on 'what is left' if one unit fails, i.e. operating redundancy without plant interruption. Integral to automatic supervisory master controller sequencing will be alternate choices in the event that one unit has failed. Excessively conservative operating modes from fear of equipment failure (esp. old equipment) can compromise optimum energy use and form a business case for air compressor replacement.

<u>Example 2: Compressor Sequencing Scenario</u>

Four compressors serve a varying air flow demand. Each has a different capacity and motor size. One of the compressors is variable speed, and the others have conventional control options of modulation and load-unload. The example illustrates the potential for overall efficiency gain when sequencing controls are used to minimize throttling losses.

Proposal A: Compressor Sequencing Scenario changes from inlet modulation mode to load-unload mode.

Proposal B: Compressor Sequencing Scenario base loads one compressor to 100% and all trim is from the variable speed unit.

> *Note*: **Proposal B** *uses less energy than the existing condition, but there are operational considerations. If the either of the two compressors fail, the remaining compressor does not have capacity to compensate. So this method relies on the next available compressor to be able to quickly start and come up to pressure which in turn relies on adequate air storage to provide the necessary time. Energy savings proposals must not impact the process.*

Notes for this example:

- Screw compressors
- The percent flow value is calculated from (flow/maximum flow) for the compressor.
- The percent of motor power (Bhp) is estimated from **Figure 14-2—Part Load Power Characteristics, Positive Displacement Air Compressors**

Measures That Can Reduce Part Load Losses

- First, identify the system flow requirements and turndown characteristic (load profile). For example, if the load is always near full load, part load losses will be minor.
- Turn off compressors when air is not needed including non-production shifts
- Turn off refrigerated dryers so they will not continue to run
- Turn off desiccant dryers. Those controlled by timers can easily purge air and cause the compressors to run for no benefit.
- If pressure must be maintained at night for certain equipment, isolate this equipment and serve with small compressors and on-off control if possible.
- Reduce pressure if possible. Leaks are reduced proportionally with pressure.

Compressor	A	B	C	D
Max air flow CFM	300	800	800	500
Bhp	67	178	178	111
kW at full load	53	144	144	90
cfm/Bhp	4.5	4.5	4.5	4.5
Full load cfm/kW	5.7	5.5	5.5	5.5
full load kW/cfm	0.176	0.180	0.180	0.180

Existing

Compressor	A	B	C	D	Overall
pct flow	90%	30%	0%	30%	
SCFM	270	240	0	150	660
Control mode	VSD	Mod.	Off	Mod.	
% kW	0.9	0.78	0	0.78	
kW	47.6	112.4	0	70.3	230
Operating cfm/kW	5.7	2.1	0	2.1	**2.9**
kWh/hr					**230**

Proposal A (Change from modulation to load/unload)

Compressor	A	B	C	D	Overall
pct flow	**90%**	**30%**	**0%**	**30%**	
SCFM	270	240	0	150	660
Control mode	**VSD**	**Load/Unl**	**Off**	**Load/Unl**	
% kW	0.9	0.47	0	0.47	
kW	47.6	67.8	0	42.3	158
Operating cfm/kW	5.7	3.5	0	3.5	**4.2**
kWh/hr					**158**
percent energy reduction, per hour				savings	32%

Proposal B (Base load at 100%, VSD trim)

Compressor	A	B	C	D	Overall
pct flow	**53%**	0%	**0%**	100%	
SCFM	160	0	0	500	660
Control mode	**VSD**	**Off**	Off	**Mod**	
% kW	0.54	0	0	1.00	
kW	28.6	0	0	90	119
Operating cfm/kW	5.6	0	0	5.5	**5.6**
kWh/hr					**119**
percent energy reduction, per hour				savings	48%

Example 2: Compressor Sequencing Scenario

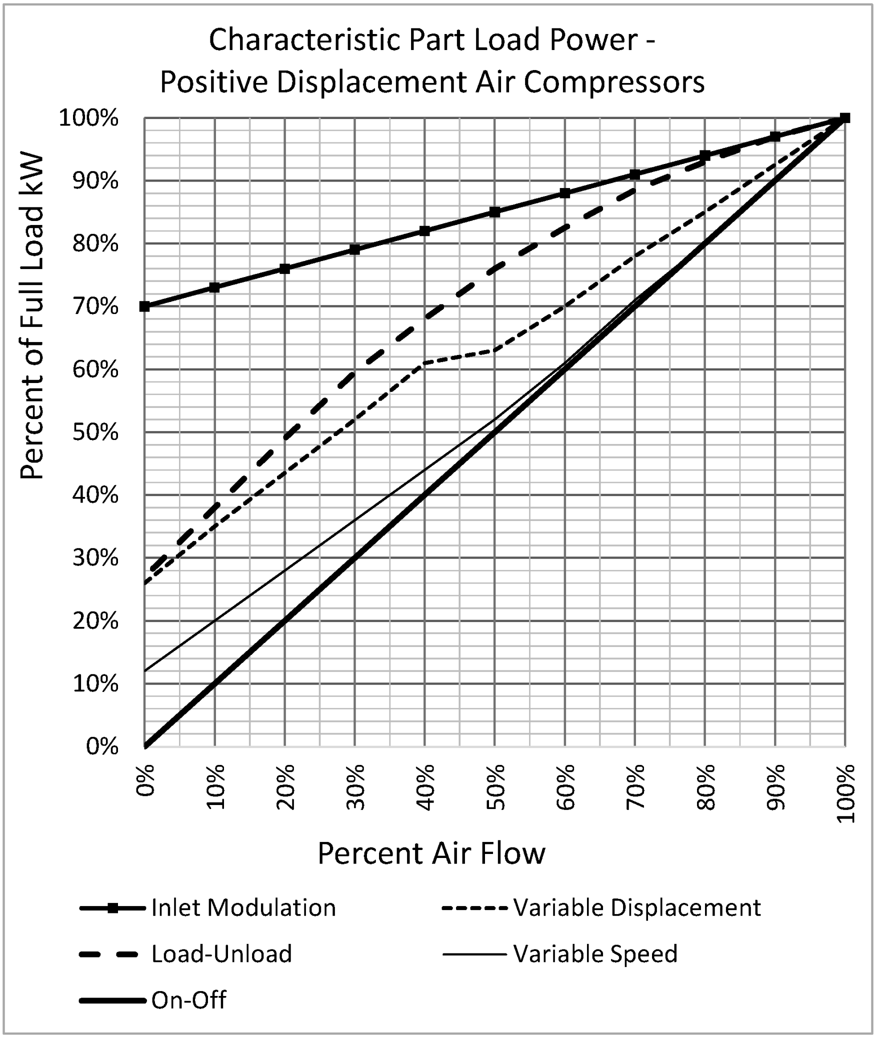


Figure 14-2—Part Load Power Characteristics, Positive Displacement Air Compressors

Source: *Improving Compressed Air System Performance*, DOE, 2003. Values estimated from charts.

Notes

- Variable displacement via slide valve, spiral valve, turn valve, or poppet valve
- Modulation assumes 3 gal/cfm storage
- Inlet modulation is without blow down

Table 14-4. Part Load Power Characteristics, Positive Displacement Air Compressors

Values for screw compressor used in Figure 14-2

Flow	On-Off / Ideal	Variable Speed	Load-Unload	Modu-lation	Variable Displ.
100%	100%	100%	100%	100%	100%
90%	90.0%	90.0%	97.0%	97.0%	92.5%
80%	80.0%	80.0%	93.0%	94.0%	85.0%
70%	70.0%	71.0%	88.5%	91.0%	78.0%
60%	60.0%	61.0%	82.5%	88.0%	70.0%
50%	50.0%	52.0%	76.0%	85.0%	63.0%
40%	40.0%	44.0%	68.0%	82.0%	61.0%
30%	30.0%	36.0%	59.5%	79.0%	52.0%
20%	20.0%	28.0%	49.0%	76.0%	43.5%
10%	10.0%	20.0%	38.0%	73.0%	35.0%
0%	0.0%	12.0%	27.0%	70.0%	26.0%

- Allow different operating strategies where different shifts have notably different load profiles. This can include using smaller compressors for lightly loaded shifts, reduced pressure, or both.
- Use on-off control where ample storage exists and when motors are amenable to cycling with reasonable off times.
- Choose load-unload in lieu of inlet throttling where only those options exist. This change may require additional storage.
- Equip compressed air plant with at least one compressor that is variable speed or other type with high part load efficiency.
- Sequence compressors for lowest overall kW/cfm power requirement. Equipment with good full load efficiency but poor part load efficiency can be viable for base loading because with percent load kept high, the part load losses never occur or are greatly minimized. Use one compressor for the remaining trim load.

COMPRESSED AIR LEAKS

Source of Data: "Compressed Air Tip Sheet #3", Industrial Technologies Program (ITP), 2004, US DOE Office of Energy Efficiency and Renewable Energy.

For well-rounded orifices, values should be multiplied by 0.97.
For sharp orifices, values should be multiplied by 0.61.

CFM of Leakage

Line Air Pressure (psig)	1/64" hole	1/32" hole	1/16" hole	1/8" hole	1 /4" hole	3/8" hole
70	0.29	1.2	4.7	19	74	168
80	0.32	1.3	5.2	21	83	187
90	0.36	1.5	5.7	23	92	207
100	0.40	1.6	6.3	25	101	227
125	0.48	1.9	7.7	31	122	276

Measuring Leaks

1. An assessment of leaks, in percent of flow, can be made using a pump up test or bleed down test (described below).
2. Overall leak flows can be identified with a permanently installed flow meter at the main piping point that represents 'all air to plant'. Like the pump up test, it requires the compressors to be running at normal pressure and the plant end uses turned off. Advantages of this method:
   - Leak value is given in SCFM directly, instead of percentage
   - Readily available for a reading without time requirements for a pump up test and calculation
   - Provides feedback to the user; i.e. a value for where we started, and now we can see the measured improvement.
3. Individual leak flow values can be identified using portable ultrasonic test equipment.

Pump Up Test Procedure

This method identifies the approximate percentage of leaks. Turn off all end loads. The test cannot be done while compressed air end uses are operating. With the end loads off, any air usage is attributed to leaks.

Run the compressor to normal operating pressure, and operate in load-unload mode (or on/off mode) automatically, so there is a repeatable cut-in and cut-out pressure. Using a stop watch, record the time for several cycles. For example, record times for three load-unload cycles and add the total seconds together as "time loaded" and "time unloaded."

$$\textbf{Percent leaks} = {}^{T1}/_{(T1+T2)} * 100\% \qquad \textbf{(eq. 10)}$$

Where:

T1 = Time loaded (seconds)

T2 = Time unloaded (seconds)

The resulting percent figure is really a gage of the piping leaks at that pressure, and is not suitable for calculating leaks as a percent of the load profile of SCFM. To convert the pump-up test percent leaks into SCFM leaks, identify the SCFM output of the compressor used during the test. For on-off and load-unload control, the 'load' cycle represents full output of the compressor; assuming the compressor is functioning properly, this can be found from manufacturer's data for the operational pressure it is working against.

For example, if the compressor used for the test was 100 Hp with a full load output capacity of 430 SCFM at 100 psig, then a test result of 22% would indicate 0.22*430 = 95 SCFM of leaks at 100 psi. Operating at 100 psi in the main piping, regardless of flow and regardless of which compressor was operating, the leakage would then be expected to be 95 SCFM.

Bleed Down Test Procedure

This method identifies the approximate SCFM of leaks. Turn off all end loads. The test cannot be done while compressed air end uses are operating. With the end loads off, any air usage is attributed to leaks. This method requires knowing the volume of the total storage which includes storage vessels and piping. Use a good quality well calibrated pressure gage.

Run the compressor to normal operating pressure. Turn off the compressor and close the discharge valve(s) so that the only way 'out' for the air is leaks. Record the time it takes for the pressure to drop to about half of the normal operating pressure.

$$\textbf{Leakage SCFM free air} = V^* \left[{}^{(P1-P2)}/_{(T^*14.7)}\right] {}^*1.25 \qquad \textbf{(eq. 11)}$$

Where:

P1 = normal operating pressure, psig
P2 = final pressure, which is P1-10psig
V = total system volume in cubic feet
T = time, minutes for pressure to drop from P1 to P2

Source: Improving Compressed Air System Performance, a Sourcebook for Industry, DOE, 2003

Notes:

- The 1.25 multiplier corrects leakage to normal system pressure, allowing for reduced leakage with falling system pressure.
- (p1-p2)/14.7 represents the portion of the initial charge of air

that leaks out, as if a level calculation (initial height–ending height)/total height; this establishes the cubic feet of the air that have left the piping.

- this calculation is not exact, but is a fair approximation. if the test procedure is done in a repeatable way, it can indicate changes in leaks from prior tests.

COMPRESSED AIR DRIERS

Dew Point requirements vary by compressed air user. Dew point temperature 35F and above can be served with a refrigerated drier. Dew point temperature below 35F requires a desiccant drier.

Refrigerated Driers use a sealed refrigeration system and a heat exchanger to cool the air indirectly, to a temperature below the required dew point; moisture is collected and discharged through a trap. Non-cycling driers operate continuously and maintain a fixed temperature; Cycling driers use thermal storage mass to allow infrequent start and stop of the compressors. Refrigerated driers use electricity for the compressor, but do not expel air other than small amounts lost from the drain trap if properly controlled by an automatic valve.

Desiccant Driers operate on chemical adsorption, 'holding' moisture using desiccant beads in a pressurized cylinder. Two cylinders are used and alternate, so one is active while the other is regenerating. Regeneration is done with air in different ways. Most systems use system compressed/dried air as a backwash to regenerate the desiccant and represent a significant compressed air burden on the system. Various control options reduce the purge air waste and its expense. **Figure 14-3 and Table 14-5** (same data)

<u>Heatless</u>. This is the standard desiccant drier type, and will use about **15%** of the maximum system output; more for very low dew point temperatures (e.g. -100F). Key is that the purge air does not reduce with demand. Conventional control is by timer only and so, when system total air output is low, the actual percentage of purge air increases. With the timer active and zero air consumption in the facility, the drier purge air can be the entire air demand for the compressor.

<u>Heated purge air.</u>
Basis of savings: Reduced system air loss.
Heating the purge air with an in-line electric heater reduces the purge air requirement by about half, to **7.5%** system air loss. This is because the purge air has a greater affinity for moisture when heated. The electric heating negates **+/- 40%** of the purge air electric savings (assumes 20% standby loss).
Source of numbers

<u>Blower purge air.</u>
Basis of Savings: Reduced system air use.
An electric blower uses room air which is much less costly to produce (lower pressure) than system compressed air. A heater is used to create the drying impetus. Heating burden is similar to heated purge approach. No system air is lost with this approach to purge, although some system air is used to cool the heated desiccant after it has been regenerated. System air consumption on average is on the order of **2%**.

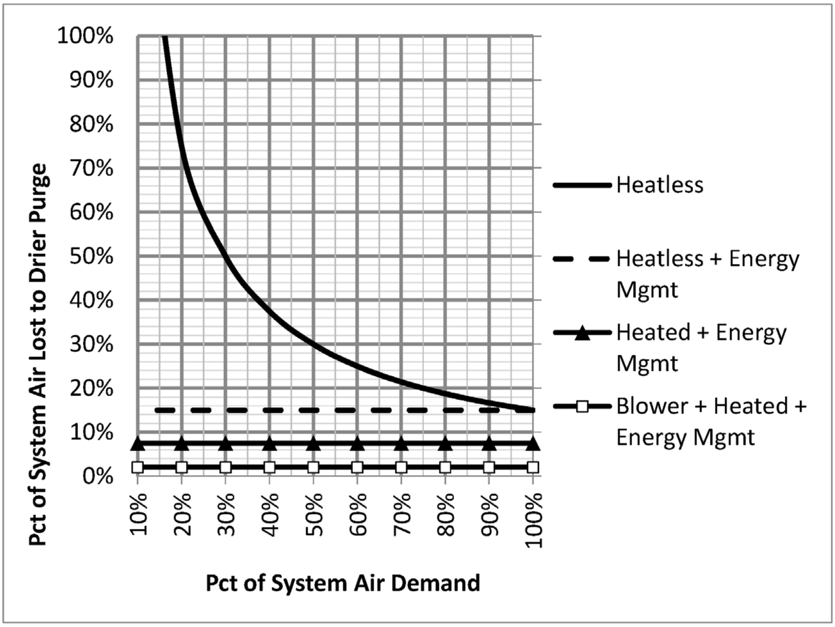


Figure 14-3. Purge Air Loss at Part Load—Desiccant Driers and Controls

Table 14-5. Data for Figure 14-3

Shows relative purge loses. Ancillary equipment (heaters, blowers) not shown

Source of data: 15% purge air for heatless desiccant dryer, 7.5% purge air for heated dryer, 2% CA air loss for purge air from blower: manufacturer's data

	SCFM 100	15%	Drier purge			7.5%	Drier purge	2.0%	Drier purge
	Compr. max capacity	**(Timer only)**	**(Timer only)**	**(Energy mgmt)**	**(Energy mgmt)**	**(Energy mgmt and heated)**	**(Energy mgmt and heated)**	**(Energy mgmt, blower + heated)**	**(Energy mgmt, blower + heated)**
System demand	Compr. delivery	Constant purge rate	**Drier purge**	Purge only when needed	**Drier purge**	Purge only when needed	**Drier purge**	Purge only when needed	**Drier purge**
percent	SCFM	SCFM	% of system	SCFM	% of system	SCFM	% of system	SCFM	% of system
100%	100	15	15%	15	15%	7.5	7.5%	2.0	2.0%
90%	90	15	17%	14	15%	6.8	7.5%	1.8	2.0%
80%	80	15	19%	12	15%	6.0	7.5%	1.6	2.0%
70%	70	15	21%	11	15%	5.3	7.5%	1.4	2.0%
60%	60	15	25%	9.0	15%	4.5	7.5%	1.2	2.0%
50%	50	15	30%	8.0	15%	3.8	7.5%	1.0	2.0%
40%	40	15	38%	6.0	15%	3.0	7.5%	0.8	2.0%
30%	30	15	50%	5.0	15%	2.3	7.5%	0.6	2.0%
20%	20	15	75%	3.0	15%	1.5	7.5%	0.4	2.0%
10%	10	15	150%	2.0	15%	0.8	7.5%	0.2	2.0%

<u>Drier Controls</u>
Basis of savings: only purge when needed, so purge air is a percent of demand instead of a percent of compressor max output. Control options:

- Active dew point measurement that inhibits purge air loss until needed. The dryer may still alternate between towers, but the actual release of air is only allowed when dew point is above the setting.
- Interlock the dryer purge to compressor operation. With this control, the drier is not allowed to purge when the compressor is stopped.

PRESSURE DROP FROM FRICTION IN PIPING

Pressure loss from friction is a function of velocity, density, and pipe roughness. Unlike liquids, gases are compressible and for this reason the pressure drop in a given length of pipe for a given mass flow depends upon the pressure. Adjustment for density at different pressure and temperature:

$$\boldsymbol{\rho 2 = \rho 1 * (P2/P1) * (T1/T2)} \qquad \textbf{(eq. 12)}$$

Where:
$\rho 1$ = Initial density, lb./ft3
$\rho 2$ = New density, lb./ft3
P1 = Initial pressure, psia
P2 = New pressure, psia
T1 = Initial pressure, R
T2 = New pressure, R

<u>Note 1:</u> Since air is compressible, each increment of pressure drop causes the air to expand further and increase velocity further, causing additional pressure drop, creating an amplifying effect. The accurate answer to calculating pressure drop involves a gradient of increasing velocity and pressure loss. Good practice of short pipelines and low velocities minimizes the runaway effect.

<u>Note 2</u>: Pressure drop increases with the square of flow at a given pressure. However, when comparing the same flow at *different* pressures, the pressure drop varies directly with the velocity (not

squared). An explanation for this is that friction is molecules scrubbing against the walls of the pipe, and although expanded and going faster, the number of molecules scrubbing against the pipe wall is the same.

In other words: For equal density, doubling flow rates produce 4x pressure drop increase. But in the case of comparing two equal mass flows at different pressures, the density goes down as the velocity goes up and the result is a direct increase instead of a squared increase. This points out a limitation in using pressure drop charts for compressible fluids, and also that using SCFM consistently has the benefit of consistent density: for a doubling of cfm for same density, the pressure drop squares; but *for a doubling of cfm because of pressure reduction, the pressure drop only doubles*.

- **Pressure Drop for Different Flow (same pressure, same pipe size)**

  If ΔP is known for one value of flow (Q) in a given pipe size, a new value $\Delta P2$ can be calculated for a new value of flow Q2:

  $$\Delta P2 = \Delta P1 * (Q2/Q1)^2 \quad Q \text{ in SCFM}, \Delta P \text{ in psi} \qquad \text{(eq. 13)}$$

  Example: At 500 SCFM, the pressure drop through a given section of pipe is known to be 2.2 psi. At 600 SCFM, the pressure drop will be $(600/500)^2 * 2.2 =$ 3.17 psi.

- **Pressure Drop for Different Pressure (same mass flow, same pipe size)**

  If ΔP is known for one value of flow (Q) in a given pipe size, a new value $\Delta P2$ can be calculated for the same flow in the same pipe at a different pressure:

  $$\Delta P2 = \Delta P1 * (P1/P2) \quad P1, P2 \text{ in psia}, \Delta P \text{ in psi} \qquad \text{(eq. 14)}$$

  Example: (ref Table 14-6) At 300 SCFM and 100 psig (114.7 psia), the pressure drop in a 1.5 inch steel pipe will be 25.4 psi per 1000 ft.

  At 125 psig (139.7 psia), the pressure drop will be 25.4*(114.7/139.7) = 20.9 psi per 1000 ft.

  At 80 psig (94.7 psia), the pressure drop will be 25.4*(114.7/94.7) = 30.8 psi per 1000 ft.

  Thus (**Table 14-6**) can serve as multiple charts for different pressures by using the pressure ratio.

- **Pressure Drop For Different Pipe Size (same flow, same pressure)**
  This method neglects some minor variations of friction factor. It is not exact but is a very good approximation.
  If ΔP is known for one value of flow (Q) in a given pipe size, a new value ΔP2 can be calculated for the same flow in the same pipe at a different pressure:
  $\Delta P2 = \Delta P1^{*} (D1/D2)^{5}$ D1, D2 in inches diameter, ΔP in psi
  Example: (ref **Table 14-6**) At 300 SCFM and 100 psig (114.7 psia), the pressure drop in a 1.5 inch steel pipe (1.610 in. inside diameter) will be 25.4 psi per 1000 ft.
    This same flow in a 2-inch pipe (2.067 in. inside diameter) will be $25.4^{*}(1.610/2.067)^{5} = 7.28$ psi per 1000 ft. For comparison, the chart says 6.72 psi per 1000 ft. (~8% error).

- Note on Accuracy of Pressure Drop Estimates
  The Darcy-Weisbach estimating method is a popular method of estimating pressure drop for fluids and is used in this text. There are others, such as Hazen-Williams. Neither are perfect. For example, the Darcy-Weisbach method requires a friction factor "f" which is a complex combination of multiple factors including equivalent length of piping in "diameters" and the roughness of the pipe walls. The equivalent length of the overall pipe system depends on the equivalent length of fittings and valves, which add to the actual footage of piping, and values for fittings are estimates. Roughness of the pipe varies considerably based on service conditions, corrosion, and deposits within the pipe, and also the pipe material itself: note that most pressure drop tables presume new, clean steel pipe. Piping pressure drop estimating methods are very useful, but the values are not absolute. Consider the values a very good approximation. Engineering judgment is needed to assure the methods are applied within their limits, and whether additional allowances are warranted for unknowns, fouling, etc.

100 psig Table 14-6 (100) CA Piping Pressure Drop (psi per 1,000 ft.)

Schedule 40 steel pipe with minor allowance for fouling, Ɛ=0.00025 ft. (0.0005 for 1/2 inch pipe)

Figures at max flow are calculated with Darcy-Weisbach equation; reduced flows are from $(CFM2/CFM1)^2$

Convert to different mass flow, same density: $\Delta P2 = \Delta P1 * (CFM2/CFM1)^2$

Convert to different pressure and density, same mass flow: $\Delta P2 = \Delta P1 * (P1/P2)$, P1 and P2 in absolute

Convert to different pipe size, same mass flow, same pressure: $\Delta P2 = \Delta P1 * (D1/D2)^5$ (approx)

100 psi	Diam.	0.622	0.824	1.049	1.360	1.610	2.067	2.469	3.068	4.026	6.065	7.981	10.02
Free Air SCFM	Compressed CFM	**0.5"** Nom	**0.75"**	**1"**	**1.25"**	**1.5"**	**2"**	**2.5"**	**3"**	**4"**	**6"**	**8"**	**10"**
2	0.256	0.205	0.0399										
5	0.641	1.28	0.250	0.0685									
10	1.28	5.12	1.00	0.274	0.0689								
15	1.92	11.5	2.25	0.617	0.155	0.0634							
20	2.56	20.5	3.99	1.10	0.275	0.113							
30	3.84	46.1	8.98	2.47	0.620	0.254	0.0672						
40	5.13	81.9	16.0	4.39	1.10	0.451	0.119						
50	6.41	**128**	25.0	6.85	1.72	0.705	0.187	0.0757					
60	7.69		35.9	9.87	2.48	1.02	0.269	0.109					
70	8.97		48.9	13.4	3.37	1.38	0.366	0.148					
80	10.3		63.9	17.5	4.41	1.80	0.478	0.194	0.0594				
90	11.5		80.8	22.2	5.58	2.28	0.605	0.245	0.0751				
100	12.8		**99.8**	27.4	6.89	2.82	0.747	0.303	0.0927				
150	19.2			61.7	15.5	6.34	1.68	0.681	0.209	0.0499			
200	25.6			**109.7**	27.5	11.3	2.99	1.21	0.371	0.0887			
250	32.0				43.0	17.6	4.67	1.89	0.580	0.139			

100 psi	Diam.	0.622	0.824	1.049	1.360	1.610	2.067	2.469	3.068	4.026	6.065	7.981	10.02
Free Air SCFM	Compressed CFM	**0.5"** Nom	**0.75"**	**1"**	**1.25"**	**1.5"**	**2"**	**2.5"**	**3"**	**4"**	**6"**	**8"**	**10"**
300	38.4				62.0	25.4	6.72	2.72	0.835	0.200			
350	44.9				84.4	34.5	9.15	3.71	1.14	0.272			
400	51.3				**110**	45.1	11.9	4.84	1.48	0.355			
450	57.7					57.1	15.1	6.13	1.88	0.449			
500	64.1					70.5	18.7	7.57	2.32	0.554	0.0643		
600	76.9					**102**	26.9	10.9	3.34	0.798	0.0927		
700	89.7						36.6	14.8	4.54	1.09	0.126		
800	103						47.8	19.4	5.94	1.42	0.165		
900	115						60.5	24.5	7.51	1.80	0.208	0.0658	
1,000	128						74.7	30.3	9.27	2.22	0.257	0.103	
1,250	160						**117**	47.3	14.49	3.46	0.402	0.148	
1,500	192							68.1	20.9	4.99	0.579	0.148	
2,000	256							**121**	37.1	8.87	1.03	0.263	0.078
2,500	320								58.0	13.9	1.61	0.411	0.122
3,000	384								83.5	20.0	2.32	0.593	0.176
3,500	449								**114**	27.2	3.15	0.807	0.239
4,000	513									35.5	4.12	1.05	0.31
5,000	641									55.4	6.43	1.65	0.49
6,000	769									79.8	9.27	2.37	0.70
7,000	897									**109**	12.61	3.23	0.96
8,000	1,025										16.5	4.21	1.25
9,000	1,153										20.8	5.33	1.58
10,000	1,282										25.7	6.58	1.95
15,000	1,922										57.9	14.81	4.40
20,000	2,563										**103.0**	**26.34**	**7.81**

125 psig Table 14-6 (125) CA Piping Pressure Drop (psi per 1,000 ft.)
Schedule 40 steel pipe with minor allowance for fouling, Ɛ=0.00025 ft. (0.0005 for 1/2 inch pipe)
Figures at max flow are calculated with Darcy-Weisbach equation; reduced flows are from $(CFM2/CFM1)^2$
Convert to different mass flow, same density: $\Delta P2 = \Delta P1*(CFM2/CFM1)^2$
Convert to different pressure and density, same mass flow: $\Delta P2 = \Delta P1*(P1/P2)$, P1 and P2 in absolute
Convert to different pipe size, same mass flow, same pressure: $\Delta P2 = \Delta P1*(D1/D2)^5$ (approx)

125 psi	Diam.	0.622	0.824	1.049	1.360	1.610	2.067	2.469	3.068	4.026	6.065	7.981	10.02
Free Air SCFM	Compressed CFM	**0.5"** Nom	**0.75"**	**1"**	**1.25"**	**1.5"**	**2"**	**2.5"**	**3"**	**4"**	**6"**	**8"**	**10"**
2	0.210	0.171	0.0335										
5	0.526	1.07	0.209	0.0573									
10	1.05	4.26	0.836	0.229	0.0574								
15	1.58	9.59	1.88	0.515	0.129	0.0528							
20	2.10	17.1	3.35	0.92	0.230	0.094							
30	3.16	38.4	7.53	2.06	0.517	0.211	0.0558						
40	4.21	68.2	13.4	3.67	0.919	0.376	0.099						
50	5.26	**107**	20.9	5.73	1.44	0.587	0.155	0.0627					
60	6.31		30.1	8.25	2.07	0.85	0.223	0.090					
70	7.37		41.0	11.2	2.81	1.15	0.304	0.123					
80	8.4		53.5	14.7	3.67	1.50	0.397	0.161	0.0492				
90	9.5		67.8	18.6	4.65	1.90	0.503	0.203	0.0623				
100	10.5		**83.6**	22.9	5.74	2.35	0.620	0.251	0.0769				
150	15.8			51.5	12.9	5.28	1.40	0.565	0.173	0.0413			
200	21.0			**91.6**	23.0	9.39	2.48	1.00	0.307	0.0734			
250	26.3				35.9	14.7	3.88	1.57	0.480	0.115			

125 psi	Diam.	0.622	0.824	1.049	1.360	1.610	2.067	2.469	3.068	4.026	6.065	7.981	10.02
Free Air SCFM	Compressed CFM	**0.5"** Nom	**0.75"**	**1"**	**1.25"**	**1.5"**	**2"**	**2.5"**	**3"**	**4"**	**6"**	**8"**	**10"**
300	31.6				51.7	21.1	5.58	2.26	0.692	0.165			
350	36.8				70.3	28.8	7.60	3.07	0.94	0.225			
400	42.1				**91.9**	37.6	9.93	4.01	1.23	0.294			
450	47.4					47.6	12.6	5.08	1.56	0.372			
500	52.6					58.7	15.5	6.27	1.92	0.459	0.0532		
600	63.1					**84.5**	22.3	9.03	2.77	0.661	0.0766		
700	73.7						30.4	12.3	3.77	0.899	0.104		
800	84						39.7	16.1	4.92	1.17	0.136		
900	95						50.3	20.3	6.23	1.49	0.172		
1,000	105						62.0	25.1	7.69	1.83	0.213	0.0545	
1,250	132						**97**	39.2	12.01	2.87	0.332	0.0852	
1,500	158							56.5	17.3	4.13	0.479	0.123	
2,000	210							**100**	30.7	7.34	0.851	0.218	0.0650
2,500	263								48.0	11.5	1.33	0.341	0.102
3,000	316								69.2	16.5	1.91	0.490	0.146
3,500	368								**94.2**	22.5	2.61	0.668	0.199
4,000	421									29.4	3.40	0.872	0.260
5,000	526									45.9	5.32	1.36	0.406
6,000	631									66.1	7.66	1.96	0.585
7,000	737									**89.9**	10.4	2.67	0.797
8,000	842										13.6	3.49	1.04
9,000	947										17.2	4.41	1.32
10,000	1,052										21.3	5.45	1.63
15,000	1,578										47.9	12.3	3.66
20,000	2,105										**85.1**	**21.8**	**6.50**

<u>Example #1:</u>

Evaluate inlet piping for restriction and compressor efficiency. Compressor rating is 1200 SCFM and has a nominal 4 inch diameter inlet pipe. For rated compressor efficiency, the manufacturer's limit for inlet piping pressure drop is 0.3 psi or less, not including the air filter. The equivalent length of the inlet piping is 100 ft. Operating pressure for the compressor is 125 psig. Motor efficiency is 94%.

Piping pressure drop is estimated using Darcy-Weisbach equation, which yields pressure drop in ducts, pipes, and tubes in units of feet of fluid. From **Table 14-6** (pressure drop in piping at 100 psig) 1200 SCFM free air in a 6 inch pipe is found by using the next lower or higher value and adjusting with affinity laws. For 1000 SCFM, the pressure drop in a 4 inch steel pipe is 2.22 psi per 1000 ft., so at 1200 SCFM the pressure drop will be 2.22*(1200/1000)^2 = 3.2 psi/1000 ft. Adjusting to sea level pressure by the factor P1/P2 the pressure drop per 1000 ft. will be 3.2*((100+14.7)/14.7)= 24.9 psi per 1000 ft. The comparative pressure loss for 100 equivalent feet of inlet pipe is then 2.49 psi vs. the manufacturer's limit of 0.3 psi, an increase of 2.19 psi. The compressor power equation **chap 14, equation 6** is used to determine the power impact of this condition. The property affecting compressor power is compression ratio P2/P1.

At design conditions	P2/P1 = (125+14.7)/(14.7-0.3) = 9.70
With restrictive inlet pipe	P2/P1 = (125+14.7)/(14.7-2.19) = 11.17

From **equation 6**, a mass flow rate of 5400 lb./hour, and the example efficiency factors of 66.175% compressor and 94% motor:

At design conditions	Power = 237 kW
With restrictive inlet pipe	Power = 254 kW
Power increase	<u>Power increase 17kW</u>, or 7% (ans)

The excess power and energy loss can be corrected with a larger intake pipe.

<u>Example #2</u>:

An existing air system is being modified for greater capacity for production increase. System pressure is 125 psig. The main distribution system for the remote compressor supply is 6 inch diameter and is 800 equivalent feet long. Data logger data shows 6,000 SCFM is a normal usage mid-shift, and the plan is to increase capacity by 25%. Pressure drop across the main distribution pipe now is measured at 10 psi. End use pressure must not be reduced, so whatever additional pressure drop exists must be compensated by higher supply pressure. Using 0.189 kWh/cfm-hour and 5,000 hours per year at 6000 SCFM, determine the annual cost penalty of using the existing piping, which will be considered against the cost of increasing the pipe size. This does not consider the cost of the added flow, just the efficiency reduction of the compressor air system. Cost of electricity is $0.12 per kWh.

A 25% increase in flow will increase existing pressure drop from 10 psi to (1.25^2)*10psi = 15.6 psi, an increase of 5.6 psig. From **figure 14-5—Savings from Reduced Compressed Air Pressure**, changes in power requirements for a 125 psi air system will be approximately 0.7 percent for each 2 psi change in pressure. Then, power increase will be approximately 0.7% * 5.6/2 = 1.96% increase. At full flow, power presently is 6000 SCFM * 0.189 kW/cfm = 1134 kW. With the increased pressure, the added kW for 6000 SCFM will be 1134 * 0.0196 = 22.3 kW. For 5000 hours per year, the added pressure drop from not increasing pipe size will cause an energy penalty of 22.3 kW * 5000 hr. = 111,500 kWh, and $13,380 per year.

STORAGE AND CAPACITANCE

Storage tanks store energy like a spring and can be seen as another compressor, able to release compressed air for a period of time to satisfy a demand. Storage levels out short term demands and helps compressor motors operate closer to an average electrical demand and avoid short term high power demands.

Supply-side

Storage is necessary for on-off control and load-unload control. The storage receiver, combined with the piping volume, serve as a flywheel for the air that is alternately compressed and then released, as with a spring. The larger the storage space, the longer time between cycles can be (on-off/load-unload), or the narrower the cut-in/cut-out pressure settings can be.

Rules of thumb for storage have been used for years, such as 2-4 gallons of storage per SCFM. With variable speed compressors, the storage requirements will be less. A very useful feature of storage is to have enough of it so that a drier purge event does not result in a compressor starting in knee-jerk fashion, only to shut off again in a couple of minutes.

Demand-side

Storage can be beneficial when there are large short-term demands at points of use. If demand cannot be supplied through the main piping fast enough (restrictive distribution), end use pressure will drop and the operational reaction is to elevate main system pressure. When this occurs, the demand-side storage can be the enabler for lowering system pressure and power.

Capacitance

Compressed air supply equipment is connected to the end use points by piping and storage. Each compressed air system has a characteristic behavior of how it reacts to a step change in end-use flow demand. It can be seen roughly as an electric circuit equivalent, with a battery, a load, and an internal resistance. An automobile analogy is when the car headlights dim while the engine is cranking: due to internal resistance, there is a voltage drop during high instantaneous demand, lowering the available voltage (for head lights, using the car example) temporarily. If the dim lights have a negative effect on the operation of the car, some provision must be made to compensate. In the case of compressed air, when intermittent low pressure causes operational, production, or quality problems, the operations staff will compensate—by raising the system pressure. When there is zero resistance in the distribution piping, the supply side (compressor and supply storage) can serve any end use instantaneously. However, there is some resistance in the distribution system. Presuming the supply side has the capacity for this short term demand, what is happening during a pressure dip at the point of use is a pressure drop in the distribution piping (akin to the battery internal resistance). One solution is to lower the resistance of the distribution piping. An alternate solution is to add a storage tank at the downstream end of the distribution piping, near the point of high demand, sized so that it can draw air locally without taxing the limited distribution piping. **Figure 14-4** illustrates the concept.

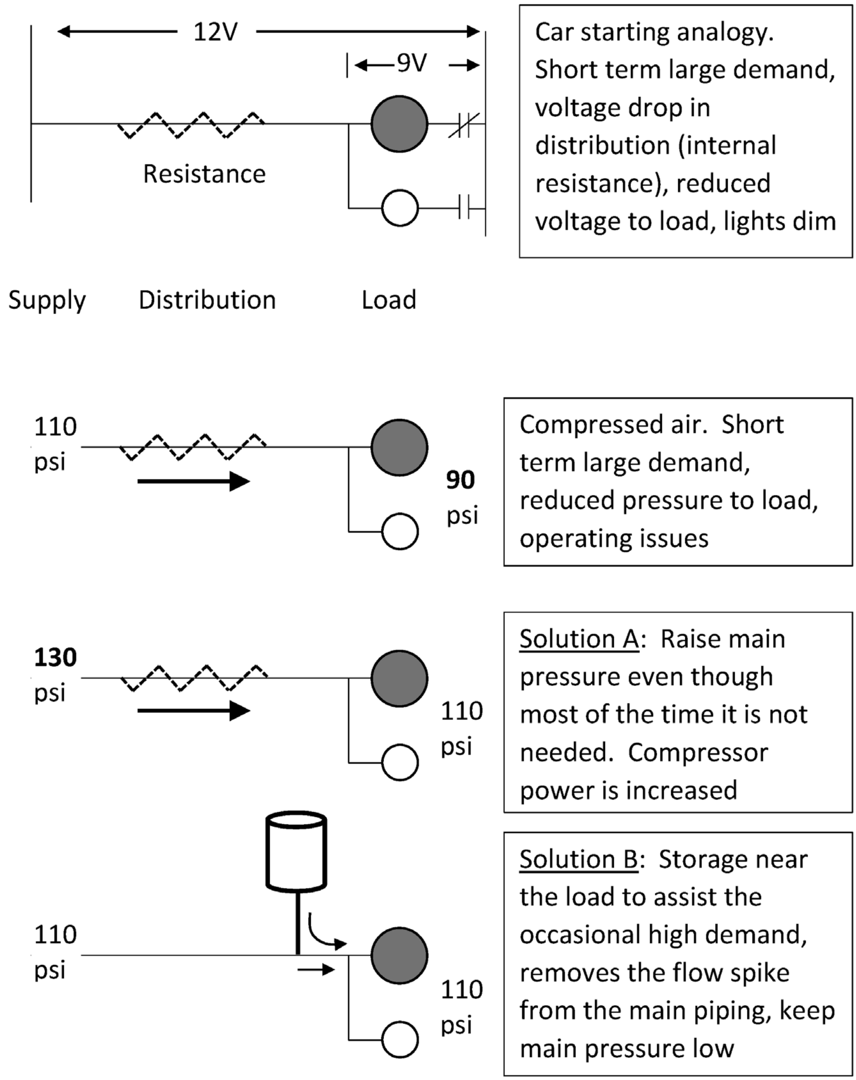


Figure 14-4. Demand-side Storage for Short-term Demands

RULES OF THUMB

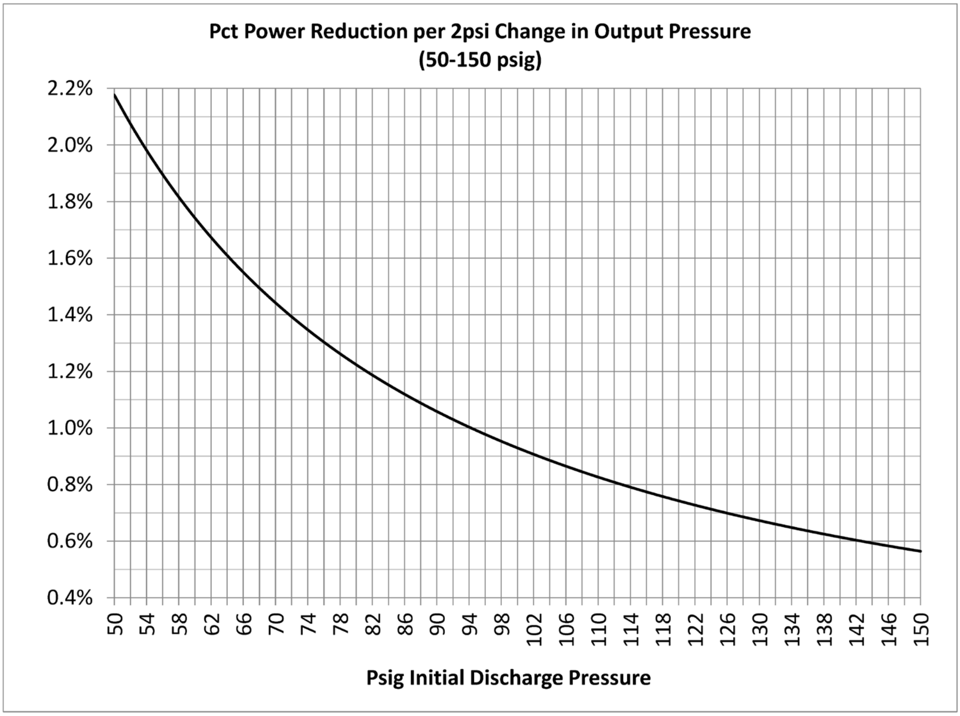


Figure 14-5. Savings from Reduced Compressed Air Pressure

Common rule of thumb of 1.0% power reduction per 2 psig reduction in discharge pressure is valid between 80psig-110psig. Discharge pressures outside this range have different values. The chart or rule of thumb are not suitable for very low pressures; these should be calculated directly with **chap 14, equation 6**.

1. Modeled with polytropic compressor equation for np=1.12, compressing air, **chap 14, equation 6**.
2. A machine efficiency factor was added and to the theoretical result and adjusted to exactly 4.5 SCFM per Hp at 100 psig discharge, at sea level input pressure, to approximate a 'real' single stage screw compressor. Values will be different for centrifugal and multi-stage compressors.
3. Taken from notes 1 and 2. Comparison for percentage change is power at given discharge pressure vs. power at the next higher 2 psig increment of discharge pressure.
4. Assumes sea level 14.7 psia inlet pressure.

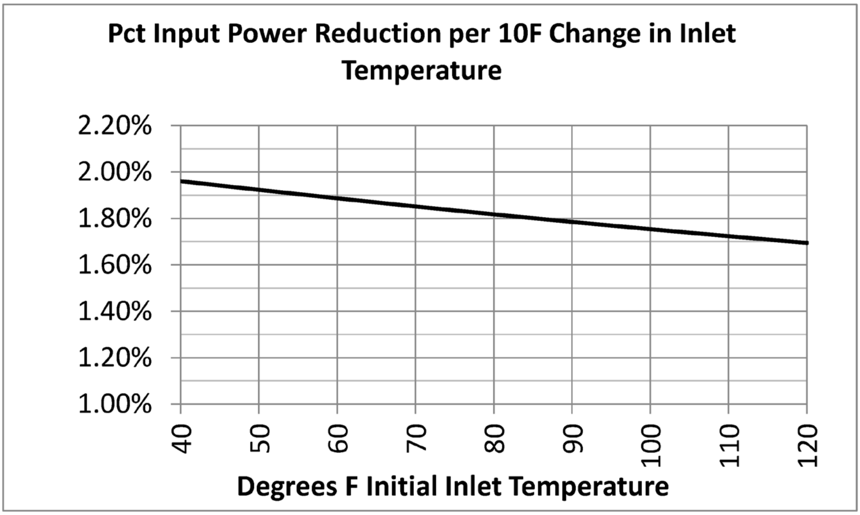


Figure 14-6. Savings from Cooler Inlet Air

Common rule of thumb of 1.9% power reduction per 10F reduction in inlet temperature is valid between 50F-70F. Inlet temperatures outside of this range have different values. Note: this should not be taken as incentive to mechanically cool the inlet air, which will increase energy consumption.

1. Modeled with polytropic compressor equation for np=1.12, compressing air, **chap 14, equation 6**.
2. A machine efficiency factor was added and to the theoretical result and adjusted to exactly 4.5 SCFM per Hp at 100 psig discharge, at sea level input pressure, to approximate a 'real' single stage screw compressor. Values will be different for centrifugal and multi-stage compressors.
3. Taken from notes 1 and 2. Comparison for percentage change is power at given temperature vs. power at the next higher 10F increment of inlet temperature.
4. Power percentage reduction values will be the same regardless of compressor operating pressure.

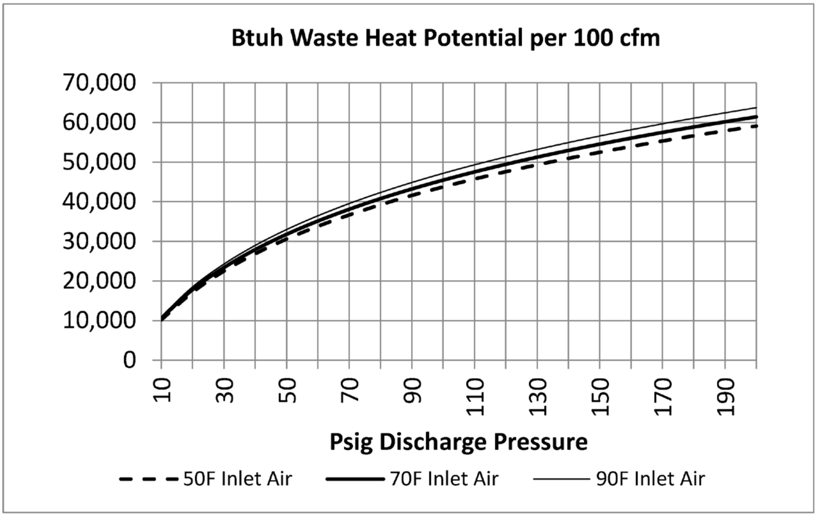


Figure 14-7. Recoverable Heat from Air Compressor

Note: Common rule of thumb of 50,000 Btuh per 100 SCFM is valid between 110 psig and 130 psig discharge pressure. Discharge pressures outside this range have different values.

1. Modeled with polytropic compressor equation for np=1.12, compressing air, chap 14, equation 6.
2. A machine efficiency factor was added and to the theoretical result and adjusted to exactly 4.5 SCFM per Hp at 100 psig discharge, at sea level input pressure, to approximate a 'real' single stage screw compressor. Values will be different for centrifugal and multi-stage compressors.
3. Taken as 80% of calculated compressor work in Btuh from notes 1 and 2, expressed as Btuh per 100 scfm.
4. Values vary slightly from inlet air temperature.

Heat recovery for an air compressor is from the appurtenances that reject heat such as oil coolers, intercoolers, and after-coolers. Heat rejection medium may be air or water and will be low grade (<200F), but represents on the order of 80% of the energy input to the compressor. In the case of packaged air-cooled screws, most of the waste heat is discharged by air-air heat exchanger from a common outlet point which is designed for conveniently ducting. When the compressor room is next to a location needing space heating, simple dampers can divert the warm air to a useful end in cold season, and to outdoors when the heat is not needed. See **Fig. 14-8**.

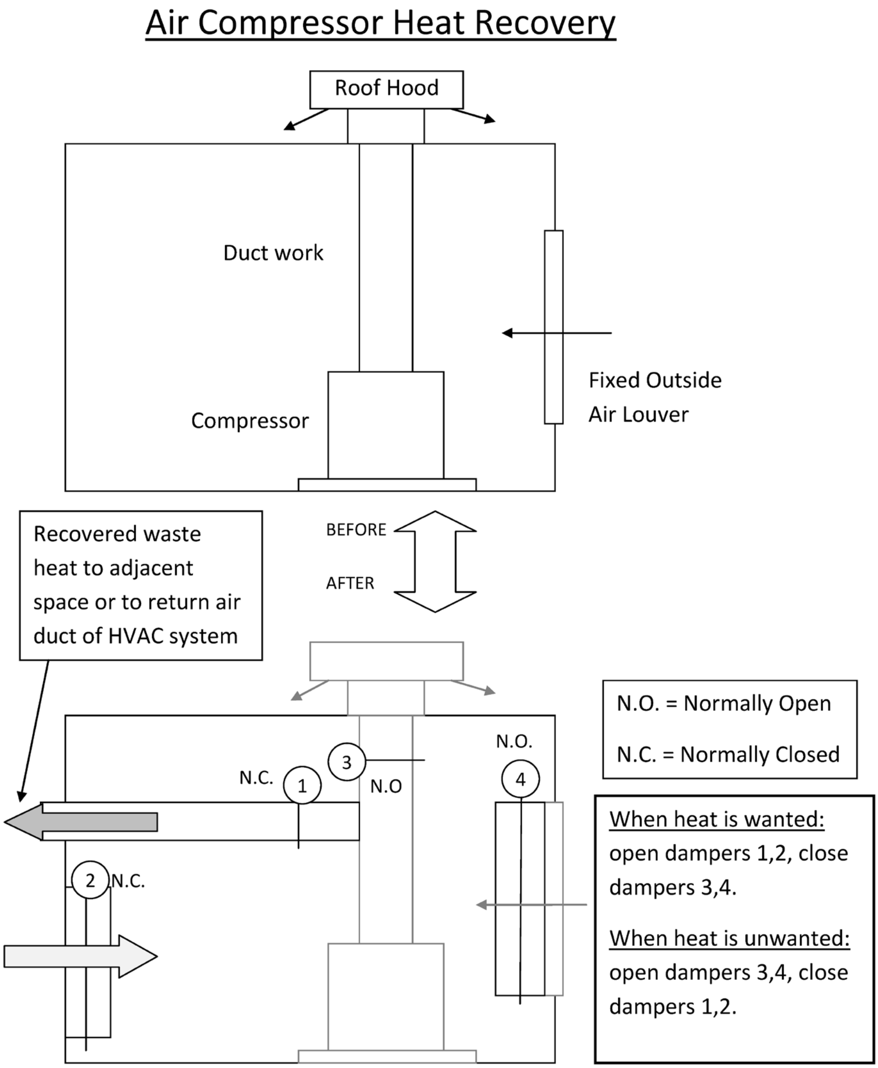


Figure 14-8. Air Compressor Waste Heat Recovery Diagram

Chapter 15

Fan and Pump Drives

FAN/PUMP CAPACITY MODULATION METHODS

Source: *Energy Management Handbook*, 7th Ed., Fairmont Press.

Table 15-1.

Method	Principle	Remarks
Discharge Damper	Energy dissipation through head loss resistance	Inexpensive. High losses relative to other options
Inlet Vane	Combination of inlet pre-rotation and resistance	Usually limited to 40% flow turndown
Eddy Current Coupling	Variable driven speed, constant motor speed	0-100% turndown
Magnetic Coupling	Variable driven speed, constant motor speed	0-100% turndown
Adjustable Speed Drive or Variable Frequency Drive	Variable motor speed	Motor cooling consideration below 20% motor speed. (a.k.a. ASD/VSD/VFD)
Variable Pitch Fan Blades (Axial)	Reduced rake of the impeller unloads the fan	Unloading curve very similar to ASD VSD/VFD
Modulating Inlet Fan Sleeve (Centrifugal Fans)	A sliding sleeve exposes all or part of the impeller, changing its effective wheel width to unload the fan	More efficient than inlet vanes and no energy penalty in wide-open position. Proprietary
Variable Pulley Systems	Modulates the speed of the driven device by moving the sides of an adjustable drive pulley in or out, changing the diameter of the pulley	This technology has generally been replaced by other methods. These were nicknamed “pulley pinchers”

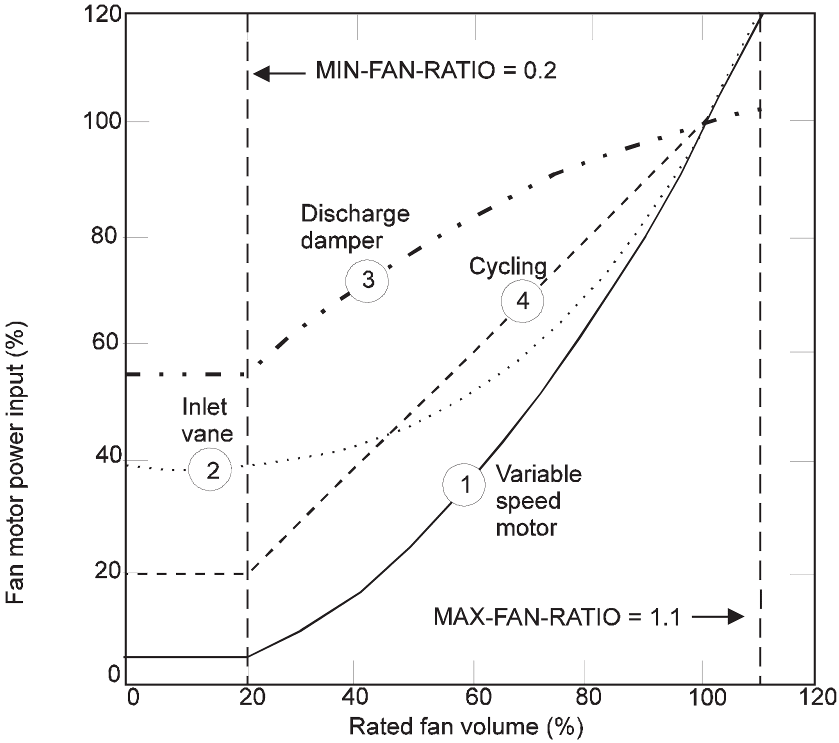


Figure 15-1A. Characteristic Fan Throttling Methods Pct Flow vs. Pct Power
Source: DOE 2.2 Building Energy Use and Cost Analysis Program Vol. 6, Hirsch, J. / LBNL / U.S. DOE

Note: The variable speed curve (1) shown approximates power reduction with flow squared instead of flow cubed, i.e. 50% flow = 0.52 = 25% power.

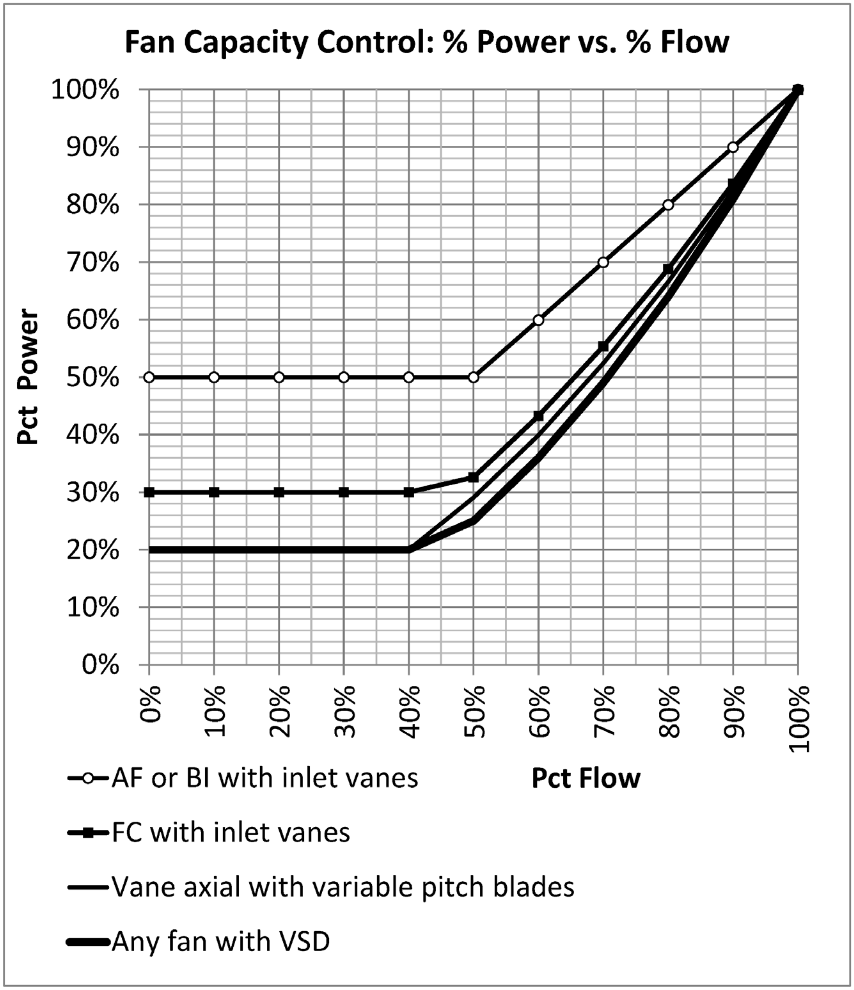


Figure 15-1B. Fan Capacity Control, VSD vs. Inlet Vanes and Variable Pitch Axial Blades

Source of data: Fan Systems, Commercial Buildings Energy Modeling Guidelines and Procedures (COMNET), New Buildings Institute, 2014

Note: Curves are for friction-only systems. Where static lift or maintained downstream pressure control is involved, power reduction will be less than shown.

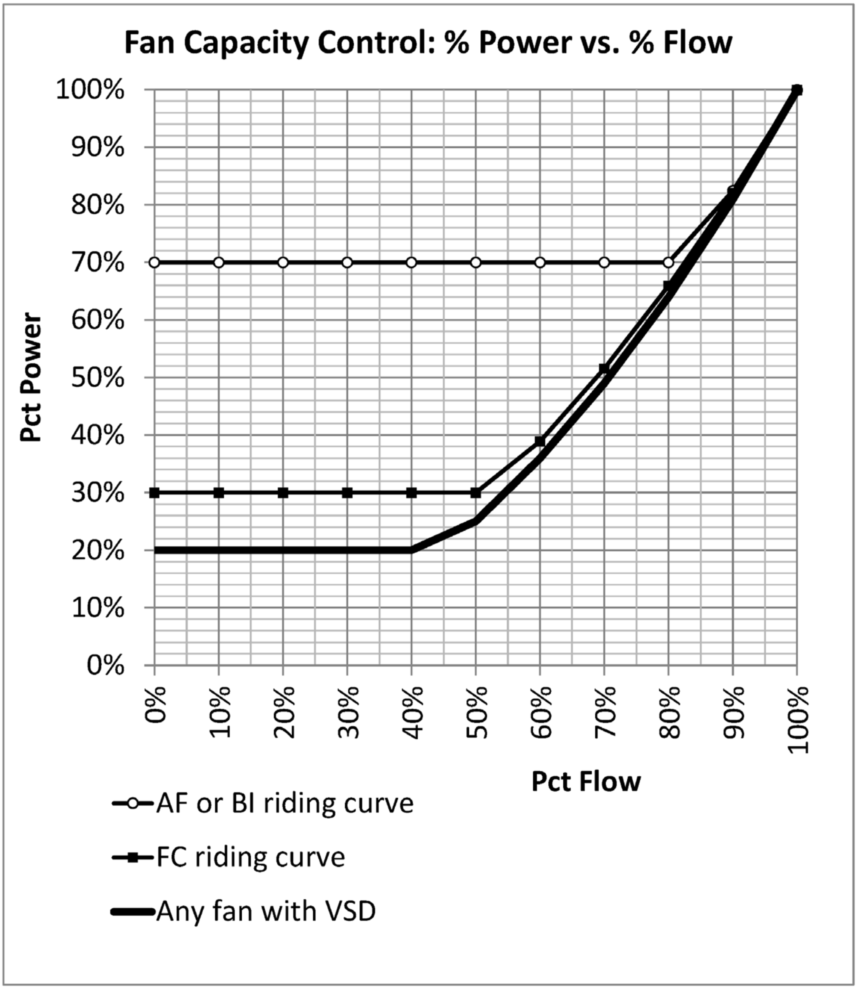


Figure 15-1C. Fan Capacity Control, VSD vs. Riding the Curve
Source of data: Fan Systems, Commercial Buildings Energy Modeling Guidelines and Procedures (COMNET), New Buildings Institute, 2014

Note: Curves are for friction-only systems. Where static lift or maintained downstream pressure control is involved, power reduction will be less than shown.

Greater of

[PLR=a+(b*FR)+(c*FR)+(c*FR)] and PLR=PowerMin

PLR	Ratio of fan power at part load conditions to full load fan power
PowerMin	Minimum fan power
FR	Fan Ratio Ratio of cfm at part-load to full-load cfm (Percent Air Flow)
a,b,c,d	Curve constants

Curve Constants	Min % power	a	b	c	d
AF or BI riding curve	70%	0.1631	1.5901	-0.8817	0.1281
AF or BI with inlet vanes	50%	0.9977	-0.659	0.9547	-0.2936
FC riding curve	30%	0.1224	0.612	0.5983	-0.3334
FC with inlet vanes	30%	0.3038	-0.7608	2.2729	-0.8169
Vane axial/ variable pitch	20%	0.1639	-0.4016	1.9909	-0.7541
Any fan with VSD	20%	0.0013	0.147	0.9506	-0.0998

AF=air foil
BI=backward inclined
FC=forward curved
VSD=variable speed drive

Charted Data

% Air Flow → % Power by Control type	0%	10%	20%	30%	40%	50%	60%	70%	80%	90%	100%
AF or BI riding curve	70%	70%	70%	70%	70%	70%	70%	70%	70%	82%	100%
AF or BI with inlet vanes	50%	50%	50%	50%	50%	50%	60%	70%	80%	90%	100%
FC riding curve	30%	30%	30%	30%	30%	30%	39%	52%	66%	82%	100%
FC with inlet vanes	30%	30%	30%	30%	30%	33%	43%	55%	69%	84%	100%
Vane axial/ variable pitch	20%	20%	20%	20%	20%	29%	40%	52%	67%	82%	100%
Any fan w/VSD	20%	20%	20%	20%	20%	25%	36%	49%	64%	81%	100%

Data used in Figures 15-1B and 15-1C (percent of full load power)
Source of data: Fan Systems, Commercial Buildings Energy Modeling Guidelines and Procedures (COMNET), New Buildings Institute, 2014

V-BELTS

The standard 'wrapped' V-belt is the mainstay of the belt drive industry. They are available as single or multiple band units. All v-belts have the ability to 'clutch' (slip) under start-up or shock loads which can be very useful in design. For example, if a machine were to jam, breaking the belts is much less costly than breaking the machine. V-belts are forgiving for minor mis-adjustments in alignment or tension. The tension undergoes an initial "break in" period, after which it should be re-tightened. All V-belts exhibit some slippage, which results in additional motor power input. This can be measured as heat build-up on belt drive sheaves, with higher amounts of slippage resulting in more heat. Sheaves eventually wear out and must be replaced to maintain drive adhesion. The belt design relies on edge friction to transmit power, thus the worn sheave that allows the belt to drop to the bottom will lose traction—increasing tension may temporarily restore friction, but will also add considerable strain to the bearings with increased wear. The same adhesion that allows the belts to slip resists "letting go" so efficiency losses include this release force as well.

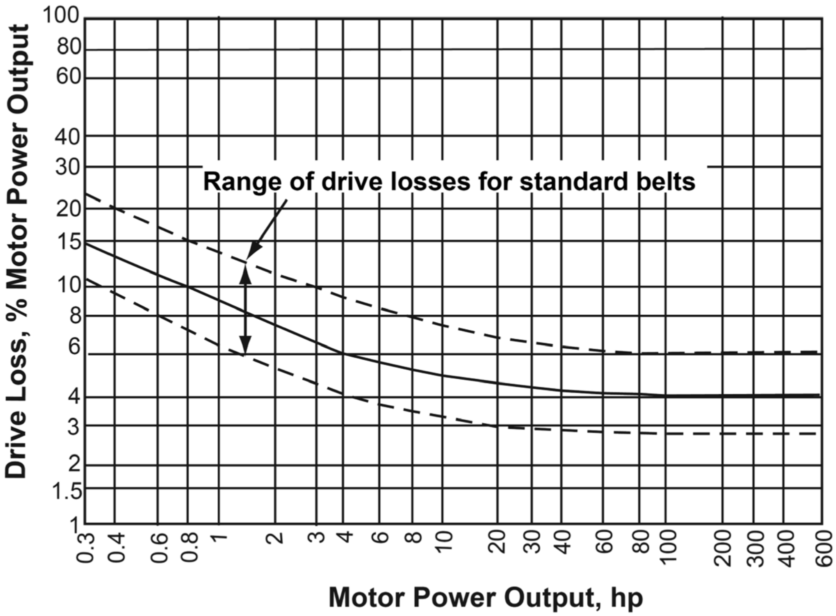


Figure 15-2. Range of Drive Losses for Standard V-Belts
Source: *Engineering Cookbook: A Handbook for the Mechanical Designer*. Loren Cook Co.

Cogged Belts

- Compared to a standard "wrapped" belt design, the cog style has less release friction. Also, the notches in the belt inner race allow the repeated bending of the belt around sheaves to occur with less effort. Power and energy consumption are normally reduced by 1-2.
- Cogged belts use standard v-belt sheaves and do not cost any more than standard V-belts, so this is easy to implement as a maintenance practice.

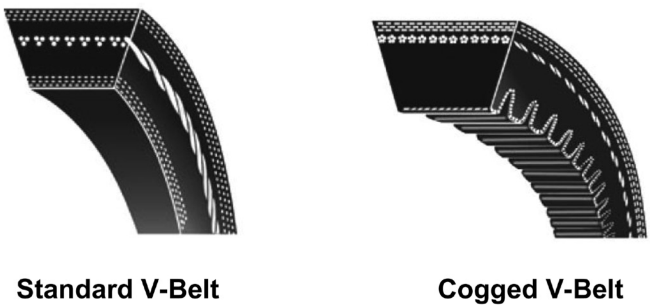


Figure 15-3. V-belt Types

SYNCHRONOUS BELTS

- Compared to V-belts, including cogged belts, the synchronous belt has no slippage at all. Power savings is typically 5%.
- Similar to large motorcycle belt drives and race car superchargers, these belt systems have a matching sheave with notches, and acts very much like a quiet chain.
- Many belt driven devices depend upon the "clutch" action during startup (some slipping) to ease the transition, especially for full voltage starts. Therefore any device that can be expected to jam would not be a good candidate for these non-slipping belts.
- To avoid concerns of damage to fan drives, limit synchronous belt replacements to fans with soft starts or VSDs.
- These belts produce a characteristic noise that may be an issue in sensitive areas, so consider visiting a site where they are in use to be sure.
- Life of the belts is a function of selection—sized to minimum, the life of the belt and sheave will be around five years; generously sizing the parts can double the life span.

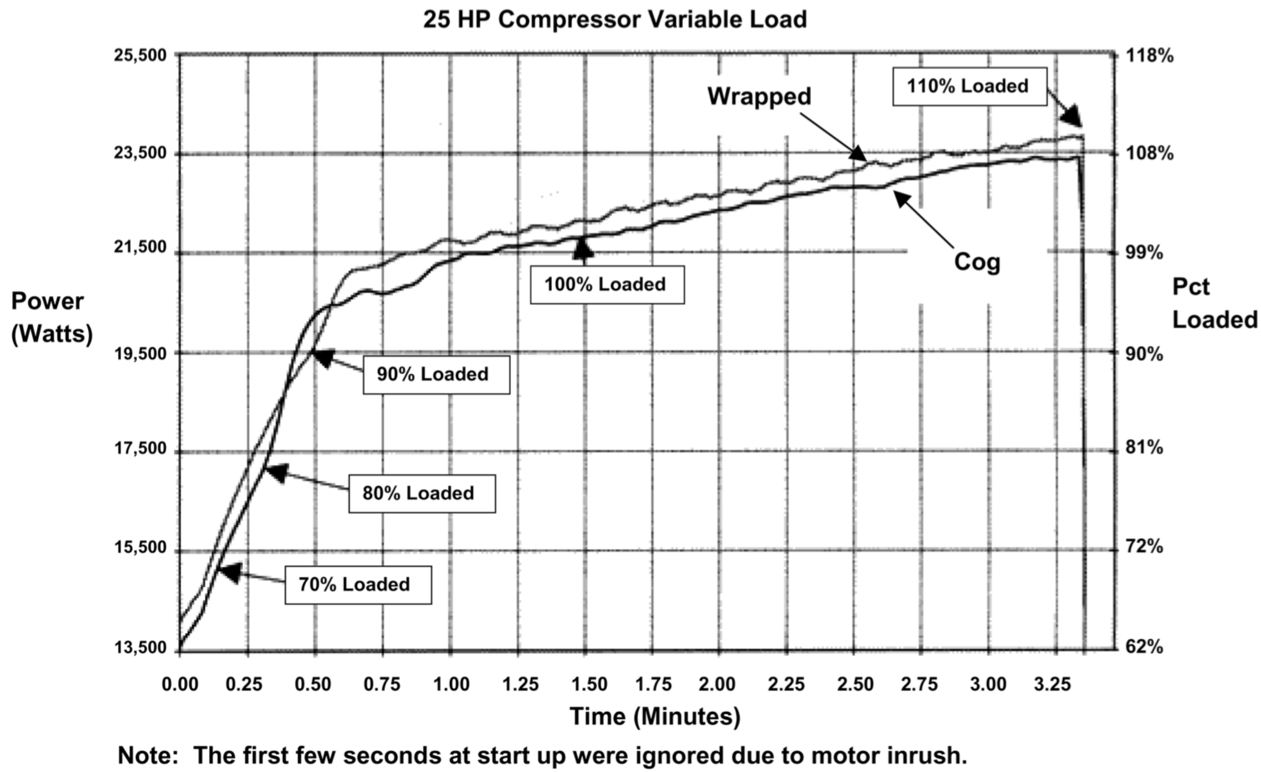


Figure 15-4. Measured energy use with V-Belt vs. Cogged Belts

Source: "Energy Comparison Study for Standard (Wrapped) Versus Cogged V-Belts," Industrial Technology Institute and Detroit Edison, November, 10, 1995.

Source of belt photos: Carlisle

Figure 15-5. Synchronous Belt/Sheave System

VARIABLE SPEED DRIVE CONSIDERATIONS

A successful VSD project is one that succeeds both functionally and financially. Here are some considerations.

The financial hurdle requires identifying the number of hours of operation at various percent of full load, and the efficiency of the VSD compared to the existing drive. The load profile is essential to avoid over-stating savings.

Load Profile

Determine hours per year in different ranges of load, such as:

- Hours/year from 0 - 60% flow
- Hours/year from 60 - 80% flow
- Hours/year from 80 -100% flow

Ranges can be more granular if sufficient data is available to support the accuracy. Most of the savings will occur in the 50-100% range since the magnitudes are highest, e.g. a large percentage of a small number is still a small number. Assuming the system is not altered other than the drive, the analysis is straight forward once the load profile is established, using the differential efficiency at each category of load, multiplied by the hours at those loads.

Functionality

Functionality considerations are important, to assure the system has no operational issues associated with the change to variable flow:

- Will the system accommodate variable flow? A constant flow system has no savings potential from variable speed, by definition. Conversions can be complex, for example, if changing 3-way control valves to 2-way control valves.

- Will variable flow affect the system performance? Impacts can be air/water distribution issues, heat transfer (hot spots), reduced ventilation, air binding, sediment fall out.
- Will air distribution elements (e.g. ceiling diffusers) perform with reduced velocity?
- Will existing motors perform with the VSD? This has been a stumbling spot for variable frequency drives (VFD's) which are supposed to have "inverter-rated" motors. An existing motor controlled by a new VFD may experience a premature failure.

BEST EFFICIENCY POINT (BEP)

Each fan/pump has a personality for efficiency, and a sweet spot also called the "Best Efficiency Point." Proper evaluation of BEP includes consideration of the motor and any throttling device used (see **Wire-to-Water Efficiency**, this chapter) as well as the system it serves.

When the operating point is fixed at the maximum design point, the equipment selection is straight forward and focuses on the best efficiency. However, many fan/pump distribution systems respond to varying loads

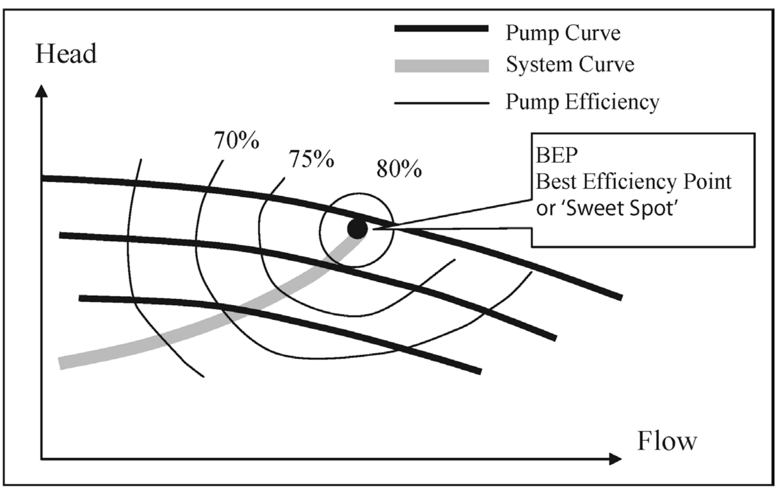


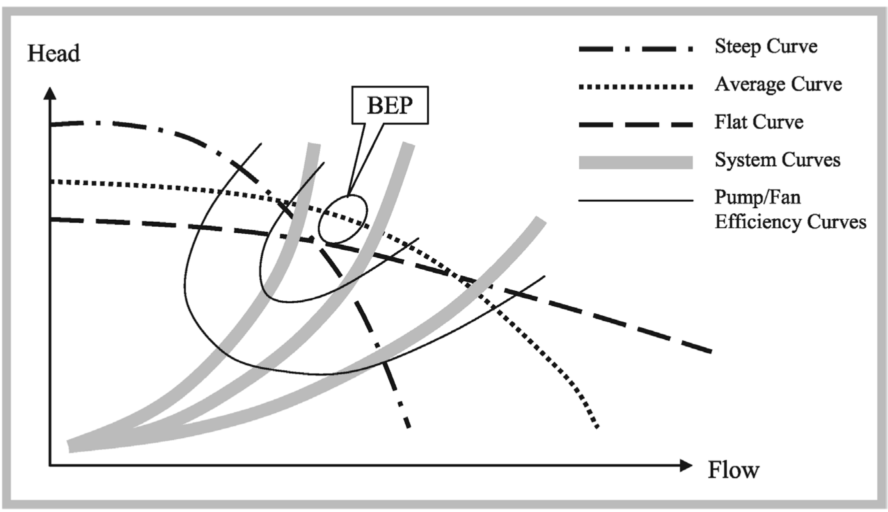
Figure 15-6. Example Pump Selection at BEP

with varying flows. When this is the case, the equipment selection will consider full flow and part flow conditions, and the prevalence of each.

PUMP/FAN CURVE CHARACTERISTICS

Fans and pumps are designed with different flow characteristics for different applications. Some have "flat curves," some "steep curves" and some in between. For new fan/pump applications general application rules of thumb are:

- Steep curve pumps are well suited for applications that expect nearly constant flow with varying head. Examples are feed water pumps and domestic water booster pumps.
- Flat curve pumps are well suited for applications that expect nearly constant head with widely varying flow. Examples are hydronic circulating pumps.



	Flow Response to System Pressure Change	Pressure Response to System Flow Change
Steep Curve	Small change in flow	Large change in pressure
Average Curve	Moderate change in flow	Moderate change in pressure
Flat Curve	Large change in flow	Small change in pressure

Figure 15-7. Pump Curve Characteristics

Many pumps are in between these two characteristics, a so-called 'medium curve'.

Simply replacing a pump with one of higher efficiency is a too-narrow focus of thought process. The characteristic curve type must also be compatible to avoid significant changes in system performance.

WIRE-TO-WATER EFFICIENCY

The overall energy efficiency of a pump or fan is a combination of the efficiency of the motor, pump/fan, and any throttling apparatus. For example, a variable speed drive may have a 97% efficiency, but if the pump/fan operates continuously at full load the VSD not only serves no purpose but adds 1/0.97 = 3.1% energy use. The collective effect becomes more significant at reduced load, especially below 70% load. Pumps/fans serving friction-only loads have relatively constant efficiency at all speeds—this is not the case for fans/pumps serving predominantly static head/lift loads. Efficiency at low speeds (<30%) is unknown.

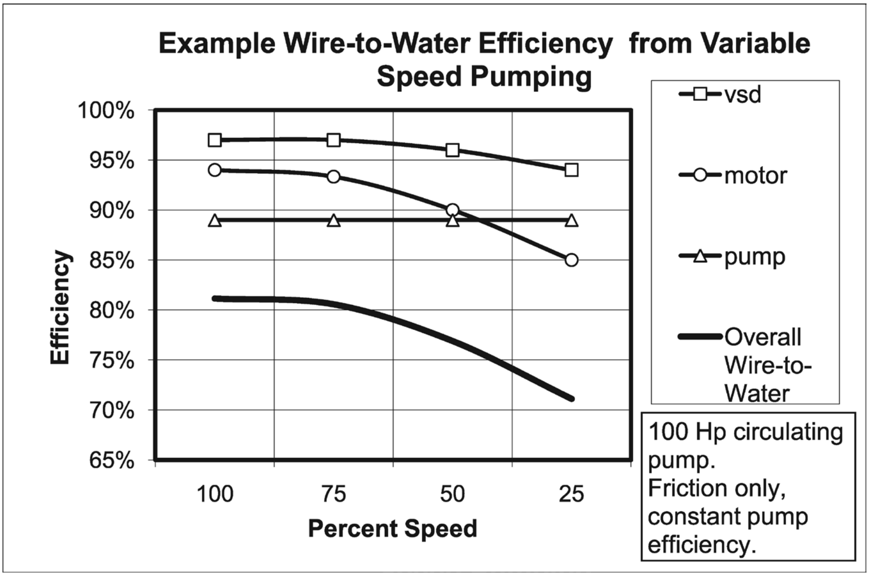


Figure 15-8. Example Wire-to-Water Efficiency

VSD SAVINGS: SQUARE INSTEAD OF CUBE

Using the affinity laws (See **Chapter 21—Formulas and Conversions "Affinity Laws"**) suggests that savings diminishes proportionally as the cube of the speed change. This is true for the fluid power for a fixed system path when only friction is involved, but does not hold true all the way up the food chain to the electrical input. The reasons for this are that motor, drive, and sometimes fan and pump efficiencies fall off at reduced speeds (for various reasons), and consume some of the savings. One way is to assume power reductions are proportional to the square of flow rate—rather than the cube. Here is a modified affinity law to consider, which will yield more realistic results.

Affinity Law, Modified for VFD Savings:

$$\mathbf{Hp2 = Hp1 \times \left(\frac{N2}{N1}\right)^{2.0}} \leftarrow \textbf{use square instead of cube}$$

AFFINITY LAW APPLICATION WHERE STATIC HEAD IS INVOLVED

Where pump or fan power is in response to pipe/duct friction, the system power response to a reduction in flow follows the affinity laws nicely, and technologies that vary the speed of the pump/fan will harvest good savings. Additional losses to potential savings occur from pump/fan inefficiencies at reduced speed and losses of the variable speed technology itself.

Static Head

When systems include static head as well as friction, only the friction portion is reduced at low speeds and low flows. Only the non-static portion of the total pump head will see exponential savings from reduced flow. This underscores the fact that very little savings are available from variable flow when the majority of the work is lift –savings are linear with reduced flow, but the delivery of the fluid takes proportionally longer. Proper analysis requires separating the static and dynamic portions of the fluid flow work. Failure to recognize this can lead to substantial errors in estimating variable flow benefits.

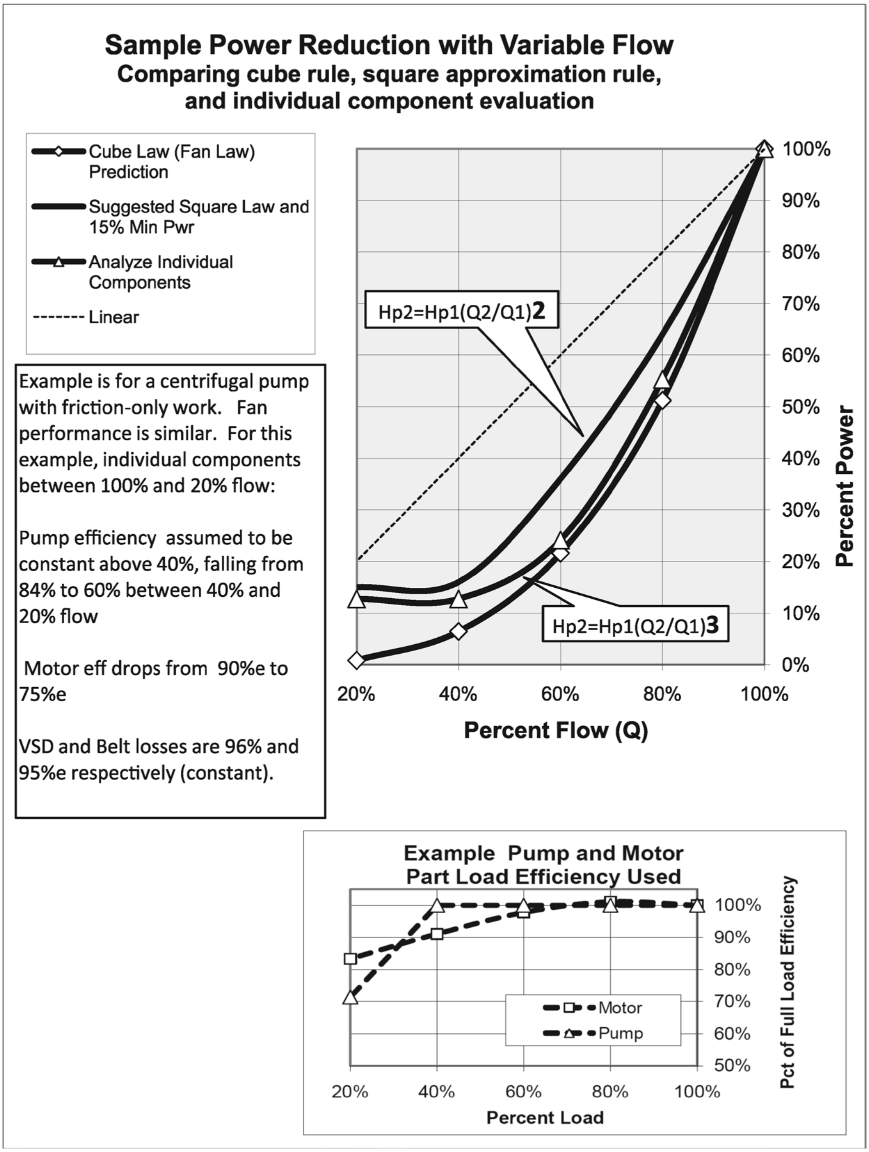


Figure 15-9. Sample Part Load Power Curve Comparing Cube Rule and Square Approximation.

Examples of static head for centrifugal fan/pump applications:

- Cooling tower (10-20 feet of static lift from basin to nozzle)
- Cooling tower, spray type (pressure at the nozzle, plus any lift)
- Domestic booster pump (static lift from pump inlet to upper floor of the building)
- Municipal pumping up hill or into storage reservoirs.
- VAV systems (constant pressure setting used in control, to assure VAV box operation)
- Variable flow hydronic pumping (constant pressure setting used in control, to assure control valve operation).
- Variable flow compressed air systems with a constant backpressure for tools.
- Boiler FD and ID fans (wind box pressure)
- Boiler feed water pumps (drum pressure)

Example variable flow systems with mixed friction and lift:

System	**Percent work from lift**
Cooling tower pump with 75 ft w.c. total pressure and 15 ft of lift.	20%
VAV air handler with 4 in. w.c. total pressure and 1 in. w.c. maintained downstream pressure.	25%
Variable flow chilled water pump with 100 ft. w.c. total pressure and 40 ft w.c. maintained downstream pressure.	40%
Municipal water pump with 300 ft. w.c. total pressure and a 180 ft vertical rise.	60%
Compressed air system with 125 psig discharge pressure and 100 psig minimum downstream pressure.	80%

See also**: Savings Impact When Controlling to a Constant Downstream Pressure—VAV and Variable Pumping**, this chapter.

Efficiencies of Centrifugal Fans/Pumps at Reduced Speed

Figure 15-10A shows system curves with varying amounts of lift involved. When lift is included, the system curve begins at something other than zero pressure and the dynamic portion rides on top of the steady lift requirement. **Figure 15-10B** shows characteristic pump/fan curves plotted on top of the system curve for systems with pure friction and varying degrees of lift. Only the portion of the system curve related to friction can be assumed as being constant efficiency. Note: Published data for efficiency vs. speed is not readily available for project-specific analysis.

Generalizations:

- Within the limits of fan/pump curve published data, the assumption of constant efficiency is reasonably close for systems that are purely friction.
- For very low flows, fan/pump efficiency will drop.
- For systems dominated by lift, the fan/pump efficiency reduces steadily at reduced load as the system curve crosses the efficiency lines.
- For systems with static head involved, including applications with a constant downstream pressure, the constant efficiency assumption is not accurate and must be adjusted depending upon the proportions of friction and static work involved. Many system fall in to this in-between category and will experience some efficiency loss.

Example 1:

A VAV air handler has a total pressure of 4.5 in. w.c. and a downstream control pressure of 1 in. w.c. Lift component: 1/ 4.5 = 22%. Ref **Figure 15-10B** and the closest curve denoted "20% lift." The fan efficiency is steady down to about 80% flow and then begins to drop. Savings will only occur for work that occurs above the static lift line.

See **Chapter 9—Load-Following Air and Water Flows vs. Constant Flow (VSD Benefit)** for solved example.

Example 2:

A single-zone VAV air handler with no downstream pressure control is has a total pressure of 3 in. w.c. and no lift component. Ref **Figure 15-10B** and the curve denoted "friction-only." The fan efficiency is nearly constant throughout the range of flow reduction. Savings will follow affinity laws directly.

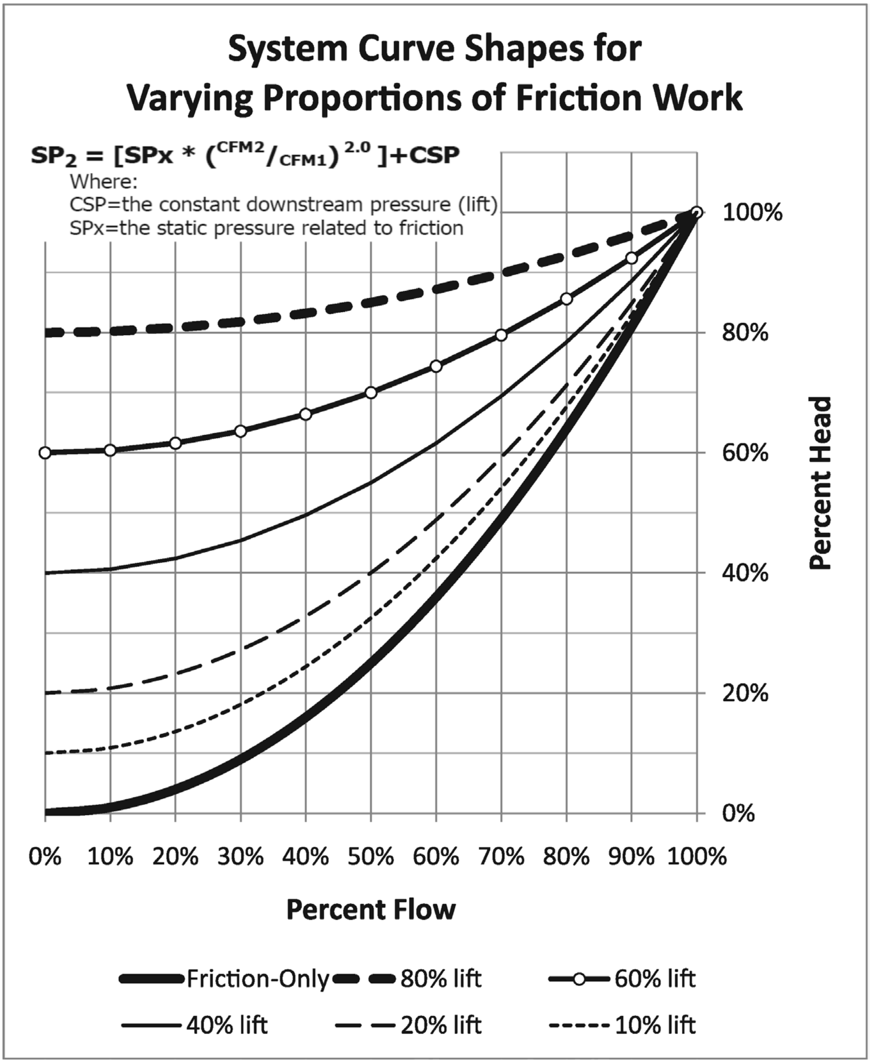


Figure 15-10.A System Curve Shapes for Varying Proportions of Friction Work.

Efficiency is relatively constant for pure-friction systems.

Efficiency degradation occurs to a greater or lesser degree depending on the proportion of lift.

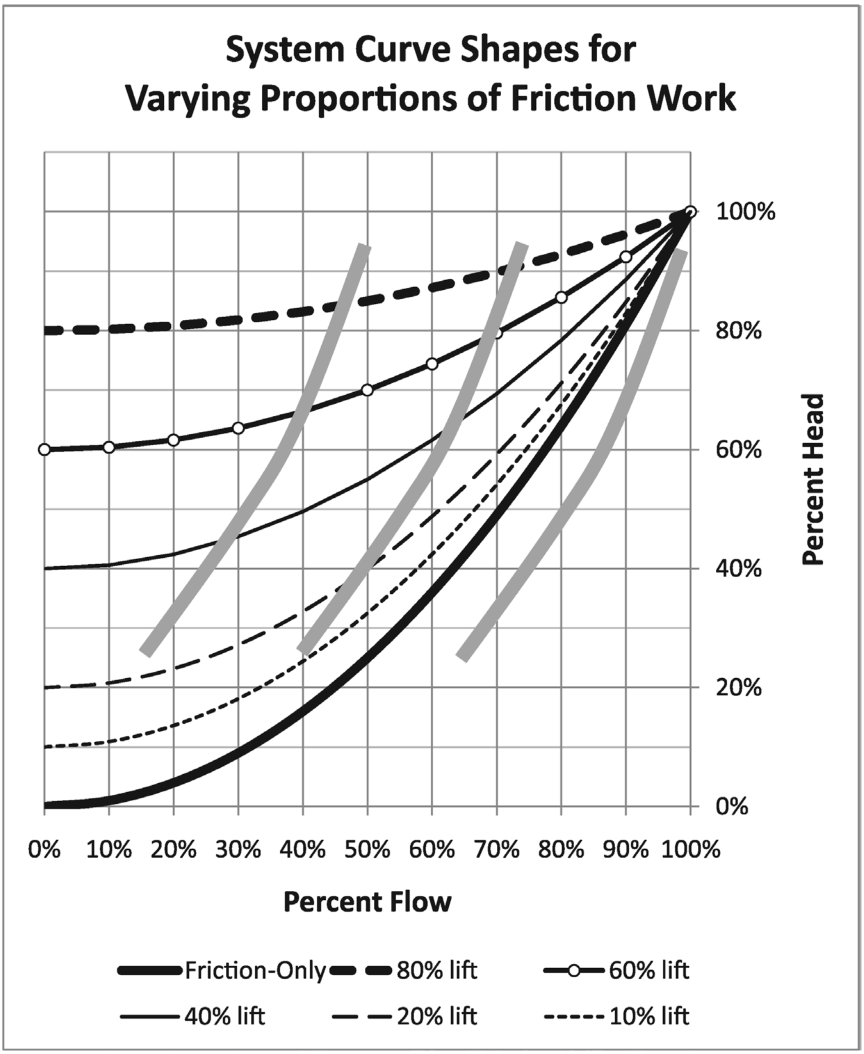


Figure 15-10B. Characteristic Centrifugal Fan/Pump Efficiency at Reduced Speed

Savings at Very Low Speeds

Predictions of savings at motor speeds less than 30% may not be realistic. In addition to the motor and drive losses, the process itself may behave differently at low loads from things like laminar flow in coils or overlapping heating and cooling. Use caution in accepting computer predictions of very low flows for variable flow water/air systems.

SAVINGS IMPACT WHEN CONTROLLING TO A CONSTANT DOWNSTREAM PRESSURE—VAV AND VARIABLE PUMPING

Predicted savings require a de-rate to account for maintaining a constant downstream pressure. For these systems, the static head represents some portion of the overall work, even though the friction portion of the fan/pump work follows the affinity laws. Were it not for the requirement of downstream pressure, the fan/pump would deliver the fluid to the terminal unit at zero pressure.

A common example of this is VAV (variable air volume) air systems, which have a downstream setting for maintained pressure. The same concept applies to variable pumping.

Since these systems are not purely friction, the assumption of constant fan/pump efficiency is not true either, and accordingly some degradation of fan/pump efficiency should be expected at reduced speed.

For variable flow hydronic systems and VAV air handling systems, there is a residual pressure requirement to operate the terminal units (VAV boxes or control valves), and so the fan/pump HP will not turn down fully according to the affinity laws without first separating the lift component. The dynamic losses for piping, ductwork, filters, coils, etc. are free to drop in pressure with the affinity laws. The analysis method reduces system pressure by the fan laws but ADDS the residual pressure. The pressure reduction curve accounting for this constant downstream pressure requirement becomes

$$\mathbf{SP_2 = \left[SPx * \left({}^{CFM2}/_{CFM1}\right)^{2.0}\right] + CSP}$$

Where:

CSP=the constant downstream pressure (lift)

SPx=the static pressure related to friction

This establishes a system curve with a non-zero bias. See **Figure 15-10A**.

SAVINGS FROM LOWERING DOWNSTREAM MAINTAINED PRESSURE SETTING

The relationship **HP2 = HP1 ×** $\mathbf{(^{SP2}/_{SP1})^{y1.5}}$ is a derivation of the affinity laws that applies to friction-based fan and pump systems without the constraint of a maintained downstream duct pressure. See **Chapter 21** for derivation.

However, many commercial HVAC fan/pump systems use a maintained downstream control pressure and so this relationship must be considered. Ideally, the system would be modeled and run with current and proposed lower residual pressure setting. See **Figure 15-10A**.

A quicker, but only approximate, method can be used to estimate savings by using the standard equation and de-rating the exponent. This de-rated exponent is a rough compensation for the fact that the pressure reduction is only occurring in the supply portion of the distribution network—losses in the balance of the system, including return duct/piping, heat exchangers, air handler casings, etc. are not affected. So, only external losses contribute to this and of those, only the supply portion.

The maximum value of the exponent is 1.5, but will normally be less. The exact value of the exponent depends on the fraction of the total system resistance affected by the setpoint reduction.

A=Fraction of total system resistance affected by setpoint reduction	Exponent = 1.5*A
100%	1.5
75%	1.1
67%	**1.0**
50%	0.8
25%	0.4

A shortcut method that works for most standard HVAC circulating air/water systems is to assume assumes 2/3 of the fan/pump fan energy is spent on the supply portion and simply drop the exponent—such that the savings from reduced pressure varies proportionally with the pressure reduction:

$$\mathbf{HP2 = HP1 \times \left({}^{SP2}/{}_{SP1}\right) \leftarrow}$$

Example: A 50 Hp VAV fan load has a maintained downstream duct pressure of 1.3 in. w.c. Lowering the setpoint from 1.3 to 1.0 in. w.c. will reduce Hp to 50*(1.0/1.3) = 38.5 HP (30% reduction).

Chapter 16

Lighting

GENERAL

- About 12% of the cost of light [in its lifetime] is from the hardware and maintenance. The other 88% is from energy use.
  Source: "Found Money: A CFO's Guide to the ROI of Lighting", Energy & Power Management, July 2005

- Because the ballast mostly determines how many watts are used, ballast choice is critical to the energy efficiency success of a project.
  Source: "FEMP Lights" Training Material, Federal Energy Management Program, 2005. Original source of data Heschong Mahone Group, Inc.

- During summer, lighting savings have a double-dip savings effect for air conditioned spaces, since the excess wattage becomes heat which becomes load on the A/C system. Depending upon the number of cooling hours, the A/C savings from lighting retrofits may be significant or may be ignored.

- During winter, the waste heat from lights act as a supplement to the building heating system, and heating cost increases, eroding some of the lighting savings. When heating systems are marginally sized it is possible for a lighting retrofit to cause the heating system to become under-sized, and this should be checked.

- Moonlight on a cloudless night on a white beach is about 1 foot candle.

- Indirect Lighting is typically 15% less efficient than direct systems, because the light must first bounce off the ceiling.
  Source: "FEMP Lights" Training Material, Federal Energy Management Program, 2005. Original source of data Heschong Mahone Group, Inc.

- Costs of turning lights off and on.
  Source: "FEMP Lights" Training Material, Federal Energy Management Program, 2005. Original source of data Heschong Mahone Group, Inc.
  — Fluorescent: The economic break-even point is typically between 5 and 15 minutes between switching.

— HID: The economic break-even point is around one hour between switching.

LIGHTING TERMS

Light output of fixtures and bulbs is rated in **lumens**, measured at the source.

Foot-candles (FC) is the standard measure of illuminance. Units are lumens per square foot, usually measured at the work surface.

Lux is the metric unit for illuminance, in lumens per square meter. To convert FC to Lux, multiply foot-candles by 10.76.

Higher **color temperatures** mean more white/blue appearance. Units are degrees K (Kelvin). A 4100K light source is seen as blue or white, and a 3000K light source is more yellow and warm color.

CRI is a 0-100 scale indicating how perceived colors match actual colors. The higher the number, the closer the match. This is an industry standard test using (8) different colors and a reference light source. Incandescent lighting has a CRI of 100.

Efficacy is a term used for evaluating lighting sources and is in units of **lumens per watt**. In general, efficacy expresses efficiency with units instead of percent.

Ballast factors can be selected to alter light output and energy use; in effect tuning the lighting system. The ballast factor is the ratio of a light output using the selected ballast, compared to the rated light output. General purpose ballasts have a ballast factor less than one; special ballasts may have a ballast factor equal to or greater than one.

Example of controlling energy use with ballast factor: A fluorescent lighting retrofit is expected to produce 55 foot candles at the work surface compared to 40 foot candles existing that is acceptable to the customer. In addition to the savings of more efficient bulbs and ballasts, selecting a ballast factor of 0.70 instead of 0.87 will reduce lighting power and light by a factor of 0.70/0.87 = 0.80, and reduce foot candles from 55 to 41, providing an additional 20% energy savings.

DIMMING

<u>Incandescent</u>: Easy, just a rheostat.

<u>Fluorescent</u>: The reduction in power is not as great as the reduction in light output, therefore efficiency declines somewhat with dim-

ming. Requires special ballast. Mixed success with Compact Fluorescents (CFL).

HID: Problematic, not common. However, bi-level switching does work well.

LED: Variety of methods, varying voltage at input.

LIGHTING VS. DISTANCE

Light intensity (foot-candles) decreases as the square of the distance.

$\mathbf{FC \sim 1/d^2}$

$\mathbf{FC2 = FC1 * {}^{1}/({}^{d2}/_{d1})^2}$

Ex. If a light source produces 50 f-c at 20 feet, that same light source will produce about $50*[1/(30/20)^2] \sim 22$ f-c

To have the same light level at a work surface, having the light source twice as far away means starting with four times as much light. Or, to have the same light level at a work surface, having the light source a fourth of the distance away means starting with 1/16th of the initial light intensity and light power. Examples leveraging this concept:

- **Task lighting** as a supplement to overhead lighting. Where required light levels at the work surface are higher than general lighting (small parts, reading, fine detail), task lighting provides 'spot lighting' at the higher intensity rather than increasing the power to the bulk overhead lighting.
- **Lowering the fixture height** in re-purposed spaces, such as a 30 foot high bay warehouse now only used for 10 foot high storage. Note spacing requirements for light coverage change and may limit this.

LIGHT COLORED SURFACES

Reflective (light) colored surfaces increase effectiveness of the lighting since less of it is absorbed. This includes floors, furniture, walls, and ceilings.

It can take up to 40% more light to illuminate a dark room than a light room with a direct lighting system.

Source: "FEMP Lights" Training Material, Federal Energy Management Program, 2005. Original source of data Heschong Mahone Group, Inc.

Reflectance Parameters for Picking Interior Surfaces and Colors

Min 80% reflective	Ceiling
Min 50% reflective	Walls
Min 25% reflective	Floor and furniture

Table 16-1. Reflective Values of Common Colors

Color/tone	Reflectivity
Bright White	80-90 %
Light Tint	60-80 %
Pastels	40-60 %
Bright Colors	20-40 %
Deep Colors	10-30 %
Dark Colors	5-10 %

Source: "FEMP Lights" Training Material, Federal Energy Management Program, 2005. Original source of data Heschong Mahone Group, Inc.

Photographic "middle gray" and Caucasian flesh tones are about 20% reflective.

A "true" black absorbs almost 100% of the light that strikes it 5-15% reflectance (85-95% absorption)—dark furniture, carpet.

LIGHTING TECHNOLOGY PROPERTIES

Table 16-2. Lighting Technology Properties - Common technologies

Conventional sources arranged by efficacy (lumens per watt)

The table is a summarized generalization of the ratings of product technology families. It is not intended to cover all manufacturer ratings for all products. For specific information, consult individual manufacturers.

Source for conventional lighting: Osram Sylvania 2012 Lamp and Ballast Product Catalog

Source for LED lighting: LED Basics, EERE / DOE, 2015

Source for LED PAR lamp life: manufacturer's data

LPW = Lumens per Watt

CFL = Compact fluorescent (self-ballasted)

Failure criteria F50" is when 50% of a large array of these lights will have failed

Failure criteria L50 is when lumen output is below 50% of initial value; similar for L70

LED life spans are projected.

LED life is strongly affected by operating temperature

Fluorescent life is strongly affected by cycle time

Note 1. DOE General Service Fluorescent Lighting Rule requires 72-97 minimum LPW as of 2012

	Mean LPW(efficacy) Lamp only	Mean LPW(efficacy) With ballast	CRI 0-100	Avg Life (hours)	Life Basis	Instant Re-Strike?
Incand. / Halogen	5-20	5-20	96-99	7.5-10k (4k com)	F50	Yes
Compact Fluor CFL	N/A	35-52	70-82	6-10k	F50	Yes
Linear Fluorescent (1)	29-99	15-98	52-90	7.5-20k	F50	Yes
High Perf Linear Fluor	90-110	90 to 100	80- 90	24-60k	F50	Yes
Mercury Vapor	19-49	15-45	20-40	16-24k	F50 or L50	No
Metal Halide	35-96	25-90	60-95	1.5-15k (10k com)	F50 or L50	No
High Press. Sodium	55-135	41-115	20-22	24k	F50	No
Low Press. Sodium	90-142	53-108	0	18k	F50	No
LED PAR (spot / flood)	N/A	67	70-90	25k	L70	Yes
LED Lay-In (2x4)	N/A	93	70-90	30-50k	L70	Yes
LED Hi/Lo Bay	N/A	90	70-90	30-50k	L70	Yes

LED TECHNOLOGY

Basis of savings: Reduced power for achieved lighting levels.
This is a rapidly emerging technology. Long term R&D goals of 250 lumens per watt are claimed to be possible (Source EERE / DOE 2015).

Economic analysis: Longevity is key to economic vitality when LEDs cost more and sometimes the O/M cost reduction (fewer replacements) is as valuable as the energy savings. Fair economic evaluation considers the differences in life spans of LED vs. competing technology. For very long business horizons (10+ years), life cycle cost analysis will be appropriate. For very short business horizons (1 or 2 years) the first cost may be prohibitive. For business horizons between very short and very long, a nominal period of "five years" or "ten years" may be chosen for the evaluation; within that time frame, the first cost, energy savings, and replacement costs are compared to see if there is meaningful value to enable a good choice.

A variety of drop-in replacement products are available, including:
- Spot and accent lighting
- Standard incandescent shapes with screw-in base
- Replacement pin-based linear tubes for fluorescent strip retrofit
- Street lights

LED lighting technology advantages:
- Long life
- Directional light output – important property for accent and spot lighting
- No mercury
- Good cold temperature operation
- Instant on – no warm up time
- Lifetime not affected by frequent switching – compatible with occupancy sensor controls
- Dimmable - with special dimmers
- No UV emissions –important property for lighting where fabric and artwork are involved

LED lighting technology challenges:
- Initial cost
- Reduced light output over time, like most light sources
- Sensitive to heat build-up

LED lighting efficiency:

The standard unit of measure is lumens per watt (efficacy). One test of a Compact Fluorescent (CFL) down light to an LED down light showed similar efficacy in measured lumens per watt (Source: Comparing LED Recessed Down lights to Traditional Light Sources, EERE, 2009) using LED test data from 2006 and 2007.

Equivalent Lighting Power for LEDs

Lighting quality is a challenging measurement. Counting photons is not enough, since the only ones that 'count' are the ones we see; further there is the perception of color and clarity. Standard CRI testing shows similar values between LED and Fluorescent, in the 75-90 range (Source: manufacturer's data) so there does not appear to be basis for using less lumens of LED light due to superior quality of light.

However, there is basis for requiring less light from LEDs and a basis to estimate it. LEDs, by nature, are 'directional'. All of the light is 'aimed' (example: from a ceiling: downward). Conventional lamps (fluorescent, incandescent, high intensity) emit light in all directions including up into the fixture where some of it is lost. Lumen output ratings are for a bare lamp, but the light that "matters" is the light that makes it to the work surface. Therefore, it is reasonable to normalize LED lumens to conventional lumens, either by a factor to decrease the conventional light source for "delivered light" or to increase the LED light source. For a directional LED source, the 'fixture efficiency' is 100%. The factor to equalize conventional to LED is the fixture efficiency of the conventional source.

Example: A conventional light source is inside a fixture with 80% efficiency. The lamp lumen output is 10,000 lumens. Estimate the lumen output for an LED replacement, to compensate for the fixture loss the LED does not have to overcome.

Usable lumens = lamp lumens * fixture efficiency

= 10,000 * 0.8 = 8,000 lumens minimum for LED

So, for comparing LED to a conventional light source in a fixture with a 70% fixture efficiency, even if the lamps have equal efficacy in lumens per watt, the LED will produce the same light at the work surface with 30% less power.

When comparing LED to other light sources for equal lumens, there are "initial" and "mean" lumens. Initial lumens (new lamp) are highest, and

Table 16-7. Representative Light Fixture Efficiencies
Source: Manufacturer's Data

Fixture Style	Fixture Efficiency (approximate)
2x4 Ceiling, open "shop light"	0.95
2x4 Ceiling, plastic lens	0.80
2x4 Ceiling, open cell	0.75
2x4 Ceiling, with mirror reflector	0.90-0.95
Recessed 'box' HID fixture with lens	0.65
Recessed can with non-directional lamp (other than reflector lamp)	0.50-0.70
Indirect Lighting	0.85-0.90

the light output (and lumens per watt) decline over time. Each technology has a different lumen depreciation curve, but the most important consideration is to either compare two technologies on initial lumens or on mean lumens. Specifically, do not compare one technology on initial lumens vs. another on mean lumens.

Economic Comparison for LED vs. Other Technologies

With all things equal, comparing price directly is appropriate. However, when important properties of technology options are different, there must be a way to normalize them to make good decisions. It is always good to evaluate light levels before any lighting replacement, rather than 1-for-1 replacement. Once this is completed, evaluating different technologies on equal delivered useful light is rational. Choosing new lights on energy savings with less delivered light is a choice, not a comparison.

- **Lumens per watt (efficacy)**. This adjustment takes into effect the directional nature of LED lighting and can be reasonably normalized to non-directional options by fixture efficiency, so that what are being compared are the lumens delivered to a work surface, where it matters.
- **Light quality**. The conventional basis for discerning light photons that are useful to human eyes is CRI (color rendering index). For example, when a light source has a very low CRI, it is reasonable to say some of the lumens reaching the work surface are not usable and do not 'count.' However, indoor lighting designs normally only include light sources proven to be viable with indoor viewing tasks; i.e., low pressure sodium lighting is not applied indoors. If light quality is proven to be meaningfully different between technologies

on a quality level, normalizing on the quality measurement is possible with an adjustment factor of $(^{CRI}2/{CRI}_1)$ although this is not conventionally done.

- **Life**. There are significant differences in life spans of different technologies. A light source that costs twice as much and lasts twice as long as it's competitor is no more expensive on a life cycle basis. However, different business horizons may or may not acknowledge the long term view. For example, comparing technology A (5 year life) to technology B (9 year life) one approach would find a point in the future when both lamps expired simultaneously (45 years hence) then add the cost of replacements and operating cost over 45 years; meanwhile, the business is pondering if they will even be around five years from now and the math exercise is useless to them. Another option is to identify a span of time that is meaningful to the customer (5 years, 10 years, etc.) and compare the cost of owning and operating the two technologies over that period.

  In some cases, the cost of O/M is a stronger determining factor than electricity cost. Consider parking lot lighting or high bay lighting where group re-lamping is outsourced and expensive; savings from this expense being less frequent can be compelling.

LIGHTING ENERGY USE, PCT OF TOTAL ELECTRIC, BY BUILDING TYPE

Table 16-3. Lighting Energy Use, Pct of Total Electric, by Building Type

Source: "FEMP Lights" Training Material, Federal Energy Management Program, 2005. Original source of data Heschong Mahone Group, Inc.

Sector	Percent
Offices	40%
Health Care	30%
Hotel/Hospitality	30%
Residential	25%
Industrial	20% (highly variable)
Retail	55%
Warehouse	40%

LIGHTING HOURS BY BUILDING TYPE

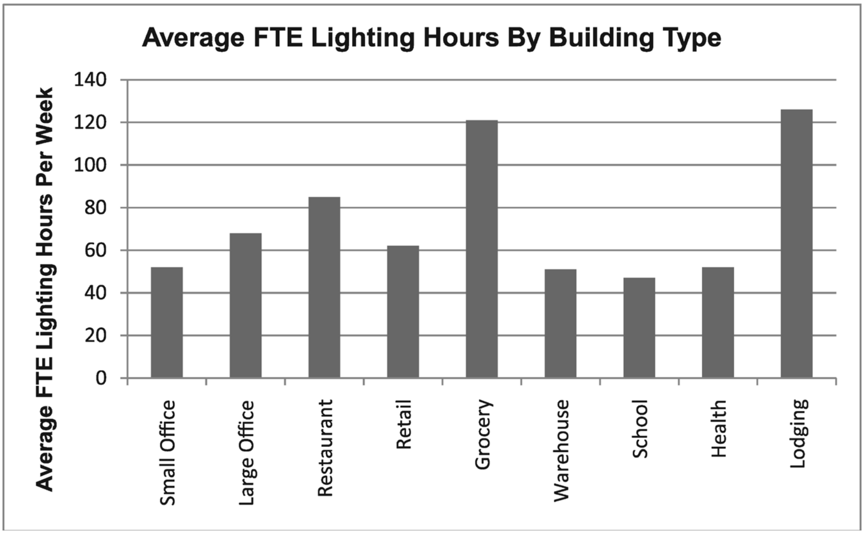


Figure 16-1. Average Lighting Hours by Building Type

FTE = Full Time Equivalent.

Source: "FEMP Lights" Training Material, Federal Energy Management Program, 2005. Original source of data Heschong Mahone Group, Inc.

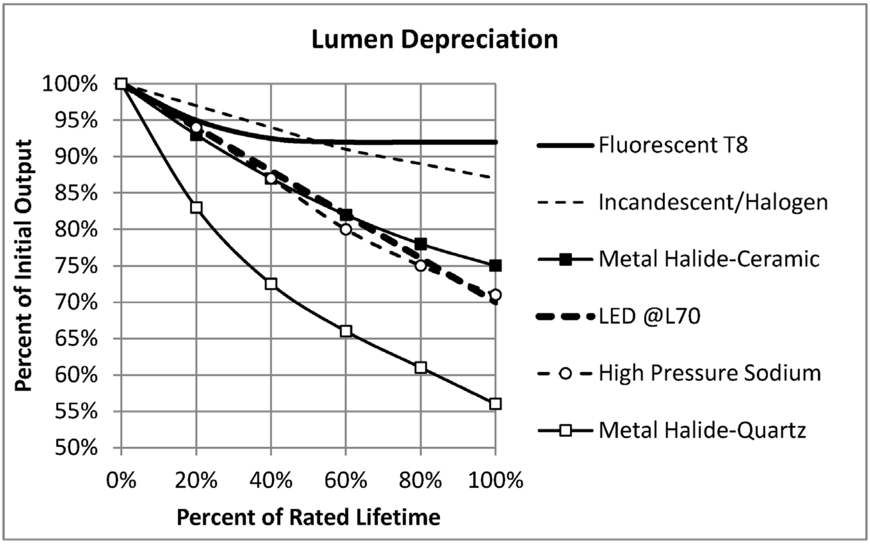


Figure 16-1a. Characteristic Lumen Depreciation Curves

Source of data: *Lumen Maintenance and Light Loss Factors: Consequences of Current Design Practices for LEDs*, 2013,PNNL/ DOE. Values used are roughly centered from the ranges of values provided. Use actual manufacturer values where available.
LED data is projected. Ref. standardized lumen maintenance measurement method for LED packages (LM-80-08 /IES 2008) to extrapolate up to six times the number of tested hours.

TYPICAL RECOMMENDED LIGHTING LEVELS

These values are only intended to help identify areas that are substantially over-lit.

Example: for the stated range of 30-50 FC, a measured value of 51 is not cause for alarm, but a measured value of 75 suggests de-lamping may be viable.

This is not intended as a substitute for a lighting designer's judgment – and the customer may want or need the additional light. Any energy conservation measures that reduce light levels need appropriate basis and documentation to avoid possible liability.

Table 16-4. Typical Recommended Lighting Levels

Business	Ambient FC	Task FC	Remarks
Office Bldg	20-50	50 Reading 30 Computer keyboards	10-20 Lobbies
Department Store	50-100	200-50 Cashier	Discount stores generally higher light levels
Jewelry	30-75	200-500 Manufacture Cashier 20-50	150-500 Accent lighting
Merchandising	30-75	20-50 Cashier	150-500 Accent lighting
Furniture showroom	10-30 Appraisal	30-100 on furniture	150-500 Accent lighting
Schools - classroom	50 Reading	30 Computer keyboards	50-100 Labs
Restaurant dining sitting area	5-10 Dining	50-100 Kitchen	10-20 Cleaning
Hotel guest room	10-20	20-50 Bathroom	10-20 Lobbies
Grocery Store	100 (vertical) High activity 75 Medium activity 30 Low activity	20-50 Cashier	
Warehouse	10 Inactive	10-50	Bulky to small items
Church	50-75	100-150	
Kitchen	50-100	50-100	
Residential	5-10	50-100	
Parking Garage (enclosed)	50 Entrance 5 General Parking	---	
Parking Lot (exterior)	0.5-2	---	

Source: Stephen W. Leinweber, L.C., C.L.E.P.

These values are only intended to help identify areas that are substantially over-lit. For example for the stated range of 30-50 FC, a measured value of 51 is not cause for alarm, but a measured value of 75 suggests de-lamping may be viable. This is not intended as a substitute for a lighting designer's judgment—and the customer may want or need the additional light.

LIGHTING OPPORTUNITIES

Source: "FEMP Lights" Training Material, Federal Energy Management Program, 2005. Original source of data Heschong Mahone Group, Inc.

Definite

- Four lamp fluorescent troffers using T-12 lamps and standard magnetic ballasts
- Mercury vapor lighting of almost all kinds
- Incandescent down-lights in public spaces, that are not dimmed for functional or aesthetic purposes
- Incandescent EXIT signs
- Spaces that are over-lit, especially if removing lamps or fixtures is appropriate
- Retail space employing R-40 or ER-40 track lighting

Maybe

- 2 + 3 lamp troffers, T-12 lamps and magnetic ballasts
- Industrial fluorescent lighting
- Private offices without occupancy sensors
- School and retail T-12 Fluorescent lighting
- Fluorescent EXIT signs
- Any location with short life lamps and high maintenance costs
- Any location being renovated for other reasons
- Locations appropriately converted to a task/ambient lighting system

Slim Chance

- Lighting with T-12 lamps and magnetic ballasts with operating hours limited by occupancy sensors or by a building automation system
- Incandescent lighting with dimming
- Any lighting in service spaces and living quarters where operating hours may be very short
- Areas where day-lighting suggests installations of photo controls
- Open offices or public rooms without occupancy sensors or other controls

Energy Cost Rate

- Buildings which enjoy very low utility rates are mediocre retrofit candidates, because payback will be longer and Savings to

Investment ratio (SIR) will be lower. A low rate suggests caution in proceeding further.

Hours of Operation

- The longer the hours of operation, the more attractive a retrofit will be.
- More than 5000 hours per year—good chance
- Less than 2500 hours per year—slim chance

OCCUPANCY SENSOR ENERGY SAVINGS

Source: Watt Stopper/Legrand, Technical Bulletin #151, 2002.

Testing was part of EPA's Green Lights Program (2001) and included a total of 158 rooms falling into 5 occupancy types: 42 restrooms, 37 private offices, 35 classrooms, 33 conference rooms and 11 break rooms.

Energy waste is the total waste from lighting on while unoccupied, and energy savings is the amount recouped by the use of occupancy sensors, with a 20 minute time delay.

Table 16-5. Occupancy Sensor Energy Savings

Application	Energy Savings (20-min. time out)	Energy Waste
Break room	**17%**	39%
Classroom	**52%**	63%
Conference room	**39%**	57%
Private office	**28%**	45%
Restroom	**47%**	68%

Source: Watt Stopper/Legrand, Technical Bulletin #151, 2002.

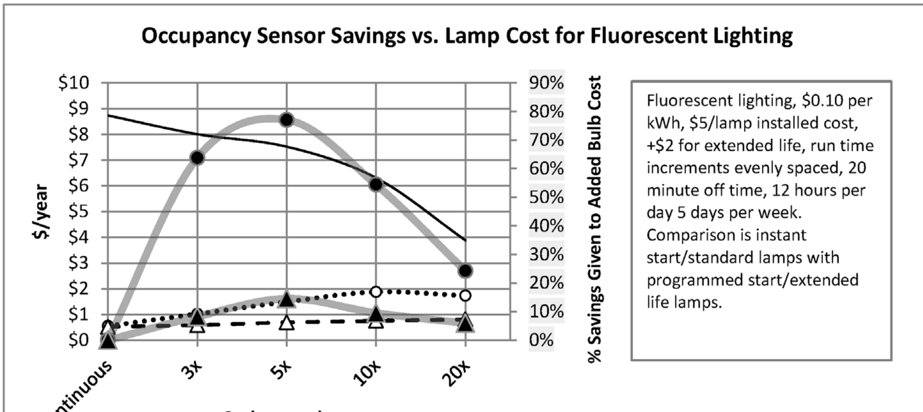
OCCUPANCY SENSOR VS. LIFE OF LIGHTING EQUIPMENT

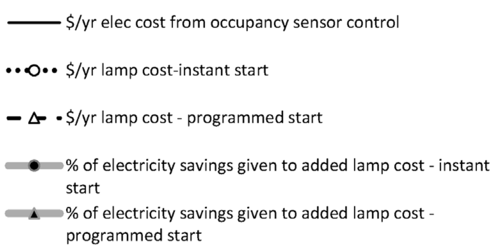
Fluorescent lighting lamp life is affected by the on/off switching frequency. For example, catalog data for fluorescent lamp life gives a value for 3-hour and 12-hour on-cycle for both instant start and program start ballast. Occupancy sensors are generally not paired with rapid start ballast or older magnetic ballast fluorescent systems. By its nature, the program start electronic ballast has the least effect on fluorescent lamp life.

LED technology does not suffer reduced life from occupancy sensor on/off switching (Source: CREE).

Occupancy sensors negate some of the energy savings from turning off fluorescent lights, the amount depending upon the ballast and the cost of electricity. One analysis showed that occupancy sensor dollar savings lost to increased fluorescent lamp replacement expense was 10% average for program start ballast, and 60% average for instant start ballast. See **Figure 16-2**. Thus, best results for occupancy sensor control of fluorescent lighting will be achieved when paired with program start ballasts.

The key to understanding this relationship is comparing continuous operation to occupancy sensor operation. While lamp life is reduced from frequent cycling, the lamps operate less hours; i.e. if a lamp life is reduced by 50% from operating only 50% of the time, the time in years before the lamp must be replaced has not changed.





Hours Life Assumed by Start Cycle

	instant start, standard lamps	programmed start with extended life lamps
15 min	4000	12000
30 min	5000	18000
1 hour	7000	24000
3 hours	15000	36000
12 hours	28000	42000
continuous	30000	42000

Figure 16-2. Occupancy Sensors vs. Fluorescent Lights.
Lamp life span from manufacturer's literature, $0.10 per kWh electricity cost, $7.00 installed cost per extended life lamp with program start ballast, and $5.00 installed cost per standard lamp with instant start ballast. Range of 3-10 starts in a 12-hour business day.

Example use of **Figure 16-2**: At 10 cycles per day, electric cost changed from $8.74/year to $6.31/year, a $2.43/year savings. Lamp cost per year (includes reduced hours life and reduced run hours) changed from $0.56/year to $1.88/yr, a $1.32/yr increase. Net savings is $2.43 - $1.32 = $1.11. Percent of energy savings given to added lamp cost is 1.32/2.43= 54%.

LIGHTING IMPACTS ON HVAC USE BY CLIMATE

Source: Advanced Lighting Guidelines, New Buildings Institute, 2003

See also **Chapter 9—Quantifying Savings, "Higher Efficiency Lighting vs. Existing Lighting."**

With more efficient lights, less lights, or better lighting controls, less heat is released inside the building. In summer this means reduced cooling load. In winter, this means increased heating load.

Table 16-6. Typical Lighting Impacts on HVAC Use by Climate

Location	Cooling Loads	Heating Loads for Large Buildings	Heating Loads for Small Buildings
Phoenix, AZ	-30%	0%	0%
Los Angeles, CA	-23%	0%	0%
San Francisco, CA	-16%	1%	2%
Denver, CO	-16%	7%	22%
Tampa, FL	-33%	0%	0%
New Orleans, LA	-29%	1%	2%
Detroit, MI	-14%	8%	23%
Philadelphia, PA	-17%	6%	18%
Providence, RI	-13%	7%	22%
Knoxville, TN	-21%	4%	11%
Seattle, WA	-7%	4%	13%

LIGHTING POWER BUDGET VALUES WATTS/SF

Most building designs achieve energy code compliance with an overall "points" method rather than prescriptive requirements such as these. However, power or energy budgets for individual items can be useful. The power budget method drives designers to more efficiency lighting technologies to achieve the desired light levels.

Prescribed values of watts/SF change often as energy codes evolve, and are not repeated here.

Table 16-7. Approximate Fraction of Indoor Lighting Operating Concurrently

Average of values from:

- Coincident Demand Factors, Work Paper WPSCNRLG0005, Southern California Edison Company, 2007
- Peak Diversity Factors, Peak Demand Program, Colorado Springs Utilities, 2006
- Coincidence Factor Study, New England State Program Working Group, 2007

Assembly	0.68
Education, K-12	0.63
Education, Secondary (High School only)	0.42
Education, College	0.61
Grocery Store	0.88
Hotel	0.50
Hotel/Motel (Non-Guest Rooms)	0.67
Medical (Clinic)	0.65
Medical (Hospital)	0.77
Office	0.77
Restaurant (Fast Food and Sit Down)	0.75
Retail	0.88
Warehouse	0.79

Chapter 17

Envelope Information

In many commercial building, envelope losses are minor compared to energy use by activities inside the building. But there are exceptions where envelope loads are a substantial part of total energy use, such as:

- Buildings with very light internal activities
- Residential use (hotels, motels, dormitories)
- All-glass buildings

BLC HEAT LOSS METHOD

Annual heating energy use, for buildings whose heating use is dominated by envelope losses, can be estimated using the building load coefficient (BLC) method with reasonable accuracy.

Source: *Energy Management Handbook* 7th Ed, The Fairmont Press.

BLC Equation

E = 24 * BLC * HDD

Where:

E=energy output, Btu (to arrive at input, divide by appropriate efficiency)

HDD = heating degree-days at the building balance temperature

24 = conversion from degree-days to degree-hours

BLC = Building Load Coefficient, Btu/hour-degF

BLC = (sum U*A) + (sum F*P) + (0.018 * Qinfilt) + (1.1 * Qvent)

U = Btu/hr-SF-degF

(sum U*A)=summation of (Uvalue*Area) for components (walls, windows, roof), e.g. the overall weighted average envelope U-factor * overall envelope surface area.

(sum F*P)=summation of (F-factor * perimeter linear feet) for slab on grade

F (un-insulated slab) = 0.73 Btu/hr-LF-degF
F (R-10 under slab) = 0.54 Btu/hr-LF-degF
Qinfilt = infiltration, cubic feet *per hour*
Qvent = ventilation, cubic feet *per minute*

Input Energy
The BLC calculation yields the output energy for the envelope. To calculate the input, divide the output by the heating/cooling equipment efficiency.

BLC and Cooling Loads
Use of the BLC method for cooling loads will understate actual loads, since this method does not include internal loads, dehumidification loads, or solar loads that add substantially and variably to actual cooling loads.

BLC Applications

- BLC with Degree Days to estimate heating loads (noted above)
- BLC + base energy use. If internal loads are quantifiable, they can be added to the envelope, ventilation, and infiltration loads predicted by the BLC equation for total energy use.
- Estimate effect of single envelope element changes in heating energy reduction

$$\textbf{Savings, Btu} = 24 * (BLC_{existing} - BLC_{new}) * HDD * 1/eff$$

Where:

$BLC_{existing}$ = UA of the element, existing condition, Btu/hr-degF

BLC_{new} = UA of the element, new or proposed condition, Btu/hr-degF

HDD = heating degree days at the balance temperature (Note 1)

Eff = average seasonal efficiency of the heater

Note 1: Upon improving the envelope, the BLC reduces (UA becomes less), *but also the balance* *temperature of the building will become less, and so the heating degree-days will be less.* Keeping HDD the same for before and after condition simplifies the equation but is neglecting this change, presuming the effect is small.

- BLC to derive balance temperature.
  The units of BLC are Btu/hr per degF. Dropping the term (24 * HDD) gives an average heat loss in Btuh for a given degree of outdoor temperature. The balance temperature is that temperature where internal heat gains exactly match the heat loss. So, ***if internal gains are known, the balance temperature can be determined.***
  To find approximate balance temperature using BLC:
  BLC * dT = Internal Gain
  dT = Internal Gain / BLC
  Balance temperature = (Indoor temperature – dT)
  and
  Balance temperature = (Indoor temperature – (Internal Gain / BLC))

See example and solution in **Figure 17-1A**. Knowing BLC and internal loads, the balance temperature can also be found by trial and error, incrementing the outside air temperature until the two values are the same (they balance there). See **Figure 17-1B**.

Envelope, Infiltration, Ventilation

Bldg area	7500	SF
Bldg perimeter	350	ft
Bldg height	14	ft
U-wall	0.15	Btuh/SF-degF
A-wall (gross)	4900	
A-wall (net)	3700	SF
U-roof	0.05	Btuh/SF-degF
A-roof	7500	SF
U-glass	0.4	Btuh/SF-degF
A-glass	1200	SF
pct glass	24%	
UA=wall	555	Btuh/degF
UA-glass	480	Btuh/degF
UA=roof	375	Btuh/degF
sum(UA)	**1410**	Btuh/degF

alternate calc

total envelope A	12400	SF
overall U	0.114	Btuh/SF-degF
Overall U*total A	1410	Btuh/degF
	OK	

F-slab (no insul)	0.73	Btuh/LF-degF
sum (F-P)	**256**	Btuh/degF

Q-infiltration	**22320**	cfh
@ 0.03cfm/SF of envelope		
Q-ventilation	**600**	cfm
@ 250 sf and 20 cfm/person		
BLC	**2727**	Btuh/degF

Internal Load

Lights	1.5	W/SF
Diversity	75%	
Lights Heat	28797	Btuh
Office equip	1.5	W/SF
Diversity	50%	
Ofc equip heat	19198	
Misc equip	0.25	W/SF
Diversity	50%	
Misc equip heat	937.5	
People	30	@250SF/person
Diversity	70%	
People Heat	5250	@250 Btuh sensible
Total Internal	**54183**	Btuh

Find Balance Temp
For some dT (inside - outside),
(BLC * dT) = internal gain, so
dT = Internal gain / BLC
Balance temp=(indoor temp (internal gain/BLC))

Internal Gain	54183
BLC	2727
dT	19.9
Inside temp	70
Outside temp at Balance	**50.1**

Figure 17-1A. Finding Balance Temperature when BLC is Known – Calculated Solution

Outside Air degF	Internal degF	dT degF	Internal Gain	BLC 2727 Loss BLC * dT	
65	**70**	5	54183	13636	
64	70	6	54183	16364	
63	70	7	54183	19091	
62	70	8	54183	21818	
61	70	9	54183	24545	
60	70	10	54183	27273	
59	70	11	54183	30000	
58	70	12	54183	32727	
57	70	13	54183	35454	
56	70	14	54183	38182	
55	70	15	54183	40909	
54	70	16	54183	43636	
53	70	17	54183	46363	
52	70	18	54183	49091	
51	70	19	54183	51818	
50	70	20	54183	54545	Balance
49	70	21	54183	57272	
48	70	22	54183	60000	
47	70	23	54183	62727	
46	70	24	54183	65454	
45	70	25	54183	68182	
44	70	26	54183	70909	
43	70	27	54183	73636	
42	70	28	54183	76363	
41	70	29	54183	79091	
40	70	30	54183	81818	

Figure 17-1B. Finding Balance Temperature when BLC is Known – Trial and Error Solution

R-VALUE REDUCTION FROM STUD WALLS

Table 17-1. Parallel Path Correction Factors
(reduction in overall R-Value of the wall)

Size of members	Framing	Insulation R-value	Metal Stud Correction Factor	Wood Stud Correction Factor
2x4	16 in. O.C.	R-11	0.50	0.76
2x4	24 in. O.C.	R-11	0.60	0.80
2x6	16 in. O.C.	R-19	0.40	0.60
2x6	24 in. O.C.	R-19	0.45	0.66

Source: *Energy Management Handbook* 7th Ed, The Fairmont Press.

Table 17-2. Effective Insulation De-Rate Effect from Stud Walls

Size of members	Framing	Metal stud example before and after R-value	Wood stud example before and after R-value
2x4	16 in. O.C.	R-11/R-5.5	R-11/R-8.4
2x4	24 in. O.C.	R-11/R-6.6	R-11/R-8.8
2x6	16 in. O.C.	R-19/R-7.6	R-19/R-11.4
2x6	24 in. O.C.	R-19/R-8.6	R-19/R-12.5

Source: Author Calculations derived from *Energy Management Handbook* 6th Ed, Turner/Doty

GLAZING PROPERTIES

Thermal

- Glass by itself has meager insulating properties. Increased thermal insulation is achieved by adding layers and pockets of trapped air or inert gas. Common center of glass U-values:

Table 17-3.

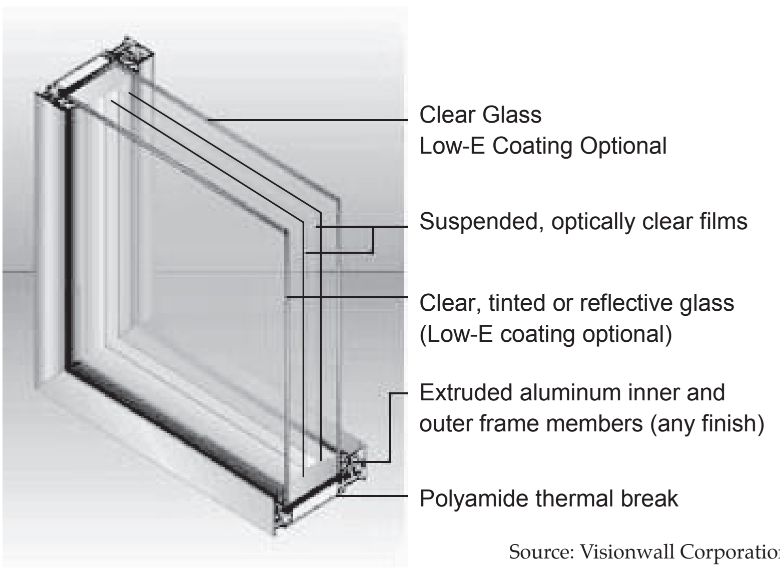
U-Value	# of Panes
1.1	single pane
0.5	standard double pane
0.3	standard triple pane (note diminishing return)

- The addition of coatings and different gases influences the overall U-value, as does the frame construction (metal, wood, vinyl, with-without thermal breaks).
- The effect of the frame on thermal performance should not be overlooked. A plain double pane window with a wood or vinyl frame has similar performance to that of Low-E coating (hard coating) with Argon gas fill and a metal frame.

  Source: *Energy Management Handbook*, 7th ed, The Fairmont Press.

High Performance Glazing

- High performance glazing systems can sometimes eliminate perimeter heating systems with equipment savings that help pay for the glazing. For ***example***, glazing systems are available and can provide an inside surface temperature of 55 deg F or higher at (–10) deg F outside temperature. The high R-values are achieved by multiple layers, each one adding a trapped air (or gas) space. These can be multi-layered glazing units or "suspended film" units.



Source: Visionwall Corporation

Figure 17-2.
The above cutaway diagram is an ***example*** of a proprietary high performance glass unit manufactured by Visionwall Corporation which achieves a combined glass/frame insulating value in excess of R-7. The increased R-value is achieved by multiple layers of glass and/or film. Note the thermal break in the frame as well.

Shading Coefficient (SC) and Solar Heat Gain Coefficient (SHGC)

Solar heating energy let through of glass elements is important for estimating heat gain through glass, beneficial or air conditioning load. The original measure of this was ***shading coefficient*** (SC).

The modern variation of this is ***solar heat gain coefficient*** (SHGC) which includes the frame as well as glass. The base units are also slightly different. Single pane clear glass has a SC rating of 1, and all tints are referenced as a percentage of this, e.g. a tint with a 0.6 SC will let through 60% as much sun heating as clear glass. The SHGC system is based on 1.0 being all available sun heat, with clear glass being less than 1, since it is not completely clear.

Both units are still in use, although the SHGC system will eventually replace SC.

Converting between the two units is approximated here:

SHGF = SC * 0.87

Glass Solar Performance Values

- Glass performance is rated by the 'shading coefficient' (SC) and 'solar heat gain factor' (SHGF). Values vary slightly by manufacturer.

Table 17-4.Some Common Values of Clear and Modified Glass Solar Performance Values.

SC	SHGF	Treatment
---	1.0	100% of light passes through (theoretical)
1	0.87	clear glass, single pane
0.81-0.86	0.70-0.74	clear glass, double pane
0.61-0.81	0.53-0.70	clear glass, single pane, Low E
0.53-0.57	0.45-0.49	light tint
0.35-45	0.30-0.39	Frittered glass, 1/8 inch dots or holes >60% opaque
0.39-0.46	0.34-0.40	heavy tint
0.23-0.40	0.20-0.34	reflective
0.32-0.52	0.28-0.45	Low E coating, clear
0.24-0.31	0.21-0.27	Low E(2) coating, clear

Low E Coatings

Low E (low emissivity) coatings are made from thin layers of metal deposited on one or more surfaces of glass. The result is a partial reflection of infrared waves and a reduction in the transmission of heat between layers; thus improving the SC (SHGF) values of all glass and U-values of multi-pane glass. LowE^2 (squared) glass is a thicker layer of metal deposit. For low-E coatings, the soft coating is superior in performance, but is subject to abrasion and so soft LowE is not applied to the outer surfaces. For multi-layered glazing, soft coating on an internal and protected layer is effective.

- Different types of low E coatings are advantageous in different climates, e.g. "southern Low E" would be designed to keep more heat out, compared to other types that would be designed to keep heat in (cold climates).

- Different locations of the low E coatings help keep heat out or keep heat in. This is specified on the particular surface the coating is to be applied. The coating applied to the warmer inboard glass panel of a double pane glass keeps radiant energy in during cold weather better than if applied on the colder outer pane—the reverse being true for summer and hot climates.

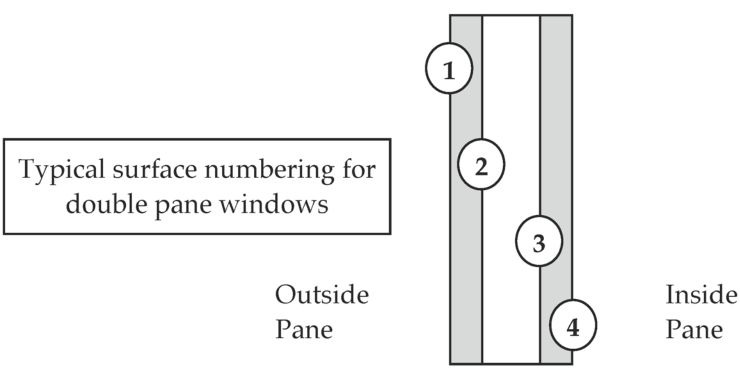
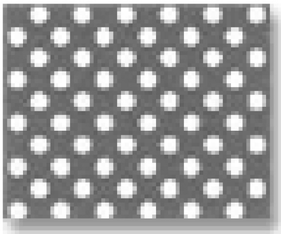


Figure 17-3. Window Surface Numbering

Silk Screened Glass

Silk screen shaded glass is made with dot patterns provide shading. Automobile windshields, near the edges, are a ready ***example*** of this technology. The fabric is embedded in the glass by the manufacturer. The percentage value is the opacity.



Window Films

Modern adhesive window coatings or solar film coatings can repel much of the solar heat gain that otherwise comes through the glass and heats the interior contents. Unlike old heavy tints, modern high performance coatings can repel more than 50% of solar heat with a minimal amount of visible light loss.

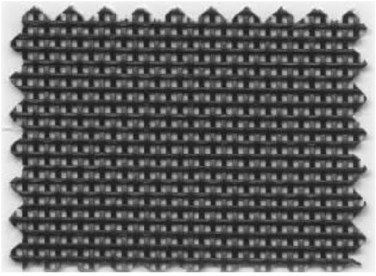
Solar films applied to the inside surface of existing glass can be cost effective in some cases. A good glazing will reduce solar load by 50% with 75% visible light transmittance, and will not become a heat sink for the reflected energy. These do not have to look like mirrored sunglasses. Spectrally selective coatings target the heat-producing wavelengths to repel. Design life is less than new windows. Films usually require re-application about every 10-15 years.

Exterior Shading

- Summer maximum cooling loads are often dictated by solar load, so reducing this directly reduces equipment size and cost. Also, occupants in the direct path of incoming sunlight will feel too warm despite the surrounding air temperature, and will lower the thermostat to compensate, adding further energy consumption.

- Fixed exterior shading elements can sometimes be used to allow solar infiltration in cold months to reduce heating. Fixed elements require analysis of sun approach angles in different seasons.

- Shading methods vary, but they share a common goal which is to keep the sun's heat out of the building. Interior shades provide some relief to occupants, but much of the heat ends up as air conditioning load since it is already inside. For ***example***, indoor blinds have been measured at 100 degrees F during 30 degree F weather, and causing air conditioning to run. For this reason, exterior shades are more effective.



- Exterior screening can be applied to certain existing glass or to skylights. These screens look like screen door material, but come in different "percent free area" patterns. A "30% FA" screen pattern blocks 70% of the light, and allows 30% through. These screens are aggressive at providing shade while allowing some visible light to come through, but are not transparent. Attachment and support of these screens is a design challenge.

Envelope Tradeoffs with Glazing

There is a balance of heat gain that is sometimes beneficial and sometimes not. See **Chapter 24, Envelope Tradeoffs—Light Harvesting, Window Tinting.**

INFILTRATION

- This is unintended outside air coming into the building due to differential pressures (mechanical, wind, and stack effect) and openings in the envelope. Energy implication is the heating and cooling energy needed to temper it once inside. In humid climates, the load of dehumidifying the extra outside air is added. Units of infiltration can include CFM, CFM/SF of envelope wall, or air changes per hour (ACH).
- When infiltration is excessive, there are usually complaints of cold drafts in winter. The best time to locate points of infiltration is during cold weather, by the use of an infrared thermometer around various exterior points in the building. Return air plenums are especially problematic if not sealed, since they operate in a slight negative pressure anyway. In extreme cases, frozen pipes in return plenums are the result of infiltration.

- Calculating infiltration is difficult and even the best formulas available rely on subjective data. A widely used relationship, called the crack method, is published in the ASHRAE Fundamentals Handbook and includes a variety of tables and factors, and is not repeated here. The only way to know for sure is by leakage testing which is seldom done due to complexity and cost.
- Infiltration values can vary by as much as a factor of 10.
- The only way to accurately quantify infiltration is with a blower test which is a major undertaking for a large building. Any new building advertising best-in-class efficiency should strongly consider this.
- A handy rule of thumb for commercial buildings for overall infiltration levels (cfm per SF of wall area) is:

Table 17-5. Approximate Infiltration Rates by Construction Quality

CFM/SF of Wall Area	Construction Quality
0.10	Tight
0.30	Average
0.60	Leaky

Table 17-6. Typical Building Element Infiltration Values

Approximate Baseline (Typical) Values @ 0.30 in. w.c. for individual elements

Element	CFM/SF
Roof	0.12
Walls	0.12
Doors (Opaque)	0.40
Loading Dock Door	0.40
Revolving Door	1.00
Swinging Glass Door	1.00
Sliding Glass Door	0.40
Windows	0.40

Source: *Infiltration Modeling Guidelines for Commercial Building Energy Analysis*, DOE, 2009
From ASHRAE SSPC 90.1 Envelope Subcommittee

Wall Construction Leakage

Median values at 0.30 in. w.c. (75Pa)
For surface area of the wall.

Conversion: M3H/M2 * 0.0547 = CFM/SF
(cubic meters per hour per square meter to
cubic feet per minute per square foot)

Wall Construction Type	CFM/SF	M3H/M2
Masonry	1.5	27
Concrete Panel	1.4	25
Frame/Masonry	3.2	58
Frame	3.0	55
Metal	1.6	30
Curtain	1.0	18

Source: *Air Tightness of Commercial and Institutional Buildings: Blowing Holes in the Myth of Tight Buildings,* Persily, A.K., 1998

Figure 17-4. Typical Infiltration Values by Wall Construction Type

AIR FLOW CREATED FROM BUILDING STACK EFFECT

Stack Effect - Cubic Feet Per Minute (CFM)
Air Loss from Vertical Height Differential

Air flow potential for equal size low/high openings when flow resistance is minimal.

$$Q = A * 60 * Cd * \sqrt{[2g * H * (Ti - Te)/Ti]}$$

Where:

Q=Cubic Feet Per Minute (CFM)

A=opening size, square feet

- For separate openings, use the average opening size.
- For evaluating chimney effect on a building of cracks, use the area of the chimney element.

CD=orifice coefficient, 0.65 for separate openings or single top opening.

H=height difference, ft. between openings

- For separate openings, use the distance between the center of the openings.
- For evaluating chimney effect on a building of cracks use the top of the chimney outlet to the center of the prevalent intake point (doors or windows).

g=32.2 ft/sec^2

Ti = internal temperature, degR (F+460)

Te = external temperature, degR (F+460)

Relationships:

Q is proportional to change in area

Q is proportional to the square root of the change in height

Q is proportional to the square root of the dT*

Double the area, double the airflow

Double the height, 141% of the airflow

Double the differential temperature, 141% of the airflow*

Half the area, half the airflow

Half the height, 70.7% of the airflow

Half the dT, 70.7% of the airflow*

In all cases, blocking the path (sealing) stops the stack effect air flow.

*(<5% error in the range of 0-100 degF)

Example: An abandoned chimney in a building measures 2′x2′ opening and the damper is open, so air flow is mostly free. The top of the chimney is 30 feet above the center of the doors, where a draft is felt, plus drafts at operable windows at the same level. Average winter temperature is 30 degF and indoor temperature is kept at 70 degF.

A = 4 SF

H=30 ft

(Ti-Te)/Ti = (70-30)/(70+460) = 0.0755

Q=4*60*0.65* SQRT[64.4*30*0.0755] = 1884 CFM (maximum)

STACK EFFECT - 10 FEET

$Q = A * 60 * Cd * \sqrt{[2g * H * (Ti - Te)/Ti]}$

CFM per SF of 'chimney' top opening

Uni-directional flow, low resistance to air movement

A: 1

CD: 0.65

H: 10

Inside temp, Ti, degF

Outside Temp, Te, degF	50	55	60	65	70	75
-10	339	352	363	374	385	394
-5	325	338	350	361	372	383
0	310	323	336	348	360	371
5	294	308	322	335	347	358
10	277	293	307	320	333	345
15	259	276	291	305	319	331
20	240	258	274	290	304	317
25	219	239	257	273	288	303
30	196	218	238	256	272	287
35	170	195	217	237	254	271
40	139	169	194	216	235	253
45	98	138	168	193	215	234
50		98	137	167	192	214
55			97	137	167	191
60				97	136	166

Figure 17-5. Stack Effect Air Flow Potential at 10,20,30 Ft. Height

STACK EFFECT - 20 FEET

$Q = A * 60 * Cd * \sqrt{[2g * H * (Ti - Te)/Ti]}$

CFM per SF of 'chimney' top opening

Uni-directional flow, low resistance to air movement

A: 1

CD: 0.65

H: 20

Inside temp, Ti, degF

Outside Temp, Te, degF	50	55	60	65	70	75
-10	480	497	514	529	544	558
-5	460	478	495	511	527	541
0	438	457	475	492	509	524
5	416	436	455	473	490	506
10	392	414	434	453	471	488
15	367	390	412	432	451	469
20	339	365	388	410	430	449
25	310	338	363	386	408	428
30	277	308	336	361	385	406
35	240	276	307	335	360	383
40	196	239	274	305	333	358
45	139	195	238	273	304	331
50		138	194	237	272	303
55			137	193	235	271
60				137	192	234

Figure 17-6.

STACK EFFECT - 30 FEET

$Q = A * 60 * Cd * \sqrt{[2g * H * (Ti - Te)/Ti]}$

CFM per SF of 'chimney' top opening

Uni-directional flow, low resistance to air movement

A: 1

CD: 0.65

H: 30

Inside temp, Ti, degF

Outside Temp, Te, degF	50	55	60	65	70	75
-10	588	609	629	648	666	683
-5	563	585	606	626	645	663
0	537	560	582	603	623	642
5	509	534	558	580	600	620
10	480	507	532	555	577	598
15	449	478	504	529	552	574
20	416	447	475	502	527	550
25	380	414	445	473	500	524
30	339	378	412	443	471	497
35	294	338	376	410	441	469
40	240	293	336	374	408	438
45	170	239	291	335	372	406
50		169	238	290	333	371
55			168	237	288	331
60				167	235	287

Figure 17-7.

AIR FLOW THROUGH OPEN DOCK DOORS

Special case of stack effect, where there is a single vertical opening.

Flow is bi-directional, cooler air entering the lower portion to replace the warm air lost from the top portion.

Potential air flow leaving the building is only sustainable if the interior space can maintain the indoor temperature. If the area near the opening gets colder, the stack effect and air loss become less due to the reduction in differential temperature is reduced and the airflow tapers off; the period in which this happens is indeterminate. In a large building open to the dock door area there is a vast supply of heated air to 'feed' the air loss the air flow out the open door could in which case the air flow could be sustained as long as the heated air supply lasted. By contrast, a small area limited to the vicinity of the dock doors would cool off quickly and air flow would eventually become zero when the indoor temperature and outdoor temperature were the same. For this reason, sectioning off the shipping/receiving areas from the balance of the building is helpful in controlling dock door losses.

Initial flow potential of an open dock door can be roughly approximated by assuming the single opening is effectively an upper and lower opening with a neutral area in the middle, and using the with the standard stack effect equation with:

- Effective 'chimney' area as 40% of the actual opening area
- Effective height differential as 50% of the actual opening height
- Coefficient Cd for bi-directional flow as Cd=0.4+(0.0025* | Ti-To |)

Note: Estimated losses are for convective air movement. Indoor/outdoor pressure differential and wind action are not considered. For doors left open long-term, use reduced indoor temperature to avoid over-stating air and heat loss.

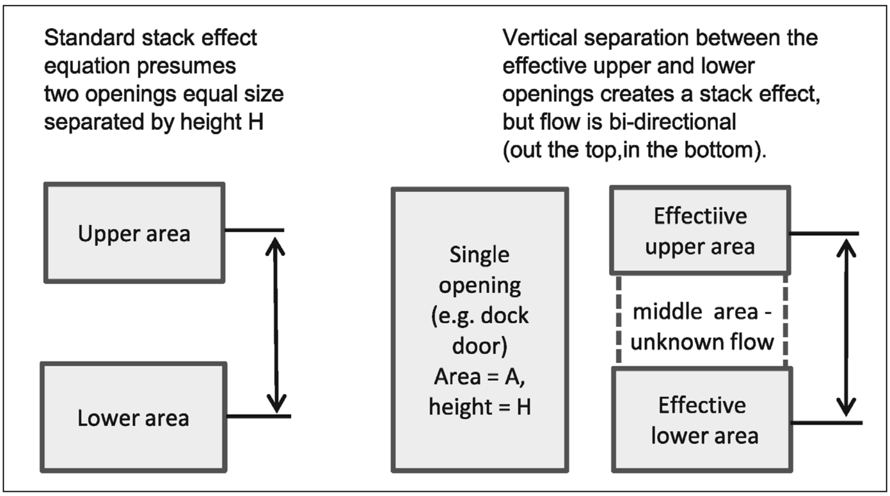


Figure 17-8. Open Dock Door Air Flow Model

Example: An 8'x10' dock door is left opened during heating season when it is 20 degF outside and 60 degF inside. Warm air is lost from the same level of the building. After a half hour, the indoor temperature has dropped to 40 degF.

Effective area: 80SF * 0.4 = 32SF
Effective height: 10 ft * 0.5 = 5 ft

<u>Initial air flow (60 degF inside):</u>
Cd = 0.50

(Ti-Te)/Ti = (60-20)/(60+460) = 0.077
Q=32*60*0.50* SQRT[64.4*5*0.077] = 4780 CFM (maximum)

<u>Air flow (40 degF inside):</u>
Cd = 0.45
(Ti-Te)/Ti = (40-20)/(40+460) = 0.040
Q=32*60*0.45* SQRT[64.4*5*0.040] = 3100 CFM

<u>Final Air flow (20 degF inside):</u>
0 CFM

Approximate Initial Air Flow Through Dock Door Opening

DOCK DOOR - 9 FT X 8 FT HIGH

$$Q = A * 60 * Cd * \sqrt{[2g * H * (Ti - Te)/Ti]}$$

CFM initial infiltration (will diminish quickly as indoor space cools off)

Uni-directional flow, low resistance to air movement

Convective air movement only, wind action not included

		factor
enter	A	0.4
actual	**72**	28.8
area,SF		area used

CD=0.4+(0.0025 * |Ti-To|)

		factor
enter	H	0.5
actual	**8**	4
height, ft		height used

Inside temp, Ti, degF

Outside Temp, Te, degF	**50**	**55**	**60**	**65**	**70**	**75**
-10	5232	5542	5851	6159	6465	6771
-5	4895	5207	5516	5823	6129	6435
0	4559	4872	5181	5489	5796	6101
5	4222	4537	4848	5157	5463	5768
10	3884	4202	4515	4825	5132	5438
15	3542	3865	4181	4493	4802	5108
20	3195	3525	3846	4161	4472	4780
25	2840	3180	3508	3828	4142	4451
30	2471	2826	3164	3491	3810	4122
35	2081	2459	2813	3149	3474	3792
40	1651	2071	2448	2799	3134	3458
45	1133	1642	2061	2436	2786	3120
50		1127	1635	2051	2424	2773
55			1122	1627	2041	2413
60				1116	1619	2032

Figure 17-8A. Dock door initial air loss (8 ft.)

Approximate Initial Air Flow Through Dock Door Opening

DOCK DOOR - 9 FT X 9 FT HIGH

$$Q = A * 60 * Cd * \sqrt{[2g * H * (Ti - Te)/Ti]}$$

CFM initial infiltration (will diminish quickly as indoor space cools off)

Uni-directional flow, low resistance to air movement

Convective air movement only, wind action not included

		factor
enter	A	0.4
actual	**81**	32.4
area,SF		area used

CD=0.4+(0.0025 * |Ti-To|)

		factor
enter	H	0.5
actual	**9**	4.5
height, ft		height used

Inside temp, Ti, degF

Outside Temp, Te, degF	**50**	**55**	**60**	**65**	**70**	**75**
-10	6243	6613	6982	7349	7714	8079
-5	5841	6213	6581	6948	7314	7678
0	5440	5813	6183	6550	6916	7280
5	5038	5414	5785	6153	6519	6883
10	4634	5014	5388	5757	6124	6489
15	4226	4611	4989	5362	5730	6095
20	3813	4206	4589	4966	5336	5703
25	3389	3794	4186	4567	4942	5311
30	2949	3372	3776	4166	4546	4919
35	2483	2935	3356	3758	4146	4524
40	1969	2471	2921	3340	3740	4126
45	1352	1960	2459	2907	3324	3722
50		1345	1950	2447	2893	3309
55			1339	1941	2436	2879
60				1332	1932	2424

Figure 17-8A. Dock door initial air loss (9 ft.)

DOCK DOOR - 9 FT X 10 FT HIGH

$Q = A * 60 * Cd * \sqrt{[2g * H * (Ti - Te)/Ti]}$

CFM initial infiltration (will diminish quickly as indoor space cools off)

Uni-directional flow, low resistance to air movement

Convective air movement only, wind action not included

		factor
enter	A	0.4
actual	**90**	36
area,SF		area used

CD=0.4+(0.0025 * |Ti-To|)

		factor
enter	H	0.5
actual	**10**	5
height, ft		height used

Inside temp, Ti, degF

Outside Temp, Te, degF	50	55	60	65	70	75
-10	7312	7746	8177	8607	9035	9463
-5	6842	7276	7708	8138	8566	8993
0	6371	6808	7241	7672	8100	8526
5	5901	6340	6775	7207	7635	8062
10	5427	5872	6310	6743	7173	7599
15	4950	5401	5844	6280	6711	7139
20	4465	4926	5375	5816	6250	6680
25	3969	4444	4902	5349	5788	6221
30	3454	3950	4422	4879	5324	5761
35	2908	3437	3931	4401	4856	5299
40	2307	2894	3421	3912	4380	4833
45	1583	2295	2880	3404	3893	4360
50		1575	2284	2866	3388	3875
55			1568	2273	2853	3372
60				1560	2263	2839

Figure 17-8A. Dock door initial air loss (10 ft.)

DOOR INFILTRATION RATES

Approximate Door Infiltration CFM		Traffic Rate Total Entrances and Exits per Door per Hour						
		5	**10**	**15**	**25**	**50**	**100**	**200**
Single Swinging Door (21 SF) No Vestibule	Htg	30	59	89	149	297	594	1189
	Clg	15	29	44	74	147	295	590
Single Swinging Door (21 SF) With Vestibule	Htg	Use no-vestibule values if trafic rate is less than 101						637
	Clg							316
Double Swinging Door (42 SF) No Vestibule	Htg	59	119	178	297	594	1189	2377
	Clg	29	59	88	147	295	590	1179
Double Swinging Door (42 SF) With Vestibule	Htg	Use no-vestibule values if trafic rate is less than 101						1274
	Clg							632
Single Automatic Door (21SF) No Vestibule	Htg	80	161	241	402	803	1607	3213
	Clg	40	80	119	199	398	796	1592
Single Automatic Door (21SF) With Vestibule	Htg	54	107	161	268	356	1071	2142
	Clg	27	53	80	133	266	631	1063
Double Automatic Door (42SF) No Vestibule	Htg	161	321	482	803	1607	3213	6426
	Clg	80	159	239	398	796	1592	3184
Double Automatic Door (42SF) With Vestibule	Htg	107	214	321	536	1071	2142	4284
	Clg	53	106	159	266	531	1063	2125
Manual Revolving Door	Htg	4	9	13	21	43	85	170
	Clg	4	9	13	21	43	85	170
Motorized Revolving Door	Htg	From 0-2 rpm, cfm=200*rpm.						
	Clg	Above 2 rpm, cfm = 400						

Figure 17-9.

Source: Commercial Load Calculation Manual N, 5th ed, Copyright ACCA, Arlington, VA Table 5F, Door Traffic Infiltration for Low Rise Commercial Buildings

Traffic Rate (TR)

- The traffic rate shall be evaluated for the hour of day used for the load estimate
- The traffic rate is zero (or close to zero) for non-business hours (night time, weekends, facilities closed to normal operations, etc.)
- For relatively steady visits: TR=(2*Avg # of visits per day) / (# of doors * duration of business day in hours)
- For a 2-4 hour visitor rush: TR = (2*Avg # of visits per rush) / (# of doors * duration of rush in hours)
- For a 1-hour visitor rush: TR=0.5*(2*Avg# of visits per rush) / # of doors
- For a 1-hour employee lunch rush: TR=0.5*(2*Avg # of employees) / # of doors
- For a scheduled employee arrival-departure rush: TR=0.5*(Avg # of employees) / # of doors

Note the effect of vestibules or revolving door for high traffic entryways. The meager effect of a vestibule is probably due to the reality that many vestibules do not act as air locks (first door should close before the next one opens); very deep vestibules that act more like an air lock would perform better.

Example: A church has 2000 visitors during a five hour period, and has conventional double doors with no vestibules. Find the winter benefit of installing a vestibule or revolving door . Average temperatures are 70 degF indoors and 30 degF outdoors, heating efficiency is 80%e, and fuel cost is $1.00 per therm. Winter is 5 months long (105 hours total), and the evaluation is only for the Sunday services.

This is a total of 4000 door openings, including entry and exit. 4000 events over five hours is 800 events per hour. There are four sets of double doors, so 200 events per hour each.

Standard double door air rate: 2377 cfm per door * 4 doors = 9508 cfm
ECM - Add a vestibule: 1274 cfm per door * 4 doors = 5096 cfm (4412 cfm saved)
ECM – Add a motorized revolving door: 400 cfm per door * 4 doors= 1600 cfm (7908 cfm saved)

Heat savings: 1.08 * cfm * dT. Or, for 40 degF dT, heat savings = 43.2 * cfm

ECM – Add a vestibule: 43.2 * 4412 cfm * 105 hours * (1/0.8) = 250 therms
ECM – Add a revolving door: 43.2 * 7908 cfm * 105 hours * (1/0.8) = 448 therms

COMPOSITE U-VALUES FOR ENVELOPE EVALUATION

A mixture of U-values can be combined into an overall U-Value using weighting, and will provide an overall effective U-Value for the composite. This works for a wall (with glass), a roof (with skylights), or a whole building envelope. The overall building envelope U-value is determined in this way to verify energy code compliance.

Example using measured values of SF:
A building has 100,000 SF of overall wall area (U-0.1), of which 40,000 SF is glass (U-0.4).
Overall U = [(40,000 * 0.4)+(60,000*0.1)] / 100,000 = 0.22 or R-4.5

Same ***example***, using percentages:
A building wall has 40% glass (U-0.4), with the rest of the wall being U-0.1.
Overall U = (0.4*0.4)+(0.6*0.1) = 0.22, or R-4.5

Example using percent of total area:
A roof has insulation with R-30 (U-0.033) and 10% skylights with U-0.7
Overall U = (0.9*0.033) + (0.1*0.7) = 0.0997, or R-10

Percent Glazing Effect on Overall Gross Wall Insulation U-Value
Example use of chart: **(Figure 17-10)**
Wall insulation is R-10 and has 40% glazing, 0.45Udouble pane glass.
Overall wall insulation effect is U 0.24 or R-4.2

PERCENT SKYLIGHT/CLERESTORY EFFECT ON OVERALL GROSS ROOF INSULATION U-VALUE

Example use of chart: **(Figure 17-11)**
Roof insulation is R-40 and has 10% skylight glazing, 0.45U
Overall wall insulation effect is U 0.068 or R-14.7

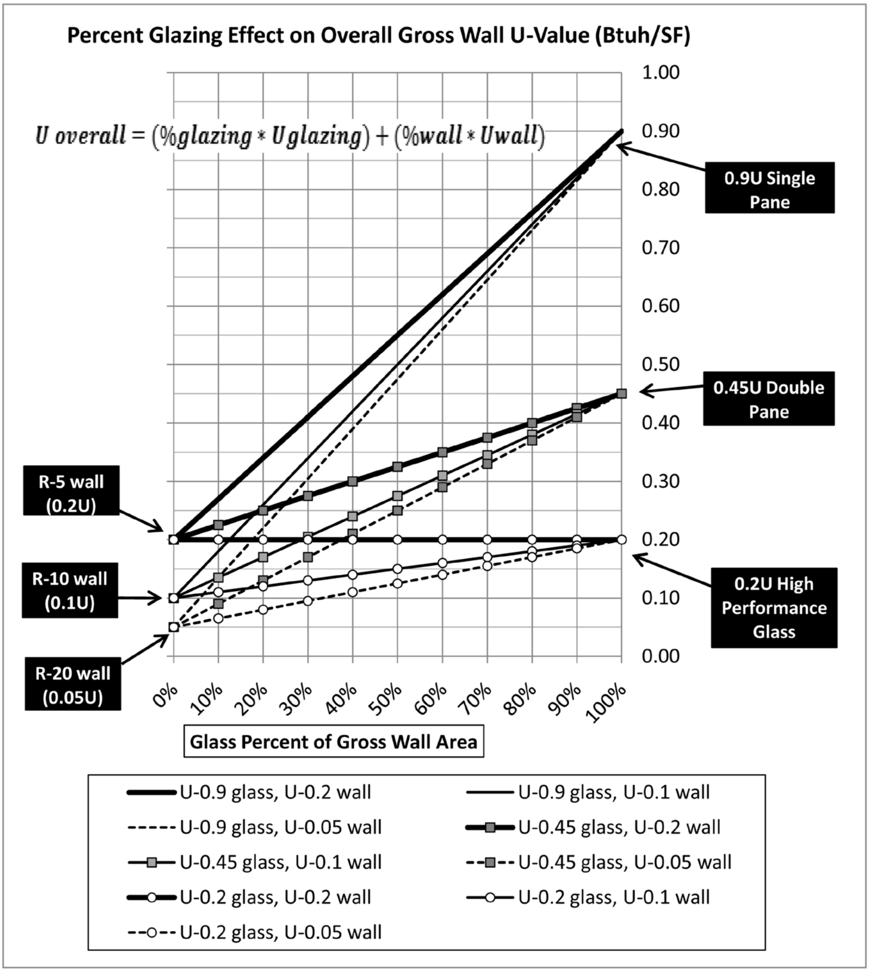
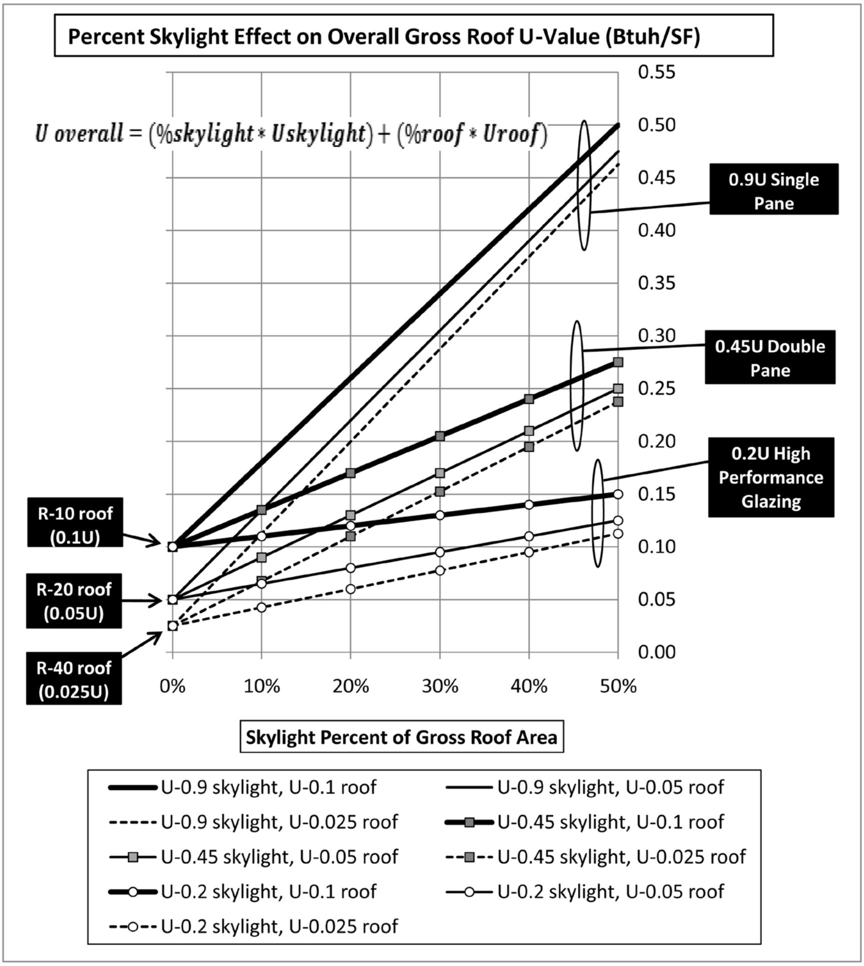

Percent Glazing Effect on Overall Gross Wall U-Value (Btuh/SF)
U overall = (%glazing * Uglazing) + (%wall * Uwall)
1.00
0.90
0.80
0.70
0.60
0.50
0.40
0.30
0.20
0.10
0.00
0.9U Single Pane
0.45U Double Pane
0.2U High Performance Glass
R-5 wall (0.2U)
R-10 wall (0.1U)
R-20 wall (0.05U)
0%
10%
20%
30%
40%
50%
60%
70%
80%
90%
100%
Glass Percent of Gross Wall Area
U-0.9 glass, U-0.2 wall
U-0.9 glass, U-0.1 wall
U-0.9 glass, U-0.05 wall
U-0.45 glass, U-0.2 wall
U-0.45 glass, U-0.1 wall
U-0.45 glass, U-0.05 wall
U-0.2 glass, U-0.2 wall
U-0.2 glass, U-0.1 wall
U-0.2 glass, U-0.05 wall

Figure 17-10.



Percent Skylight Effect on Overall Gross Roof U-Value (Btuh/SF)
U overall = (%skylight * Uskylight) + (%roof * Uroof)
0.55
0.50
0.45
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0.00
0%
10%
20%
30%
40%
50%
0.9U Single Pane
0.45U Double Pane
0.2U High Performance Glazing
R-10 roof (0.1U)
R-20 roof (0.05U)
R-40 roof (0.025U)
Skylight Percent of Gross Roof Area
U-0.9 skylight, U-0.1 roof
U-0.9 skylight, U-0.05 roof
U-0.9 skylight, U-0.025 roof
U-0.45 skylight, U-0.1 roof
U-0.45 skylight, U-0.05 roof
U-0.45 skylight, U-0.025 roof
U-0.2 skylight, U-0.1 roof
U-0.2 skylight, U-0.05 roof
U-0.2 skylight, U-0.025 roof

Figure 17-11.

Chapter 18

Domestic Water Heating

See also **Chapter 22 Water Efficiency**

DOMESTIC WATER HEATERS

Gas or Electric

In almost all locations, natural gas is less expensive to use for heating purposes.

Gas heating efficiencies vary. Plain gas heaters have a nominal 80%e but quickly decline because the heat transfer surfaces are not designed to be cleaned and thus are never cleaned. 70-75% is more likely for standard commercial gas heaters.

Condensing gas heaters have excellent potential for efficiency since the water temperatures are generally lower than for heating water use, allowing the condensation to occur. 95% efficiency is easily obtainable through condensing for 120 deg F water.

Heat Pump Water Heaters

When gas is not available, or when there is a coincident cooling demand that is always there, heat pump water heaters can be effective. A COP of around 2.0 is expected at 140 degrees condensing, which means it costs half as much as electric resistance.

In locations where ambient temperatures get cold in winter season, the heat pump water heater would lose efficiency, since it needs a source of heat. Savings will be reduced during times when the heat pump water heater absorbs heat from an environment that is being heated by another system.

A manufacturer's rating of "percent standby loss" is no longer in use, but may still be found on older labeled heaters. For example, a water heater with a 360,000 Btuh heater rated for 3.33% standby loss per hour with a 60 gallon tank would be expected to lose 1200 Btuh per hour. Since this same 60 gallon tank could come with a variety of burners, the percentages also vary for different units using the same tank, and this statistical unit can be confusing.

DOMESTIC WATER HEATER STANDBY LOSSES

Storage Tanks for Domestic Hot Water Heaters

Storage tanks are used as thermal flywheels to accommodate temporary high loads that are in excess of average loads, thereby allowing a smaller burner or heater. The designers use a combination of heating capacity (gallons of hot water recovery rate) and tank size to meet the customer usage requirements. The tank has a standby loss from its surface area.

Any un-insulated tanks or bare sections can be justified by insulating them. Also, any un-insulated valves and piping reduces standby loss.

Estimating these losses can be set approximately equal to the heat loss of a cylinder or pipe of equal diameter and length, although this understates losses since there will be bare fittings and they will short circuit the insulation barrier. Loss is proportional to the differential temperature between inside (fluid) and outside (air), so lowering water temperature reduces standby loss.

One rule of thumb for tank loss is **6.5 Btuh/SF** for an 80 degree differential inside to outside.

Source: Table 404.1—Minimum Performance of Water Heating Equipment, DOE, 10 CFR Ch. II, 2005.

Domestic Hot Water Recirculation

Pumps and a return line are used to reduce the waiting time from opening a faucet to hot water delivery. Pump energy is small and the recirculation line is small. The benefit of a hot water recirculation ECM lies in the thermal losses.

With the pump on, the loss in Btu/h is continuous. With the pump off, the loss continues until the recirculation line is equal to ambient temperature, then stops, so turning off the circulating pump when not needed reduces the standby loss. The two common methods used to control the pump are an aquastat and timer.

An aquastat stops the pump when the water returning from the system is sufficiently high. Once it cools off, it starts again. Some users complain that this method short cycles their pumps.

A timer stops the pump during unoccupied hours, but lets it run continuously during occupied times.

Refer to tables in the Appendix "Heat Loss from Un-Insulated Hot Piping and Surfaces" and "Heat Loss from Insulated Piping" for heat losses of piping to help quantify potential savings.

Instantaneous Water Heaters

By generating hot water at the point of use, distribution losses for domestic hot water can be eliminated. In the case of a small point of use that is a long way away, this is very advantageous. A point of use water heater applied at a central location addresses tank losses but does nothing to address distribution piping losses which are usually the larger of the two. Also, this ECM eliminates the thermal flywheel design feature of the tank method, so consideration of larger heating capacity and infrastructure to support it become design questions.

Note: Instantaneous electric or gas demand can easily double when point of use heaters are applied compared to storage tank heaters.

Tank Water Heaters

Approximate Standby Loss, Btuh

burner	gallons	elec	gas/oil
15,000	20	177	511
30,000	20	177	529
30,000	40	241	733
60,000	40	241	771
100,000	40	241	821
100,000	75	323	1078
200,000	75	323	1203
100,000	100	370	1225
200,000	100	370	1350

Instantaneous Water Heaters

Approximate Standby Loss, Btuh

burner	gallons	elec	gas/oil
5,000	0.1	31	41
10,000	0.1	31	47
20,000	0.5	45	103
40,000	1	55	160
100,000	5	98	371
200,000	10	131	598

Basis: 20+(35*sqrtV) Electric
(Q/800)+(110SqrtV) Gas/Oil
V=gal storage, Q=nameplate input Btuh

Source: Water heater performance criteria, City of Houston, TX Commercial Energy Code. (based loosely on formula in ASHRAE 90.1-2004).

Approximate values if manufacturer data is not available.

Figure 18-1. Domestic HW standby loss

Chapter 19

Weather Data

DEGREE-DAYS

Units of degree-day weather data are: $(^{\text{degF-days}}/_{\text{period}})$

Period is usually months or years. City locations are often compared in tables of heating degree-days per year. More detailed weather data list degree-days by month.

One heating degree-day is one degree below the base temperature consistently for a full day.

One cooling degree-day is one degree above the base temperature for a full day.

Calculate average temperature from degree-days:

Base temp - $(^{\text{HDD}}/_{\text{no. of days in the period}})$

Base temp + $(^{\text{CDD}}/_{\text{no. of days in the period}})$

Example: if November has 600 degree-days (65 degree base), the average temperature would be 65- (600/30) = 45 degrees.

Cautions for Using Degree-days

1. The common 'base' number for heating degree-days (HDD) is 65 degrees, which indicates that at temperatures below 65 the building will likely begin to need heat. This is seldom a good assumption in commercial buildings, other than for hotel guest rooms and other residential occupancies, or where internal loads are very small and envelope loads dominate. Thus, accuracy of degree-day values in non-residential energy analysis requires using a balance temperature other than 65F. Additional information on Building Balance temperature: **Chapter 17—Envelope, BLC Applications, Chapter 11—Mechanical Systems, Thermal Balance Concept for Buildings.**
2. The definition of "degree-days" includes the average daily temperature, which is the (daily high—daily low)/2. So, two cities with similar daily average temperatures can have much different weath-

er. For **example**, if two cities have a high of 50 and a low of 10 they will have an "average" temperature of 30 degrees. But if one city has 16 hours at 10 degrees and one hour at 50 degrees and the other city has the reverse, their heating requirements are obviously much different. For this reason, using modeled heating data for one city and projecting it to another city (as if the city were picked up and moved) using degree-days is risky and not recommended. A better expression of weighted average weather data is "bin" data.

3. Where cooling degree-days (CDD) are published, they are usually with the same 65 degree base. Since most commercial buildings have significant internal loads with thermal balance temperatures much lower than 65, the use of CDD information will usually underestimate cooling loads. Note that ASHRAE 90.1 climatic data tables are based on CDD50.

For example:

Location	**CDD65**	**CDD50**
Atlanta	1,246	5,038
Denver	434	2,732

Source of CDD65: NOAA, 1949-2006 avg
Source of CDD50: ASHRAE 90.1 - 2001

4. Degree-days, in any event, are limited to conduction loss/gain and ventilation load loss/gain. Degree-days do not provide information to estimate internal loads or solar loads.
5. Weather data software can produce temperature data in terms of degree-days in specified base temperatures.

BIN WEATHER DATA

This method uses historical data, usually hourly, to express weather parameters such as dry bulb temperature, wet bulb temperature, etc.

Using dry bulb temperature as an **example**: for each hour that 70 degrees F is recorded at a weather station, the 70 degF 'bin' is incremented by one (1). So, if the 70 degF bin has a value of 100, this means there were 100 hours recorded at 70 degrees F during that period.

Bin data reports are very useful since they establish the proportions of operational time that will be expected for different ambient weather conditions. Accuracy of calculations is enhanced with such data, since ambient conditions also affect equipment loads and efficiencies, i.e. the corresponding equipment efficiency at 70 degrees (and during the 100 hours) will be unique at that temperature and different at other temperatures.

Additional Bin weather information:

Appendix: Bin weather for 5 cities.

Appendix: Hours per year below outside dry bulb and wet bulb temperature

Cautions for Using Bin Weather Data

1. While bins indicate the number of hours at a certain temperature, they do not indicate when this occurs.
2. Some of the hours may be transient. For example, 10 bin-hours at 35 degF wet bulb may be the combination of 8 hours all at once, and two days where the wet bulb was only at 35 for one hour as the weather changed from 35 to 40 degF wet bulb. If a process, such as a plate frame heat exchanger, is designed to take advantage of free cooling in dry weather but takes an hour or two to switch operating modes to the free cooling mode, this 'hour' is of no real use and including it overstates savings.
3. Some of the hours may be during unoccupied periods. To correct for this, use a weather data source that allows the user to select appropriate time periods, such as 6am-10pm instead of all 24 hours, for facilities that close at night, weekends, etc. Customizing the bins increases the accuracy of the conclusions drawn for them.

WEATHER DATA BY DAYS AND TIMES

This format is used for hourly computer analysis programs. It is also useful in quantifying savings when time-of-day information is available. For many ECM proposals, this method is the quickest.

For example, the cell for January 10am is the average temperature observed in this city for each day in January at that time. For 7 day per week operation, it can be said that the temperature shown for January 10am occurs 30 times (30 days per month).

This form of data representation can be useful for measures that are weather-dependent. For **example**, if a boiler is left idling all summer currently and a measure is proposed to turn it off above 60 degrees F, the number of hours it can be turned off can be easily determined. Referring to the second sample chart, the periods that are greater than 60 degrees F are highlighted. The number of boxes that are highlighted are counted—in this case there are 132, and then multiplied by 30 (days in a month). For this **example**, there are 3960 hours the boiler could be turned off.

Similar tables are available for wet bulb temperature.

Limitations: ECM proposals that consider more than one parameter (i.e. dry bulb and wet bulb temperature) are not conveniently used with this form of table. For **example**, to find the number of annual hours when it is below 55 degrees dry bulb and also below 35 degrees wet bulb is a manual process of comparing the charts cell-by-cell and would be better suited to bin data with coincident wet bulb.

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
0000	25.2	29.2	40.6	50.6	59.6	66.6	69.6	69.6	63.6	53.6	43.2	31.2
0100	23.9	27.9	39.3	49.3	58.3	65.3	68.3	68.3	62.3	52.3	41.9	29.9
0200	22.7	26.7	38.1	48.1	57.1	64.1	67.1	67.1	61.1	51.1	40.7	28.7
0300	21.7	25.7	37.1	47.1	56.1	63.1	66.1	66.1	60.1	50.1	39.7	27.7
0400	20.9	24.9	36.3	46.3	55.3	62.3	65.3	65.3	59.3	49.3	38.9	26.9
0500	20.7	24.7	36.1	46.1	55.1	62.1	65.1	65.1	59.1	49.1	38.7	26.7
0600	21.2	25.2	36.5	46.6	55.6	62.6	65.8	65.6	59.6	49.6	39.2	27.2
0700	22.4	26.4	37.8	47.8	55.8	63.8	66.8	66.8	60.8	50.8	40.4	28.4
0800	24.7	28.7	40.1	50.1	59.1	66.1	69.1	69.1	63.1	53.1	42.7	30.7
0900	27.9	31.9	43.3	53.3	62.3	69.3	72.3	72.3	66.3	56.3	45.9	33.9
1000	31.7	35.7	47.1	57.1	66.1	73.1	76.1	76.1	70.1	60.1	49.7	37.7
1100	35.9	39.9	51.3	61.3	70.3	77.3	80.3	80.3	74.3	64.3	53.9	41.9
1200	39.9	43.9	55.3	65.3	74.3	81.3	84.3	84.3	78.3	68.3	57.9	45.9
1300	42.9	46.9	58.3	68.3	77.3	84.3	87.3	87.3	81.3	71.3	60.9	48.9
1400	44.9	48.9	60.3	70.3	79.3	86.3	89.3	89.3	83.3	73.3	62.9	50.9
1500	45.6	49.6	61.0	71.0	80.0	87.0	90.0	90.0	84.0	74.0	63.6	51.6
1600	44.9	48.9	60.3	70.3	79.3	86.3	89.3	89.3	83.3	73.3	62.9	50.9
1700	43.1	47.1	58.5	68.5	77.5	84.5	87.5	87.5	81.5	71.5	61.1	49.1
1800	40.4	44.4	55.8	65.8	74.8	81.8	84.5	84.8	78.8	68.8	58.4	46.4
1900	37.1	41.1	52.5	62.5	71.5	78.5	81.5	81.5	75.5	65.5	55.1	43.1
2000	33.9	37.9	49.3	59.3	68.3	75.3	78.3	78.3	72.3	62.3	51.9	39.9
2100	31.2	35.2	46.6	56.6	65.6	72.5	75.8	75.6	69.6	59.6	49.2	37.2
2200	28.7	32.7	44.1	54.1	63.1	70.1	73.1	73.1	67.1	57.1	46.7	34.7

Figure 19-1. Sample Weather Data in Month and Hour Format

Source: "Hourly Analysis Program" (HAP), Carrier Corporation, 2003

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
0000	25.2	29.2	40.6	50.6	59.6	66.6	69.6	69.6	63.6	53.6	43.2	31.2
0100	23.9	27.9	39.3	49.3	58.3	65.3	68.3	68.3	62.3	52.3	41.9	29.9
0200	22.7	26.7	38.1	48.1	57.1	64.1	67.1	67.1	61.1	51.1	40.7	28.7
0300	21.7	25.7	37.1	47.1	56.1	63.1	66.1	66.1	60.1	50.1	39.7	27.7
0400	20.9	24.9	36.3	46.3	55.3	62.3	65.3	65.3	59.3	49.3	38.9	26.9
0500	20.7	24.7	36.1	46.1	55.1	62.1	65.1	65.1	59.1	49.1	38.7	26.7
0600	21.2	25.2	36.5	46.6	55.6	62.6	65.8	65.6	59.6	49.6	39.2	27.2
0700	22.4	26.4	37.8	47.8	55.8	63.8	66.8	66.8	60.8	50.8	40.4	28.4
0800	24.7	28.7	40.1	50.1	59.1	66.1	69.1	69.1	63.1	53.1	42.7	30.7
0900	27.9	31.9	43.3	53.3	62.3	69.3	72.3	72.3	66.3	56.3	45.9	33.9
1000	31.7	35.7	47.1	57.1	66.1	73.1	76.1	76.1	70.1	60.1	49.7	37.7
1100	35.9	39.9	51.3	61.3	70.3	77.3	80.3	80.3	74.3	64.3	53.9	41.9
1200	39.9	43.9	55.3	65.3	74.3	81.3	84.3	84.3	78.3	68.3	57.9	45.9
1300	42.9	46.9	58.3	68.3	77.3	84.3	87.3	87.3	81.3	71.3	60.9	48.9
1400	44.9	48.9	60.3	70.3	79.3	86.3	89.3	89.3	83.3	73.3	62.9	50.9
1500	45.6	49.6	61.0	71.0	80.0	87.0	90.0	90.0	84.0	74.0	63.6	51.6
1600	44.9	48.9	60.3	70.3	79.3	86.3	89.3	89.3	83.3	73.3	62.9	50.9
1700	43.1	47.1	58.5	68.5	77.5	84.5	87.5	87.5	81.5	71.5	61.1	49.1
1800	40.4	44.4	55.8	65.8	74.8	81.8	84.5	84.8	78.8	68.8	58.4	46.4
1900	37.1	41.1	52.5	62.5	71.5	78.5	81.5	81.5	75.5	65.5	55.1	43.1
2000	33.9	37.9	49.3	59.3	68.3	75.3	78.3	78.3	72.3	62.3	51.9	39.9
2100	31.2	35.2	46.6	56.6	65.6	72.5	75.8	75.6	69.6	59.6	49.2	37.2
2200	28.7	32.7	44.1	54.1	63.1	70.1	73.1	73.1	67.1	57.1	46.7	34.7
2300	26.7	30.7	42.1	52.1	61.1	68.1	71.1	71.1	65.1	55.1	44.7	32.7

Figure 19.2 Example use of Weather Data in Month and Hour Format

Example use of table: times when outside air is above 60 degrees.

Chapter 20

Pollution and Greenhouse Gases

Since energy use and greenhouse gas emission are directly linked the potential impact of the application of this book is very large.

POLLUTION—EMISSION CONVERSION FACTORS BY STATE
Use local factors if available. Representative data, changes over time.

CO_2 = Carbon Dioxide
SOx = Sulfur Oxides
NOx= Nitrogen Oxides

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Emission Factors by State

If local data is not available

units are lbs per MWh

* = Value is less than half of the smallest unit of measure . For values with no decimals the smallest unit is 1 and values under 0.5 are shown as *.
Source: DOE/EIA-0348(01)/2
Date of Release: March 2010. Date of Data: 2008

State	Primary Fuel Source	SOx (lbs/MWh)	NOx (lbs/MWh)	CO2 (lbs/MWh)
Alabama	Coal	5.5	1.7	1253
Alaska	Gas	1.3	4.9	1421
Arizona	Coal	0.8	1.3	1078
Arkansas	Coal	3.1	1.6	1220
California	Gas	0.0	0.9	663
Colorado	Coal	2.3	2.6	1711
Connecticut	Nuclear	0.3	0.5	684
Delaware	Coal	9.4	3.2	1931
District of Columbia	Petroleum	*	*	2134
Florida	Gas	2.7	1.7	1214
Georgia	Coal	8.7	1.9	1449
Hawaii	Petroleum	4.1	4.3	1753
Idaho	Hydroelectric	1.1	0.6	187
Illinois	Coal	3.8	1.3	1169
Indiana	Coal	9.4	3.3	2116
Iowa	Coal	6.4	2.9	1904
Kansas	Coal	4.1	2.3	1753
Kentucky	Coal	7.1	3.3	2116
Louisiana	Gas	2.2	1.8	1302

Figure 20-1.

State	Primary Fuel Source	SOX (lbs/MWh)	NOX (lbs/MWh)	CO2 (lbs/MWh)
Maine	Gas	3.4	1.4	685
Maryland	Coal	10.5	1.9	1356
Massachusetts	Gas	2.3	1.0	1154
Michigan	Coal	6.7	2.2	1478
Minnesota	Coal	3.5	2.8	1510
Mississippi	Gas	3.2	1.9	1184
Missouri	Coal	6.1	2.0	1869
Montana	Coal	1.6	2.1	1505
Nebraska	Coal	4.8	2.8	1519
Nevada	Gas	0.6	1.4	1139
New Hampshire	Nuclear	3.3	0.6	653
New Jersey	Nuclear	1.2	0.7	695
New Mexico	Coal	1.2	3.8	1827
New York	Gas	1.3	0.8	740
North Carolina	Coal	4.2	1.1	1325
North Dakota	Coal	8.4	4.2	2217
Ohio	Coal	10.4	3.2	1851
Oklahoma	Coal	2.8	2.3	1535
Oregon	Hydroelectric	0.5	0.5	405
Pennsylvania	Coal	7.7	1.8	1228
Rhode Island	Gas	*	0.9	892
South Carolina	Nuclear	3.4	1.0	928
South Dakota	Coal	4.0	4.0	1249
Tennessee	Coal	5.4	2.0	1423
Texas	Gas	2.5	1.2	1373
Utah	Coal	1.0	3.1	1862
Vermont	Nuclear	*	*	2
Virginia	Coal	4.3	1.7	1255
Washington	Hydroelectric	0.2	0.4	271
West Virginia	Coal	6.9	2.2	2044
Wisconsin	Coal	6.0	2.3	1713
Wyoming	Coal	3.9	3.5	2206
U.S. Total	Coal	4.2	1.8	1326

Figure 20-1. (Cont'd)

Table 20-1. Fossil Fuel Emissions

Units: Pounds per Billion Btu of Energy Input from Burning Fossil Fuels
Source: Energy Information Administration - Natural Gas Issues and Trends 1998, Chap 2, Table 2

Pollutant	Natural Gas	Oil	Coal
CO_2	117,000	164,000	208,000
CO	40	33	208
NO_x	92	448	457
SO_x	0.6	1,122	2,591
Particulates	7	84	2,744
Mercury	0.000	0.007	0.016

POLLUTION—CONVERSION TO EQUIVALENT NUMBER OF AUTOMOBILES

Sometimes it is effective to present pollution numbers in terms of automobiles removed from the road.

The numbers vary depending on vintage of the car, miles per gallon, miles per year, and fuel type. The standard for US EPA reporting follows.

Table 20-2. Fossil Fuel Emissions of Vehicles

Source: US EPA Office of Transportation and Air Quality, Feb 2005

Note:

Carbon dioxide, while not regulated as an emission, is the transportation sector's primary contribution to climate change. Carbon dioxide emissions are directly proportional to fuel economy--each 1% increase (decrease) in fuel consumption results in a corresponding 1% increase (decrease) in carbon dioxide emissions.

Average Vehicle	Total Annual CO_2
Passenger Car	5.03 metric tons of CO_2
	543 gallons of gasoline (22.1 mpg avg.)
Light Truck	6.32 metric tons of CO_2
	682 gallons of gasoline (17.6 mpg avg.)

OTHER ENVIRONMENTAL CONSIDERATIONS

Water-cooled Systems

See **Chapter 22—Water Efficiency**.

Fluorescent Lighting

A common stabilizing ingredient in fluorescent lights is mercury. Proper disposal is necessary to avoid landfill contamination. Widespread use of fluorescent lighting, especially compact fluorescents, introduces the potential for land and water pollution from improper disposal, which is a paradigm shift for residential customers who simply 'throw away' light bulbs that have failed.

Source Emission Amplification

It is common for energy savings to be expressed in <u>site</u> energy units, since that is most closely associated with the direct cost of the energy seen by the customer. For electricity produced by fossil fuel, <u>source</u> energy is higher than site energy, amplified by the conversion efficiency of the power plant where the electricity came from.

Source Energy = Site Energy / Conversion Efficiency

Another factor that makes source energy higher than site energy is the distribution loss that occurs between source and site. For electricity these include line losses and transformer losses; for natural gas, these include line losses and pumping energy used to transport the fuel. Similarly, fuel oil and coal have embedded energy related to of processing and transportation of the fuel.

> Example: A facility uses 1,000,000 kWh on site in one year, which is 3413 MMBtu/year (site). The electricity comes from a power plant with a 37% conversion efficiency and a 7% distribution loss. Source energy = 3413 / (0.37 * 0.93) = 9919 MMBtu source (ans)

Source energy is a better representation of the environmental impact of energy use, and the impact on available energy supplies. For example, compare electric resistance heating to natural gas heating – site energy will be nearly the same for each, but source energy for electric resistance heating is roughly 3x that of fossil fuel heating due to the conversion losses to make the electricity. For greenhouse gas accounting, the electricity conversion losses are called 'Scope 2 emissions' and are considered emissions by the user of the electricity. Renewable electricity sources have an improved business case over fossil fuel generation options where source energy is a metric of comparison or is monetized.

Chapter 21

Formulas and Conversions

EFFICIENCY

Efficiency = Output/Input
Output = Input x Efficiency
Input = Output/Efficiency

COP, EER, KW/TON

COP= cooling capacity/work input
COP= Btu out/Btu in or Work in
COP is unitless so input and output must be in the same units

EER=cooling capacity/work input
EER = Btu out (gross cooling) /W-H in or Btuh out/Watts in.
Watts input includes auxiliary equipment such as indoor and outdoor fans. For split systems, or if the indoor fan energy is unknown, ARI 210 adds 1,250 Btu/h per 1,000 cfm and adds by 365 W per 1,000 cfm.

kW/ton=work input/cooling capacity

COP for chillers includes only the compressor, not the auxiliaries or cooling tower fans.

EER=COP*3.413=12/kW/ton

kW/ton=12/EER=3.517/COP

COP=3.517/kW/ton=EER/3.413

The constant 3.517 is 12000/3413

SEER = weighted average of EERs for air-cooled equipment that consider performance improvements in mild weather.

HEAT-CONVERSION FACTORS

Calorie (Cal) = heat required to raise 1 g water 1 deg C.
Large Calorie (kCal) = heat required to raise 1 kg water 1 deg C.
Food Calorie = same units as kCal.

Btu = heat required to raise 1 lb water 1 degree F.
Therm = 10^5 Btu.
Dekatherm = 10^6 Btu.

AFFINITY LAWS

All of these relationships are for friction-only and a fixed system.
Q is flow, GPM or CFM

$(CFM2/CFM1) = (N2/N1)$

$(GPM2/GPM1) = (N2/N1)$

$(Q2/Q1) = (N2/N1)$

$(SP2/SP1) = (N2/N1)^2$

$(HP2/HP1) = (N2/N1)^3$

$Q2 = Q1 * (N2/N1)$

$SP2=SP1 * (N2/N1)^2$

$HP2 = HP1\ (N2/N1)^3$

Variation 1: HP in terms of flow change
Q is proportional to N, so
$HP2 = HP1\ (Q2/Q1)^3$

Variation 2: HP in terms of Variable flow downstream static pressure setting change

$HP2 = HP1 \times (^{SP2}/_{SP1})^{y1.5}$

Derivation of Variation 2

Fan Law 1: $HP2 = HP1 \times (\ ^{N2}/_{N1}\ )^{y3}$

Fan Law 2: $SP2 = SP1\ (\ ^{N2}/_{N1}\ )^{y2}$

Rewrite Fan Law 2: $\sqrt{SP2} = \sqrt{SP1}\ (^{N2}/_{N1})$

$(^{N2}/_{N1}) = {}^{\sqrt{SP2}}/_{\sqrt{SP1}}$

$(^{N2}/_{N1}) = \sqrt{(^{SP2}/_{SP1})}$

$(^{N2}/_{N1}) = (^{SP2}/_{SP1})^{y0.5}$ {A}

substitute {A} into Fan Law 1: $HP2 = HP1 \times [(^{SP2}/_{SP1})^{y0.5}]^{y3}$

finally: $\mathbf{HP2 = HP1 \times (^{SP2}/_{SP1})^{y1.5}}$

ELECTRICAL FORMULAS

Motor kW input = Input Volts * Input Amps * (√3) * PF * (1/1000)

Note: factor (√3) is for 3-phase loads only

Motor kW input = Motor Hp output * $(^{1}/_{\text{Motor Efficiency}})$

Motor Hp output = Input kW * Motor Efficiency

kVA = Volts * Amps * (√3) * (1/1000)

Note: factor (√3) is for 3-phase loads only

Power Factor (PF):

PF = real/apparent power

PF = kW/kVA

PF = COS θ

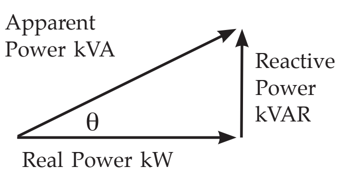
kW = kVA * COS θ

kVAR = kVA * SIN θ

kVAR = kW * TAN θ

$\theta = COS^{-1}\ (^{kW}/_{kVA})$

$\theta = TAN^{-1}\ (^{kVAR}/_{kW})$



Right Triangle Solutions
$kVA^2 = kW^2 + kVAR^2$
$kVAR = \sqrt{(kVA^2 - kW^2)}$

Table 21-1. Common Electrical Formulas
A = amps; V = Volts; Hp = horsepower; Eff = efficiency

TO FIND	DC	AC Single Phase	AC 3 PHASE
Amps when Hp is Known	$\frac{Hp \times 746}{V \times Eff}$	$\frac{Hp \times 746}{V \times Eff \times PF}$	$\frac{Hp \times 746}{\sqrt{3} \times V \times Eff \times PF}$
Amps when KW is known	$\frac{kW \times 1000}{V}$	$\frac{kW \times 1000}{V \times PF}$	$\frac{kW \times 1000}{\sqrt{3} \times V \times PF}$
Amps when kVA is known	$\frac{kVA \times 1000}{V}$	$\frac{kVA \times 1000}{V}$	$\frac{kVA \times 1000}{\sqrt{3} \times V}$
KW	$\frac{A \times V}{1000}$	$\frac{A \times V \times PF}{1000}$	$\frac{A \times V \times \sqrt{3} \times PF}{1000}$
KVA	$\frac{A \times V}{1000}$	$\frac{A \times V}{1000}$	$\frac{A \times V \times \sqrt{3}}{1000}$
Hp (output)	$\frac{A \times V \times Eff}{746}$	$\frac{A \times V \times Eff \times PF}{746}$	$\frac{A \times V \times Eff \times \sqrt{3} \times PF}{746}$
Efficiency	$\frac{746 \times \text{Output Hp}}{\text{Input Watts}}$	$\frac{746 \times Hp}{V \times I \times PF}$	$\frac{746 \times Hp}{\sqrt{3} \times V \times I \times PF}$
Power Factor		$\frac{\text{Input Watts}}{V \times A}$ Or $\frac{kW}{kVA}$	$\frac{\text{Input Watts}}{V \times A \times \sqrt{3}}$ Or $\frac{kW}{kVA}$

Voltage imbalance (three phase only):

$$\% \text{ imbalance} = \frac{(\text{max voltage on any line}) - (\text{average voltage})}{\text{average voltage}}$$

LOAD FACTOR

"Load Factor" is: $(^{\text{average demand}}/_{\text{maximum demand}})$.

Commonly applied to electrical demand, but equally applicable to gas or water. Poor load factors are related to high demand charges and improving a low load factor can reduce customer bills by reducing demand charges.

For ***example***, if a customer uses 100,000 kWh in a month and has a 300 kW maximum recorded demand, find the load factor.

Average demand is 300,000/30 days/24 hours = 416 kW

Maximum demand is 900 kW

Load Factor = 416/900 = 46%.

ENERGY TRANSPORT (CIRCULATING WATER AND AIR)

Pump brake horsepower (Bhp) = $^{\mathbf{gpm * head * sp. Gravity}}/_{\mathbf{3960 * eff}}$

Fan Bhp = $^{\mathbf{CFM * TSP * FA}}/_{\mathbf{6356 * eff}}$

Differential Temperature

- Differential temperature, or Delta-T, or dT, is the motive force for all heat transfer.
- Heat transfer will continue to occur until both sides are at equal temperature.
- Insulation practices will reduce heat transfer by increasing resistance to heat flow.
- Another equally effective method to reduce heat transfer is to reduce the dT.
- For a given insulation, half the dT results in half the heat transfer. This is the basis of savings for energy conservation measures like space temperature reset, hot water reset, cool roofs, and passive shading.
- For a given value of thermal energy transported by air or water, dT determines the required mass flow rate. For a given heating or cooling load, twice the dT results in half of the flow rate.

HEAT TRANSFER FORMULAS

The full description of heat transfer includes convection and radiation mechanisms. Only conduction is described in this text, for brevity and since it applies to insulation, a common energy reduction measure.

Conduction heat flow RATE for a building envelope (Btuh):

q=U * A * dT

U = heat transfer coefficient, Btu/SF-degF
A = SF
dT = differential temperature, degrees F

"U" can be for an individual area, or can be a combined, weighted U-value for the "overall" envelope.

For ***example***, if there are 25,000 SF of roof at U = 0.1, 100,000 SF of opaque wall at U = 0.2, 20,000 SF of opaque wall at U = 0.4, and 40,000 SF of glass at U = 0.7, find the overall U-Value

$$\frac{(25{,}000 * 0.1) + (100{,}000 * 0.2) + (20{,}000 * 0.4) + (40{,}000 * 0.7)}{185{,}000}$$

= 0.31 Overall U-Value

This also demonstrates which of the constituent envelope pieces contributes the most to heat loss/gain.

For this ***example***:
Roof: 2500/58,500 = 4%
Walls: (20,000 + 8000)/58,500 = 48%
Glass: 28,000/58,500 = 48%

Conduction heat flow RATE for glass (Btuh):

Q = A * SC * SG
A = SF
SC = shading coefficient, a fraction of 1 or less. 1 = no shade
SG = solar gain, Btuh/SF

Conduction heat flow ENERGY (Btu):

$\mathbf{Q = M * C_p * dT}$
M = Mass, lb
C_p = Btu/lb-degF
dT = differential temperature, degrees F

Envelope conduction heat transfer ENERGY from Degree Days (Btu):

$\mathbf{Q_{cooling} = U * A * 24 * CDD}$

Envelope conduction heat transfer ENERGY from Degree Days (Btu):

$\mathbf{Q_{heating} = U * A * 24 * HDD}$

HVAC FORMULAS AND CONVERSIONS

Distinction is made between energy (Btu/kWh) and rate (Btuh) since these are commonly confused.

Air Heating RATE (Btuh—sensible):

$\mathbf{q=1.08 * F_a * cfm * dT}$

dT = differential temperature, degrees F

F_a = altitude factor to account for changes in air density

See Item "**Altitude Correction**" in this section for air density ratios (altitude correction factors) at standard temperature, and the **Appendix "Altitude Correction factors at Different Temperatures."**

Note: For a given load (Q), when conveyed by air, a decrease in density from higher altitude or higher temperature requires an increase in volumetric air flow (CFM) for the same mass flow.

Total heat flow RATE of air (Btuh):

4.5 * cfm * dH

dH = differential Enthalpy, Btu/lb, taken from a psychrometric chart

Water heating RATE (Btuh):

500 * gpm * dT

dT = differential temperature, degrees F

Cooling energy from known ton-hours (kWh):

kWh = (Ton-hours) x (kW/ton)

Convert air changes per hour (ACH) to cubic feet per minute (CFM)

The volume of air in the enclosure is V (cubic feet)

One air change is one volume-worth of air moved per hour

so

CFM = ACH * V/60

V = volume of the enclosure

ALTITUDE CORRECTION

Air gets thinner at higher altitude and this fact affects many energy consuming equipment items, especially those involving combustion and convective heat transfer. This is significant in multiple ways:

1. Packaged heating and cooling equipment capacities are usually less than nameplate capacities would suggest. This is because they are rated at standard conditions, e.g. 95F ambient, 80F DB / 67F WB entering air, sea level.
2. Unless accompanied by higher air flows, heat transfer surfaces rejecting heat must increase in temperature and surfaces absorbing heat must lower in temperature to achieve the same heat transfer. For heat transfer driven by the refrigeration cycle, where the condensing and evaporating temperatures are created by the compressor, the thermodynamic lift increases and the compressor input power increases, i.e. efficiency is less at higher altitudes.
3. Fan power is reduced for lower density air. Fan rating are usually 70F at sea level.
4. Combustion equipment capacities (engines, burners) will have reduced capacity unless compensated with additional air flow. A rule of thumb for this de-rate is 4% per 1000 feet above sea level. Forced draft burners and turbo charged engines may be able to compensate with additional air flow, but anything 'naturally aspirated' will be de-rated.

Fans are affected significantly from altitude, but the effect on pumps is negligible—this is because water is largely non-compressible and its density change is small with altitude change. Refrigeration cycle equipment (compressors), and other 'closed' circulating systems are not affected by altitude, other than any heat transfer points.

Some equipment is manufactured with a degree of excess air flow or coil surface area to allow selection without de-rate for the first 2000-3000 feet.

Thinner air (less pounds of it for each cubic foot) means:

- HVAC calculations for air heat transfer are reduced directly as the ratio of <actual air density> to <sea level density>, since the basic formulas are for sea level air density. A table of altitude correction factors for standard temperature follows.
- Fans selected for sea level operation move the same CFM, but less

pounds of air at higher altitude, and use less brake horsepower.

- HVAC equipment: higher air temperature rise is required for air-cooled condensers, air coils, and dry coolers, with a corresponding higher condensing temperature. A higher air temperature drop is required for air-cooled evaporators, with a corresponding lower evaporator temperature. The approximate relationship for this is 2-4 percent per 1000 feet.
- Less fuel can be added to the mix to maintain the proper air-fuel ratios for combustion. This means combustion equipment is de-rated at altitude, including heaters, boilers, and automobiles. The approximate relationship for this is 4 percent per 1000 feet. Boilers with forced draft fans and engines with superchargers or turbo-chargers can compensate for this with oversized fans, to artificially raise the atmospheric pressure that the equipment sees.
- Higher air temperature rise is required for air-cooled equipment, with a corresponding higher component or heat exchanger surface temperatures. This affects equipment of all types that is cooled from surrounding air. The approximate relationship for this is 2 percent per 1000 feet for natural convection and 4% per 1000 feet for forced convection. Capacity de-rates apply to variable frequency drives (VFDs) due to the cooling heat sinks and other equipment with air-coils designed for sea-level air density.

For this text "Fa" is a density factor combining the effects of altitude and temperature, and are presented as a multiplying factor, e.g. CFM * TSP * **Fa**/6356 = Air Hp.

Table 21-2. Air Density Ratios (Altitude Correction Factors-Fa)

Altitude (ft) 0=Sea Level	Absolute Pressure at 70 degF (psiA)	Altitude Correction Factor at 70 degF Fa
0	14.70	1.000
1000	14.13	0.962
2000	13.61	0.926
3000	13.13	0.893
4000	12.67	0.862
5000	12.05	0.820
6000	11.76	0.800
7000	11.31	0.769
8000	10.89	0.741
9000	10.50	0.714
10000	10.14	0.690

Example: A naturally aspirated cast iron boiler rated at 1,000,000 Btuh at sea level, operating at 5,000 feet elevation, would be de-rated by a factor of approximately 5 * 4% = 20%, and have an output of closer to 800,000 Btuh at this altitude.

See also **Appendix "Altitude Correction Factors at Different Temperature**s."

HUMIDIFICATION

Adding moisture to air:

Humidifier Load (lbs/hr) = 60* Cfm * PCF(air) *ΔM

PCF(air) = lbs/cubic foot air density
ΔM = (lbs moisture/lb dry air)

Note also that each pound of evaporated water used for humidification absorbs about 1000 Btu of heat, either by the heater used to boil it or from the surrounding air.

DEHUMIDIFICATION

Removing moisture from air:

Dehumidification Load (lbs/hr) = 60* Cfm * PCF(air) *ΔM

Where:

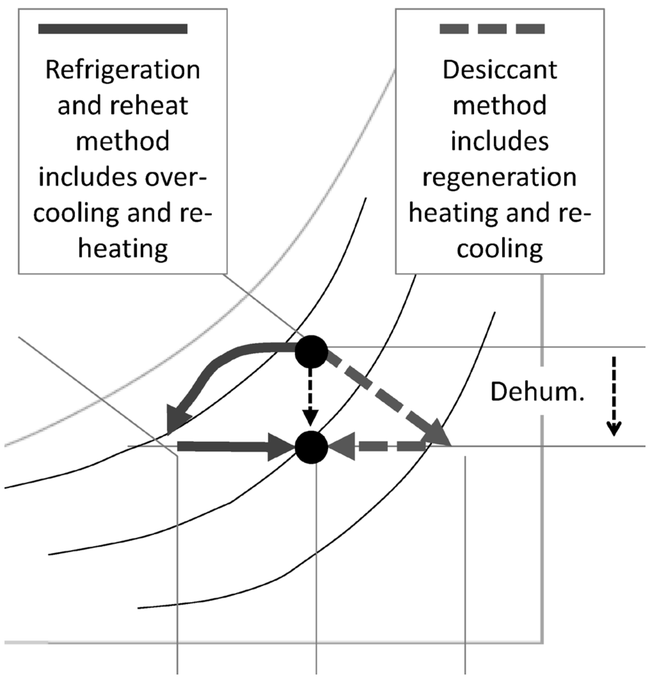
PCF(air) = lbs/cubic foot air density
ΔM = (lbs moisture/lb dry air)

This represents the water removal, pounds of water per hour. Unlike humidification, the dehumidification processes available are inefficient and the actual work required is greater than implied by the pounds of water removed. Psychrometrically, we would like to simply move the air condition vertically up and down, but the available processes do not allow this.

Energy penalties with dehumidification:

- Refrigeration and reheat: ~ twice the over-cooling amount
- Desiccant: regeneration and re-cooling energy
- Both are in the 20-40% efficient range
- Both can use waste heat if available

1. Dehumidification by refrigeration and reheat
The air is cooled to a point below dew point which removes the moisture. Presuming the initial air was at a desired temperature, the air is over-cooled to accomplish this and then must be reheated back its original state. The amount of over-cooling is accomplished at the COP of the cooling apparatus, and the equal and opposite amount of reheating is accomplished at the COP of the reheating apparatus.
2. Dehumidification by desiccant
The desiccant chemical absorbs water. The desiccant is regenerated by heating, to drive off the moisture so the desiccant can be re-used. The hot desiccant heats the air which must then be cooled back to its original state.



STANDARD TEMPERATURE AND PRESSURE (STP)

Ideal gas laws do a pretty good job of predicting the changes. (Not exact, but close)
Use absolute pressure and temperature

- Isobaric (constant pressure, change in temperature predict change in volume)
  - At higher temperatures the gas expands; lower temperatures the gas compresses
  - V2=V1*(T2/T1)

- Isothermal (constant temperature, change in pressure predict change in volume)
  - At lower pressure the gas expands; higher pressure the gas compresses
  - V2=V1*(P1/P2)

- Pressure and Temperature Compensation
  - Change in volume with change in both temperature and pressure V2=V1*(P1/P2)*(T2/T1)

    Utility billing and BTU meters adjust for this when gases are measured.

    Combustion is all about counting molecules, so optimized air/fuel adjusts P/T for both

Other examples of ideal gas laws are a gas in a fixed volume. Changes in pressure will change the temperature.

- Isochoric (constant volume, rigid container, change in pressure predicts change in temperature and vice versa)

T2 = T1(P2/P1)

- Increase pressure on the tank, temperature rises
- Decrease pressure, temperature drops

P2 = P1(T2/T1) Increase temperature on the tank, pressure rises

- Decrease temperature on the tank, pressure drops
- Increase temperature on the tank, pressure rises

PROPERTIES OF AIR, WATER, ICE

Air density: 0.075 lbs/$_{ft3}$ at STP
Specific heat of dry air at STP: 0.24 Btu/$_{lb\text{-}degF}$
Density of water at STP: 8.34 lb/gal = 62.4 lbs/$_{ft3}$ at STP
Specific heat of water at STP: 1 Btu/$_{lb\text{-}degF}$
Specific heat of ice: 144 Btu/$_{lb}$
Volume of water: 7.48gal/$_{ft3}$
7000 grains of moisture = 1 lb of moisture

SPECIFIC HEAT OF AIR AND WATER

Also called heat capacity, this is the heat that can be absorbed by the material for a 1-degree F temperature rise.

Specific heat of water is approximately 1.0 Btu/lbm (pounds mass).

Specific heat of air is approximately 0.24 Btu/lbm at 70 degrees F.

Note: The fact that one pound of water carries four times the heat energy of a pound of air is the reason energy transport using water is more efficient than air: air requires four times the 'pounds per hour' to be circulated, and four times the circulation energy is required to move it.

HEATING VALUES OF COMMON FUELS

Values are higher heating value (HHV) or gross heat release, LHV subtracts the heat of vaporization of water in the reaction. For most fuels, the difference is 2-10% (natural gas about 10%), but wood can have up to 40% moisture content and heating values per unit weight can vary considerably. Natural gas STP conditions are 60degF and 14.7psiA. (**See Figure 21-3**.)

LATENT HEAT OF WATER

This is the phase change heat properties of a material. Water is of interest for boilers and ice makers.

Heat of vaporization for water is approximately 960 Btu/lbm.

Heat of fusion for water is approximately 144 Btu/lbm.

Table 21-3. Heating Values of Common Fuels

Source: Biomass Energy Data Book, 2011, US DOE

Gaseous Fuels (32F / 1 atm)	Lower Heating Value		Higher Heating Value	
	Btu/ft3	Btu/lb	Btu/ft3	Btu/lb
Natural gas	983	20,267	1,089	22,453
Hydrogen	290	51,682	343	61,127
Still gas (in refineries)	1,458	20,163	1,584	21,905

Liquid Fuels	Lower Heating Value		Higher Heating Value	
	Btu/gal	Btu/lb	Btu/gal	Btu/lb
Crude oil	129,670	18,352	138,350	19,580
Conventional gasoline	116,090	18,679	124,340	20,007
U.S. conventional diesel	128,450	18,397	137,380	19,676
Low-sulfur diesel	129,488	18,320	138,490	19,594
Petroleum naphtha	116,920	19,320	125,080	20,669
NG-based FT naphtha	111,520	19,081	119,740	20,488
Residual oil	140,353	16,968	150,110	18,147
Methanol	57,250	8,639	65,200	9,838
Ethanol	76,330	11,587	84,530	12,832
Butanol	99,837	14,775	108,458	16,051
Liquefied petroleum gas (LPG)	84,950	20,038	91,410	21,561
Liquefied natural gas (LNG)	74,720	20,908	84,820	23,734
Methyl ester (biodiesel, BD)	119,550	16,134	127,960	17,269
Liquid Hydrogen	30,500	51,621	36,020	60,964
Butane	94,970	19,466	103,220	21,157
Isobutane	90,060	19,287	98,560	21,108
Isobutylene	95,720	19,271	103,010	20,739
Propane	84,250	19,904	91,420	21,597

Solid Fuels	Lower Heating Value		Higher Heating Value	
	Btu/ton	Btu/lb	Btu/ton	Btu/lb
Coal (wet basis)	19,546,300	9,773	20,608,570	10,304
Bituminous coal (wet basis)	22,460,600	11,230	23,445,900	11,723
Coking coal (wet basis)	24,600,497	12,300	25,679,670	12,840
Farmed trees (dry basis)	16,811,000	8,406	17,703,170	8,852
Forest residue (dry basis)	13,243,490	6,622	14,164,160	7,082
Sugar cane bagasse	12,947,318	6,474	14,062,678	7,031
Petroleum coke	25,370,000	12,685	26,920,000	13,460

Notes for **Table 21-3**

1. **HHV / LLV**: The heating value of any fuel is the energy released per unit mass or per unit volume of the fuel when the fuel is completely burned (ANSI/ASABE S593.1 2011). The term calorific value is synonymous to the heating value. Typical units for expressing calorific or heating value are MJ/kg in SI units or Btu/lb in English units. The heating value of a fuel depends on the assumption made on the condition of water molecules in the final combustion products. The higher heating value (HHV) refers to a condition in which the water is condensed out of the combustion products. Because of this condensation all of the heating value of the fuel including sensible heat and latent heat are accounted for. The lower heating value (LHV), on the other hand refers to the condition in which water in the final combustion products remains as vapor (or steam); i.e. the steam is not condensed into liquid water and thus the latent heat is not accounted for. The term net heating value (NHV) refers to LHV (ANSI/ASABE S593.1 2011). The term gross heating value (GHV) refers to HHV.
   Source: Biomass Energy Data Book, 2011, US DOE
2. **Wet basis and dry basis** are applied to solid fuels that contain moisture. The difference between the fuel weight as delivered and bone dry is the moisture weight.
   Moisture weight divided by the wet weight represents percent moisture "wet basis"
   Moisture weight divided by the dry weight represents percent moisture "dry basis".
3. **Conversions for Heating Values**:
   To convert from Btu/lb to kcal/kg multiply Btu/lb by 0.5556
   To convert from Btu/lb to MJ/kg multiply Btu/lb by 0.002326
   Source: Coal Conversion Facts, World Coal Institute, 2007
4. For typical bituminous coal with 10% moisture and 25% volatile matter, the difference between gross and net calorific values (difference between HHV and LHV) is approximately 470 Btu/lb.
   Source: Coal Conversion Facts, World Coal Institute, 2007

INSULATION FORMULAS

An excellent treatment of practical application of insulation is found in: *Energy Management Handbook*, 6th ed., Chap 15, The Fairmont Press.

R= $^{1}/_{U}$

U= $^{1}/_{R}$

Thermal Resistivity "R"

The higher the R-value, the higher the insulating value

Units: **(degF * SF * hr)/Btu**

R-values are commonly used to express performance of a chosen thickness. I.e. a 9-inch insulation batt has an R-value of 30 and a 3-inch batt of the same material has an R-value of 11. The "per inch" general parameter has been dropped since the thicknesses have been chosen.

R-values are tabulated for a wide variety of materials in many texts and are not repeated in this text. One source for material R-values of common materials is ASHRAE Fundamentals Handbook.

Overall Heat Transmission Coefficient "U"

The lower the U-value, the higher the insulating value

U= 1/R

Units: **Btu/(hr * SF * degF)**

U-Factor is commonly used for evaluating composite layered insulating assemblies. The amount of heat in Btu per hour that is transmitted through one square foot of a surface (wall, floor, roof), per degree F differential temperature.

Thermal Conductivity "K"

The lower the K-value, the higher the insulating value

K= $^{\text{thickness}}/_{R}$

K is the inverse of R. To convert K to R: R= $^{\text{thickness}}/K$

Units: $^{(Btu * inch)}/_{(hr * SF * degF)}$

K Factor is heat transfer intensiveness of a material, usually for one inch of thickness.

This is an industry standardized unit used by insulation manufacturers. The "per inch" is a convention that allows different materials to be readily compared.

Note: the K- values are temperature dependent and will vary slightly depending on temperature of service, i.e. the k-value at 100 degrees will be different than at 500 degrees.

Multiple Layers of Insulation

Calculate the overall effect by adding the individual "R"-values, then the reciprocal (1/Rt) is the component U-value. Individual U-values in a layered system cannot be added directly for the composite U_{total}.

$$Ut = 1/(R1 + R2 + \ldots Rn)$$

$$Ut = 1/\left(\frac{1}{U1} + \frac{1}{U2} + \ldots \frac{1}{Un}\right)$$

OTHER USEFUL FORMULAS

Equivalent Hydraulic Diameter of a Rectangular Shape

D = hydraulic diameter

$$D = 1.3 * \left(\frac{(w * h)^{0.625}}{(w + h)^{0.25}}\right)$$

Duct and Fitting Pressure Losses using "C" Factor

Pressure Loss = $(^{V}/4005)^2 * C$

where:

V = velocity in fpm

C=loss coefficient

Duct and Fitting Pressure Losses using Equivalent Diameters (L/D)

Pressure Loss = f*(L/D * (^{V}/4005)^2

where:

f = friction factor

L/D = equivalent diameters

Estimating Unitary HVAC EER from Nameplate Compressor Full Load Amp (FLA) Data

This gives a good approximation for existing equipment, but is not exact. Use manufacturer's literature for Btuh and watts instead of this method, if it is available.

NOTES.

1. Formula is shown for three phase. If single phase, omit the 1.732 factor. FLA = full load amps of the compressor.
2. Power factor PF is approximately 0.8 for motor-compressors.

Step 1. Btuh Cooling Capacity

Nominal capacity can sometimes be inferred from the model number labeling (e.g. ….036 = 3 tons for small unitary equipment, …075 = 75 tons for large unitary equipment) but this is voluntary by manufacturer and not consistent. Look it up to be sure.

The capacity is 'nominal' but so is the nameplate FLA

Step 2. Watts

A. Compressor watts: V * FLA * 1.732 * PF
(Sometimes RLA is shown instead of FLA)
B. Condenser fan watts (for air cooled equipment): add 10% of compressor watts.
C. Evaporator fan watts: 365 watts per 1000 CFM. This is from the ARI rating standard that defines EER ratings.
D. Power Factor (PF) can be assumed to be 0.8 for compressor motors for EER calculations.

Step 3. **EER = Btu/Watts**

FUEL SWITCHING— ELECTRIC RESISTANCE HEAT VS. COMBUSTION HEAT

The economics of this fuel switching option vary by locale. For heating purposes, the combustion option must include the efficiency loss, so the input is the calculated value. For electric resistance, the efficiency of conversion to heat is assumed to be 100%. Common units can be Btu or therm (for gas heating)

The following ***example*** compares natural gas heat to electric heat. Similar approach for other fuels.

Step 1. Convert combustion heat to common units

Natural gas cost is $1.00 per therm, or $10.00 per MMBtu
Efficiency is 80%
Adjusted cost is 10/0.8 = $12.50 per MMBtu

Step 2. Convert electric heat to common units

Electric cost is $0.07 per kWh
1 kWh = 3413 Btu
Efficiency is 100%
Adjusted cost is $0.07 *1,000,000/3413 = $20.50 per MMBtu

Factor: ($/kWh) * 293 = $/MMBtu

Factor: (/kWh) * 29.3 = $/therm

Step 3. Compare fuel cost savings to switch from electric to gas
(20.50 – 12.50)/20.50 = 39% savings to fuel switch for this example.

HEAT PUMP—APPROXIMATE COP FROM HIGH/LOW REGION TEMPERATURES

$$\text{COP} \sim 0.6 * \frac{460 + \text{Tcond}}{(\text{Tcond} - \text{Tevap})}$$

where:
(0.6 approximates the efficiency of industrial heat pump equipment.)
Commercial equipment less, e.g. 0.5)
temperatures are degF

Tcond = condensing temperature (higher temperature zone)
Tcond = evaporating temperature (lower temperature zone)

NOTE: the temperatures used are the temperatures of the refrigeration cycle, which are not the same as the high/low region temperatures, due to **heat exchanger approach**. For ***example***, a water source heat pump with a 70 degF water source and a 120 degF air sink may expect a 10 degF approach water-side and 25 degF approach air side, so the temperatures used in this formula would be 60 degF and 145 degF respectively. Failure to include approach will yield COPs that look better than they really will be.

HEAT PUMP—APPROXIMATE KW POWER FROM COP

$$kW = \frac{\text{heat duty}}{3413 * \text{COP}}$$

where:
heat duty is heat pump heat output in Btu/hr
3413 is Btu per kWh constant

CHIMNEY EFFECT

This natural effect is driven by height and differential density/buoyancy of warmer vs. colder air, but is commonly estimated using temperatures.

$$dP = Cah * \left[\left(\frac{1}{to} - \frac{1}{ti}\right)\right]$$

where:

dP = available pressure difference, in psi
C = 0.0188
a = atmospheric pressure, in psi
h = height, in ft
To = absolute outside temperature, in °R
Ti = absolute inside temperature, in °R

Quantifying stack effect can be useful when a stack unintentionally draws air into a building, since that air must be heated or cooled.

$$dP = Ca \sqrt{2gh * \left(\frac{Ti - To}{Ti}\right)}$$

where:

Q	=	stack effect draft/draught flow rate, ft^3/s
C	=	discharge coefficient (usually taken to be from 0.65 to 0.70)
A	=	area, ft^2
g	=	gravitational acceleration, 32.2 ft/s^2
h	=	height, ft
Ti	=	average inside temperature, °R
To	=	outside air temperature, °R

OTHER CONVERSION FACTORS

Conversions for Big Numbers

10^3	thousand
10^6	million
10^9	billion
10^{12}	trillion
10^{15}	quadrillion

Beware of "M"

Sounds like "million," but is often "thousand." If you're not sure, ask for the zeros.

In English units, M=1,000
In SI units, M=1,000,000

Examples:

MCF	=	1000 cubic feet
MBH	=	1000 Btu per hour
MGD	=	1,000,000 gallons per day
$M	=	1,000,000 dollars

MHz = 1,000,000 Hz
MJ = 1,000,000 Joules
MWh = 1,000,000 watt-hours = 1,000 kWh

Other letter designations
D=10
C=100
K=1,000
MM=1,000,000

1 MBtu = 1000 Btu
1 MMBtu = 1,000,000 Btu = 10^6 Btu
1 Quad = 10^{15} Btu

1 MW = 10^6 watts
1 kW = 3413 Btu/h
1 kWh = 3413 Btu
1 kWh = 3.6 E^6 Joule

1 ton-hour = 12,000 Btu

Table 21-4. Pressure Unit Equivalents

1 in. w.c.	0.0360 psi
1 in. w.c.	5.18 psf
1 in. w.c.	0.0733 in. Hg
1 in. w.c.	1.86 mm Hg
1 psi	144 psf
1 psi	2.31 ft. w.c.
1 psi	27.8 in. w.c.
1 psi	2.04 in. Hg
1 psi	51.7 mm Hg
1 atm	14.7 psi
1 atm	408 in. w.c.
1 atm	29.92 in. Hg
1 atm	760 mm Hg

1 therm = 100,000 Btu
1 therm = 105.5 E^6 Joule
1 ft^3 natural gas = approx. 1000 Btu at STP = 10^3 Btu at STP
1 ccf natural gas = 100 ft^3 = approx. 100,000 Btu at STP = 10^5 Btu = approx 1 therm at STP
1 MCF natural gas = 1000 ft^3= approx. 1,000,000 Btu at STP = 10^6 Btu at STP

1 barrel (petroleum measure) = 42 U.S. gallons

1 boiler HP = 33,475 Btu/h (33.5 Mbh) ***this is output***
= 9.809 kW
= 34.5 lbs of water evaporated at 212 degF

Approximate conversions using drops

20 drops = 1 ml
20,000 drops = 1 liter
75,680 drops = 1 gallon

Tons

Short Ton (U.S.) = 2000 LB
Metric Ton = tonne = 1000 kg = 2204.6 LB
Imperial ton (Long ton) = 2240 LB

Infiltration—Air

CFM/SF = 0.0547 * M^3H/M^2 (cubic meters/hr per square meter)

Human Power

A value of 0.075kW was used to compute the human energy expenditure in all unit operations, representing the average power a normal human can supply in tropical climates.
Source: Megbowon & Adewunmi, 2002

CONVERSION FACTOR TABLES

Source: Engineering Cookbook: A Handbook for the Mechanical Designer. Loren Cook Co.

Multiply Length		By		To Obtain
centimeters	x	.3937	=	Inches
fathoms	x	6.0	=	Feet
feet	x	12.0	=	Inches
feet	x	.3048	=	Meters
inches	x	2.54	=	Centimeters
kilometers	x	.6214	=	Miles
meters	x	3.281	=	Feet
meters	x	39.37	=	Inches
meters	x	1.094	=	Yards
miles	x	5280.0	=	Feet
miles	x	1.609	=	Kilometers
rods	x	5.5	=	Yards
yards	x	.9144	=	Meters

Multiply Area		By		To Obtain
acres	x	4047.0	=	Square meters
acres	x	.4047	=	Hectares
acres	x	43560.0	=	Square feet
acres	x	4840.0	=	Square yards
circular mils	x	7.854×10^{-7}	=	Square inches
circular mils	x	.7854	=	Square mils
hectares	x	2.471	=	Acres
hectares	x	1.076×10^{5}	=	Square feet
square centimeters	x	.155	=	Square inches
square feet	x	144.0	=	Square inches
square feet	x	.0929	=	Square meters
square inches	x	6.452	=	Square cm.
square meters	x	1.196	=	Square yards
square meters	x	2.471×10^{-4}	=	Acres
square miles	x	640.0	=	Acres
square mils	x	1.273	=	Circular mils
square yards	x	.8361	=	Square meters

Multiply Volume		By		To Obtain
cubic feet	x	.0283	=	Cubic meters
cubic feet	x	7.481	=	Gallons
cubic inches	x	.5541	=	Ounces (fluid)
cubic meters	x	35.31	=	Cubic feet
cubic meters	x	1.308	=	Cubic yards
cubic yards	x	.7646	=	Cubic meters
gallons	x	.1337	=	Cubic feet
gallons	x	3.785	=	Liters
liters	x	.2642	=	Gallons
liters	x	1.057	=	Quarts (liquid)
ounces (fluid)	x	1.805	=	Cubic inches
quarts (fluid)	x	.9463	=	Liters

Multiply Force & Weight		By		To Obtain
grams	x	.0353	=	Ounces
kilograms	x	2.205	=	Pounds
newtons	x	.2248	=	Pounds (force)
ounces	x	28.35	=	Grams
pounds	x	453.6	=	Grams
pounds (force)	x	4.448	=	Newton
tons (short)	x	907.2	=	Kilograms
tons (short)	x	2000.0	=	Pounds
Multiply Torque		**By**		**To Obtain**
gram-centimeters	x	.0139	=	Ounce-inches
newton-meters	x	.7376	=	Pound-feet
newton-meters	x	8.851	=	Pound-inches
ounce-inches	x	71.95	=	Gram-centimeters
pound-feet	x	1.3558	=	Newton-meters
pound-inches	x	.113	=	Newton-meters
Multiply Energy or Work		**By**		**To Obtain**
Btu	x	778.2	=	Foot-pounds
Btu	x	252.0	=	Gram-calories
Multiply Power		**By**		**To Obtain**
Btu per hour	x	.293	=	Watts
horsepower	x	33000.0	=	Foot-pounds per minute
horsepower	x	550.0	=	Foot-pounds per second
horsepower	x	746.0	=	Watts
kilowatts	x	1.341	=	Horsepower

Multiply		By	To obtain
acres	x	0.4047	= ha
atmosphere, standard	x	*101.35	= kPa
bar	x	*100	= kPa
barrel (42 US gal. petroleum)	x	159	= L
Btu (International Table)	x	1.055	= kJ
Btu/ft^2	x	11.36	= kJ/m^2
Btu·ft/h·ft^2·°F	x	1.731	= W/(m·K)
Btu·in/h·ft^2·°F (thermal conductivity, k)	x	0.1442	= W/(m·K)
Btu/h	x	0.2931	= W
Btu/h·ft^2	x	3.155	= W/m^2
Btu/h·ft^2·°F (heat transfer coefficient, U)	x	5.678	= W/(m^2·K)
Btu/lb	x	*2.326	= kJ/kg
Btu/lb·°F (specific heat, c_p)	x	4.184	= kJ/(kg·K)
bushel	x	0.03524	= m^3
calorie, gram	x	4.187	= J
calorie, kilogram (kilocalorie)	x	4.187	= kJ
centipoise, dynamic viscosity, μ	x	*1.00	= mPa·s
centistokes, kinematic viscosity, v	x	*1.00	= mm^2/s
dyne/cm^2	x	*0.100	= Pa
EDR hot water (150 Btu/h)	x	44.0	= W
EDR steam (240 Btu/h)	x	70.3	= W
fuel cost comparison at 100% eff.			
cents per gallon (no. 2 fuel oil)	x	0.0677	= $/GJ
cents per gallon (no. 6 fuel oil)	x	0.0632	= $/GJ
cents per gallon (propane)	x	0.113	= $/GJ
cents per kWh	x	2.78	= $/GJ
cents per therm	x	0.0948	= $/GJ
ft/min, fpm	x	*0.00508	= m/s

Multiply		By	To obtain
ft/s, fps	x	0.3048	= m/s
ft of water	x	2.99	= kPa
ft of water per 100 ft of pipe	x	0.0981	= kPa/m
ft^2	x	0.09290	= m^2
$ft^2 \cdot h \cdot °F/Btu$ (thermal resistance, R)	x	0.176	= $m^2 \cdot K/W$
ft^2/s, kinematic viscosity, v	x	92 900	= mm^2/s
ft^3	x	28.32	= L
ft^3	x	0.02832	= m^3
ft^3/h, cfh	x	7.866	= mL/s
ft^3/min, cfm	x	0.4719	= L/s
ft^3/s, cfs	x	28.32	= L/s
footcandle	x	10.76	= lx
$ft \cdot lb_f$ (torque or moment)	x	1.36	= N·m
$ft \cdot lb_f$ (work)	x	1.36	= J
$ft \cdot lb_f$ / lb (specific energy)	x	2.99	= J/kg
$ft \cdot lb_f$ / min (power)	x	0.0226	= W
gallon, US (*231 in^3)	x	3.7854	= L
gph	x	1.05	= mL/s
gpm	x	0.0631	= L/s
gpm/ft^2	x	0.6791	= $L/(s \cdot m^2)$
gpm/ton refrigeration	x	0.0179	= mL/J
grain (1/7000 lb)	x	0.0648	= g
gr/gal	x	17.1	= g/m^3
horsepower (boiler)	x	9.81	= kW
horsepower (550 $ft \cdot lb_f/s$)	x	0.746	= kW
inch	x	*25.4	= mm
in of mercury (60°F)	x	3.377	= kPa
in of water (60°F)	x	248.8	= Pa
in/100 ft (thermal expansion)	x	0.833	= mm/m
$in \cdot lb_f$ (torque or moment)	x	113	= mN·m
in^2	x	645	= mm^2

Multiply		By	To obtain
in^3 (volume)	x	16.4	= mL
in^3/min (SCIM)	x	0.273	= mL/s
in^3 (section modulus)	x	16 400	= mm^3
in^4 (section moment)	x	416 200	= mm^4
km/h	x	0.278	= m/s
kWh	x	*3.60	= MJ
kW/1000 cfm	x	2.12	= kJ/m^3
kilopond (kg force)	x	9.81	= N
kip (1000 lb_f)	x	4.45	= kN
kip/in^2 (ksi)	x	6.895	= MPa
knots	x	1.151	= mph
litre	x	*0.001	= m^3
micron (μm) of mercury (60°F)	x	133	= mPa
mile	x	1.61	= km
mile, nautical	x	1.85	= km
mph	x	1.61	= km/h
mph	x	0.447	= m/s
mph	x	0.8684	= knots
millibar	x	*0.100	= kPa
mm of mercury (60°F)	x	0.133	= kPa
mm of water (60°F)	x	9.80	= Pa
ounce (mass, avoirdupois)	x	28.35	= g
ounce (force of thrust)	x	0.278	= N
ounce (liquid, US)	x	29.6	= mL
ounce (avoirdupois) per gallon	x	7.49	= kg/m^3
perm (permeance)	x	57.45	= $ng/(s \cdot m^2 \cdot Pa)$
perm inch (permeability)	x	1.46	= $ng/(s \cdot m \cdot Pa)$
pint (liquid, US)	x	473	= mL
pound			
lb (mass)	x	0.4536	= kg
lb (mass)	x	453.6	= g
lb_f(force or thrust)	x	4.45	= N

Multiply		By	To obtain
lb/ft (uniform load)	x	1.49	= kg/m
$lb_m/(ft \cdot h)$ (dynamic viscosity, μ)	x	0.413	= mPa·s
$lb_m/(ft \cdot s)$ (dynamic viscosity, μ)	x	1490	= mPa·s
$lb_f s/ft^2$ (dynamic viscosity, μ)	x	47 880	= mPa·s
lb/min	x	0.00756	= kg/s
lb/h	x	0.126	= g/s
lb/h (steam at 212°F)	x	0.284	= kW
lb_f/ft^2	x	47.9	= Pa
lb/ft^2	x	4.88	= kg/m^2
lb/ft^3 (density, *p*)	x	16.0	= kg/m^3
lb/gallon	x	120	= kg/m^3
ppm (by mass)	x	*1.00	= mg/kg
psi	x	6.895	= kPa
quart (liquid, US)	x	0.946	= L
square (100 ft^2)	x	9.29	= m^2
tablespoon (approx.)	x	15	= mL
teaspoon (approx.)	x	5	= mL
therm (100,000 Btu)	x	105.5	= MJ
ton, short (2000 lb)	x	0.907	= mg; t (tonne)
ton, refrigeration (12,000 Btu/h)	x	3.517	= kW
torr (1 mm Hg at 0°C)	x	133	= Pa
watt per square foot	x	10.8	= W/m^2
yd	x	0.9144	= m
yd^2	x	0.836	= m^2
yd^3	x	0.7646	= m^3

METRIC CONVERSION FACTORS. SOURCE: ASHRAE

Multiply	By	To Obtain
acre	0.4047	ha
atmosphere, standard	*101.325	kPa
bar	*100	kPa
barrel (42 US gal, petroleum)	159	L
Btu, (International Table)	1.055	kJ
Btu/ft^2	11.36	kJ/m^2
Btu·ft/h·ft^2·°F	1.731	W/(m·K)
Btu·in/h·ft^2·°F		
(thermal conductivity, k)	0.1442	W/(m·K)
Btu/h	0.2931	W
Btu/h·ft	0.9615	W/m
Btu/h·ft^2	3.155	W/m^2
Btu/h·ft^2·°F		
(heat transfer coefficient, U)	5.678	W/(m^2·K)
Btu/lb	*2.326	kJ/kg
Btu/lb·°F (specific heat, c_p)	4.184	kJ/(kg·K)
bushel	0.03524	m^3
calorie, (thermochemical)	*4.184	J
calorie, nutrition (kilocalorie)	*4.184	kJ
candle, candlepower	*1.0	cd
centipoise, dynamic vicosity, μ	*1.00	mPa·s
centistokes, kinematic viscosity, ν	*1.00	mm^2/s
clo	0.155	m^2·K/W
dyne/cm^2	*0.100	Pa
EDR hot water (150 Btu/h)	44.0	W
EDR steam (240 Btu/h)	70.3	W
fuel cost comparison at 100% eff.		
cents per gallon (no. 2 fuel oil)	0.0677	$/GJ
cents per gallon (no. 6 fuel oil)	0.0632	$/GJ
cents per gallon (propane)	0.113	$/GJ
cent per kWh	2.78	$/GJ
cents per therm	0.0948	$/GJ
To Obtain	**By**	**Divide**

Multiply	By	To Obtain
in/100 ft (thermal expansion)	0.833	mm/m
in·lb$_f$ (torque or moment)	113	mN·m
in^2	645	mm^2
in^3 (volume)	16.4	mL
in^3/min (SCIM)	0.273	mL/s
in^3 (section modulus)	16 400	mm^3
in^4 (section moment)	416 200	mm^4
km/h	0.278	m/s
kWh	*3.60	MJ
kW/1000 cfm	2.12	kJ/m^3
kilopond (kg force)	9.81	N
kip (1000 lb$_f$)	4.45	kN
kip/in^2 (ksi)	6.895	MPa
litre	*0.001	m^3
MBtuh (1000 Btu/h)	0.2931	kW
met	58.15	W/m^2
micron (μm) of mercury (60°F)	133	mPa
mil (0.001 in.)	*25.4	mm
mile	1.61	km
mile, nautical	1.85	km
mph	1.61	km/h
mph	0.447	m/s
millibar	*0.100	kPa
mm of mercury (60°F)	0.133	kPa
mm of water (60°F)	9.80	Pa
ounce (mass, avoirdupois)	28.35	g
ounce (force of thrust)	0.278	N
ounce (liquid, US)	29.6	mL
ounce (avoirdupois) per gallon	7.49	kg/m^3
perm (permeance)	57.45	ng/(s·m^2·Pa)
perm inch (permeability)	1.46	ng/(s·m·Pa)
pint (liquid, US)	473	mL
To Obtain	**By**	**Divide**

(Cont'd)

Multiply	By	To Obtain
ft	*0.3048	m
ft	*304.8	mm
ft/min, fpm	*0.00508	m/s
ft/s, fps	*0.3048	m/s
ft of water	2.99	kPa
ft of water per 100 ft of pipe	0.0981	kPa/m
ft^2	0.09290	m^2
$ft^2 \cdot h \cdot °F/Btu$ (thermal resistance, R)	0.176	$m^2 \cdot K/W$
ft^2/s, kinematic viscosity, ν	92 900	mm^2/s
ft^3	28.32	L
ft^3	0.02832	m^3
ft^3/h, cfh	7.866	mL/s
ft^3/min, cfm	0.4719	L/s
ft^3/s, cfs	28.32	L/s
footcandle	10.76	lx
$ft \cdot lb_f$ (torque or moment)	1.36	$N \cdot m$
$ft \cdot lb_f$ (work)	1.36	J
$ft \cdot lb_f/lb$ (specific energy)	2.99	J/kg
$ft \cdot lb_f/min$ (power)	0.0226	W
gallon, US (*231 in^3)	3.785	L
gph	1.05	mL/s
gpm	0.0631	L/s
gpm/ft^2	0.6791	$L/(s \cdot m^2)$
gpm/ton refrigeration	0.0179	mL/J
grain (1/7000 lb)	0.0648	g
gr/gal	17.1	g/m^3
horsepower (boiler)(33,470 Btu/h)	9.81	kW
horsepower (550 $ft \cdot lb_f/s$)	0.746	kW
inch	*25.4	mm
inch of mercury (60°F)	3.377	kPa
inch of water (60°F)	248.8	Pa
To Obtain	**By**	**Divide**

Multiply	By	To Obtain
pound		
lb (mass)	0.4536	kg
lb (mass)	453.6	g
lb_f (force or thrust)	4.45	N
lb/ft (uniform load)	1.49	kg/m
$lb_m/(ft \cdot h)$ (dynamic viscosity, μ)	0.413	$mPa \cdot s$
$lb_m/(ft \cdot s)$ (dynamic viscosity, μ)	1490	$mPa \cdot s$
$lb_f \cdot s/ft^2$ (dynamic viscosity, μ)	47 880	$mPa \cdot s$
lb/min	0.00756	kg/s
lb/h	0.126	g/s
lb/h (steam at 212°F)(970 Btu/h)	0.284	kW
lb_f/ft^2	47.9	Pa
lb/ft^2	4.88	kg/m^2
lb/ft^3 (density, ρ)	16.0	kg/m^3
lb/gallon	120	kg/m^3
ppm (by mass)	*1.00	mg/kg
psi	6.895	kPa
quad (10^{15} Btu)	1.06	EJ
quart (liquid, US)	0.946	L
revolutions per minute (rpm)	*1/60	Hz
square (100 ft^2)	9.29	m^2
tablespoon (approx.)	15	mL
teaspoon (approx.)	5	mL
therm (100,000 Btu)	105.5	MJ
ton, short (2000 lb)	0.907	Mg; t (tonne)
ton, refrigeration (12,000 Btu/h)	3.517	kW
torr (1 mm Hg at 0°C)	133	Pa
watt per square foot	10.8	W/m^2
yd	*0.9144	m
yd^2	0.836	m^2
yd^3	0.7646	m^3
To Obtain	**By**	**Divide**

NOTES: Where deg k is used, it is for temperature intervals; deg C may be used for this purpose as well.
*Denotes the conversion is exact.

Chapter 22

Water Efficiency

Acknowledgment: Process water sections of this chapter were co-authored by Paul Larson,
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Contents

INTRODUCTION

This book is about energy auditing and is a source for common and helpful information to enable the task of optimization and finding energy and cost savings. Water is not energy, but has been included here for the practical reasons that it is not inexpensive anymore and professionals with the background, training, and tools in place to scout for energy savings will find that the methodologies used with energy can be applied to finding and quantifying water savings. This chapter primarily covers water use in manufacturing and water use as a partner to energy efficiency. Standard residential water use reduction measures are well defined and mentioned only briefly.

This text provides valuable insight and general approaches to serve the goal of reducing water use. The information is condensed as a general reference for water efficiency considerations. Like the energy information in this text, information related to water efficiency assumes and depends on good knowledge of water technology and the processes involved. Without a good background in water system technology, seek the assistance people who do understand water, to include on-site water experts. It is essential to understand that saving water should only occur when there is no impact to the process. Whether the water is used for manufacturing, a boiler, cooling equipment, or a car wash the specifics of the water are essential to process success. It is very possible to reduce water use and create a problem elsewhere as an unintended consequence. The more that is known about the process, process water requirements, water chemistry, water treatment technologies, measurement technologies and O/M aspects of water, the better.

Topics Not Covered

For domestic water heating, see **Chapter 18 Domestic Water Heating**.

For brevity, some topics were not included but are noted here.

- Residential water use and fixtures (brief mention)
- Storm water reuse
- Gray water reuse
- Sea or river water use
- Ground water (well water) use
- Irrigation and landscaping
- Landscaping

- Water laws, rights
- Environmental requirements, restrictions
- On-site wastewater treatment and neutralization
- Municipal water and wastewater treatment processes
- On site potable water treatment
- Distillation
- Electrical discharge ionization
- Closed loop circulating systems (sealed, no water consumption): chilled water, heating water, process cooling loop

A PHILOSOPHY OF WATER

As far as I know, the total amount of water on the planet is a constant. So, when a cooling tower evaporates 100,000 gallons of water in a day, it is recycled somewhere else as snow or rain and does so without having to wait a million years. But 'where' it is returned is significant, and the usual location is "somewhere else." So, a practical aspect of water is whether it is conveniently available and using less of it in a given location assists water supplies in that location. In fact, when water is not available locally, a common response is to move. For urban collections of people, water supply is an important concern.

Whether we should flush toilets with drinking water, water our yards with drinking water, or count decorative turf irrigation as a process efficiency gain are related questions, but not engineering questions, and not discussed here. For commercial and industrial water surveys, the goal is usually to find ways that reduce costs related to purchased water.

WATER TECHNOLOGY COMPARED TO ENERGY TECHNOLOGY

Understanding some fundamental similarities and differences are useful when adapting energy optimization concepts and practices to water optimization.

Compared to energy, there are similarities and differences. Like water, the total amount of energy on the planet is nearly constant, save the balance of heat loss to space and heat received from sunlight. Like water, energy accumulates and is stored, sometimes directly (thermal storage) but more often indirectly either in the form of a chemical process serving as 'fuel', or potential energy. Like water, energy is much more practical to access when it is in large reservoirs. Like water that is drawn from one spot and returned in the form of rain over a thousand square miles, energy that is in the form of low grade heat or sunlight may be plentiful but is difficult to retrieve economically. Unlike water, the return cycle for conventional fossil fuels is nowhere near close to the rate at which it is being consumed and so, while the energy balance remains, the practical reality is fossil fuel used for energy is not sustainable. Both water and energy supplies create pressure for growth and concentrated uses—without water people and plants will die; without energy, commerce will die. Energy has the additional pressure of fundamental change for sources that depend upon a fuel supply that will run out. An engineering concern for both water and energy is the predicting and planning and infrastructure so that human endeavors (life and commerce) receive a minimum of bumps. Unlike electricity, water pricing convention has not included time of use, water use "demand charges" are not a consideration for cost optimization. Water use definitely has seasonal and daily swings in load profile, so it is not out of the question for utility water rate designs to incorporate time of use and demand rates.

Like energy, most water cost management efforts fall into basic categories: using less to begin with, more efficient equipment, and less expensive sources of supply. See **Table 22-1.** For evaluation and analysis, water and energy are not a lot different. See **Table 22-2.**

REDUCED PRESSURE

(Table 22-3) Some water dispensing points of use will see reduced usage with reduced pressure (e.g. 40-50 psig vs. 60-80 psig). Reduced pressure reduces water flow through outlet points by about a third with half the supply pressure. When water flow is used for wetting, washing, rinsing and is left running during washing there is idle water flow time that is reduced with pressure. Examples are hand washing, dish washing, car washing, showering.

Note that volume-sensitive operations will not save water with this approach. Filling a pot sink or laundry wash machine with reduced pressure only takes longer and will impact productivity.

One residential study showed an **average annual water use reduction of 6% for homes that received water at lower pressures compared to higher pressures**. (Source: *How to Conserve Water and Use it Effectively*, EPA, 2015). Values of pressure were not given; in this location there are notable differences in site elevation which cause differences in service pressure and usage estimates were based on homes in different pressure zones.

Table 22-1. Basic Water Management Strategy Categories

Category	Water Examples	Energy Examples
Using less to begin with	Using water only when needed (automatic control and behavior), reduced evaporation and leak losses, and selecting processes that inherently do not need water. Reusing waste water rejected by one process in another process, storage.	Using energy only when needed (automatic controls, behavior), insulation, maintenance. Heat recovery between processes, storage
More efficient processes and equipment	Equipment with higher input/output efficiency (higher cycles of concentration, reduced blow down, regeneration, backwash).	Equipment with higher efficiency input/output efficiency, reduced part load throttling losses, reduced distribution losses.
Less expensive sources of supply	Well water, non-potable water, river water, etc.	De-regulation, fuel switching, energy storage where time of use rates exist.

Table 22-2. Water Calculation General Considerations

Calculation Consideration	Water Calculation	Energy Calculations
Transport costs	• Major for municipal water distribution. • Minor within buildings.	• Major for HVAC within a building (air/water circulation) • Major for district heating and cooling, distribution losses
Part load losses	No	Yes
Mass-energy balance involved	No	Yes. Common to evaluate mass flow and ΔT together
Compounding inefficiencies	Yes	Yes
Interacting measures	Yes	Yes
Evaporation	Direct loss	Cooling effect of evaporation can be an energy penalty or benefit
Time of use / seasonal rates	Yes (time of year) • Summer irrigation season • Sometimes blended rates are appropriate	Yes (time of day) • "On peak / off peak" rates • Sometimes blended rates are appropriate
Demand separate from consumption	No	Yes. Also power factor costs
Impacts to design intent, building function, O/M, health and safety	Sanitation and manufacturing quality risks from reuse water	Indoor air quality risks from ventilation and temperature resetting, productivity loss from temperature changes, manufacturing quality risk from process change

Table 22-3. Representative Water Flow Change with Pressure Reduction
Source: Water Pressure Reducing Valves, Watts, Inc. 2014

Supply Pressure psi	Water Volume in 10 Minutes
50 psi	30 gallons
100 psi	45 gallons
150 psi	56 gallons

WATER GRADES

Some Commercial Uses for Different Grades of Water

Definitions vary, some applications may overlap. This is to illustrate tiered water grades.

There are different ways to achieve water quality levels. Economics dictate using the lowest grade of water that will serve the task. Economics also suggest evaluating the discharge water from one process that may be suitable for a secondary, lower grade, use before discharging to the sewer. The concept of reusing water within a facility can be rewarding.

Highest grades

- Semiconductor fabrication (mineral residues create defects)
- Laboratory
- Reagent (chemical analysis or other reactions)
- Computer cooling
- Pharmaceutical manufacturing

Medium grades

- Pre-conditioning for higher grade water processing
- Beverages (contents affecting taste removed for consistency)
- Car and window washing (reduce spots/streaks after drying)
- Clothes washing, bathing
- Boiler feed water

Standard grade potable water

- Drinking water
- Food preparation
- General domestic use

Less critical water uses, potable or non-potable

- Irrigation (non-food)
- General washing operations (parts, street sweeping, non-food)
- Cooling tower and other evaporative processes
- Process cooling water/hydronic fill water
- Fire suppression

Water Purity Levels

Pure water is H_2O and nothing else, although in practice this is difficult to achieve.

Water is nicknamed the universal solvent since it is able to dissolve most chemical compounds; it also supports most life forms on this planet, and so it is no surprise that water can have most anything in it and that the impurities will vary by location and even what time of the year it is. "Drinking water" aka "potable water" describes water that is safe to drink; this can be accomplished by either removing or neutralizing unhealthy items in the water.

Water can be refined mechanically and chemically to different criteria depending upon what properties are needed. Water properties include suspended solids, dissolved solids, biological contaminants, and others. Various metrics are used to establish water quality. For example, when it comes to injecting water into people pretty much everything is a concern. In manufacturing, particulates and mineral content are common concerns. pH and dissolved solids can be trouble when water is conveyed in metal pipes and where heat transfer occurs. Dissolved gases cause problems when released in a heating process. Mineral build-up blocks pipes. And so on.

Distillation. Water distillation occurs naturally on earth and deposited as rain. Distilled water can be manufactured by heating water until it evaporates and then condensing it in a clean container. The process relies on water boiling at a lower temperature than anything else within the water, such that only H_2O is condensed. Distilling water requires considerable energy and is costly when the heat energy has a price tag. Distillation effectively separates water from minerals and biological contaminants, but will carry with it any substance with a boiling point equal to or lower than water such as solvents or oils.

Conductivity. Pure water is a poor conductor of electricity, and one of the measurements used to validate the quality of water is measuring its resistivity or conductivity (**Table 22-4**).

Table 22-4. Pure Water Grades Based on Resistivity

Laboratory (ISO 3696-1995)	Grade 1	Grade 2	Grade 3
Resistivity	>10 MΩ-cm	>1 MΩ-cm	>0.2 MΩ-cm
Conductivity	<0.1 µs/cm	<1 µs/cm	<5 µs/cm
Reagent Grade (ASTM D1193-2006)	Type I	Type II	Type III
Resistivity	>18 MΩ-cm	>1 MΩ-cm	>0.25 MΩ-cm
Conductivity	<0.056 µs/cm	<1 µs/cm	<4 µs/cm

Resistivity unit is megohm
Conductivity unit is microsiemens, which is equal to micromho, µ℧=µS

FILTERS AND STRAINERS

Filters and strainers are used for removing suspended solids (not dissolved solids) including debris, grit, sand, and other particles. Strainers capture larger pieces and use either perforations or wire screens. Filters capture finer particles and can even clarify cloudy water. Centrifugal filters separate particles that have a specific gravity greater than water.

Sand filters are beds of sand, in successively finer layers. Water is moved through the filter bed at low velocity like a press, and the suspended solids are mechanically removed. Periodically, the flow through the filter is reversed to 'lift' the sand and loosen the trapped material which is then is discarded to drain. Backwash flow must be sufficient to clean the sand, but not so high as to upset the stratification of the sand layers. Backwash/blow down cycles can be 5-20 minutes duration. Some facilities use a large drainage pipe (e.g. 8 inch) for the intermittent large volumes of backwash water, others use a storage tank to hold the backwash water which is then drained out through a small pipe over time.

Many filters and strainers are equipped with a **backwash** provision for occasional clearing of the filter using a portion of the system water. The purge water is normally routed to drain.

- Municipal water treatment
- Swimming pools

- Cooling towers
- Car washing

Water saving measures for filters:

1. **Recover sand filter backwash water.** Municipal water treatment facilities use sand filters and route the backwash water to a settling pond where time and gravity separate a lot of the solids from the water, after which the upper portion of the water is recovered and re-introduced into the plant water inlet. Other process steps that remove solids share the recovery pond (blow down, rinse water). The same concept of settling and re-using can be applied commercially where permitted for lower tier uses like cooling towers and other evaporative equipment on the property, or irrigation. Sand filters used with chemically laden water, such as cooling towers, would not be suitable for any reuse other than the system where it came from.
2. **Cartridge filters in lieu of sand filters.** Equal capacity cartridge filters are obtainable but are not used in practice other than small private pools. Where cartridge filters are allowable, there will still be some blow down for general sanitation, but much less and some claims have been made that overall blow down flow using cartridge filters can be reduced by half when compared to sand filters.
3. **Recover filter blow down water.** Where automatic blow down filters are used, such as suspended solids separator units that removes sediment by centrifugal force, the units periodically (usually on a timer) blow down to clear the collected solids. These units can be retrofitted with cleanable micron filters allowing the filter blow down water to be returned to the same system, leaving only the solids behind.
4. **Compress blow down duration.** Automatic controls for blow down may be invoked by time or differential pressure, but the blow down duration will be based on time. Blow down events that are too short will be self-reporting (e.g. cloudy water), but blow down events that are too long will not. Manually blowing down has the advantage of ending the blow down cycle when the leaving water is clear. If fully automatic, such as *xyz minutes twice a day every day*, settings will be conservative 'to be sure' which will then usually discharge more water than is needed. Where automatic controls are used for blow down, periodic checks of timer equip-

ment and validation of settings will help reduce un-necessarily long blow down events.

COOLING TOWERS

The bulk of water loss from a cooling tower is evaporation and is unaffected by water treatment. It takes 1000 Btus of heat (give or take) to evaporate a pound of water and the heat load of the building or process dictates the amount of heat to reject. So, the direct approach to reducing cooling tower water use is reducing the heat load at the source. If the heat load is considered unmovable, the next object of attention in reducing cooling tower water use is the blow down flow.

Water Consumption for Water-Cooled Mechanical Refrigeration Equipment

1 ton-hour = 12,000 Btu output, but the rejected heat has the heat of compression in it, so there is more heat to reject than 12,000 Btu. Factors for heat of compression at normal condensing temperatures are around 1.25, so 12,000 * 1.25 = 15,000.

1 ton-hour = 1.8 gallons of water

or

1 ton-hour = 15 lbs of water

One lb of water requires about 1000 Btu to evaporate, hence 1 ton-hr = 15 lbs of evaporated water to cool the condenser.

Combined:

- For cooling energy: (15 lbs/ton-hour) * 1 gal/8.34 lbs) = **1.8 gal/ton-hour water use from mechanical refrigeration**
- For cooling rate: (15 lbs/hr-ton) * (1 gal/8.34 lbs) * (1 hr/60 min) = **0.03 gpm/ton evaporation rate**

For actual water used, include blow down. The amount of blow down to add to the evaporation load depends on the cycles of concentration which in turn depends on initial water hardness and type of water treatment used.

Since the cooling tower leaves minerals behind as water is evaporated, mineral concentrations increase and will cause problems when the concentrations are too high. To control like scale and corrosion, a portion of the water is discharged as blow down, and replaced with cleaner water thereby diluting the remaining minerals. Automatic blow down maintains the mineral content at a prescribed level that is a balance between water cost and reliable cooling water operation. *Measures* *that reduce blow down water use do so by increasing cycles of concentration.*

To visualize "cycles of concentration," imagine whatever minerals there are in a glass of tap water. Evaporate the water and leave the minerals behind and fill it with more tap water. That is now two cycles of concentration. How many cycles can be used before dumping the water depends on the initial concentration levels of the minerals and the point of concentration where the minerals create problems.

Ex. If 1000 ppm is the red line for corrosion or scaling problems, using water with 100 ppm of minerals allows 10 cycles but using water with 500ppm of minerals only allows two cycles.

To determine the maximum cycles, identify the maximum mineral concentration of the circulating water (ppm TDS or μS conductivity), and divide that value by the make-up water that will be used for dilution.

Max cycles = Max ppm TDS/make-up TDS

or

Max cycles = Max μS/make-up μS

Ex. If 1000 ppm is the red line for corrosion or scaling problems, using water with 100 ppm of minerals allows 10 cycles but using water with 500 ppm of minerals only allows two cycles.

Make up water requirements to a cooling tower come mostly from evaporation, which is based on heat rejection (~ 1000 Btu/lb.); the other significant source of water use is blow down (dumped water based on minerals that must be replaced). Once cycles of concentration are known, the "blow down" flow rate can be determined from the evaporation rate according to the following formula.

Blow down flow:

$$\mathbf{BD = {}^{E}/_{(cycles\ -1)}}$$

Where:

E= evaporation flow rate, gal, gpm, gph, gpd, etc.

BD=blow down flow rate, same units as E

Cycles = cycles of concentration

Note: Blow down flow is not limited to cooling towers. In general, whenever water is evaporated, some form of dumping and replacing water is needed to prevent excessive mineral deposits. Humidifiers, swamp coolers, steam boilers, all control mineral deposits within some acceptable level of total dissolved solids, normally with blow down and making up with fresh water to dilute the remainder.

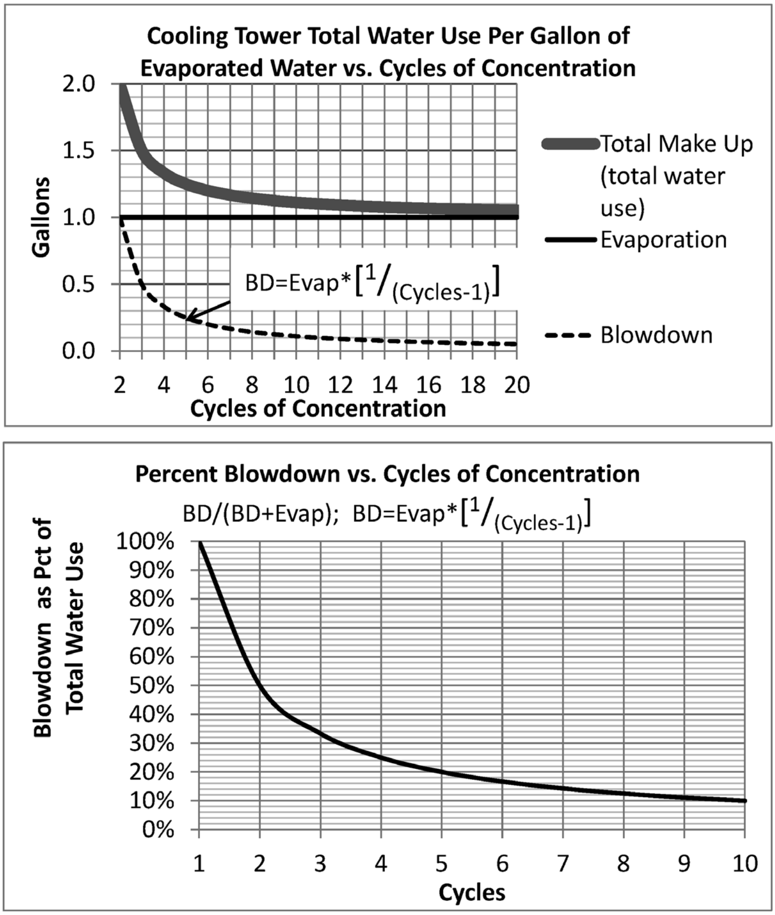


Figure 22-1. Cooling Tower Water Use vs. Cycles of Concentration
Total water use per gallon of evaporated water

The higher the cycles of concentration, the less blow down flow, although the water used by evaporation remains the same. With very low cycles of concentration, water use is increased dramatically and blow down flows equal evaporation rates (doubling the total use) at two cycles. Overall water use improvements decrease with higher cycles and there are diminishing returns above 6-8 cycles of concentration. See **Figure 22-1 and Table 22-5.**

Conventional treatments use chemicals to allow operation with higher levels of total dissolved solids (TDS):

Table 22-5. Cooling Tower Blow Down Flow vs. Cycles of Concentration

	BD/Evap	BD/Total
Cycles	BD as % of evaporation water use	BD flow as % of total water use
2	100%	50%
3	50%	33%
4	33%	25%
5	25%	20%
6	20%	17%
7	17%	14%
8	14%	13%
9	13%	11%
10	11%	10%
11	10%	9.1%
12	9.1%	8.3%

- Chemicals that help keep the dissolved solids in solution
- Chemicals that help prevent minerals from sticking to materials exposed to the water (especially heat exchangers)

Alternative treatment of cooling tower water

Multiple attempts have been made to increase cycles of concentration without chemicals, with mixed results. Some methods are passive, such as a magnet or copper rod lowered into the sump. Others rely on electrostatic or electromagnetic insertion devices. Others generate ozone or hydrogen peroxide in the water. And other methods will be devised. Some of the alternative water treatment methods have shown promise, others have caused severe metal damage, some have successfully replace biocides, and others have done nothing.

Attempts to use soft water for cooling towers see less mineral deposits because the calcium and magnesium have been removed by the softener unit. At normal cycles, there can be operational benefit from less scale (less minerals); attempts to use soft water at high cycles of concentration have encountered corrosion and other issues and should be considered a risk unless specifically and carefully designed for soft water use. Water chemistry is complex and the equipment it serves is expensive. Whatever the water treatment system, it is good practice to monitor with corrosion coupons and regular internal inspections.

Water-Saving Measures for Cooling Towers

1. Reduce rejected heat. Reducing cooling load is reflected in evaporation water use directly and in blow down according to cycles of concentration. Note: The bulk of cooling tower water use is for evaporation.
2. Use non-potable water or process hand-me-down water when available.
3. Increase cycles of concentration. Higher cycles do not affect evaporation rate, but will reduce the blow down flows. Cycles can be increased by using water with less minerals, water purification to remove minerals, chemical treatment that keeps minerals in solution, or piping and heat exchanger materials that don't corrode or scale. Savings from increased cycles depend upon what the cycles of concentration are currently; greatest improvements occur when initial cycles are low. See **Table 22-6**.

BOILERS

Heating water boilers are a sealed system and water consumption is limited to replenishing leaks. The water is merely a heat transfer medium.

Steam boilers that use 'live' steam for process must replace the expelled steam, pound for pound. Here, minerals are left behind from evaporation and are periodically blown down to drain; some blow down being from the top of the boiler water reservoir, but most blow down being from the bottom where the 'mud' ends up.

Steam boilers that serve heat exchangers use steam as a heat transfer medium and are mostly sealed, but there are sources of constant water loss. Condensate return lines are notorious for leaks and the 'tightness' of the condensate system is an important question for analysis. Hot wells, condensate tanks, and deaerator tanks are all system 'openings' where steam is allowed to vent.

Steam boiler elements of construction are very sensitive to minerals and gases in the water they boil, and the control limits will vary depending upon steam pressure and temperature, with less tolerance at higher operating pressures and temperatures (more strict requirements). Water treatment systems usually treat the water in the boiler as well as the incoming feed water. Some boiler water treatment methods:

- Deaerator: removes air and other gases from feed water by heating and venting
- Softener: removes minerals from feed water, especially calcium and magnesium, exchanging them for sodium

Table 22-6. Cooling Tower Water Reduction from Increased Cycles

Cycles Before	Cycles After	% Cooling Tower Water Use Reduction
2	3	25.0%
	4	33.3%
	5	37.5%
	6	40.0%
	7	41.7%
	8	42.9%
	9	43.8%
	10	44.4%
3	4	11.1%
	5	16.7%
	6	20.0%
	7	22.2%
	8	23.8%
	9	25.0%
	10	25.9%
4	5	6.2%
	6	10.0%
	7	12.5%
	8	14.3%
	9	15.6%
	10	16.7%
5	6	4.0%
	7	6.7%
	8	8.6%
	9	10.0%
	10	11.1%
6	7	2.8%
	8	4.8%
	9	6.3%
	10	7.4%
7	8	2.0%
	9	3.6%
	10	4.8%

BD=Blow down
E=Evaporation
Total water use = E+BD
BD = $^{E}/_{(\text{cycles -1})}$
Pct reduction =
[(Total water use 1 – Total water use 2)/Total water use 1]

- Chemicals: additives in the boiler water may include an oxygen scavenger, pH control and other ingredients to inhibit scale, pitting and corrosion.

Even with stellar water treatment blow down will be required because the evaporation process leaves behind virtually everything except H_2O. Some things that exist in boiler steam besides water include:

- Gases liberated from the water upon boiling
- Liquids that evaporate at steam temperature.
- Material that are mechanically entrained in the steam flow leaving the boiler

Boilers without any form of water treatment are rare, and invariably will have problems as a result, such as mineral accumulation in 3/4 inch tubes you can't fit a pencil through. Not only is this an obvious heat transfer and efficiency impact, it can also be dangerous when sensing lines to safety controls become blocked.

Blow down is used to prevent problematic concentrations of contaminants in the boiler. Often, the blow down rate is determined by the total dissolved solids (above xyz level of TDS, blow down); sometimes other contaminants dictate the need for blow down. When blow down occurs, the lost water is replaced with make-up water.

Blow down separators mix additional water with hot water to achieve an acceptable sewer drain discharge temperature, increasing actual water use and creating additional water and sewer charges. When the blow down is released from the boiler, it drops to ambient pressure and a portion of the blow down is dissipated as flash steam. **Table 22-7** indicates additional water use for blow down, presuming a discharge temperature limit of 140F to drain.

Note that in some cases cooling of blow down water in order to put it into a drain doubles the blow down water usage. This then doubles the savings value of any measure that reduces blow down.

The percent blow down is a function of the controlled minerals in the make-up water, and the cooling tower concept of 'cycles of concentration' applies equally well. If the make-up water has a concentration of a controlled mineral that is nearly as high as the maximum limit, there will be few cycles of concentration and a lot of blow down. Thus, the percent of blow down is a function of the level of impurities in the make-up water.

Table 22-7. Approximate Tempering Water to Cool Boiler Blow Down to a Final Discharge Temperature of 140F, Gallons per Gallon of Blow Down

BD=blow down Temper=tempering

BD at Boiler Saturated Liquid Temp, F	Pct of BD Flashed	BD Flow After Flashing, Gal	Cold Water Used for Temper, F	Temper Flow, Gal	Added water to Dispose of BD, Gal per Gal of Original BD	Total Water Use Including Make-Up, Gal per Gal of BD	Total Water to Drain, BD and Cooling Water
500	32%	0.68	50	0.55	0.55	1.55	1.23
			60	0.61	0.61	1.61	1.30
			70	0.70	0.70	1.70	1.38
400	20%	0.80	50	0.64	0.64	1.64	1.44
			60	0.72	0.72	1.72	1.52
			70	0.82	0.82	1.82	1.62
300	9.3%	0.91	50	0.73	0.73	1.73	1.63
			60	0.82	0.82	1.82	1.72
			70	0.93	0.93	1.93	1.84
220	0.8%	0.99	50	0.79	0.79	1.79	1.79
			60	0.89	0.89	1.89	1.88
			70	1.02	1.02	2.02	2.01

Note: Higher pressure boilers have higher temperature blow down water however this is flashed in the tempering unit and vented, so the actual cooling duty for the auxiliary cold water is given as 212F for all cases. Mass reduction from steam flashing is incorporated in the table values according to (Hb-Hf)/Vf, where:

Hb=heat of liquid at boiler pressure
Hf=heat of liquid at flash pressure
Vf=latent heat of vaporization at flash pressure

Boiler Blow Down Example 1: Boiler with a limit of 2000 ppm TDS

B=blow down	**Make up condition 1**			**Make up condition 2**		
	TDS ppm	**Minerals final**		**TDS ppm**	**Minerals final**	
Cycle 2	200	400		400	800	
Cycle 3	200	600		400	1200	
Cycle 4	200	800		400	1600	
Cycle 5	200	1000		400	2000	**B**
Cycle 6	200	1200				
Cycle 7	200	1400				
Cycle 8	200	1600				
Cycle 9	200	1800				
Cycle 10	200	2000	**B**			

TDS=total dissolved solids, ppm (parts per million)

The term 'cycles of concentration' isn't normally used with steam boiler blow down, however the same concepts that affect a cooling tower blow down rate apply so this is a legitimate way to visualize it. **Boiler Blow down Example 1** clearly shows that having twice the content of the offending item in make-up results in twice the blow down flow; so it follows that the blow down flow can be reduced by any means that reduces the make-up water content of the offending items. The conventional way to evaluate blow down as a function of make-up water is a simple fraction:

$$^{\textbf{Qty of A in feed water}}/_{\textbf{Max Qty of A in boiler water}} = \%\ \textbf{blow down}$$

Returning to **Boiler Blow Down Example 1:**
Make-up condition 1: 200 ppm TDS in make-up and 2000 limit.
200/2000 = 10% blow down

Make-up condition 2: 400 ppm TDS in make-up and 2000 limit.
400/2000 = 20% blow down

When one boiler water contaminant of concern is in much higher relative concentration than the others, it becomes the deciding factor for blow down. When this happens, there is just enough blow down for the one item and too much blow down for all the others. When there is imbalance in the items of concern, it is possible to reduce blow down by proactively removing some of the driving contaminant from the make-up water. **Boiler Blow Down Example 2** presents a mix of four items of interest, one of which is the more demanding and which sets the blow down for the boiler. Without being a water chemistry expert, concepts like this can be applied in practice by interviewing the water chemistry expert with questions that begin with "what are the conditions you monitor and blow down for and which are the dominant ones that determine overall blow down." There may not be an effective ancillary treatment for the dominant contaminant, but there may.

Boiler Blow Down Example 2

	Make up content	**Boiler water limit**	**% Blow Down**	**System Blow Down**
	ppm	**ppm**		
Before				
Contaminant A	350	900	38%	
Contaminant B	200	1500	13%	
Contaminant C	5	50	10%	
Contaminant D	75	500	15%	**39%**
After Pre-Treating for "A"				
Contaminant A	**150**	900	**17%**	
Contaminant B	200	1500	13%	
Contaminant C	5	50	10%	
Contaminant D	75	500	15%	**17%**

Boiler blow down can be manual, automatic, or continuous. Manual blow down alone represents risks of too much blow down (wasted water and chemicals) or too little blow down (equipment harm) and so automatic blow down is preferable to manual. Automatic blow down uses key measurement parameters and initiates a blow down event when the concentration of the item of interest reaches a threshold. Continuous blow down has the potential to create a steady contaminant level (as opposed to cycling up and blowing down) and is useful for heat recovery (steady flow instead of large infrequent batches) but a constant blow down flow paired with a variable load will create excess water use when the boiler operates at reduced load.

Sources of savings from reduced boiler blow down:

- Water and sewer costs, including the amplification factor for cooling the blow down before discharging to drain
- Replenishing water treatment chemicals
- Heat contained in the blow down water, including the amplification factor of firing efficiency
- Heat required to bring make-up water to boiler temperature
- Burner and induced draft fan energy
- Pump energy for feed water, elevated to boiler pressure

Water-Saving Measures for Boilers

1. Reduce demand for heat or steam.
2. Reduce condensate losses, especially the condensate return piping. Strive for no leaks in the condensate piping system.
3. Reduce blow down volumes
    - Pre-treat make up water to remove minerals to then reduce blow down requirements, i.e. water softener, reverse osmosis, demineralizer.
    - Automatic controls to allow water to cycle up while still within acceptable limits
    - Separate pre-treatment of make-up water when blow down extent is determined by one measured variable that is significantly closer to its threshold than all the others.

PROCESS WATER PURIFICATION

Note: A complete treatment of water purification equipment involves chemistry which can be daunting and is not the focus of this text. When measures are proposed that change operating characteristics, additional review is needed by the water chemistry specialists which may include equipment suppliers and operations.

Water that has been sanitized and made clean enough to drink may not be suitable for process use. And the reverse is true. Suitability of a water treatment process depends largely upon the use of the water, e.g. drinking water vs. process water, what the items of concern.

Other than a filter or strainer, water purification methods rely on chemistry. Discussions here attempt to relay some basic concepts that will aid an energy or process professional in pursing water optimization without claiming water chemistry expertise. As with any manufacturing or other process, proposals that suggest changes to a working water treatment system or processes should be reviewed by the customer and their water specialist, and then implemented carefully.

Terms specific to water purification have equivalents in more basic terms, which can help when first reviewing the technology.

Technology	**Term**	**Functional Equivalent**
Softener	Regenerant waste water	Discharge or Blow down
Deionizer (resin)	Regenerant waste water	Discharge or Blow down
Reverse osmosis	Reject water or concentrate	Discharge or Blow down
	Feed water	Input
	Product or permeate	Output

Water Softeners

Water softeners use a chemical exchange mechanism, trading sodium ions for calcium and magnesium ions (primarily) in water. Removing the "hardness" makes the water more "soft." Terms "hard" and "soft" are generally defined in **Table 22-8.** Some water softener uses:

- Reduce mineral deposits in boilers
- Reduce fouling in water atomizing nozzles
- Reduce the necessary amount of detergent in cleaning operations
- Pre-treatment for reverse osmosis on highly scaling water sources

Table 22-8. Relative Degrees of Water Hardness

	Grains/gal as CaCO3
Soft	<1
Moderately Hard	3-7
Very Hard	>10

The salt in a softener is consumed in the process and replenished. The mineral tank for the softener is periodically flushed to remove minerals. The discharge waste water contains the discarded minerals as well as brine (sodium or potassium). The regeneration (flush) event is triggered by water flow. A rule of thumb for standard water softener water performance is 5-7 gallons of discharge water per 1000 grains removal.

Note: perfect world conditions may be capable of 5 gallons but typical practice is to set up the regeneration event 10-15% inside the maximum limit, which increases the waste water slightly for quality assurance.

The more 'work' the softener must do upon the inlet water, the sooner its capacity to hold minerals is exhausted and must be flushed. Thus the water efficiency of a softener will be lower with harder inlet water.

Deionizers

This section is limited to resin bed deionizers.

Industry names include resin bed deionizers, demineralizers, ion exchangers or DI water units. Each has the process goal of removing ion impurities of water. These are often the last in a series of purification steps and are preceded by particulate filtration and RO units. Post filtering is also common. In ultra pure water, the water becomes aggressive to metal and is carried in plastic piping.

Deionizer types vary depending on what the water will be used for. Basic types:

Strong Acid Cation/Weak Base Anion (SAC/WBA). The first bed (strong acid) exchanges hydrogen ions for cations from the water and resin in the second bed (weak base) absorbs the acid. The quality is roughly 20-100 KΩ (20,000-100,000 ohms) resistivity and is used for things like boiler feed water and ice making.

Strong Acid Cation/Strong Base Anion (SAC/SBA). The first bed functions like SAC/WBA but the second bed is different. In the second resin bed the resin replaces the anions with hydroxide ions that combine with the hydrogen ions to form H2O. The quality is roughly 1 MΩ (1 million ohms) resistivity and is used for things like metal plating rinses and power plant make-up. SAC/SBA remove silica and other weakly charged anions.

Mixed Bed. Here, the resin beads are a mixture of acid/base. The quality is up to 18.2 MΩ (18.2 million ohms) resistivity and is used as ultra-pure water for chemical reagent and semiconductor rinsing. Mixed bed DI units will utilize a pre-treatment upstream such as SAC/SBA deionizer or reverse osmosis to prevent damage to the resin bed. Mixed-bed DI is equivalent to thousands of SAC/SBA units in series.

Resin deionizer units use a variation of the water softening process, where ions are removed by chemical attraction. These units are easily fouled and are used after upstream treatments like filters and RO processes. Like a softener, the resin bead affinity for ions eventually becomes 'full' and must be regenerated. Key differences between DI and softener regeneration are:

- The regeneration water and rinse water is of equal quality to normal influent water, not tap water like a softener.
- The regeneration water includes acid and caustic before being rinsed.

The time between resin regeneration is determined by its capacity to remove grains of ions, and the less work the DI unit has to do in removing impurities, the longer it can operate between regeneration cycles.

Calculating Water Use for Resin Bed Softeners and Deionizers

On a per-GPG removal rate basis, a basic relationship for softener waste water is

Gallons per 1000 gallons of finished water = w/b

Where:

w = waste water flow, gallons waste water per cubic foot of resin

b = kgal batch size per regeneration event = (r/GPG removal rate)

r = threshold when the resin needs to be regenerated, grains capacity

See **Figure 22-2A, Figure 22-B and examples**

Softener Example 1

Removal capacity	25,000 grains (1 cubic foot of resin) before regen
Removal rate	20 grains/gal CaCO3
Gal to reach limit (b)	25,000/20 = 1250 gallons (1.25 kgal) finished water
Regeneration rate (w)	150 gallons per cu ft of resin
Regeneration volume	150 gal/cu ft * 1.0 cubic ft of resin = 150 gallons
Discharge per kgal (w/b)	150/1.25 = 120 gal per 1000 gal finished water

Total water used per kgal	1000+120 = 1120 gal
Water process efficiency	output/input = 1000/1120 = 89% eff

Deionizer Example 1

Application	SAC/WBA DI (strong acid, weak base)
Removal capacity	21,000 grains (1 cubic foot of resin) before regen
Removal rate	15 grains per gallon
Gal to reach limit (b)	21,000/15 = 1,400 gallons (1.4 kgal) finished water
Regeneration rate (w)	300 gallons per cu ft of resin
Regeneration volume	300 gal/cu. ft. * 1.0 cubic ft. resin = 300 gallons
Discharge per kgal (w/b)	300/1.4 kgal = 214 gal per kgal finished water
Total water used per kgal	1000+214 = 1214 gal
Water process efficiency	output/input = 1000/1214 = 82% eff

Compound Treatment Loss

It is important to note that DI regeneration and rinse water use finished DI water. The embedded water losses associated with regenerating DI equipment reduces the efficiency of the DI process. When water input comes from a pre-treatment step (series processes), the overall efficiency is the combination of the individual efficiencies. The overall efficiency for two process in series is determined by

Overall Process efficiency = (Process 1%efficiency) * (Process 2 efficiency)

An alternate method is to identify the collective input and output flows, the difference being any reject, purge, backwash, or blow down flows. The two methods should yield identical answers and one can be used to double check the other.

Overall efficiency = total output/total input

Total DI output is to the process, and does not include the parasitic DI water use related to regeneration.

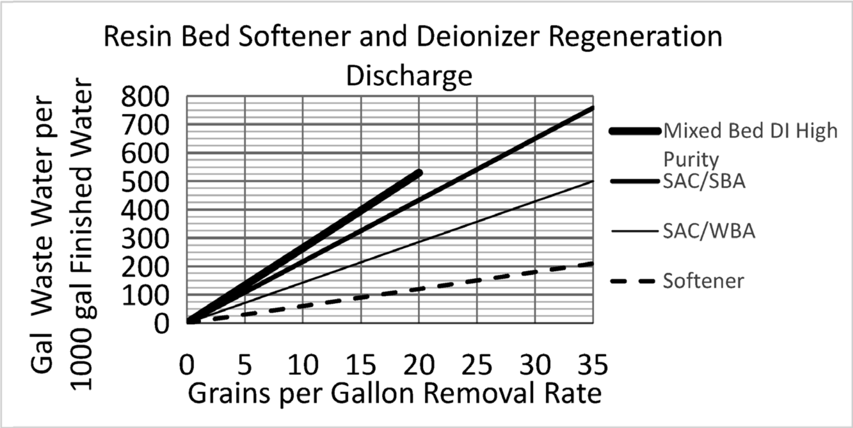
For example, a SAC/SBA DI process is required to produce finished water at 0.00292 gpg. Two options are explored for water efficiency. Case 1 takes city water to the DI unit directly, and Case 2 pre-treats the city water with an RO unit before entering the DI unit.

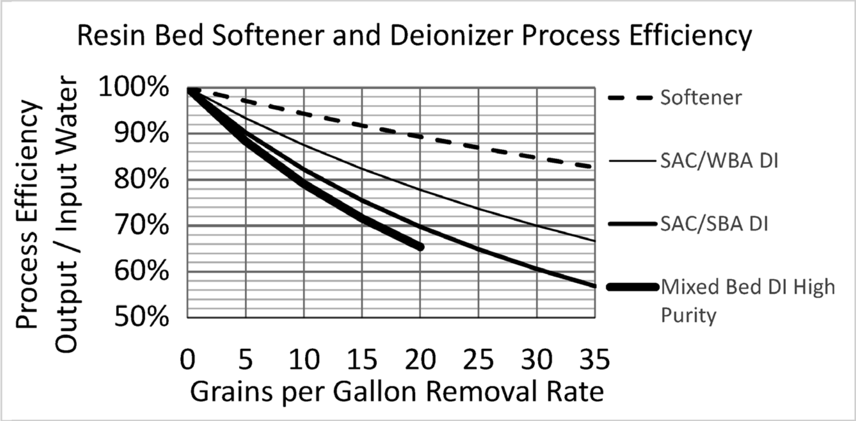
<u>Case 1</u>. DI alone, using city water directly that contains 600 ppm

TDS with a final DI output of 0.05 ppm. This is a change from 35.1 gpg to 0.00292 gpg, ~35 gpg reduction. From **Fig 22-2A**, the DI regeneration would be about 750 gallons per 1000 gallons of output. Process efficiency is output/input, 1000/(1000+750)=57%. 1750 total gallons used to produce 1000 gallons of finished water.

Case 2. RO process in series with DI. Same initial water content of 600 ppm TDS and same final output of 0.05 ppm, but the mineral removal is in two steps with RO taking most of it. The reverse osmosis step removes 95% of the dissolved solids in exchange for losing 30% of the feed water. The RO output water TDS is approximately (1-0.95)*600 = 30 ppm which then becomes the DI input water. Now, the DI work load is 30ppm to 0.05 ppm, which is a change from 1.75 gpg to 0.00292 gpg, ~1.75 gpg reduction. From **Fig 22-2B** (low range chart) the DI regeneration would be about 37 gallons per 1000 gallons of output. Process step efficiency is output/input, 1000/(1000+37)=96%. Overall efficiency for the two step process includes the RO water loss to pre-condition the DI input water. Process efficiency is output/input, 1000/(1037/0.7)=67%, compared to 57% for DI alone. 1481 total gallons used to produce 1000 gallons of finished water is a 15% reduction in water use.

> Note that there are other considerations for pre conditioning DI inlet water with RO, namely the chemical costs, labor burden and waste water treatment costs associated with servicing the DI units which occurs much less frequently when pre-treating with RO.



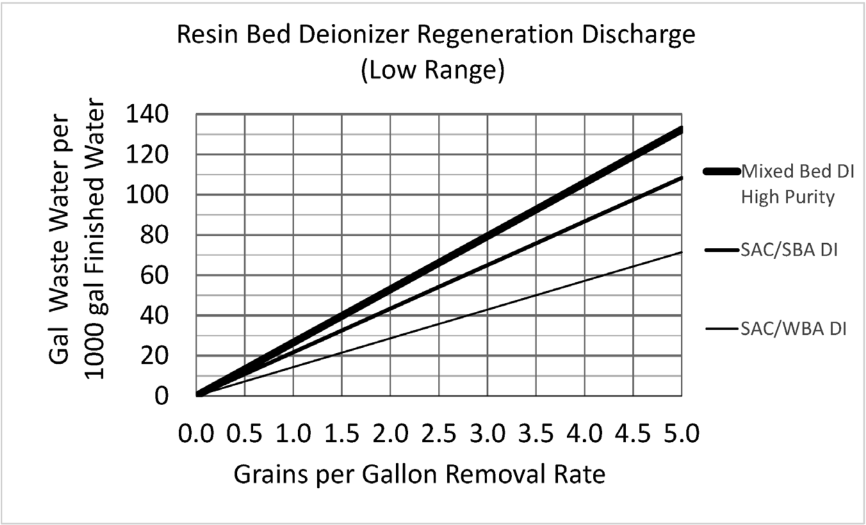


Softener		GPG removal = incoming water CaCO3
		(All hardness of incoming water removed)
	w	150 gal discharge/cf of resin
	r	25,000 grains/cf resin capacity at regen
SAC/WBA DI		20,000-100,000 ohm resistivity
	w	300 gal disch./cf of resin (total both tanks)
	r	21,000 grains/cf resin capacity at regen
SAC/SBA DI		1 Meg ohm resistivity
	w	325 gal disch./cf of resin (total both tanks)
	r	15,000 grains/cf resin capacity at regen
Mixed Bed DI High Purity		15-18 Meg ohm resistivity
	w	225 gal discharge/cf of resin
	r	8,500 grains/cf resin capacity at regen

gal/kgal = w/b (waste/batch)
w=waste flow per cu ft of resin for regen
b=(r/gpg removal rate) / 1000
r=grains removal per batch

Values are approximate but a good Indicator of magnitude. Values are not Corrected for the added water use from Using DI or RO water as the regen and rinse water. Water flows include all steps including final rinse. Water flows per cu ft of resin are typical but will vary by operator and chemical dosing, ~10% or more.

Figure 22-2A. Resin Bed Softener and Deionizer Discharge



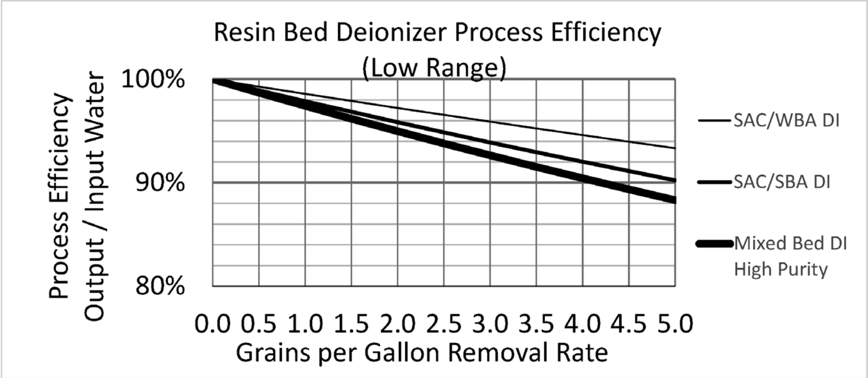


Figure 22-2B. Low Range Resin Bed Deionizer Discharge

Reverse Osmosis (RO)

RO units operate by leveraging a natural process called osmosis, which is diffusion across a semi-permeable membrane. RO membranes are used to remove particles in the sub-micron size (e.g. 0.0001 micron). The main feature of RO is the ability to remove dissolved solids, which filtration alone cannot achieve. The standard measurement of dissolved solids is TDS (Total Dissolved Solids), and reducing levels of TDS have a variety of uses in industry. **Table 22-9** gives examples of TDS levels for common water sources.

Table 22-9. Relative Total Dissolved Solids for Some Water Sources

Water Type	TDS (mg/L)
Potable water	<500
Fresh water (not treated)	<1500
Brackish water	1500-5000
Sea water	>5000

Source: Reverse Osmosis Optimization, FEMP/PNNL, 2013

Osmosis is a naturally occurring phenomenon where a dilute solution will travel across a semi-permeable membrane to an area of higher concentration, until they are in equilibrium. What moves through the membrane is the solvent (water) and the impetus for the movement is the "osmotic pressure" created by the lack of equilibrium in the two fluid concentrations. Osmosis is at work when we eat something salty and then feel thirsty. **Reverse osmosis** is a machine that reverses the natural process using a pump to raise the fluid pressure above the osmotic pressure, overcoming the natural force and fluid flow direction. In so doing, solvent (water) moves away from the concentrated minerals instead of toward it (reverse flow direction) and the impurities collect on the pressurized side where they are rejected. The result is purified water leaving the machine. Production RO units remove as much as 98-99% of dissolved solids or 'salts' in this way. The RO process can be visualized as the pump (piston) pushing water through the membrane, although the actual mechanism is a combination of porosity and diffusion. See **Figure 22-3**.

Note: The correct terms for RO flows are feed water (inlet), permeate or product (outlet), and reject (waste). This text interchanges names but includes explanations in terms of inlet, outlet, and waste because they are intuitive and assist in making the analogy connection with other familiar *processes. Once familiar with water purification concepts, it is appropriate to use standard industry terms.*

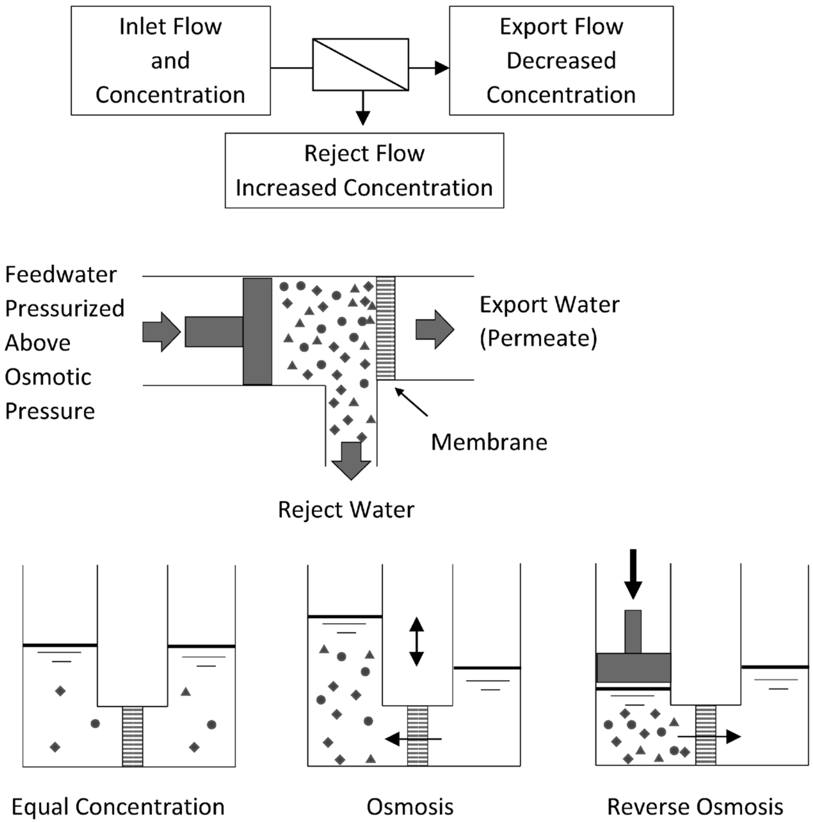


Figure 22-3. Reverse Osmosis Functional Diagram

Osmosis is about *concentration levels*. The RO process increases the concentration the total dissolved solids in the waste water while simultaneously decreasing the concentration of dissolved solids from the finished water. The **rejection rate** is the performance statement for the machine regarding its ability lower the mineral concentration of the leaving water. The discharge from RO units is a continuous process, and is called **reject water**.

RO is commonly used for desalinization of sea water and brackish water, and for purifying drinking water in general. RO is a popular partner for pre-treating water prior to deionization purification, and as a water supply for humidifiers, car wash rinsing and window washing. The principle treatment of RO applications in this text is for commercial and process use, within the context of efficient use of water.

Recovery rate is calculated from product flow/feed water flow (output flow/input flow), and varies by inlet water condition. A high recovery rate is desirable because it means more of the input water become usable output water and less of the input water (presumably paid for) goes to waste.

Rejection fraction expresses the portion of inlet water minerals removed from product water.

% **Rejection = 100% - (product conc./inlet conc.)**

Mass balance exists as the flows divide through the RO process. Since the total amount of water and the total amount of minerals is conserved, useful relationships exist.

Product (output) flow * product conc. + reject (waste) flow * reject conc. = feed water flow * feed water conc.

Since conductivity is a reflection of mineral concentration:
Product (output) flow * product conductivity + reject (waste) flow * reject conductivity = feed water flow * feed water conductivity

Inlet minerals = product (output) minerals + reject minerals

Reject minerals = reject flow * reject conc.

Product (output) minerals = product flow * product conc.

Concentration when mineral count is known:
Concentration = minerals/flow
Ex: grains/gallons=gpg; mg/liters = mg/L

Mineral count when concentration and flow are known:
Minerals = concentration * flow
Ex: gpg*gallons = grains; mg/L*liters=mg

Note: In most cases with RO, the objective is to determine the concentration of one stream when the other two are known. In this calculation, an interim step is the qty of minerals, but only to enable the follow-on step of concentration. If used consistently, the un-matching units are harmless when used in this way. Shorthand:

- Consider flow rates to be "volumes" (e.g. gpm treated as gallons)
- Consider concentration units to be "units of minerals," e.g. gpg, ppm, mg/L are all units of minerals.
- Then, gpm and ppm can be used together, if consistent. Alternately, the back and forth conversions to matching units will produce the same values.

<u>Given</u>: Inlet flow = 100 gpm, inlet concentration = 600 ppm, flow recovery = 75%, mineral rejection = 98%
<u>Step 1</u>: Calculate flow for product and reject. Product flow = inlet flow * recovery. 100 gpm * 0.75 = 75 gpm. Reject flow = inlet flow – product flow. 100 gpm – 75 gpm = 25 gpm.
<u>Step 2</u>: Calculate product concentration. Product conc. = (1-mineral rejection)*inlet conc. (1-0.98)*600 ppm = 12 ppm.
<u>Step 3</u>: Calculate reject concentration from the other two known quantities. For mass balance treat flow rates as 'volume', concentration units generically, and minerals generically as "units."

Inlet minerals = ppm*volume. 600 ppm*100 gpm = 60,000 units
Outlet minerals = ppm*volume. 12 ppm*75 gpm = 900 units
Reject minerals = inlet minerals – outlet minerals. 60,000-900=59,100 units
Reject concentration = minerals/volume. 59,100/25=2364 ppm

See **Figure 22-4**

Sanity Check for Flow Meters and Conductivity Meters is another good application of mass balance in reverse osmosis work. Reverse Osmosis units often utilize digital magnetic impulse flow meters for manufacturing economy. These flow meters are often in-accurate and require field calibration. Most operators have no way other than barrel method (a large bucket and a stop watch; cumbersome and messy) to measure the actual flow rate being discharged to calibrate the meter. Therefore the meters do not get proper calibration in the field and the factory calibration is assumed by the operator to be correct. This can and

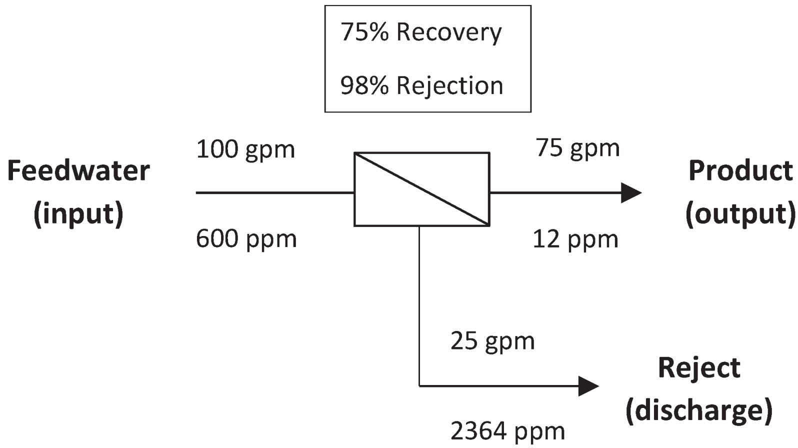


Figure 22-4. Sample RO Calculation Using Mass Balance

does result in poor recovery efficiency and wasted water.
For a +/- 5% range of acceptable accuracy:

> **(Inlet flow)*(inlet conductivity)/[(output flow*output conductivity)+(reject flow*reject conductivity)] = 0.95-1.05**

If outside the range of 0.95-1.05, one of the meters is compromised

The following example illustrates how to verify flow meter accuracy using the mass balance principle and conductivity readings. This presumes conductivity meters are accurate and the membrane is functioning to its mineral rejection fraction rating—the only unknown is flow. This method will verify proportions of flow (i.e. the recovery rate) but not actual flow. Verifying proportions is useful because that is what flow recovery factor is.

Specified	
Inlet flow	100 gpm
Inlet concentration	150 ppm
Recovery	0.77
Mineral rejection	0.98
Flow meters indicate	
Inlet flow	100 gpm
Outlet flow	77 gpm
Reject flow	23 gpm (implied)

<u>Conductivity readings</u>

Inlet conductivity	250 µS
Outlet conductivity	5.0 µS
Reject conductivity	821 µS

<u>Step 1: Gross check</u>

(Inlet flow)*(inlet µS)/[(output flow*output µS)+(reject flow*reject µS)] should = 0.95-1.05 ← should be if all is well (100*250)/[(77*5.0)+(23*821) = 1.30 (30% error). ← **Something is wrong**.

<u>Step 2: Find flow proportions conductivity</u>

<u>Note A:</u> this method uses conductivity readings. The math can be done in ppm or mg/L units but it is not required.

<u>Note B:</u> Concentration in reject water will vary with recovery flow rate, but concentration will not vary with product water provided the membrane is functioning to its mineral reject rating.

<u>Note C:</u> Remaining minerals method (mass balance) is used to determine design value of reject concentration.

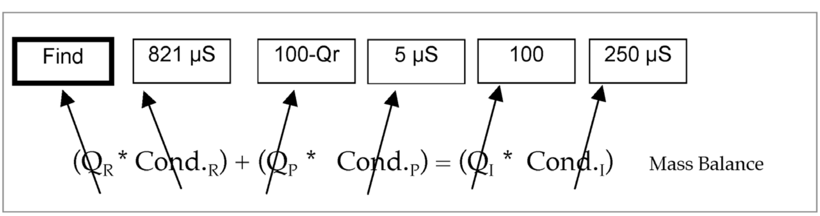
<u>Note D:</u> This process establishes the proportions of flows based on mass balance, inferring mineral content from conductivity. The actual flows can only be known when there is one accurate flow reading.

Determine what the conductivity readings should be if the flow is correct

% **rejection=100% - (product conc./inlet conc.)**

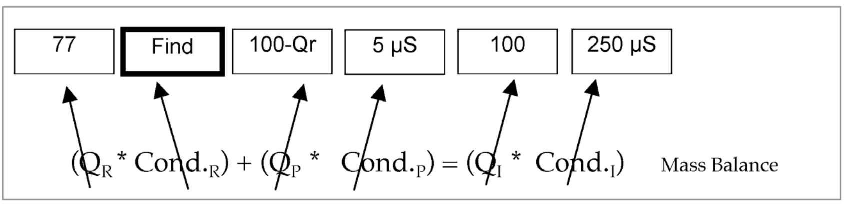
% rejection given as 98% in this example

Find what reject concentration should be at flow proportions given by the flow recovery factor. Look at what is known and solve for the remainder. Using "Q" for flow and "100" for the input flow (since this is only solving for proportions), I for input, P for product, and R for reject:



Depending upon your mood, you can solve this using algebra, or you can set the equation up in a spreadsheet and iterate until solved **(Figure 22-5)**. In this case, the proportion of flow rate implied by conductivity is <u>30.05%</u> reject flow instead of the design intent of 23%. For the presumed 100 gpm inlet flow, this is a continuous 7 gpm flow to waste that is attributable to flow meter error. Over the course of a year this would represent 3.6 million gallons with a cost of $22,000 at $6.00 per kgal.

<u>Step 3: Adjust to Proper Flow Proportion</u>
For proper flow, determine what the conductivity of the reject water "should be" at design flow recovery and adjust to same. The same method applies with a different unknown. Remember, the inlet conductivity and product conductivity will not change with flow, only the reject conductivity will change.
Product conductivity = Inlet conductivity * (1-mineral rejection)
Product conductivity = 250 * (1-0.98) = 5 µS.



This one solved for <u>1070 µS</u> at flow division of 77/23 and 98% mineral rejection. **See Figure 22-5.**

Concentration factor (CF) expresses the ratio of mineral concentration in the reject water compared to inlet feed water. The higher the recovery rate, the higher the concentration factor and the tendency to foul the membrane with scale, so the recovery rate is a balance. Concentration factor is exactly:

CF =Reject concentration/feed water concentration

Recovery rate and concentration factor are cousins and *approximated* by

$$\mathbf{CF = \left[\, {}^{1}/_{(1\text{-}\%\,recovery)} \,\right]}$$

The relationship between concentration ratio and flow ratio is a good approximation when mineral rejection ratios are high, but not exact unless recovery is 100%. Some sources offer convenient RO formulas

Figure 22-5. Sample Calculations to Verify Flows Using Conductivity

Find Flow Proportions from Conductivity

Mineral Rejection %		0.98	given
Reject	QR	**30%**	**← adjust**
	Cond.R	821	measured
	+		
Product	QP	70%	100%-reject flow
	Cond.P	5	measured
	=		
Inlet	QI	100%	nom. 100% flow
	Cond. I	250	measured
			Subtract two sides
		-0.1184	Iterate QR to zero

Find Conductivity for Desired Flow

Flow recovery		0.77	given
Reject	QR	23%	Input-product flow
	Cond.R	**1070**	**← adjust**
	+		
Product	QP	77%	100% * recovery
	Cond.P	5	measured
	=		
Inlet	QI	100%	nom. 100% flow
	Cond. I	250	measured
			Subtract two sides
		-0.05	Iterate Cond.R to zero

that use CF to derive values without a mass balance; while tempting, it is important to remember the resulting values will inherit the inaccuracies of the original assumption of 100% mineral rejection. RO reject concentration factor vs. recovery rate is shown in **Table 22-10**. Note the exponential rise in concentration for reject water. For example, for a 98% mineral rejection factor, a recovery rate of 75% causes the reject flow to have 3.94x the mineral concentration compared to the feed water, and increasing from 75% to 90% recovery more than doubles the concentration. The escalating mineral concentration creates a practical barrier to increased recovery rates. 90% recovery (nearly 10x mineral concentration) is nearly always unattainable. 80% on a very low TDS feed water or softened feed water is possible.

Table 22-10. Concentration Factors from Flow Recovery and Mineral Rejection

	Mineral Rejection Factor										
	1.00	**0.99**	**0.98**	**0.97**	**0.96**	**0.95**	**0.94**	**0.93**	**0.92**	**0.91**	**0.90**
Flow Recovery	Conc. Factor	Conc. Factor	Conc. Factor	Conc. Factor	Conc. Factor	Conc. Factor	Conc. Factor	Conc. Factor	Conc. Factor	Conc. Factor	Conc. Factor
30%	1.43	1.42	**1.42**	1.42	1.41	1.41	1.40	1.40	1.39	1.39	1.39
35%	1.54	1.53	**1.53**	1.52	1.52	1.51	1.51	1.50	1.50	1.49	1.48
40%	1.67	1.66	**1.65**	1.65	1.64	1.63	1.63	1.62	1.61	1.61	1.60
45%	1.82	1.81	**1.80**	1.79	1.79	1.78	1.77	1.76	1.75	1.74	1.74
50%	2.00	1.99	**1.98**	1.97	1.96	1.95	1.94	1.93	1.92	1.91	1.90
55%	2.22	2.21	**2.20**	2.19	2.17	2.16	2.15	2.14	2.12	2.11	2.10
60%	2.50	2.49	**2.47**	2.46	2.44	2.43	2.41	2.40	2.38	2.37	2.35
65%	2.86	2.84	**2.82**	2.80	2.78	2.76	2.75	2.73	2.71	2.69	2.67
70%	3.33	3.31	**3.29**	3.26	3.24	3.22	3.19	3.17	3.15	3.12	3.10
75%	4.00	3.97	**3.94**	3.91	3.88	3.85	3.82	3.79	3.76	3.73	3.70
80%	5.00	4.96	**4.92**	4.88	4.84	4.80	4.76	4.72	4.68	4.64	4.60
85%	6.67	6.61	**6.55**	6.50	6.44	6.38	6.33	6.27	6.21	6.16	6.10
90%	10.0	9.91	**9.82**	9.73	9.64	9.55	9.46	9.37	9.28	9.19	9.10
95%	20.0	19.8	**19.6**	19.4	19.2	19.1	18.9	18.7	18.5	18.3	18.1

Membrane materials determine the RO unit's ability to allow movement of water vs. the dissolved salts, as well as its compatibility with the environment in which it is placed, especially pH and temperature limits. RO membrane service life depends on feed water, pretreatment, and service conditions. The commonly stated life of RO membranes is

3-5 years although this will depend upon service conditions. During its service life, the membrane performance will decline, with one estimate being 20% in three years (Source: *Design Parameters Affecting Performance*, Hydranautics/Denko Corp, 2001).

Fouling reduces performance and is a design and operation challenge. In general, RO fouling comes from one or more of these:

- Silt (suspended particulates/mud on the membrane)
- Bio (biological growth on the membrane)
- Organic (e.g. oil or grease on the membrane)

The potential for fouling can be predicted by specific water testing techniques which are mentioned here for completeness, but not fully described

- Silt fouling potential is indicated by a high SDI value, **(silt density index.) SDI** is determined by flowing water through a 0.45 micron membrane and measuring pressure drop accumulation. Most if not all membrane manufacturers list maximum SDI of <5.
- Scaling potential is determined by detailed water analysis and concentration, (recovery rate.) This is the **Langeliers saturation index, (LSI.)** LSI determines how high of a recovery can be achieved without reaching the point of saturation for any of the ions. LSI takes into account temperature and pH.

Scaling is from precipitation of dissolved minerals or salts onto the membrane. The greater the concentration the greater the tendency to foul by scaling. As more and more water is removed, the dissolved materials eventually precipitate out onto the membrane surface, forming scale.

CIP (clean-in-place) procedures use ancillary equipment to circulate strong chemicals to clean the membrane. For example, one method would begin with a high pH (caustic), 10.5 - 11.0 pH, to remove the fouling, followed by low pH (acid) to remove any scale. Repeated cycles of chemical cleaning reduce the membrane effectiveness. Other things that take a toll on membranes are material detriment from high or low pH and general particulate fouling. pH limitations of membrane material are coordinated with feed water chemistry, required water treatment (pH adjustment) and cleaning methods to assure long service life.

Pre-treatment varies according to feed water. In most/all cases there will be filtration for suspended solid particulate removal (1-5 micron capture size) and can be augmented with flocculants that encourage small particles to stick together and form bigger particles to be more easily removed by sand filtration or centrifugal separators, although flocculants will quickly block cartridge filters. Scaling inhibitors are helpful in achieving higher recovery rates, by reducing the tendency for calcium scale to precipitate out and deposit on the membrane surface. Other pre-treatments may include pH adjustment, softening, and temperature control. Integral to scale or fouling control is maintaining a minimum velocity across the membrane. This forms the basis for minimum reject flow rates and maximum recovery rates. Langelier saturation index or LSI will pre-determine the maximum concentration of the feed water at a given temperature and pH.

Water savings vs. operational costs

Efforts to conserve water by raising recovery rates increase concentration levels in the reject water and encourage scale fouling with associated increase in maintenance cost to clean the membrane and possibly reduced membrane service life. Conversely, operating with generous amounts of reject flow may be tempting for reduced maintenance cost but do so by incurring higher water costs. In some cases, high concentrations of minerals in reject water can result in additional wastewater treatment costs (high TDS), eroding water cost savings. Generally, identifying the highest recovery rate without undue maintenance cost and without premature membrane failure will yield good overall economy. The relative cost of water vs. maintenance will identify the proper economic balance. Membrane manufacturers usually provide a software program that uses an accurate feed water analysis to project the safe and recommended recovery rate in each individual application.

RO Piping Configurations

The basic RO can be combined and nested. Stages and passes can be integral to a pre-manufactured single RO unit or built-up with separate RO units and inter-connecting field piping. The benefits of series and parallel piping can be efficiency, performance or redundancy.

Multiple Stages (Concentrate Staging). Increasing the number of **stages** puts the reject flow of one RO process as the inlet of another one. **See Figure 22-6.** The overall waste flow is reduced, increasing the recovery rate. With each successive stage, the inlet flow is reduced (because it is the reject flow of the stage before it) so the area of membrane is reduced to maintain necessary water velocity. Traditional design reduces the

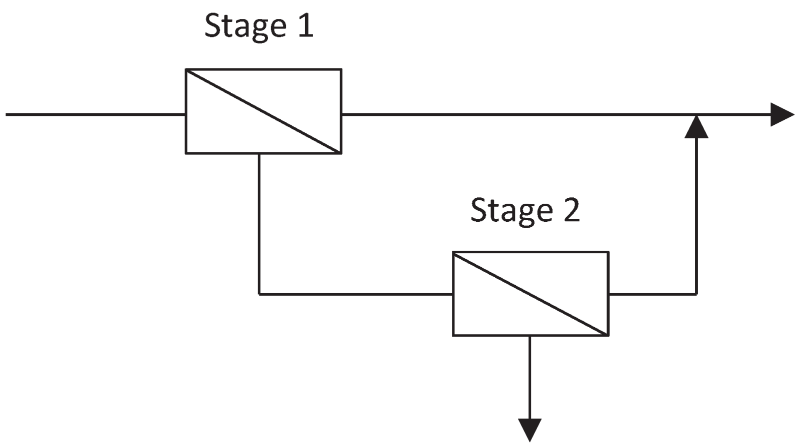


Figure 22-6. Two stage RO Configuration

number of elements by half for each successive stages to accomplish this. Also, with each successive stage the water conditions become more concentrated. Additional passes meet diminishing returns. Careful design with vendor software is needed to assess water conditions at each point for reliable operation.

> *Note: Multi-stage RO requires increased pressure to the first stage to allow sufficient pressure for the second stage and so on; the higher overall recovery and reduced water consumption is gained at the expense of higher electrical cost.*

Multiple Passes. Increasing the number of **passes** puts RO units in series, with the outlet of one RO process becoming the feed water to the next one. This can be used to amplify the mineral rejection fraction of the RO process. **See Figure 22-7.** Since the second pass has low mineral concentration to begin with, the reject water quality will be as good as or better than the feed water to the first pass and is commonly recycled. When second pass reject water is returned to first pass feed water, that amount of water subtracts from purchased feed water. The recycled water flow is not large, but does form a runaround that never leaves the system and so adjusting for high recovery rates in the second pass (e.g. 90%) will additionally increase RO product (output) flow.

> *Note: Two pass RO requires increased pressure to the first pass to allow sufficient pressure for the second pass, and higher product water quality is gained at the expense of higher electrical cost. In the case of an RO*

two-pass unit pre-treating a high purity DI unit, savings in water from reduced DI regeneration can justify the increase in electricity cost.

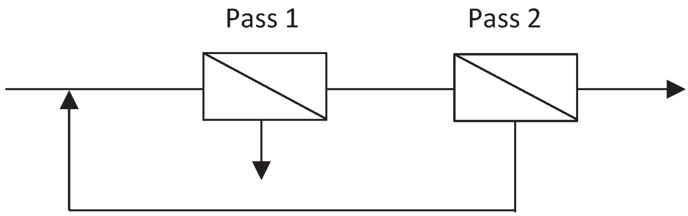


Figure 22-7. Two pass RO Configuration

Overall Recovery Rate is the water efficiency metric for RO production when there is more than one unit and they are interconnected. Whether series, parallel, blending, etc. this requires knowing only the *overall inputs and output flows.* Where multiple intermingled units exist, the overall inputs and outputs are found by drawing a box around all the collective RO system, to determine the product and reject flow values leaving the box. See "**Box Method**" in this chapter. Then:

Overall Recovery
= Total Output Flow/Total Input Flow, or
= Total Output Flow/(Total Output Flow + Total Reject Flow)

Quantifying RO Performance

Accurate RO performance assessment requires assistance from the vendor or software and requires water analysis. Minimum velocity across the membranes is a key factor in preventing fouling, and minimum operating flows are defined.

Useful RO Formulas

Terms

Product water = permeate, output (different names, same thing)
Reject = concentrate, waste (different names, same thing)
ppm = parts per million
GPG = grains per gallon
mg/L = milligrams per liter

Abbreviations

Concentration = conc.
Conductivity = μS

Flow Recovery (portion of the inlet water makes it to output)

% flow recovery = product flow (output)/inlet flow

% flow recovery = (inlet flow – reject flow)/inlet flow

Mineral Rejection Fraction (portion of inlet water minerals removed from product water)

% rejection = 100% - (product conc./inlet conc.)

% rejection = (inlet conc. – product conc.)/inlet conc.

Concentration Factor (CF)

(CF=how much more concentrated the reject is vs. inlet)

Concentration factor = reject conc./inlet conc.

Concentrations by ratios

Product (output) concentration = inlet conc. * (1-mineral rejection fraction)

Conductivity

Conductivity (μS) can be in any consistent units

% Mineral rejection fraction = (inlet μS– product μS)/inlet μS

Concentration factor = reject μS/inlet μS

Concentration factor = (reject μS– product μS)/(inlet μS - product μS)

Conversion factors

7000 grains = 1 pound

1 GPG = 1 pound per 7000 gallons

1 GPG = ppm/17.14 (17.13761522) **A**

1 GPG = mg/L/17.12 (17.11806121) **B**

Factor **F** = **A/B** = 1.0011423

1 ppm = (mg/l)/**F** (essentially interchangeable)

1 ppm = 0.05835 GPG (1/**A**)

1 mg/L = 0.05842 GPG (1/**B**)

1 mg/L = ppm* **F** (essentially interchangeable)

1 Liter = 0.26417 gallons

1 gallon = 3.785 Liters

1 gallon = 8.34 lbs

1 M^3 = 264 gallons

1 ft^3 water = 7.48 gallons

Conductivity

Conductivity µS/cm2	Resistivity Meg-Ohm/ cm2
0.056	**18.0**
0.063	**16.0**
0.071	**14.0**
0.083	**12.0**
0.100	**10.0**
0.125	**8.00**
0.167	**6.00**
0.250	**4.00**
0.500	**2.00**
1.00	**1.00**
1.33	**0.750**
2.00	**0.500**
3.00	**0.333**
4.00	**0.250**
5.00	**0.200**
10.0	**0.100**
20.0	**0.0500**
30.0	**0.0333**
40.0	**0.0250**
50.0	**0.0200**
100	**0.0100**
200	**0.00500**
300	**0.00333**
400	**0.00250**
500	**0.00200**
1,000	**0.00100**
2,000	**0.000500**
3,000	**0.000333**
4,000	**0.000250**
5,000	**0.000200**
10,000	**0.000100**

Concentration Equivalence

TDS ppm CaCO3	TDS mg/L CaCO3	TDS GPG
0.0500	**0.0501**	**0.00292**
0.100	**0.100**	**0.00585**
0.200	**0.200**	**0.0117**
0.300	**0.300**	**0.0175**
0.400	**0.400**	**0.0234**
0.500	**0.501**	**0.0292**
1.00	**1.00**	**0.0585**
2.00	**2.00**	**0.117**
3.00	**3.00**	**0.175**
4.00	**4.00**	**0.234**
5.00	**5.01**	**0.292**
10.0	**10.0**	**0.585**
11.0	**11.0**	**0.643**
12.0	**12.0**	**0.702**
13.0	**13.0**	**0.760**
14.0	**14.0**	**0.819**
15.0	**15.0**	**0.877**
20.0	**20.0**	**1.17**
30.0	**30.0**	**1.75**
40.0	**40.0**	**2.34**
50.0	**50.1**	**2.92**
100	**100**	**5.85**
200	**200**	**11.7**
300	**300**	**17.5**
400	**400**	**23.4**
500	**501**	**29.2**
1,000	**1,001**	**58.5**
2,000	**2,002**	**117**
3,000	**3,003**	**175**
4,000	**4,005**	**234**
5,000	**5,006**	**292**
10,000	**10,011**	**585**

Figure 22-8. Conductivity and Concentration Values

Charts are independent.
Do not read across to connect rows, e.g. do not correlate concentration with conductivity

Figure 22-9. Concentration Value Correlation to Conductivity Italicized Columns: It is desirable to link conductivity (μS/cm) and **TDS (ppm) but no exact factor exists because it depends on what is dissolved.** Experimental values are sometimes used to approximate the link from conductivity to ppm. The approximations are site and water specific and not universal. For water with high sodium chloride content, the conversion is about 0.5, e.g. 0.5*(μS/cm) = ppm; for other water other constants are used depending on what is in the water. One way to arrive at the correlation is to use utility water quality reports or other reports and assume their instruments are accurate; however, water sources do change.

Concentration Equivalence at Given Correlation of Conductivity

Conductivity		ppm/µS= 0.5			ppm/µS= 0.55			ppm/µS= 0.60			ppm/µS= 0.65		
Conductivity µS/cm2	Resistivity Meg-Ohm/cm2	TDS ppm CaCO3	TDS mg/L CaCO3	TDS GPG	TDS ppm CaCO3	TDS mg/L CaCO3	TDS GPG	TDS ppm CaCO3	TDS mg/L CaCO3	TDS GPG	TDS ppm CaCO3	TDS mg/L CaCO3	TDS GPG
0.056	**18.0**	0.0278	0.0278	0.00162	0.0306	0.0306	0.00179	0.0333	0.0334	0.00195	0.0361	0.0362	0.00211
0.063	**16.0**	0.0313	0.0313	0.00183	0.0344	0.0344	0.00201	0.0375	0.0375	0.00219	0.0406	0.0407	0.00238
0.071	**14.0**	0.0357	0.0358	0.00209	0.0393	0.0393	0.00230	0.0429	0.0429	0.00251	0.0464	0.0465	0.00272
0.083	**12.0**	0.0417	0.0417	0.00244	0.0458	0.0459	0.00268	0.0500	0.0501	0.00292	0.0542	0.0542	0.00317
0.100	**10.0**	0.0500	0.0501	0.00292	0.0550	0.0551	0.00322	0.0600	0.0601	0.00351	0.0650	0.0651	0.00380
0.125	**8.00**	0.0625	0.0626	0.00366	0.0688	0.0688	0.00402	0.0750	0.0751	0.00439	0.0813	0.0813	0.00475
0.167	**6.00**	0.0833	0.0834	0.00487	0.0917	0.0918	0.00536	0.1000	0.1001	0.00585	0.1083	0.1085	0.00634
0.250	**4.00**	0.125	0.125	0.00731	0.138	0.138	0.00804	0.150	0.150	0.00877	0.163	0.163	0.00950
0.500	**2.00**	0.250	0.250	0.0146	0.275	0.275	0.0161	0.300	0.300	0.0175	0.325	0.325	0.0190
1.00	**1.00**	0.500	0.501	0.0292	0.550	0.551	0.0322	0.600	0.601	0.0351	0.650	0.651	0.0380
1.33	**0.750**	0.667	0.667	0.0390	0.733	0.734	0.0429	0.800	0.801	0.0468	0.867	0.868	0.0507
2.00	**0.500**	1.00	1.00	0.0585	1.10	1.10	0.0643	1.20	1.20	0.0702	1.30	1.30	0.0760
3.00	**0.333**	1.50	1.50	0.0877	1.65	1.65	0.0965	1.80	1.80	0.1053	1.95	1.95	0.1140
4.00	**0.250**	2.00	2.00	0.117	2.20	2.20	0.129	2.40	2.40	0.140	2.60	2.60	0.152
5.00	**0.200**	2.50	2.50	0.146	2.75	2.75	0.161	3.00	3.00	0.175	3.25	3.25	0.190
10.0	**0.100**	5.00	5.01	0.292	5.50	5.51	0.322	6.00	6.01	0.351	6.50	6.51	0.380
20.0	**0.0500**	10.0	10.0	0.585	11.0	11.0	0.643	12.0	12.0	0.702	13.0	13.0	0.760
30.0	**0.0333**	15.0	15.0	0.877	16.5	16.5	0.965	18.0	18.0	1.053	19.5	19.5	1.140
40.0	**0.0250**	20.0	20.0	1.17	22.0	22.0	1.29	24.0	24.0	1.40	26.0	26.0	1.52
50.0	**0.0200**	25.0	25.0	1.46	27.5	27.5	1.61	30.0	30.0	1.75	32.5	32.5	1.90
100	**0.0100**	50.0	50.1	2.92	55.0	55.1	3.22	60.0	60.1	3.51	65.0	65.1	3.80
200	**0.00500**	100	100	5.85	110	110	6.43	120	120	7.02	130	130	7.60
300	**0.00333**	150	150	8.77	165	165	9.65	180	180	10.53	195	195	11.40
400	**0.00250**	200	200	11.7	220	220	12.9	240	240	14.0	260	260	15.2
500	**0.00200**	250	250	14.6	275	275	16.1	300	300	17.5	325	325	19.0
1,000	**0.00100**	500	501	29.2	550	551	32.2	600	601	35.1	650	651	38.0
2,000	**0.000500**	1,000	1,001	58.5	1,100	1,101	64.3	1,200	1,201	70.2	1,300	1,301	76.0
3,000	**0.000333**	1,500	1,502	87.7	1,650	1,652	96.5	1,800	1,802	105.3	1,950	1,952	114.0
4,000	**0.000250**	2,000	2,002	117	2,200	2,203	129	2,400	2,403	140	2,600	2,603	152
5,000	**0.000200**	2,500	2,503	146	2,750	2,753	161	3,000	3,003	175	3,250	3,254	190
10,000	**0.000100**	5,000	5,006	292	5,500	5,506	322	6,000	6,007	351	6,500	6,507	380

RO Reject Waste Water

A chief complaint for water cost control related to reverse osmosis is the reject water. Many attempts have been made to reduce this flow, although it is in the nature of the process. When the RO unit is amidst other water-using equipment that doesn't mind the minerals, reuse of the reject water is an elegant and cost effective approach. In reusing RO reject it is no longer wasted water because it replaces purchased water use in another process.

If reuse opportunities do not exist, it is also possible to utilize a separate RO unit to re-process the reject water, reducing the waste flow. The higher dissolved mineral content of this "clean up" or "roughing" RO feed water compared to regular feed water means membrane material may be different and maintenance will be higher and membrane service life will be lower. An example of a 'roughing' membrane a lower cost membrane with a lower mineral rejection such as 80-90%. These could serve as roughing duty for reusing waste water from RO reject or other waste water sources. Alternately, expired pure water RO membranes can be re-purposed for roughing work before discarding. Reclaiming waste water requires strict filtration at the micron level like any RO feed water and will require increased cleaning. Care must be used to determine what additional contaminants may be present in the waste water before choosing to reuse it. These may include oils, grease, and organic contaminants from other processes.

Electricity Use in Reverse Osmosis

See also **Appendix A "Embedded Energy in Water and Waste Water"**

Units are kWh input per 1000 gallons RO *output*. Variability comes from pumping energy, throttling means for part load conditions, auxiliary equipment such as pre-treatment and distribution, and energy recovery in some larger installations. Fundamental pump energy varies according to water contents, for example, sea water has a much higher osmotic pressure than potable water and so the reverse osmosis effect requires a much higher pressure. Between potable water purifying and sea water desalination, the difference in pump pressure can be from 200 psi to 1000 psi respectively. Inland brackish water pressures are in between. The example is provided to illustrate the calculation method.

Example of RO pump energy: Potable water purification for process, two pass RO, 75% recovery, 105 gpm product (output) flow using 44a, 480V, three phase power. This is approximately 29 kW. On a kW per-kgal basis, 1000/105/60=0.16 hours to produce one kgal, and 29*0.16=4.6 kW per 1000 gallons per hour (gph) flow rate, or 4.6 kWh per kgal of finished water.

When water is purchased, the water costs per gallon produced will be higher than electric costs. Using the above example with 70% recovery, water cost at $5.00/kgal and electricity cost at $0.07 per kWh:

1000 gallons of product (output) requires 1 kgal/0.7= 1.4 kgal feed water which costs $5.00.

5 kW per kgal of finished water * 0.07 = $0.35.

For this example, water cost is >90% of the combined electric+water charge

Water-saving measures for water purification equipment

1. Reduce demand for purified water volume.
2. Integrate source measures with end use reduction measures, to allow downsizing of equipment.
3. Reduce strictness requirements for the purification levels
4. Choose water purification technology options with the smallest waste water signature (gallons waste per 1000 gallons finished water).
5. Reuse discarded water from one process in a lower tier process that can accept the higher mineral content, such as cooling tower, scrubber, evaporative cooler, non-critical wash water, or irrigation
6. Resin Bed Softeners and Deionizers
   - Regenerate when needed instead of on a timed basis.
   - Measure and verify water use does not exceed normal values (Figure 22-2A, B).
   - Evaluate overall efficiency including pre-treatment water use that exchange water loss at upstream step for water savings at downstream step
7. Reverse Osmosis
   - Increase recovery rate to maximum allowable concentration of the reject water. This may invoke the use of scale inhibitors.
   - Cross check reject and product flow ratios with conductivity meters to assure the flows and recovery rates are what they should be.

- Use multi-stage RO equipment to re-use the RO reject water and increase overall recovery factor

8. Reverse Osmosis (electrical savings)
   - Use variable pump speed rather than head dissipation, to reduce throttling losses
   - Avoid over-pressurization and control to no higher than manufacturer's requirements
   - Select membranes that require lower pressure for each unit of product flow
   - For high pressure applications, such as seawater desalination, use energy recovery from decompressing water

WATER EFFICIENCY FOR MECHANICAL COOLING SYSTEMS

Water-Cooled HVAC and Refrigeration

The refrigeration cycle of mechanical cooling systems are almost universally more efficient when cooled with water rather than air. Once through water cooling allows condensing temperatures near the temperature of the water source which may be city water, river water, sea water, etc. When city water is used, once through water cooling is expensive and places large demands upon local water supplies so, to conserve water, most cooling systems use a cooling tower which evaporates a portion of the water to cool the rest of the water. With a cooling tower, the water temperature is a function of wet bulb temperature which is lower than dry bulb temperature. The amount of savings over air-cooled systems depends upon the wet bulb depression, which is determined from weather tables for the given location (psychrometrics). Aiding the saving of water-cooled systems are the reduced heat exchanger approach values compared to tube-in-air heat exchangers and higher economizer hours where cooling occurs without a compressor. Subtracting from the savings of water-cooled systems are the ancillary pumps used to move the water around. Specific power for a cooling tower fan is similar to that of an air-cooled condenser.

Once Through Cooling Water Systems

Few large cooling systems use once through water cooling, but many small refrigeration systems do (walk in coolers, ice machines). Converting these to a shared cooling tower can produce the percent water savings noted in the cooling tower example.

Example: Water savings potential of a cooling tower vs. once through water cooling
100 ton cooling load heat rejection
With heat of compression factor of 1.25, this is 1.5 MMBtuh (1000*12,000*1.25)
Option 1: Straight through cooling, with a 40F temperature rise
Note: Q●= heat flow rate, m●=mass flow rate
[Q●=m● c dT, m●=Q●/c dT, c=1]
Water used = 1.5EE6/40 = 37,500 lbs/hour = 74 gpm

<u>Option 2: Cooling tower</u>, with 4 cycles of concentration
[E=Q/1000 Btu/lb] = 1.5 MMBtuh/1000 = 1500 lbs/hr = 3 gpm
[BD=E/(cycles-1)] = 3/3 = 1.0 gpm
Water used = 4.0 gpm
Note: This explains a design rule of thumb for cooling tower water use of 0.04 gpm per ton
Water use reduction = 74-4 = 70 gpm.
Water use % reduction = 70/74 = <u>94% reduction by using a cooling tower</u>.

Evaporative Cooling

Evaporating water requires heat, about 1000 Btu/lb give or take. Evaporative cooling uses the cooling effect (heat absorption) of evaporating water to cool an air stream, and these systems are used to create a cool air source. The air is cooled adiabatically (no change in total heat), exchanging air temperature for moisture content. (See also **Chapter 11 "Evaporative Cooling"** and the psychrometric explanations).

Direct use of the cooled air supply can be for air conditioning or spot cooling in drier climates—high humidity areas have little benefit from evaporative cooling. Using the cool and moist air as-is is termed "**direct evaporative cooling**." "**Indirect evaporative cooling**" is essentially a cooling tower, using the evaporative effect to cool water which then cools the air sensibly with a heat exchanger. As with all heat exchangers, heat transfer relies on a differential temperature and approach which limits the hours the effect can be utilized.

Sometimes HVAC designs combine indirect and direct evaporative cooling processes in series and are intuitively named "**indirect-direct evaporative cooling**" with the indirect step being the first of the two; indirect-direct systems are capable of achieving lower discharge temperatures and have less moisture content than direct evaporation and less 'humid feeling' occupant complaints when used for comfort cooling.

Direct evaporative cooling also finds use as an **evaporative pre-cooler** for conventional air-cooled equipment, so the equipment "thinks" it is cooler outside than it really is. The conventional air-cooled equipment is fitted with evaporative pads or spray nozzles to pre-cool the air. Care must be taken to prevent water from wetting the air-cooled heat exchanger to prevent corrosion issues. Each evaporative cooling applications can be used to supplant or eliminate mechanical cooling (vapor compression) and provide considerable energy savings, in exchange for water consumption. Evaporative pre-cooling is also used to reduce capacity loss in combustion turbine generators by cooling the air and would be considered for other air-cooled or air-consuming equipment that suffers capacity or efficiency degradation on hot dry days.

Cost Tradeoff between Decreased Electricity Use and Increased Water Use

Water is used frequently to make mechanical cooling systems more efficient. There is an investment in water/waste water cost for a return in electric savings. The tradeoff methodology is akin to fuel switching projects where it is all about dollars.

The lower refrigeration cycle temperatures from cooling with water instead of air create energy savings which can be considerable in summer and whenever the ambient air wet bulb temperature is lower than dry bulb temperature. Sometimes, in dry climates, the evaporative cooling effect is utilized directly, as in 'swamp coolers'. Here the energy use of mechanical cooling apparatus is completely supplanted and cooling burden in kW/ton is nearly zero. Cooling systems with economizers will see additional hours of 'free cooling' each year for water-cooled systems. While there is no doubt water-cooled HVAC&R makes saves energy in cooling systems, it may not be clear how much of the energy savings are given up to water and waste water costs. Even when energy savings estimates are accurate, dollar savings will be over-stated unless water and waste water costs are factored in.

If energy savings calculations also include corresponding water and waste water use for each hour or weather bin, then the tradeoff in dollars can be identified directly and this is the preferred approach.

Direct calculations are more accurate because they account for weather and load variations, and economizer operation.

Generalizations and quick views are possible for various systems but are only as accurate as the assumptions used. If a value of gallons water per kWh can be reasonably determined, the applicable cost factors for water, waste water, and electricity can determine the financial merit of the proposal. And it requires a crystal ball to determine what utility rate changes will occur during the life of the measure. A convenient way to measure the give-and-take of water for efficiency proposals is shown in **Figure 22-10,** with results being "percent of electric savings given up to water and waste water costs."

Note: The challenging part about using ***Figure 22-10*** *is determining a reasonable value of gallons per kWh saved. For cooling, this requires an estimate of specific water use (gallons per ton-hour) and an alternative design (e.g. air cooled) that does not use water. There is a differential kWh (water cooled using less) and a value of water use that must be paid for with the electric savings. The ratio of the two becomes the indicator of gallons per kWh savings. For the water-cooled option, water consumption will be proportional to the load and should not vary; however for the air-cooled option the seasonal efficiency will vary with outdoor ambient temperature, so a seasonal weighting will improve accuracy.*

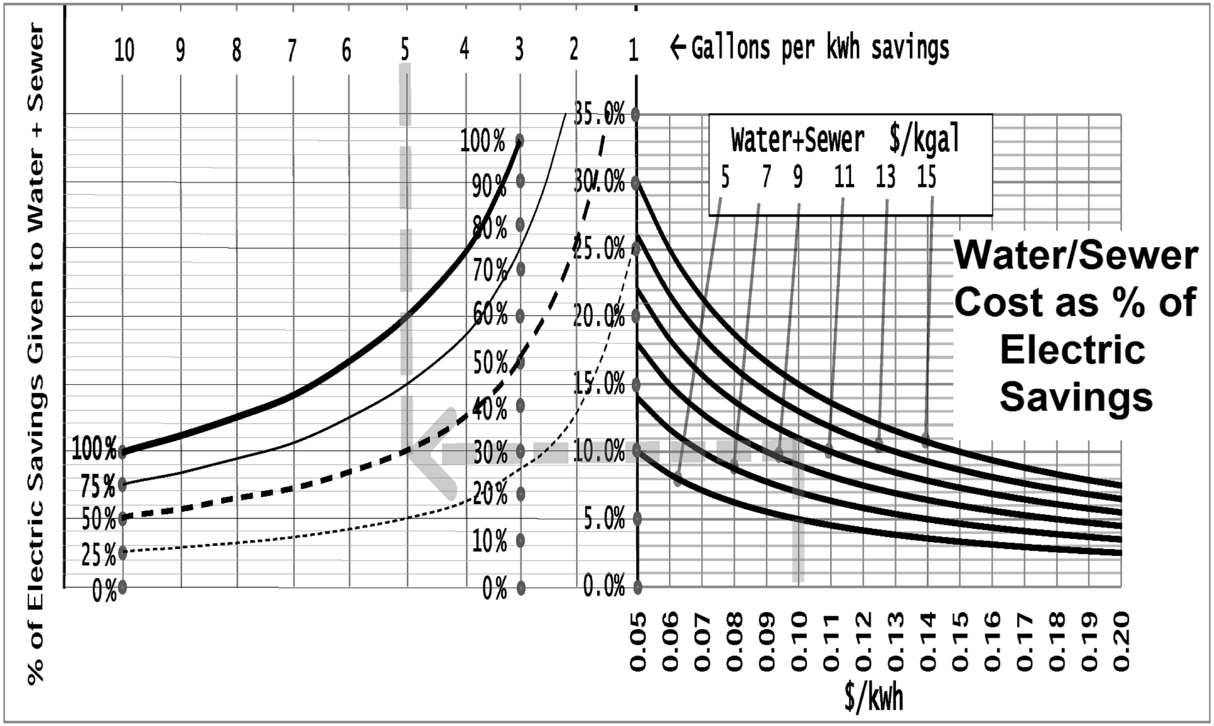
Table 22-11 includes representative values of gallons per kWh saved for various systems.

Since utility rates change and are different by location, conclusions on what makes sense today may change over time. Clearly, when electricity is expensive and water is inexpensive water-cooling to improve efficiency is a good deal, but the same energy efficiency measure applied where the relative costs of the two were reversed would yield a 'no' instead of a 'yes' for the proposal.

Figure 22-10. Percent of Electricity Savings Given to Water and Waste Water Cost

1. Enter the chart with the cost per kWh and intersect the combined cost per kgal (1000 gal) for water + sewer.
2. Proceed to the left to intersect the vertical line gallons per kWh savings.
3. Read the percent of electric savings lost to water and waste water cost.

Example shown is for $0.10 per kWh electric cost, $10.00 per 1000 gallons water and sewer cost, and 5 gallons water used per kWh saved. Here, 50% of the electricity savings are lost to water and sewer charges.



Data used for **Figure 22-10** (continued)

Formula: **% of electric savings lost to water + sewer charges**
= $ Water and Sewer/$ Electric
= (G/1000 * W)/E

Where:
G = Gal/kWh
W=$/kgal for Water+Sewer
E = $/kWh

When there are different water rates for summer and winter*: Use summer rates for measures that use water primarily in summer. For measures that use water throughout the year either use an average annual water/sewer rate or evaluate summer and winter separately.*

1 Gal/kWh

Elec cost	Water/Sewer Cost $/kgal										
$/kWh	5	6	7	8	9	10	11	12	13	14	15
0.05	10.0%	12.0%	14.0%	16.0%	18.0%	20.0%	22.0%	24.0%	26.0%	28.0%	30.0%
0.06	8.3%	10.0%	11.7%	13.3%	15.0%	16.7%	18.3%	20.0%	21.7%	23.3%	25.0%
0.07	7.1%	8.6%	10.0%	11.4%	12.9%	14.3%	15.7%	17.1%	18.6%	20.0%	21.4%
0.08	6.3%	7.5%	8.8%	10.0%	11.3%	12.5%	13.8%	15.0%	16.3%	17.5%	18.8%
0.09	5.6%	6.7%	7.8%	8.9%	10.0%	11.1%	12.2%	13.3%	14.4%	15.6%	16.7%
0.10	5.0%	6.0%	7.0%	8.0%	9.0%	10.0%	11.0%	12.0%	13.0%	14.0%	15.0%
0.11	4.5%	5.5%	6.4%	7.3%	8.2%	9.1%	10.0%	10.9%	11.8%	12.7%	13.6%
0.12	4.2%	5.0%	5.8%	6.7%	7.5%	8.3%	9.2%	10.0%	10.8%	11.7%	12.5%
0.13	3.8%	4.6%	5.4%	6.2%	6.9%	7.7%	8.5%	9.2%	10.0%	10.8%	11.5%
0.14	3.6%	4.3%	5.0%	5.7%	6.4%	7.1%	7.9%	8.6%	9.3%	10.0%	10.7%
0.15	3.3%	4.0%	4.7%	5.3%	6.0%	6.7%	7.3%	8.0%	8.7%	9.3%	10.0%
0.16	3.1%	3.8%	4.4%	5.0%	5.6%	6.3%	6.9%	7.5%	8.1%	8.8%	9.4%
0.17	2.9%	3.5%	4.1%	4.7%	5.3%	5.9%	6.5%	7.1%	7.6%	8.2%	8.8%
0.18	2.8%	3.3%	3.9%	4.4%	5.0%	5.6%	6.1%	6.7%	7.2%	7.8%	8.3%
0.19	2.6%	3.2%	3.7%	4.2%	4.7%	5.3%	5.8%	6.3%	6.8%	7.4%	7.9%
0.20	2.5%	3.0%	3.5%	4.0%	4.5%	5.0%	5.5%	6.0%	6.5%	7.0%	7.5%

2 Gal/kWh

Elec cost	Water/Sewer Cost $/kgal										
$/kWh	5	6	7	8	9	10	11	12	13	14	15
0.05	20.0%	24.0%	28.0%	32.0%	36.0%	40.0%	44.0%	48.0%	52.0%	56.0%	60.0%
0.06	16.7%	20.0%	23.3%	26.7%	30.0%	33.3%	36.7%	40.0%	43.3%	46.7%	50.0%
0.07	14.3%	17.1%	20.0%	22.9%	25.7%	28.6%	31.4%	34.3%	37.1%	40.0%	42.9%
0.08	12.5%	15.0%	17.5%	20.0%	22.5%	25.0%	27.5%	30.0%	32.5%	35.0%	37.5%
0.09	11.1%	13.3%	15.6%	17.8%	20.0%	22.2%	24.4%	26.7%	28.9%	31.1%	33.3%
0.10	10.0%	12.0%	14.0%	16.0%	18.0%	20.0%	22.0%	24.0%	26.0%	28.0%	30.0%
0.11	9.1%	10.9%	12.7%	14.5%	16.4%	18.2%	20.0%	21.8%	23.6%	25.5%	27.3%
0.12	8.3%	10.0%	11.7%	13.3%	15.0%	16.7%	18.3%	20.0%	21.7%	23.3%	25.0%
0.13	7.7%	9.2%	10.8%	12.3%	13.8%	15.4%	16.9%	18.5%	20.0%	21.5%	23.1%
0.14	7.1%	8.6%	10.0%	11.4%	12.9%	14.3%	15.7%	17.1%	18.6%	20.0%	21.4%
0.15	6.7%	8.0%	9.3%	10.7%	12.0%	13.3%	14.7%	16.0%	17.3%	18.7%	20.0%
0.16	6.3%	7.5%	8.8%	10.0%	11.3%	12.5%	13.8%	15.0%	16.3%	17.5%	18.8%
0.17	5.9%	7.1%	8.2%	9.4%	10.6%	11.8%	12.9%	14.1%	15.3%	16.5%	17.6%
0.18	5.6%	6.7%	7.8%	8.9%	10.0%	11.1%	12.2%	13.3%	14.4%	15.6%	16.7%
0.19	5.3%	6.3%	7.4%	8.4%	9.5%	10.5%	11.6%	12.6%	13.7%	14.7%	15.8%
0.20	5.0%	6.0%	7.0%	8.0%	9.0%	10.0%	11.0%	12.0%	13.0%	14.0%	15.0%

Data used for **Figure 22-10** (continued)

3
Gal/kWh

Elec cost	Water/Sewer Cost $/kgal										
$/kWh	5	6	7	8	9	10	11	12	13	14	15
0.05	30.0%	36.0%	42.0%	48.0%	54.0%	60.0%	66.0%	72.0%	78.0%	84.0%	90.0%
0.06	25.0%	30.0%	35.0%	40.0%	45.0%	50.0%	55.0%	60.0%	65.0%	70.0%	75.0%
0.07	21.4%	25.7%	30.0%	34.3%	38.6%	42.9%	47.1%	51.4%	55.7%	60.0%	64.3%
0.08	18.8%	22.5%	26.3%	30.0%	33.8%	37.5%	41.3%	45.0%	48.8%	52.5%	56.3%
0.09	16.7%	20.0%	23.3%	26.7%	30.0%	33.3%	36.7%	40.0%	43.3%	46.7%	50.0%
0.10	15.0%	18.0%	21.0%	24.0%	27.0%	30.0%	33.0%	36.0%	39.0%	42.0%	45.0%
0.11	13.6%	16.4%	19.1%	21.8%	24.5%	27.3%	30.0%	32.7%	35.5%	38.2%	40.9%
0.12	12.5%	15.0%	17.5%	20.0%	22.5%	25.0%	27.5%	30.0%	32.5%	35.0%	37.5%
0.13	11.5%	13.8%	16.2%	18.5%	20.8%	23.1%	25.4%	27.7%	30.0%	32.3%	34.6%
0.14	10.7%	12.9%	15.0%	17.1%	19.3%	21.4%	23.6%	25.7%	27.9%	30.0%	32.1%
0.15	10.0%	12.0%	14.0%	16.0%	18.0%	20.0%	22.0%	24.0%	26.0%	28.0%	30.0%
0.16	9.4%	11.3%	13.1%	15.0%	16.9%	18.8%	20.6%	22.5%	24.4%	26.3%	28.1%
0.17	8.8%	10.6%	12.4%	14.1%	15.9%	17.6%	19.4%	21.2%	22.9%	24.7%	26.5%
0.18	8.3%	10.0%	11.7%	13.3%	15.0%	16.7%	18.3%	20.0%	21.7%	23.3%	25.0%
0.19	7.9%	9.5%	11.1%	12.6%	14.2%	15.8%	17.4%	18.9%	20.5%	22.1%	23.7%
0.20	7.5%	9.0%	10.5%	12.0%	13.5%	15.0%	16.5%	18.0%	19.5%	21.0%	22.5%

4
Gal/kWh

Elec cost	Water/Sewer Cost $/kgal										
$/kWh	5	6	7	8	9	10	11	12	13	14	15
0.05	40.0%	48.0%	56.0%	64.0%	72.0%	80.0%	88.0%	96.0%	**104%**	**112%**	120%
0.06	33.3%	40.0%	46.7%	53.3%	60.0%	66.7%	73.3%	80.0%	86.7%	93.3%	**100%**
0.07	28.6%	34.3%	40.0%	45.7%	51.4%	57.1%	62.9%	68.6%	74.3%	80.0%	85.7%
0.08	25.0%	30.0%	35.0%	40.0%	45.0%	50.0%	55.0%	60.0%	65.0%	70.0%	75.0%
0.09	22.2%	26.7%	31.1%	35.6%	40.0%	44.4%	48.9%	53.3%	57.8%	62.2%	66.7%
0.10	20.0%	24.0%	28.0%	32.0%	36.0%	40.0%	44.0%	48.0%	52.0%	56.0%	60.0%
0.11	18.2%	21.8%	25.5%	29.1%	32.7%	36.4%	40.0%	43.6%	47.3%	50.9%	54.5%
0.12	16.7%	20.0%	23.3%	26.7%	30.0%	33.3%	36.7%	40.0%	43.3%	46.7%	50.0%
0.13	15.4%	18.5%	21.5%	24.6%	27.7%	30.8%	33.8%	36.9%	40.0%	43.1%	46.2%
0.14	14.3%	17.1%	20.0%	22.9%	25.7%	28.6%	31.4%	34.3%	37.1%	40.0%	42.9%
0.15	13.3%	16.0%	18.7%	21.3%	24.0%	26.7%	29.3%	32.0%	34.7%	37.3%	40.0%
0.16	12.5%	15.0%	17.5%	20.0%	22.5%	25.0%	27.5%	30.0%	32.5%	35.0%	37.5%
0.17	11.8%	14.1%	16.5%	18.8%	21.2%	23.5%	25.9%	28.2%	30.6%	32.9%	35.3%
0.18	11.1%	13.3%	15.6%	17.8%	20.0%	22.2%	24.4%	26.7%	28.9%	31.1%	33.3%
0.19	10.5%	12.6%	14.7%	16.8%	18.9%	21.1%	23.2%	25.3%	27.4%	29.5%	31.6%
0.20	10.0%	12.0%	14.0%	16.0%	18.0%	20.0%	22.0%	24.0%	26.0%	28.0%	30.0%

Data used for **Figure 22-10** (continued)

5 Gal/kWh

Elec cost	Water/Sewer Cost $/kgal										
$/kWh	5	6	7	8	9	10	11	12	13	14	15
0.05	50.0%	60.0%	70.0%	80.0%	90.0%	**100%**	**110%**	120%	130%	140%	150%
0.06	41.7%	50.0%	58.3%	66.7%	75.0%	83.3%	91.7%	**100%**	**108%**	117%	125%
0.07	35.7%	42.9%	50.0%	57.1%	64.3%	71.4%	78.6%	85.7%	92.9%	**100%**	**107%**
0.08	31.3%	37.5%	43.8%	50.0%	56.3%	62.5%	68.8%	75.0%	81.3%	87.5%	93.8%
0.09	27.8%	33.3%	38.9%	44.4%	50.0%	55.6%	61.1%	66.7%	72.2%	77.8%	83.3%
0.10	25.0%	30.0%	35.0%	40.0%	45.0%	50.0%	55.0%	60.0%	65.0%	70.0%	75.0%
0.11	22.7%	27.3%	31.8%	36.4%	40.9%	45.5%	50.0%	54.5%	59.1%	63.6%	68.2%
0.12	20.8%	25.0%	29.2%	33.3%	37.5%	41.7%	45.8%	50.0%	54.2%	58.3%	62.5%
0.13	19.2%	23.1%	26.9%	30.8%	34.6%	38.5%	42.3%	46.2%	50.0%	53.8%	57.7%
0.14	17.9%	21.4%	25.0%	28.6%	32.1%	35.7%	39.3%	42.9%	46.4%	50.0%	53.6%
0.15	16.7%	20.0%	23.3%	26.7%	30.0%	33.3%	36.7%	40.0%	43.3%	46.7%	50.0%
0.16	15.6%	18.8%	21.9%	25.0%	28.1%	31.3%	34.4%	37.5%	40.6%	43.8%	46.9%
0.17	14.7%	17.6%	20.6%	23.5%	26.5%	29.4%	32.4%	35.3%	38.2%	41.2%	44.1%
0.18	13.9%	16.7%	19.4%	22.2%	25.0%	27.8%	30.6%	33.3%	36.1%	38.9%	41.7%
0.19	13.2%	15.8%	18.4%	21.1%	23.7%	26.3%	28.9%	31.6%	34.2%	36.8%	39.5%
0.20	12.5%	15.0%	17.5%	20.0%	22.5%	25.0%	27.5%	30.0%	32.5%	35.0%	37.5%

6 Gal/kWh

Elec cost	Water/Sewer Cost $/kgal										
$/kWh	5	6	7	8	9	10	11	12	13	14	15
0.05	60.0%	72.0%	84.0%	96.0%	**108%**	120%	132%	144%	156%	168%	180%
0.06	50.0%	60.0%	70.0%	80.0%	90.0%	**100%**	**110%**	120%	130%	140%	150%
0.07	42.9%	51.4%	60.0%	68.6%	77.1%	85.7%	94.3%	**103%**	**111%**	120%	129%
0.08	37.5%	45.0%	52.5%	60.0%	67.5%	75.0%	82.5%	90.0%	97.5%	**105%**	113%
0.09	33.3%	40.0%	46.7%	53.3%	60.0%	66.7%	73.3%	80.0%	86.7%	93.3%	**100%**
0.10	30.0%	36.0%	42.0%	48.0%	54.0%	60.0%	66.0%	72.0%	78.0%	84.0%	90.0%
0.11	27.3%	32.7%	38.2%	43.6%	49.1%	54.5%	60.0%	65.5%	70.9%	76.4%	81.8%
0.12	25.0%	30.0%	35.0%	40.0%	45.0%	50.0%	55.0%	60.0%	65.0%	70.0%	75.0%
0.13	23.1%	27.7%	32.3%	36.9%	41.5%	46.2%	50.8%	55.4%	60.0%	64.6%	69.2%
0.14	21.4%	25.7%	30.0%	34.3%	38.6%	42.9%	47.1%	51.4%	55.7%	60.0%	64.3%
0.15	20.0%	24.0%	28.0%	32.0%	36.0%	40.0%	44.0%	48.0%	52.0%	56.0%	60.0%
0.16	18.8%	22.5%	26.3%	30.0%	33.8%	37.5%	41.3%	45.0%	48.8%	52.5%	56.3%
0.17	17.6%	21.2%	24.7%	28.2%	31.8%	35.3%	38.8%	42.4%	45.9%	49.4%	52.9%
0.18	16.7%	20.0%	23.3%	26.7%	30.0%	33.3%	36.7%	40.0%	43.3%	46.7%	50.0%
0.19	15.8%	18.9%	22.1%	25.3%	28.4%	31.6%	34.7%	37.9%	41.1%	44.2%	47.4%
0.20	15.0%	18.0%	21.0%	24.0%	27.0%	30.0%	33.0%	36.0%	39.0%	42.0%	45.0%

Data used for **Figure 22-10** (continued)

7
Gal/kWh

Elec cost	Water/Sewer Cost $/kgal										
$/kWh	5	6	7	8	9	10	11	12	13	14	15
0.05	70.0%	84.0%	98.0%	**112%**	126%	140%	154%	168%	182%	196%	210%
0.06	58.3%	70.0%	81.7%	93.3%	**105%**	117%	128%	140%	152%	163%	175%
0.07	50.0%	60.0%	70.0%	80.0%	90.0%	**100%**	**110%**	120%	130%	140%	150%
0.08	43.8%	52.5%	61.3%	70.0%	78.8%	87.5%	96.3%	**105%**	114%	123%	131%
0.09	38.9%	46.7%	54.4%	62.2%	70.0%	77.8%	85.6%	93.3%	**101%**	**109%**	117%
0.10	35.0%	42.0%	49.0%	56.0%	63.0%	70.0%	77.0%	84.0%	91.0%	98.0%	**105%**
0.11	31.8%	38.2%	44.5%	50.9%	57.3%	63.6%	70.0%	76.4%	82.7%	89.1%	95.5%
0.12	29.2%	35.0%	40.8%	46.7%	52.5%	58.3%	64.2%	70.0%	75.8%	81.7%	87.5%
0.13	26.9%	32.3%	37.7%	43.1%	48.5%	53.8%	59.2%	64.6%	70.0%	75.4%	80.8%
0.14	25.0%	30.0%	35.0%	40.0%	45.0%	50.0%	55.0%	60.0%	65.0%	70.0%	75.0%
0.15	23.3%	28.0%	32.7%	37.3%	42.0%	46.7%	51.3%	56.0%	60.7%	65.3%	70.0%
0.16	21.9%	26.3%	30.6%	35.0%	39.4%	43.8%	48.1%	52.5%	56.9%	61.3%	65.6%
0.17	20.6%	24.7%	28.8%	32.9%	37.1%	41.2%	45.3%	49.4%	53.5%	57.6%	61.8%
0.18	19.4%	23.3%	27.2%	31.1%	35.0%	38.9%	42.8%	46.7%	50.6%	54.4%	58.3%
0.19	18.4%	22.1%	25.8%	29.5%	33.2%	36.8%	40.5%	44.2%	47.9%	51.6%	55.3%
0.20	17.5%	21.0%	24.5%	28.0%	31.5%	35.0%	38.5%	42.0%	45.5%	49.0%	52.5%

8
Gal/kWh

Elec cost	Water/Sewer Cost $/kgal										
$/kWh	5	6	7	8	9	10	11	12	13	14	15
0.05	80.0%	96.0%	**112%**	128%	144%	160%	176%	192%	208%	224%	240%
0.06	66.7%	80.0%	93.3%	**107%**	120%	133%	147%	160%	173%	187%	200%
0.07	57.1%	68.6%	80.0%	91.4%	**103%**	114%	126%	137%	149%	160%	171%
0.08	50.0%	60.0%	70.0%	80.0%	90.0%	**100%**	**110%**	120%	130%	140%	150%
0.09	44.4%	53.3%	62.2%	71.1%	80.0%	88.9%	97.8%	**107%**	116%	124%	133%
0.10	40.0%	48.0%	56.0%	64.0%	72.0%	80.0%	88.0%	96.0%	**104%**	112%	120%
0.11	36.4%	43.6%	50.9%	58.2%	65.5%	72.7%	80.0%	87.3%	94.5%	**102%**	109%
0.12	33.3%	40.0%	46.7%	53.3%	60.0%	66.7%	73.3%	80.0%	86.7%	93.3%	**100%**
0.13	30.8%	36.9%	43.1%	49.2%	55.4%	61.5%	67.7%	73.8%	80.0%	86.2%	92.3%
0.14	28.6%	34.3%	40.0%	45.7%	51.4%	57.1%	62.9%	68.6%	74.3%	80.0%	85.7%
0.15	26.7%	32.0%	37.3%	42.7%	48.0%	53.3%	58.7%	64.0%	69.3%	74.7%	80.0%
0.16	25.0%	30.0%	35.0%	40.0%	45.0%	50.0%	55.0%	60.0%	65.0%	70.0%	75.0%
0.17	23.5%	28.2%	32.9%	37.6%	42.4%	47.1%	51.8%	56.5%	61.2%	65.9%	70.6%
0.18	22.2%	26.7%	31.1%	35.6%	40.0%	44.4%	48.9%	53.3%	57.8%	62.2%	66.7%
0.19	21.1%	25.3%	29.5%	33.7%	37.9%	42.1%	46.3%	50.5%	54.7%	58.9%	63.2%
0.20	20.0%	24.0%	28.0%	32.0%	36.0%	40.0%	44.0%	48.0%	52.0%	56.0%	60.0%

Data used for **Figure 22-10** (concluded)

9 Gal/kWh

Elec cost	Water/Sewer Cost $/kgal										
$/kWh	5	6	7	8	9	10	11	12	13	14	15
0.05	90%	**108%**	126%	144%	162%	180%	198%	216%	234%	252%	270%
0.06	75%	90%	**105%**	120%	135%	150%	165%	180%	195%	210%	225%
0.07	64%	77%	90%	**103%**	116%	129%	141%	154%	167%	180%	193%
0.08	56%	68%	79%	90%	**101%**	113%	124%	135%	146%	158%	169%
0.09	50%	60%	70%	80%	90%	**100%**	**110%**	120%	130%	140%	150%
0.10	45%	54%	63%	72%	81%	90%	99%	**108%**	117%	126%	135%
0.11	41%	49%	57%	65%	74%	82%	90%	98%	**106%**	115%	123%
0.12	38%	45%	53%	60%	68%	75%	83%	90%	98%	**105%**	113%
0.13	35%	42%	48%	55%	62%	69%	76%	83%	90%	97%	**104%**
0.14	32%	39%	45%	51%	58%	64%	71%	77%	84%	90%	96%
0.15	30%	36%	42%	48%	54%	60%	66%	72%	78%	84%	90%
0.16	28%	34%	39%	45%	51%	56%	62%	68%	73%	79%	84%
0.17	26%	32%	37%	42%	48%	53%	58%	64%	69%	74%	79%
0.18	25%	30%	35%	40%	45%	50%	55%	60%	65%	70%	75%
0.19	24%	28%	33%	38%	43%	47%	52%	57%	62%	66%	71%
0.20	23%	27%	32%	36%	41%	45%	50%	54%	59%	63%	68%

10 Gal/kWh

Elec cost	Water/Sewer Cost $/kgal										
$/kWh	5	6	7	8	9	10	11	12	13	14	15
0.05	**100%**	120%	140%	160%	180%	200%	220%	240%	260%	280%	300%
0.06	83%	**100%**	117%	133%	150%	167%	183%	200%	217%	233%	250%
0.07	71%	86%	**100%**	114%	129%	143%	157%	171%	186%	200%	214%
0.08	63%	75%	88%	**100%**	113%	125%	138%	150%	163%	175%	188%
0.09	56%	67%	78%	89%	**100%**	111%	122%	133%	144%	156%	167%
0.10	50%	60%	70%	80%	90%	**100%**	110%	120%	130%	140%	150%
0.11	45%	55%	64%	73%	82%	91%	**100%**	109%	118%	127%	136%
0.12	42%	50%	58%	67%	75%	83%	92%	**100%**	108%	117%	125%
0.13	38%	46%	54%	62%	69%	77%	85%	92%	**100%**	108%	115%
0.14	36%	43%	50%	57%	64%	71%	79%	86%	93%	**100%**	107%
0.15	33%	40%	47%	53%	60%	67%	73%	80%	87%	93%	**100%**
0.16	31%	38%	44%	50%	56%	63%	69%	75%	81%	88%	94%
0.17	29%	35%	41%	47%	53%	59%	65%	71%	76%	82%	88%
0.18	28%	33%	39%	44%	50%	56%	61%	67%	72%	78%	83%
0.19	26%	32%	37%	42%	47%	53%	58%	63%	68%	74%	79%
0.20	25%	30%	35%	40%	45%	50%	55%	60%	65%	70%	75%

Table 22-11. Gallons of Water per kWh Saved for Some Cooling Systems

Technology	Compared to	Type of Comparison	When used	Possible Business	Gal / kWh
Water-cooled AC (cooling tower)	Air cooled	Efficiency	Summer	Commercial/ Industrial	**8.8**
Evaporative pre-cool AC (pads)	Air cooled	Efficiency	Summer	Commercial/ Industrial	**7.3**
Evaporative pre-cool AC (atomize)	Air cooled	Efficiency	Summer	Commercial/ Industrial	**4.4**
Water economizer	Compressor	In lieu of	All year	Commercial/ Industrial /Data Center	**4.0**
Evaporative cooling (swamp cooler)	Air cooled	In lieu of	Summer	Residential, manufacturing	**3.0**
Evaporative fluid cooler	Dry cooler	Efficiency	All year	Data center	**8.8**
Condenser, once through to drain	Air cooled	Efficiency	All year	Restaurant	**206**

Notes for Table 22-11

1. Colorado Springs weather. Values are not universal.
2. Gal/kWh calculated from gallons of water used divided by the corresponding kWh saved.
3. Water-cooled AC (cooling tower) efficiency improvement, compared to air-cooled:
   Base: comfort cooling for commercial load, average summer air-cooled 1.05 kW/ton (1.1 kW/ton design, 0.885 SEER factor, averaged to presume it is off at night). Water-cooled efficiency 0.8 kW/ton including auxiliaries, 18% cooling tower blow down.
4. Evaporative pre-cooled AC efficiency improvement, compared to air-cooled:
   Uses water and wetted pads to cool the air entering an air-cooled chiller.
   Base: Comfort cooling for commercial load, average 0.99 kW/ton for operation above 70F dry bulb outside temp. Evaporative pre-cool pad system assumes water only used above 70F, 60% efficient evaporative media, 25% air flow reduction through the unit, 0.25 kW circulator pump for pads, overall 3.6 gal per ton-hour including 20% blow down.
5. Evaporative pre-cooled AC efficiency (atomized): Spray active above 60F, variable spray system follows variable condenser air flow, de-mineralized water, all mist utilized, no blowdown, no air flow restriction, base 1.05 kW/ton air cooled, 95% water absorption/contact with air, spray pump energy subtracted from compressor savings.
6. Water economizer in lieu of mechanical cooling compressor in a water chiller:
   Used to turn the compressor off for cooling load in winter and cool weather.
   Base: water cooled with 0.75 kW/ton cooling plant eff. including auxiliaries, 18% blow down. Economizer operation 0.20 kW/ton cooling plant eff. including auxiliaries, 18% blow down.
7. Evaporative cooling (swamp cooler) in lieu of mechanical cooling air-cooled:
   Base comfort cooling for commercial load, average summer air-cooled 1.05 kW/ton (1.1 kW/ton design, 0.885 SEER factor, averaged to presume it is off at night). Evaporative pad system assumes 0.1 kW/ton to allow for additional air flow (air is not as cool, need more of it), 60% eff. evaporative media, overall 3.54 gal/ton-hour including10% blow down.
8. Evaporative fluid cooler efficiency improvement, compared to a dry cooler:
   Year-round operation for a data center
   Base: Standard CRAC units with water cooled indoor condenser and dry coolers on the roof, no economizer, 1.0 kW/ton seasonal average, CRAC head pressure control. Fluid cooler system uses same pump with 0.75 kW/ton annual average cooling eff, 18% blow down.
9. Refrigeration, once through water cooling efficiency improvement compared to air-cooled:
   Medium temp refrigeration; Base: 1.1 kW/ton over whole year (1.5 in summer, 0.8 in winter). Cooling tower option yields 0.8 kW/ton annually including auxiliaries, single pass, 30F temperature rise. No blow down, it all goes to drain.

Water-Saving Measures for Mechanical Cooling Systems

1. Reduce demand for cooling
2. Use direct air economizer instead of water economizer
3. Increase cycles of concentration to reduce blow down
4. For cold weather heat rejection, dry coolers instead of evaporative cooling
5. Evaluate cost balance between saved energy and purchased water to create the savings
6. For systems that can selectively use air-cooling or evaporative assist air cooling, determine the economic point to "switch modes" to plain air-cooled, based on water cost vs. benefit.

EVAPORATION LOSS

Water loss from evaporation can be deliberate or unintentional. In the case of evaporative cooling the evaporation serves a purpose and provides value. Some sources of intentional evaporation:

- Cooling towers, scrubbers, fluid coolers, evaporative condensers
- Evaporative coolers
- Air washers
- Humidifiers
- Waste water treatment
- An evaporator unit designed to drive off moisture for the sake of recovering the solids or when disposal is more convenient in solid form.
- Water-based inks or paints evaporate water after being applied.
- Baking (food)

When evaporation is unintentional, water loss (and sometimes the attendant cooling effect) is a parasitic loss of water which, and curbing these losses can be a source of savings.

Some sources of unintended evaporation are shown in **Table 22-12**, with potential measures to reduce evaporation. Covers may be attached, hinged, motorized, or removable. A cover in place for 12 hours and off to the side for 12 hours reduces evaporation losses by half.

Table 22-12. Example Sources of Unintended Evaporation

	Measure	Remarks
Swimming pools	Cover, reduce temperature, liquid surface treatment, high humidity, wind break	See also **Chapter 5 "ECM Descriptions – Swimming Pools"**
Open tanks	Cover, locate in high humidity area	See also **Appendix "Evaporation Loss from Water in Heated Tanks"**
Open trenches	Cover	
Ponds	Cover, floating plastic balls	
Leaks	Repair	

Evaporation is driven by the difference in vapor pressure between the fluid and the ambient atmosphere where they meet. When the differential is zero, there is no evaporation. Hotter water and lower relative humidity surrounding air amplify evaporation, so evaporation is less in humid areas with lower temperature water. Outdoor evaporation is accelerated with wind speed. A generalized value of evaporation rate in swimming pools has been widely used

(See Chapter **5 "ECM Descriptions – Swimming Pools"**) and has provided justification for pool covers over and over. Since energy loss is linked to water loss from evaporation, formula simultaneously identifies energy and water loss from surface evaporation of the pool. Heat losses predicted by the pool water loss formula (and therefore the water predicted water losses) have been verified in pool facilities where the heating fuel supply has been sub metered (source: doty,s.).

Example: if 70% of the heat loss is from evaporation, 70% of the fuel use is replacing heat lost to evaporation and that heat can be equated to evaporated water by 1000 Btu per pound of water.

Representative heat loss from heated tanks is given in **Appendix "Evaporation Loss from Water in Heated Tanks."** Given that most of the heat loss is from evaporation, water losses for heated water tank losses can be estimated from the Btuh heat losses from the following equation:

Btuh * factor/(1000*8.34) = gallons per hour

Where:
The *'factor'* is an approximation based on what heat losses are unrelated to evaporation, and will vary depending on insulation and ambient temperature. A value of 0.7 is not unreasonable in most cases. An uninsulated heated tank in a very cold area would have a lower factor, while a heated tank in a hot dry area would have a higher factor. For improved accuracy, test values for specific conditions of water temperature, ambient dry bulb, ambient wet bulb, and insulation would be required.

Liquid pool covers. Little or no independent test data available. The product floats on water and forms a layer that reduces water evaporation. It will be less effective than a non-permeable physical cover, but has the advantage of providing some benefit at all times, which a physical cover cannot claim. For saving water and heat, it is certainly better than nothing which is often the case when users refuse to mess with the pool covers. The economics are not clear, specifically how long the product lasts before deteriorating and needing to be replenished and ultimately how much of the utility savings are given to buying the liquid.

Pond balls (a.k.a. bird balls). The original impetus for developing these polyethylene balls was to keep birds from landing in ponds where they weren't wanted. They also offer evaporation reduction by covering a portion of the surface. Attractive because they are easy, attempts to utilize these in covered plating and process tanks have been met with difficulty such as impeding conveyors that dip parts and blemished plated products from bumping into a ball.

Water-Saving Measures for Evaporation Losses

1. Cover tanks, especially heated tanks
2. Anything that reduces the differential water vapor pressure between the water surface and the contacting air reduces the evaporation rate:
   - Adjust ambient air conditions around tanks for higher humidity
   - Adjust ambient air conditions around tanks for narrower differential temperature (tank vs. air)

     Note that dehumidifying or exhausting rooms containing a heated tank will increase evaporation losses. Containing such rooms and allowing high humidity reduces evaporation losses.

- Contain tank areas with non-permeable room construction, like any other humidified space.
- Where chemicals are mixed with the water and also evaporate and ventilation is required, variable ventilation based on contaminant levels will allow reduced ventilation during off-shift. This is akin to reducing blow down use in a softener, DI unit, or cooling tower, controlling blow down to allowable levels.

  Note: Safety considerations trump energy and water considerations. Sometimes requirements are xyz air changes, period, occupied or not. If safety can be maintained, there is nothing wrong with requesting permission to use an alternate method that brings economy to the operation.

3. Allow tanks to cool during off-shift if process allows it
4. Turn off any tank agitation units during off-shift, as these increase evaporation
5. Municipal supply concepts:
   - Distribute water in pipes instead of open ditches or canals
   - Covered storage
6. Irrigation design concepts: Irrigate directly to roots instead of wetting the surface
   - Select nozzles that reduce evaporation losses from scattered mist

DOMESTIC HOT AND COLD WATER SYSTEMS

Residential water conservation methods are well documented. This text focuses on commercial and industrial applications of energy and ways to economize. Certain commercial buildings provide residential uses (hotel, dormitory) and so the request to address water use concurrent with energy use surveying can be expected.

Costs of purchased water and heating water are proportional to the amount of water they process. For the same 10 minute shower, a 1.25 gpm shower head will reduce shower water heating cost by half compared to a 2.5 gpm shower head. Similarly, bathtubs with reduced volume reduce usage proportionally. When a common building water supply element is replaced, such as a water heater or booster pump, it is often fruitful to use an integrated design approach and retrofit points of use with lower flow devices, allowing the equipment to be downsized. In new designs, this approach can downsize water and wastewater piping as well.

It is important to note that default flow values for plumbing fixtures not otherwise specified will usually be code maximum "standard" values. High efficiency (low water use) water fixtures must be specifically asked for, even in new buildings.

See **Table 22-13.** for some water-saving measures for residential uses.

Table 22-13. Example Residential Water Reduction Measures

Basis of Savings	Water Measure	Energy Cousin
Behaviors that lead to reduced usage	• Less frequent wash loads, less frequent bathing, shorter showers • Metering, sub metering, and accountability for feedback	• Dressing seasonally, using window blinds for direct sunlit glass, turning off lights when not in use • Metering, sub metering, and accountability for feedback
Using less to begin with	• Wash machines and bathtubs that inherently hold less water • Restricting flow outlet fixtures (spouts, shower heads, aerators – **note 1**) • Reduced supply pressure – **note 2** • Lawn and landscape choices that require less water	• Envelope improvements • Less glazing • Lower light levels • Smaller homes, buildings, cars
More efficient equipment	• Toilet fixtures and flush valve retrofits with reduced requirement for 'gallons per flush' • Irrigation equipment (less evaporation)	• Equipment energy efficiency ratings (lights, motors, heating and cooling equipment)
Cost efficient supply sources	• Gray water, well water	• Fuel switching
Providing enough, but just enough	• Metering faucets that automatically shut off, soil moisture sensor, rain sensor	• Equipment right-sizing, variable speed drives, occupancy sensors, timed overrides, automatic control scheduling, photocell, spot cooling
Prevent backslide of savings	• Ongoing maintenance	• Ongoing maintenance

Notes for Table

1. Standard fixtures have maximum flows that are less than legacy replacements, however more aggressive water flow reduction fixtures can be specified for deeper savings. Aeration as a method to reduce spout usage may not be allowed in certain uses such as hospitals, and will limit flow reduction options.
2. When uses are significantly related to washing/rinsing etc. vs. filling.

WATER REUSE OPPORTUNITIES

Waste-not want-not. Hand-me-down.

It's always easiest to use new water for everything and put waste into the sewer. But sometimes economy can be identified by reusing water for a less demanding task before finally discharging it. Even when water rights laws restrict diverting natural rain and drainage flows, reusing water within a facility will normally be acceptable.

All things equal, when multiple sources of waste water and multiple reuse end use candidates exist, match the least picky end use with the lowest quality waste stream. For example, using ultrapure DI rinse water to water the grass overlooks significant residual value if it can be reused as RO input or DI regeneration water. A cooling tower or scrubber is usually the final destination.

The reuse concept requires a review of the discharge of one water user or step with the water needs of another water user or step. When connecting water use between different processes, the same concepts of heat recovery apply which are water quality, quantity and concurrent timing.

- **Water quality tiers**. This concept pairs waste water sources with users that have less strict water quality requirements.
- **Concurrent timing** limits reuse when source and sink are at different times, and incur the cost of storage. Short term storage with multiple batches will have better economic return than seasonal use storage.

  Note: A way to resolve storage cost is in terms of gallons of 'free' water used per year/gallons stored. The longer it is stored before being used, the more costly the storage.
- **Matching quantities** often limits utilization. If two units of 'free' water are available but the usage point only needs one unit, the utilization is 50%. Sometimes the reduced quality of the 'free' water results in using more of it and 1-to-1 comparison may not apply.

Reuse Water Suitability

Contents of discarded water determine what the waste water may be suitable for. See **Table 22-14**. In many cases, it is not suitable for anything and is put to sewer. Unless the contents of the waste water are well understood, it poses risk. Reuse water as irrigation is always an environmental question and, if irrigating food crops is a health question.

It is not uncommon that a waste water source is sometimes suitable, sometimes not; wash vs. rinse cycle, normal vs. maintenance cycle, etc. Especially where cleaning events are manual, such as regenerating a resin deionizer, successful recovery of some reusable water depends upon the operator moving the proper valve at the proper time and is inherently risky. Using recovered water in a process whose ultimate destination is sewer is one way to re-use water with confidence, i.e. as make up to a cooling tower or scrubber or something else at the bottom of the water food chain.

Table 22-14. Some Contents of Concern for Discarded Water
Batch flows usually require storage and may require sanitizing.
All have potential for contamination from maintenance/cleaning events.
None of these is suggested for use as drinking water as is.

Source of Discarded Water	Flow	Contents of Concern
Sand filter or automatic strainer back wash	Batch	Suspended particles. Potential biological contamination (pool, cooling tower, etc.).
Cooling tower blow down	Batch	Contaminants removed from the air stream, water treatment chemicals and biocide, and high levels of dissolved solids TDS.
Scrubber blow down	Batch	Contaminants removed from the air stream, water treatment chemicals and biocide and high levels of dissolved solids TDS.
One pass cooling water for refrigeration or cooling jacket	Continuous while the machine runs	Contamination upon equipment failure, debris from cooling jacket.
Softener regenerant waste water	Batch	Salt brine, removed minerals, especially calcium and magnesium.
Resin bed deionizer regenerant waste water	Batch	Acid and caustic, and removed minerals. Final rinse water is a candidate for reuse.
Reverse osmosis	Continuous	Inlet water dissolved solids, concentrated. Some chemicals from pH control and scale inhibitor. Clean-in-place chemicals.
Washing waste water	Batch or continuous	Soap, suspended solids from what is being washed.
Rinsing waste water		Suspended solids from what is being rinsed.

Commercial and Industrial Water Reuse Examples

Note: All water reuse contains some risk and requires careful thought and monitoring, and not all jurisdictions will allow all of the measures.

A useful concept is "**cascade washing**," nicely illustrated with a "tunnel" clothes washer, but applies to multiple wash processes. A "tunnel" washer is a long cylindrical clothes washer with an auger that pulls the laundry along the length of the machine. *The final rinse water discharge is repurposed as upstream rinse or wash water and so on, moving progressively from clean step to dirty step.* A value of 2.0 gallons per pound of dry laundry total water use has been estimated for the tunnel washer compared to conventional washer-extractor machines using 3-4 gallons per pound. (source: Alliance for Water Efficiency, 2014). Regardless of the current standard for gallons per pound, considering the basic process of wash, rinse, final rinse, the water use can characteristically be reduced by a third or half when the final rinse is reused for a wash cycle of another batch.

It is convenient to reuse water within the same process, but not required. Manufacturing facilities often include multiple products and multiple process steps and with a sharp eye and some homework, *opportunities can be identified for using the waste water from one process to supplant new water use in another process*.

Clothes washing reuse

For new purchases, some commercial wash machines are available with water recycling features (reusing final rinse water) and this would be a very good strategic choice. Water retrofits to emulate a tunnel washer by reusing final rinse water for the next initial wash cycle are possible where conventional fill/extract machines discharge indirectly to open grates; the modification would likely include storage and may require filtering and heating and sanitation requirements may restrict or prevent this.

Dish washing reuse

Like clothes washing, reusing final rinse water for the wash cycle of the next batch is desirable for reducing water and energy usage. Basic washers have a single reservoir that is shared for all cycles and drained, while conveyor washers have multiple tanks. For new purchases, some commercial wash machines are available with water recycling features (reusing final rinse water) and this would be a very good strategic choice. Regardless of the current standard for gallons per rack (values change constantly), considering the basic process of wash, rinse, final rinse, the water use can characteristically be reduced by a third when the final rinse is reused for a wash cycle of another batch. For retrofits, reusing final rinse water may be easier to implement on a multi-compartment style unit since the storage feature already exists. However, sanitation requirements may restrict or prevent this.

Pre-rinse sprayers are also at the 'dirty end' of the washing food chain and are large consumers of purchased water. However the pre-rise activity creates spray that may be a health concern for the worker. Final rinse water could be suitable for this.

Car washing reuse

Filtering and reusing wash water from one car to the next is viable, with 'new' water for the final rinse. The technology to convert existing car washes to recycle water are not complicated, but will be difficult to the extent that space does not exist for the additional equipment and piping.

Vegetable rinse water reuse

Prepared vegetables are trimmed and rinsed and then disposed in a trench drain. The water has pieces of leaves, etc. and is not potable, but the primary contaminant is suspended rather than dissolved solids, so it could be filtered and used as cooling tower water.

Parts washing reuse

There are a variety of parts washing machines. This reference is to parts washers used in conjunction with metal fabrication facilities where the machined pieces are washed to remove oil and prepare for coating. These units have an integral basin that the (dirty) wash water is kept in and warmed, presumably with solvent or soap. When a final rinse is used before the parts leave the washer, the rinse water—much cleaner than the wash water - can be captured in the washer tank instead of a floor drain, and serve as make-up instead of new water. Savings may be small, but cost is also small. Contamination in the rinse water stays with the process.

Reverse osmosis reject reuse

RO concentrates minerals and rejects them in a water stream but adds little or nothing to the water that wasn't already there, so RO reject waste water is more flexible than some other waste streams. When RO feed water is sea water, RO reject waste water is probably returned to the sea. When RO feed water is potable water and purified for process use, RO reject waste water can be used for a number of purposes, including irrigation and cooling tower or scrubber make up. For RO units in series, the reject water for the second stage will be similar to the inlet water to the first stage and can be easily reused in process instead of lower tier uses like irrigation and cooling towers. Note that higher recovery factors reduce the discharge flow and increase the mineral concentration; in some cases, the reject water will not be usable for anything and must be discarded.

Purified rinse water reuse

Very low mineral water is sometimes used for rinsing when any type of residue or blemish on the dried part is unacceptable. When the parts are rinsed in clean areas it may well be that the "drain" water after the rinsing activity is cleaner than the potable water from which it was originally created. Whatever the volume of this "high dollar water," reusing it is attractive for process economic efficiency due to the higher volume of purchased water it represents. Unless it has added contaminants from the rinse process that cannot be easily removed (oil, solvent), collecting this ultra clean waste water allows it to be used as RO input or DI input water.

Process Water Reuse Case Study

Note: This case study gives an example of what can be considered for reuse. Careful water analysis is critical for any application of water for direct reuse.

The example facility processes semiconductor printed circuit board and the process includes a rinsing step using ultrapure deionized water so as to leave no residue that would affect the board operation. The fabrication machines use 18 megohm ultrapure RO/DI water for both wash and rinse functions. The wash section of the machines reuse rinse water in counterflow direction to the direction of travel for the part, progressively to from rinse to initial wash step, and then to sewer; the discarded water is still low resistance (low minerals) but does have some solvent in it from the cleaning operation. The final RO/DI ultrapure water rinse water is also put directly to sewer. The ultrapure water process is mixed bed type with pre-treatment from a two pass RO unit. The DI waste water chemically laden and considered unusable. The RO reject water contains about 400 ppm of dissolved solids but is suitable for low tier reuse. The RO/DI ultrapure water is the largest water end user. Other water consuming support processes include cooling towers for process heat, scrubbers for process exhaust, and humidifiers for make-up air. The property has a lawn that is irrigated in summer. The plant operates continuously, three shifts per day. The existing system captures RO reject water for scrubber use year round and irrigation in summer, with a storage tank. Total 'new' water from the utility connection for the preceding 12 months was 74 Mgal with a cost of $555,000 for water and sewer combined. The existing water system is shown in **Figure 22-11.**

Note: Sketching the end uses in a general hierarchy shows the end uses with higher water quality requirements from lower ones and the ultimate destination (sewer) at the bottom.

End use review: The various end uses were reviewed to reconcile usage with utility meter data. An end use was defined as usage that results in a corresponding inflow of purchased water to replace it; thus anything consuming purchased water or discharging purchased water to drain is an end use. In this case:

- Fab tool use was estimated with a customer sub meter for RO product (test confirmed the sub meter was reasonably accurate). This was divided into wash and rinse portions using customer estimate for 70% wash, 30% rinse.
- Cooling towers and scrubbers were sub metered by the utility. Cooling towers were operating on new water at this time. Scrubbers were currently operating with 100% RO reject water.
- Humidification use was estimated from approximate air flow and bin weather. Humidifier uses RO water.
- Domestic use was estimated from **Table 22-15**, using # of shifts, average workers, and plant hours.
- Irrigation flow was estimated based on local data for similar grass according to amount of turf. About 90% of irrigation was being supplied by RO reject water.
- The RO unit overall recovery is 67%.

- The DI unit average recovery is 96%, owing to the very low conductivity of the upstream RO pre-treatment.
- RO reject was calculated from the recovery factor provided by the customer. Only the RO reject flow that is actually going to drain was included, since some of it was being reused currently.
- DI regeneration water from **Table 22B** based in TDS of inlet water from RO.
- Other incidental uses (leaks, general washing) were neglected.

Reuse Case Study: Baseline Purchased Water End Use and Meter Reconciliation

Reconcile Meter Readings and End Uses (kgal)	Baseline End Use Actual	Baseline Waste Gross Waste	Baseline End Use New Water	Baseline End Use % of New Water
Fab Wash	26,711		26,711	39%
Fab Rinse	11,448		11,448	17%
RO Reject Waste		19,546	9,991	15%
DI Regeneration		1,526	1,526	2%
Cooling Tower	12,864		12,864	19%
Scrubber	5,366		-	0%
Irrigation	4,656		466	1%
Humidifier	3,914		3,914	6%
Domestic	1,685		1,685	2%
Total End Use			68,604	100%
Total Metered Water			73,987	
% Accounted For			93%	

Modifications: (**Figure 22-12)**

1. Filter and reuse DI rinse water as RO inlet water.
2. Reuse DI wash water for cooling towers and scrubbers. No longer use RO reject water for scrubbers.
3. Reuse RO reject water is used for irrigation as now.

Notes:

- The reduction in RO/DI water end use creates further reductions in RO reject and DI regeneration water. Water savings for cooling towers and scrubbers is conservative; low mineral content will yield high cycles of concentration.

- A new storage tank is needed to store DI wash water that is still suitable for cooling/tower scrubber use. This water is not suitable for irrigation or RO feed water, due to solvents in the wash water.
- The existing storage tank would continue to store RO reject water for irrigation use.

Methodology was built in a spreadsheet. Some of the calculations:

- End uses were identified a point where water was either consumed or put to drain. End uses were summed and confirmed a close match to metered records for the last year. Calculations and assumptions for identifying the end uses are indicated on the section "**End Use Review**."
- Pairing: Fab rinse water quality made it suitable for direct reuse as RO feed water. Fab wash water is suitable only for cooling tower and scrubber use. RO reject water is best suited to lawn irrigation.
- Fab Rinse water to RO feed water is real time and is a direct reduction in feed water flow. Fab wash water supply and demand are handled with a "checkbook," where the "deposits" are from the wash water and the "withdrawls" are from the cooling tower and scrubber demands. RO reject water supply and demand are handled with a second "checkbook," where "deposits" are the RO reject stream and the "withdrawls" are the irrigation demands. For DI rinse water, the utilization is 100%. For the two "checkbook" accounts, the portions that create a mutual match are found with trial and error. Granularity was by month, so there will be some adjustment and the tanks will be very useful for this. Tank sizes were not determined but will be based on the end uses of the reuse water, as both "accounts" have surplus "deposits," with the remainder going to drain.
- The RO and DI were tabulated side by side. The DI output was calculated as the sum of the end uses and the DI input was calculated from the given average efficiency for that process. The DI input is equated to the RO output and, in turn, the RO input is calculated from the RO efficiency (recovery factor). This arrangement allows end use reductions from recycling DI final rinse water to show the effect through the RO and DI source efficiencies. For example, with 67% RO recovery and 96% DI

efficiency, each gallon of reduction in DI end use becomes a purchased water savings of 1/0.96/0.67=1.55 gallons saved.

Results

- Savings estimated to be 23.4 million gallons of purchased water per year.
- At $7.50 per 1000 gallons combined water/sewer charges this is $175,000 per year.
- 100% utilization of DI final rinse water as RO feed water, with approximately 28% of purchased RO feed water supplanted by this reuse.
- 72% utilization of DI wash water for cooling towers and scrubbers.
- Cooling towers and scrubbers use no purchased water.
- 34% of total water use would be from reuse water compared to 12% now. This includes irrigation as a reuse destination, but does not include fabrication tool reuse within the machines.

Reuse Case Study: Results and Savings

Results (kgal)	Baseline Purchased Water	Proposed Purchased Water
Fab Wash	26,711	26,711
Fab Rinse	11,448	1,804
RO Reject Waste	9,991	9,493
DI Regeneration	1,526	1,068
Cooling Tower	12,864	0
Scrubber	0	0
Irrigation	466	466
Humidifier	3,914	3,914
Domestic	1,685	1,685
Total Purchased Water	68,604	45,141
Savings		23,463

Additional opportunities:

1. Most of the fabrication tool rinsing stations utilized counterflow reuse of water within the tool, but some did not. Modifications to recycle rinse water as wash water in the same tool will create addi-

tional savings that are amplified throughout the water purification system.

2. A 'roughing' RO could recover part of the remaining RO reject for use in the humidifiers.

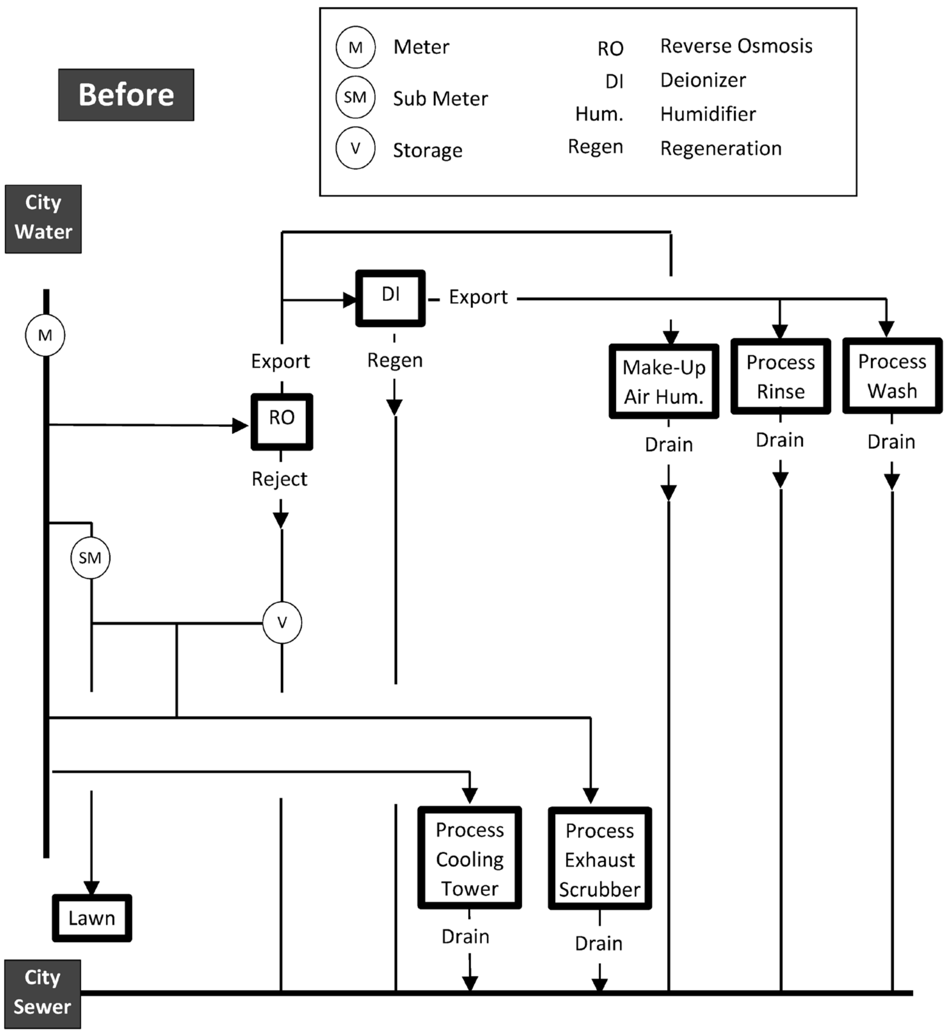


Figure 22-11. Case Study/Modification Flow Diagram (Before)

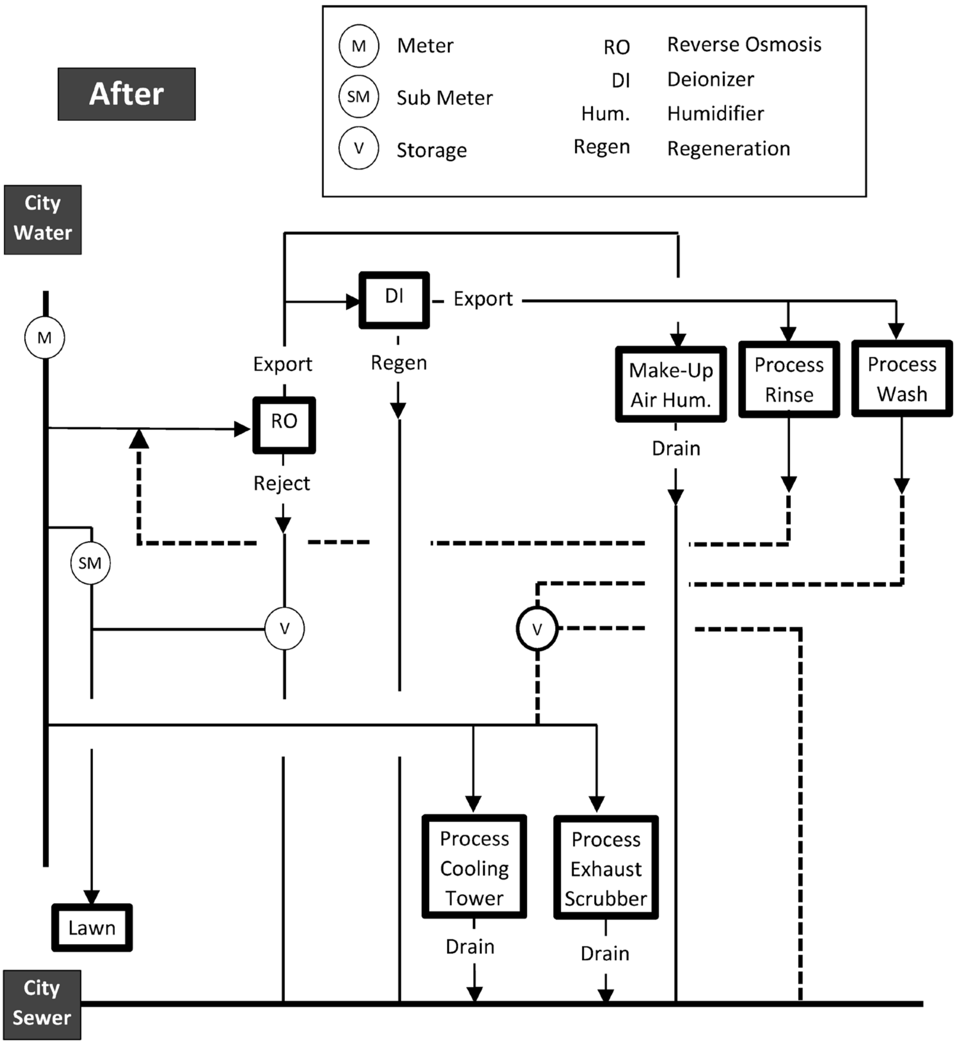


Figure 22-12. Case Study/Modification Flow Diagram (After)

POTABLE WATER SUBSTITUTES

Some water uses require potable water (chlorinated, drinkable, tap water, city water), some do not. Where alternate non-potable sources are available, dollar savings can be achieved even when water savings are not (and savings compound when also using less). Non-potable water is any water not suitable for human consumption. It can be recycled waste water, well water, rain water, etc.

Examples where non-potable water could potentially be substituted for potable water are listed. This is not a definitive list and merely points out applications where drinking water is commonly used other than for human consumption.

- Irrigation
- Toilets
- Cooling towers
- Scrubbers
- Cooling Jackets (injection mold machines, vacuum pumps, etc.)
- Parts washing
- Quenching
- Car washing
- Animal washing
- Clothes washing
- Street and parking lot washing
- Where water is embedded in non-food product, such as paper and concrete

Some non-potable water use practical notes:

- Sanitation concerns for non-potable use are understandable, any municipal attempts to distribute non-potable water create infrastructure challenges which are essentially double piping.
- Requirements vary as far as the commercial use of non-potable water. For example, it may be interpreted by a building official that designs must presume a patron drinks water from a commercial washing machine (stranger things have happened). As a minimum, non-potable water use will involve independent piping, conspicuous labeling, and isolation from potable water systems by backflow prevention equipment.
- Potable water systems are assumed to be the primary supply of water and non-potable systems are secondary. For this reason, when contemplating a switch to non-potable water supplies it is usually presumed to not be available continually and a potable supply is also installed as backup; the design is not complex, but there is a cost increase for redundant piping and double supply connections.

Non-potable water used for cooling towers or other evaporative processes (fluid coolers, humidifiers, steam boilers) may yield disappointing savings if not properly researched. When the non-potable supply is treated waste water, the mineral content (dissolved solids) will be higher than the prior generation water which was initially used. If the make-up water to a cooling tower has double the minerals in it when running on non-potable water, it will require twice as much blow-down and the savings will be eroded. An example illustrates. <u>Step 1</u> identifies the necessary cycles of concentration to control total dissolved solids with the two different water streams. <u>Step 2</u> determines the quantity of water and cost for each water stream.

<u>Example Step 1:</u> Find available cycles of concentration for equal blow down threshold. Potable water was able to use 5 cycles of concentration, N=5; non-potable water was able to use 2.5 cycles of concentration, N=2.5

Assumed values:

- Potable water make up water source: 200 ppm TDS
- Non-potable water make up water source: 400 ppm TDS
- Ultimate concentration of minerals before blow down is 1000 ppm TDS
- Maximum cycles for potable water: 1000/200 = 5 cycles
- maximum cycles for non-potable water: 1000/400 = 2.5 cycles

B=blow down	**Potable**			**Non-Potable**		
	Minerals initial	**Minerals final**		**Minerals initial**	**Minerals final**	
Cycle 2	200	400		400	800	
Cycle 3	200	600		400	**1200**	**B**
Cycle 4	200	800				
Cycle 5	200	1000	**B**			

<u>Example Step 2:</u> Find the actual operating cost savings for using non-potable water.

BD = $^{\mathbf{E}}/_{\mathbf{(cycles\ -1)}}$

Potable: BD=1000/(5-1) = 250; total water use is 1250 gallons per hour

Non-potable: BD=1000/(2.5-1) = 667; total water use is 1667 gallons per hour

Using non-potable water for this cooling tower required 33% more water.

Assumed values:

- Evaporation rate for the cooling tower is 1000 gallons per hour
- Relative cost of non-potable water is 50% of potable water

	Use Factor	**Unit cost Factor**	**Overall Operating Cost Factor**
Potable	1.0	1.0	1*1=1
Non-Potable	1.33	0.5	1.33*0.5 =0.67

Operating cost savings from non-potable water = (1-0.67)/1 = <u>33% (ans)</u>

In this example, non-potable water cost per gallon was 50% of potable, but actual operating cost was only 33% less since more of it was used. Note that if non-potable water cost per gallon were only 25% less than potable, this measure would lose money.

When reuse water is free, it is easy to find a business case for its use, even when using more of it, but when the reuse water is purchased it is not always a clear choice when additional costs are considered. For irrigation use, the reuse water is merely dispensed onto the grass and the comparison is limited to the unit cost of the water supply options. But for process or evaporative use, chemical treatment cost may be significant especially if the water content is such that more chemicals are needed than for 'regular' water.

WATER ACCOUNTING

Identify how the suggestion will reduce water cost by stating "Basis of Savings."

Many of these follow the similar process review and optimization patterns as energy surveys and opportunity hunts.

- Annual usage savings are rate * time according to load profile.
- Where reuse occurs, focus on avoided new water purchase.
- Where water is sold in units of cubic feet, there are 7.48 gallons in one cubic foot.
- For domestic use, water consumption often takes the form of
  Gallons per minute * minutes
  Gallons per flush * flushes, etc.
  A compound unit is 'shower-minutes'

Savings Amplification

See **Figure 22-13.** The value of reusing water or 'using less to begin with' is amplified when the water it replaces has embedded losses, which is to say the use is downstream of something with an efficiency less than 1. For example, a process using RO water may lose 25% of its input water to reject which may not be reusable. In this case each unit of reused or never used water that came from the RO output is the equivalent of 1/0.75%=1.33 units of avoided new water.

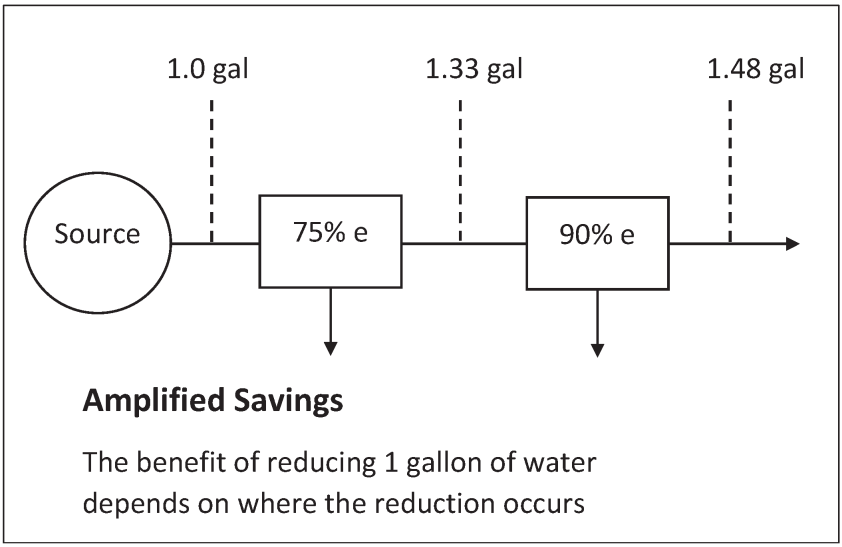


Figure 22-13. Water Savings Amplification

System Approach

For optimizing unit operations, more systems knowledge is always better but sometimes it can be acceptable to know "what" a machine or process does without necessarily knowing all of the details of "how" it does it. In fact, even when the internal intricacies are understood, overall efficiencies and opportunities can be more easily identified without them. What is important to know is what it does: the reason the thing exists, its objective and how it relates to the other elements that collectively are trying to accomplish some objective. A constructive step in process optimization for water or energy is mapping the process in steps or boxes with inputs, outputs and waste streams including any dependencies. A follow on step is to identify the magnitude of the inputs, since for most operations the bulk of the consumption occurs in a few places. Focusing only on individual machines is quicker and more convenient but usually leaves savings opportunities untapped. The more fruitful recommendations will come from investigations at both micro and macro views.

Box Method

The overall process performance can be identified using the "Box Method." With this method, the output of the box is the total delivered product at the point where it is needed (i.e. heat to the occupied space, water to the end use), and input is the item of interest (i.e. purchased fuel, purchased water). All of the equipment and sub processes contributing to the end use delivery goes inside the box. The 'box' method will identify the overall input, output, and waste streams as if the thing were a single unit. This method creates clarity of overall efficiency which is what matters to the business. See **Figure 22-14.** Having established the existing state of overall efficiency and measurement points for ongoing measurement, individual measures (boilers, insulation, controls) can be proposed and implemented, with the success being measured both individually and collectively, always keeping an eye on overall system performance. For systems with seasonal load change, such as heating and cooling, ongoing measurements will identify efficiency in different seasons, including part load.

> Energy example: The heating system's reason to exist is to heat the building. The heating system includes the boilers, pumps, pipes, tunnels, all of which are outside the building itself, so all of this goes into the box. The focus is purchased natural gas. What matters to the business is the heat delivered to the building, so that is the output, measured with Btu meters at the end use. In this example, boiler combustion efficiency consistently tests 80%, but delivered *useful* heat is less and only 70% of the heat in the purchased fuel is utilized in winter and 40% in summer.

> Water example: The pure water system exists to provide rinse water for the delicate parts at the end of fabrication. The water system consists of two reverse osmosis (RO) units in series, filters, pumps, a mixed bed deionizer, 1000 gallon tank and circulating system from the end of line, so all of this goes into the box. The fo-

cus is purchased water. RO reject water is recycled from the second pass to the inlet of the first pass, with no discharge. The first pass RO reject is discharged at 25% rate (75% recovery), which is 200 gallons per kgal of product water (output). The deionizer regenerates in batches based on grains of minerals it has collected, and is averaged to an equivalent continuous discharge of 20% of its output or 100 gallons per kgal. With a given end use of the finished water, the overall water system efficiency is output/input, (1000-200-100)/1000 = 70% recovery. This calculation could also be done with a measurement of delivered pure water flow to the end use.

Box Method vs. Repurposed Water. Referencing **Figure 22-14**, the box *can* be extended to include reuses of the "Loss" water when that water is reused. There are no rules for this, but the approach in this text will be to evaluate process water as shown, and evaluate repurposed water separately, as avoided purchased water.

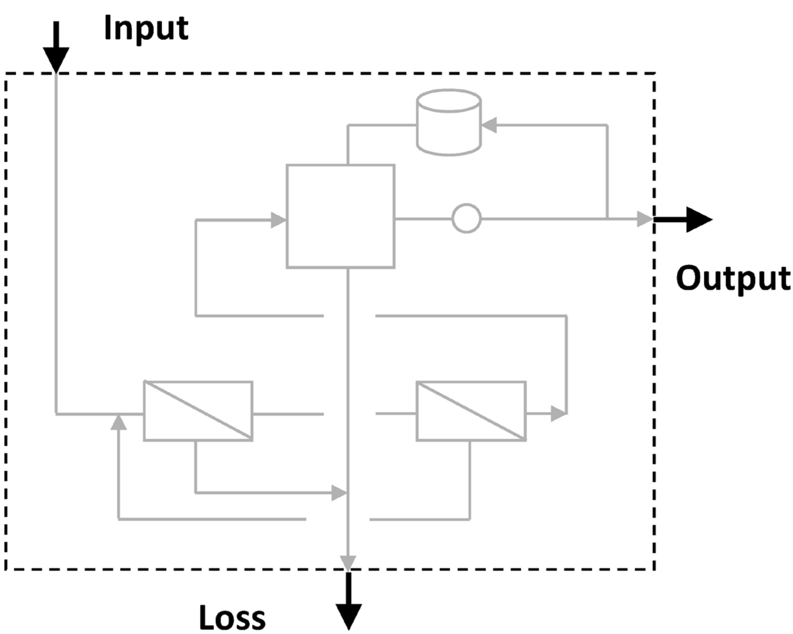


Figure 22-14. 'Box' Method Applied to Process Water
Output is the useful delivered product at the end use.
Efficiency is overall output/overall input

Approach for Interacting Measures

Water measures, just like energy measures, interact when a measure related to a more efficient source is paired with a measure that saves by reduced end use and both are implemented. Evaluating savings independently and adding overstates savings. The method to avoid this is to ***apply the new source efficiency to the reduced new end use***.

Example:

Existing water use is 1 Mgal per month.

Measure A: raise cycles of concentration on the cooling tower from 4 cycles to 8 cycles. This will reduce water use for the cooling tower by 14.3% (from **Table 22-6**), which is 0.143 Mgal

Measure B: add shading elements to sunlit windows and reduce interior light levels in summer. This will reduce cooling load by 15% which, in turn, will reduce cooling tower heat rejection and water use by 15%, which is 0.15 Mgal.

Integrating the two measures: Reduced end use after implementing measure B will be 85% * 1 Mgal = 0.85 Mgal. The cooling tower efficiency (measure A) will reduce the new use to 0.857*0.85 Mgal = 0.728 Mgal, or **0.272 Mgal** savings.

Simply adding Measure A and B: total savings = 0.15*0.143= **0.293 Mgal**, a 7.7% error, and this error is on the high side (exaggerating savings), a reputation professional consultants do not want.

Not everything interacts. Measures that are unrelated simply add.

Contrasting Water Reuse with Heat Recovery

Heat recovery and water reuse follow similar patterns. Waste heat is categorized as high, medium, and low grade which refers to the temperature. Obviously 80F waste heat, even if there is lots of it, is not able to heat a 200F process. To be viable, the waste heat must be at a higher temperature than what is needed in the neighboring process, close enough so as to practical in moving it there, and available at the same time. Within the limits of coincident source and sink, savings are proportional to the supplanted 'new' heat.

Requiring source and sink timing to match eliminates many heat recovery opportunities, and storage is desirable. While heat *can* be stored, but it is costly and diminishes the overall benefit because each storage step involves heat transfer which depends upon differential temperature. Storage then lowers the ultimate benefit; e.g. preheating a 100F process with 200F waste heat directly requires one unit of mass, assuming the heat is delivered in an air or water stream. With storage, part of the differential temperature is spent 'filling' the storage container, say 50F dT. Then there is only half of the original dT available to move the heat to the heat sink and consequently it takes twice the mass flow ($Q=mc\Delta T$). Water is easier to store for long periods than energy. Both energy storage and water storage have high first costs.

The water re-use strategy is very similar to the heat recovery model.

- Quality of waste water must be suitable for use by the recipient. Like heat transfer that goes from higher to lower temperature, water re-use applications go from high cleanliness to low cleanliness, high mineral content to low mineral content, etc. Actual savings may be reduced when the re-use water quality is such that more is required.
- The 'free' waste water must be available at the same time the recipient needs it, unless it is being stored.
- Savings are equal to the portion of 'new' water supplanted, as well as associated sewer costs.

Water Accounting

Accounting for water use can be for matching source and sink for reuse, or simply for accounting for 'where it all goes' compared to the water bill. For water reuse it requires knowledge of the times, quantities and qualities of the sources and sinks. A given waste water stream may have multiple places it can be used, and multiple times. The more you know about how and when water is used the better, ideally creating a water use signature that allows time stamping for sources and sinks. Whether hourly, daily or monthly depends on the steadiness of the loads and whether storage is or is not available as a flywheel buffer. Each unit of time becomes a column or row in a table, and within that time slot is some available excess water flow and the desire to find a good use for it. Steady uses, especially within the same process, can hand down water in real time, but intermittent uses paired with constant sources require 'banking' the surplus water in a tank, or discarding it.

For matching source and sink, it is useful to set up a "checking account" where the surplus water is "deposited" and the reuse water consumption is a "withdrawl." From the available surplus 'free water' stream, a 0-100% portion is deposited into the account until the "balance" is positive, but small—this establishes the portion of the surplus water that has found a use. There can be multiple 'accounts' depending on water quality and what is paired with what.

Knowing 'how much' is either available or required can be a challenge. If a process is sub metered, and the meter is accurate and recorded, you know the load profile very well. In some industrial processes this is well defined, but in others it will require back-calculating or estimating. Examples:

- **Cooling towers**. If you know the cooling load profile in ton-hours, this equates directly to water used by the cooling tower. 1 ton-hour = 12,000 Btu of cooling and about 15,000 Btu to reject at the cooling tower, and evaporating water absorbs about 1000 Btu per pound. Allowing for blow down, cooling tower water requirements can be estimated from either Btu for long term quantities, or from Btuh (Btu per hour) for a flow rate:

  Tons*Btuh/ton*1 lb/1000Btu*1 hr/60 min*1 gal water/8.34lb = gpm per ton for evaporation (E). To this, add the blow down which is E/(cycles-1).

  Refrigeration cycle heat rejection by evaporation (heat of compression is included)

  gpm=0.030*tons, and calculate blow down separately
  gpm=0.034*tons for 8 cycles of concentration
  gpm=0.040*tons for 4 cycles of concentration

  Heat rejection by evaporation without heat of rejection

  gpm=0.024*tons, and calculate blow down separately
  gpm=0.027*tons for 8 cycles of concentration
  gpm=0.032*tons for 4 cycles of concentration

These relationships are easily adaptable for weather-related cooling loads by estimating a load profile (in tons) using bin weather or, if you're lucky, a trend log of cooling load. See **Chapter 9** for discussion on load profile from weather. There can also be cooling in winter for high internal loads and a water based economizer cycle.

- **Humidifiers.** Estimating humidifier loads is determined from how hard the humidifier works. Humidifying a closed space is difficult because it requires knowing the leakage loss from the vapor barrier. A short term trend or data logger to learn how it cycles on and off, or throttles will give some idea. When an outside air stream is humidified, the load can be found from bin weather data for grains moisture per pound entering vs. the humidifier, and the air flow.

The leaving condition will be close to saturation. If serving a space this way, it will catch up and cycle or throttle. If serving a process exhaust it will follow the outside air conditions closely and constantly.

Humidifier Load (lbs/hr) = 60*air density*ΔM

Air density in units of lbs/cubic foot

ΔM = rise in moisture level, lbs moisture/lb dry air

- **Scrubbers.** These take air from the space and 'wash it' before discharge. They are usually constant volume and often run continuously. By knowing or estimating the indoor rH and temperature, the moisture level can be found (lbs moisture/lb dry air) and it becomes a humidifier calculation. The leaving condition will be close to saturation.
- **Domestic use-residential.** When people live somewhere, water use is fairly predictable. A value of **100 gallons per person per day** has worked faithfully for a lot of years. This value represents all water used by the home, which includes laundry and irrigation, cooking, cleaning, all of it. Variability exists for leaks, efficiency of plumbing fixtures, amount of irrigation, and when people are gone a lot vs. always home. To gain confidence, try it out with your own water bill during non-irrigation season (winter), being sure to correct for the days of the bill.
- **Restroom use-commercial.** Restroom water use is normally defined as toilet, urinal, faucet, and shower use—of those four, showers are a wild card. An industry unit is **gallons per employee day** or GED. GED data is derived from total water use divided by number of work days and multiplied by assumed water end use percentages (end use pies) for the sector. Different GED values exist for different sectors; offices are a convenient sector because the work product of an office does not involve water and true restroom water use can be reasonably isolated. See **Table 22-15**.

Another approach is to use utility bills for water usage in an office where restroom usage can be inferred from total water use. Then it is a matter of knowing the number of people and the work hours per month.

— Eliminate irrigation usage by choosing only buildings where irrigation water is metered separately or only counting months in winter when there is no irrigation

— Eliminate water-based cooling usage by choosing buildings using only air-cooled HVAC and refrigeration.

Table 22-15. Approximate Commercial Restroom Water Use by Person

Sector	Description	Gal/ employee-day (GED)	**Gal/ person-hour**	Source
Office building	Air-cooled HVAC	11.5	**1.3**	Annual utility data for three office buildings with known occupancy. Avg. monthly data in non-irrigation season divided by monthly work hours, divided by # of people. Average of the three results which were 1.5, 1.1, 1.2 gal/person-hour. Two buildings had a cafeteria, one had a shower (sparse usage), one had neither. Data from non-irrigation months.
Office building	Hourly use modified for hourly use assuming 9 hours per work day	9-16	**1.0-1.8**	*Commercial and Institutional End Uses of Water, Dziegielewski, B. et al, 2000, table ES.4, American Water Works Association*
Office building	Hourly use modified for hourly use assuming 9 hours per work day	11.6-33	**1.3-3.7**	*Waste Not, Want Not: The Potential for Urban Water Conservation in California*, Pacific Institute, 2003, Appendix E, table E-2. Range is 11.6 (modeled) and 33.0 (GED-derived) gal/employee-day

There are still variables because some office buildings have cafeterias and some have showers, but these are minor. Occupancy density in offices varies but is often in the range of 250 to 700 SF per person which means domestic water use estimates based on building SF will be weak as will estimates based on number of fixtures and flushes.

The conversion from daily to hourly is useful for identifying restroom use in facilities that have different hours or number of shifts, allowing the numbers to be scaled. The basis for this is the fact that **restroom use depends on the number of people and how long they are at the building**.

Example: A manufacturing facility operates with two 8.5-hour shifts, 6 days per week, which is 428 hours per month occupancy. The average number of people working is 90. Using 1.3 gal/hr-person, this is 1.3*428*90 ppl = 50,076 gallons per month.

- **Hotel, dormitory or multi-family building**, see "residential" but apply "percent occupancy" to estimate number of people.
- **Irrigation.** Estimating irrigation has a lot of variables. Soil moisture holding properties, ambient temperature and humidity, type of vegetation, spray or drip distribution, when watering occurs, and if it is watered frugally or excessively. The best estimate is a measurement. In any given city, find a few buildings that have the sprinkler irrigation on a separate meter, read the consumption on the bills (correcting for number of days in the bill period), and compare that the SF or acres of grass and a pattern will emerge. The pattern will follow growing season and outside air temperature. It will also follow rain, so unseasonably wet or dry months will skew the pattern. If using utility bills, irrigation usually can be identified by high usage in irrigation months and 'cut off the top' of the chart as a basic estimate. But if irrigation is amidst large process uses, it may not stick out at all.

Jan	6,250
Feb	8,350
Mar	6,600
Apr	7,150
May	**65,750**
Jun	**82,450**
Jul	**73,450**
Aug	**54,800**
Sep	**65,150**
Oct	18,040
Nov	18,660
Dec	6,500

- **Steam Boilers.** Water use comes from process use of the steam, blow down, steam leaks, and condensate leaks. Sometimes a boiler will have a meter on the make-up water. For all private sub-meters, question accuracy.

- **Swimming Pools**. Water use comes from evaporation, sand filter blow down, and leaks. Evaporation can be estimated from pool temperature and ambient air temperature. See **"Evaporation Loss"** in this chapter.
- **Heated tanks**. Heat loss is mostly from evaporated water, so the heat loss is a reflection of water use. See **"Evaporation Loss"** in this chapter.
- **Laundries and Dishwashers**. Water use per load varies by machine, as much as +/- 100%, so rules of thumb are no good here.
- **Process Use** will vary. Even when the process is understood, water use may or may not follow production levels. If product is in batches, it may be countable especially when water is a controlled ingredient. If process is continuous, there may be a flow meter (if it works). When there isn't an identifiable pattern for processes, it may be that the only data point is the utility meter. If this is the case, you may be able to use process of elimination for other uses you *can* quantify. Similar to energy, in most industrial settings there are a few areas where the lion's share of the water are used. If you can account for 80% of it, consider that a success.

Cost tradeoffs

It is common for process alternatives or technology change proposals to include cost tradeoffs and these need to be evaluated for effective business decisions. Other measures may exchange water loss in one place for water savings in another place. A good example of this is resin deionization alone vs. pre-treatment with RO. This example was given in "**Compound Treatment Loss**," in the **Deionizer** section. Some measures exchange electricity input in one place for water savings in another place.

<u>Example</u>: Evaluate a proposal to use a sand filter and roughing RO unit to recover cooling tower water for use back in the cooling tower. The cooling towers operate on purchased water at 5 cycles of concentration and blow down at 1000 ppm. Electric input to run the RO unit is 5 kWh per kgal. Electricity cost is $0.10 per kWh, water cost is $7.00 per kgal and sewer cost is $3.00 per kgal for the blow down. The RO recovery rate would be 50%. Annual heat rejection load is 5 million ton-hours (1000 ton average load and 5000

hours per year). Maximum evaporation rate is 30 gpm, average blow down rate is 7.5 gpm and maximum blow down rate is 15 gpm. An existing storage tank can be re-used for the cooling tower blow down, so the average blow down rate can be used for RO sizing. There is space in the existing equipment room and the new equipment can be operated by current employees. Processed reuse water will be fed by gravity to the cooling tower with the balance being supplied by purchased new water.

Solution: Cooling load is 5.0EE6 *12,000 Btu* 1.3 for heat of compression = 7.8EE10 Btu to reject. At 1000 Btu/lb, this is an evaporation load of 7.8EE7 lbs of water per year. At 8.34 lbs/gallon, the evaporation load is 9352 kgal of water. Blow down will be E/(5-1), or 9352/4=2338 kgal per year. The RO unit would supplant 50% of that (2338/2 = 1169 kgal) at $10.00/kgal water+sewer charge which is a savings of $11,690 per year. Electricity cost to process the input flow of used water is 2338 kgal/yr * 5 kWh/kgal * $0.10 per kWh = $1169. Combining water savings and electric cost, the net savings is 11,690-1169=**$10,521 annual savings**. The water savings far outweighs the electricity cost, but there are equipment and operational costs to consider. Using 2015 prices:

- Packaged sand filter estimated at $2,500 with a 20 year life.
- Packaged 10 gpm RO unit with membranes is priced at $25,000, with a 20-year life.
- 500 gallon plastic tank for RO product, with level control switches, estimated at $2,000.
- Installation is estimated at $15,000.
- Membrane replacement cost is estimated to be a yearly expense at $1000 per year (based on $500 per 5 gpm-rated membrane)
- Chemical costs are estimated at $500 per year.
- Equipment servicing costs during the 10-year service life are estimated at $1000 per year.
- Collective first cost: 2,500+25,000+2,000+15,000 = **$44,500 first cost**
- Collective annual operating cost: 1000+500+1000 = **$2,500 annual cost**
- Subtracting the annual operating costs directly from the annual savings, this measure provides a simple payback of 44,500/(10,521-2,500) = 5.5 years and an internal rate of return of 17%.

Chemical Treatment Costs

Some measures achieve water savings through the use of chemicals, such as raising cooling tower cycles of concentration and raising recovery rates for reverse osmosis. Processes such as resin bed deionization require addition of chemicals (acid/base) to renew the resin and this water is then stored and neutralized with additional chemicals before discharging to sewer. For measures that create water savings from the addition of chemicals, the cost of the chemicals subtracts from the savings. When chemical are inherent in the water being used, reducing end use and reducing blow down flow, regeneration flow, and reject flow also reduces chemical cost and adds to water savings. Conversely, chemical treatment costs associated with reuse water decrease the apparent water savings, especially when the process requires more of the salvage water (and more chemicals) compared to 'new' water.

Savings De-rating

- Any new active technology that will not be met with a corresponding level of O/M to sustain the performance can be expected to backslide. Passive measures like bathtubs that simply will not hold any more water should not require a de-rate for sustained savings, just like extended overhangs on a building continue to shade summer sun year after year in the same way.
- Low flow shower heads may require a longer shower, or invite one because of the sensation of free showering to squander. The standard calculation involves a combination unit "shower-minutes" which is the number of showers per day or month and how long each one is on average. Presuming the shower minutes (esp. the 'minutes' part) is a constant may overstate actual savings.
- Low flow toilets use a whole lot less water than their old school counterparts, but they sometimes require double flushing. This de-rate would not apply to a low flow urinal.
- Behavior savings are difficult to quantify and assure long term. Self-reported results are often biased.
- Savings with backslide tendency reduce the economic merit. These include degradation of equipment from normal wear or lack of maintenance, measures defeated by users, lack of O/M training, staffing and funding. When these are anticipated, a subjective derate is suggested.

Quality Assurance

As always, take care to avoid solving one problem only to create another one. Adjustment and savings measures are suggestions or recommendations or proposals and are subject to customer review, vendor review, for their comfort level. Examples of problems associated with water treatment are easy to find and expensive to correct. Even exchanging a restroom fixture for a water-saver type should consider the drainage pipe system to which it connects. Making changes to a working process is a business risk.

APPENDIX A. CONSUMPTIVE USE

Consumptive use or **evaporation credit** is available from some utilities, and discounts the sewer charges for water uses that don't put all the water back into the sewer. Common consumptive use items are evaporation (cooling towers, pools, laundries), processes where the water built-into the product (concrete, exported beverages), and irrigation. Factors apply and sub metering is sometimes required to achieve the credit. While sewer is not "water," it does represent a water cost management action item because the cost factor for water is almost always "water + sewer" combined. Thus, the consumptive use credit reduces the water + sewer unit cost.

APPENDIX B. EMBEDDED ENERGY IN WATER AND WASTE WATER

(**Table 22-A1**) A facet of water-energy study is the embedded energy in water or wastewater, which is the energy cost to transport and treat the water. The common unit of embedded water and wastewater costs is kWh per million gallons. Water and waste water embedded energy values are released by many organizations for voluntary benchmarking. Some waste water treatment processes are different than others and may be more or less energy intensive, but a large variance is not seen. For water systems, a large variability exists in the 'raw water conveyance' category when large pumping units are used vs. living by a lake or having water appear automatically by gravity. Water produced from sea water or brackish water by reverse osmosis introduces additional variability.

Table 22-A1. Representative Embedded Energy in Municipal Water and Waste Water

Water Supply or Technology	Remarks	Notes	(kWh/MGal)	(kWh/kGal)
Municipal Potable Water	Fresh water harvesting, typical	1	1500-3500	1.5-3.5
	Fresh water harvesting, high raw water pumping costs	1	10,000	10
Municipal Waste Water	Typical aeration process	2	1500-3000	1.5-3.0
Reverse Osmosis	Potable water purified for process	3	3000-5000	3-5
	Brackish water, conventional	4	7000-11,000	7-11
	Sea water, conventional	5	14,000-21,000	14-21
Ground Water, Well	Per 100 ft lift	6	650-800	0.65-0.8

This table is intended only to provide magnitude comparison.
Actual use varies by technology and inlet water conditions.
RO performance is based on output flow, so the kWh/gal varies by recovery rate.

Notes for table

1. Values gathered in 2013 from various municipalities in Colorado and California Included in water embedded electricity:
    - Total electricity = used for raw water conveyance, treatment, and distribution. Distribution includes potable and non-potable water.
    - Total flow = Mgal of finished water product output (potable, non-potable export flow). Flow value is leaving the water treatment facility. Lost water from leaks or evaporation are not included.
    - These are freshwater harvesting facilities. Desalination costs not included
    - Energy cost other than electricity are not included
2. Values gathered in 2013 from various municipalities in Colorado and California Included in waste water embedded electricity:
    - Total electricity = used for treatment, sanitation, and sludge disposal. This includes treatment of non-potable water
    - Total flow = Mgal of waste water input
    - Energy cost other than electricity are not included
3. Varies by configuration. One single-stage two-pass water purification unit at 300 psi and 75% recovery measured at 4.6 kWh/kgal. One single-stage single-pass RO unit at 180 psi and 70% recovery calculated at 3.6 kWh/kgal.
4. Comparison of Configurations for High-Recovery Inland Desalination Systems, Davies, A., 2012. Includes pumping and ancillary equipment.
5. *Desalination and Water Treatment*, Ludwig,H., 2009. Includes energy consumption of the whole plant (extraction, intake facilities and feed pumping, pre-treatment, RO systems, sterilization, waste water treatment, product water conditioning). Distribution pump energy to consumers not included.
6. This is just the energy to lift the water.

Chapter 23

Using Feedback for Energy Management

CONTENTS

INTRODUCTION

Conservation is using less. There are two ends of this rope, and they are *more efficient systems* and *using less to begin with*. The equipment approach produces big improvements with single line items and represents doing the same things but with better equipment. Lower cost approaches (using less to begin with) involve a multitude of items, many of which involve people and choices. Human nature proves over and over that conservation is enhanced when there is feedback, such as getting a bill. When isolated from energy cost, there is no consequence for a good or bad choice, which leads to disinterest. The keys to savings from behavior measures are encouragement and feedback.

Behavior items can be identified wherever a **choice** exists, and some sources of behavior savings are more obvious than others. The obvious example is an end user choosing to leave the lights on or not. Other choices include performing maintenance for efficiency or only upon failure, operational control strategies that are optimal vs. easy, and whether saving energy is a significant company goal or not.

Savings that come from end user behavior involve people and so are fickle and less predictable than machines bolted to the ground. Since they involve people, they are fickle and unreliable compared to machines bolted to the ground. The easy answer is to dismiss these savings as 'more trouble than they are worth' and focus elsewhere. When the people are customers (e.g. shopping mall, hotel, airport) it is fair to consider behavior as an independent variable. However when people are attached to the property (office employees, factory workers, students) savings possibilities do exist. A common theme in behavior-related savings attempts is feedback that associates a behavior with a result. When the feedback is meaningful and timely, the effect can be self-regulating much like an automatic control loop. Results for measures that include feedback will be much better than those without it.

Savings opportunities from existing equipment have always been there. Harvesting them requires either occupants who care, or feedback measures that cause them to care. And, unlike new equipment savings, behavior measures require ongoing work or savings will quickly decline.

Avoided costs associated with behaviors such as justifying smaller equipment are possible but risky. One scenario is replacing an aging unit with a smaller one, or accommodating growth without adding more capacity; taking credit for the end user behavior. In utility jargon, this is the opposite of 'firm"; it is risky because the equipment size is being based on something that is relatively uncontrollable (if the savings don't materialize, or they go away later and the unit is too small, then what?). This is like asking a design engineer if they will size a glass office building cooling unit presuming the occupants will close their blinds to the west-facing sun. It's not a bad idea, but unless the owner is willing to bear the risk, the engineer probably won't either.

THE NEED FOR FEEDBACK

For good energy management to take place, the energy use must be visible and personal. For behavior savings measures, feedback is applied as steady encouragement and reinforcement so the savings continue. The goal is to make the connection between choices and results. A very effective form of feedback is paying for what is used; when end users are paying the bills associated with the behavior, cost is a good form of feedback unless the cost of energy happens to be cheap. A distinct difference between residential and commercial customers is that end users in commercial and industrial (C&I) facilities are not the ones paying the bill, making apathy toward energy use in the workplace common. When using more or less has no apparent consequence, and whenever energy is provided 'free', it should be expected that usage will be higher than if personally paid for. Metering individual use would be ideal as with individual residential customer, but this is not practical for businesses. Still, the more closely the feedback is representative of personal choice, and the sooner the feedback comes after the choice, the better the results will be for self-regulating behavior.

Using control terms, 'closing the loop' provides a sense of accountability and control, combating apathy. **Figure 23-1** shows standard open-loop and closed-loop control diagrams, modified so the controller is a person, the set point is replaced by their check book, and the output is the person's choice of behavior. The closed-loop variant of this diagram illustrates the importance of feedback: energy use creates a corresponding cost which provides feedback to the end user who can then gauge their future actions accordingly. Open loop control exists whenever there is no consequence, regardless of choosing to conserve or waste.

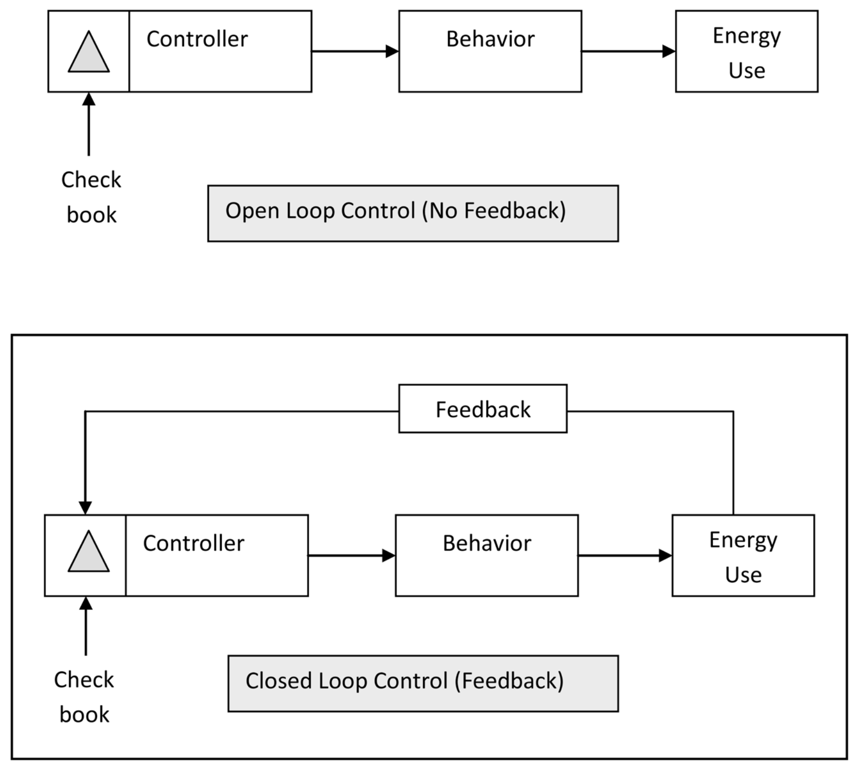


Figure 23-1. Standard Process Control Diagram Adapted to Behavior Choices
In this adaptation, the control set point adjustment is the checkbook

OBSTACLES TO BEHAVIOR SAVINGS

Behavior approach is different when applied to employees or customers. If company management is uncomfortable with seeking savings from measures focused on behavior, it may be best to politely shift focus back to project-based energy reduction strategies, since programs involving employees will fail without management support.

- **Customer/visitor behavior** for conservation has almost no control, nor will it be likely to get management support, because customers are in control of the business and can leave as quickly as they arrive; customers are catered to. Propose a water saving shower head in a five-star hotel and you'll probably be shown the door, because that customer group enjoys excess. Retail and food service businesses strive for an inviting environment, so customers enjoy being there and spend some money. So, it *is* about behavior, just not related to energy efficiency.

- **Employee behavior** for conservation has some additional feedback and control. Employees working inside buildings have a stake in the company because it is their source of income. If they walk away, there is a consequence. Employee managers want productivity out of employees, and one tool for that is a pleasant workplace. Again, behavior is not unnoticed by management, but priorities may be somewhere other than energy savings. Thus, office building operations staff are tasked with assuring comfort and will do almost anything to prevent a comfort complaint. This is not a bad approach, since labor costs are usually much higher per SF-year than energy costs. However, if energy costs increase at a faster rate than salaries, the distance between them will narrow. The hypothesis is that behavior savings can be harvested with minor intrusions into the work day. Prospects of success are further improved when approaching a group that is more receptive; either resonating with the linkage between energy and environment, or the linkage between company profits and one's own job.

Some obstacles to keep in mind when seeking to enlist the building occupants as partners.

- **Accountability.** Whether choosing to save or waste, there is no direct benefit or consequence. Costs and savings are not felt.
- **Apathy.** Small effects from one person's choices are invisible in the grand scheme. A sense of futility comes from the diluting effect of one caring occupant amidst a large number of non-caring occupants.
- **Entitlement.** For some, there is the sense that it is acceptable to be wasteful at work (a measure of fun to spend other people's money). For others, wasteful behavior may simply be from not seeing or paying the bills.
- **Superiority status.** Sometimes people may have the sense that their value to the company vastly outweighs all other concerns, and energy savings are seen as petty. From the viewpoint that everyone must participate to conserve energy, this may seem like a personal criticism. From a business standpoint, it may well be that certain individuals possess vast earning potential for the company and treating them like royalty may be an acceptable business approach despite the ill feelings it creates from "the lower class." Remember, energy

management serves the business, not the planet. When this is identified, the task becomes finding polite ways to work around the minority.

- **Privacy.** People are different. Consider marketing methods of "opt out" vs. "opt in"; e.g. unless you actively un-check this box, you will automatically by default be sent promotional material. Some people don't mind that, some people see it as rude and aggressive. The same emotion, can result from behavior change campaigns or technology. Forcing one's will on others is offensive to many and can be demoralizing in the workplace.
- **High maintenance.** Pursuing behavior savings requires a commitment for ongoing attention. This is much different than capital projects with beginning and ending dates. Tangible factors for management consideration are time involved and money saved; intangible factors are employee morale, and company image; crossover factors include turnover cost and employee productivity for the hours spent at work.
- **Volatile savings.** Behaviors are the least controllable of all low cost measures which explains why many energy management programs do not pursue them. Unlike machines, people cannot be 'told to care'; rather, they are offered encouragement and feedback in hopes they will 'choose to care'.
- **Backsliding.** Savings from behavior measures tend to decay over time because they rely on choices by people. Constant attention such as encouragement and feedback is needed to sustain the savings.

BEHAVIOR CHOICES AND FEEDBACK

When people act like they are paying the utility bill, some things will start to happen:

- Turn lights out when done
- Turn off equipment when not in use
- Dress seasonally
- Draw shades in cold weather unless sunlit
- Draw shades when sunlight is strong against the glass in summer
- Close dock doors except during truck loading and unloading
- Turn off personal items when peak electrical events are announced

Enablers for Behavior Savings

- Occupancy sensors to reduce lighting, ventilation, heating, and cooling when vacant

- Override sensors on zone thermostats to allow compressed schedules (turn off promptly at closing time, and if you stay late or come in on a weekend just push the button for a couple hours of comfort)
- Task lighting for early birds/late birds to prevent large bays of lighting being turned on
- Heated floor pads in lieu of space heaters – a fraction of the power
- Dashboards or other feedback mechanisms and the measurement equipment they draw data from
- Staff and time allotment to engage occupants and make measurements necessary to verify results

Feedback for Customer and Visitor End Users

In cases where conservation at the end user level has a negative effect on business, the added cost of wasteful end users is simply a cost of doing business. In this case, energy cost reduction efforts will focus on equipment rather than behavior, such as insulation, lighting, heating and cooling equipment.

In cases where the end user will not be insulted by the suggestion to participate, there are options. Some people quietly behave frugally on their own. Tastefully displaying a high performing building award will encourage frugal behavior in some who recognize the reward and want to be part of the achievement. Other examples include elevator messages, recycle bins, dashboard displays (especially when there are buttons to push), and even suggestions to use hotel sheets and towels more than one time. While there is no feedback, the messages relay the fact that the company cares and the user can make their own choice.

Feedback for Employee End Users

Feedback can serve to create accountability; it can also maintain interest by having a choice be met with a result – '*I caused that*.' In contrast, a lack of feedback provides an open control loop where the choice to waste doesn't hurt. If choices are made as if the cost (dollars or otherwise) is our own, those choices will be different than if the expense belongs to someone else. When people can see the impact they have on energy use, their choices are no longer anonymous, and wasteful practices may be reduced. With such a program, the company is clearly asking for partners to control energy use; the use of feedback will engage more people and amplify the results.

Some examples of feedback to end users:

- **Awards.** Prominent displays such as Energy Star® or LEED® building rating plaques, local competition or recognition, or other community involvement associated with energy can strike a chord with occupants who then adopt a sense of personal pride and ownership. Refreshing the displays demonstrates ongoing commitment from the company.
- **Praise.** Having requested something that is optional, say 'thank you' and make some noise when it has happened and it worked. For example, requests for the use of window blinds, seasonal dress, or other targeted behaviors will be new to many; acknowledge the willingness to help and describe the benefits. Other 'impositions' worth acknowledging cooperation could be things that are new and noticed, such as occupancy sensors, lights, or control settings. Occupants are your partners.
- **Flyers.** Pamphlets or newsletters reminding customers of energy concepts, tips, efficient light options, maintenance benefits to energy use, thermostat settings, etc.
- **Charts.** Pictures help grasp the message from data usage patterns, such as comparing one month to a prior month, or same month last year. These can be posted, emailed, or displayed in areas like a cafeteria, coffee room, break room, lobby.
- **Energy Dashboards**.

Updates and Reports. Communicate what is being done, with results, future plans, tips and encouragement. Include activities at all levels, including end user, equipment, and O&M levels. Especially include explanations that illustrate linkage between choices and energy use, so each person feels a connection to results and feels that what they do matters. This is an art, and there is a sweet spot between too much and too little; a steady diet of small bites will work better than large blasts of information. Media for communication will depend on the audience and can include paper, email, website, or social media.

Feedback Messaging

Messages are the tender in this transaction, and the types of messages are important. What messages are persuasive and will motivate people to a desired behavior? We can appeal to the person's sense of doing the right thing (leading by example) or the collective benefit of the healthy company they work for. Messages that convey social norms can be very effective [1]. Social norm messages provide comfort that the suggested behavior will be readily accepted by others already there, e.g. '*join the others who are already doing this*' instead of *'please consider doing this'*.

In addition to types and venues for feedback, there is the consideration of how much; there is evidence that regular feedback is more effective than periodic feedback [2]. Of course it is possible to overwork a particular subject and create a repelling effect. In a business setting, the business interests and priorities are communicated to the employees so activities are of like mind, so heavy emphasis on one subject may be interpreted as de-emphasizing others. Interaction with end users is a blend of art and science.

Communication concepts for conservation messaging:

- A servant mentality is well received. We are there to explain and encourage, not to hear our own voice.
- Use single messages.
- Brevity: The more brief the statement, the more emphatic the point.
- Bring the terms down to earth.
- Combine energy skills with people skills. Enthusiasm and the ability to relay technical information to a non-technical audience are essential.
- Link conservation and emissions to dollars. It is no criticism of a business to worry about money.
- A sense of what the audience cares about is essential and so expect a tailored approach. Some occupants will be motivated by dollar savings, others by energy unit savings, and some will not care about any of it – and that's OK. With patience, interest may develop over time.
- Getting someone's attention once isn't hard, but keeping it over time will be a challenge. Something about it has to provide motivation. Just remember, people are busy, people are different, and their cooperation is optional.
- Make communication friendly, inviting and encouraging but not zealous, pushy, or judgmental.
- Acknowledge energy savings as a company profit boost. For employees that are company stockholders, this provides feedback of dividends. For all employees, a boost to company health implies a long life for the company and implies job stability.

Controversial Feedback Mechanisms

Some things can be personal, political, or too much of an interruption, and use will depend upon the group and management support. For example:

- **Suggestion box.** Asking for ideas (open forum or anonymously) involves people and conveys respect. Suggestions from lay persons may not be viable but responding to them is important even when not acting on them. Whenever a conservation measure can be attributed to others, do so.
- **Green teams**. Enthusiastic and passionate people use these to express themselves and are excited at the prospect of influencing company actions. With some visible results morale will be boosted, but without action on recommendations apathy will creep in.
- **Voluntary reporting of carbon emissions,** with displayed results. This shows trends, comparison to others, and reinforces the message of a caring and positive company. The de facto venue for this is the International Greenhouse Gas Protocol. The controversy is twofold. First, greenhouse gas is weakly connected to company profit. More importantly, focusing on greenhouse gas is political, since it directly connects to a liberal point of view and brings to mind the sentiment of imposing personal beliefs on others, a behavior that is commonly offensive.
- **Bounty.** Some attempts have been made to pay for successful ideas. Initially, savings would be predictable initially, but they would be expected to fade in a short amount of time. The race to be first and telling people 'no' to their idea and can have a demoralizing effect to some. Non-cash awards, like a preferred parking spot for a month would evoke more friendly competition.
- **Party.** Sharing a small portion of the savings can be a good investment when it serves to bolster people's attitudes which are the source of the savings which is people and their attitudes. However to those that view the savings as sacred, hard to come by, and who are satisfied with the internal award of doing the right thing, a party can be viewed as squandering.
- **Seed money.** Investing savings in additional improvements provides a mechanism to 'see' fruits of the effort and can build momentum. However, this decreases the financial merit for the measure and should be moderated.
- **Friendly competition** among groups. Extroverts will have a differ-

ent view of this than the introverts. Not everyone has a competitive nature.

OPERATIONS AND MAINTENANCE CHOICES AND FEEDBACK

O&M Contribution Potential

O&M staff is in a very good position to reduce energy cost in a facility and will produce results when incentives and feedback exist allowing them care. See **Table 23-1A and 1B**.

When buildings are closed and manufacturing operations are stopped, they can be viewed as machines that are incurring cost without producing value – from a business viewpoint, it is prudent to reduce the residual operating costs as much as practical. A certain amount of residual power is common for building and plants; some equipment is kept warm, there are parking lot lights and emergency egress lights, minor plug loads, some occasional HVAC use for setback control. Beyond that, things start to become a mix of necessary and optional. It is not uncommon to find that half of the existing residual energy use can be eliminated.

- **Lights left on that could be turned off.** Some operational practices rely on the practice of security staff or cleaning crews to turn them off, especially shared spaces. Operational tours to turn things off will produce results but may or may not be sustainable. Automation in the form of scheduled control or occupancy sensors make these savings more secure but are an investment.
- **Computers left on that could be turned off.** Some practices are based on central IT computer operation habits, for updates, or to be sure there is no delay when the computers are accessed in the morning by the end users. The "always on" practice is sometimes justifiable, such as military or emergency operations that must be able to act instantly, but more often than not it's just a habit. Part of this practice stems from the IT department not being responsible for paying the bill.
- **Air compressors left on that could be turned off.** Sometimes, justification is presented that a certain machine needs continuous air pressure or that air needs to be available for an emergency use. In these cases, leaks from the main system can represent a much larger 'use' than the minor or anomaly use and the practice is very costly.

The easy answer (behavior) is to accept that what is, is. The assertive operational behavior is to ask "why" and investigate. Pursuing these savings is a choice – a behavior. Management would be wise to provide incentive for the behavior, get feedback for results, and then give feedback for the performance. One activity to create feedback is reviewing consumption data. The activity of monitoring interval data and investigating usage during off times is new to many operators, but is often fruitful. It is one thing to seek out the savings, and another thing to sustain them over time. Automatic monitoring will go a long way to keeping this useful bird-dogging of night or off-shift power use active after the novelty wears off. The chance of sustaining the savings increases when the management feedback is also sustained.

Enablers for O&M Savings

- **Training**. Not being required to care about energy use is not knowing what things influence it. Many of the links between O&M choices and energy savings will be quickly picked up once there is a compelling reason to care.
- **Tools.** These may include instruments attached to the automatic control system, or handheld instruments used by technicians for spot checks. This feedback is essential for justifying heightened O&M activities. Quantified results are needed for management support; some can be measured, such as before/after conditions; others are calculated.

Case Study of O&M Savings – Ghost Loads

Example: A logical first step is accounting for usage, which might include counting the things you can see running (exterior lights, interior lights, and equipment) and searching for other loads until the residual usage is accounted for. Then, each usage line item can be evaluated. Once the optional items have been eliminated, subsequent measurement indicates what the residual usage should be, which then forms the basis for ongoing measurement to sustain it, i.e. if usage is found to be above the expected baseline usage, the feedback prompts the question of 'why'. See **Figure 23-2.** Here, the usage in the middle of the night was found to be almost 60% of daytime usage.

Automatic Controls and Optimization

Computer operated building controls are a potent tool for operation-

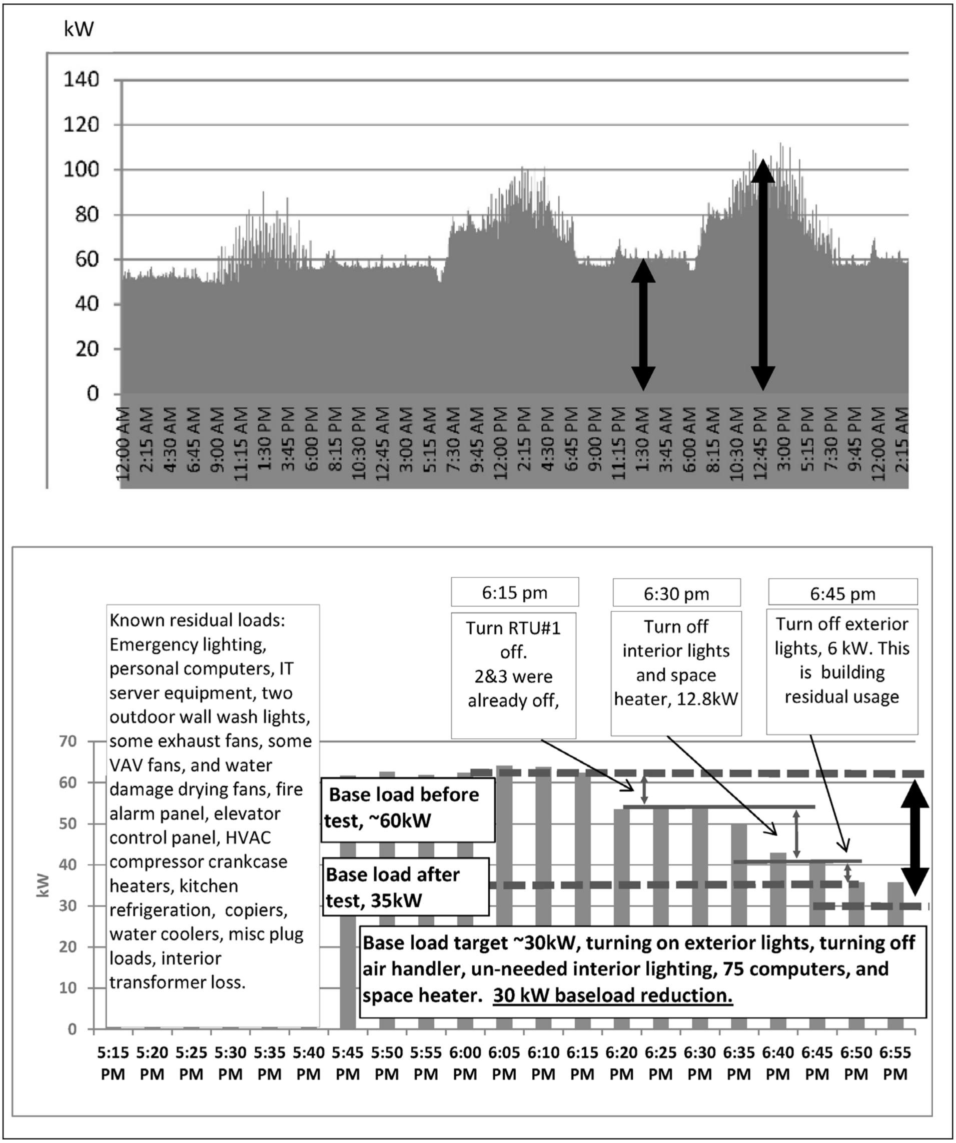


Figure 23-2. Ghost Loads Identified with O&M Practice and Interval Data

Data is five minute electrical demand for a building in its unoccupied state. Top chart shows Sunday, Monday, Tuesday usage pattern. Bottom chart shows a one hour usage pattern in detail, before and after test changes made. Part of the residual load and expense is necessary (emergency lights, parking lot lights, freeze protection, equipment that cannot be turned off). The rest of it is related to occupant behavior choices (leaving things running), and operational choices.

al savings. The more energy systems that can accept automatic control, the greater the potential savings when properly applied.

- Motor controls and lighting with remote on/off capability.
- Motor driven loads that can be throttled (slowed).
- Temperature settings that can be altered locally or remotely.
- Equipment that can be remotely load limited, load shed, or reset.
- Mechanism for user override request to allow compressed schedules
- Controls that include both the source and end use points. For example, digital control of an air handler paired with standalone control of terminal units significantly increases opportunities for optimization.

Controls have their own barriers and enablers, including training, and time allowance to focus on controls.

The guiding light for optimization routines is to *provide enough, but just enough* of whatever it is, which is why end use feedback is so useful. Almost any simplified approach and constant value control setting represents an opportunity for improvement. Knowing how the upstream and downstream elements of the system interact helps identify the necessary feedback. Process optimization is the subject of many texts, all of which center on the full view of the process, energy inputs, and feedback.

Examples of operating control optimization:

- **Process:** A large blower (10 psi, very high volume of air) at a waste water treatment facility has a means of varying its capacity, and serves an array of air outlet points, each with a control valve to modulate the air to what is needed.
- **Conventional operation**: The system maintains a constant pressure setting equal to the highest estimated demand day of the year, and the machine capacity modulates to maintain that set point at all times. Control valves at each point of use throttle back to get the flow they need. This functions fine.
- **Optimized operation:** This functions equally well with less energy. Feedback from each point of use is sampled, either the valve position (hardware position sensing device) or controller output to each end use control valve (implied valve position). The information becomes feedback from the end of the process that allows optimization at the beginning of the process. This control method is called 'most open valve' and provides enough, but just enough pressure: upstream

main pressure is constantly adjusted so at least one control valve is 95% open…supplying just enough pressure. Savings come from avoiding dissipation losses that result from a constant pressure and control valves to dissipate/waste whatever is not needed. The same approach applies to a multitude of processes, but challenges some established control practices, namely by combining upstream and downstream control strategies into one.

Maintenance-Based Savings

Certain maintenance tasks have identifiable energy savings. The savings of a maintenance task depends upon how inefficient the unit was to begin with. For example, increasing the frequency of a task will cause less inefficient operating hours. Maintenance savings have good prospects for long term savings, since they are a matter of procedure.

Examples of maintenance optimization:

- **Equipment:** Heat exchanger serves a process
- **Conventional operation:** The heat exchanger is operated until there is indication of a loss of capacity, to where the served process is ineffective or slowed.
- **Optimized operation:** When a heat exchanger is fouled, there is an energy penalty. Fouling is evidenced by high approach temperatures, so temperature sensing points identify this; can be thermometers with markings showing clean state, or can be automated if desired. Heat exchanger service is prompted by approach temperature which occurs after fouling begins but before performance is impacted. Calendars can do reasonably well if processes allow adjustment based on site conditions. For example, when a heat exchanger is opened and found to be in bad shape, the interval to the next cleaning would be shortened.

Feedback for O&M

Operations-based savings involve how the building or plant is operated. For energy management, these measures are often the first to be implemented because they do not rely on large capital expense. Thus, they are the counterpart to project-based approaches. Often, the measures for operational savings amount to 'doing things differently'. This becomes a choice (a behavior change). With feedback and reinforcement, the behavior change can be sustained and valuable; without feedback,

Table 23-1A: Example O&M Measures with Energy Benefits (Maintenance)

Prevent backsliding. Maintain equipment and systems already in place to sustain energy savings
Maintain clean heat transfer surfaces of all types, for low approach temperatures
Repair duct leaks
Repair compressed air leaks
Repair steam traps
Repair insulation and seals (piping, ducts, ovens, tanks, doors)
Repair door seals for docks, ovens, partitions between climate controlled areas. This also includes covers for heated tanks
Repair piping expansion loops and fittings where expansion/contraction leaks have evolved into continuous operation rather than repairs
Verify refrigerant charges
Check/adjust/repair economizer controls for packaged HVAC equipment
Check/adjust/tune burners for combustion efficiency
Calibrate instruments

Table 23-1B: Example O&M Measures with Energy Benefits (Operations)

Prevent backsliding. Maintain operating strategies already in place to sustain energy savings. This includes economizer controls, resets, etc.
Compress run schedules for lighting and equipment, either through automation or practice. Anything that is "all the time" is a flag for possible improvement. Off is better than idling. In comfort applications, operating boilers in summer and chillers in winter is energy intensive. Measures can include programmed schedules for start/stop, occupancy sensing for intermittent processes (e.g. paint booth, hotel rooms), and idling equipment. Compress schedules and rely on requests for extended hours of operation to avoid unnecessary run time.
Elegant operations rather than brute force. Reduce inherent energy burdens in primary services such as refrigeration head pressure, compressed air pressure, distribution air and water pressure, heating and cooling temperature, and ventilation amount. This also includes minimizing overlapping/fighting systems, especially when control is from heat/cool mixing or pressure dissipation. Best operating practice provides enough, but just enough of each service with resets or feedback from points of use. Optimize by assessing the demand, and adjusting the initial service pressure or temperature to just meet that demand.
Fuel switching can save money but requires some diligence. A practical example for commercial buildings is locking out electric heat in unoccupied and set back periods when an alternate source of heat is available, e.g. electric heaters in VAV boxes vs. natural gas heaters in rooftop equipment. Where demand rates are high and low load factors exist (load factor = average / max kW), peak shaving can sometimes pay off.
Set back temperatures. Set-backs should coincide with building usage. It is common to see customer setbacks that are only a few degrees below occupied settings. Other than freeze protection, aggressive setbacks (e.g. 50F) are a good opportunity.
Reduce ghost loads. Feedback for this requires a sub meter or utility interval data (e.g. every 15 minutes) and are identified when the energy use persists while the building or plant is closed. These loads can be significant. Lights, computers and air compressors are common items left to idle. The energy insult of ghost loads is that money is being spent without any resulting productivity: parasitic business loss.
Respond to utility price signals. Shift processes to off-peak Turn off non-essential loads during peak times Schedule any optional testing (such as electric fire pump) to off-peak time Schedule maintenance run times for generators to on-peak time Duty cycle or load limit provided production/productivity/air quality is met
Review control settings, schedules, and overrides frequently

the savings may not be properly identified and may deteriorate over time. Some operational choices are implemented with automatic controls and can be durable. Other operational choices involve habits which are subject to a form hysteresis where the choice to change needs initiative and has a natural tendency to revert to the original behavior. The initiative to attempt a change is easier to achieve than the initiative to sustain the new behaviors, and feedback is the key. For some, simply knowing the behaviors are ideal for low energy use is enough, but when additional effort is involved, the default assumption would be that it creep back to the old ways over time. Implementing measurement methods and feedback is prudent for maintaining these savings. Goals, charts showing before/after or what it would have been without the activity, and reward, are examples of feedback. Most workers will see their value improve by increasing their value to the company; value can be in the form of pay or improved prospects of long term employment.

Connecting energy use to job duties provides the motivation for activities to consider energy cost as well as repairs, comfort, and complaints. In turn, an environment is created that seeks better ways to do things. Showing O&M tangible value will have a sustaining effect from shifting the age old and inaccurate paradigm that O&M staff are a necessary evil, sunk cost, overhead cost, or other traditional labels, to a valuable company asset that controls and reduces operating costs.

- <u>Recognition</u>. O&M staff will readily embrace continuous improvement when they believe it will be noticed. Noticing is feedback and the enabler for being noticed is documented savings. Feedback begets feedback.

  Note that recognition one time for an extra task done 100 times can expect gradual decline for the extra duty.
- <u>Support.</u> Flexibility in management allows freedom to seek out savings opportunities within the general scope of O&M activities. O&M budget and staffing levels beyond bare bones, respectable work quarters and funding for training are examples of support from management. Voicing confidence in the O&M staff to departments they interact with will spread the support effect to peers and occupants.
- <u>Accountability</u>. "It's easy to spend someone else's money." When accountable for the utility costs, O&M staff will align their priorities accordingly. But when energy costs are paid for in other departments, there is no motive to pursue energy savings. If the O&M de-

partment priorities are limited to "keep it running" and "prevent hot/cold complaints," reducing energy use is seen as meaningless or even a detriment. For example, comfort issues will be less if a building is never set back and after-hours phone calls are reduced when all equipment is kept running over the weekend.

MANAGEMENT CHOICES AND FEEDBACK

The nature of business requires clear definition of mission and goals which become what managers care about and focus on. A best practice for business organizations is to delegate energy cost accountability down to all levels of middle management. Tools for this are budgets with energy cost accountability, sub meters, and performance metrics that connect department usage to department manager pay. When the incentive is meaningful, the department manager will respond to the feedback by exerting influence and control to their employees to achieve the goal, and activities will change from reactive to proactive as a result. Businesses without delegated accountability for energy use can expect higher usage and costs compared to management structures with such feedback. Resistance to change is common, and opportunities are sometimes missed simply because it is easier to make no changes.

Enablers by Management

A primary enabler management provides is a commitment to make organizational changes that provide energy accountability and tools to measure it.

Financial backing is an essential enabler for the time, training, instruments, and controls to implement changes and verify the results. O&M departments on a starvation budget are disabled from producing optimal results and lost savings will usually exceed budget reductions. Incorporating energy benefits can form the business justification for O&M program scope beyond 'keep it running'.

Feedback Created by Management Structure

Accountability

Each business endeavor has an identified reason to exist; this is true for commercial, manufacturing, security, healthcare, military, data center, and other endeavors. It is good to hold dear what is important. The flaw is equating *'this is important'* to *'this is important and nothing else is important.'* Without the feedback that comes from accountability, a business mission can be used as a convenient excuse to ignore savings opportunities

– sometimes even as an arrogant view that energy conservation is beneath them. For example, data center energy evaluations routinely focus on cooling energy, and ignore the data center energy entirely as untouchable; the convention of PUE (ratio of total data center energy to IT energy) encourages this and is, itself, a fundamental flaw in energy management because it treats IT equipment energy as unchangeable. Similar unbalanced or aloof choices can be seen in security, occupational safety, and medicine. If management lives in fear of these groups, then there may be no solution and operating costs that are higher than necessary are a cost of doing business. But this is risky where competition exists. A preferred approach is to maintain the high standards of the endeavor output while also incorporating energy savings. In most cases, the barrier is only a paradigm – we do it this way because we've always done it that way – and paradigms can change with management help. Of course, the pressure of reduced operating cost can be taken too far and impact performance, quality, or safety. These are not easy choices; the concept is balance and feedback is the enabler. This represents a unique difference between residential and non-residential energy conservation: if I choose to waste money at my house, it has no impact on the finances of my neighbor. See **Table 23-2**.

Table 23-2. Examples Where Lack of Accountability Discourages Conservation Practices

(Where adding accountability encourages conservation practices)
Local government entities where bills for individual buildings or departments are paid centrally.
College and military campuses with multiple buildings where billing is paid centrally or master metered.
Multi-family facilities where utilities are built-into the rent.
Leased office or warehouse space where utilities are built-into the rent.
Data centers.

As long as the department cost of utilities is paid for "by others," energy use will be higher than it needs to be. But if a department pays their own utility bills, the viewpoint changes. If budgets or goals are not being met and the manager is facing a consequence, the feedback causes a basic behavior of "what can I do to reduce this?"

Management and the workers in their watch have many things they are accountable for. If energy cost control is a company goal, having specific groups see and be responsible for their energy use makes sense and the dysfunction without it is obvious. On the other hand, if energy is a tiny expense compared to other expenses, then the choice to care only trivially may be appropriate. Remember that energy is a tool for production and as long as production is high and energy cost is the enabler and relatively low, it may be rational to not care too much. For reference, **Table 23-3** and **Table 23-4** show typical values of energy cost as a percent of total O&M costs.

Table 23-3. Typical Commercial Energy Expense as a Percent of Total Operating Expense (3)

Values are purchased utility costs divided by total operating expenses. Energy presumed to be the largest utility cost. Payroll costs are included in the O&M cost except as noted.

Business Type	Percent Operating Expense from Energy	Remarks
Colleges	3.2%	Includes teaching staff payroll
Healthcare - Hospitals	1.3%	
Hotels and Motels	5.3%	
Restaurant	5.1% (full service) 4.6% (fast food)	
Retail	2.7%	
Office Buildings	1.25% (salaries included) 20% (occupant salaries not included)	No-occupant salaries category is for a leased office building

Feedback for Management

Within a business there may be multiple defined groups that are good candidates for compartmentalized energy cost and goals; these can be in separate buildings or in different sections of the same building. Whether gauged by the main utility meter or a sub meter, the feedback is the measured usage for the business section involved – the needle that points to good or bad results. A good way to visualize the practice is to draw a box around it. See **Figure 23-3** and **Table 23-5.**

How management views energy savings results can be either an enabler or disabler. If savings "today" are not acknowledged "tomorrow," energy conservation gains will be seen as asymptotic and even self-defeating. A more accurate representation uses two separate sets of values for

Table 23-4. Typical Manufacturing Energy Expense as a Percent of Total Operating Expense (4)

Values are cost of electricity + fuel, divided by total operating expenses, including payroll.

Category	% Operating Expenses from Energy	Remarks
Textile product mills	1.9	NAIC 314
Paper pulp mill	11.2	NAIC 322110
Nitrogenous fertilizer	14.4	NAIC 325311
Tire manufacturing	2.8	NAIC 326211
Alumina refining	18.3	NAIC 331311
Iron and steel forging	4.6	NAIC 332111
Automobile manufacturing	0.5	NAIC 336111
Semiconductor and related device manufacturing	2.9	NAIC 334413
Electroplating, plating, polishing	4.8	NAIC 332813
Heat treating	9.2	NAIC 332811
Ready mix concrete	2.2	NAIC 327320

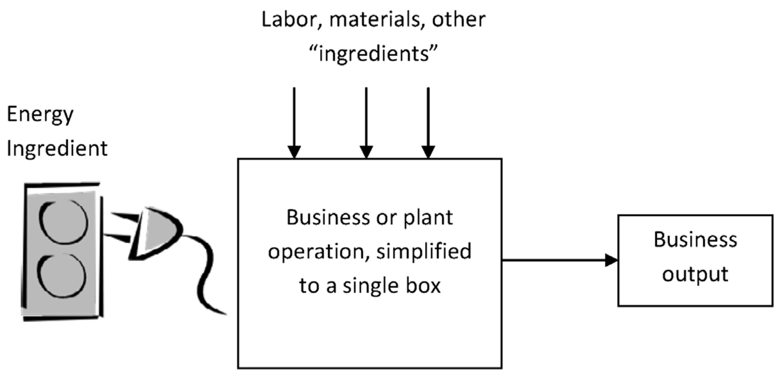


Figure 23-3. Simplified 'Box View' of Overall Energy Efficiency

consumption and cost, contrasting actual consumption and cost to "what it would have been." See **Figure 23-4**. From a charting standpoint, this requires two sets of books, with the "would have been" numbers including the benefit of savings. With a single measure and sustained savings, the two lines will be parallel but offset; with a continuous improvement culture, the two lines will diverge. The credibility of the savings may be questioned so the better the numbers, the better received this will be. Examples

Table 23-5. Direct and Indirect Business Energy Metrics

Type	Application	Description
Direct	Commercial Building	Energy use per SF, per meal, per student, etc.
Direct	Manufacturing facility or sub process area	Overall utility input per unit of production (per part, per semiconductor chip, per ton, per million gallons, etc.)
Direct	Data center	Useful computer output per kWh (Note: this measure is elusive and varies by computer application, but when computers are used for a purpose, that purpose can be measured. Some suggestions include number of cycles, calculations, or transactions)
Indirect	Manufacturing	Btu per machine 'move'
Indirect	Manufacturing	scfm of compressed air per part
Indirect	Manufacturing	kW per scfm of compressed air
Indirect	Manufacturing	Facility demand or load factor
Indirect	Cooling	Overall kW/ton including all auxiliaries
Indirect	Heating	Overall thermal efficiency including all thermal loss other than the final end use
Indirect	Ventilation	Outside air volume per person

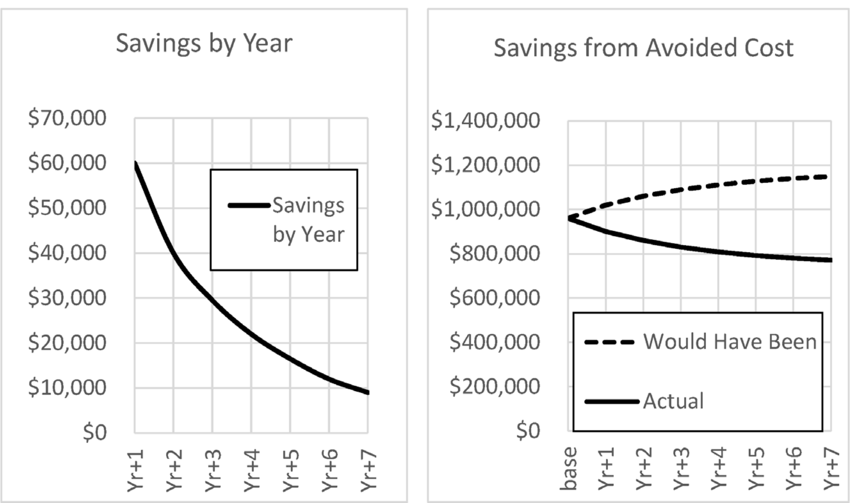


Figure 23-4. 'What it Would Have Been' Charting

Value of assuring continued savings of measures in place are measures that backslide from original savings and measures that interact and do not exactly add; these can be accounted for with careful energy management practice. Repeated commissioning or maintenance savings serve to safeguard the initial savings from eroding.

UTILITY CHOICES AND FEEDBACK

Basis of savings can come from various places. The traditional source is the avoided cost of additional infrastructure. Sometimes there is no business case for efficiency incentives other than the utility oversight authority telling the utility to provide them, and then collect what it cost. Sometimes there is a business case for avoided fuel and O&M cost that comes from using less, although normally using more is better for the business the utility is in. Time of use is a motivator for utilities since maximum demand determines infrastructure sizing, and portions of infrastructure that exist for small periods of time are expensive to recover. Feedback is in the form of dollar savings or dollar penalties. Customer choices that impact demand and consumption are all about behavior. An advantage utilities have is large populations; statistics tell us that results will be more firm with large populations than small ones and can be viewed as a collective customer entity. Some utility measures exchange money for a switch to control customer load at will, but most rely on financial incentive as the feedback to encourage a change in behavior. Utility measures applied to residential customers will be more likely to succeed because the customers are, in fact, paying the bill; applied to commercial customers, the end users do not directly pay the bills and will be less likely to change behavior.

Some utility feedback mechanisms utilities provide

- **Metering.** This does not change the reality of a utility needing to recover its costs, but does promote conservation because the customers see their usage as feedback to their choices. Without metering, wastefulness is not traceable. A similar 'free utilities' effect exists anywhere personal choices for use are not accompanied by feedback to indicate the results of the behaviors. Examples are multi-family, apartment, and dormitory buildings with central services, and master-metered campus settings with multiple buildings such as colleges and military bases.
- **Utility prices.** If a service is more expensive, overall use will pare down. Locations where energy cost is low will have less success with

conservation measures in general than areas where energy cost is high.

- **Inverted block rates.** Where inverted (inclined, or increasing) block rates are used, each successive block of use is more expensive. The intent of this rate design is to provide cost feedback to the end users causing them to change behavior to avoid the penalty. Inclined rates are nicknamed 'conservation rates' because they encourage conservation. Managing to operate in the first tier or block provides reward and is actually at unit pricing below the cost to produce it, while usage in the higher tiers produces negative feedback. The extent of conservation produced by inverted block rates is a concern for utilities who must remain whole in collecting overall operating costs. The potential exists for reduced consumption to cause reduced revenues [5] and may require an iterative increase in rates to compensate. The utility business case for time of use incentives is clearer than general conservation.
- **Time of use.** High on-peak pricing discourages use; low off-peak pricing encourages use. The differential between the two defines the business case for load-shifting measures. Low off peak rates also creates apathy for conservation during those times, invoking the quiet reality that it is cheap to waste at night.
- **Interruptible rates.** Sometimes a physical interruption – disconnecting load. More often, stinging price increases that cause the customer to disconnect it voluntarily, albeit with a few choice words. These are usually accompanied by a reduced general rate for the product during the balance of the year, providing the mix of positive and negative feedback.
- **Incentives.** Rebate designs consider the uptick cost of a high efficiency measure. When the customer is faced with a replacement, the choices may be 'like kind' or the efficient alternative which usually costs more; the rebate intends to make the efficient upgrade more attractive. These are paid for through rates, but serve the communication need of overall public energy awareness.
- **Awareness programs.** From billboards to energy audits, utilities reach out to customers with messages related to behavior, 'conserving to save money' being the most common. While it may seem obvious, associating choices with utility cost is foreign to many customers, residential and C&I alike. Beneath each specific incentive and recommendation is the simple concept that a portion of the energy consumed is

discretionary. Once customers understand the linkage, they are empowered to find ways to save on their own, beyond the specific measures and recommendations given. Behavior changes range from personal choices for energy use to how the next building is built or what to look for in the next lease option. Information is power.

ENERGY DASHBOARDS

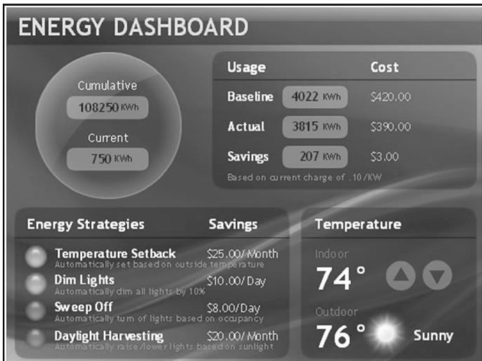
A 'dashboard' is a video terminal configured to display pertinent energy metrics. These are akin to company displays of profit, share value, or productivity; utility water reduction goals, and fund raising thermometers. As a minimum, these raise awareness and communicate that the company is paying attention to energy consumption. Additional benefit occurs when the viewer gains the sense that they have some control of the outcome from their personal choices. Miniature dashboards and messages can also be created for computer screen 'gadgets' and phone 'apps'. Communicating time-of-use periods (high priced utility times) can be valuable and can be done in a variety of ways.

See **Figure 23-5.** Dashboards represent the 'pulse' of the building or facility, and the use of real time values and bottom-line metrics. Values can be presented in the form of numbers, bars, and dials. Content can include:

- Demand, energy, carbon, dollars
- Current and cumulative savings
- Comparative values, goals, or benchmarks
- Impending events like on-peak periods
- Consumption of different end uses
- Supplemental information screens

The dashboard can become a battleground between audience interests. It is possible to alternate displays daily or weekly. Displayed items will often include a visual means to see whether the value is good or bad, improving, etc. Dashboards add value by bridging the gap between data and information in an inviting way. An energy dashboard can be engaging to an end user when unattended in a common area, such as in a lunch room or lobby; here the environment for the curious is without pressure. Of course, the same data points can be logged and trended with conventional DDC controls.

Pictures aside, the use of current data is a fundamental improvement in feedback effectiveness. Seeing the results of a choice a month later is literally 'old news'. The quicker the feedback comes, the tighter the control.



Energy Dashboard Highlighting Installed Conservation Measures, and Their Savings Results.
Source: beaconenergy.net

Energy Dashboard Highlighting Actual vs. Benchmark Energy Use
Source: dglogik.com

Energy Dashboard Highlighting Current Demand and Various End Uses (Sub Metered)
Source: nrel.gov

Figure 23-5. Energy Dashboard Examples

DEPUTY EFFECT

Discovering the triggers for energy savings requires knowledge of the equipment, systems, controls, and processes, as well as the 'what and why' of energy management science. There is an educational effect from an energy survey that adds depth to the staff; engaged customers will retain and use the new insight going forward to identify additional savings opportunities. Here, the change in behavior is increased awareness of energy use and cost, and hunting for improvements. This 'deputy effect' can occur with O&M staff, green-team members, or end users. Some examples:

- Task lighting can provide considerable savings when work is stationary by reducing overhead lighting power
- Exterior lights may not need to run all night
- Does there really need to be five copy machines on this floor?
- What is using power at night?
- The paint curing oven is twice as big as the product cart we put inside it
- Plating operation can be moved to off-peak times
- If operations can be scheduled, load factor will improve and electricity cost will reduce
- If equipment with waste heat is located next to a process, can we use it?
- The stack temperature is high, which may mean there is fouling
- Can we lower the condenser water temperature in winter?
- The large air compressor runs all night and weekend for the sake of one area that needs some standby pressure
- The new oven can be gas-fired and less costly to operate than electric
- If we switch to air-dry paint, can we eliminate the baking step entirely?

A weakness of any energy reduction program is when the guidance comes from an outside consultant and the expertise drives away at the end. Without this expertise, impacts to existing measures and new opportunities may go unnoticed. The best chance of sustained savings exists when expertise exists in-house; however this is not always practical, especially for smaller companies. Solutions may include remote readings and reports (a measurement and verification task), repeating the energy survey work every few years, training, or a retained part-time energy professional. In general, the more the concepts involve and are shared with the people who remain at the facility, the better chance of the effect lasting.

SUB METERS

The discussion of sub meter hardware, installation and O&M considerations is beyond the scope of this text. However, it is useful to point out conditions where feedback from sub metered energy flows have merit. Some examples are shown in **Table 23-6**.

Table 23-6. Conditions That Can Benefit from Sub Metering

Individual buildings that are on a master meter
Individual tenants that are served from a common meter
A manufacturing facility with more than one significant function or profit center
A commercial office building housing a data center that consumes 50% of the electricity in 2% of the floor space. The mixed occupancy defies quantifying the separate uses for benchmarking or monitoring improvements unless sub metered
District heating and cooling in a campus setting with multiple buildings sharing the common supply
A single equipment item within a facility that is the single largest energy load
When overall facility use is to be accounted for by end use

SAVINGS FROM O&M AND BEHAVIOR

O&M Savings

See **Chapter 8 – "Building Operations and Maintenance"**

Behavior Savings

Savings depend upon the extent of current wastefulness, the percent of the waste that behavior can influence, and the percent that is captured from a change. It also depends on what constitutes a behavior-related measure. To gauge the potential of behavior savings, multiple metrics were used and collectively they suggest a range of possible savings. Values in **Table 23-7** suggest a range of 5-15% for commercial and industrial low cost measures and roughly half of that for end user behavior measures. There may be exceptional case studies where a perfect storm of problems existed, but those should not set expectations for normal results.

Additional factors affecting savings values in **Table 23-7**:

- Customers seldom implement all proposed measures. Even when low cost measures are listed separately, it is normal that less than half are implemented [14].
- Low cost measures seldom affect all facility end uses. For example, a 10% reduction to an end use that represents 10% of the total energy use is a 1% reduction.
- With regard to cost, it makes a difference whether the savings occurs with electricity or fuel so when a Btu of natural gas energy has half the purchase price a Btu of electric energy, dollar savings represent half of the energy savings.

Table 23-7. Approximate Range of Savings for Commercial and Industrial Low Cost and End User Behavior Measures

Values given here are not firm, but are provided to show a pattern of general consensus. Low cost measures include savings from end users, O&M, and management choices. For end user behavior measures alone, a best guess of half of the stated values is used.

Metric Used	Low Cost Savings	End User Behavior Savings
A rule of thumb for typical potential savings through energy management, based on consensus and years of experience, has been 5-15 % from low cost measures, 15-30% savings for 3-5 year project return, and 30-50% for long term intensive measures [6, pp. 5].	5-15%	2-8%
Manufacturing energy waste has been estimated at 37% [7, 8, Note 1]. Of the total possible savings, most of it is in the form of waste heat and most of that not economically recoverable. *Author assumption is that 1/3 of energy waste is not from waste heat and half of that is accessible from O&M and behavior savings combined. Of the waste heat portion, half is recoverable and a fourth of that can be harvested with O&M practices.*	8-10%	4-5%
Benefits of feedback from utility metering vs. non-metered collections have been estimated to be 20% [9]. Less savings would be anticipated for C&I sectors compared to residential because fewer end uses are discretionary *Author assumption is that residential savings will be double C&I savings, for a 1/3-2/3 split of the given 20%*	-	4%
Conservation as a result of utility inclined block rates has been estimated by one source at 10% [10]. Results of water use inverted block rates has been demonstrated to produce system reductions of 5-8% [11]; however, for electricity there is very little firm evidence on the magnitude of this tradeoff, and none that is based on a large-scale systematic empirical study [12, pp 3]. It is also unclear as to the permanence of inverted block rate behavior changes *Author assumption is that residential energy response will be half of the water benefit*	-	2-4%
Electric consumption reduction of 6% was reported for a municipal water plant as a result of monitoring process metrics that included energy use [13].	-	6%
A study showed continuous feedback compared to monthly feedback increased residential natural gas usage savings from 5% to 12% [2].	-	5-12%

- Off peak electric savings, such as ghost loads, have low dollar value. Where the customer electric rate includes demand charges, nighttime savings do not affect demand; further, off-peak energy rates may be half of regular rates. Collectively, energy saved in off-peak hours may have only a fourth of the dollar value of daytime electric savings.
- Sometimes savings claims are in the form of percent of total available savings are available from low cost measures, and this can be true within the context applied. For example, '50%' represents '50% of whatever the total savings are', a different unit of measure than "percent of energy use." The newer the building, the less capital projects are considered and the greater the proportion of total opportunities rest with low cost measures and, for a brand new building, 100% of available savings will be from low cost measures. For older buildings with attractive capital expense projects (system change or equipment replacement), low cost measures may represent 10% of total savings, with only a portion of that from behavior measures.

ADDITIONAL RELATED TOPICS

For brevity, some topics were not included but are noted here for related additional study

- End use disaggregation (computer algorithms to derive end use patterns from interval data)
- Adaptive occupancy control for patterned end uses
- Just-in-time manufacturing
- Carbon incentives
- Site vs. source emission metrics
- Data normalizing methods

Summary

Energy management efforts that are related to behavior seek to influence choices from employees, customers, and visitors that create energy savings. Persistence of behavior measures are less firm than bolted-down projects, but are still attractive because they are low cost. A general barrier to behavior changes exists when the end users who create the saving are not the ones receiving the savings. Feedback combined with communication is effective when it links choices to results and creates accountability and ownership, either real or perceived. Feedback can be in the form of dollars, emissions, pride, security, or other attributes. The more closely the feedback connects individual choices to results the better, helping remove anonymity. The sooner the feedback occurs after the choice the better, encouraging self-regulation. Operations and management (O&M) departments can achieve very good savings with proper tools, feedback and incentive. Other energy cost control strategies with behavior components include utilities, and business management structure.

In all cases sustaining savings from behavior over the long term requires ongoing attention and incentives, and there is a strong tendency to backslide. The best hope of maintaining these savings relies on ongoing feedback that give people with choices a reason to care about energy use. Whether sub metered machine usage vs. production, long term trends, utility data comparing year-to-year usage, or regional or sister operations performance, something needs to form the basis to identify whether usage is reasonable or not, is backsliding or not, and is improving or not.

Citations:

(1) *Crafting Normative Messages to Protect the Environment*, Cialdini, R., Dept. of Psychology, Arizona State University, 2003
(2) *The Effect of Goal-Setting and Daily Electronic Feedback on In-Home Energy Use*, van Houwelingen, J.H., and W.F. van Raaij, Journal of Consumer Research 16(1), 1989
(3) *Commercial Energy Auditing Reference Handbook*, 2e, Fairmont Press
(4) *Business Expenses*, U.S. Census Bureau, 2007, Detailed Statistics by Subsectors and Industries
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(6) *Energy Management Handbook*, 8e, Fairmont Press
(7) *U.S. Manufacturing Energy Use and Greenhouse Gas Emissions Analysis*, Nov 2012, Energetics Inc., prepared for Oak Ridge National Laboratory for U.S. DOE
(8) *Manufacturing Energy Consumption Survey*, 2010, Released March 2013, U.S. EIA
Note 1: Loss assumptions from (5) were applied to corresponding end uses in (6). Ranges of loss for two end use categories were averaged to arrive at a single figure
(9) *Water Plan: Water Conservation Assumptions*, California Dept. of Water Resources, 1989
(10) *Household Electricity Demand*, Revisited, Reiss, P. and White, M., June 2002
(11) *Effectiveness of Conservation-Oriented Water Rates in Tucson*, Cuthbert, R., American Water Works Association journal 81, no. 3, 1989
(12) *Equity Effects of Increasing-Block Electricity Pricing*, Borenstein, S., Center for the Study of Energy Markets (CSEM), University of California Energy Institute, 2008
(13) *Innovation around the PI System*, Gallon, G. and Sepehr, M. OSIsoft Users Conference, 2014
(14) *Customer Implementation Survey Data*, 2007-2011 Energy Audit Low Cost Measures, Colorado Springs Utilities, 2013

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Chapter 24

Special Topics

A—DATA CENTER EFFICIENCY

CONTENTS

HIGH ENERGY USE INTENSITY IN DATA CENTERS

Data center energy use per SF is high, usually an order of magnitude higher than other facilities except for some manufacturing applications. Patterns and rules of thumb are elusive since computer technology changes usually find ways to put more capacity in a smaller space. A key indicator is watts per SF, and values of 100+ watts/SF are not uncommon, while older sites are 20-50 watts/Sf. Where data centers are embedded inside an office building, the energy use for the data center area dominates the rest of the building. In such sites, establishing EUI for the building vs. the data center is difficult unless sub-metered. (**Fig 24-A1**).

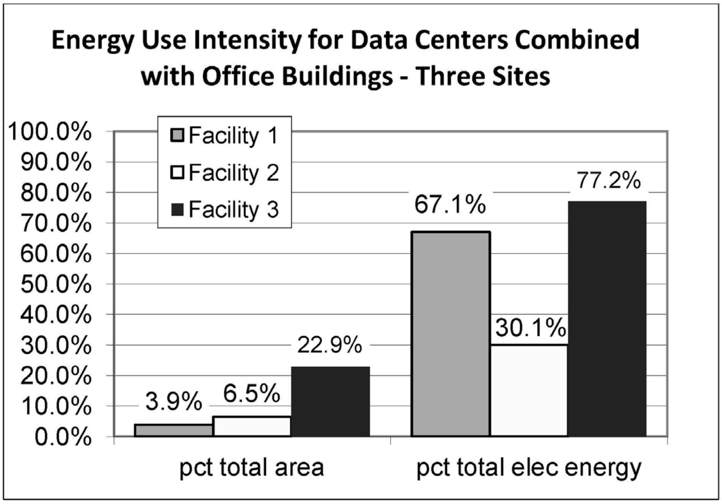


Figure 24-A1. Portion of Building Area vs. of Electric Use for Three Data Centers

Source of data: sub meter records. Facility 1 was a heavy use site.
Facility 2 and 3 were average use.
Electricity values are total of IT load and mechanical load.

Energy Use Intensity (EUI)—Per SF

EUI for data centers varies directly with the equipment density. With installed densities observed in the field from 20-100 watts/SF, the EUI will vary widely. For example, a data center with future provisions may have large underutilized areas initially. The best way to gauge energy intensity is from meter data. Utility data serves this purpose when the building is entirely a data center but when data centers occupy only a portion of a building, energy use will be elusive unless recorded on a sub meter or UPS units.

	kBtu/SF-yr
Facility 1	4704
Facility 2	1053
Facility 3	1122

Energy Use Intensity for Three Data Centers

Source of data: sub meter records. Facility 1 was a heavy use site.
Facility 2 and 3 were average use.
Electricity values are total of IT load and mechanical load.

Steady Load Shape

Computer rooms/data centers are uniquely steady in load shape, weather independent, all sensible cooling, and humidity sensitive **(Fig 24-A2**). They are also unique in HVAC as one of the few loads that need cooling and humidification at the same time. Calculations for computer room energy savings calculations are often easy since the loads are steady. The largest electric loads are ones that practically nothing can be done about. The loads that can be influenced are HVAC and lighting. Lighting loads add directly to the cooling load so they should be as low as is practical. HVAC total energy input (cooling, fans, and pumps) can be nearly equal to the computer energy use. The bulk of the mechanical load is cooling which is a direct reflection of the heat dissipated in the data center room and is usually 15-30% of the total electric load, depending on the cooling efficiency, so cooling efficiency is a natural target. This is usually achieved by lowering refrigeration condensing head pressure or raising refrigeration suction pressure, or both. Humidification is very energy intensive, especially since there will always be simultaneous humidification and dehumidification and since these units are normally electric powered. Ironically, a large humidification demand comes from the unintentional dehumidification caused by the cooling coils—in some systems 10% or more of the cooling energy spent is on unintended dehumidification. Therefore the cooling apparatus dew point is a natural target, by either lowering the space rH, or by raising the chilled water temperature if chilled water is used. Results of altering data center rH can be dramatic. The difference between theoretical cooling load and total mechanical load is related to pump/fan energy and the additional parasitic load it creates, plus cooling load for lighting and UPS equipment. Where oversized or redundant equipment is operated, fan energy can be a significant portion of a data center cooling load and variable speed control of fans can be viable.

DATA CENTER POWER FOOD CHAIN

In data centers there is a 'food chain' of energy use, and at the top of the food chain is the computers themselves. The series-effect of how energy is used in a data center produces an amplifying effect that is shown in the diagram. *Thus, each unit of energy reduction at the computer itself will yield 1.2 to 1.6 units of energy savings at the meter.* Because of the

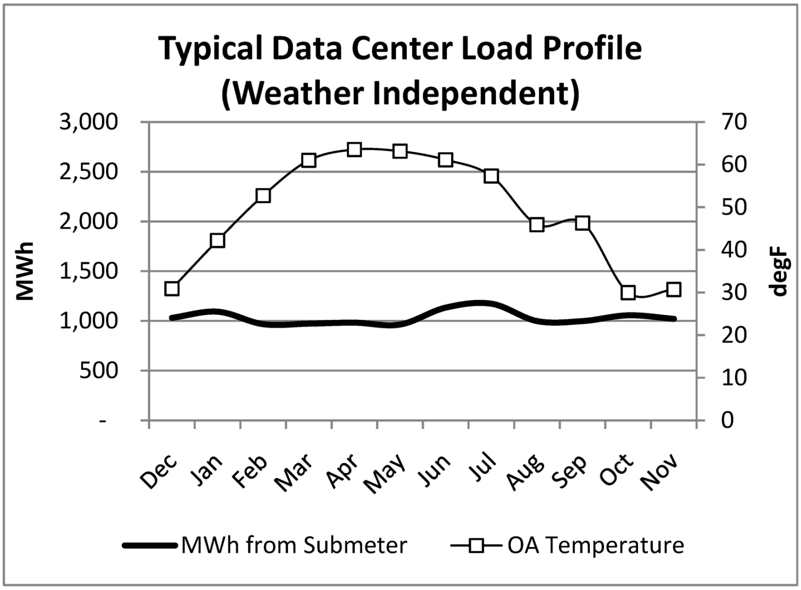


Figure 24-A2. Data Center—
Representative Load Shape vs. Outside Air Temperature
Source of Data: Agilent Inc.

amplifying effect, the #1 thing to look at for energy reduction in a data center is the computers themselves.

Another viewpoint helps illustrate the value of focusing first on the computers: If cooling energy is 20% of total data center annual energy use, a 10% reduction in cooling energy would reduce overall data center energy use by 2%, while a 10% reduction in computer energy use would reduce overall data center use by 12-16% (**Figure 24-A3**).

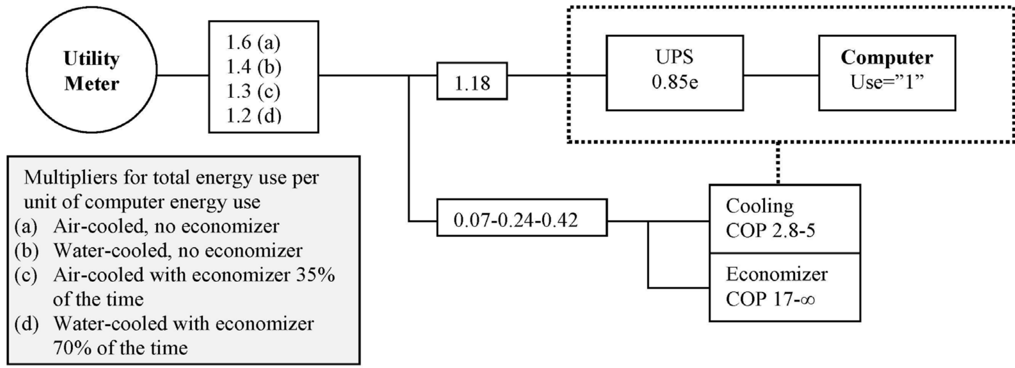
SERVER PART LOAD ENERGY USE

Since they are at the top of the food chain, and their use is amplified through the ancillary systems, computer end use is the first place to seek improvement.

MEASURES TO REDUCE COMPUTER ENERGY USE

Since they are at the top of the food chain, and their use is amplified through the ancillary systems, computer end use is the first place to seek improvement.

Figure 24-A3. Example Amplifying Effect of Computer Energy Use in a Data Center



- Factors vary by system particulars.
- Amplification factor varies by UPS efficiency, cooling efficiency, and economizer hours.
- UPS efficiency depends on type and load. Typical double-conversion UPS may have a 95% efficiency at full load but are normally employed in an array or are oversized such that actual load is 50% or less in service, causing lower UPS efficiency. Example shown assumes 85%e.
- COP of 2.8 is typical for air-cooled refrigeration equipment at design summer conditions, somewhat higher annually but head pressure relief is limited by false loading controls such as flooded evaporators and condenser bypass. COP 3.0 assumed for air cooled systems operating year round in mechanical cooling mode
- COP of 6.0 is typical for water-cooled refrigeration equipment, but is closer to COP 5 with pumps and cooling towers incorporated at the heat removal system
- Economizer power use varies; some systems do not have economizers at all. Water cooled system economizers (pumps and cooling tower) typical 0.2 kW/ton and COP-17. Various other economizers have similar efficiency. A true air-economizer (just opening dampers) is considered zero energy or COP-infinite
- Economizer hours vary by local and design style. For this example, the 35% for air-cooled assumes refrigerant run-around with 20F approach and 60F supply air, limiting hours to <40F outside air dry bulb and an annual cooling COP of 7.8. The 55% for water-cooled assumes chilled water cooling and a combined 10F wet bulb approach and 60F supply air, limiting hours to <50F outside air wet bulb and an annual COP of 13.4. Both estimates were based on Colorado Springs weather.
- Indoor data center fan energy excluded

Develop Computers That Use Less Power

An obvious benefit would be equipment that uses less to begin with, reducing the scale of the usage. Using computers less is also obvious, but beyond the scope of this text.

Control Part Load Computer Losses

Conventional server design over-builds processing power so it is "never slow" and thus is at part load most of the time. Unlike personal computers, data center server power requirements for busy and non-busy are almost indistinguishable.

Some computing tasks may be truly constant in which case there is nothing to change. (In mechanical terms, applying a variable speed drive to a constant flow pump is a waste of money). Some applications are critical or regulated and best served with traditional 'dedicated iron'. However, many (perhaps most) commercial data centers do have variability in the computer work flow and non-critical data and can benefit from technology improvements that address part load power use. Ask any data center if their data is critical and they will answer 'yes', so that is another paradigm.

One source estimates 70% of server power requirement at 0% CPU utilization (Blackburn, 2008). A study of power for various servers indicated >80% power usage at 20% CPU load, but a sharp power decline below 20% CPU load (Vasan, A. et al, 2009). A data center in 2014 recorded power use on a rack where new server hardware was being added, indicating 80% power usage with the CPU idle (**Figure 24-A4**). With this, a generalized curve can be formed to illustrate the potential for savings by improved throttling means for computers, with the savings being the area between the two lines (**Figure 24-A5**).

A simplified example illustrates the concept of optimizing computer idle power. Consider a data center with equipment that has the characteristic persistent load as shown in **Figure 24-A5** and currently operating with a load profile that includes 50% of total time at 30% computer loading. Re-organizing computer workload so that the computers operate at 60% load during these times (half the computers) will provide considerable savings.

IT load amplifies computer load due to UPS loss, assumed to be 90% efficient.

IT load = computer power * 1/0.9 UPS e

Existing
- Full load, 50% of the year, all computers run at 70% load and 90% power
- Part load, 50% of the year, all computers run at 30% load and 77% power

Proposed
- Full load, 50% of the year, all computers run at 70% load and 90% power
- Part load, 50% of the year, half the computers run at 60% load and 87% power

		A	B	C	D	E	F
		computer load	qty of computers	time at this load	computer pwr	UPS e	IT power
Existing 1	full load	70%	100%	50%	90%	90%	100%
Existing 2	part load	30%	100%	50%	77%	90%	86%
	annual power = (B1*C1*F1)+(B2*C2*F2)						93%
Proposed 3	full load	70%	100%	50%	90%	90%	100%
Proposed 4	part load	**60%**	**50%**	50%	**87%**	90%	97%
	annual power = (B3*C3*F3)+(B4*C4*F4)						74%
	Annual savings						19%

Example calculation for optimizing computer idle power

Virtual Servers

Also known as virtual machines (VM) or load balancing. Unlike a household personal computer, data center servers are not good at 'throttling' energy use up and down as the computing work varies—a fully loaded vs. idling server may only see little change in power requirements.

Virtual server technology, aka virtual machine or VM, uses supervisory software to coordinate and concentrate the work of multiple servers and use fewer servers when load permits. For example, ten lightly loaded servers doing very little could have their work moved to 2 or 3 servers, allowing the other seven to be turned off. The benefit of this technology is allowing computer energy use to reduce when computing activity reduces. Not all data is suitable for virtualization, but a lot of it is. It is not a stretch for virtual server technology to reduce data center energy consumption by 10-20%.

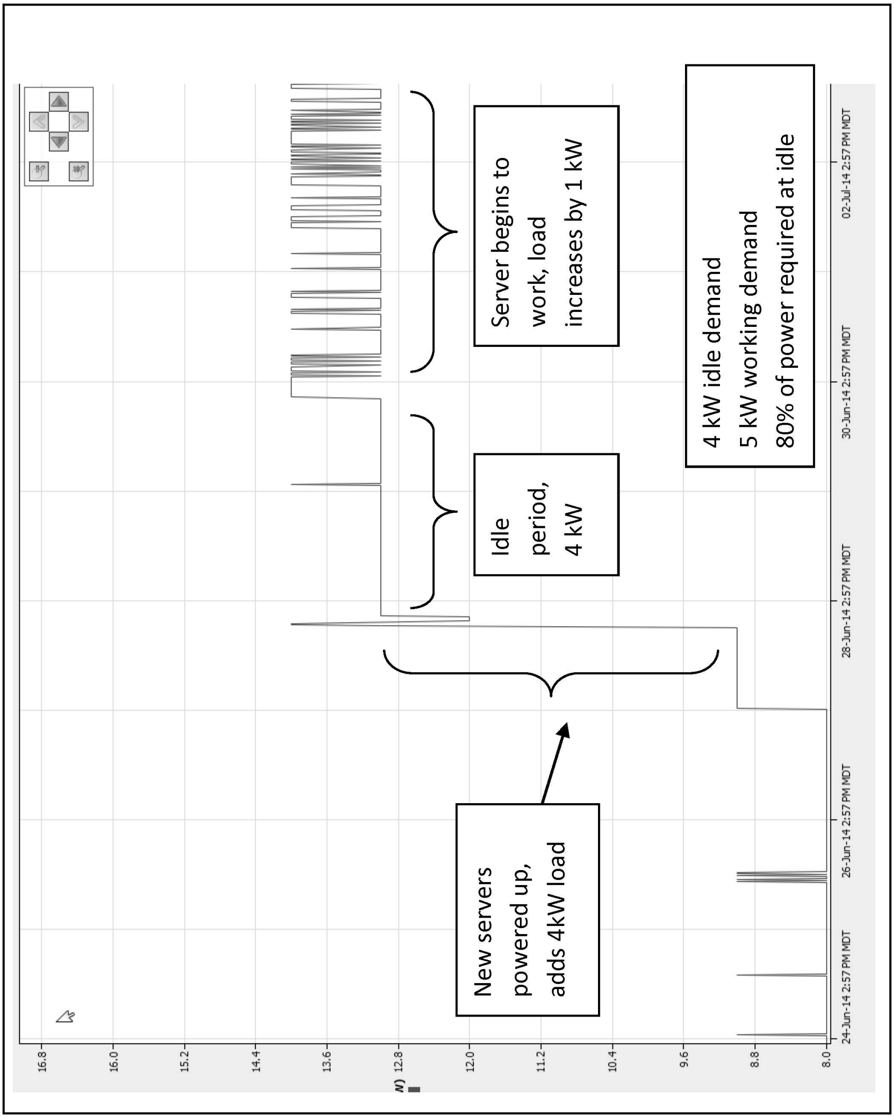


Figure 24-A4.
Server Idle Power
(Source: Oracle)

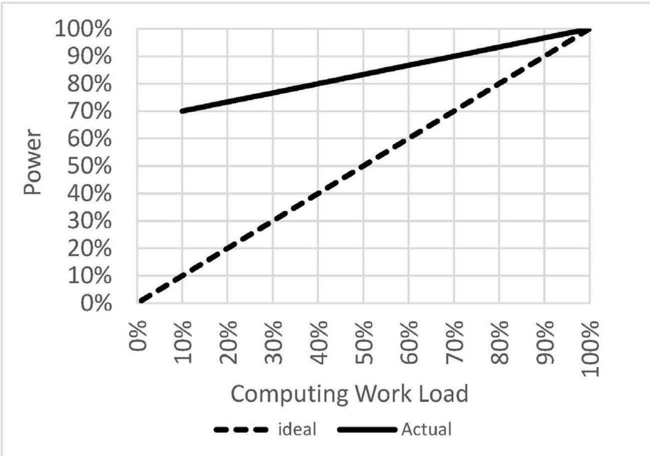


Figure 24-A5. Characteristic Server Power vs. Computing Work Load

One large server application reported *"about 60 watts of power when idle (0 requests-per-second, or RPS), 130 watts when at low-level CPU utilization (small RPS), and 150 watts at medium-level CPU utilization." Source: Making Facebook's Software Infrastructure More Energy Efficient with Autoscale, Internet article, Qiang Wu, August 2014.* While 'small' and 'medium' RPS values were not given, it illustrates that computer power at low work load is disproportional; in this example, distributing workload in order to avoid computer operation below 'medium RPS' levels showed energy reduction of 10-15%. Savings occur in periods of reduced server load and come from shifting load to make some servers inactive and keeping the remaining servers busy.

In another case, implementing virtual servers has prevented the expense of up-sizing a UPS unit with an expanding data center, a clear indication of power requirement reduction.

Supervisory software coordinates the loads on an array of machines, starting them, moving work between computers based on the load. This is analogous to staging an array of pumps, cooling units, boilers, air compressors, generators, even employees, as plant load changes; it is an old concept applied to complex computer arrays.

Arguments for virtual servers are the reduction in standby losses. Arguments against include added complexity and interactions that create new failure modes, critical data security, client confidentiality, and expensive proprietary software. It is reasonable that not all data is suitable for VM. One generalization maybe that data deemed suitable for cloud may also be suitable for a virtual server.

Hardware Throttling

Solutions are possible on a per-machine basis and would benefit smaller computing systems. Here, the processing load is gauged by the machine itself and additional processor cores are activated and de-activated as load increases. Another hardware approach for some servers uses a CPU clock multiplier to throttle processor speed to match the load, reducing power.

Cold Data Storage

This is a natural extension of variable load, where loads that are predominantly long term storage with infrequent access are segregated, stored, backed up, and then fully powered down. This is the equivalent of Backup tapes or CDs in a box or large scale sleep mode. Retrieval will be slowed, but power usage can approach zero.

Control UPS losses

UPS losses add directly to computer energy use. Changes here affect everything downstream, which is usually "all computers." To put this loss in perspective, consider all the attention given to efficiency improvements in cooling. Cooling energy is a fraction of dissipated heat in a data center, determined by the COP (coefficient of performance) of the cooling system. Depending upon the cooling system, 20-40% of data center electric use comes from cooling. Compare this to a UPS system operating at 80% efficiency which, by itself, adds 25% to the computer energy use and, paired with a COP=3 cooling system to remove the UPS heat, adds 33% energy use to the computers. When UPS operating loads are very low (25% or less), the energy use from UPS losses can equal all cooling energy use in a data center associated with the computers.

Computer load=1, cooling COP=4, UPS efficiency=0.8
UPS loss=(1/0.8)-1 = 0.25
Cooling load = 1/3 = 0.25

Efficiency of UPS equipment is stressed in this document, but must be balanced with other considerations, such as reliability (king), power quality, and ride-through.

UPS efficiency noted in manufacturer's literature is often not achieved in practice because operating loads are less than 100% and UPS efficiency drops off at reduce loads. UPS loads are less than 100% for a couple of reasons; (1) Data centers grow, so 'headroom' over-sizing for future expansion is a natural choice, and (2) UPS's reason to exist is reliability and reliability designs utilize multiple units to accommodate a failure without disruption of electric service to the computers.

To avoid single points of failure and have high reliability, UPS units are normally in pairs or arrays (N+1, 2N, 3N/2, etc). Consider a UPS sized for 70% of maximum load for future expansion, then connected to a hot spare such that if one fails the other takes over; this creates two UPS units operating at 35% load…and a throughput efficiency around 85%. This same scenario with a data center operating at 40% of computer capacity can yield a 15% load and 75% efficiency. Many UPS displays indicate input and output power, so the efficiency is easily monitored.

Double conversion UPS technology converts all of the power from AC to DC and back again. Batteries are integrated into this design and provide ride-through capability upon power loss, e.g. 5 minutes, 15 minutes, 30 minutes, etc. This technology is widely used.

Flywheel (rotary) UPS technology uses the energy stored in a spinning mass to provide electricity upon power loss—the flywheel acting like a short term generator. These can be used with or without batteries. Without batteries, on-site generators need to start and accept the load in a very short time, e.g. 15 seconds or so compared to 15 minutes or so with batteries.

Delta-conversion UPS technology is a variation on double conversion UPS design. An important difference is that in normal mode AC and reconstituted DC power work in tandem, compared to "all" AC power being converted to DC and the AC again all the time. During a power loss, the two systems behave similarly by inverting battery power to standard AC power.

Most UPS equipment manufacturers provide efficiency data for full and part load operation. **Figure 24-A7** shows a typical curve for a double conversion UPS. **Figure 24-A8** shows field UPS efficiency measurements along with the UPS loss contribution to total computer power.

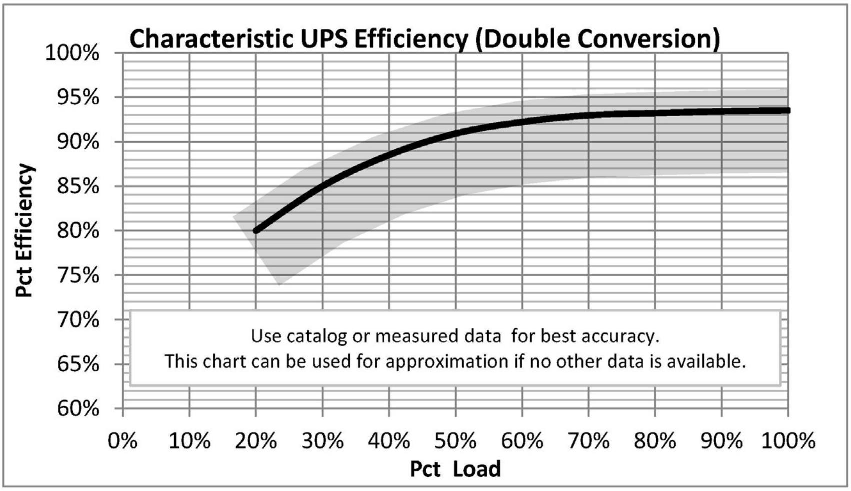


Figure 24-A7. Characteristic UPS Efficiency Curve

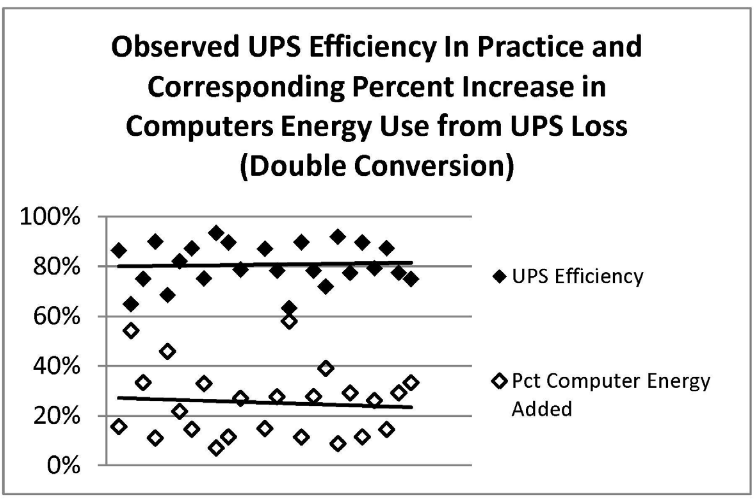


Figure 24-A8. Observed UPS Efficiencies

UPS efficiency expressed as UPS output kW/UPS Input kW).
Percent of computer energy added from [1/Efficiency)-1].
Trend lines for this data set indicates approximately 80% efficiency and 25% power add for computers.

POWER USAGE EFFECTIVENESS (PUE)

PUE is a metric used to determine the energy efficiency of a data center. PUE is determined by dividing the amount of power entering a data center by the power used to run the computer infrastructure within it. PUE is therefore expressed as a ratio, with overall efficiency improving as the quotient decreases toward 1. PUE was created by members of the Green Grid, an industry group focused on data center energy efficiency. Data center infrastructure efficiency (DCIE) is the reciprocal of PUE and is expressed as a percentage that improves as it approaches 100%.

PUE=total data center power divided by IT equipment power (ups power). Values should normally be <2; 1.25 would be very good; 1.5 is common. When cooling is all free, no lights, just a data center in a wheat field, PUE=1.

See **Figure 24-A9**. PUE measures the ratio of total power to computer equipment power, thus is a measure of ancillary power use only. In this metric, the computer energy use is considered "1," a value that the other systems revolve around but do not challenge.

> *Author's Note: There is a need for a different computer overall system metric. The PUE metric is the ratio of IT power to total data center power; this metric does a good job of making visible the non-computer loads such as UPS losses, lights and cooling. However, PUE treats the IT power as immovable, and underscores the paradigm of constant use assumption. For example, implementing virtual server technology would save significant power and energy, but would not result in an improved PUE score.*

Interpreting PUE values strives for a value of "1" which means there is no energy use except for the computers. Higher values represent higher non-computer loads, such as cooling.

PUE	*Level of Efficiency*
3.0	Very Inefficient
2.5	Inefficient
2.0	Average
1.5	Efficient
1.2	Very Efficient

Source: Green Grid, 2013

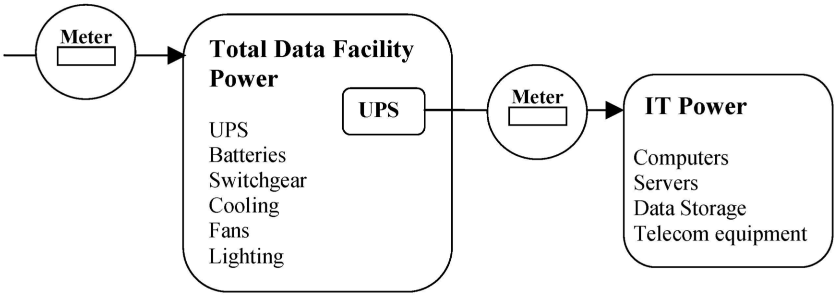


Figure 24-A9. Power Usage Effectiveness (PUE)

PUE = Total Facility Power/IT Equipment Power

Part Load Computer Rating System Needed

A metric to gage the part load computer efficiency is elusive, but needed. With it, computers could be as easily rated as a washing machine with a yellow label on it. Each case won't exactly fit the mold, but with *some* benchmark test in place, consumers will be able to shop for this performance item (like city and highway gas mileage, EER and SEER for cooling equipment, combustion efficiency and AFUE for combustion equipment) and manufacturers will respond. Possible units of measure could be transactions per second per watt ($^{TPS}/_{watt}$) or requests per second per watt (RPS/watt), but units to measure computer load are not universal. The ideal machine is one that has energy use turn down directly with demand, and the request of computers is the same. Many machines can turn down effectively to 50% load and the usage starts to flatten below 50%; this is not a failure. If a machine spends a large amount of time in the region below 50% load, it is probably oversized; and "percent" losses applied to lower loads have lower magnitude losses anyway. The point is to assert the need for power requirements to turn down proportionally when the computer is doing less.

MECHANICAL COOLING ENERGY REFLECTION

Other than conveyance systems (fans, pumps) that move the heat to the point of removal, mechanical cooling energy is a reflection of the data center heat load. Since the data center heat came from electrical consumption, the mechanical cooling kW can be predicted accurately from data center kW, if the coefficient of performance (COP) is known. COP is a measure of refrigeration cycle efficiency and defined as Btu out/Btu in. So, a machine cooling a 10 kW load with a COP of 2.0 will require 5 kW. Water-cooled refrigeration equipment will see reduced "lift" (difference between condenser temperature and evaporator temperature, plus heat exchanger approach values) compared to air-cooled machinery, and has a correspondingly higher COP. Summertime efficiencies for air-cooled equipment are low (low COP) because of the higher outside air temperature, but efficiency gains are seen at night and in winter. Since data center loads are continuous, seasonal efficiencies matter more than summer efficiencies.

Fig 24-A10 illustrates the reflection effect between kW load in a space and the mechanical cooling kW required to get rid of it. This is accurate for refrigeration cycle power and does not include fan and pump energy from auxiliary systems.

WATER COST

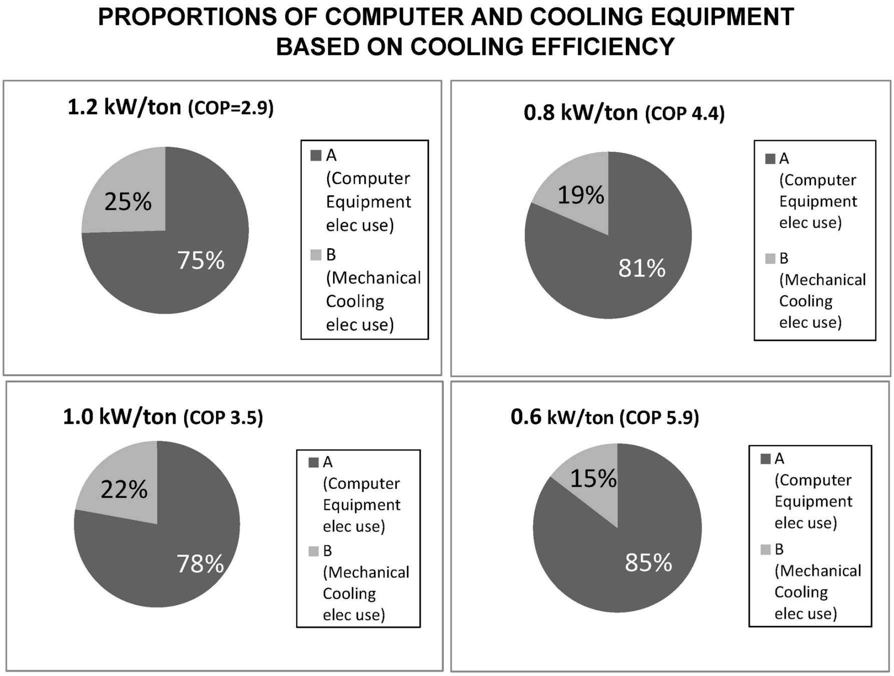
See **Chapter 22 Water Efficiency "Cost Tradeoff Between Decreased Electricity Use and Increased Water Use**." Water cooling offers improved refrigeration cycle efficiency, reduced hot weather demand, and increased cooling output for a given compressor. Aside from considerations of "water" in the data center, water is not free. The economics of anything that is water-cooled must consider the cost of the water and sewer charges. Whether there is net gain from the competing interest of energy savings and water cost depends upon their relative cost. Where electricity costs are very high, the choice is easy. When electricity cost is relatively low and water cost is relatively high the electric savings can be substantially or fully offset by water cost.

Example: Consider the default cooling system (that does not use water) as an "investment" in water/sewer costs to save electricity. Identify the required water use in gpm per kWh saved and translate to dollars spent vs. dollars saved according to the cost of each.

COOLING DESIGNS

Brute force mechanical cooling can be applied anywhere. Economizers and alternative cooling designs take advantage of local conditions, and benefits are amplified by other technologies such as equipment capable of higher temperatures at the outlet, and equipment capable of relative humidity swings.

Author's note: Taken to conclusion, computer equipment designed to run normally at with 100F and 10-90% rH entering air could eliminate all



Eff COP	Eff kW/ton	A (Computer Equipment elec use)	B (Mechanical Cooling elec use)
2.9	1.2	0.75	0.25
3.5	1.0	0.78	0.22
4.4	0.8	0.81	0.19
5.9	0.6	0.85	0.15

$$\text{"A"} = \left[{}^{1}/_{1+(0.284 * \text{kW/ton})}\right]$$

or $\left[{}^{1}/_{1+(1/\text{COP})}\right]$

$$\text{"B"} = 1- \left[{}^{1}/_{1+(0.284 * \text{kW/ton})}\right]$$

or $1- \left[{}^{1}/_{1+(1/\text{COP})}\right]$

Total Use, Computer Equipment + cooling = 1
Total kW(h) * A = Computer Equipment kW(h)
Total kW(h) * B = Cooling System kW(h)

Figure 24-A10, Theoretical Proportions of Cooling Power to Total Computer Room Power

Source: "Energy Efficiency in Computer Data Centers," Energy Engineering Journal, Vol. 103, No. 5, 2006.

Constant 0.284 is 3413 Btu per kWh/12,000 Btu per ton-hour; units are ton-hours/kWh

Each kWh of computer equipment use (dissipated in the data center room) has a corresponding 0.284 Ton-Hours of cooling load that follows it religiously.

mechanical cooling equipment and energy in most areas, by using only general ventilation and air exchange. I consider this a goal.

Computer Enablers for Cooling Efficiency

Purchase computers/servers that are tolerant of higher ambient temperatures.

The ability to operate normally at higher temperatures is an enabler for higher room temperatures, higher supply air temperatures, hot/cold aisle design, etc. The switch from ambient air temperature guidelines to server inlet air temperature guidelines allows de-coupling of room and equipment conditioning.

Purchase computers/servers that are tolerant of lower ambient relative humidity.

The ability to operate normally at lower relative humidity is an enabler for reducing humidification loads. This is especially valuable when operating at higher space temperatures where larger amounts of moisture are required to maintain a given relative humidity. 30% rH has been achieved in multiple data centers. The lower, the better.

Potential for Reliability Impact

Reliability is a huge consideration in data centers including the cooling provisions. Some data centers have 'sister' sites that make any one data center non-imperative, but other sites require operating as if shutting down is not an option. Some strategies for energy efficiency can impact overall reliability by introducing new 'single point of failure' points—these should be identified and discussed and may be unacceptable to the customer.

Interestingly, the CRAC unit with remote condenser one of the least energy efficient cooling systems offering large energy savings by a number of design alternatives, but also has among the least single points of failure of any cooling system. Depending upon the existing cooling design, optimization options may increase, decrease or have no change to overall reliability. Examples of measures that can have reliability impact to a data center:

- Air-cooled to water cooled. Common piping introduces a single point of failure. Large water pipes in the data center with leak potential. Backup source of water needed.
- Dry cooler conversion to evaporative fluid cooler. Ten or twenty dry coolers replaced by two fluid coolers will create a larger im-

pact upon a single point failure.

- In-row cooling and hot-aisle containment where server racks are fully enclosed will overheat more quickly if cooling is interrupted than those open to a large cold room. A justification for early design centers being so cold was additional minutes afforded by having a very cold room, allowing time for an orderly shutdown if cooling systems failed.

Retrofit Obstacles

- Change introduces uncertainty which is equal to risk in a business where reliability is priority #1.
- Many data centers are strategically located where electricity costs are low, which works against the business case for efficiency proposals.

Customers resist replacing existing equipment that is relatively new and working fine; expensive.

- Since equipment specifications are integral to cooling savings opportunities, cooling improvement opportunities may be limited in retrofit applications as long as legacy computers remain in service; for example one that must operate with 68F ambient air all around it, and 50% rH. It is possible for a few sensitive equipment items (that need colder ambient temperature or higher relative humidity to operate normally) to 'drive' the environmental requirements of an entire data center. In this case, equipment replacement or supplemental cooling for the sensitive equipment may be fruitful.
- Retrofit work introduces people not normally in the data center, which can be a security issue. Designs with the mechanical systems remote have an inherent advantage here.
- There may not be room for new technology, especially when being 'added' to some existing technology.
- New cooling systems that augment existing systems can have smooth retrofits when the new technology simply makes the heat load look smaller; then, if the new technology is being turned off for service or isn't working properly, the original system capacity is forgiving. This is much different than a new system replacing an existing one.
- Cooling systems, like any design, are usually designed to operate in a specific way. In turn, the equipment in place has operating

limitations that fit the original design but may not be agreeable to different strategies. A few examples are given here, but the message is to evaluate the system carefully to avoid creating problems while saving energy.

— Conversions involving water in an operating data center are risky.
— Anything involving cooling system down time will be difficult to accommodate when the data center cannot be shut down.
— Variable air flow may not be successful with existing raised floor supply air plenum, without some re-balancing. It is not uncommon for 'hot spots' to arise when reducing air flow. Sometimes these can be overcome by re-balancing and clearing out cables that block air pathways, but other times the air flow reduction will be limited by the supply air path conditions.
— Cooling equipment in place may not be amenable to optimization strategies and may delay savings until the equipment is at end of life. Examples:
— Split system CRAC units retrofits to add economizer modes are often cost prohibitive, and may be better candidates for unit replacement, one at a time.
— Chilled water systems shared with comfort cooling serving adjacent spaces will limit chilled water reset for the 'process cooling' without splitting the system.
— Direct expansion (DX) cooling can be difficult to convert to variable air volume if not designed that way to begin with (refrigeration issues). Where they can be converted, air flow reduction may be limited to 50% of air maximum flow which can limit fan savings.
— Motors serving constant flow duty may fail prematurely if mated to a variable speed drive, if it is of the electronic type.
— Water pipe sizes in place can limit cooling water reset efforts.
— Water throttling valves built-into cooling equipment may thwart savings from condenser water reset.

INTERACTION OF HVAC MEASURES

Many measures can be combined to advantage. Some examples:

- Raising room temperature allows a corresponding increase in supply air temperature which allows a corresponding increase in chilled water temperature (if applicable) and economizer hours.

- Hot/cold aisle containment concentrates the heat; the higher temperatures enable higher supply air temperatures, higher supply water temperatures, and reduced circulating flow energy.
- Some water cooled systems or pre-coolers can operate with evaporative cooling in summer and dry cooling in winter.

Interactions can also work against energy savings. Some examples:

- When room temperature is increased, humidification loads can increase because, for a given setting of *relative* humidity, higher temperature air holds more moisture. Things that affect moisture loss from a data center are increased as a result, so measures that tighten the envelope and reduce dehumidification become more important to prevent increased humidifier energy use.
- When computer equipment temperatures rise too much, rack cooling fans will start or speed up—these fans are small but collectively their energy use is not trivial. Some measures with the potential to have savings offset by rack fans are increased room temperature, increased supply temperature, and liquid cooling where supply water temperature has been elevated.

BASIC HVAC STRATEGIES

Increase room temperature

<u>Basis of savings:</u> Reduced heat gain through envelope. Reduced refrigeration cycle lift and economizer hours when supply air temperature or chilled water supply temperature increases.

Decrease room relative humidity

<u>Basis of savings:</u> Reduced humidity loss through envelope.

Excess dryness encourages electrostatic discharge (ESD) and computer hardware problems, therefore, some humidification will normally be needed for Data Centers to reduce static and sparks. Depending on the equipment manufacturer, 30% rH may be enough to prevent spark issues; humidifying to 50% rH should be discouraged. Since humidification consumes energy, raising humidity levels higher than necessary should be avoided, and lowering humidity levels is an opportunity to reduce energy use. In one large data center, changing set points to achieve 30%rH instead of 45%rH allowed multiple humidifiers to turn off, reducing electric load by 70kW (**Fig 24-A11**).

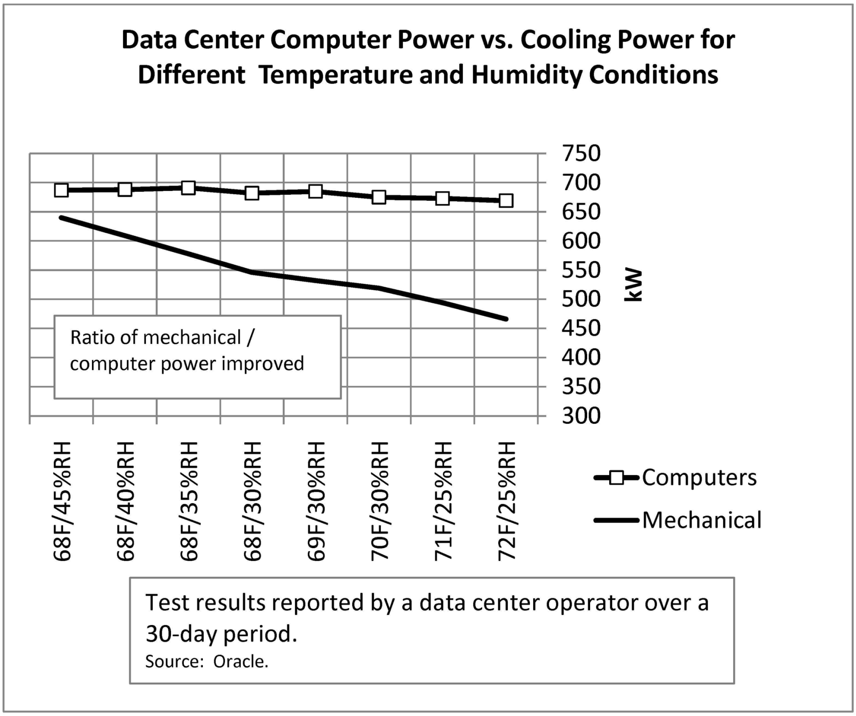


Figure 24-A11. Effect of Raising Temperature and Lowering Relative Humidity Settings in a Data Center

Decrease unintended dehumidification

Basis of savings: Reduced moisture loss from the space.

Without deliberate design to prevent it, much of the humidification can be due to the natural *dehumidification* effect of the cooling coils. To reduce unintended dehumidification and resulting extra humidification to replace it requires **raising the cooling coil apparatus dew point or lowering the room dew point**. Either approach, or both, spreads the two apart and stops the condensation.

- Lower the room dew point by increasing temperature and lowering relative humidity, after verifying that the environmental conditions meet the requirements of the equipment it serves.
- Increase supply air temperature. This will have a corresponding increase in coil temperature (chilled water or DX).
- Increase coil size so the average coil surface temperature is elevat-

ed. Although not currently a standard offering, it may be possible to request a DX computer room unit with a mismatched compressor/evaporator coil pair, e.g. a 20 ton compressor and a 30 ton coil. Increasing coil size is done at the design stage, and may increase unit size and cost.

- Increase chilled water temperature for chilled water systems. This is an integrated design choice also, and requires coil evaluation to assure the needed heat transfer be available at the higher temperature entering water, e.g. 50 degF entering Chilled Water (CHW). Increasing chilled water temperature is a design decision also, and will steer the design to a **segregated system** whereby the computer room chilled water is operated at a higher temperature than the building systems, through either a separate system entirely or a heat exchanger and blending valve arrangement. Combining process cooling with comfort cooling invariably creates energy efficiency compromises.
- Adjust controls for wider tolerance of "cut-in" and "cut-out" settings, allowing indoor humidity levels to swing by 10% rH or more. The savings from this measure come from reduced control overlap between adjacent COOLING units, i.e. the controls of one machine calling for humidification while the controls of the neighboring machine calling for de-humidification.
- Coordinate the unit controls to act more like "one big cooling unit" than multiple independent cooling units. The savings from this measure are similar to widening the control settings, which are from reduced control overlap between adjacent cooling units. The standard use of multiple cooling units, each with their own "stand-alone" controls, each with tight tolerance control settings, is a built-in opportunity for simultaneous heating/cooling and humidification/de-humidification. The overlapping controls are readily observed in the field and function, but with energy penalty. If such overlap can be avoided, energy savings will result. Note: depending upon computer hardware heat density, there will naturally be different conditions at different cooling units, so independent temperature control at each cooling unit is appropriate.
- Increase air flow to raise average coil surface temperature and air temperature. Note that this measure increases fan energy sharply and may exceed the avoided humidification savings.

Increase chilled water temperature
Basis of savings: Reduced refrigeration lift, reduced unintentional dehumidification, increased economizer hours.

An enabler for this measure is having the data center chilled water system separate from other chilled water systems. Data center cooling loads are "sensible only" and effectively process cooling. Elevating chilled water temperature from traditional 45 degF to 50-55F degF will raise the air handler apparatus dew point and eliminate unintended dehumidification, reducing load. Remember, it is only the sensible cooling that is of value in a data center. In one case, eliminating the unintended dehumidification load showed a 6% reduction in annual operating cost due to reduced load and a further 2% reduction for chiller efficiency gain. This reinforces the cost of unintended dehumidification in data center efficiency. For a given supply air temperature, increasing chilled water temperature requires larger chilled water coils (larger or deeper coil for more surface area) for equal capacity, which will limit this measure when applied to existing equipment. However, existing equipment can accommodate chilled water temperature increase in conjunction with supply air temperature increase.

Fundamental limitation for raising chilled water temperature.

Aggressive increases in chilled water temperature from higher rack inlet temperatures can see a limit in refrigeration cycle savings when there is a simultaneous low condenser temperature. Conventional refrigeration machinery requires a minimum separation between condenser and evaporator pressure, related to refrigerant and oil management within the machine.

Note that oil-free refrigeration solves a lot of this.

The required minimum differential pressure is usually expressed in temperature units, and may be 15-25F, depending upon the chiller design and refrigerant used. This value becomes a fundamental limit.

> Example: A chiller has a 17F fundamental separation between condensing and evaporating temperatures and the cooling coil has a 10F approach. Dry climate allows cooling tower to produce 65F condenser temperature in summer. Hot/cold aisle design allows 70F inlet air temperature which is produced from 60F chilled water. Refrigeration savings are promising based on 1 pct per degF, but the differential temperature at the chiller is only 10F and the chiller will not operate this way. Either the condenser

temperature must be increased or the chilled water temperature decreased until the 17F split is achieved, and the fundamental limit of the chiller increases the data center cooling energy about 1% per degree of compromise in cooling mode. **It is important to note that the chiller limitation does not apply to economizer mode** and so it is viable to have different air and water temperature control set points for mechanical cooling and economizer modes—when the compressor is off, compressor concerns no longer apply. For this example, leaving condenser temperature at 65F, the chilled water temperature in summer would be 48F (65-17), and the 70F supply air temperature would be achieved by throttling. However in economizer mode the chiller limit does not apply and the chilled water temperature can be 60F. The increase from 48F to 60F extends the hours of the year the water economizer is viable, keeping the compressor off much longer. See section "**Economizer Hours**."

When there is a control provision to protect the chiller against low internal differential pressure, it is possible for conservation measures like raising chilled water temperature or lowering condenser water temperature to be thwarted. Spending additional cooling tower fan energy to lower condenser water becomes a waste when it is met with provisions to blend the too-cold condenser water to maintain a given head pressure. Coordinating automatic control systems prevents this.

Increase supply air temperature

<u>Basis of Savings:</u> Reduced refrigeration lift (same condensing temperature, higher evaporating temperature), and increased economizer hours.

For this to work, the computers that operate normally with elevated inlet air temperature (e.g. 65-70F). For chilled water systems, refrigeration savings only occur if there is a corresponding increase in chilled water supply temperature, likely only possible with a separate chilled water system dedicated to just the data center process cooling.

Refrigeration savings can be estimated from the amount of supply air temperature rise seen at the refrigeration machinery. Reduced refrigeration 'lift' occurs when the same cooling load is processed using warmer supply air temperature or warmer chilled water temperature with same condenser temperature. Savings are approximately 1% power decrease per degree F rise (per reduction in lift). Refrigeration cycle savings assume the condensing temperature has not changed, which it should not. For example, if supply air temperature was formerly 57F and is now 64F, this is rise of 7F. If the refrigeration system is direct expansion CRAC units, expected reduction in compressor power is 7%. If the refrigeration system is chilled water and the supply water temperature is increased 7F, expected reduction in compressor power is 7%. If the refrigeration system is shared with comfort cooling for another area and supply water temperature is not increased, savings will be zero in compressor mode.

Reduce chilled water flow

Basis of savings: Reduced pump energy.

With constant load, there may/may not be variable flow provisions. If chilled water capacity is oversized or includes substantial reserve for expansion, flow may be excessive and can be identified with low differential temperature (dT = return – supply temperature). Remedies can include balancing valve adjustment, impeller trim, or adding variable speed drives and controls to the pump motors.

Reduce supply air flow

Basis of savings: Reduced fan energy.

Fan energy creates heat which is counterproductive in cooling a data center. At full load cooling, 10% of total cooling load is commonly attributed to the circulating fans, which is parasitic since the fan motors add heat and create even more load. Further, the 10% becomes a larger and larger portion of the room load at part load, which occurs when equipment is oversized. If cooling unit loads are high, fan energy will be in reasonable proportion, but if cooling units are oversized (redundant units running = oversized), constant speed fans will create unnecessary cooling load. Each 5 Hp of fan motor energy creates about 1 ton of extra cooling load. Like chilled water, the symptom for excess air flow is low dT (dT = return temperature – supply temperature). Remedies can include sheave adjustment, fan staging, and adding variable speed drives and controls to the fan motors.

Separate data center HVAC air systems from adjoining systems

Basis of savings: Reduced moisture loss from the space.

If the data center shares the building with other functions, any connections to a "house" HVAC system should be removed since whatever air is pushed in is removed along with moisture added in the data center. Coupled systems result in humidifying the entire building and excess humidifier energy use.

Calibrate and coordinate cooling unit controls

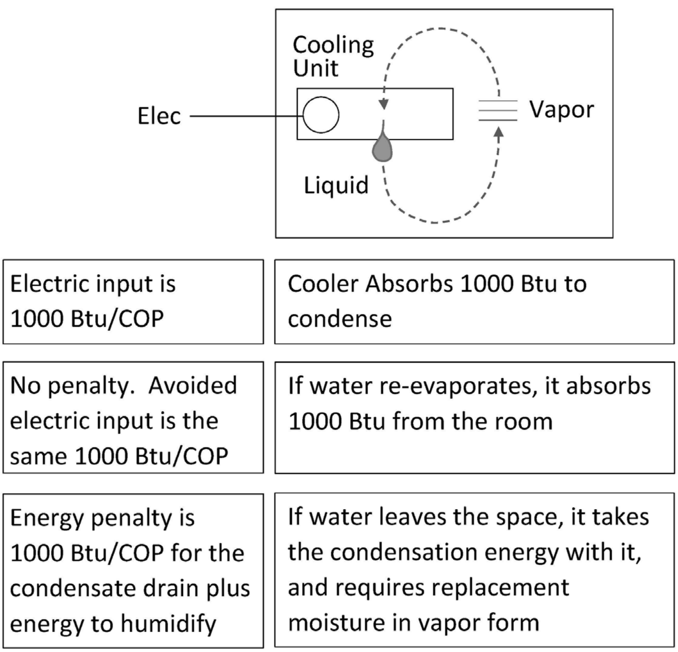
Basis of savings: Reduced overlapping heating/cooling, overlapping humidification/dehumidification from control fighting.

Calibrate sensors and controls every two years.

Note on overlapping humidification/dehumidification:

- Dehumidification accompanied by over-cooling and reheating is energy waste equal to the over-cooling and the matching reheating.
- When moisture is condensed on a cooling coil and then re-evaporated from the cooling coil—e.g. when condensate does not leave the data center envelope—it is not an energy loss. The energy used to condense water is released when the water is re-evaporated, using the water as a thermal flywheel. Assuming the re-evaporation is almost immediate, there is no parallel humidification load added.
- When condensed water is removed from the data center envelope, it is an energy loss: The energy to condense the water is lost, as no cooling benefit has occurred in the data center. Simultaneously, the drop in room moisture levels will create a drop in relative hu-

Determining Energy Penalty from HVAC Moisture Condensation in a Humidified Closed Volume



midity for a given temperature and the control response is usually to turn on a humidifier to replace it.

Clean heat exchanger surfaces

Basis of savings: Reduced approach temperatures and reduced refrigeration cycle lift; also extended economizer hours.

Applies to all forms of heat exchangers. Extent of fouling opportunity determines frequency. Things that require more frequent cleaning included open water systems (cooling towers), close fin spacing, exposure to air contaminants (no filters or low efficiency filters).

Reduce ventilation

Basis of savings: Reduced humidification in winter.

There should be sufficient ventilation for the few people in a data center, but no more.

Verify correct operation of cold weather controls

Basis of savings: Reduced false loading energy in mild weather.

For air-cooled CRAC units, limit the false-load condenser bypass and flooded condenser control action to cold temperatures. For water-cooled systems, prevent the cooling tower or evaporative cooler

This device is intended to bypass air cooled condensers for cold winter operation, but are sometimes found operating and raising head pressure in 70F weather.

basin heater from operating while the cooling tower is active; indoor sump is ideal and will eliminate the heaters.

HVAC SYSTEM VARIATIONS

Hot/Cold Aisle Containment

Basis of Savings: Reduced air flow from higher differential temperature felt by the cooling unit entering air, resulting in reduced fan energy. Increases benefits of other measures.

Refer to **Figure 24-A12**. This approach uses a reduced air flow through the computer racks which translates to reduced fan energy to circulate that air. Where widened inlet/outlet air temperatures are met with similarly widened inlet/outlet water temperatures (for water cooled systems), water circulation power savings will be created as well as air circulation power savings. For the savings to fully materialize, the low flow/high differential temperature air stream must be conveyed from the computer discharge point to the air circulating fan (remote or in-row). If the concentrated heat is diluted from room air or excess supply air, the savings will be reduced proportionally. Segregation methods include alternating hot/cold aisles, barriers to separate hot air from general room air, enclosed server enclosures with ducted supply air, outlet chimneys (outlet air ducts), vestibules that create hot or cold aisle plenums, etc. The common theme for all variations of this technology is controlling the cooling air medium such that the cooling supply air cannot reach the return air without passing through an active server rack, and the heated return air is kept segregated to the point of heat extraction.

With the two air streams controlled and kept separated, the differential air temperature drives the savings and is determined by:

- Rack warm air leaving temperature: This is limited by equipment environmental limits for normal operation
- Rack cool air entering temperature: Above some temperature (e.g. 65-70F) the cooling fans in the racks will start and erode some of the savings.

For water-cooled systems, the differential water temperature drives the savings and is determined by:

- Coil warm return water temperature: This is limited by the approach temperature of the coil, which is determined from coil selection and cleanliness. The approach temperature (how close the leaving water temperature can get to the entering air temperature) will never be zero and can be in the range of 5-10F. For example, 90F warm air returning to a chilled water coil may result in 80F return water temperature.

- Coil cool supply water temperature: Coil approach means cooling water inlet must be at least 5-10F colder than leaving air temperature. If the cooling water system serves nothing but the data center, the air and water system differential temperatures can be the same, just offset downward from heat exchanger approach. However there is nothing preventing colder than ideal supply water from being used, with throttling. The circulating savings will not be impacted (and would even be improved by colder supply water temperature), but refrigeration savings potential will be affected.

- Refrigeration compressor power (lift reduction) is not a source of energy savings with this measure, unless coupled with elevated evaporation temperature in the refrigeration cycle.

  Example: Elevating rack air temperature from 55F to 75F will produce refrigeration benefit in a CRAC unit since the air temperature rise has a corresponding temperature and pressure rise in the DX coil. But in a chilled water cooling system, the air temperature rise only produces a refrigeration benefit if the chilled water supply temperature also increases. If it is kept low for comfort cooling or other process use and merely throttled at the computer air handler coil to elevate the leaving air temperature, no refrigeration benefit will occur. This underscores the fundamental benefit of separating process cooling from comfort cooling.

Air flow benefits are highest when fans are variable speed and throttle with load, although care must be taken to detect and satisfy the zone of highest demand. Since computer heat will vary by rack, an 'average' hot air return could include a mix of under-loaded and overloaded servers which would not be good for equipment. Optimally, air flow through each server rack would be continually adjusted by modulating air flow to attain the target leaving air temperature, providing enough but just enough air flow. Attempts have also been made to control air flow automatically at each perforated floor tile. However, the control of a bulk inlet flow to satisfy a bulk outlet temperature can be flawed.

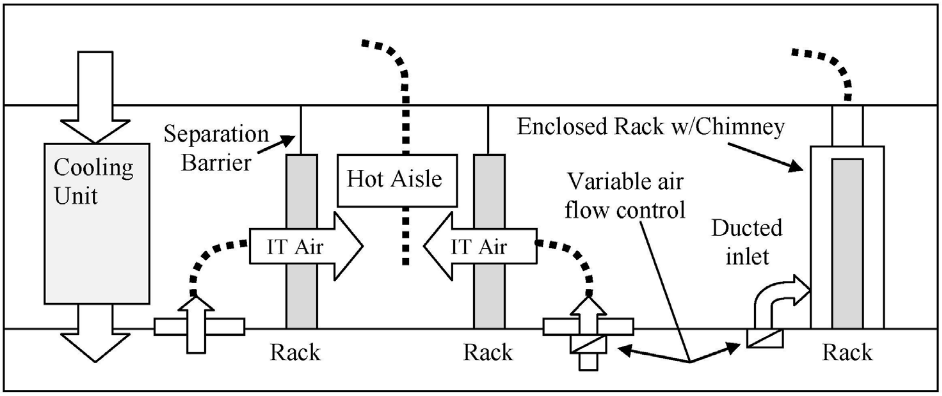
> Example: If one rack is too hot and the other rack has been turned off from server virtualization and is no longer producing heat, the bulk average temperature would read 'normal'. Having ample cooling air available to all racks will allow some cooled air to pass through a low loaded server, but the hot server will adapt by starting its on-board fans to increase local flow.

If supply air is discharged to the room from perforated tiles instead of ducting, it will behave like primary-secondary water flows and any excess primary air will "find a way" back to the fan suction, mixing with the carefully collected hot air; mixing path (shunt from supply to return) may be between racks, through ceiling registers, or through the racks themselves. This excess supply air flow means excess fan energy, which is contrary to the basis of savings of the measure itself.

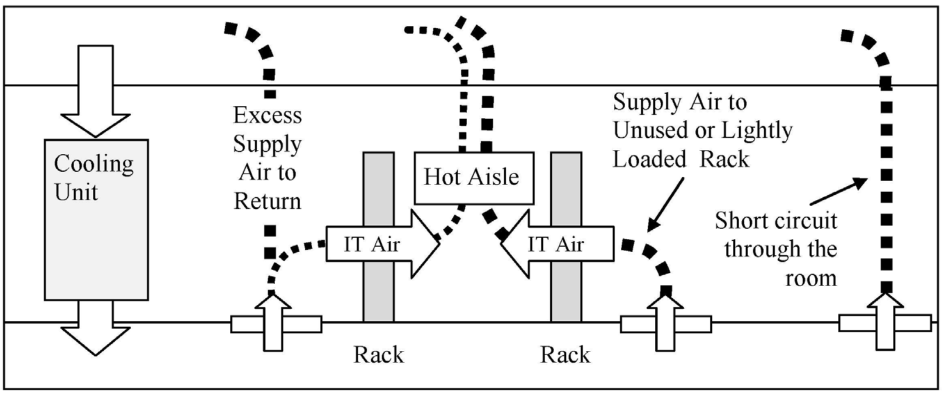
Conventional racks beside sealed enclosures with hot air chimneys.

Cold aisle containment vestibule. Cold air has no way out of here except through the server rack.

Variable flow, combined with elevated supply temperature, must be implemented in harmony with the computer equipment. If cooling air flow is reduced too much (too warm, not enough of it), the rack response will be to start or speed up the on-board cooling fans to protect the equipment. Energy use of a large array of small fans is not trivial and small fans are typically less efficient than large fans—thus energy savings at the main cooling fan can be offset by energy use of in-rack fans.



Hot Aisle Benefit Methods. Increasing air ΔT decreases air flow and fan power.



Hot Aisle Benefit Degradation. Excess supply air lowers ΔT and increases air flow.

Fig 24-A12. Hot/Cold Aisle Containment Methods

Adiabatic humidifiers

Basis of savings: Using the heat of the data center to evaporate the water instead of a new source of heat.

Use adiabatic equipment in lieu of steam or infrared pan heaters. Humidification in a data center is discussed in a separate paragraph Data Center Humidification, within this section. Humidifier types and energy characteristics are shown in **Chapter 11—Mechanical Systems, "Humidifiers."**

Dry cooler conversion to evaporative cooling

Basis of savings: Reduced refrigeration lift, increased economizer hours.

The dry cooler by itself creates high head pressure conditions in summer because the refrigerant heat must pass through two heat exchange steps before being released, and the condensing temperature is driven by ambient dry bulb temperature. On a 100F day, the "cooling" water temperature will likely be 120F. If evaporatively cooled, the condensing temperature will be driven by wet bulb temperature. For example, if summer wet bulb temperature is 70F, cooling water would be about 80F (with a respectable cooling tower or fluid cooler), a 40F reduction on that 100F day, with about 40% reduction in compressor power.

However these savings only exist in hot weather, while water usage would be year-round.

Direct air exchange economizer

Basis of savings: Cooling without ancillary energy related to heat exchangers. Increased economizer hours from no heat exchanger approach penalty.

The system would use conventional air handling units with mixing dampers. Increased contaminants from outside air would require aggressive filtration, with increased air moving costs.

In moderate and humid weather, cooling and humidifying could be met with outside air directly with very low energy input. However, cold dry climates would require non-adiabatic humidification and the energy input from the humidifiers would negate the energy savings from turning off the cooling equipment. Introduction of computer equipment that is indifferent to low relative humidity would make this technology instantly viable.

Liquid cooling

Basis of savings: Reduced heat conveyance energy (pumping vs. fans).

This system separates equipment and room heat loads, allowing flexibility to optimize both independently. This method is incorporated with the computer/server design (e.g. liquid cooled computer) to draw the heat away directly with liquid instead of fans, and will be a likely solution for cooling when computer power density is high. Heat pipes can be used to wick heat to a heat sink for removal by water, or the heat-generating computer components can be mounted on a tube-plate heat exchanger that forms a water wall.

Conveyance savings exist because, for each unit of thermal energy, moving it from place to place via air requires about one fourth the number of pounds of water compared to air (specific heat of water is ~1.0 Btu/lb-degF compared to ~0.24 Btu/lb-degF for air). Thus, pump energy is fundamentally less than fan energy. Once the heat is drawn away from the computer equipment, it must be rejected. The heat-bearing liquid can be connected to a conventional water-cooled chiller, air-cooled chiller, or dry cooler. In dry climates, heat rejection can be via a fluid cooler or cooling tower/heat exchanger directly.

Obvious operational concerns exist for water amidst sensitive electrical equipment. Design approaches include low water pressure to reduce the chance of leaking, and distilled water to reduce conductivity in case of a spill.

In-row cooling

Basis of savings: Reduced fan power by eliminating friction loss of raised floor air plenum.

Good potential for built-in hot/cold aisle design provisions (segregating hot and cold air). Cooling equipment is interspersed with data equipment racks. Concerns include proprietary tendency and scalability.

Evaporative pre-cooling for air-cooled heat rejection

Basis of savings: Reduced refrigeration lift, and increased economizer hours.

Evaporation in wetted media or direct spray cools the air entering the heat rejection device adiabatically, causing the device to "think" it is cooler outdoors. The cooling effect, whether on a condenser or dry cooler, allows the compressor to reduce power consumption. For example, power consumption on a 90F day may be the same as a 70F day in normal air-cooled mode. Common equipment for this is evaporative pads and spray nozzles.

> Note: An important consideration for evaporative pre-cooling is the reduced air flow from evaporative pads—due to the continuous heat rejection nature of a data center, there will be consider-

able hours when water is not being applied and the air resistance of the dry pads is parasitic, reducing heat transfer, increasing fan power, reducing economizer hours.

Indirect evaporative cooling supplement

Basis of Savings: Pre-cooling with via evaporation reduces load on mechanical cooling. For systems without any economizer, this can create savings from compressor-off hours.

This is a piggyback design, supplementing a conventional cooling design. The cooling effect is similar to back outlet cooling, making all or part of the load "disappear." The approach temperature of the evaporative cooler and air heat exchanger becomes a limiting factor.

Example: If it is 60F wet bulb outside, the fluid cooler (or cooling tower/heat exchanger combination) may produce 70F leaving water temperature, which is pumped to the indirect air-cooling coil. The coil may have a 5F approach temperature of its own, which means it could produce 75F air temperature—probably not enough to cool the computer. But with 65F supply air temperature and hot/cold aisle containment with 95F rack outlet air, the example indirect coil can reduce the load by (95-75)/(95-65) = 2/3.

With the cooling coil located at the air handler, this method has the operational advantage of keeping water away from the computers. Fan energy from CRAC/CRAH remains and will increase slightly from the air resistance of the coil that is in series with everything else in the air stream. As the only source of cooling, the water system will probably be supplemented with conventional chilled water cooling to 'top off' whatever the evaporative cooler cannot finish. As an add-on device, the original air-cooling system processes whatever heat the pre-cooling coil cannot absorb. The extent of cooling capability will depend upon the wet bulb depression for the local climate. The higher the supply air temperature allowed in the data center air system, the better the system works.

Back outlet heat exchanger

Basis of savings: Eliminates or reduces load on air handlers and their fan energy. In drier climates, when coupled with hot/cold aisle containment, heat removal can be achieved without any mechanical cooling at all.

This is a strap-on liquid cooling accessory to a standard air-cooled server rack. Using water as the conveyance fluid instead of air removes most of the fan energy used in data center cooling. This figure is fundamental from the specific heat of water vs. air (1.0 vs. 0.24). Essentially spot cooling, this approach eliminates the need for large air handlers, mechanical rooms and raised floor air plenums. The computer rack fans push the heat out the back, through the heat exchanger, which is hinged to the computer rack for access. Heat rejection would be from conventional chiller and cooling tower, which incorporates economizer mode.

Operational concerns from water-bearing equipment attached directly to the server racks can be mitigated with non-conductive working fluids and high reliability pipe designs.

ADDITIONAL OPPORTUNITIES FOR DATA CENTERS

Lighting

<u>Basis of savings</u>: Reduced internal load which reduces cooling load.

Measures include less overhead lighting, task lighting, higher efficiency lighting and occupancy sensors.

Envelope (vapor barrier)

<u>Basis of savings</u>: Reduced humidification load in winter

Seal the data center envelope with a contiguous <u>vapor barrier</u>, including above ceilings, below floors, and all doors to adjacent spaces and outdoors, penetrations, cracks, floor, wall and ceiling coatings to keep the moisture inside the room.

Sub metering

<u>Basis of savings</u>: Early detection of dysfunction once baseline is established.

Monitoring large distinct uses separately is a basic energy management strategy. For data centers, the entire data center building is a unique use. Within the data center, IT loads are the single largest use and can be measured independently of total data center load at the UPS output; some UPS equipment is configured to export this data directly for reporting and historical use. Where water is consumed (evaporative cooling), sub metering water use will provide information for water cost management and accounting.

Table 24-A1, Part 1. Summary of Data Center Energy Savings Opportunities—Measures Common to Most Cooling Systems (continued)

	Measure	New or Retrofit	Basis of Savings
Common to most Cooling Systems	Increase room temperature	Either	Reduced heat gain through envelope. Reduced refrigeration cycle lift and economizer hours when supply air temperature or chilled water supply temperature increases.
	Decrease room relative humidity	Either	Reduced humidity loss through envelope.
	Decrease unintended dehumidification	Either	Reduced moisture loss from the space.
	Increase chilled water temperature	Either	Reduced refrigeration lift, reduced unintentional dehumidification, increased economizer hours.
	Separate data center chilled water system from comfort cooling and other process cooling	Either	Enabler for chilled water optimization, preventing unintentional dehumidification.
	Refrigeration units that have a minimum required separation between condensing and evaporating temperatures	New	Enabler for increased savings from chilled water and condenser water reset without false loading.
	Cooling coils with low approach temperature (large surface area, multiple passes)	New	Enabler for heat exchange with low temperature penalty. Savings from refrigeration cycle, evaporative cooling hours, economizer hours.
	Amply sized heat rejection equipment (cooling towers, dry coolers, evaporative coolers) selected for low approach temperatures and low specific power for fans (kW per ton)	Either	Enabler for refrigeration efficiency measures and increased economizer hours. Reduced energy use for heat rejection.

Table 24-A1, Part 1. Summary of Data Center Energy Savings Opportunities—Measures Common to Most Cooling Systems (concluded)

	Measure	New or Retrofit	Basis of Savings
Common to most Cooling Systems	Increase supply air temperature	Either	Reduced refrigeration lift (same condensing temperature, higher evaporating temperature), and increased economizer hours.
	Reduce chilled water flow	Either	Reduced pump energy.
	Reduce supply air flow	Either	Reduced fan energy.
	Separate data center HVAC air systems from adjoining systems	Either	Reduced moisture loss from the space.
	Calibrate and coordinate cooling unit controls	Either	Reduced overlapping heating/cooling, overlapping humidification / dehumidification from control fighting.
	Clean heat exchanger surfaces	Either	Reduced approach temperatures and reduced refrigeration cycle lift; also extended economizer hours.
	Reduce ventilation	Either	Reduced humidification in winter.
	Verify correct operation of cold weather controls	Either	Reduced false loading energy in mild weather.
	Reduce lighting	Either	Reduced internal load which reduces cooling load.
	Envelope (vapor barrier)	Either	Reduced humidification load in winter
	Sub metering	Either	Early detection of dysfunction once baseline is established.

Table 24-A1, Part 2. Summary of Data Center Energy Savings Opportunities—Measures for Cooling System Variations

	Measure	New or Retrofit	Basis of Savings
Cooling System Variations	Hot/cold aisle containment	Either	Reduced air flow from higher differential temperature felt by the cooling unit entering air, resulting in reduced fan energy. Reduced cooling water circulation energy if the high differential temperature is also felt in the water system. Increases benefits of other measures.
	Adiabatic humidifiers	Either	Using the heat of the data center to evaporate the water instead of a new source of heat.
	Dry cooler conversion to evaporative cooling	Retrofit	Reduced refrigeration lift, increased economizer hours.
	Direct air exchange economizer	Either	Cooling without ancillary energy related to heat exchangers. Increased economizer hours from no heat exchanger approach penalty.
	Liquid cooling	New	Reduced heat conveyance energy (pumping vs. fans).
	In-row cooling	New	Reduced fan power by eliminating friction loss of raised floor air plenum.
	Evaporative pre-cooling for air-cooled heat rejection	Either	Reduced refrigeration lift, and increased economizer hours.
	Indirect evaporative cooling supplement	Either	Pre-cooling with via evaporation reduces load on mechanical cooling. For systems without any economizer, this can create savings from compressor-off hours.
	Back outlet heat exchanger	Either	Eliminates or reduces load on air handlers and their fan energy. In drier climates, when coupled with hot/cold aisle containment, heat removal can be achieved without any mechanical cooling at all.

ECONOMIZERS

Water economizer on central water-cooled chilled water system: Chilled water systems already using a cooling tower can be augmented with a plate/frame heat exchanger to allow the cooling tower to provide cooling directly with the compressor off when wet bulb temperatures are low enough.

Water economizer on central air-cooled chilled water system: A parallel dry cooler or fluid cooler can be integrated into existing chilled water piping that is outdoors. In cold weather, the dry cooler or fluid cooler can run instead of the chiller.

Pumped refrigerant economizer on air-cooled CRAC unit: This is an efficiency upgrade to conventional CRAC units, including operational improvements by simplified refrigeration equipment. Conventional CRAC units have high energy use in summer like any other air-cooled system and, significantly, must run in winter weather. Compressor hours are long and cold weather operating efficiency is hampered by false loading provisions to allow stable operation, including flooded condensers (making the condenser behave as if smaller) and condenser bypass valves. The pumped refrigerant design allows the compressor to be stopped whenever outdoor air is sufficiently low (e.g. 50F). At that time, a refrigerant pump is started, using the indoor and outdoor coils as a runaround heat recovery unit—cooling indoor air by virtue of outdoor air, with no compressor. The pumped refrigerant system economizer hours are increased directly as supply air temperature is increased. This design still suffers from high summer demand, but allows compressor off-hours in evenings and winter months for good savings.

Indirect air economizer: For split systems, a run-around coil can be installed to pre-cool return and reduce load on the compressors in winter, possibly turning them off. Equipment can be dry cooler or evaporative cooler.

Pre-cooling coil within a water-cooled CRAC unit: In lower temperature/drier weather, the fluid cooler (or cooling tower with heat exchanger) is capable of making much colder water than in summer mode. The pre-cooling coil can accept this cold water to pre-cool incoming air and then use the water throttling valves to allow the compressors to run as well if needed.

Table 24-A2. Economizer Comparison

Values are approximate. Where specific values of kW/ton are available, use them but be sure to incorporate auxiliary equipment and not just compressor power. Best accuracy will incorporate adjustable values at different temperatures. Criteria 'when viable' can be used to estimate annual hours of economizing. Tons load, hours of economizer operation, with the difference between baseline and economizer kW/ton determine annual savings. Water cost offset not included but may be significant depending on relative cost of electricity vs. water.

System Application	Economizer type	When Viable	Baseline cooling system kW/ton winter operation	Economizer mode kW/ton
Water-cooled chilled water system	Heat exchanger parallel to water-cooled chiller	OA wet bulb >10F below chilled water supply temperature	0.7 kW/ton including auxiliaries	0.25 kW/ton including auxiliaries
Air-cooled chilled water system	Dry cooler in parallel to air-cooled chiller	OA dry bulb >20F below chilled water supply temperature	0.9 kW/ton including pumping cost. Assumes head pressure control is active	0.2 kW/ton for chilled water pump and dry cooler fans
Air-cooled chilled water system	Evaporative fluid cooler or cooling tower/heat exchanger parallel to air-cooled chiller	OA wet bulb >10F below chilled water supply temperature	0.9 kW/ton including pumping cost. Assumes head pressure control is active	0.2 kW/ton for chilled water pump and dry cooler fans
Air-cooled DX CRAC unit	Pumped refrigerant run-around (dry cooling)	OA dry bulb >20F below supply air temperature	0.9 kW/ton (assumes flooded condenser and condenser bypass control)	0.2 kW/ton, refrigerant pump and condenser fans
Air-cooled DX CRAC unit	Pumped water run-around pre-cooling coil (dry cooling)	OA dry bulb >20F below supply air temperature	0.9 kW/ton (assumes flooded condenser and condenser bypass control)	0.2 kW/ton, just the water pump and dry cooler fans
Air-cooled DX CRAC unit	Pumped water run-around pre-cooling coil (evaporative cooling)	OA wet bulb >20F below supply air temperature	0.9 kW/ton (assumes flooded condenser and condenser bypass control)	0.2 kW/ton, just the water pump and evaporative cooler
Water Cooled DX	Pre-cooling coil (evaporative cooling)	OA wet bulb >20F below supply air temperature	0.8 kW/ton including auxiliaries	0.4 kW/ton presumes half load displaced

Economizer Hours

"Free cooling" hours are longer for data centers than most other facilities because (a) the cooling load is always there and (b) the facility is always operating, including nights and weekends. Economizer hours vary by climate. Without attempting to list cities, generalizations can be made based on prevailing dry bulb and wet bulb conditions compared to HVAC economizer operating parameters. For systems that turn compressors fully off in economizer mode, hours of benefit can be estimated using bin weather for the respective city or climate zone. Each cooling system with an economizer will have a defined temperature (dry bulb or wet bulb), below which the economizer will allow the compressor to turn off. Knowing the target temperature below which the economizer is viable allows estimating hours benefit from bin weather data. See **Appendix: Hours per Year Below Outside Dry Bulb and Wet Bulb Temperatures**. Systems that increase economizer hours (more savings) will be those with higher supply air or water temperatures, and equipment with lower heat exchange approach temperatures.

Note: Both examples reference **Appendix: Hours per Year below Outside Dry Bulb and Wet Bulb Temperatures, 8760 hours.**

Example 1: A chilled water system operating in Albuquerque is operating with 45F chilled water. A water economizer is in place with an overall heat exchange approach of 10F, allowing chillers to turn off below 35F wet bulb. A proposal is being considered for hot/cold aisle containment that would allow cooling with 55F water instead of 45F water. Using the same cooling equipment, this will allow chillers to be turned off at 45F wet bulb. In addition to data center fan savings, find the economizer savings. From the chart:

Economizer hours before (below 35F wet bulb)	2606 hours
Economizer hours after (below 45F wet bulb)	4792 hours

Increase of 2186 hours per year, compressors off

Example 2: Conventional CRAC units are to be replaced in Chicago due to end of life. Two economizer options are considered: one is a pumped refrigerant system that will allow compressors to stop below 40F dry bulb outside air (65F supply air, 20F outdoor coil approach, 5F indoor coil approach); the other is a pre-cooling coil connected to a fluid cooler that will allow compressors to shut off below 50F wet bulb outside air (65F supply air, 10F fluid cooler approach, 5F indoor coil approach). Compare hours of economizer operation. From the chart:

Economizer hours below 40F dry bulb	3140 hours
Economizer hours below 50F wet bulb	4143 hours

B— PERCENT PER DEGREE RULE OF THUMB FOR REFRIGERATION CYCLE IMPROVEMENT

***Statement on Rules of Thumb**: Rules of thumb are derived from observed patterns, within some boundaries. Simple rules of thumb can be claimed when, the behavior is reasonably linear. For all rules of thumb, the basis and context behind rules should be understood so that the limitations are taken into account. For example, some rules of thumb are limited to a certain range of temperatures or pressures considered 'normal' by the creator of the rule. Others may apply to a particular fluid or material. A rule of thumb that works in one climate may not work in another climate.*

> *Example: The rule of thumb of 1-1.5% savings per degree of change in a refrigeration cycle is actually a pretty good one. But consider applying it to an ultra-low temperature freezer, converting the unit into a cooler by raising its operating temperature from -70F to +40F. This would probably not work very well, but if it did, estimating energy savings using the rule of thumb would suggest the cooler would now be creating power on its own, which it obviously cannot.*

Rules of thumb are very useful but do not replace knowledge of fundamentals and systems.

A value of *1-1.5% power reduction per degree Fahrenheit* (F) has been used successfully for years to estimate the effects of either lowering condenser temperature or raising evaporator temperature in a mechanical cooling system. This concept shows up in a variety of energy conservation measures, all with the goal of reducing power requirements. Reviewing the underlying science will allow confident use of this rule of thumb and explain the range of values given.

The principles involved with this rule of thumb include lift, heat exchanger approach, coefficient of performance, refrigeration cycles, and sources of error. Since the work involves differences, pressure and temperature units do not have to be in absolutes. These terms and a Mollier Diagram (pressure-enthalpy, or p-h) were used to evaluate this rule of thumb.

Manufacturer's data is the most accurate source of power reduction from a change in operating conditions because it captures all the various influences in a bottom line live test. A Mollier (p-h) diagram can be used with before/after system conditions for accurate results. The rule of thumb can be used for reasonable accuracy, especially when incorporating the baseline lift (system lift before the change is made). Using the traditional 1-1.5% rule of thumb can overstate savings, but is 'safe' at the 1% level.

Review of important terms:

Lift is the difference between the high and low pressure regions a compressor works against. Compressor power is proportional to lift. If all the existing lift is removed, all of the existing work associated with lift is removed. So, 0-100% lift power reduction will follow a 0-100% reduction in lift.

> Note: The compressor has additional power requirements besides lift, such as fluid friction, bearing losses, and motor losses. If 'all' power came from lift, the task would simply be to find the change in lift as a portion of total lift.

In a refrigeration cycle, saturated conditions allow interchanging pressure and temperature. For measures that lower condensing temperature or raise evaporating temperature, power reduction can be estimated from temperature difference even though what the compressor really sees is a change in *pressure* difference. An example of low refrigeration lift is a water-cooled chiller (~40-55F lift). An example of high refrigeration lift is an air-cooled walk-in freezer with a blower coil (~120-140F lift).

Approach is a heat exchanger term that describes how close the leaving fluid on one side can "approach" the entering or ambient fluid on the other side of the heat exchanger. Oddly, there are differences in industry definitions of approach - what-compared-to-what depends on heat exchanger style - but the basic concept is the same. An infinitely large heat exchanger or infinite contact time will produce an approach temperature of zero but for practical purposes there is always some differential. The value of approach will vary by heat exchanger type, fluid, and sizing. See **Chapter 8 Building Operations and Maintenance "Notes on Heat Exchangers" and "Heat Exchanger Approach Diagrams."** Approaches for liquid heat exchangers are generally lower than those using air or gas. Turbulent flow (scrubbing at the boundary layer) and higher Reynolds numbers reduce approach; greater surface area or part load operation reduces approach; longer contact time reduces approach, and fouling increases approach. Examples of using approach to calculate lift:

- 45F chilled water with 5F approach, and 80F condenser water with 10F approach → (80+10) - (45-5) = 50F lift.
- 55F supply air with a 15F approach, condensing at 95F with a 20F approach → (95+20) - (55-15) = 75F lift.

  (Note the lift advantage using water cooling vs. air cooling for the same 40F heat exchanger leaving fluid temperature).

Approach values can be derived from manufacturer's data or measured in the field, but will vary depending upon load. This is because heat exchangers become effectively oversized as load decreases. For example, one chiller was found to have a condenser approach of 10F at full load, 7F at 75% load, and 4F at 50% load.

Coefficient of Performance (COP) is the ratio of output to input energy or power and is unitless since both terms are the same, e.g. Btu output/Btu input, and Btus cancel. Applied to refrigeration systems, COP is the ratio of refrigeration output to power input (in same units). This can be measured with the actual machine in service, and can also be derived from the cycle states of a Mollier (p-h) diagram using COP= (h1-h4)/(h2-h1). Compressor power can be derived from COP using kW/ton=3.517/COP.

State 1-2	Vapor compression
State 2-3	Condenser
State 3-4	Liquid expansion
State 4-1	Refrigeration effect

Fig 24-B1 compares theoretical results to values of % power reduction predicted by the conventional rule of thumb. The theoretical values consistently fall within the range of 1% and 1.5% power reduction per degree F change. The uncertainty error band becomes larger with larger changes. Estimates of 1.5% per degree F will likely overstate savings.

Fig 24-B2 incorporates baseline lift. The baseline lift is the range of high-to-low refrigeration temperature boundaries, approach included, before a change is made. By first calculating the baseline lift, a *single value* of percent power reduction per degree F is found on the chart.

Example: A water cooled chiller system is found making 38F chilled water with 75F condenser water. An efficiency measure would raise water temperature by 7F to 45F. An approach temperature of 3F is assumed for the evaporator and 5F for the condenser. Baseline lift, in temperature terms, is then (75+5) – (38-3) = 45F. From **Fig 24-B2**, power reduction would be ~1.1 percent per degree F, or 7.7% power reduction.

Fig 24-B3 shows refrigeration cycle efficiency for equal values of lift at different temperatures. Additional error is introduced when the rule of thumb is applied equally to measures affecting the low and high temperature regions of the cycle. Refrigeration efficiency is affected somewhat more at lower pressures and will respond differently to a lift change – a detailed explanation is beyond the scope of this article but part of it is visible in non-parallel lines shown in **Fig 24-B3**.

Sources of error:

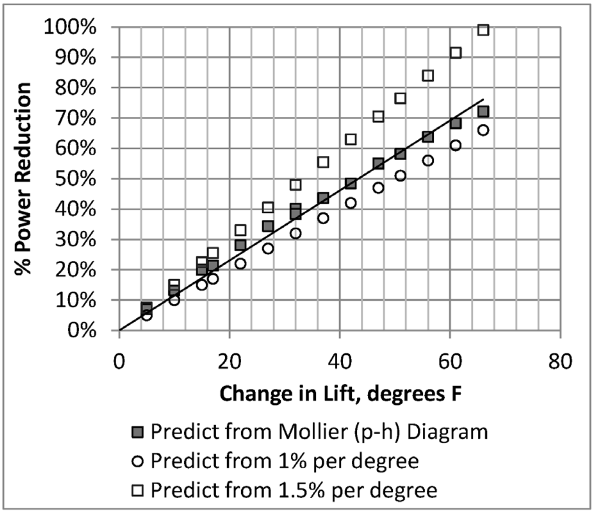
- Not incorporating baseline lift. This is the dilemma of working in "percentages" and is a watch-out for any/all rules of thumb using percentages. For lift-related power, the same one degree F change will make a 10% impact for an original lift of 10F, but only a 1% impact if the original lift is 100F.
- Not incorporating heat exchanger approach will understate lift. 55F supply air and 95F ambient temperature is incorrectly identified as 40F lift. If both air-cooled heat exchangers have a 20F approach, the true lift would be (95+20) – (55-20) = 80F lift. For this example, the error in savings would be ~14% (From **Fig 24-B2**, 1.08% reduction vs. 1.22%).
- Refrigerant used – the Mollier (p-h) diagrams used were for R-717 (ammonia). Different working fluids have different properties.
- Range of load. At part load, some things get better, some get worse. Seasonally, it's a mix.
- Machinery type. This affects actual energy use at the meter. Machine losses are not considered in a Mollier (p-h) diagram.

Summary: The refrigeration rule of thumb "1-1.5% power reduction per degree F of change" is useful if taken with a grain of salt. The range given is a catch-all for a number of variables. The most influential variable is lift which is responsible for the majority of refrigeration energy input. Actual machine test data is best for evaluating different conditions. Direct use of the 1-1.5% rule of thumb can overstate savings and so adhering to 1% is suggested. By incorporating baseline lift, a single value of percent power change per degree F is available. Other than actual machine testing, each method has unknowns. Experience and good judgment are needed to determine if savings de-rates are appropriate when using any rule of thumb.

Figure 24-B1: Rule of Thumb Error Band % Compressor Power Reduction vs. Degree F Lift Reduction (No Regard for Baseline Lift)

Dotted lines shows actual vs. estimated savings using the 1-1.5% rule of thumb without regard to baseline lift. Uncertainty increases with the size of the change in lift.

- Mollier (p-h) diagram prediction for power based on COP = (h1-h4)/(h2-h1)
- Rule of thumb prediction = degrees F change in lift * assumed % per degree F (1% or 1.5%)
- % power reduction = (P1-P2)/P1



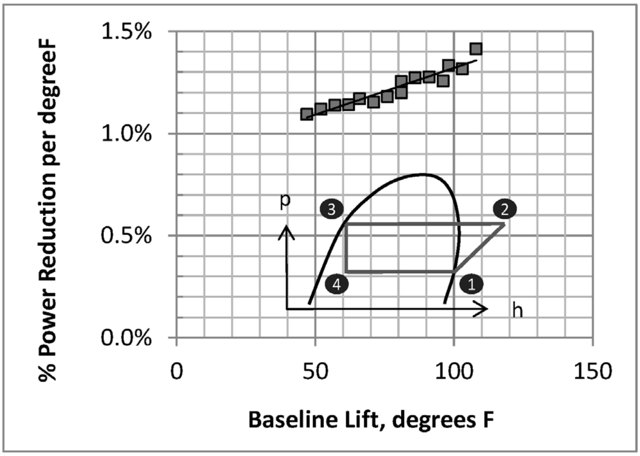


Figure 24-B2: Modified Rule of Thumb for Single Value of % Compressor Power Reduction vs. Degree F Lift Reduction (By Referencing Baseline Lift)

By first determining baseline lift, the % power per degree F value can be taken directly.

- Baseline lift = T high – T low temperature boundaries of the refrigeration system, including approach values.
- Mollier (p-h) diagram prediction for power based on
- COP = (h1-h4)/(h2-h1)
- % power reduction = (P1-P2)/P1

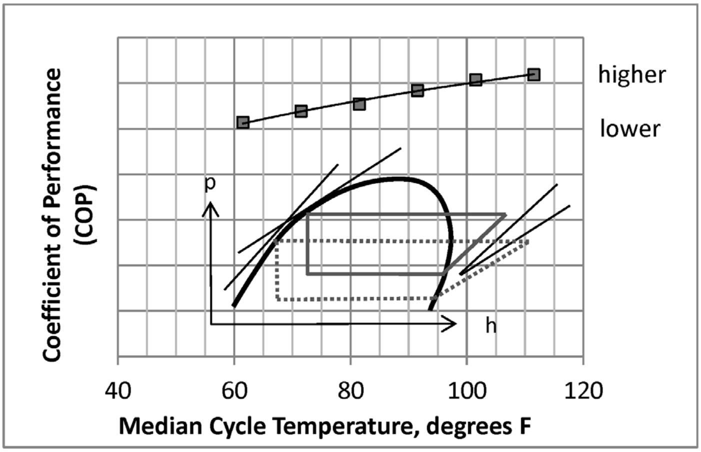


Figure 24-B3: Coefficient of Performance vs. Median Cycle Temperature

Refrigeration cycle power requirements are more sensitive to lift change on the low temperature side than the high temperature side. Measures affecting low side lift will have greater benefit, per degree F change, than on the high side.

- Median temperature = [(Condenser HX fluid + approach) + (Evaporator HX fluid – approach)]/2
- (COP) = (h1-h4)/(h2-h1)

C—EARLY REPLACEMENT BUSINESS CASE

Energy savings alone can seldom pay for full normal replacements, but can often be shown to pay for early replacement costs or upgrades during normal replacement. This is analogous to comparing the cost of basic equipment (new or replacement) against the energy savings of efficiency upgrades to justify the upgrade investment.

Example: if a rooftop unit has a design life of 18 years and is 16 years old and running fine, the default is to leave it alone, and early replacement is only justified if savings are significant. In this example, 88% of the anticipated life is already used, so 88% of a replacement project cost would be attributed to normal replacement. Thus, 12% of the replacement project cost would need to be weighed against the energy savings of the new equipment to justify the early replacement. Clearly equipment that is almost new is very difficult to justify replacing, but the older it gets the more compelling the business case gets. A characteristic of "normal expected life" is when the repair costs have escalated sufficiently that keeping the thing running is no longer practical. By charging the measure payback with the value of the remaining life of the equipment, replacement at any time can be justified if the savings are high enough.

Existing: EER-9, AFUE-80%
Available: EER-12, AFUE-92%
Total Cost: 100,000
Age: 16 years
Expected Life: 18 years
Early Replacement portion (2/18=11%) = $11,000
Total energy savings = $2900 per year
Payback: 3.8 years

If energy savings are charged with the full replacement cost
Total cost: $100,000
Total energy savings = $2900 per year
Payback: 34.5 years

D— LEASE ARRANGEMENTS— EFFECT ON ENERGY PROJECT INTEREST

There are many variations of lease arrangements, including what is included in the lease and what is not. When certain lease provisions exist, there can be a polarity of interests of the building owner and tenant with respect to investments in energy efficiency and energy conservation. Unfortunately, there is little uniformity in the definition and use of lease types, sometimes even in neighboring cities, and so a characterization of who-pays-what by lease type is not possible. However, some useful generalizations can be made for certain combinations of lease provisions with respect to where incentives do/do not exist for energy improvement projects.

Table 24-D1. Some Basic Lease Arrangements

Applications of the lease terms vary widely, so presume nothing by the name or general description, and focus on the fine print.

Type Basic	Description
Fully Serviced	Tenant rental payment includes all operating expenses and maintenance, including common areas and grounds. Common in multi-tenant buildings and hi-rise offices.
Gross Lease	The tenant pays a fixed amount of rent and the landlord is responsible for payment of taxes, insurance and other costs associated with owning the property, including common areas.
Net Lease	The tenant pays base rent for their leased space, plus a portion of the maintenance fees, insurance premiums and other operating expenses for the property and common areas.
Triple Net Lease	Tenant pays base rent for their leased space, all fees and operating expenses associated with their leased space, and a portion of all fees and operating expenses associated with common areas. Often this lease is used for a free-standing facility.

Lease Provision Effect on Incentives for Energy Improvements

1. Any lease arrangement where the building owner pays for utility costs (e.g. the Gross Lease) the building owner is motivated to invest in energy efficiency and conservation measures where such financial investment will:
   - generate a return on investment in the form of an enhanced cash flow for the building owner;
   - provide the property with a relative competitive leasing advantage over similar properties due to reduced energy costs for prospective tenants; and/or
   - result in an increased asset value which exceeds the cost of the investment (i.e., long term return on investment for the building owner).

2. Any lease arrangement where the tenant pays for utility costs, but the building owner pays for capital improvements (i.e., the Net Lease),
   - the tenant has no incentive to make building improvements, since the building is not theirs.
   - the building owner has little or no motivation to invest in energy conservation measures, because the building owner incurs the cost, while the tenant reaps the benefit of the building owner's investment in the form of reduced utility bills.

Although the building owner may see increased value for the building over similar properties from being less expensive to operate, the building owner's benefit from the investment is generally deferred to a later date and, therefore, not traditionally seen as a particularly valuable return on investment. In this scenario, where a cost is incurred by the building owner but the benefit is received primarily by the tenant, there is a fundamental disconnect between building owner and tenant, with the usual result being a resistance by building owners to invest in energy conservation measures.

Because of the widespread use of Net Leases, major sectors of the leased property market are largely untapped with respect to energy efficiency and energy conservation projects. To correct this requires addressing the landlord tenant arrangement to create scenarios for mutual benefit, connecting

- the landlord's interest in positive cash flow, asset preservation and

full occupancy,
- the tenant's interest in avoiding capital expense in leased space and reducing monthly out-of-pocket operating costs.

Such "win-win" outcomes can be achieved with a Net Lease, by using alternative lease contract language which allows building owners to pass through to tenants costs incurred by the building owner which result in cost savings to tenants. Language such as this allows building owners to recover costs incurred from energy conservation projects when the tenant receives benefits from the reduced utility costs, better aligning the interests of building owners and tenants and accommodating the transient nature of the tenants. This type of language would become an enabler for landlords to make improvements that have a capital return within their business horizon.

The following is one example of how to numerically establish what is fair between the two parties. Conventional economic measures for energy efficiency projects are not a good fit for leased spaces. Landlord and tenant have different business horizons (how far into the future they look) which makes simple payback period (SPP) an 'uncommon denominator.' Using internal rate of return (IRR) improves on this since it blends SPP with measure life. A special use of IRR uses the measure life for the landlord's economic view, while the tenant's IRR uses a measure life set equal to the life of the lease, reflecting their business horizon. With this method, IRRs can be aligned by iterating the percentage of shared savings between the two parties.

Scenario to illustrate the economic method that can align landlord and tenant:

Landlord is leasing a building to a single tenant and a price has been agreed upon for the building as-is. Tenant pays all utilities. An efficiency proposal is discussed whereby improvements would be paid for by the landlord and the utility savings shared between landlord and tenant.
- Length of lease: 5 years
- Life of measure: 20 years
- (P) Project cost: $160,000
- (A) Annual savings: $30,000
- Division of first cost: proportional to lease life/measure life (5/20),

payable to landlord at beginning of lease (shared investment)

- Division of savings during lease term (landlord-tenant): The percent split for the energy savings between landlord and tenant is iterated until the internal rate of return is equal for both parties

Results (**See Figure 24-D1**):

- In this example, the two IRRs aligned when <u>61%</u> of the utility savings went to the landlord.
- **Tenant** pays landlord $40,000 at the beginning of the lease. Tenant receives all utility savings during the lease (5 yrs. * $30,000 per year = $150,000) and refunds the landlord $91,500 during the lease (61% * $30,000 = $18,300 per year = $91,500 total during the lease); keeping the remainder of utility savings over the life of the lease (39% * $30,000 = $11,700 per year = $58,500). Thus the tenant spends $40,000 and receives $58,500 during the lease. Simple payback = $40,000/$11,700 = 3.4 years for a 5 year lease. With a '**measure life' of 5 years** (the tenant's business horizon= lease length), <u>IRR = 14.2% for the tenant.</u>
- **Landlord** funds the project, less the tenant's one-time payment, paying $160,000 – $40,000 = $120,000. The landlord also receives 61% of the utility savings as a refund from the tenant, or $91,500. Thus the landlord spends $120,000 and receives $91,500 during the lease ($18,300 per year). Simple payback = $120,000/$18,300 = 6.6 years. With a **measure life of 20 years** (the landlord's business horizon), <u>IRR = 14.2% for the landlord.</u>
- IRRs are the same. With this method (equal IRR), both tenant and landlord share investment and reward fairly.
- Additional iterations can identify the exact IRR crossing point if desired.

Figure 24-D2 is a series of charts that illustrate the landlord tenant alignment for different values of overall simple payback period, measure life, and lease term length, using the process outlined in the given example and **Figure 24-D1**.

Aligning internal rates of return using different business horizons is the primary tool to bring landlord and tenant together. To succeed, lease arrangements such as these need to be mutually beneficial. Discussions and provisions in the agreement may be similar to guaranteed savings performance contracts.

- Third party review or standardized impartial agreement terms.
- Clarify whether economic analysis is based on total project cost or incremental cost above normal replacement costs.
- Clarify where risks are being shared, and how they will be controlled.
- Clarify how savings will be verified initially and ongoing, and how to reconcile savings shortfalls or excesses.
- Identify ongoing maintenance responsibilities. For example, it would be unfair to require a tenant to make a consistent refund of energy savings when those savings are being eroded from inadequate maintenance.
- Adjust for utility cost escalation during the life of the project and lease terms.

Adjust for variations in weather effects and building usage changes.

Additionally, the landlord can become 'stuck with it' if the next tenant to lease the space does not agree to the shared funding arrangement, or if the space remains vacant after the first tenant's lease has expired. And if the building is sold, there is a question of whether the invested money will be recovered in building value. These types of concerns may limit such lease provisions to larger and longer term tenants.

Figure 24-D2
Landlord-Tenant Division of Project Savings for Equal Internal Rate of Return

Notes for Figure **24-D2** chart series
Simple payback period for the chart title is the cost of the project divided by the total savings, as if the landlord did this without partnering with the tenant.

For each value of overall simple payback period, there is one chart showing the savings division, and a second chart showing the internal rate of return. For each chart pair, the corresponding values are found for measure life and lease length.

Basis, to find equal value for landlord and tenant:

1. Measure life for landlord is the full measure life; measure life for tenant is the length of the lease.
2. Initial cost of the efficiency project is pro-rated based on length of lease and life of measure. For example, a project has a 15 year life and the lease is 5 years; the tenant pays 5/15 = 33% of the project cost.
3. Measure savings proportions for landlord and tenant identify how much of the total savings each party receives. The proportions are iterated until internal rate of return (IRR) for landlord and tenant are equal. For each iteration, the savings split forms separate landlord and tenant simple payback periods; this, combined with the respective landlord and tenant measure life, is the basis for the IRR calculations for each party.
4. At the point where IRRs match, the percent of savings to the landlord are identified. Tenant receives the balance of the utility savings.
5. Charted conditions are limited to:
   Measure life >lease term
   Lease duration 2-10 years
   Minimum SPP=2 years
   IRR for both parties >0%

Minimum Attractive Rate of Return (MARR) can be applied if desired. For each value of overall simple payback period, there are two charts. The second chart is the matching IRR point. Applying the MARR, if desired, can rule out solutions that are below the threshold.

Chart Use Example:

Project Cost	$ 100,000
Savings/yr	$ 20,000
Overall SPP	$100,000/20,000$ per year = 5 years
Project Life	10 years
Lease Term	5 years

From Charts for 2 year Simple Payback (5 year lease/10 year life):

Pct of savings to Landlord	37.3%
IRR for both Landlord and Tenant	8.0%

Iteration #1, increments of 10% landlord portion of savings

L=Landlord	20	Measure Life (years) = Landlord Project Term View
T=Tenant	5	Lease term (years) = Tenant Project Term View

IRR	-6%	61%	0%	53%	4%	44%	8%	35%	11%	25%	14%	15%	17%	4%	19%	-9%	22%	-26%
	L	T	L	T	L	T	L	T	L	T	L	T	L	T	L	T	L	T
Total project cost, K$	160																	
Pct cost to each party	75%	25%																
Cost to each party, K$	120	40																
Total annual savings, K$	30																	
Pct savings to each party	10%	90%	20%	80%	30%	70%	40%	60%	50%	50%	60%	40%	70%	30%	80%	20%	90%	10%
Savings to each party, K$	3	27	6	24	9	21	12	18	15	15	18	12	21	9	24	6	27	3

At 60/40 split of savings the tenant's IRR is higher; at 70/30 split of savings, the landlord's IRR is higher. The crossing point is between 60-70%.

Cash Flow, iteration 1. Use to find the point where IRR values cross

Savings Division	10% L	90% T	20% L	80% T	30% L	70% T	40% L	60% T	50% L	50% T	60% L	40% T	70% L	30% T	80% L	20% T	90% L	10% T
Project cost	(120)	(40)	(120)	(40)	(120)	(40)	(120)	(40)	(120)	(40)	(120)	(40)	(120)	(40)	(120)	(40)	(120)	(40)
Year 1	3.0	27.0	6.0	24.0	9.0	21.0	12.0	18.0	15.0	15.0	18.0	12.0	21.0	9.0	24.0	6.0	27.0	3.0
2	3.0	27.0	6.0	24.0	9.0	21.0	12.0	18.0	15.0	15.0	18.0	12.0	21.0	9.0	24.0	6.0	27.0	3.0
3	3.0	27.0	6.0	24.0	9.0	21.0	12.0	18.0	15.0	15.0	18.0	12.0	21.0	9.0	24.0	6.0	27.0	3.0
4	3.0	27.0	6.0	24.0	9.0	21.0	12.0	18.0	15.0	15.0	18.0	12.0	21.0	9.0	24.0	6.0	27.0	3.0
5	3.0	27.0	6.0	24.0	9.0	21.0	12.0	18.0	15.0	15.0	18.0	12.0	21.0	9.0	24.0	6.0	27.0	3.0
6	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
7	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
8	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
9	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
10	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
11	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
12	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
13	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
14	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
15	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
16	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
17	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
18	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
19	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-
20	3.0	-	6.0	-	9.0	-	12.0	-	15.0	-	18.0	-	21.0	-	24.0	-	27.0	-

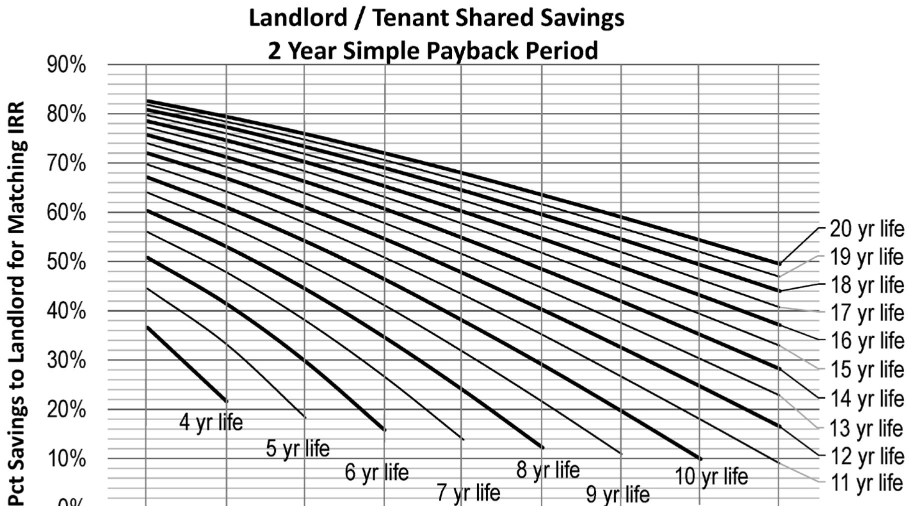
Figure 24-D1 (continued)
Example Iterative Solution for Aligning Landlord and Tenant Internal Rate of Return (IRR) for Shared Cost and Savings

Iteration #2, increments of			1%	landlord portion of savings								
L=Landlord				20	Measure Life (years) = Landlord Project Term View							
T=Tenant				5	Lease term (years) = Tenant Project Term View							
IRR	13.9%	15.2%	**14.2%**	**14.2%**	14.5%	13.1%	14.7%	12.0%	15.0%	10.9%	15.3%	9.8%
	L	T	L	T	L	T	L	T	L	T	L	T
Total project cost, K$	160											
Pct cost to each party	75%	25%										
Cost to each party, K$	120	40										
Total annual savings, K$	30											
Pct savings to each party	60%	40%	**61%**	**39%**	62%	38%	63%	37%	64%	36%	(	
Savings to each party, K$	18	12	18	12	19	11	19	11	19	11		

Cash Flow, Iteration 2, between 60% and 70% savings to landlord

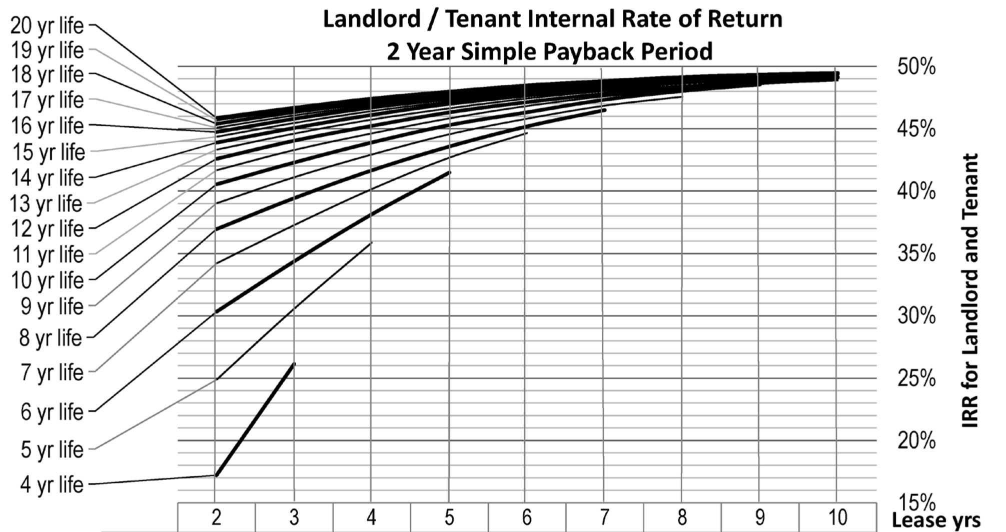
Savings Division	60% L	40% T	**61% L**	**39% T**	62% L	38% T	63% L	37% T	64% L
Project cost	(120)	(40)	(120)	(40)	(120)	(40)	(120)	(40)	(120
Year 1	18.0	12.0	18.3	11.7	18.6	11.4	18.9	11.1	19
2	18.0	12.0	18.3	11.7	18.6	11.4	18.9	11.1	1
3	18.0	12.0	18.3	11.7	18.6	11.4	18.9	11.1	
4	18.0	12.0	18.3	11.7	18.6	11.4	18.9	11.1	
5	18.0	12.0	18.3	11.7	18.6	11.4	18.9	11.1	
6	18.0	-	18.3	-	18.6	-	18.9	-	
7	18.0	-	18.3	-	18.6	-	18.9	-	
8	18.0	-	18.3	-	18.6	-	18.9		
9	18.0	-	18.3	-	18.6	-	18.9		
10	18.0	-	18.3	-	18.6	-	18.9		
11	18.0	-	18.3	-	18.6	-	18.9		
12	18.0	-	18.3	-	18.6	-	18.		
13	18.0	-	18.3	-	18.6	-	1		
14	18.0	-	18.3	-	18.6	-			
15	18.0	-	18.3	-	18.6	-			
16	18.0	-	18.3	-	18.6	-			
17	18.0	-	18.3	-	18.6	-			
18	18.0	-	18.3	-	18.6	-			
19	18.0	-	18.3	-	18.6				
20	18.0	-	18.3	-	18.6				

Figure 24-D1 (concluded)
Example Iterative Solution for Aligning Landlord and Tenant Internal Rate of Return (IRR) for Shared Cost and Savings



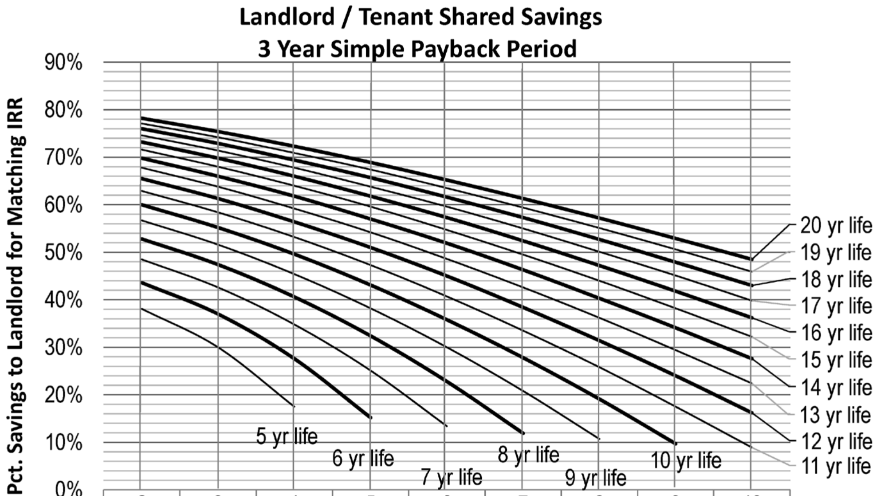
	2	3	4	5	6	7	8	9	10
20 yr life	82.6%	79.4%	75.9%	72.0%	67.9%	63.5%	59.0%	54.3%	49.5%
19 yr life	81.8%	78.4%	74.7%	70.6%	66.2%	61.6%	56.8%	51.9%	46.9%
18 yr life	80.8%	77.3%	73.3%	69.0%	64.4%	59.5%	54.5%	49.3%	44.0%
17 yr life	79.7%	76.0%	71.8%	67.3%	62.4%	57.2%	51.9%	46.3%	40.7%
16 yr life	78.5%	74.6%	70.2%	65.3%	60.1%	54.6%	48.9%	43.1%	37.1%
15 yr life	77.2%	73.0%	68.3%	63.1%	57.6%	51.7%	45.6%	39.3%	32.9%
14 yr life	75.7%	71.2%	66.2%	60.7%	54.7%	48.4%	41.9%	35.1%	28.2%
13 yr life	74.0%	69.2%	63.8%	57.8%	51.4%	44.6%	37.5%	30.2%	22.8%
12 yr life	72.0%	66.9%	61.0%	54.6%	47.6%	40.2%	32.5%	24.6%	16.5%
11 yr life	69.7%	64.2%	57.8%	50.8%	43.2%	35.1%	26.6%	17.9%	9.0%
10 yr life	67.1%	61.0%	54.1%	46.4%	37.9%	29.0%	19.6%	9.9%	
9 yr life	64.0%	57.4%	49.7%	41.1%	31.6%	21.5%	10.9%		
8 yr life	60.3%	53.0%	44.4%	34.6%	23.8%	12.2%			
7 yr life	56.0%	47.8%	38.0%	26.6%	13.8%				
6 yr life	50.8%	41.4%	29.7%	15.8%					
5 yr life	44.5%	33.2%	18.3%						
4 yr life	36.6%	21.6%							

Figure 24-D2 for 2 Year Overall Simple Payback Period: Savings Division
See notes for **24-D2** chart series



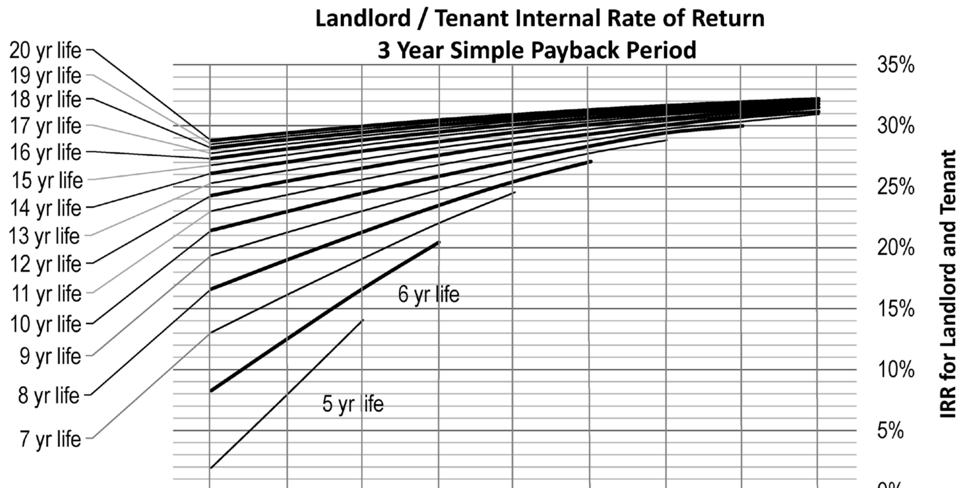
	2	3	4	5	6	7	8	9	10
20 yr life	45.9%	46.7%	47.4%	48.0%	48.5%	48.8%	49.2%	49.3%	49.5%
19 yr life	45.7%	46.5%	47.3%	47.9%	48.3%	48.7%	49.0%	49.3%	49.5%
18 yr life	45.4%	46.3%	47.1%	47.7%	48.3%	48.6%	49.0%	49.3%	49.5%
17 yr life	45.1%	46.1%	46.9%	47.6%	48.2%	48.6%	49.0%	49.1%	49.4%
16 yr life	44.7%	45.8%	46.7%	47.4%	48.0%	48.4%	48.8%	49.2%	49.4%
15 yr life	44.4%	45.5%	46.4%	47.2%	47.9%	48.3%	48.7%	49.0%	49.2%
14 yr life	43.9%	45.1%	46.1%	47.0%	47.7%	48.2%	48.7%	49.0%	49.2%
13 yr life	43.3%	44.6%	45.7%	46.6%	47.4%	48.0%	48.5%	48.8%	49.1%
12 yr life	42.6%	44.0%	45.2%	46.3%	47.1%	47.8%	48.3%	48.8%	49.1%
11 yr life	41.7%	43.3%	44.6%	45.8%	46.8%	47.6%	48.1%	48.6%	48.9%
10 yr life	40.5%	42.3%	43.9%	45.3%	46.3%	47.3%	48.0%	48.6%	
9 yr life	39.0%	41.1%	42.9%	44.6%	45.8%	46.9%	47.6%		
8 yr life	37.0%	39.4%	41.7%	43.6%	45.2%	46.5%			
7 yr life	34.2%	37.3%	40.2%	42.7%	44.7%				
6 yr life	30.3%	34.4%	38.1%	41.5%					
5 yr life	24.9%	30.6%	35.9%						
4 yr life	17.2%	26.1%							

Figure 24-D2 for 2 Year Overall Simple Payback Period: Internal Rate of Return
See notes for **24-D2** chart series



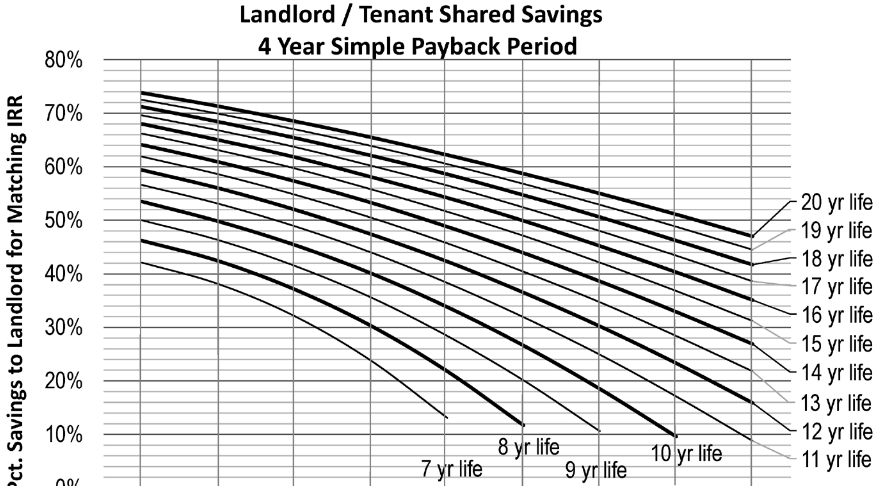
	2	3	4	5	6	7	8	9	10
20 yr life	78.2%	75.4%	72.3%	68.9%	65.2%	61.3%	57.2%	52.9%	48.5%
19 yr life	77.1%	74.2%	70.9%	67.4%	63.5%	59.4%	55.1%	50.6%	45.9%
18 yr life	76.0%	72.9%	69.4%	65.7%	61.6%	57.3%	52.7%	47.9%	43.0%
17 yr life	74.6%	71.4%	67.8%	63.8%	59.6%	55.0%	50.1%	45.1%	39.8%
16 yr life	73.2%	69.8%	66.0%	61.8%	57.3%	52.4%	47.2%	41.8%	36.2%
15 yr life	71.6%	68.0%	64.0%	59.6%	54.7%	49.5%	44.0%	38.2%	32.2%
14 yr life	69.8%	66.0%	61.8%	57.0%	51.9%	46.3%	40.3%	34.1%	27.6%
13 yr life	67.8%	63.8%	59.2%	54.2%	48.6%	42.6%	36.2%	29.4%	22.4%
12 yr life	65.5%	61.3%	56.4%	51.0%	45.0%	38.4%	31.4%	24.0%	16.2%
11 yr life	62.9%	58.4%	53.2%	47.3%	40.7%	33.5%	25.8%	17.5%	8.9%
10 yr life	60.0%	55.2%	49.6%	43.1%	35.8%	27.8%	19.1%	9.7%	
9 yr life	56.7%	51.6%	45.4%	38.2%	30.0%	20.8%	10.7%		
8 yr life	52.8%	47.4%	40.6%	32.4%	22.8%	11.9%			
7 yr life	48.5%	42.6%	34.8%	25.1%	13.4%				
6 yr life	43.6%	37.0%	27.6%	15.2%					
5 yr life	38.1%	30.0%	17.5%						

Figure 24-D2 for 3 Year Overall Simple Payback Period: Savings Division
See notes for **24-D2** chart series



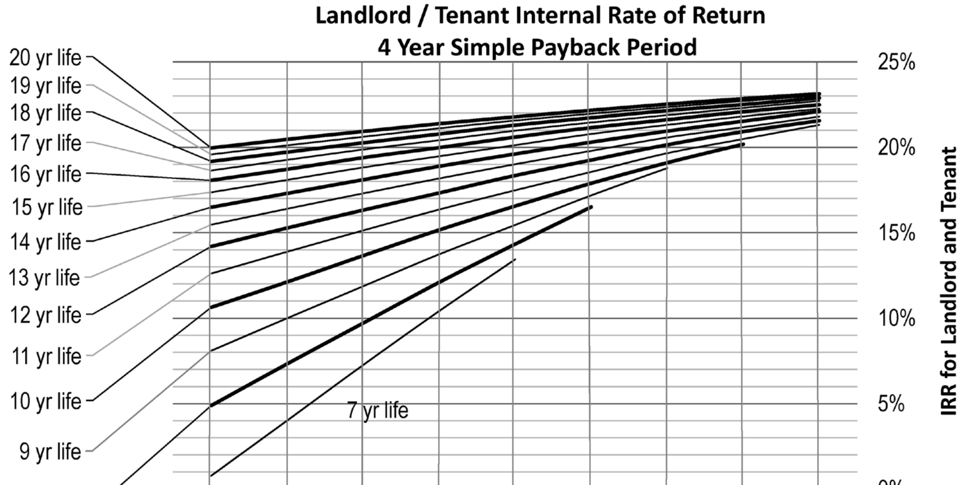
	2	3	4	5	6	7	8	9	10
20 yr life	28.8%	29.4%	30.0%	30.5%	30.9%	31.3%	31.6%	31.9%	32.2%
19 yr life	28.5%	29.1%	29.7%	30.3%	30.7%	31.2%	31.6%	31.9%	32.1%
18 yr life	28.2%	28.9%	29.5%	30.1%	30.5%	31.0%	31.4%	31.7%	32.0%
17 yr life	27.7%	28.5%	29.2%	29.8%	30.4%	30.8%	31.2%	31.6%	31.9%
16 yr life	27.3%	28.1%	28.8%	29.5%	30.1%	30.6%	31.1%	31.4%	31.8%
15 yr life	26.8%	27.6%	28.4%	29.2%	29.8%	30.4%	30.9%	31.3%	31.7%
14 yr life	26.1%	27.0%	27.9%	28.7%	29.5%	30.1%	30.6%	31.1%	31.5%
13 yr life	25.3%	26.3%	27.3%	28.2%	29.0%	29.7%	30.4%	30.9%	31.4%
12 yr life	24.3%	25.5%	26.5%	27.6%	28.5%	29.3%	30.1%	30.7%	31.1%
11 yr life	23.0%	24.3%	25.6%	26.8%	27.8%	28.8%	29.7%	30.3%	31.0%
10 yr life	21.4%	23.0%	24.5%	25.9%	27.1%	28.3%	29.4%	30.0%	
9 yr life	19.4%	21.2%	23.0%	24.7%	26.3%	27.8%	28.8%		
8 yr life	16.6%	19.0%	21.3%	23.5%	25.4%	27.1%			
7 yr life	13.0%	16.1%	19.1%	22.0%	24.5%				
6 yr life	8.3%	12.5%	16.6%	20.4%					
5 yr life	1.9%	7.9%	14.1%						

Figure 24-D2 for 3 Year Overall Simple Payback Period: Internal Rate of Return
See notes for **24-D2** chart series



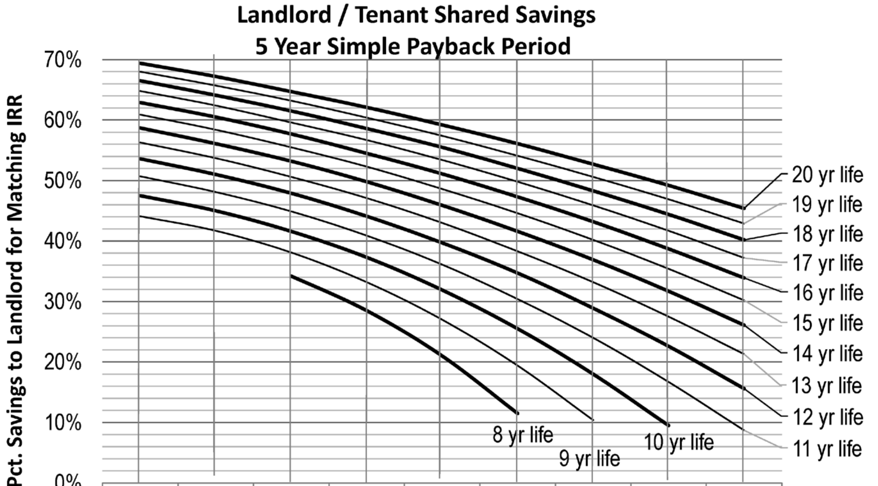
	2	3	4	5	6	7	8	9	10	Lease yrs.
20 yr life	73.8%	71.3%	68.5%	65.5%	62.2%	58.7%	55.0%	51.1%	47.0%	
19 yr life	72.5%	69.9%	67.0%	63.9%	60.5%	56.8%	52.9%	48.8%	44.5%	
18 yr life	71.2%	68.4%	65.4%	62.1%	58.6%	54.7%	50.6%	46.2%	41.7%	
17 yr life	69.6%	66.8%	63.7%	60.2%	56.5%	52.4%	48.0%	43.4%	38.6%	
16 yr life	68.0%	65.0%	61.8%	58.1%	54.2%	49.9%	45.2%	40.3%	35.1%	
15 yr life	66.2%	63.1%	59.7%	55.8%	51.6%	47.1%	42.1%	36.8%	31.2%	
14 yr life	64.1%	60.9%	57.3%	53.3%	48.8%	43.9%	38.6%	32.9%	26.9%	
13 yr life	61.9%	58.6%	54.8%	50.5%	45.7%	40.4%	34.7%	28.4%	21.8%	
12 yr life	59.4%	56.0%	52.0%	47.4%	42.3%	36.5%	30.2%	23.3%	15.9%	
11 yr life	56.6%	53.1%	48.9%	44.0%	38.3%	31.9%	24.9%	17.1%	8.8%	
10 yr life	53.5%	49.8%	45.4%	40.1%	33.8%	26.6%	18.5%	9.6%		
9 yr life	50.0%	46.3%	41.5%	35.6%	28.4%	20.1%	10.6%			
8 yr life	46.2%	42.4%	37.1%	30.3%	21.8%	11.7%				
7 yr life	42.1%	38.1%	32.1%	23.8%	13.1%					

Figure 24-D2 for 4 Year Overall Simple Payback Period: Savings Division
See notes for 24-D2 chart series



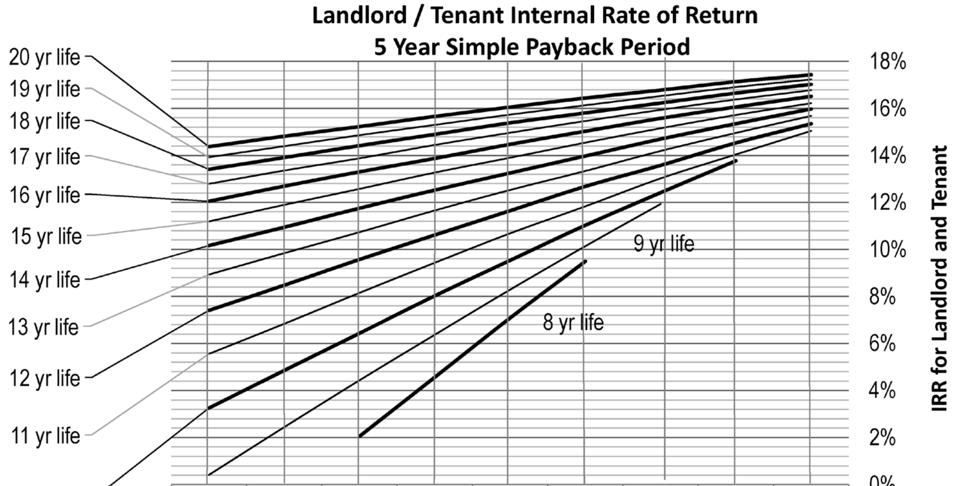
	2	3	4	5	6	7	8	9	10
20 yr life	20.0%	20.5%	20.9%	21.4%	21.8%	22.2%	22.5%	22.8%	23.1%
19 yr life	19.6%	20.1%	20.6%	21.1%	21.6%	22.0%	22.3%	22.7%	23.0%
18 yr life	19.2%	19.7%	20.3%	20.8%	21.3%	21.7%	22.1%	22.5%	22.9%
17 yr life	18.6%	19.3%	19.9%	20.4%	21.0%	21.5%	21.9%	22.3%	22.7%
16 yr life	18.1%	18.7%	19.4%	20.0%	20.6%	21.1%	21.6%	22.1%	22.5%
15 yr life	17.4%	18.1%	18.8%	19.5%	20.1%	20.8%	21.3%	21.8%	22.3%
14 yr life	16.5%	17.3%	18.1%	18.9%	19.6%	20.3%	20.9%	21.5%	22.1%
13 yr life	15.5%	16.4%	17.3%	18.2%	19.0%	19.8%	20.6%	21.2%	21.8%
12 yr life	14.2%	15.3%	16.3%	17.3%	18.3%	19.3%	20.1%	20.9%	21.6%
11 yr life	12.6%	13.9%	15.1%	16.4%	17.5%	18.6%	19.7%	20.5%	21.3%
10 yr life	10.6%	12.1%	13.7%	15.2%	16.6%	17.9%	19.1%	20.2%	
9 yr life	8.1%	10.0%	11.9%	13.7%	15.5%	17.2%	18.8%		
8 yr life	4.9%	7.3%	9.7%	12.1%	14.3%	16.5%			
7 yr life	0.8%	4.0%	7.3%	10.4%	13.4%				

Figure 24-D2 for 4 Year Overall Simple Payback Period: Internal Rate of Return
See notes for **24-D2** chart series



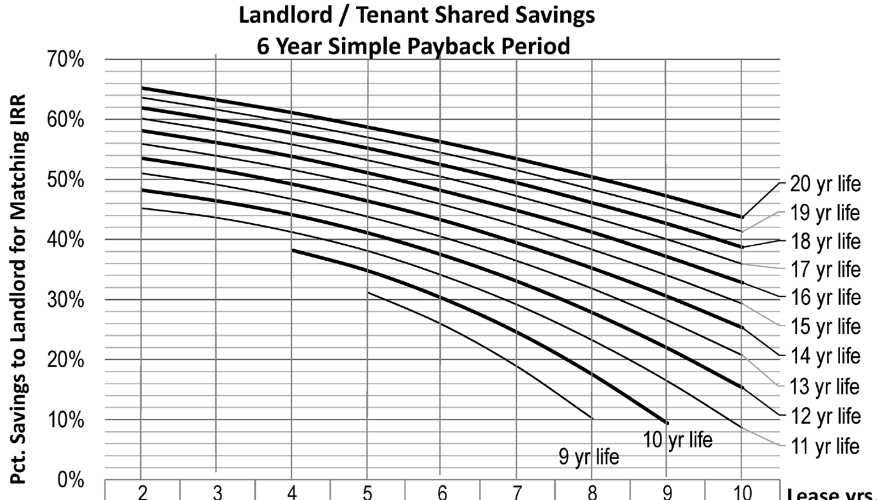
Lease yrs.	2	3	4	5	6	7	8	9	10
20 yr life	69.4%	67.2%	64.7%	62.1%	59.2%	56.1%	52.7%	49.2%	45.4%
19 yr life	68.0%	65.7%	63.2%	60.4%	57.4%	54.1%	50.6%	46.9%	42.9%
18 yr life	66.5%	64.1%	61.5%	58.6%	55.5%	52.0%	48.3%	44.4%	40.2%
17 yr life	64.8%	62.4%	59.6%	56.7%	53.4%	49.8%	45.9%	41.7%	37.2%
16 yr life	62.9%	60.5%	57.7%	54.5%	51.1%	47.3%	43.2%	38.7%	33.9%
15 yr life	60.9%	58.4%	55.5%	52.3%	48.6%	44.6%	40.2%	35.4%	30.2%
14 yr life	58.7%	56.1%	53.2%	49.8%	45.9%	41.6%	36.9%	31.7%	26.1%
13 yr life	56.3%	53.7%	50.6%	47.1%	43.0%	38.3%	33.2%	27.5%	21.3%
12 yr life	53.6%	51.0%	47.9%	44.1%	39.7%	34.7%	28.9%	22.6%	15.6%
11 yr life	50.7%	48.1%	44.9%	40.9%	36.1%	30.4%	24.0%	16.7%	8.7%
10 yr life	47.5%	45.0%	41.6%	37.3%	31.9%	25.5%	18.0%	9.5%	
9 yr life	44.1%	41.7%	38.1%	33.2%	27.0%	19.4%	10.4%		
8 yr life			34.2%	28.5%	21.0%	11.5%			

Figure 24-D2 for 5 Year Overall Simple Payback Period: Savings Division
See notes for **24-D2** chart series



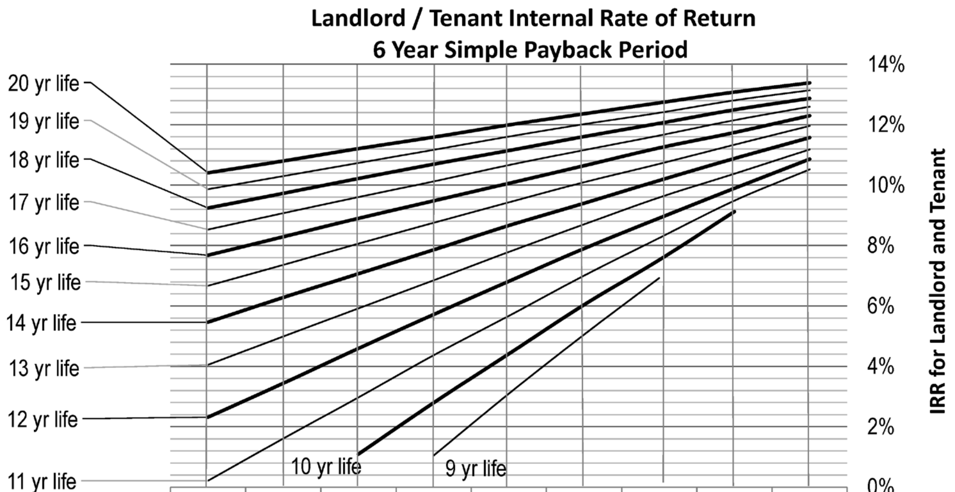
	2	3	4	5	6	7	8	9	10
20 yr life	14.4%	14.8%	15.2%	15.7%	16.1%	16.4%	16.8%	17.1%	17.4%
19 yr life	13.9%	14.4%	14.9%	15.3%	15.7%	16.1%	16.5%	16.9%	17.2%
18 yr life	13.4%	13.9%	14.4%	14.9%	15.4%	15.8%	16.2%	16.6%	17.0%
17 yr life	12.8%	13.4%	13.9%	14.4%	15.0%	15.5%	15.9%	16.4%	16.8%
16 yr life	12.0%	12.7%	13.3%	13.9%	14.5%	15.0%	15.6%	16.1%	16.5%
15 yr life	11.2%	11.9%	12.6%	13.3%	13.9%	14.5%	15.2%	15.7%	16.2%
14 yr life	10.2%	10.9%	11.8%	12.5%	13.3%	14.0%	14.7%	15.3%	16.0%
13 yr life	8.9%	9.8%	10.7%	11.7%	12.5%	13.3%	14.2%	15.0%	15.7%
12 yr life	7.4%	8.5%	9.6%	10.6%	11.6%	12.7%	13.6%	14.5%	15.3%
11 yr life	5.6%	6.8%	8.2%	9.4%	10.7%	11.8%	13.0%	14.0%	15.0%
10 yr life	3.3%	4.9%	6.4%	8.0%	9.5%	11.0%	12.4%	13.8%	
9 yr life	0.4%	2.4%	4.4%	6.4%	8.3%	10.1%	11.9%		
8 yr life			2.0%	4.6%	7.1%	9.5%			

Figure 24-D2 for 5 Year Overall Simple Payback Period: Internal Rate of Return
See notes for **24-D2** chart series



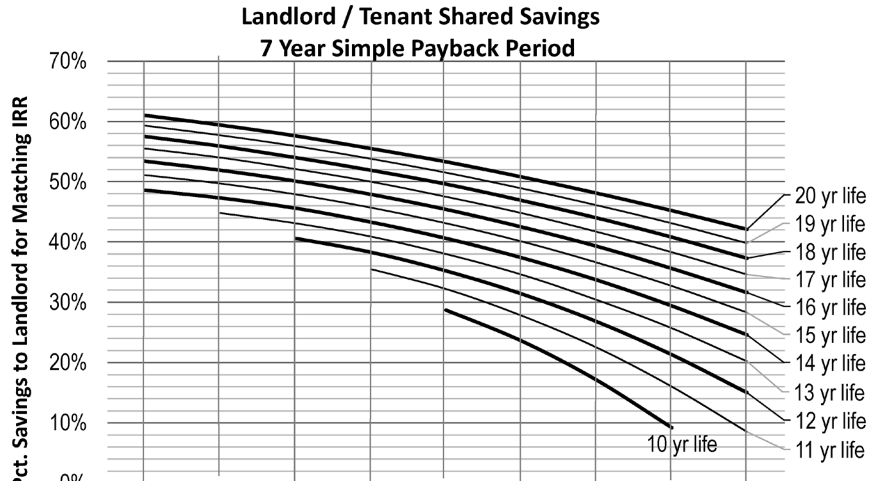
	2	3	4	5	6	7	8	9	10
20 yr life	65.2%	63.2%	61.1%	58.7%	56.2%	53.4%	50.4%	47.2%	43.7%
19 yr life	63.6%	61.6%	59.4%	57.0%	54.4%	51.5%	48.3%	45.0%	41.3%
18 yr life	61.9%	59.9%	57.7%	55.2%	52.4%	49.4%	46.1%	42.6%	38.7%
17 yr life	60.1%	58.1%	55.8%	53.2%	50.4%	47.2%	43.7%	40.0%	35.9%
16 yr life	58.1%	56.1%	53.8%	51.1%	48.1%	44.8%	41.2%	37.1%	32.8%
15 yr life	55.9%	53.9%	51.6%	48.9%	45.8%	42.3%	38.3%	34.0%	29.3%
14 yr life	53.5%	51.6%	49.2%	46.4%	43.2%	39.4%	35.2%	30.5%	25.3%
13 yr life	51.0%	49.1%	46.7%	43.8%	40.4%	36.4%	31.8%	26.5%	20.7%
12 yr life	48.2%	46.4%	44.1%	41.1%	37.4%	33.0%	27.8%	21.9%	15.3%
11 yr life	45.2%	43.6%	41.2%	38.1%	34.0%	29.1%	23.2%	16.4%	8.6%
10 yr life			38.2%	34.8%	30.2%	24.5%	17.5%	9.4%	
9 yr life				31.2%	25.8%	18.8%	10.2%		

Figure 24-D2 for 6 Year Overall Simple Payback Period: Savings Division
See notes for **24-D2** chart series



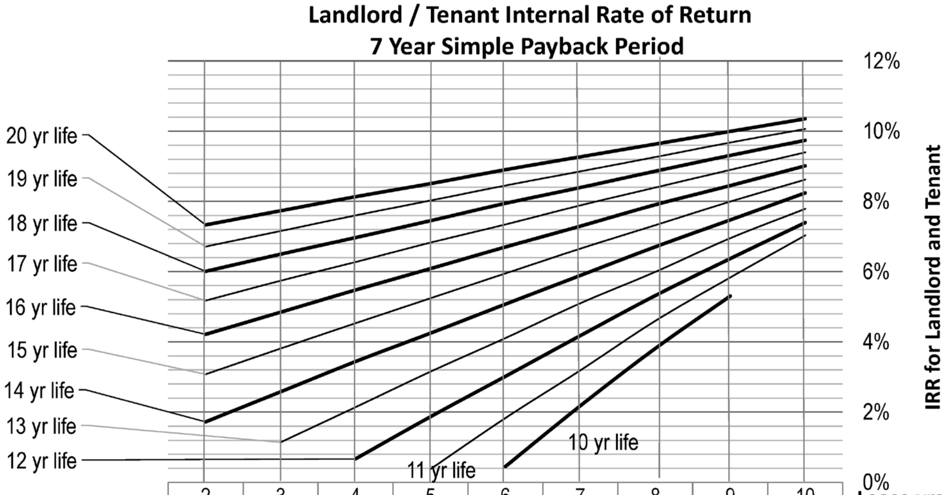
	2	3	4	5	6	7	8	9	10
20 yr life	10.4%	10.8%	11.2%	11.6%	12.0%	12.4%	12.7%	13.1%	13.4%
19 yr life	9.9%	10.3%	10.7%	11.2%	11.6%	12.0%	12.4%	12.8%	13.1%
18 yr life	9.2%	9.7%	10.2%	10.7%	11.1%	11.6%	12.0%	12.5%	12.9%
17 yr life	8.5%	9.1%	9.6%	10.1%	10.7%	11.2%	11.6%	12.1%	12.6%
16 yr life	7.7%	8.3%	8.9%	9.5%	10.1%	10.6%	11.2%	11.7%	12.3%
15 yr life	6.7%	7.4%	8.1%	8.8%	9.4%	10.1%	10.7%	11.3%	12.0%
14 yr life	5.5%	6.3%	7.1%	7.9%	8.7%	9.4%	10.2%	10.9%	11.6%
13 yr life	4.0%	5.0%	5.9%	6.8%	7.8%	8.7%	9.6%	10.4%	11.2%
12 yr life	2.3%	3.4%	4.6%	5.7%	6.8%	7.9%	8.9%	9.9%	10.9%
11 yr life	0.2%	1.6%	3.0%	4.4%	5.7%	7.0%	8.2%	9.5%	10.5%
10 yr life			1.1%	2.8%	4.4%	6.0%	7.5%	9.1%	
9 yr life				1.0%	3.1%	5.0%	6.9%		

Figure 24-D2 for 6 Year Overall Simple Payback Period: Internal Rate of Return
See notes for **24-D2** chart series



Lease yrs.	2	3	4	5	6	7	8	9	10
20 yr life	61.0%	59.4%	57.6%	55.5%	53.3%	50.8%	48.1%	45.2%	42.1%
19 yr life	59.3%	57.7%	55.9%	53.8%	51.5%	48.9%	46.1%	43.1%	39.8%
18 yr life	57.5%	55.9%	54.0%	51.9%	49.6%	46.9%	44.0%	40.8%	37.3%
17 yr life	55.5%	54.0%	52.1%	50.0%	47.5%	44.8%	41.7%	38.3%	34.6%
16 yr life	53.4%	51.9%	50.1%	47.9%	45.4%	42.5%	39.3%	35.6%	31.6%
15 yr life	51.1%	49.7%	47.9%	45.7%	43.1%	40.1%	36.6%	32.7%	28.3%
14 yr life	48.6%	47.3%	45.6%	43.3%	40.6%	37.4%	33.7%	29.4%	24.6%
13 yr life		44.8%	43.1%	40.9%	38.0%	34.6%	30.4%	25.7%	20.2%
12 yr life			40.6%	38.3%	35.2%	31.4%	26.8%	21.3%	15.0%
11 yr life				35.5%	32.2%	27.8%	22.5%	16.0%	8.5%
10 yr life					28.7%	23.6%	17.1%	9.2%	

Figure 24-D2 for 7 Year Overall Simple Payback Period: Savings Division
See notes for **24-D2** chart series



	2	3	4	5	6	7	8	9	10
20 yr life	7.3%	7.7%	8.1%	8.5%	8.9%	9.3%	9.6%	10.0%	10.4%
19 yr life	6.7%	7.2%	7.6%	8.0%	8.5%	8.9%	9.3%	9.7%	10.1%
18 yr life	6.0%	6.5%	7.0%	7.4%	7.9%	8.4%	8.9%	9.3%	9.7%
17 yr life	5.2%	5.7%	6.3%	6.8%	7.3%	7.9%	8.4%	8.9%	9.4%
16 yr life	4.2%	4.8%	5.5%	6.1%	6.7%	7.3%	7.9%	8.4%	9.0%
15 yr life	3.1%	3.8%	4.5%	5.2%	5.9%	6.7%	7.3%	8.0%	8.6%
14 yr life	1.7%	2.6%	3.4%	4.2%	5.1%	5.9%	6.7%	7.5%	8.2%
13 yr life		1.1%	2.1%	3.2%	4.1%	5.1%	6.0%	7.0%	7.8%
12 yr life			0.7%	1.9%	3.0%	4.2%	5.3%	6.4%	7.4%
11 yr life				0.4%	1.8%	3.2%	4.6%	5.8%	7.0%
10 yr life					0.5%	2.2%	3.8%	5.3%	

Figure 24-D2 for 7 Year Overall Simple Payback Period: Internal Rate of Return
See notes for **24-D2** chart series

E—COORDINATING UPSTREAM/DOWNSTREAM SETPOINTS

- Larger HVAC systems that include central station equipment and terminal equipment are very often not coordinated in their control. The central station has its control agenda, and the terminal units have a separate one (usually final control). Coordinating the two is an opportunity for savings in many, many systems. The advanced routines that include polling do this automatically.
- This concept applies to any system that includes a main "upstream" air handler with associated "downstream" terminal units with heating capability. The concept is simple enough, but will require some programming. The objective is to provide enough, but just enough, cooling and heating from the upstream device and to avoid heating and cooling fighting between upstream and downstream components. Typical control system design does not include interaction between air handler and terminal unit and each behaves as if the other does not exist. Because of this, heat/cool overlap can easily occur and not be detected. The same concept applies to fan pressure when variable fan capacity is used (VAV).
- ***Examples***:
  1. Resetting AHU supply temperature upwards until at least one terminal unit heating coil is at least 95% closed.
  2. Resetting AHU fan static pressure downwards until at least one VAV box damper is at least 95% open.
  3. Resetting boiler hot water supply temperature downward until at least one heating control valve is 95% open.
  4. Resetting chilled water supply temperature upward until at least one chilled water control valve is 95% open.
  5. etc.

  The polling routine works with "most open" or "most closed" signals, but is normally referred to as the "most open" valve/damper routine. The point of use feedback can be in the form of actual position or control signal. Tuning is necessary to prevent or reduce hunting.
- The benefit of these measures can sometimes be observed with two control system screens open side by side (example here is for an air handler): one for the main air handler and another for a terminal unit when several are in heating mode. If the terminal unit is operating in heating mode and the supply air is relatively low (less than 60 degF), slowly raise the supply air temperature, 2 degF at a time, and wait several minutes per step—you may see the reheat coil stop heating, which demonstrates that they were fighting each other, silently consuming energy. The effort taken to coordinate the entire system is

really just doing what logic suggests and correcting measures taken out of convenience.

- Before and After examples are shown in **Figure 24-E1**.

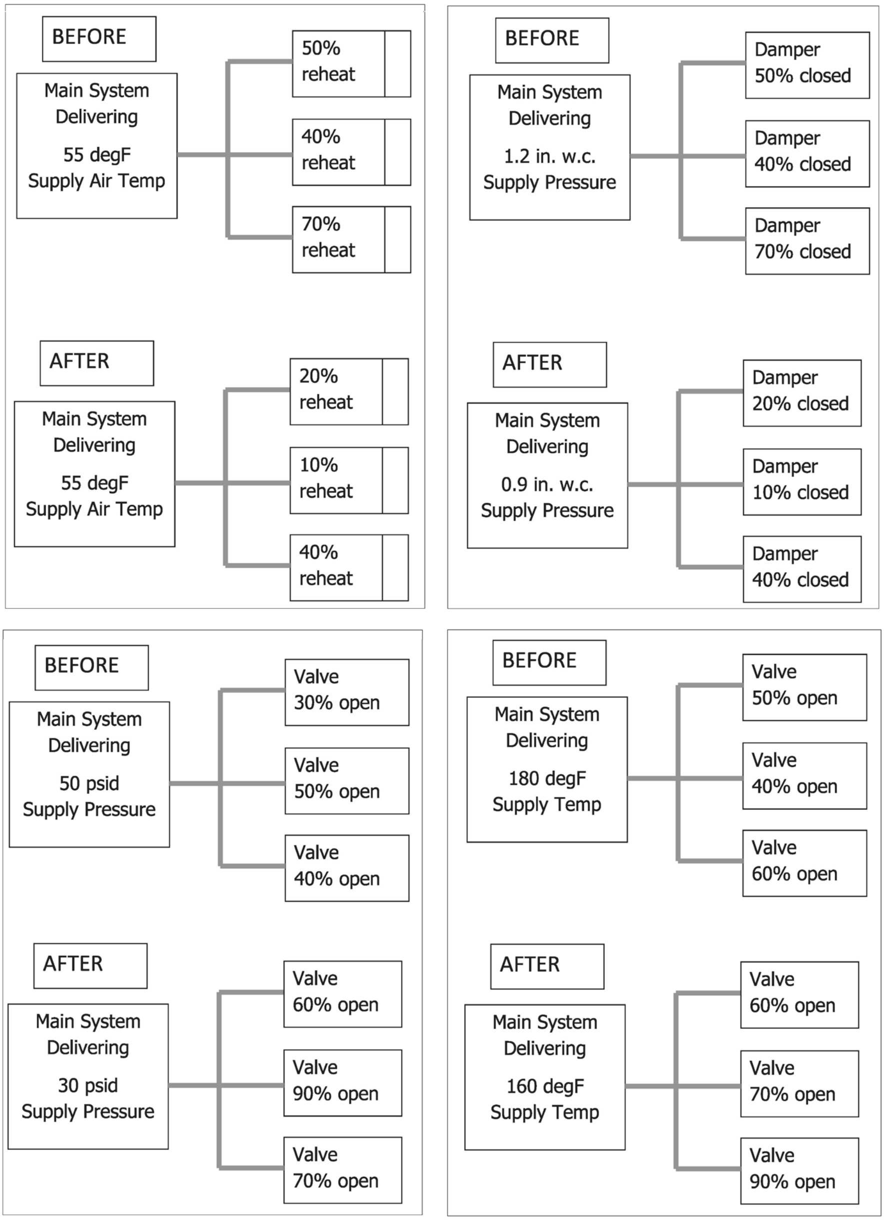


Figure 24-E1. Example Applications of Optimizing Upstream/Downstream Settings

Example benefit from coordinating upstream/ downstream settings between main air handler and terminal units. In this example the air handler has a fixed set point of 55 degF air year round. The example diagram shows the economizer providing the cooling, however whether free cooling or mechanical cooling, the heating cost is not free.

Same inlet temperature, same outlet temperature. The waste from heat/cool systems fighting is silent, but costly.

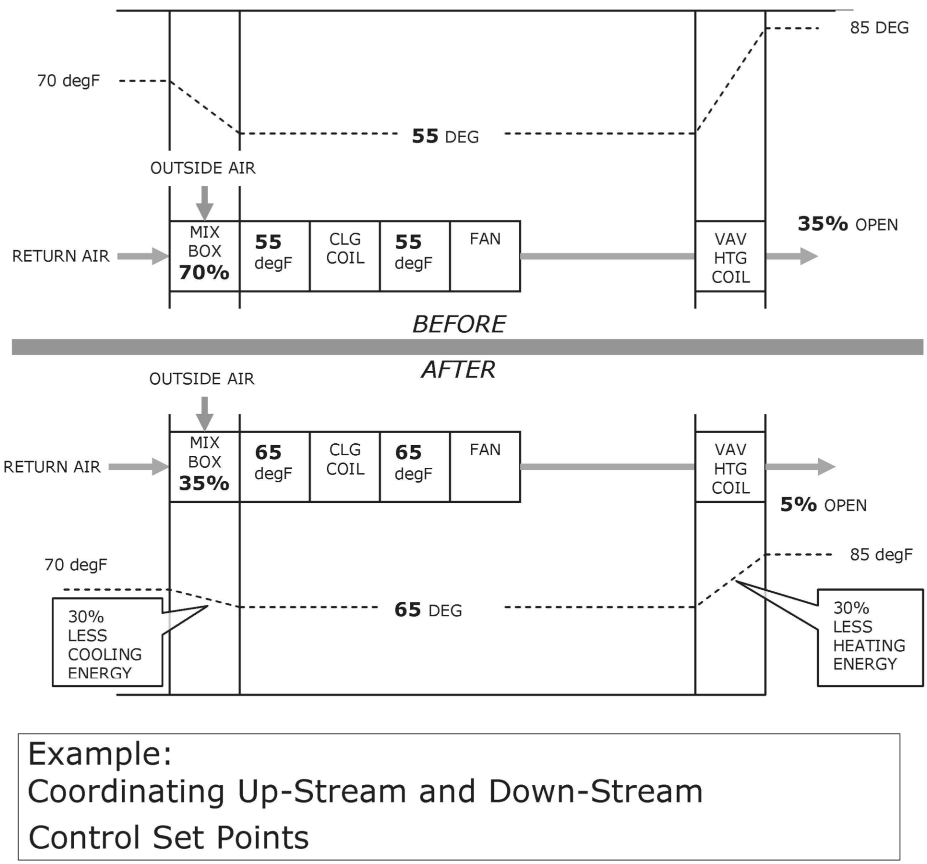


Figure 24-E2. Air Handler Example

F—SEMICONDUCTOR FAB MULTI-STAGE HVAC AIR TEMPERING

High exhaust rates coupled with sequential heating, cooling, humidifying, dehumidifying, dew point control, and final tempering operations create very large HVAC loads. Such operations are common in semi-conductor fabrication facilities, printing operations, etc.

The HVAC energy use is at least as much as the machine drive (tool) energy use and, incorporating process heating and cooling, can represent half of the total energy input into the product.

Breakdown of Energy Use for Computer and Electronic Product Manufacturing (NAICS 334)

Source: E-Source, Boulder Colorado. Original source of data: U.S. EIA

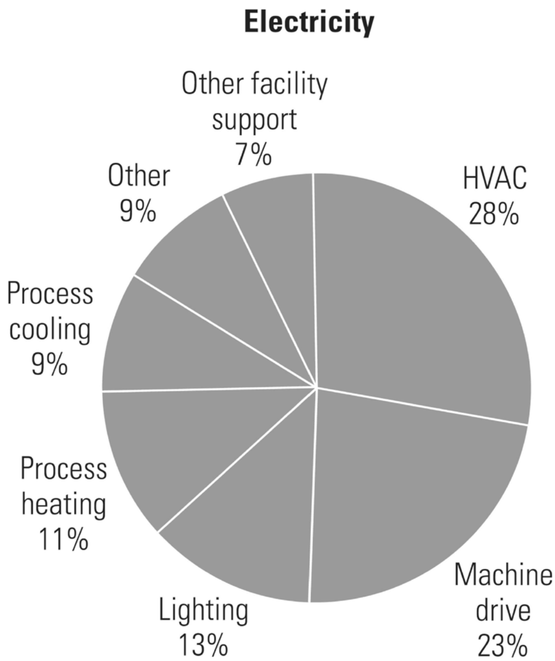


Figure 24-F1

General Suggestions:

- Closely evaluate each step on a psychrometric chart for opportunities.
- Determine necessary levels of cooling, heating, moisture, and compare to actual operations. Where actual operations go beyond the theoretical levels, ask why and identify the cost implications of doing so. Significant overlap beyond what is theoretically required is very common.

- If the final process is self-heating (semi-conductor fabrication), and upstream supply air is being heated, adjust supply air temperature to reduce load on the final cooling step.
- If dew point control is being used, examine the instrumentation and calibration frequency. Most dew point and relative humidity instruments have a sizeable +/- tolerance, drift over time, and should be calibrated frequently to avoid errors that translate into control overlapping costs. For ***example***, a 5% error in relative humidity at can translate into a 5 degF error in dew point temperature calculation at 80degF and a 10 degF error 120 degF which, in turn, drives heating and cooling equipment an additional 5 degrees for naught.
- Where "anticipation" is used for weather changes in large outside air processing operations, evaluate these settings closely since they are often overly conservative at the expense of energy input from un-necessary simultaneous heating and cooling.

Example Approach

Facility manufacturers semiconductors and has established air flow rates, temperature and humidity levels.

Step 1: Verify environmental control parameters.

These all go under the heading of "begin by using less" and will lower the bar of energy use before any other modifications are considered. These are sensitive subjects for manufacturing control, but worth asking since they drive the mechanical system energy use. Also, it is possible that the manufacturer has changed product lines or inherited a building from another manufacturer and requirements have become less strict.

See **Table 24-F1** for common questions to ask with the intent of using less to begin with.

Step 2: Primary Cooling, Heating, and Air-Water Transport Efficiencies
These approaches are straight forward and discussed in earlier chapters.

Step 3: Control Optimization for the 100% Outside Air Systems—Dehumidification Control

The combination of large air flows and close tolerances in the clean rooms create operational realities that lead to energy waste. A warm sunny afternoon at 50% rH followed by a cloudburst is like a pig moving through a python for a 100% outside air unit. If the dew point control is

Table 24-F1.

	Basis of Savings
Can air changes be reduced?	Reduced exhaust and make up air quantities Reduced fan horsepower Reduced filtration cost
Can humidity levels be reduced? (dry climate)	Reduced humidifier operating expense Reduced pre-heating and re-heating expense for adiabatic humidification
Can humidity levels be increased?	Reduced dehumidification (cooling) load (humid climate) Reduced reheat load
Can temperature and humidity level tolerances be widened?	Reduced heat/cool overlap left in place continuously to handle intermittent outdoor moisture step changes (e.g. cloudburst)
Can the allowable rate of change of clean room temperature and humidity be increased?	This is discussed in more detail below

lost at the supply air discharge point, the humidity level in the clean room will rise, and may rise quickly, risking product quality problems. Even when a control valve opens fully with no delay, the large coils do not reach temperature instantaneously and the moisture slips by. The air handlers process air on a real time basis and are asked to behave instantaneously which they cannot. To overcome the equipment limitation, operators will deliberately establish overlapping heating and cooling so the coil is always cold enough to handle the cloudburst, reheating for all other hours. Hence, there is considerable waste from false loading. For example, if there are 500 hours per year of rapidly increasing outside air dew point and 4 months of winter, the system may operate inefficiently for 5000 hours per year.

The following is an example evaluation. Conditions evaluated are:

- Warm, moist day
- Warm, dry day

- Cool, dry day
- Cool, moist day

The settings and air flow for each fab is different, and the hours of each point are different. What is consistent is the nasty doubling effect. Note: in climates where water economizers are used, winter impact will be reduced.

Savings Potential

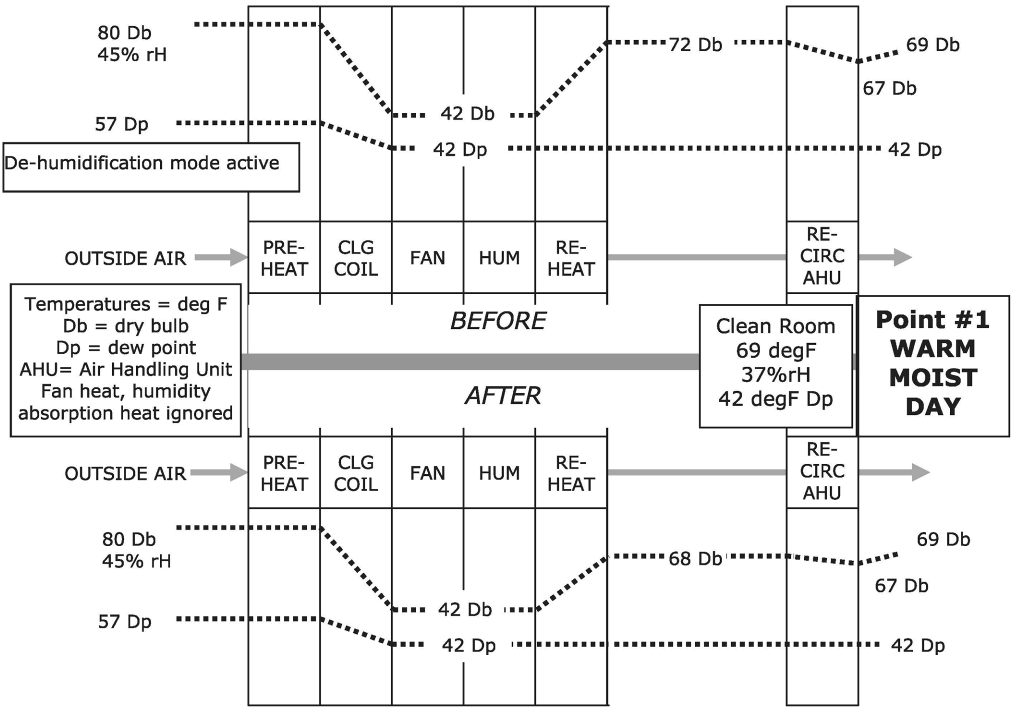
Table 24-F2.
Order of Magnitude Hourly Savings for
Semiconductor Fab HVAC Optimization for four Day Types Studied

Total energy is for heating and cooling only, and does not include fans, pumps, or humidification. Values are outputs; input will vary by heating and cooling equipment efficiency. Descriptions of each Day Type and basis of savings follow. Since each location varies, these values cannot be used directly. The message is that there are meaningful savings available from optimization in semiconductor fabrication plants.

Point		Total Energy (MBh)	Energy Reduction from Optimization	Pct Reduced
1	Warm, Moist Day	12,564	864	7%
2	Warm, Dry Day	9,855	2,549	26%
3	Cool, Dry Day	6,264	1,080	17%
4	Cool, Moist Day	8,856	3,672	41%

Assumptions:
- Clean room final conditions 69 degF, 37% rH, 42 degF dew point
- 100% outside air, 100,000 CFM
- Preheat set point 55 degF
- Cooling coil set point 55 degF (non-dehumidification mode)
- Cooling coil set point 42 degF (dehumidification mode)
- Final supply air temperature set point for make up air units 72 degF
- Supply air temperature set point for re-circulating air handlers (over clean rooms) 67 degF
- Dehumidification mode set point >29 degF dew point

Figure 24-F2. Warm Moist Day



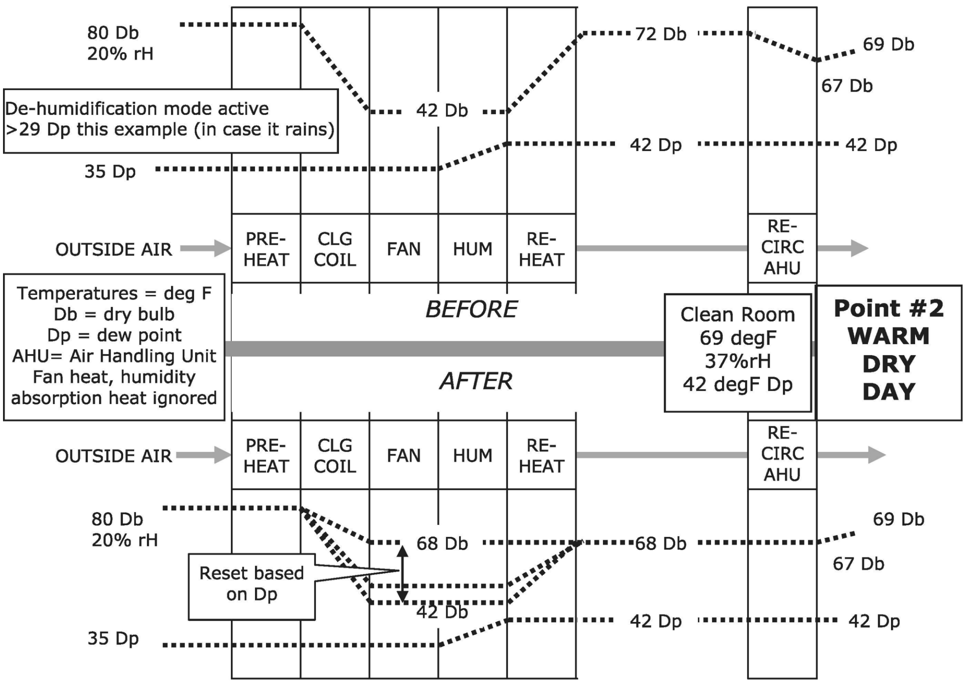
Lower supply air temperature to (1) degF above normal re-circ AHU supply temp. Reduces the need for re-circ cooling.
Avoids 4 degF over-heating and matching re-cooling

Savings **for 100,000 CFM for 4 degF of overlap:**
*Presumes the re-circulation fans move twice the total air as the make-up air units do.

Heating: 432 Mbh
Cooling: 432 MBh (36 tons)

*Heating savings: 1.08*CFM*(dT)/1,000 = MBh*
*Cooling savings: 1.08*CFM*(dT)/1,000 = MBh*

Figure 24-F3. Warm Dry Day



For 100,000 CFM:
Raise dehumidification mode limit to (5) degF below the clean room final condition. Watch approaching weather more closely. If rain is anticipated within the next 4-6 hours, push the system into dehumidification mode to protect product. Reset cooling coil leaving temperature from outside air dew point, with an override of 37 degF dry bulb and below requiring no dehumidification regardless of dew point.

Non-dehumidification mode cooling temperature is reset from dew point:

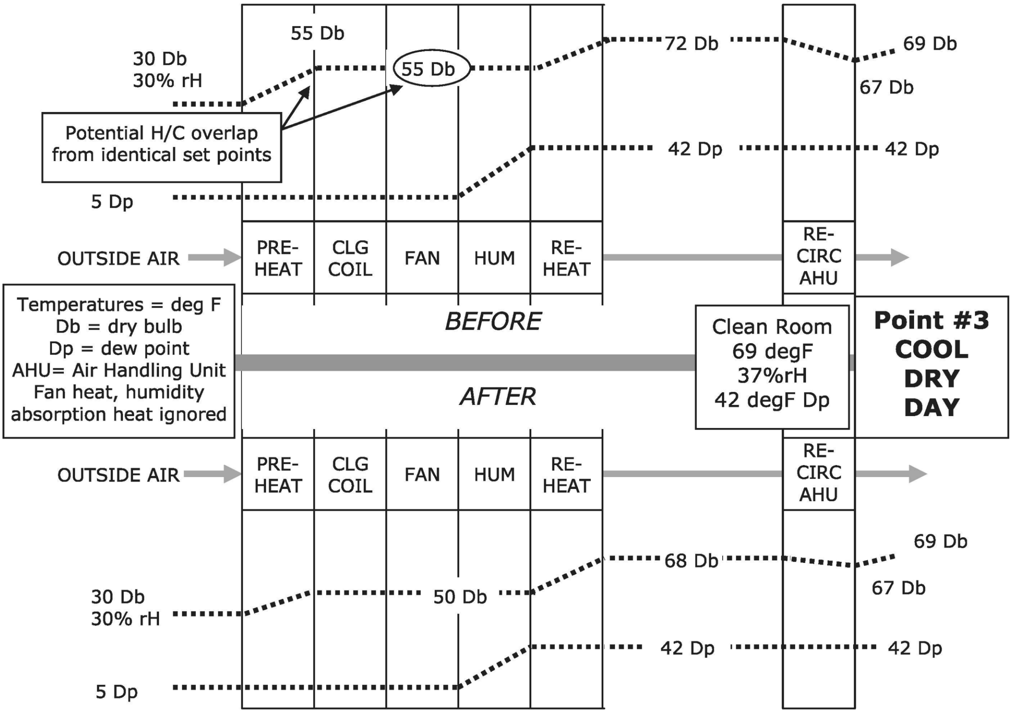
Outside Air Dew Point (degF)	Cooling Coil Leaving (degF)
De-humidification mode limit	De-humidification mode supply air temp
Nominal 15 deg below the upper limit	(1) deg above the re-circ AHU discharge air

Savings **for 100,000 CFM** from resetting supply air temperature **(5-13 degF overlap).**
Additional savings from 68 degF final temp instead of 72 not shown. See point #1.

Outside Air Dew Point (degF)	Cooling Coil Leaving Proposed (degF)	Cooling Coil Leaving Now (degF)	Cooling Saved	Heating Saved
37 Dp	**42 Db**	42 Db	0	0
34 Dp	47.2 Db	42 Db	562 Mbh (47T)	562 MBh
31 Dp	52.4 Db	42 Db	1123 Mbh (94T)	1123 Mbh
28 Dp	57.6 Db	55 Db	281 Mbh (23T)	281 MBh
25 Dp	62.8 Db	55 Db	842 Mbh (70T)	842 MBh
22 Dp	**68 Db**	55 Db	1404 MBh (117T)	1404 MBh

*Heating savings: 1.08*CFM*(dT)/1,000 = MBh*
*Cooling savings: 1.08*CFM*(dT)/1,000 = MBh*

Figure 24-F4. Cool Dry Day

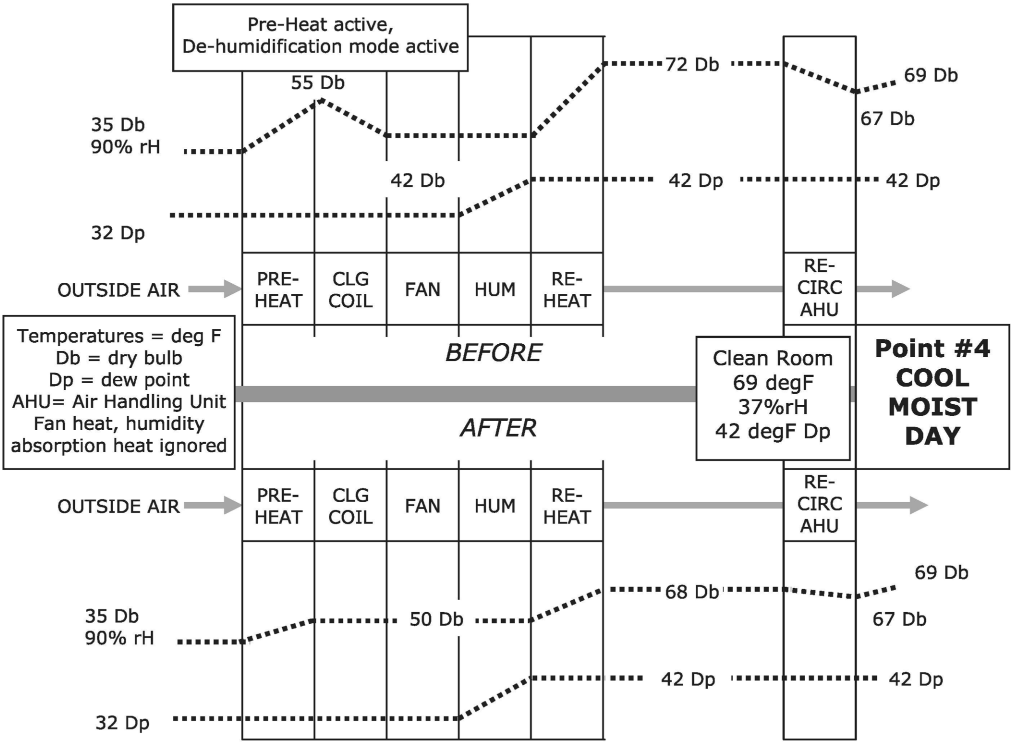


Lower preheat set point to 50 degF to provide dead band between it and cooling, to prevent overlap.
Additional savings from 68 degF final temp instead of 72 not shown. See point #1.

Savings **for 100,000 CFM with (1) degF of H/C overlap**:
Heating: 108 MBh
Cooling: 108 MBh (9 tons)

*Heating savings: 1.08*CFM*(dT)/1,000 = MBh*
*Cooling savings: 1.08*CFM*(dT)/1,000 = MBh*

Figure 24-F5. Cool Moist Day



Low temperature dehumidification override (point #2) corrects the conflict between preheat and dehumidification modes, made worse by the 55 degF preheat setting (point #3).
Additional savings from 68 degF final temp instead of 72 not shown. See point #1.

Savings **for 100,000 CFM for 13 degF overlap:**
Heating: 1404 MBh
Cooling: 1404 MBH (117 tons)

*Heating savings: 1.08*CFM*(dT)/1,000 = MBh*
*Cooling savings: 1.08*CFM*(dT)/1,000 = MBh*

G—HVAC RETROFITS FOR THE THREE WORST SYSTEMS

- Constant Volume Reheat
- Multi-Zone
- Double Duct

Figure 24-G1. Constant Volume Reheat Retrofit to VAV Reheat

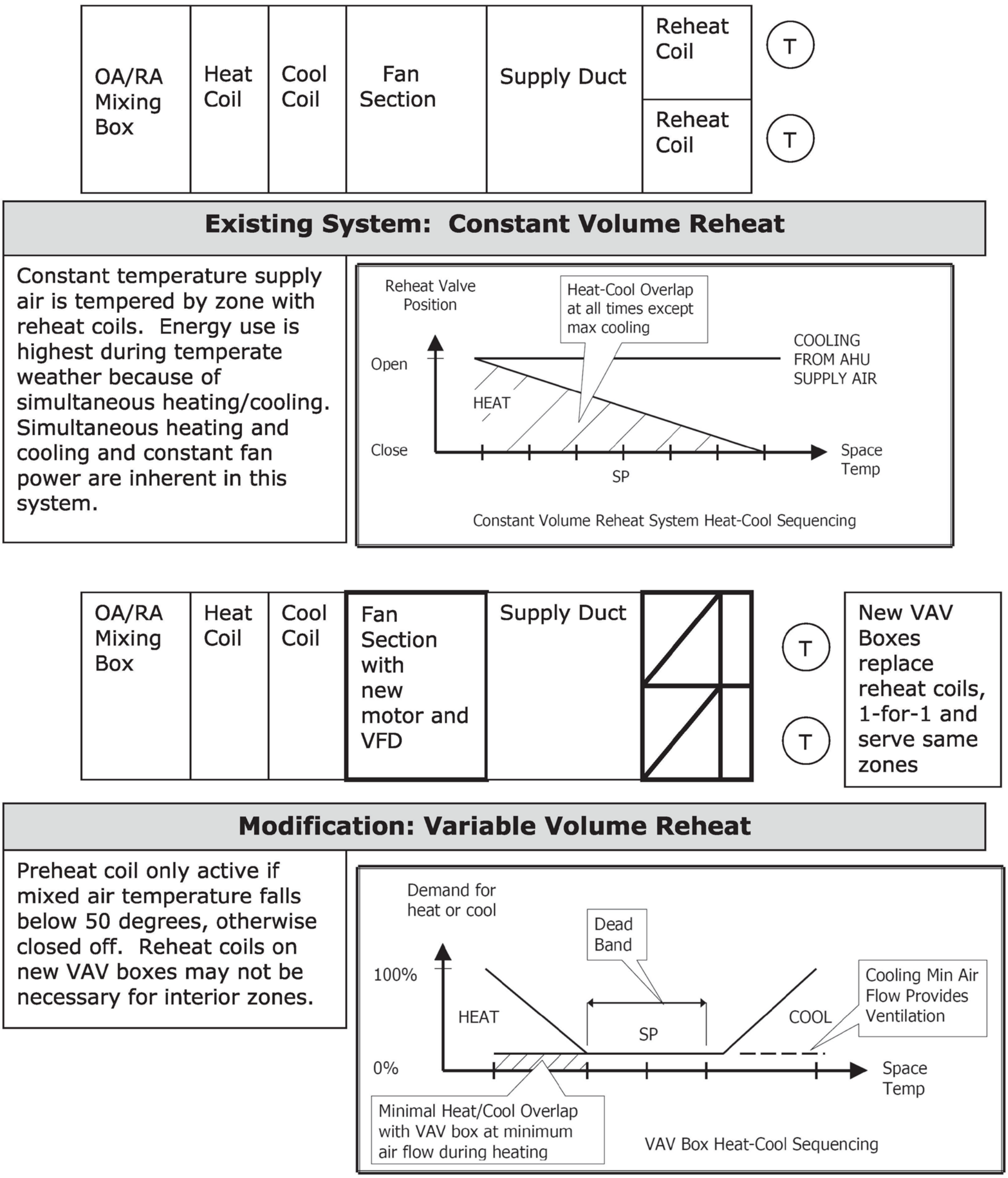


Figure 24-G2. Multizone Retrofit to VAV Reheat

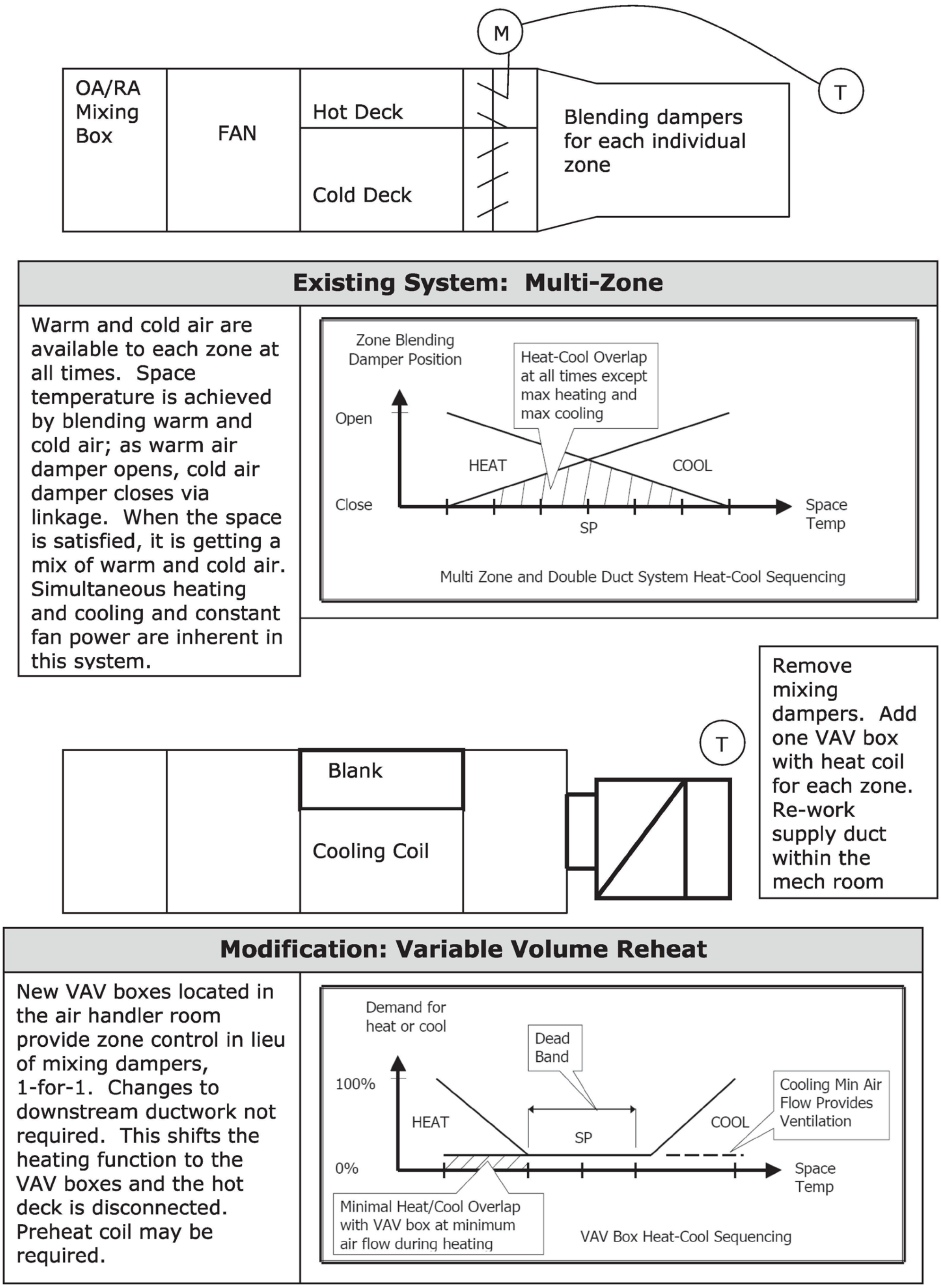
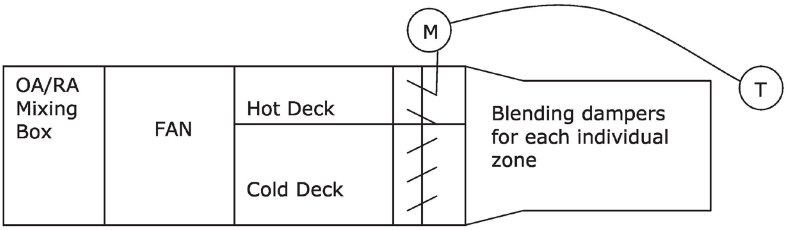
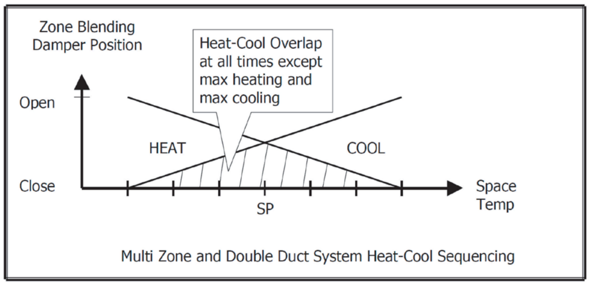


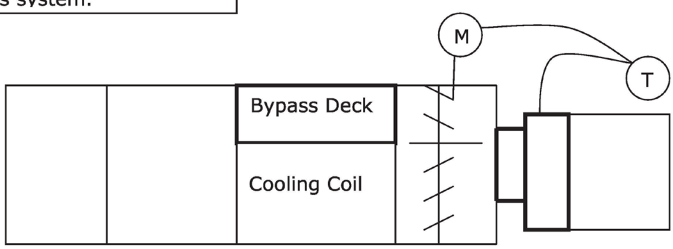
Figure 24-G3. Multizone Retrofit to Texas Multizone



Existing System: Multi-Zone

Warm and cold air are available to each zone at all times. Space temperature is achieved by blending warm and cold air; as warm air damper opens, cold air damper closes via linkage. When the space is satisfied, it is getting a mix of warm and cold air. Simultaneous heating and cooling and constant fan power are inherent in this system.





Keep the mixing dampers. Add one heating coil for each zone.

Modification: Constant Volume Texas Multi-Zone

New heating coils located in the mech room provide zone heating control in lieu of mixing dampers, 1-for-1. Heating occurs only after cooling coil is bypassed, eliminating overlap, but still constant volume. Changes to downstream ductwork not required. Hot deck is disconnected. Preheat coil may be required.

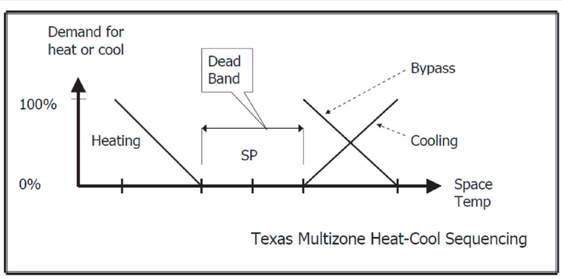
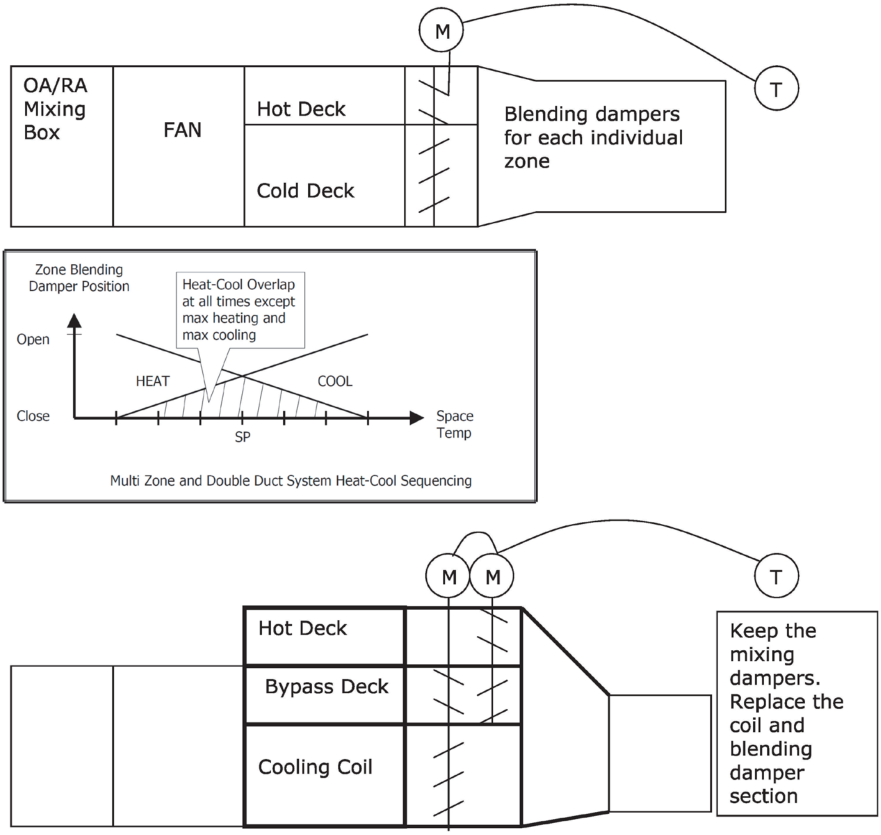


Figure 24-G4. Multizone Retrofit to Three-deck Bypass



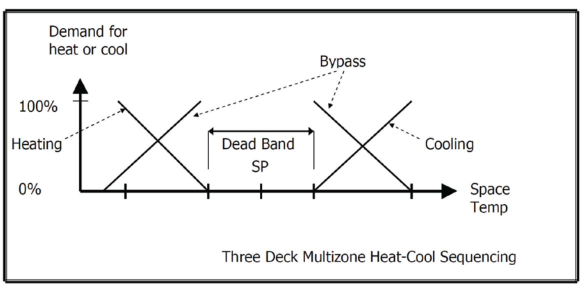
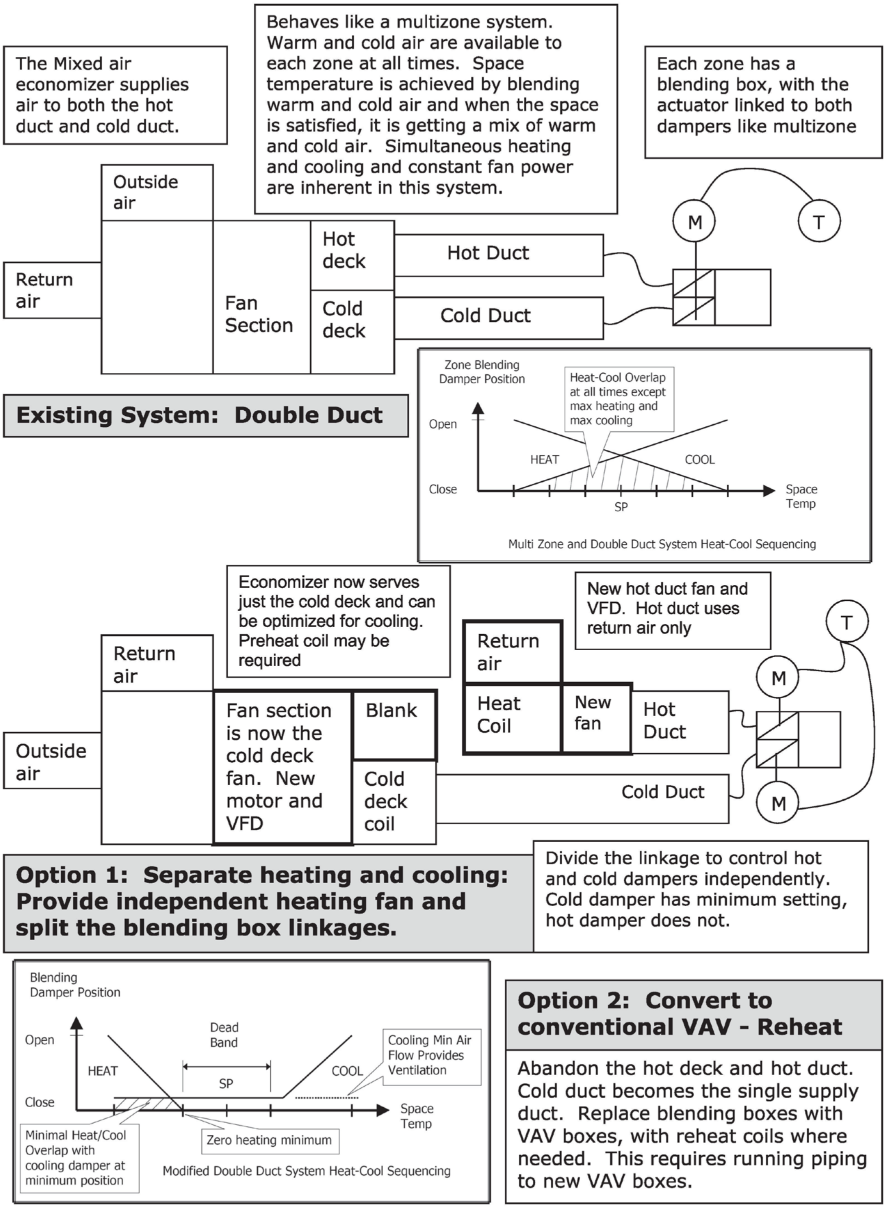
Modification: Three Deck Multi-Zone	
Bypass deck allows the use of two actuators – note that the cooling and heating dampers are no longer linked. Changes to downstream ductwork not required. This allows individual heatiing and cooling control by zone without overlap, but is still constant volume. Preheat coil may be required.	

Figure 24-G5. Double Duct Retrofit Options



H—CHILLED WATER SYSTEM DISCUSSION AND ECMS

Understanding chilled water system behavior is important for maximizing energy benefits of ECM applications. These plants are often treated alone, but are closely related to the systems they connect to. Energy savings of a replacement chiller unit can be increased by up to 20% when other measures are implemented to raise the system dT.

For example:

- Excess chilled water pumping at any location creates added heat and chiller load.
- Variable primary flow creates a trade-off between pump energy and chiller energy, and the same is true for variable condenser water flow.
- Excess secondary pumping (distribution) creates corresponding high primary flows.
- Low chilled water differential temperature (dT) causes high chilled water flows, excess chiller run time, and low chiller loading at inefficient operating points.

A common thread in chilled water system improvements is the dT. Raising dT creates immediate energy benefits throughout the chilled water system but achieving the goal can be elusive and usually involves work beyond the chilled water system itself. This discussion is limited to the conventional primary/secondary pumping methods used for chilled water systems.

PROPORTIONAL PUMPING ENERGY USE

The goal:

- Minimize parasitic energy expense related to cooling energy transport.
- Keep distribution energy cost proportional to plant load.
- Delta T disease undermines both of these objectives.

System Terms

Pumping schemes are generally in these categories, sometimes with different names. They are all affected by Delta T.

- **Primary loop**: pumps that move water through the primary cooling equipment in the plant. This is often a short section of large pipe that resides only in the main plant, but the primary pumping loop can also extend remotely as far as desired—the determining factor is the location of the secondary pump units. Primary piping almost always consists of a single 'loop', with branches, and can include storage. It is in the primary loop that the chilled water is "made."

- **Secondary loop**: pumps that move water from the primary loop through the building in one or more 'loops'. For example, a chilled water plant can have a secondary loop for each of several buildings and a new loop and pump set can be added for each additional building constructed over time. In most cases, the secondary loop delivers the heating and cooling energy directly to the load (the air handlers).
- **Tertiary loop**: In some cases, a third tier of pumping is provided. This may be the result of buildings that originally operated stand-alone, with their own chiller, that have since been incorporated into a large central distribution network—and leaving the original building pumps in place may be done for convenience. In some very large systems (e.g. district cooling systems) the additional pumping tiers may be needed to separate primary equipment and buildings from high pump discharge pressures—in extreme cases, heat exchanges may be used for this hydraulic isolation.

Energy Implications

Chilled water system differential temperature dictates required flow. For a given cooling load, the flow rate will go up as the dT goes down, and vice versa.

This familiar equation

$$\mathbf{Q=500 * Gpm * dT} \qquad \textbf{eq. 1}$$

yields

$$\mathbf{Gpm= Q/(500*dT)} \qquad \textbf{eq. 2}$$

Where:

Q = cooling load, Btuh
Gpm = gallons per minute flow rate
dT = differential temperature, degrees F

Example: For a load of 500 tons (6,000,000 Btuh), a 10 degree dT will require 1200 gpm, while a 5 degree dT will require 2400 gpm.

Power savings are proportional to flow, with some limiting factors like part load efficiency losses and maintaining a minimum pressure in the system, but within a fixed pipe size, velocity is proportional to flow, and power increase can be estimated with affinity laws.

Hp2 = Hp1 * (Flow 2/Flow 1)^2 **eq. 3**

Pump flow rate varies inversely as the dT (eq. 2). For a fixed cooling load and variable dT, (eq. 2) becomes:

Gpm 2 = Gpm 1 * (dT1/dT2) **eq. 4**

Substitution of (eq. 4) into (eq. 3) yields:

Pump Power 2 = Pump Power 1 * (dT1/dT2)^2 **eq. 5**

Actual Pump Energy = Design Pump Energy *

For a given load profile throughout the year, a certain number of tons are needed for a certain number of hours, regardless of the form it takes getting there. Therefore, the change in pump energy is proportional to pump power (eq. 5), allowing annual operating cost and the impact of reduced dT to be identified.

(dT design/dT actual)^2 **eq. 6**

Example: A chilled water system is designed to deliver full cooling capacity at 10 degF dT, but is only achieving 7 degF differential. Find the increase in water flow and pumping power required to deliver the full load to this system. Original pump flow and power is taken as 100%.

New pump flow (eq. 4): 100% (10/7) = 142%
New pump power (eq. 5): 100% * (10/7)2 = 204%
New pump energy (eq. 6): 100% * (10/7)2 = 204%

All this fuss over a few degrees? Yes, indeed.

Extra pumping in chilled water systems creates a parasitic loss, because all of the pump energy ends up as heat in the water, creating more load and even more pumping. Each 5-Hp of pump power (Bhp) creates about one extra ton of parasitic cooling load. This makes chilled water pumping (and cooling air distribution for that matter) unique from heating systems, where distribution energy heating is beneficial.

Tons = Pump BHp * 0.746 * (3413/12,000) **eq. 7**

Where:
Pump BHp = pump shaft input power, or (water horsepower/ pump efficiency)

Chiller Plant Disease: Low System Differential Temperature (dT)

Chillers don't create low dT, they simply have to deal with it. Increased pumping energy (read savings opportunity) is inversely proportional to low chilled water dT. This applies to all tiers of pumping for chilled water systems.

Consequences of low chilled water system dT:

- High chilled water flows—rivers of water going in circles.
- Chiller efficiency is low—especially when operated below 50% of maximum load.
- Chiller run time and annual maintenance costs increase when primary equipment expects a higher dT than what is delivered from the secondary system. Wear and tear costs are directly related to run hours.
- Chilled water pumping costs increase in proportion to cooling load (transport energy penalty).
- Chilled water load increases from added pump heat; approximately 1 ton load per 5-Hp pumping power.

FLOW MATCHING

In primary-secondary chilled water flow design, the building load is reflected in the secondary distribution system—the chilled water segment that touches the load. Ideally, the secondary flow will be proportional to load, but whether it is or is not, the primary flow must match it. This is another key difference between chilled water and heating water loops—in a heating loop, heat transfer is sensible only, and primary/secondary flows need only roughly match to function. However, chilled water cooling loads will be adversely affected by rising chilled water temperatures, especially in humid climates since dehumidification of cooling coils drops sharply with a rise in entering chilled water temperature. For this reason, primary water flows are almost always kept higher than secondary flows, to avoid any warm system return water from bypassing the chillers and heading back out into the system, mixing with 'fresh' chilled water.

Whether by temperature, flow, or a combination, automatic control systems for primary-secondary chilled water systems will start/stop or throttle primary flow pumps in response to changes in secondary flows to maintain equal or surplus primary flow, thereby assuring 'fresh' chilled water supply reaches its ultimate destination. When a primary pump is started a chiller must also be started; otherwise the pump will inject return water into the supply and raise supply water temperature. This points out a disconnect in these chilled water plants: primary equipment is started in response to flow and not load.

Each chiller is specified and built to achieve its stated cooling capacity for a given dT and flow. Similarly, the air handlers that use the chilled water are selected based on design flow and dT. When these are equal, the secondary distribution network is merely the connecting link between the two and things work fine. But when, for whatever reason, the design flows and dTs of primary, air handler, and secondary systems do not match, there are issues. The dysfunction and energy waste of primary chilled water pumping systems is directly proportional to this mismatch.

Consequences of dT mismatch:

- System dT higher than chiller: primary equipment operates beyond its capacity and either overloads or cannot keep up. This case is short lived because operation and comfort are not sustainable.
- System dT lower than chiller: This is a common blight of central cooling plants. From the chiller viewpoint, the water differential is low and the primary equipment does not load. For large systems with large flows of low dT water, multiple chillers will be operated to balance high secondary flows with high primary flows—i.e. starting extra machines simply to process all the water. The symptom of this condition is multiple chillers operating with low percent load—e.g. 2 chillers operating at 40% load instead of one at 80% load. Additional run-hours on machines is an identifiable cost. Energy use increases come from:

1. Excess pumping creates added load because all pump work and pump inefficiencies end up as heated water (Eq. 7).
2. Chiller operation at low load/low efficiency points. Chiller efficiency is usually low when operated below 50% of maximum load.

Understanding the effect of Delta T throughout the chilled water system gives insight into ECM interactions and the benefits of approaching energy improvements from a system basis.

Chiller plants do not operate independently. Refer to **Figure 24-H1**. Individual chiller plant ECMs are described here; interactions vary. While replacing a chiller is often the first or only ECM, the presentation will follow the hierarchy and build the best performing system from the base, up. This also points out that the system ECMs can improve existing chiller efficiency all by themselves.

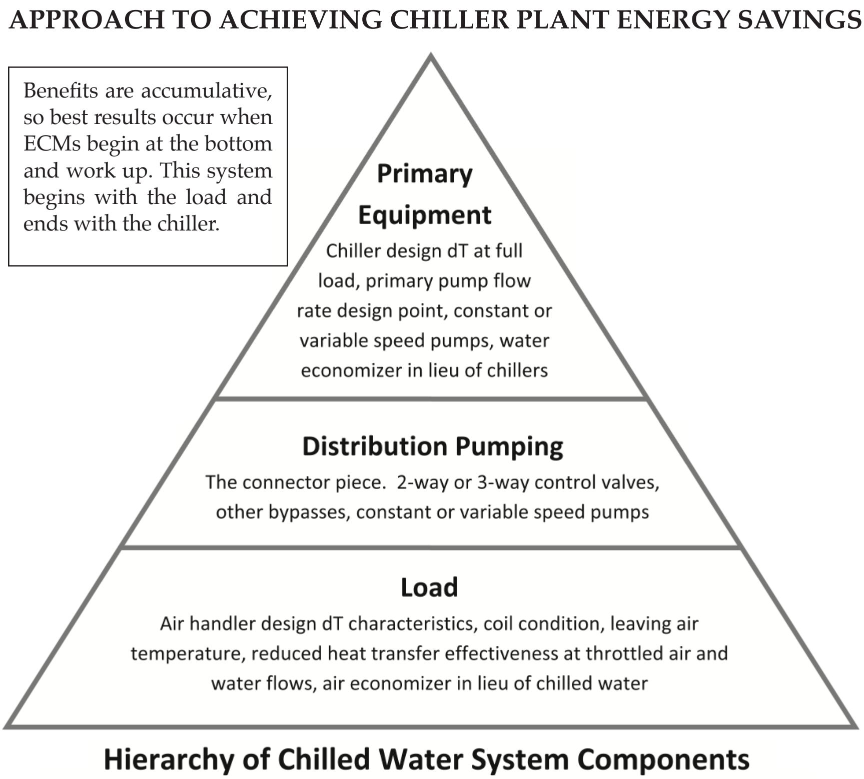


Figure 24-H1. Chilled Water System Hierarchy

MODIFYING AIR HANDLING SYSTEMS TO INCREASE SYSTEM DT

In all cases, the preferred solution is to have a building air-side system (the interface point between chilled water and warm air) be such that it automatically provides the necessary dT,—when this happens, variable flow systems work as intended, chillers load properly, and all is well. When the load dT and primary pumping dT are not aligned, dysfunction of the chiller plant occurs because staging becomes a matter of flow instead of load. The extent of the dysfunction and pumping energy increase is noted in (eq. 6) squared ratio of higher/lower dT value. The usual case is the primary dT assumption (chiller and primary pumps) are higher than the load they will actually see. For a 10-degree dT chiller plant, an 8 degree dT load will increase pumping costs by (10/8)2=56% (eq. 6) with attendant energy waste from pump heat and poorly loaded chillers. Make no mistake: low dT is costly.

Raising the chilled water dT on an existing system requires addressing the source of the load. A common remedy is air handler coil replacements; larger coils with longer water/air contact time and surface area allow higher extraction, higher dT, and lower water flows.

The expense and downtime related to overhauling the building air handlers is a deterrent, and gives rise to heating/cooling plant low dT syndrome that has been there for years and the various workaround strategies that are less costly to try. For such systems, a migration approach is possible, whereby any air handler replacement is mandated to have a high dT—sometimes even higher than the desired system dT, over-compensating for an accelerated effect on the overall loop. For long-lived institutions, it is helpful to quantify the annual energy penalty related to the do-nothing option as a savings incentive for making system upgrades.

The best way to determine what the system dT is comes from a review of operations logs, either from an energy management system (trends) or from clipboard daily readings. Measurements of plant loads, chiller loads, primary/secondary flows, or differential temperatures are all indicators of whether distribution pumping energy is in proportion to load. Other things that pull down a system dT:

- Undersized air handler coils, either originally or from increased cooling duty over the years.
- Dirty heat transfer surfaces—fouling on the outside or inside of the tubes.

- Water balance, creating excess water flow.
- Control valves that do not close tightly, bleeding by internally.
- 3-way valves that divert supply water to return rather than throttle.
- Control settings for supply air temperature too high (heating systems) or too low (cooling systems) causing the control valve to be wide open all the time in a futile attempt to achieve the unreasonable setting.

Interaction: Load to secondary flow. The secondary pumps are the linkage between the load and the chillers. For a given load, the flow (and pump energy and extra cooling load from it) is proportional to the differential temperature (dT). So, the variable secondary pumping system will automatically have less to do if changes are made to increase the dT.

With the load set for variable flow and high dT, variable flow secondary pumping will respond well and energy transport costs will be minimized. With the chiller and primary pumping 'tuned' to the same frequency, pumping mismatch will be corrected, chillers will load fully, and refrigeration unit efficiency will be high.

VARIABLE SECONDARY FLOW

Building cooling load changes are normally accommodated with automatic control valves that throttle the amount of chilled water into the air coil. Constant flow systems divert excess water to the return and are termed constant flow-variable temperature systems and works well for smaller systems, especially with a single chiller. But for larger systems with multiple chillers, the change to a variable flow-constant temperature system will bring good savings. For the constant flow design, pumping cost is constant, even at zero load, while the variable flow system aims to keep the pumping cost in proportion to load. Variable flow secondary pumping begins with replacing 3-way control valves with 2-way control valves in constant flow systems. This change reduces system flow rate when control valves close, allowing the distribution pumps to slow down in response. Some systems are a mix of 2 and 3-way valves and improvements can be made by removing remaining 3-way valves.

> ***Note***: *simply capping off one of the ports of the 3-way valve may or may not work, since the valve/actuator combination was designed to merely*

'steer' the water in a different direction rather than stop the flow altogether—at the very least, the actuator may need to be replaced, and an all-new valve is better.

Included in this measure are the variable speed drives (VSD) and controls to throttle the secondary water flow according to need, usually based on a minimum downstream pressure that is sufficient for loads (air handlers) to get needed water flow and cooling when a valve opens.

Energy savings estimates require a load profile, where the percent flow is assumed to track the percent load.

Example: Chilled water secondary distribution pump at full flow requires 35 Hp (29 kW) of pump power. Half of the control valves are 3-way and so at 50% load the system is still circulating 75% of the flow. Removing all remaining 3-way valves will allow 50% flow at 50% load. Find savings for 2000 hours of operation at 50% load with this modification.

Existing pump power:	29 * (0.75)2 = 16.3 kW	**(eq. 3)**
New pump power:	29 * (0.50)2 = 7.3 kW	
Pump power reduction:	16.3-7.3 = 9 kW	
Energy savings:	2000 hours * 9kW = 18,000 kWh (ans.)	

As with all chilled water pumping, the pumping energy ends up in the water and adds to the cooling load in parasitic fashion, so reduced pump energy augments system savings by reducing load. From (Eq. 7), cooling load reduction at the 50% flow point is 9 kW * (3413/12000) = 2.6 tons.

The concept of secondary loop flow reduction savings is not unfamiliar, but the interactions to/from related systems are of interest.

Interaction: Secondary flow to primary flow. With few exceptions, chilled water systems must deliver 'fresh' chilled water to the point of use. With a primary-secondary pumping arrangement, the primary flow rate (through an operating chiller) must be at least equal to the secondary flow, or else mixing will occur and chilled water temperature delivered to the point of use will warm up (bad). Thus, the two flow paths are very related and a decrease in secondary flow allows a reduction in primary flow. In many, many chilled water systems, extra chillers and primary pumps are started to process all the secondary flow, instead of based on load; the result being chillers running at low loads at low efficiency points. Reductions in secondary flow that are accompanied by a steady and sufficiently high dT will allow chillers to run according to load as well as flow, thereby running less chiller/pump sets and operating them at higher loads and higher efficiency.

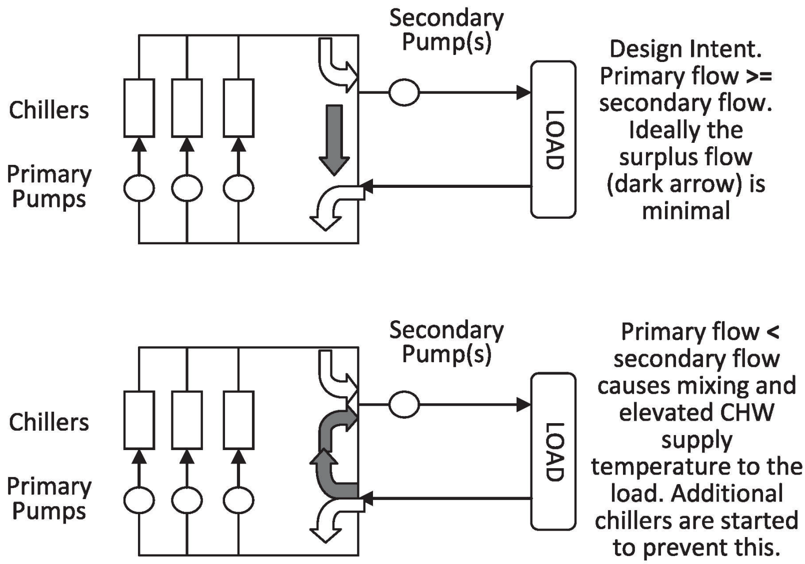


Figure 24-H2. Primary-secondary flow diagram

HIGH EFFICIENCY CHILLER

- Evaluate hours at part load, and include part load economic and efficiency evaluation at the dominant loads that will be experienced, not just full load.
- Variable speed units can provide exceptional part load efficiency.
- Identify hours of operation that will occur below 50% load and provide alternate operational modes or jockey equipment to avert high part load losses that are inevitable at very low loads.
- Select equipment that is compatible with the system, especially the system differential temperature (dT). Many, many systems are poorly applied in this regard.
- Select equipment with low pressure drops, to reduce circulating pump energy input. The large amount of heat transfer that takes

place requires a combination of heat exchanger surface area and turbulence, so easy-flowing systems will necessarily cost more and the benefit of reduced pumping cost requires evaluation. Also, any intentions of varying the flow rate through a chiller will immediately change the heat transfer characteristics and the chiller selected for lowest water pressure drops will be affected more.

- Make ample provisions for heat exchanger service and cleaning, to encourage these activities and achieve higher overall average heat exchanger performance during equipment life.
- Consider oil-less designs for the heat exchanger enhancement benefits they produce, but only if machine reliability is assured. Conventional designs of refrigeration equipment circulate oil with refrigerant and oil films on the insides of heat exchangers impede heat transfer significantly. Thus, the absence of oil would be an inherent advantage in improving heat transfer and efficiency.
- The parasitic cooling load imposed by hermetic designs is avoidable by an external motor. All motor input work becomes cooling load, however the motor inefficiencies (on order of 5% of motor input power) becomes additional cooling load for hermetic designs

  Example: a 1000 Hp motor driving a chiller with a 5% motor loss creates an extra 50 Hp of heat which translates to 10 tons of cooling load, increasing power requirements and reducing net cooling output of the machine by 1%. Conventional designs rely on a shaft seal for such applications and avoiding worries about refrigerant leaks at the seal may be a chosen tradeoff for the energy penalty.
- Choose equipment that can leverage optimization strategies for additional savings such as:
  - Low condenser water in dry climates—equipment that cannot accept cold condenser water and must operate on artificially higher head pressures for the sake of refrigerant or oil management can give up considerable savings.
  - Provisions to tolerate switching back and forth to water economizer operation in climates where these are used.
  - Tolerance to resetting chilled water and condenser water temperatures where it is advantageous to do so. Systems that need

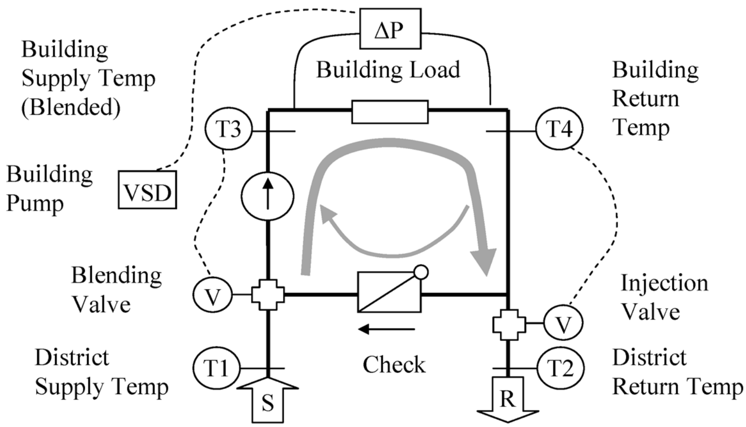
to maintain wide differential pressures or maintain minimum head pressures represent machine design limitations that impose restrictions to energy saving measures.

— Remote reset of leaving water temperature setting and demand limiting.

OTHER CHILLED WATER SYSTEM ECMS

TERTIARY PUMPING

For larger district cooling systems with an additional layer of pumping, the concepts of managing dT and energy transport costs apply. The energy focus is that these systems pass the load on to the upstream secondary flow system and must be compatible to do this efficiently. For example, a distribution pipeline may deliver chilled water to various buildings, each with their own cooling loop. The coupling of the two may be a heat exchanger, but is more commonly another "bridge" that provides cooling as needed. Excess pumping within the tertiary loops adds parasitic load like any other pumping scheme and efficiency gains are available by eliminating them. Attention is often given to the coupling "bridge" between secondary and tertiary loops, with limited results. Decoupler 'bridge' monitoring controls attempt to force a given dT onto the distribution system by restricting the water flow—usually based on only accepting return water of sufficient temperature rise. In operation, only as much chilled water is allowed to the loop or building as it can use and return the set dT, and heating/cooling flow is denied until this happens. These methods do save energy at the pump and plant level, but should be watched closely to avoid climate control issues in the building. For example, if there are old air handler coils or marginally sized coils that cannot achieve proper air conditions with the optimum dT, then controls that force them to live with not enough water flow will create a loss of air temperature control and comfort issues—a case of solving one problem and creating a new one.



<u>Basis of Savings:</u> Maintains building load ΔT, enabling variable flow pump savings.

<u>Operation:</u> Injection valve is controlled by building return temperature T4 to only allow new chilled water when the existing chilled water has been fully utilized, i.e. when T4=design value of T2. Building supply temperature is controlled by T3 and blending valve. Variable flow pump control in the building (VSD or throttling valve) maintains differential pressure sufficient for building cooling load (2 or 3-way valves) and prevents excessive mixing which otherwise can cause dehumidification impact from elevated building supply temperature.

<u>Application:</u> Scenario is separate buildings formerly had their own water chillers and have since been adopted into the district cooling service territory. If district design supply and return temperatures are the same as building design supply and return temperatures and district pumps have enough pressure for the buildings, tertiary building pump, bridges, and controls are not needed and the system will work better without them. When tertiary pumps are needed, they must be de-coupled from the district pumps with the cross over pipe which then creates potential for short circuiting district flow S-to-R without a check valve. When district design supply temperature is lower than building design supply temperature, the blending valve is used along with the building pump to raise building supply temperature. If district design supply temperature and building design supply water temperatures are the same, the blending valve is not needed. As with all chilled water systems, economical pumping relies on coils that perform at the intended system ΔT.

<u>Diagnostics</u>
If T2<T4, check valve is leaking.
If T3> cooling coil design temperature, humidity control is impacted.
If T2<T1, district connection pipes to the building are reversed.

Figure 24-H3

VARIABLE PRIMARY CHILLED WATER FLOW (DEDICATED PUMPS)

Modulating flow through chillers is directly in line with the overall desire to have pumping costs kept in proportion to load, however some caution is necessary.

If flow reductions are moderate, savings are available, but when reduced too far, chiller efficiency is reduced—mostly from a loss of heat transfer at the tube boundary layer, creating higher heat exchange approach and greater lift for the refrigeration cycle. The answer often depends upon the full flow tube velocity, turbulence, and heat transfer characteristics—the sooner the chillers 'lose' the boundary layer heat transfer, the sooner they will exhibit a drop in efficiency. The answer may also depend upon the chiller manufacturer, understandably conservative with the application of their machine, and how much testing they have done to confirm ranges of reliable operation. Where serious doubts or warranty issues arise, defer to the manufacturer and focus on other ECMs. Some other considerations of low flow operation should be mentioned:

- Low flows that can aggravate freezing of an evaporator when operating at low water temperatures
- Low flows that can make proof-of-flow switches unreliable

Included in this measure are flow controls for the water through the chiller which may include a flow meter or differential pressure sensor across the tube barrel, and a water throttling means such as a variable speed drive or modulating control valve.

To quantify savings of variable chilled water flow requires a case-by-case comparison, with the help of the chiller manufacturer, comparing the kW saved from the pump with any kW increase at the chiller. Results vary, so always include this research step and do not make implementations on rules of thumb for this ECM. While a full range of proportional flow reduction is bound to have issues, flow reductions to 50% are usually tolerable and fruitful. One chiller manufacturer's statement confirms this: Chiller kW/ton is basically unaffected by evaporator flow reduction down to 50 percent of design flow (assuming flow reduction is in proportion to chiller load reduction with a constant dT (1).

(1) "Will Variable Evaporator Flow Negatively Affect Your Centrifugal Chiller?," *Engineering System Solutions*, April 2000, McQuay International.

Bearing in mind that the chiller power requirement is an order of magnitude higher than the pump, it makes little sense to reduce pump energy when there is a chiller penalty. Still, it is clear that a chiller operating at 30% load does not need 100% of water flow –so there is a balance. The direct approach maintains a fixed "gpm per ton" specific flow rate at all times; but this neglects the reality that reduced flow rate means reduced heat transfer, so a chiller penalty is predictable. A minimum flow rate of 50% is prudent to stay solidly in the turbulent zone of fluid movement through the tubes, avert freeze damage, etc.

Some amount of increase from the full load specific gpm/ton flow value is needed to minimize chiller penalties and strike the necessary balance. The amount of bias depends on the chiller, and each is a unique case. This is because chillers can be selected at any number of conditions for evaporator flow, and therefore upon flow reduction some will begin to lose heat transfer effectiveness in the tubes sooner than others. Tube velocity is an indication of this, but not the final determination—for example a given chiller and tube configuration designed for 6 feet per second velocity will fare better at flow reduction than one selected for full capacity at 4 feet per second.

Savings of variable speed pumping depends upon the load profile, and how many hours are spent at reduced loads. Savings are tangible but not large, in the range of 2-5% overall energy savings, with the most savings occurring at low load. Building loads with a wide distribution of percent plant load will have opportunities for variable flow control in general, while process cooling chillers with steady loads may not benefit at all.

VARIABLE CONDENSER WATER FLOW

The condenser-half of the chiller is responsible for heat rejection. The heat exchange process in the condenser are affected by flow reduction just like in the evaporator, and so the same study in trade-off between pump energy saved and chiller energy penalty apply. Pump heat is rejected to the cooling tower, so does not increase cooling load, other than some additional water the cooling tower will use to dissipate it. Savings for variable condenser water flow control are similar to variable chilled water flow, with the same caveats. A difference is that the condenser pumping circuit, if connected to a cooling tower, is an open circuit with a static lift component, and so the pump savings will be less

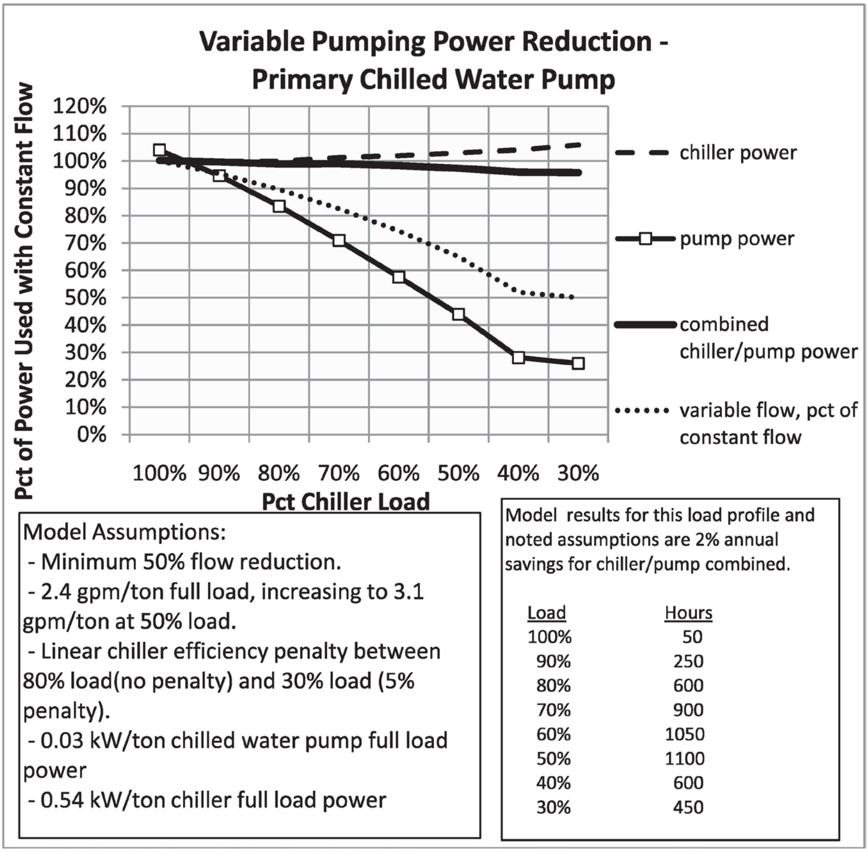


Figure 24-H4. Example Analysis of Variable Chiller Flow Pumping for One Set of Assumptions
Compares pumping benefit to loss of chiller performance.
This example shows only a slight net gain.

than for a friction-only system.

A practical consideration for variable condenser water flow is the cooling towers, where the varying water flow goes. Most cooling towers distribute water over the fill with rudimentary distribution pipes with holes in the bottom, and flow reductions below some point produce uneven flow and undesirable cooling tower performance, which can include reduced thermal performance, icing, and even fan overload. However, most cooling towers can accommodate a flow turndown of 50% without much trouble (still need to ask).

PRIMARY-ONLY PUMPING

In this scenario, all the water always goes through a chiller and bypassing is inherently eliminated. Flow is reduced with load until a minimum flow is reached, at which time water is circulated through the chiller to sustain minimum flow rates. Like all variable primary pumping schemes, analysis is required to compare gains in pump energy to losses in boiler/chiller efficiency from reduced tube heat transfer effectiveness (Reynolds Number changes). This is a complex system to design and usually includes significant piping modifications, flow stations, and controls to always assure minimum flows through the primary equipment even as distribution load drops. A distinct advantage of this system is compactness—without the layers of pumping, the number of equipment items is less, along with the required space.

Savings are achieved from higher proportionality of flow to load than primary-secondary systems.

Example: A 1000-ton chiller operating with a 500-ton load using primary-secondary pumping will have a primary pump running that is delivering twice the water needed. The primary-only system conserves these flows, as well as the surplus flows that occur at decoupler bridges.

Like all chilled water systems, the performance is limited by the load it serves and the signature dT.

CHILLED WATER ECONOMIZER

When there is a need for cooling inside a building coincident with dry conditions outside, a water-side economizer can be useful. This arrangement uses the evaporative cooling power of a cooling tower to lower the condenser water to a level lower than the chilled water. Then, through the use of a large heat exchanger, the chilled water is made without the chiller. Another name for the water-side economizer is "flat plate" system, so named because a plate-frame heat exchanger is normally used due to the very low approach temperatures they can offer. A good use for a water-side economizer is to delay the start of a chiller in dry climates, operating at dry bulb temperatures that do not accommodate a standard air-side economizer. The other viable use for a water-side economizer is when the persistent cold weather cooling loads cannot tolerate the dry air that comes with the air economizers, e.g. a data center. It is important to note that using a water-side economizer in cold weather when an air-side economizer is available is not good economics. Presuming the air-side fans are in use anyway, cooling from an air-economizer is truly free, while the water-side economizer can take 25% or more of the energy the chiller system would use since it uses all the auxiliary pumps and cooling tower fans. Cooling efficiencies of 0.2 kW/ton are common and so the water-side economizer is definitely not free. At very low temperatures, some facilities experience problems with air economizer mixing boxes and prefer the water economizer for added freeze protection—in this case the savings of using an air economizer can be used to fund modifications to poor performing mixing boxes. See **Chapter 5 Water Economizer vs. Air Economizer. See Chapter 9** for a method of quantifying savings.

VARIABLE FLOW DURING WATER SIDE ECONOMIZER OPERATION

For systems with a water-side economizer (flat plate heat exchanger), the extended hours of operation of the condenser and chilled water pumps during winter months offer additional savings potential when switching to variable speed pumping. It is common to find constant flow condenser and chilled water pumps running at 2 degrees F dT during flat plate mode, raising the cost of the 'free' cooling system unnecessarily. In this case, the pump heat of both the condenser and chilled water pumps are a detriment to the system, by raising the temperature on both sides of the flat plate.

The measure consists of modulating both pumps based on load, with savings occurring in colder months when the heat exchanger has excess capacity.

An added bonus of adding variable speed VSD control to the condenser pumps is in making the transition from flat plate back to conventional cooling when the initial condenser water temperature is too cold for chiller operation. Throttling water flow in response to chiller head pressure will help get the chiller through this transient event.

CONDENSER WATER RESET

Chillers are very responsive to reduction in head pressure and can use significantly less energy when colder condenser water is supplied. This is of course a tradeoff between cooling tower energy and chiller energy and the answer to where the break-even point is requires cooling tower specific power analysis. In most cases (unless cooling tower bodies are undersized and are compensating with excess fan power) good savings are available from "condenser water reset" when outdoor conditions are favorable and lower condenser water can be easily obtained. For example, a chiller designed to use 100% of power at full load with 85 degree F entering condenser water can provide full cooling capacity with 20-30% less power when the condenser water is 65 degrees F; a striking reduction. To get the most of this valuable control routine requires knowing the balancing point between added tower fan kW and saved chiller kW, and a chiller that accepts colder condenser water. Where condenser water reset is anticipated, a valuable performance specification for a replacement chiller is the ability to operate normally at full design condenser flows of 3 gpm per ton at 60 degF condenser water temperature at all chiller load—this capability will facilitate good savings at part load conditions. Significant to the topic of tradeoffs is that reducing flow rate through a condenser while lowering its temperature may in fact negate the savings of both; the end result in kW/ton will depend on the head pressure the compressor works against, and this is driven by the mean temperature of the water. Thus, very cold water entering is offset by very warm water leaving, etc. Generally minor reductions in flow and aggressive reduction in cooling tower water leaving temperature yield good results, again, provided the cooling tower is properly sized to provide the extra cooling without undue extra fan Hp. Determining the balance point of energy expense and savings for cooling tower reset requires knowing the specific power requirements (kW/ton) of the cooling tower and the chiller. Using a ratio of the two, the analysis can be resolved into some basic rules of thumb and the higher the ratio of chiller to tower specific power, the lower will be the economic approach value of the cooling tower where overall energy is still being saved—i.e. the more you can push the cooling towers. See **Chapter 9** for a method of quantifying savings.

Table 24-H1. Evaluating Condenser Water Reset Using Chiller/Tower Specific Power Ratio

Chiller kW/ton	Tower kW/ton	Ratio of Chiller / Tower kW/ton	Typical Lowest Economical Tower Approach For Condenser Water Reset, degF
0.5	0.10	**5:1**	**15**
0.5	0.085	**6:1**	**13.5**
0.5	0.07	**7:1**	**12**
0.5	0.06	**8:1**	**11**
0.5	0.05	**10:1**	**10**
0.5	0.04	**12.5:1**	**8.5**
0.5	0.03	**17:1**	**7**

I—COMMISSIONING

Resource for additional information: Simplifying the Commissioning Process, paper presented at WEEC (World Energy Engineering Congress), Washington, DC 2006, Association of Energy Engineers.

Third Party

While not essential, it is good practice to have commissioning or other quality control advice come from an independent third party, since this eliminates the potential for conflicts of interest.

Rigor

The level of effort and cost of commission is variable, and is usually determined by the owner based on perceived risk that commissioning can mitigate.

Full Commissioning

A process that creates a quality control advocate to the owner called the commissioning agent. The overarching goal and value of the service is verifying that the facility systems meet the functional and operational needs of the building owner and occupants. Key to the process is identifying the project intent which defines in measurable terms what the owner's requirements are; all commissioning activities then serve to assure the project intent comes to pass. For the owner, this process increases confidence during the period between conceptual planning and final construction, and increases the chances of the owner being satisfied with the end result. Early detection techniques are used heavily throughout each phase of the work, including design reviews, equipment submittal reviews, construction process reviews, and functional testing (start-up review). For example, early detection of a repetitive construction process can avoid large amount of re-work and associated delays if only detected at the very end of the project via a punch list. When early detection mechanisms identify potential issues, the owner is counseled and has the opportunity to make a correction to minimize impact. The early detection can prove very valuable to an owner depending upon what issues are discovered—especially those that build-in problems for the life of the building. Commissioning (Cx) can be applied to any building component or manufacturing process, but is usually applied to building systems such as HVAC and control systems since these have historically been problematic. Sampling techniques are used to spot-check work processes. A common byproduct of commissioning is improved coordination between trades. A common complaint of commissioning is the added cost of the QC process and the resistance to change for established processes of designers and contractors. Commissioning often enhances sustainable building operations by focusing on practical matters such as service access, documentation, and operator training.

Start-Up Commissioning

The final phase of a full commissioning project, this activity is sometimes the limit of the commissioning scope. It exercises the various systems to demonstrate they work and work together, and verifies the design is functioning as intended. The functional testing will usually include a formal written plan that serves to document the work and allows repeat commissioning at a later date or periodically to maintain operating service level. The assumption is that the design is appropriate (no design peer review) and that construction QC is acceptable (no construction oversight). This option reduces commissioning cost, but includes no early detection of design or construction issues and so offers reduced value in project intent protection. With start-up commissioning it is possible to have a fully functioning system that was not a great idea to begin with.

Retro Commissioning

This process focuses on an existing building that has slipped from its intended operation, or is being used differently than originally intended. Retro commissioning provides an evaluation of the existing facility and an assessment of overall functional parameters. Recommendations are made to restore good operation to the existing facility, and to identify opportunities for improvement. Retro commissioning in a general sense is a "tune up" of an existing building, with the goal of making it perform like it was intended by design. Typical tasks include identification of the original project's intentions, basis for the original design, verification of documents (as-built drawings), functional testing, and owner training. Facility changes since inception, and new opportunities for improved operation are also considered. This service may also include testing adjusting and balancing of the HVAC system or additional engineering services for cost/savings evaluation. This process often uncovers deferred maintenance and repairs and major repairs are essential to the success of any retro commissioning attempt.

The following have been identified by owners as the primary objectives for retro-commissioning a project:
Source: Retro commissioning Handbook for Facility Managers, Oregon Office of Energy, 2001, Prepared by PECI.

- Bring equipment to its proper operational state
- Reduce complaints
- Reduce energy and demand costs
- Increase equipment life
- Improve indoor air quality
- Increase tenant satisfaction
- Improve facility operation and maintenance
- Reduce staff time spent on emergency calls

Re-Commissioning

Once an established 'good as new' condition is achieved, it is common for building systems to back slide over time, with resulting deterioration of operating efficiency and/or comfort conditions to some degree. Repeating the commissioning process is intended to periodically pull the building or process back to the baseline state, thereby reducing losses from reduced efficiency and having a higher confidence level that intended performance is maintained. The documentation and test procedures created in the original commissioning activity should always be scrutinized for repeatability to accommodate re-commissioning over time. Common items requiring periodic attention include:

- Sequence of operation
- Setpoints and equipment run schedules
- Start-up and shutdown procedures
- Recalibration frequency
- Heat exchanger condition and service frequency
- Identifying needed repairs
- Annual energy use review

Ongoing Commissioning

The logical conclusion of repeat commissioning, this includes real time oversight of the systems in question, usually with data logging and complex software. With a well established baseline, immediate notification can occur for system operation that strays off course, allowing prompt correction and no loss of project intent. This application also helps guard against the natural tendency for backsliding or erosion of savings over time.

J— ENVELOPE TRADEOFFS— LIGHT HARVESTING, WINDOW TINTING

Many, many energy related choices involve give and take, and this includes envelope elements. Analysis is required to arrive at the "answer" in each case, because of the magnitude and variability of each give-and-take item in the mix. Give and takes are what integrated designs are all about. While certainly the most elegant of designs, they are also the most susceptible to value engineering, substitutions, and building use changes, so documenting the basis of design is helpful in sustaining the functions. Some tradeoffs are thermal for thermal, some are thermal for light, and some are influenced by cost of different energy sources as well. In all cases, the availability of alternative energy sources or waste heat can help considerably.

Many thermal tradeoffs related to envelope are climate dependent. In cold areas, the reduced cooling load in the relatively short cooling season will be overshadowed by the lost opportunity for passive solar heating assist during day lit hours of heating season. This is underscored by energy code recommendations for residential glazing properties, where envelope loads are the dominant factor influencing HVAC energy use; the more prevalent the cooling burden (presumably from sun loads) the more tinting is recommended and the more prevalent the heating burden—colder climates—the less tinting and the more U-value is required. This is akin to deciding whether to pitch one's tent in the shade or in the open.

The occurrence of beneficial light on an envelope section is intermittent, while the choice of building materials and insulating properties is fixed. Within limits, the overall wall/roof U-value can be stabilized by increasing insulation in non-glazed areas. This is a simple weighted-average calculation, but will reach diminishing returns with envelope sections with high percentages of glazing. For example, the amount of insulation needed to raise an overall wall R-value to R-19 when glazing is 30% of the wall area is impossible without high performance glazing. See also **Chapter 17 Composite U-Values for Envelope Evaluation.**

The residential model assumes most of the winter heat gain can be utilized by soaking into the house as a free supplement. However the residential model does not fully overlay onto the commercial environment , further complicating the choices. Consider a large cube-shaped building with interior and exterior zones and significant glazing in a moderate climate. Energy modeling may suggest a large surplus of beneficial heating, but unless care is taken in the model the benefits may be overstated. With

Baseline

20%	% glazing		
80%	% wall		
2.5	R-glazing	U-glazing	0.40
19	R-wall	U-wall	0.053
8.2	U-overall	U-Total	0.12

Forcing overall wall to R-19

20%	% glazing		
80%	% wall		
2.5	R-glazing	U-glazing	0.40
100	R-wall	U-wall	0.010
11.4	U-overall	U-Total	0.09

Futile without special glazing

0.2	% glazing		
0.8	% wall		
2.5	R-glazing	U-glazing	0.4
1,000,000	R-wall	U-wall	0.0000010
12.5	U-overall	U-Total	0.0800008

With high performance glazing

0.2	% glazing		
0.8	% wall		
7.0	R-glazing	U-glazing	0.1428571
33	R-wall	U-wall	0.0300752
19	U-overall	U-Total	0.0526316

Figure 24-J1. Sample Envelope Trade-off Calculation

the door closed to the perimeter office, what often happens in sunlit hours of cool weather is that particular room becomes overheated, invoking a call for cooling in the zone, either from an economizer or mechanical cooling. The cooling system serves the comfort call and sweeps the offending heat away even as an area on the opposite side of the building is calling for heating—in this example, much of the potential benefit is not realized. Obviously the same cube shape building with a single central open area would receive a greater portion of the benefit in this example. Thus, the building geometry, HVAC zoning , and HVAC system ability to distribute or store the energy windfall is as important as how much and when it occurs. Further, the temperature requirements for a house are different than in a building—there are people in the house at all times, while most commercial buildings are closed at night and the HVAC turned down or off. For example, if a building cools off to 50 degF overnight even without a skylight, it will simply cool off to that temperature sooner with the skylight and the rate of heat loss below that temperature is slowed proportionally to the (now reduced) differential temperature. These items are all definable with effort but easily overlooked from simplistic models. Most HVAC design approaches provide heating and cooling capacity for worst case and provide comfort on a per-zone basis, and are not well suited to storing or distributing dynamic envelope heat/cool influences.

In the case of daylight harvesting, additional variables are introduced. The cooling impact (plus or minus) effect is balanced by an increased solar load and interior cooling load may or may not be reduced. Ideally, just the visible light would be admitted and certain glazing coating properties can be specified to minimize the effect.

Some heat-give and take generalities for commercial buildings:

- Average or annual solar heating potential will not be realized in internal areas of the building when there are interior/exterior partition and HVAC zones. This makes the commercial building (other than an atrium or warehouse) fundamentally different than a house.
- In climates dominated by cooling and strong sunlight, solar repelling technologies are effective and cooling benefits outweigh.
- In climates dominated by heating, any source of free heat is good. In these areas solar absorbing technologies are effective and heating benefits outweigh.
- In moderate climates, it depends and a one-size-fits-all approach is a project risk. The give-and-take is more closely balanced and require analysis. Different answers can be appropriate in the same city de-

Table 24-J1. Envelope Give and Take Examples

Envelope Give and Take Examples

	Energy Pro	Energy Con	Remarks
Skylights / clerestories	Daylight harvesting reduces energy use for natural light when switched off during periods of abundant free light. Note: systems that do not shut off lights in response to day lighting eliminate most of the potential savings	Each skylight is effectively a 'hole' in the roof insulation system, increasing heat loss, especially at night when there is no offsetting energy benefit. With free light comes heat and without intervention via glazing properties the avoided cooling load from less interior light energy expense can be replace, or more, with the sunlight.	Comparison of heat loss vs. reduced lighting energy is needed to decide. Light tube systems reduce the thermal loss if the curb and tube are sealed and well insulated. Very effective in buildings where energy use is dominated by lighting, such as retail, and where temperature control is not critical (warehouse).
Light shelves	Daylight harvesting reduces energy use for natural light. Design can leverage sun positions throughout the year, e.g. positioning for passive solar heating in winter	Over-heating is a common complaint in summer. Larger window expanse is a larger heat loss in winter. With free light comes heat and without intervention via glazing properties the avoided cooling load from less interior light energy expense can be replace, or more, with the sunlight.	Optimized systems also include light sensing equipment to control artificial light in conjunction with free light. Upper section of the window usually will have a different chosen value of visible light transmittance, while the section below the light shelf could include stronger tinting and interior shades. Over-lighting (glare) is also very possible during brightest hours of day.
Exterior shading	Reduces solar gain in summer. Design can leverage sun positions throughout the year, e.g. positioning for passive solar heating in winter	Reduces solar gain in winter.	Very effective in climates where building load is dominated by cooling load from solar gain.
Window tinting and film	Reduces solar gain in summer.	Reduces solar gain in winter.	These are effectively solar screens that are non-adjustable. Obviously, optimization is easier with the ability to open a blind or close it when there is an advantage to do so.
Cool roofs and light/dark colored walls	Reduces heat gain in summer. Reflective roofs reduce passive pre-heating of outside air ventilation loads, reducing cooling burden.	Reduces heat gain in winter. Reflective roofs reduce passive pre-heating of outside air ventilation loads, increasing heating burden.	Additional "heat island" environmental building in dense urban areas. Effect of cool roof to the energy use within a particular building is strongly dependent on the roof insulation. Additional tangible benefit to building HVAC comes from the heating of the ambient air that package rooftop units live in, impacting condensing temperatures and cooling efficiency. Outside air ventilation loads are also influenced, but oppositely depending on season.

pending on building geometry and siting, interior zoning, and use.

- Light-admitting elements with manual or automatic controls to open/close them will increase overall annual utilization of the greatly enhance the ability to harvest and manipulate the energy to advantage.
- The magnitude influence of solar heating vs. thermal losses through 'holes' in the insulation is proportional to the percent of glazing.
- When ambient daylight is characteristically overcast, solar overheating concerns are greatly reduced.

K—HVAC OVERLAPPING HEATING AND COOLING

Source: Simultaneous Heating and Cooling—the HVAC Blight, paper presented at WEEC (World Energy Engineering Congress), Washington DC, 2008, Association of Energy Engineers.

SOME SOURCES AND REMEDIES OF HVAC HEAT–COOL OVERLAP

Notes for Tables 1- 5

1. Safety for people and property, and preventing equipment damage always has higher priority than energy savings. Consider all aspects of any system change.
2. Reference to resetting from "zone demand" refers to finding the highest call for heating or cooling, e.g. the 0-100% demand based on how far off set point it is. This is not the same as resetting from zone temperature, which would only be equivalent if all zone temperature setpoints were identical. This is sometimes referred to as "zone of greatest demand," "most open valve," etc. Care must be taken with reset schedules based on demand since one zone with a very high demand (such as a thermostat jammed to its lowest setting) will drive the reset routine inappropriately. Various methods are possible to inhibit this wild-card effect, such as limiting user adjustments to +/- 2 degrees, omitting the two highest and lowest readings, etc. Averaging allows the least disruption from wild cards, but can sacrifice comfort unless individual uses are very close to building averages.
3. VAV conversions require close attention to building air balance, pressurization, ventilation standards, and are generally an engineered solution. Adding VFDs to standard motors can cause premature motor failures. Systems with electric reheat coils may require higher minimums than otherwise desired, to prevent overheating.
4. Variable water flow conversions require close attention to chiller/boiler manufacturer requirements, air control and sediment (which are velocity dependent), and control valves. Adding VFDs to standard motors can cause premature motor failures. Simply capping off the third port of a 3-way control valve and calling it a 2-way control valve can introduce actuator close-off problems.
5. Altering perimeter heating systems is an integrated design task that must consider heat loss rates at the perimeter. For example, large sections of single pane glass may

not achieve comfort with any system other than a baseboard, unless the glazing is also changed.

6. Alterations of constant volume air flow systems to VAV must consider the effects upon the cooling and heating equipment from reduced air flows, especially unitary refrigeration equipment.
7. The terms thermostat and temperature sensor are used interchangeably when referring to zone level temperatures, for convenience in writing.

Table 24-K1. Sources and Remedies of Heat/Cool Overlap

Table 1 - CENTRAL SYSTEMS

Item	Sources of Waste	Remedy
CAV Reheat	For zones calling for heat, the SA must first be heated to room temperature before any room heating can occur. The lower the SA temperature, the higher the reheat penalty. In cooling mode, constant fan energy means constant heat; at part cooling load the fan heat is disproportionate.	Reset supply air temperature from zone demand (best) or from OA temperature. VAV conversion. This provides fan energy proportional to cooling load, and also reduces the magnitude of the reheat penalty, by not allowing heating until the zone air flow is at minimum.
VAV Reheat	For zones calling for heat, the SA must first be heated to room temperature before any room heating can occur. The lower the SA temperature, the higher the reheat penalty.	Reset supply air temperature from zone demand (best) or from OA temperature.
Multi-Zone (Blending) **and** **Double-Duct (Blending)**	Provides automatic heating and cooling in every case except maximum cooling or maximum heating. The two system types are essentially the same, except the blending function for the double duct occurs remotely with blending boxes instead of at the air handler. Blow-through fan blast upon coil faces creates turbulence and eddy currents that violate the separation of cold and hot air streams. Temperature measurements have shown thermal mixing at the top of the cold coil and the bottom of the hot coil from turbulence. For Multi-Zone system, constant volume nature requires hot deck and cold deck dampers to be linked; while linked, one opens as the other closed. For most systems, a single fan is used and the mixed air section is shared by both the warm and cool air streams. This forces compromises that limit the benefits of air-side economizer and supply air reset. For best heating efficiency in winter, OA would be at minimum, but for best cooling efficiency the economizer would be active. This usually results in a constant 55 or 60 degree SA temperature in winter and a 15-20 degF reheat burden on the hot air stream.	Reset hot deck and cold deck temperature from zone demand (best) or from OA temperature. Same for Double-Duct hot *duct* and cold *duct* temperatures Ideally there would be two fans or draw through to prevent this. A sheet metal divider plate half the height of the coil extending toward the fan will help. Perforated baffle well in front of the coils (if there is room) will even the air flow distribution at the expense of additional pressure drop. VAV air handler conversion and split the linkages with independent control actuators. Allow hot deck damper to open only after cold deck damper is at minimum position. Where practical, especially larger systems – and most applicable to Double-Duct systems: Split the air streams and provide two fans: one for the hot air stream / one for the cold air stream. Assign the mixed air section to the cooling side and allow air-side economizer to optimize cooling savings and also to provide ventilation. The heating air stream becomes entirely re-circulating, and the inherent reheat penalty goes away. Sometimes it is easier to create a new air handler for the warm air and blank off part of the Double-Duct casing than to field modify.

Table 24-K1. Sources and Remedies of Heat/Cool Overlap (*Cont'd*)

Table 1 - CENTRAL SYSTEMS (cont'd)

Item	Sources of Waste	Remedy
Constant Volume Cooling Air Flow **Also** **Constant Flow Chilled Water Pumping**	Constant pump/fan energy means constant heat; at part cooling load the fan heat is disproportionate. In heating systems or heating modes, the heat loss is not a significant issue since the heat released from motor horsepower is beneficial heat, delivered at the cost of electric resistance heating. Savings are then the differential between electric resistance heating and other available heat sources.	VAV / variable flow pumping conversion so flow rate and energy transport is load following.

Table 2 - TERMINAL UNITS

Item	Sources of Waste	Remedy
Double-Duct Mixing Box	Constant volume nature requires hot duct and cold duct blending box dampers to be linked; while linked, one opens as the other closed.	In conjunction with VAV air handler conversion and Hot Duct/Cold Duct reset control: Split the blending box linkages with independent control actuators for hot and cold ducts. Allow hot duct damper to open only after cold duct damper is at minimum position.
VAV Box "Heating CFM" Settings	Single Path VAV systems inherently have some reheat, due to the ventilation air and the cooling minimum setting. VAV box "Heating CFM" settings are seldom specified differently from the cooling minimum by design engineers, but are sometimes adjusted upwards by controls suppliers or building operators for various reasons; always increasing energy use in the process.	Ideally, the outside air would be provided independently of the cooling/heating air and the VAV box minimums would be set to zero. Economics often prohibit the use of this optimal system. Verify that "Heating CFM" air flow values are no higher than "Cooling Minimum CFM" values for each VAV box.
Baseboard Heat Not Sequenced with Air Terminal	Baseboard heating with independent thermostat control sharing a zone with an air system creates conflicting control action. Depending upon the settings, one or the other does too much heating or cooling the other compensates. Either way it s overlap.	Sequence the baseboard heat with the terminal unit, as either first or second stage heating, and integrate the baseboard controller "call for heating" with air system SA reset routines.
Baseboard Heat Zones Crossing VAV Cooling Zones	When VAV air system zones physically do not match the baseboard zones (as in after an interior alteration or tenant improvement), conflicting control action will result. This happens when one baseboard or fin tube adds heat to multiple VAV air zones. The portion of the baseboard linked with the VAV thermostat and within its area of air distribution works fine, but the section of fin tube in a foreign VAV air zone is the issue. When it needs heat and doesn't get it, there is a comfort complaint; when it gets heat and didn't need it, the VAV in that area will compensate by increasing cooling air flow and energy.	Re-segment the baseboard heat to match VAV box coverage, or add/increase reheat coil capacity in the VAV boxes and eliminate the baseboard system.

Table 24-K1. Sources and Remedies of Heat/Cool Overlap (*Cont'd*)

Table 3 - CONTROLS

Item	Sources of Waste	Remedy
Boilers running in summer	The boiler (assume hot water for this paper) is not the source of overlap, but is the enabler for the other sources of overlap.	Turn off the boiler when possible. Whenever the heating water flow stops, the overlapping waste from inadvertent heating during cooling season are a mute point.
Chillers or flat plate running in winter	The chiller is not the source of overlap, but is the enabler for the other sources of overlap. Some facilities operate a flat plate in winter. While the chilled water is now produced at a discount rate (chiller off, auxiliaries on) the negating effect of the heating water still exists.	Turn off the chiller or flat plate when possible. Whenever the chilled water flow stops, the overlapping waste from inadvertent cooling during heating season are a mute point.
Lack of Dead Band Between Sequential Heat/Cool Equipment	As long as the heating and cooling valves are not open at the same time, there is no overlap. The dead band measure forms a safety zone between the two to provide assurance it won't happen. Controls drift, controls have operating ranges, etc. so the space in between allows for these practical realities.	Adjust controls for a 5-degree dead band between sequential heating and cooling equipment, such as between a preheat coil and a mixing damper control set point or between a preheat coil and cooling coil.
Controlling Methods Can Allow Overlap of Sequential Heat/Cool Equipment	Applies to air handlers, and also to terminal units with heat / cool capability. Example is preheat, mixing damper, and cooling coil all in a row. Using a shared signal for multiple end devices has the advantage of simplicity but depends upon accurate and repeatable response from the actuators Using independent signals for each end device eliminates the dependency on critical actuator adjustments. Each device has its own setting and control loop. However, if the independent settings are changed overlap can be created without knowing it. With PI control, there can be significant time lag to return to set point and overlap occurs when near a transition between heating and cooling operations unless a wide dead band is maintained between the settings.	Arrange control software that sequences several components under a master control routine that will keep them proportionally spaced apart. This is sometimes referred to as a 'receiver controller' routine, since it was commonly employed that way using pneumatics. Or Create additional code that allows manual adjustment of only one of the settings, and the others are calculated – maintaining the space (dead band) between them. Or Create additional code that allows only one of the sequential independent controller outputs to be "non-zero" at a time. This approach of mutual exclusive heat/cool control, combined with transition delays, is an active approach to the problem. In all cases, extra temperature sensors downstream of each heating or cooling device allows monitoring and alarms if heating and cooling occur sequentially

Table 24-K1. Sources and Remedies of Heat/Cool Overlap (*Cont'd*)

Table 3 – CONTROLS (cont'd)

Item	**Sources of Waste**	**Remedy**
Air Handler and Terminal Unit Controls Not Coordinated	Independent control loops upstream and downstream, but part of the same duct system, each with their own set point, are the source of waste. Each acts without regard to the other This is very common with VAV Reheat and CAV Reheat systems.	Reset supply air temperature based on zone demand. For VAV air handlers with reheat VAV boxes and for CAV Reheat systems, the goal is to provide air that is cool enough for any zone needing cooling, but just cool enough and no more – thereby reducing reheat.
Preheated Air Stream Common to Multiple Systems	Source of waste is lack of upstream-downstream set point coordination.	Instead of a fixed set point, adjust controls to reset the preheated temperature based on demand. Normally, preheating just above freezing (to 45 degF) will eliminate the un-necessary re-cooling by a mixing damper or cooling coil.
Preheat coil operation while mixing damper is past minimum setting	Cooling action of the mixing dampers negates the heating energy input just upstream.	Modify controls to make cooling and heating operations within the air handler mutually exclusive, including economizer cooling.

Table 4 - VALVES, DAMPERS AND ACTUATORS

Item	**Sources of Waste**	**Remedy**
Actuator Insufficient Close-Off Rating	Without sufficient available actuator power, system pressure can force open a control valve while in the closed position. This is especially true of globe valves. This is a common 'gotcha' for variable flow pumping conversions where, instead of installing new 2-way valves, the bypass leg of the existing 3-way valve is simply capped off. The actuator of the 3-way valve is normally sized for just diverting the flow and not stopping it, and consequently the actuator cannot provide proper close-off. Often this is detected by a whistling at the valve.	Actuator and valve/damper assemblies have published close off ratings. Verify these are well in excess of (e.g. 50% higher than) expected pressures. For two way valves, this is often full system pressure. If unable to hold closed against system pressure, replace the actuator with a properly sized one.
Electronic Actuator Residual Close-Off Force Feature	For positive seating, additional force is required once the device is at its closed position. Pneumatic controls and actuators do this very well: e.g. an 8-13 psi spring range with a 4psig signal has a residual force applied beyond the "just closed" point. Good quality electronic actuators have a mechanism to allow "over-travel" or utilize "force-sensing" circuitry to create this residual force. *Some electronic actuators do not have this!*	Replace the actuator if it will not tightly close, making sure the new one has the residual force feature.

Table 24-K1. Sources and Remedies of Heat/Cool Overlap (*Cont'd*)

Table 4 - VALVES, DAMPERS AND ACTUATORS (cont'd)

Item	Sources of Waste	Remedy
Electronic Actuator Adjustment	Valve not seated, allowing leak-by internally even while being commanded fully closed. Usually from improper set-up, or the zero adjustment is improper or has drifted. When coupled to the valve or damper for the first time, the technician strokes the device and sets the zero and span adjustment of the actuator so it knows where full open and closed is. If care is not taken to adjust for residual close-off force, the errant adjustment may allow the leak by for the life of the valve or damper. This is more of an issue with valves than dampers since dampers don't fully close anyway.	Adjust / replace the actuator
Pneumatic Actuator Adjustment	Some pneumatic (air-powered) actuators have adjustable spring pre-loading. Others are made adjustable through pilot positioners with their own zero and span adjustments. If not properly adjusted, valves or dampers may not close. More commonly, sequenced devices may overlap.	With a variable pressure air source or squeeze bulb, slowly modulate the pressure to the actuator(s) and observe the opening and closing points. If too close to zero or max pressure at either end, or if overlapping of sequenced devices, adjust the spring ranges or replace the springs. Ultimately, there should be several psi of pressure above and below the travel end points and a gap (dead band) between any sequenced devices sharing the same signal.
Pneumatic Transducer Adjustment	For "hybrid" DDC controls – formerly pneumatic that now have digital control signals interfaced to existing pneumatic end devices – the transducer is the linking component between digital and pneumatic technologies. These have a zero and span which correspond to the minimum and maximum travel of the device and minimum and maximum command of the controller. If improperly set, or if they have drifted, lack of closure force or overlap can result. For example: Adjustments that simply align a digital "no call for heating" command value to the transducer with a pneumatic output that brings the device to a "just closed" position has not allowed for residual tight seating and will very likely leak by.	With the DDC signal at a value to fully close the final device (valve or damper), the transducer output should be a pressure several psi beyond the actual "just closed" point to create the residual force for tight seating. Adjust as required. When a device is told to be "full closed", programming to add additional pressure from the transducer (e.g. 0 or full pressure) is an excellent way to assure maximum available close-off force is applied.
Leaking Multi-Zone or Dual-Duct Blending Dampers	Multi-Zone blending dampers are generally un-accessible after installation of ductwork, and it is a fair assumption that they leak after 10 years of service. Dual duct blending terminal boxes usually have access doors and the dampers can be visually inspected.	Field test by commanding the dampers full closed to heating (full open to cooling). Compare the downstream zone temperature to the upstream cold duct / cold deck temperature. If it is higher, the hot damper is leaking. Repeat for the hot damper test. Remedy is usually to replace the dampers. For Multi-Zone this requires removal of the zone ductwork. For Dual-Duct systems, replacement of the terminal may be more cost effective.

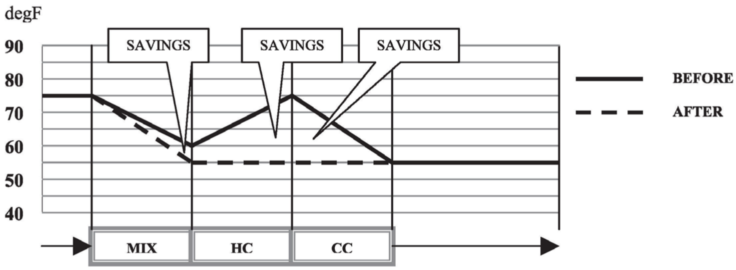
Table 24-K1. Sources and Remedies of Heat/Cool Overlap (*Cont'd*)

Table 4 - VALVES, DAMPERS AND ACTUATORS (cont'd)

Item	Sources of Waste	Remedy
Leaking Control Valves or Dampers	Conventional globe-style control valves are metal seated. Debris or wear can damage the seats, preventing full close-off. Large sections of dampers linked together, old dampers, and manual dampers used as motorized dampers will not close securely. These are seldom used to regulate heating quantities, but can admit excessive outside air when "closed", creating an overlap condition and undue burden on the heating system.	Standard test for valves is to command them closed and verify no heating / cooling is occurring downstream through temperature measurements, allowing some time for residual heat to dissipate. The most reliable measurement for air systems is the air temperature rise/drop through the coil. For un-insulated piping of sufficient length, this can be detected by pipe temperature as well. Damper leakage can normally be verified visually and with air temperature measurements, with the device commanded closed. It is good practice to attempt to move the damper blades with your hands, since sometimes they have become free from the axles and are free wheeling. If the device won't move freely or won't close, and actuator issues are ruled out, replace the valve or damper.

Table 5 - OTHER

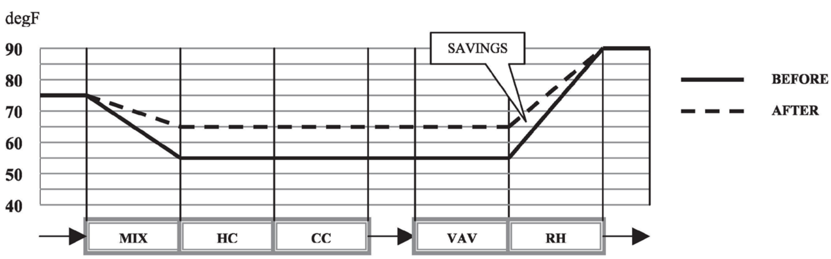
Item	Sources of Waste	Remedy
Fan heat from excessive static pressure	During cooling mode any heat from fan or pump transport equipment is parasitic and adds to cooling load.	Lower duct static pressure as much as possible, with the goal of providing enough, but just enough pressure. Ideally, this is reset from VAV box demand with the most-open VAV box damper 90% open indicating "just enough" pressure.
Improper zoning and addition of duct heaters or space heaters	When large single zone equipment is used, a common mistake is to cross zones. For example if one unit serves both east and west exposures with the thermostat in the west zone, it will be over-cooled in the east zone in the afternoon. Often the solution to this is to add duct heaters or space heaters in the over-cooling areas.	Ideally, correct the zoning root cause with additional equipment and split the ducts by exposure. Alternately, provide self-compensating diffusers that will close off and reduce the over-cooling air flows.
Over-cooling of VAV interior zones	Without significant supply air reset, it is predictable that interior zones of a VAV system will be over-cooled. Unless building operators quietly adjust interior VAV boxes to zero minimums and ignore the ventilation standards, people will first complain and then will bring in space heaters.	Ideally, interior and exterior duct systems will be separate, allowing greater tempering of air to interior zones that have no heating provisions. Where interior and exterior zones share a common supply air temperature, the reset is a compromise between interior zones that want it warmer and some perimeter zones that need it cooler. Any reset is better than no reset at all.
Fan heat from excessive static pressure	During cooling mode any heat from fan or pump transport equipment is parasitic and adds to cooling load.	Lower duct static pressure as much as possible, with the goal of providing enough, but just enough pressure. Ideally, this is reset from VAV box demand with the most-open VAV box damper 90% open indicating "just enough" pressure.
Overlap from adjacent zones of control	For open plan offices with multiple VAV zones of control, allowing one zone to be set significantly different than a neighboring zone will create interaction between zones and waste from overlap.	Limit user adjustment to +/- 2 degF. Combined with a 4 or 5 degree dead band between zone heating and cooling operations, overlap of this type will be minimized.



Mixed Air Single Path

- Air Handler heat / cool settings not coordinated.
- Economizer savings missed.
- Un-necessary heating and counteracting cooling.

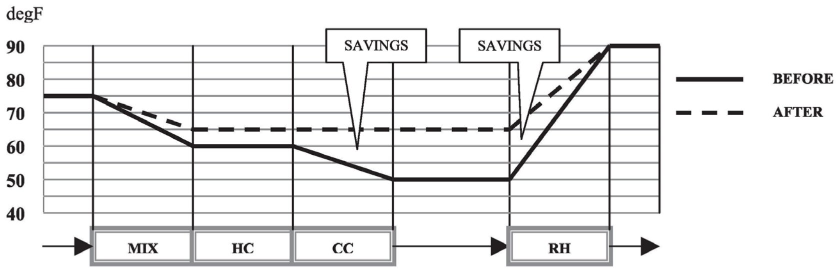
Figure 24-K1. Overlap Example—Mixed Air Single Path



VAV with Perimeter Reheat

- Upstream / Downstream settings not coordinated.
- The year-round constant 55 requirement increases building energy use in winter.
- When heat is required, elevating the supply air temperature set point reduces reheat energy.
- In this example the over-cooling is free but the counteracting reheat is not.

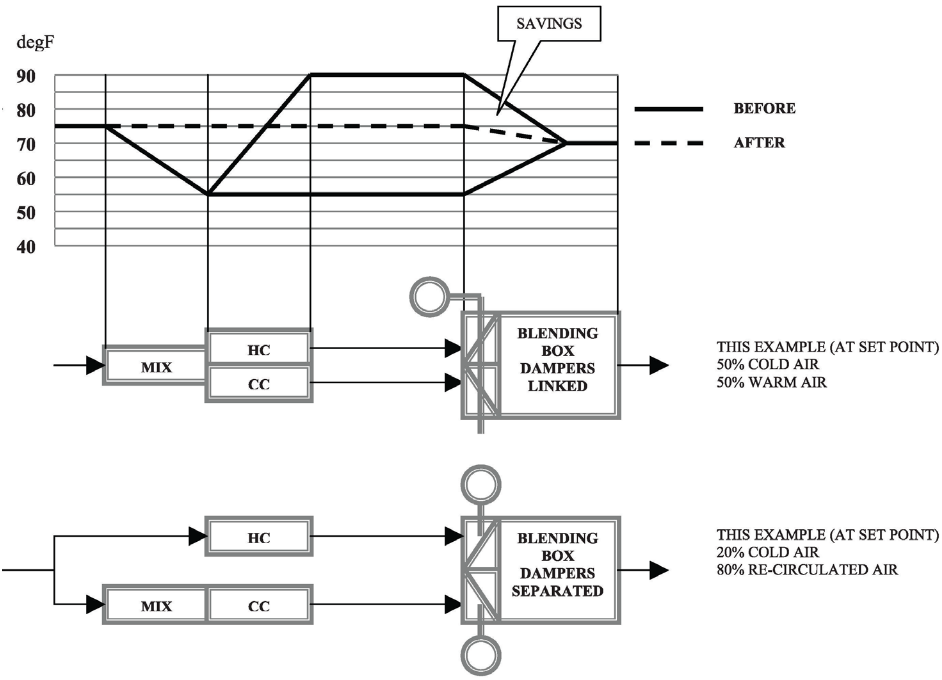
Figure 24-K2. Overlap Example—VAV Perimeter Reheat



Constant Volume Reheat

- Upstream / Downstream settings not coordinated.
- The year-round constant 55 requirement increases building energy use in winter. When heat is required, elevating the supply air temperature set point reduces reheat energy.
- Mixed air control setting not coordinated with cooling coil setting.
- Mechanical cooling not locked out (it was 30 degrees outside).
- Additional reheat energy to counteract over-cooling.

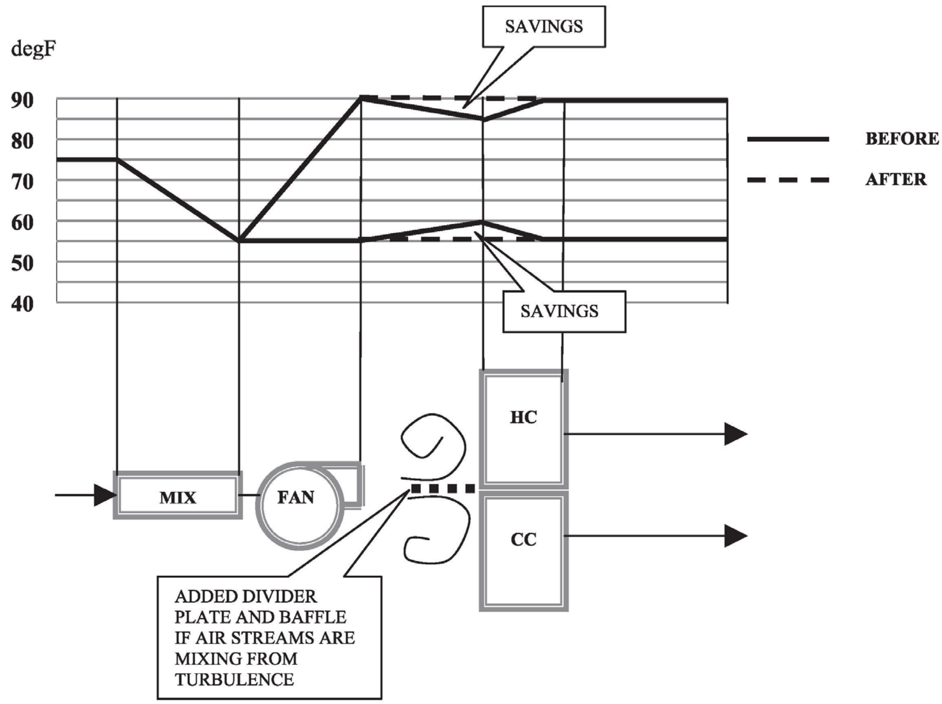
Figure 24-K3. Overlap Example—Constant Volume Reheat



Double Duct

- Common AHU path creates compromise in energy use.
- Standard blending box links both dampers and creates continuous overlap of heat-cool except at max cooling / max heating conditions.
- System change splits the AHU paths and splits the blending box dampers.
- Principle is the same for Multi Zone systems.

Figure 24-K4. Overlap Example—Double Duct



Blow Through Mixing

- Turbulence at coil faces 'bounces' cold air onto heating coil and warm air onto cooling coil, false loading both coils.
- Two draw through fans is the ideal solution.
- A sheet metal divider plate and perforated baffle also helps.
- Principle is the same for Multi Zone systems.

Figure 24-K5. Overlap Example—Multizone Blow Through Mixing

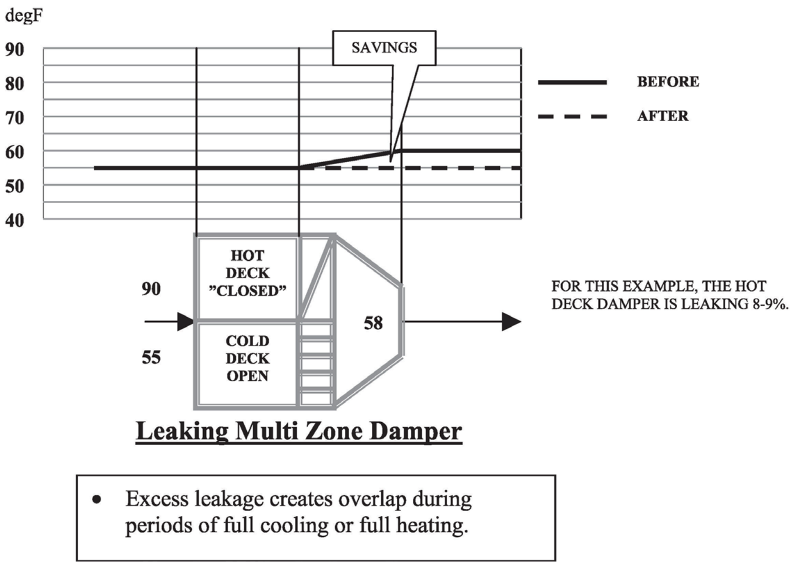


Figure 24-K6. Overlap Example—Leaking Multizone Damper

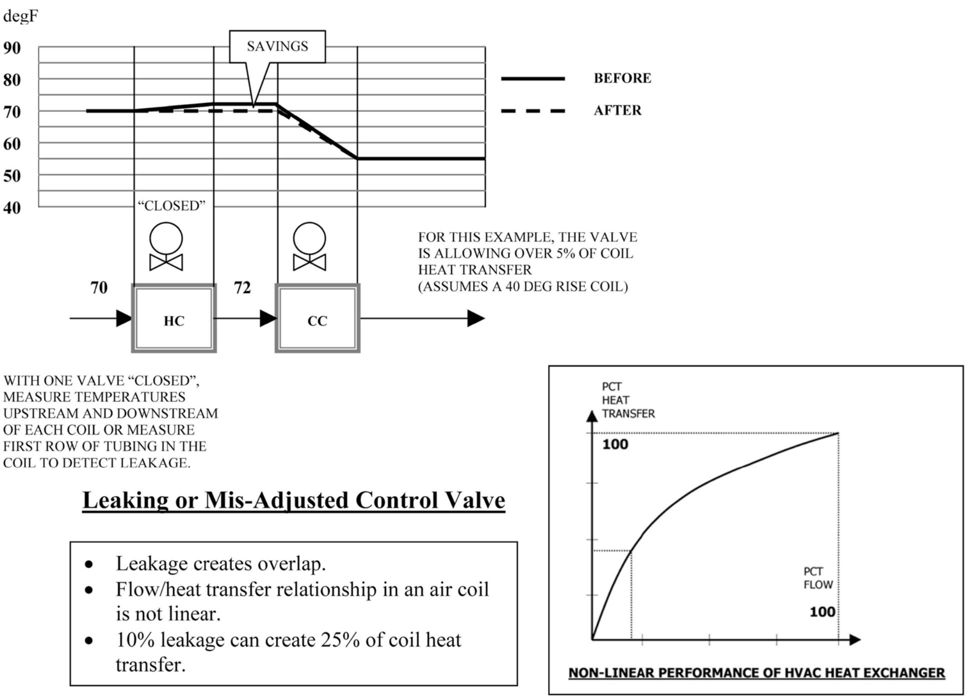


Figure 24-K7. Overlap Example—Leaking Control Valve

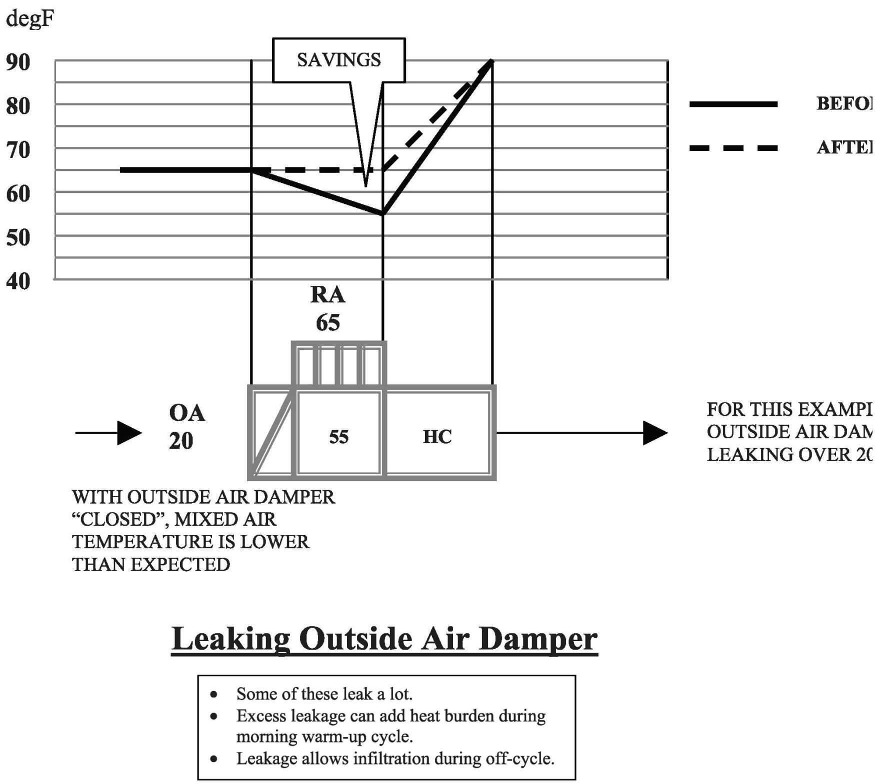


Figure 24-K8. Overlap Example—Leaking Outside Air Damper

L—PART LOAD HVAC EFFICIENCY

Source: Part Load HVAC Efficiency, Energy Engineering Journal, Vol. 107, No. 3, 2010.

Additional Items Where Part Load Performance has Energy Implications

- Capacity Control with Inlet Vanes, Discharge Dampers, Slide Valves, Hot Gas Bypass
- Magnetic Bearings/Oil-Free Water Chilling Equipment
- Boiler Air-Fuel Mixture Control at Part Load to Reduce Losses from Excess Combustion Air

- Stack Dampers—Thermal Siphoning from Flue Vents
- Low Ambient Cooling Equipment Operation—False Loading
- Parallel Boiler, Chiller Flow Isolation and Staging
- Permanent Magnet AC and Brushless DC motors
- Chilled Water System Low Delta T and Part Load Pumping Costs
- Centrifugal Compressor Maps
- Demand Controlled Ventilation
- Separating Interior and Perimeter VAV Air Systems
- Overlapping Heating and Cooling
- Transformer Part Load Efficiency
- Plating Rectifiers at Part Load
- Motor Efficiency and Power Factor at Part Load

Common Themes in Identifying and Approaching Part Load Energy Efficiency

- Standby losses are small at full load but become an increasingly large percentage of energy use at part load.
- Right size equipment uses less energy because it stays in its efficient range longer.
- In most cases, efficiency drops off rapidly for equipment loaded below 50% of capacity.
- It is usually more efficient to run one machine at full load than several at low load.
- Systems with a variable load and a constant fluid flow represent an opportunity for part load energy improvement.
- The operating cost of the auxiliaries, especially the pumping, drag down the overall system efficiency at part load.
- Quantifying part load energy consumption requires knowing the number of hours for each value of load and the coincident system efficiency.

Actual vs. Ideal

Figure 24-L1. In most systems energy use does not track the load profile except at the top end. The goal of part load design is for the two to track as closely as possible. Many of the characteristic curves within this article follow a pattern: below about 50% load the efficiencies begin to drop rapidly. The reasons vary, but the results are similar. Keeping this tendency in mind is useful as HVAC designs are evaluated.

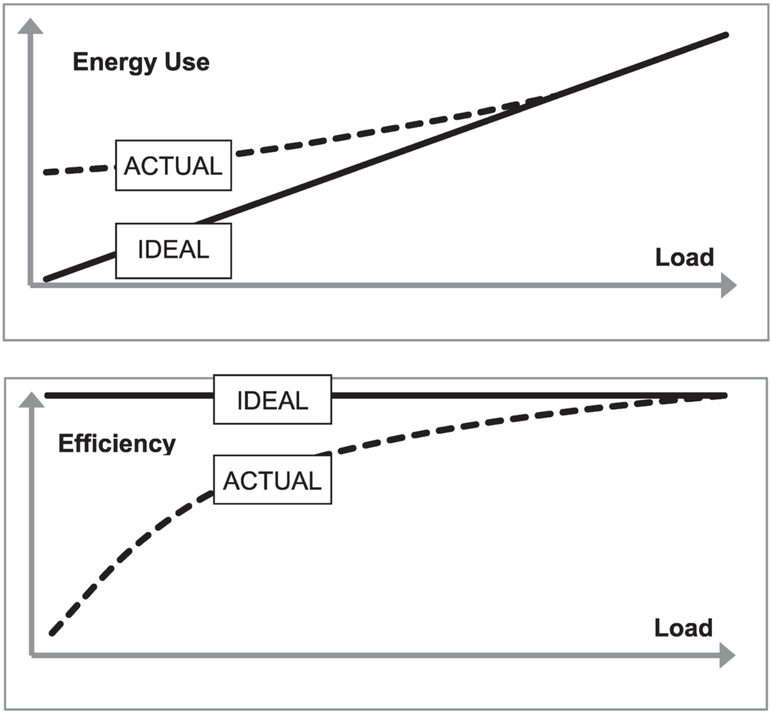


Figure 24-L1. Characteristic Ideal and Actual Energy Use and Efficiency Curves

Calculating Annual Energy Use

Quantifying part load energy use requires a load profile (how many hours at what percent load) coupled with the associated system energy efficiency at that moment. The concept of "bins" is useful—e.g. the number of hours at 80 percent load defines the 80 percent bin in the load profile.

Using HVAC cooling as an example, the energy use for each load profile bin becomes the power consumption at that condition multiplied by the number of hours it occurs:

(IP) **Tons * kW/ton * Hours = kWh**
(SI) **kWSI cooling * 1/COP * Hours = kWh**

Cooling energy conversions:
[IP: COP cooling efficiency = 3.517/(kW/ton)]
[SI: kW cooling load = tons * 3.517]

Estimating part load energy consumption requires knowing the efficiency for the active equipment at each value of load. In some cases a simplified approach can be used. Two simplified examples are provided.

In complex systems, simplified approaches may not work and estimating will be difficult even with modeling software, including cases where:

- The load profile is variable but not coincident with weather, production rate, etc., such as sporadic use of large equipment.
- Equipment efficiencies are variable but independent of load, such as a mixture of available equipment in a plant.

Example Factors Affecting a Commercial Building Load Profile

Example 1: Weather-Dependent Systems.
Ref Figure 24-L2. In a commercial office building, the internal activities are fairly constant and the overall cooling load follows weather and peaks on hot days. From **Figure 24-L2**, there are 400 hours at 60% load, and the system efficiency at that load is 1.25 kW/ton. If the maximum load of this system is 200 tons, the energy use of this load bin is (200

Table 24-L1.

Factor	Example Influential Items
Climate	• Patterns (Bins) of Dry/Wet Bulb Temperatures • Solar Incidence
Envelope	• % Glass and Shading • Insulation and Building Weight • Aspect Ratio (Surface Area to Volume) • Site Orientation • Leakage and Infiltration • Bottom line: extent to which envelope loads influence the overall heating and cooling load
Occupancy	• Occupied Days and Hours Per Week • Occupancy Times of Day • Unoccupied Periods • Areas of Continuous Operation
Internal Activity	• Interior Climate Conditions Maintained • Humidification / Dehumidification • Additional Ventilation for Process Exhaust • Primary Business Activity • Persistent Process Loads

tons)*(0.6)*(400 hours)*(1.25 kW / ton) = 60,000 kWh. The other load bins are evaluated in the same way to arrive at an annual total energy consumption value. By comparing different HVAC equipment efficiency profiles with a given load profile, energy consumption of design alternatives can be compared.

Note on **Figure 24-L2**: This example method is suitable for evaluating HVAC energy use when both the load profile and system efficiencies are weather dependent. In this case the load and equipment efficiency both vary directly with outdoor dry bulb temperature, e.g. for the hours of 100% load it will be hot outside (with efficiency x) and partial loads will occur in milder weather (at efficiency y). This is a similar assumption model for the seasonal efficiency rating of some air-cooled HVAC systems (SEER).

The method illustrated in **Figure 24-L2** cannot be used unless both load and efficiency can be correlated on weather. Do not use when:

- The load profile is weather independent, such as a data center.
- The equipment efficiency profile is weather independent, such as a water-cooled HVAC cooling unit with constant condenser water temperature.

Example 2: Production-Dependent Systems.

Refer to Figure 24-L3. When HVAC loading in a factory is a strong function of production rates, a similar approach can be taken by defining the hours at each load and the corresponding efficiency of the cooling system. This example is for process cooling energy use, but can be applied to other plant equipment (e.g. compressed air) if the efficiency at each load bin is predictable. This method is limited to applications where both the load and equipment efficiency are a strong function of plant capacity, and where the HVAC function is largely weather independent.

Protecting the Design

Design documents serve multiple purposes. They illustrate design intent, allow bidding and construction, and serve as training basis for building operators. They also can serve to bring order to the equipment substitution process.

Documentation of both full load and part load performance values establishes a benchmark for expected annual energy consumption, and requiring substitutions to have equal or better full load and part load efficiency ratings is good practice. Noting these performance parameters on the design drawing equipment schedules, along with corresponding conditions, will communicate the design intent and enable equipment suppliers to match performance for any proposed substitutions.

But this is not enough. Efficiency ratings speak to equipment at standardized test conditions. The HVAC System overall efficiency can be impacted by equipment changes even when the statistics appear equal and so careful review of substitutions is required.

Table 24-L2.
Impact of Equipment Substitution on Part Load Energy Efficiency

Designed	Substitution	Reason for Part Load Energy Use Increase
8:1 Burner	4:1 Burner	Increased cycling of boiler at low load
50%Lo Fire 2-Step furnace	66% 2-step furnace	Increased cycling of furnace at low load
4-Stages of DX Capacity	2-Stages of DX Capacity	Increased amount of run hours at lower loads
50 tons cooling capacity	60 tons of cooling capacity	Over sizing increases run hours at lower loads
VFD Screw Compressor	Slide Vane Capacity Control	Increased energy use at part load
7degF(3.8degC) CT approach	10degF(5.5degC) CT approach	Increased refrigeration power at all loads

DX Direct expansion
VFD Variable Speed Drive
CT Cooling Tower

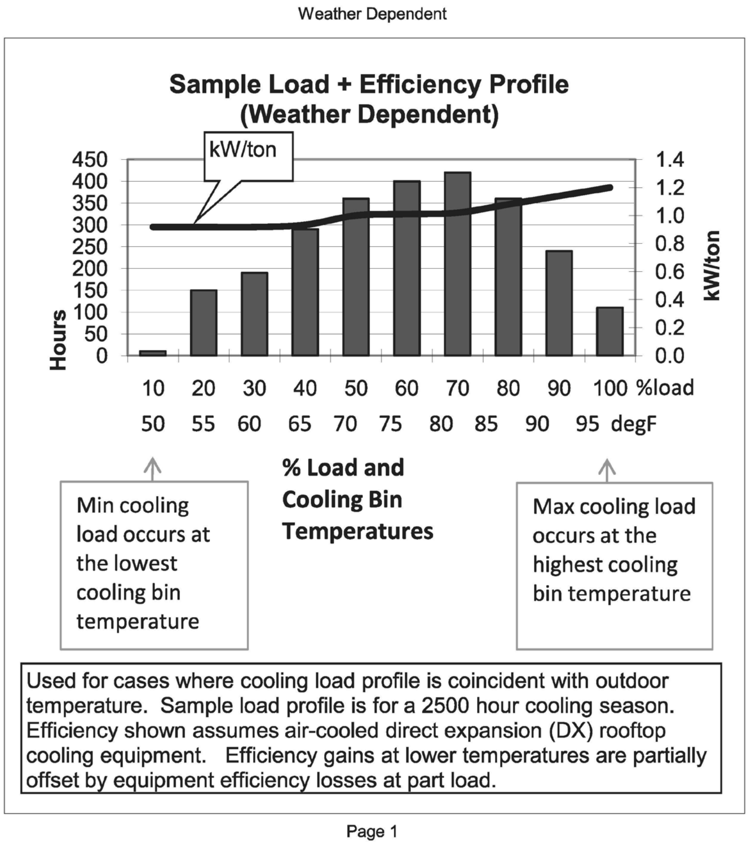


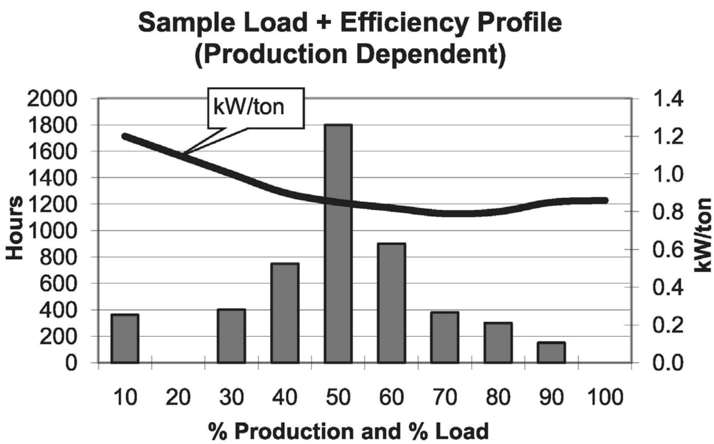
Figure 24-L2. Combining Load Profile with Efficiency Profile—Weather Dependent

Examples of HVAC Efficiency at Part Load

Note: Many of the charts that follow have been created to illustrate general patterns of part load performance found in a variety of HVAC systems. Where these generalized graphs are used, the assumptions are noted.

Equipment Right Sizing

No engineer wants to show up with a shortage of capacity. There is always some uncertainty in calculating maximum loads, and there may be



Used for cases where percent equipment load correlates to percent production. Efficiency profile assumes a water-cooled chiller with constant condenser temperature and constant flow pumps with 0.86 kW/ton overall efficiency at full load increasing to 1.2 kW/ton at 20% load.

	% Load	10	20	30	40	50	60	70	80	90	100
1 shift	Jan			50	75	150	75	10			
1 shift	Feb			50	75	150	75	10			
1 shift	Mar			50	75	150	75	10			
1 shift	Apr			50	75	150	75	10			
1 shift	May			50	75	150	75	10			
2 shifts	Jun				50	200	100	100	100	50	
2 shifts	Jul				50	200	100	100	100	50	
2 shifts	Aug				50	200	100	100	100	50	
1 shift	Sep			50	75	150	75	10			
1 shift	Oct			50	75	150	75	10			
1 shift	Nov			50	75	150	75	10			
closed	Dec	0	0	0	0	0	0	0	0	0	0
	total	0	0	400	750	1800	900	380	300	150	0

Figure 24-L3. Combining Load Profile with Efficiency Profile—Production Dependent

requests for reserve or future capacity. These all play into the decision for equipment sizing.

When equipment is oversized, it will operate at less than full load. The extent to which this happens determines the extent of the energy penalty associated with the choice. During part load operation, all equipment is effectively oversized. For on-off equipment, oversizing increases the off-cycle time and cycling losses at part load. For equipment with modulating capacity, oversizing at full load makes the inefficient portion of the curve show up sooner (**Figure 24-L4**).

Table 24-L3. Basic Strategies for Good HVAC Efficiency at Part Load

Item	Remedies
General	• Strive for energy reduction that is proportional to load reduction • Select equipment with highest available full load and part load efficiency • Recommend maintenance practices to sustain efficient operation • Establish values of maximum and minimum load and design for efficient operation at minimum load. Guard the design intent during value engineering and substitution processes • Avoid operating equipment below 50% capacity • Avoid systems with inherent poor part load efficiency • Avoid hot gas bypass and other false loading • Zone systems to avoid running large systems for small loads during partial occupancy • Right size equipment and avoid over sizing • Minimize energy transport and standby losses
Unitary Heating and Cooling equipment with fixed capacity	• Use multiple compressors controlled in stages in lieu of cycling; or use multiple smaller units
Heat Pumps	• Consider two-stage compressors to prevent excessive cycling
Compressors	• Multiple units or modular, to keep running equipment highly loaded • Variable speed in lieu of slide valve for screw compressors
Boilers	• Staging to keep load above 50% • Isolate and bypass equipment that is not running • Stack dampers for fired equipment • Parallel positioning (separate control of air/fuel) for modulating burners • Separate domestic water heating from space heating equipment
Chillers	• Staging to keep load above 50% • Isolate and bypass equipment that is not running
Large Central Heating and Cooling Plants	• Unequal size array of equipment • Jockey equipment for light load • Modular equipment arrays • Correct drooping delta T conditions
Fans / Pumps	• Staging to keep load above 50% • Evaluate system curve so operating point stays near best efficiency point with single or multiple units running • Trim impellers of constant speed pumps if excess throttling is required • Select pumps/fans slightly to the right of the BEP when multiple units serve a common pipe/duct
Motors	• Use VSDs instead of inlet vanes or throttling dampers/valves
Distribution Piping and Ductwork	• Move equipment closer to the load to reduce distribution losses • Insulate distribution pipes and ducts • Variable flow so distribution pumps/fans are load-following • Temperature reset in mild weather
Controls Optimization	• Monitor overall system output efficiency as well as individual units • Select best equipment to run from load curves or from pct. operating load • Coordinate upstream and downstream control settings • Reset from demand, such as "most open valve" • Prevent heat/cool overlap false loading • Prevent sequential pressurization/ dissipation false loading

Single Stage HVAC Equipment and Cycling

In its simplest form, part load conditions cause fixed capacity unitary heating and cooling equipment to cycle off and on, with the off-cycle increasing in proportion as the load becomes less. For unitary cooling equipment, cycling losses are mostly from inefficient compressor performance at low loads. The condenser and evaporator coils are effectively oversized at part load and partially offset the energy increase, but it is still a net loss. Annual losses of 5-10% have been shown in field studies for cycling packaged cooling equipment (Source 1). For furnaces, the losses during the off-cycle periods are small, and much of the heat remaining in the heat exchanger during the off-cycle will be picked up in the building air stream (Source 2).

Source 1. *Improving DOE-2's RESYS Routine: User Defined Functions to Provide More Accurate Part Load Energy Use and Humidity Predictions* (2000), Henderson, Hugh I. et al, Lawrence Berkeley National Laboratory (LBNL) Paper LBNL-46304.

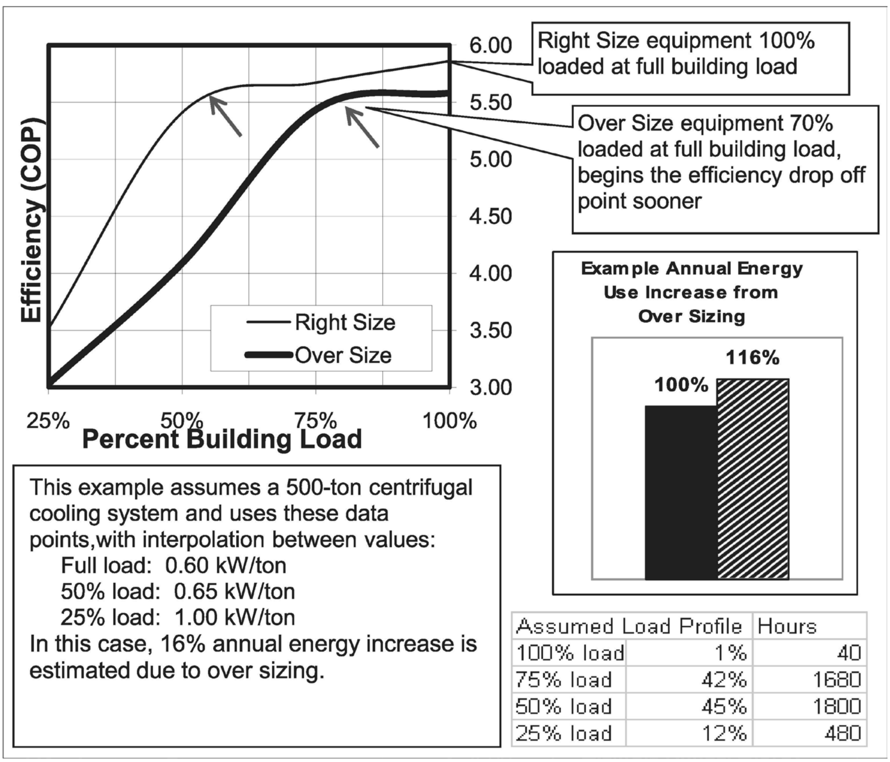


Figure 24-L4. Example Efficiency Loss from Oversized Cooling Equipment

Source 2. Screening Analysis for EPACT-Covered Commercial HVAC and Water-Heating Equipment (2000): Pacific Northwest National Laboratory (PNNL), for the United States Department of Energy (DOE).

With single stage unitary equipment, cycling will be proportional to load. Part load efficiency can be improved by using multi-stage equipment or multiple smaller units that are sequenced. Note that ventilation considerations must be integrated with capacity reduction increments where cycling equipment is used.

Motor, Fan, Pump, and Drive Losses

Motors, fans and pumps exhibit the same drooping characteristic as many other equipment items. As with other equipment, overhead losses that are a small percentage of full load energy use become a larger fraction at part load, reducing overall machine efficiency. In addition to the motor, part load losses affect driven loads and power transmission equipment.

See Chapter 15—Efficiencies of Centrifugal Fans/Pumps at Reduced Speed

Boiler Radiation Loss (Skin Loss)

Standby loss for hot equipment is a function of surface area, temperature difference, and insulation. At full load, these losses are a small percentage of total heat transfer (1-2%). At reduced load, the losses become a greater percentage of total energy use and overall efficiency drops. For example, if operated at 30% load, a packaged boiler rated at 80% efficiency and 1.5% of full load casing loss will operate at 75% thermal efficiency—a fuel use increase of 6.5%. Modular boilers produce energy savings by operating smaller machines more closely matched to the load.

Oversized operation also occurs when domestic water heating is coupled to a boiler with a 'side arm' auxiliary heat exchanger. In this application the boiler must run at part load in summer to satisfy a small heating demand when it would otherwise be turned off.

See Chapter 13—Boiler Standby Heat Loss (Boiler Skin Loss) at Full Load

Equipment Sequencing

Where multiple equipment sets are used for maximum load, choosing which one or combination to run at part load can influence energy use. Some machines are more efficient than others and would be a logical choice when they have sufficient capacity. In all combinations, control decisions that strive to keep the machines operating in their 'sweet spot' will optimize energy use.

Zoning for Partial Occupancy

When a building is fully occupied, many options for zoning will work similarly. But when only a portion of the building is occupied, some designs behave differently than others. When building usage is non-uniform, designs that require whole-building systems to run for a small portion of the load will provide the comfort, but at increased energy use. By contrast, package units can turn on only what is needed, such as individual rooms in a hotel. Duct zoning can also influence part load efficiency.

Ref. **Figure 24-L5**. Note that all three HVAC systems must run at reduced load in Method A, compared to one HVAC unit at full load in

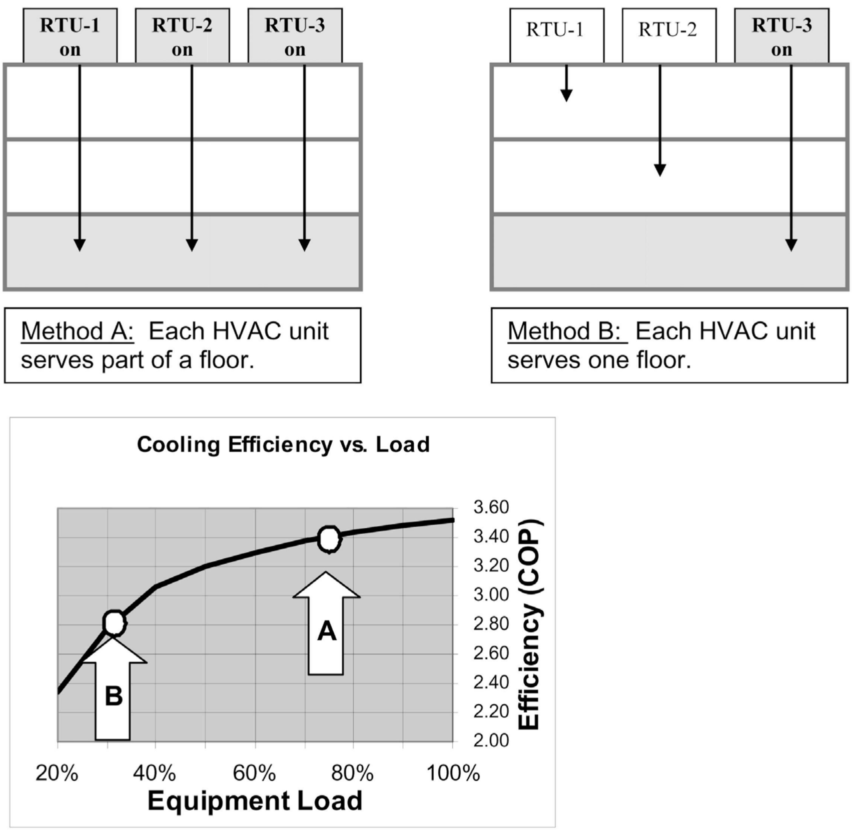


Figure 24-L5. Part Load Efficiency Improvements from Zoning

Curve shown is for standard HVAC rooftop compressors, with characteristic efficiency degradation from 100% at full load to 150% of kW/ton power requirement at 20% load (6). Values are for constant condenser temperature.

Method B. Besides low load/cycling losses, additional energy waste comes from conditioning other areas of the building needlessly. Designs that allow basic segmentation of the building can more efficiently accommodate the variable use. With little added cost, basic air/water zoning can be broken down by floor or quadrant.

Energy Transport Losses

There's the heating or cooling equipment, and then there's the pipe or duct delivering it to the point of use, complete with the insulation trying to keep the energy inside the conduit.

The concept of variable flow water and air systems is effective at reducing the transportation motive force overhead cost, since the energy expense for transmission follows the heating/cooling load (load-following) instead of being a constant expenditure of energy. Any system with a variable load and a constant fluid flow represents an opportunity for part load energy improvement. Designs that isolate and bypass off-line equipment from distribution loops reduce pumping energy as well. Any time there is a fluid being moved with no benefit from the work it represents a part load inefficiency that can be designed out.

Note that variable flow designs address pumping energy transport losses but do not address the thermal losses of the distribution system; these are mitigated through insulation. At constant temperature, the insulated duct/pipe losses are essentially constant at all velocities and flows; temperature resets mitigate but do not eliminate these losses.

The most effective way of reducing energy transport losses at full and part load is to locate the equipment closer to the point of use and design for low friction ducts and pipes. The lowly PTAC or other ductless systems are an excellent example of inherently low energy transport losses. Unnecessary energy transport losses are parasitic to the system as a whole. Extra pump or fan energy in a cooling system adds to the cooling load directly and is a double expense. The extra transport energy in heating systems is usually beneficial heat and represents an extra cost only due to the differential fuel cost between main heating and electric heating.

See Chapter 11—Thermal Energy Transport Notes

Chilled Water Plants and Auxiliaries

Chillers do not operate by themselves. Interestingly, there is no provision in chiller efficiency ratings (COP, IPLV) for the auxiliary pumps and cooling tower fans that serve them. At full load the auxiliary pumps may consume 10-15% of the total cooling plant input energy but may use 50% or more of the total at part load and drag down the overall efficiency. This is especially true for smaller systems with one chiller. Designs that utilize variable flow pumping or modular/staged systems with incremental pump capacity help keep the auxiliary energy use in proportion at lower loads. Cooling towers are another auxiliary but tend to track the plant load profile, either by speed change (if VSD controlled) or by cycling. Note that condenser water reset helps the chiller part load efficiency considerably.

Simply adding variable speed controls to the pumps does not cure chiller plant part load losses. Variable flow reductions through a chiller are limited by tube velocity and Reynolds number which impact tube heat transfer at low flows. Thus, as flow is reduced and pump energy is conserved, chiller power requirements per ton increase, eroding some of the pump savings. The amount of flow reduction before break-even will vary by chiller—some can be reduced to half flow and some manufacturers do not allow any reduction at all. Each chiller is different.

See Chapter 11—Part Load Chilled Water System Performance

Central Heating and Cooling Plants—Distribution Losses

Large central systems are more efficient at full load than most unitary systems but have energy disadvantages at part load:

1. Since they are the sole source of heating or cooling it is common for central plants to run year round including low-load months. Operating large equipment to serve small heating or cooling needs is inefficient and can lead to cycling. Jockey equipment can be used for these very small seasonal loads, either at the central plant or at a particular point of use.
2. Flow reductions in chilled water secondary distribution systems are limited by low system differential temperatures that accompany low facility cooling loads. This low delta T phenomenon increases the flow rate and pump energy cost per ton at reduced loads, pulling down overall plant efficiency at part load. The roots of this issue are in the building air-side equipment and correcting it involves air-side as well as water-side intervention.
3. Distribution thermal piping losses become more significant at reduced load, reducing the fraction of heating and cooling energy delivered to the point of use. This is especially true with heating systems due to higher differential temperatures between the

fluid and ambient surroundings. Thermal losses are normally in reasonable proportion at full load but disproportionate at part load. Temperature reset and more insulation can reduce this parasitic loss, but the core issue remains and the best remedy for part load energy efficiency is often distributed systems nearer the points of use.

See Chapter 5—ECM Descriptions—Boilers and District Heating

HVAC Air System Types

HVAC designers have a choice of systems to apply and there is no one-size-fits-all solution. Of the systems capable of satisfying the design load, there will be differences in full load equipment efficiency, energy transport efficiency, and construction cost to consider. But there are also part load efficiency differences.

Several HVAC system types accomplish part load control through tempering or mixing. These systems suffer significant losses from simultaneous heating and cooling at part load. Modern systems use much less false loading from mixing and heat/cool overlap, but many of these legacy systems are still in service. In evaluating part load performance for an HVAC air system, a primary criterion is the extent to which hot/cold mixing and tempering occurs. Another determinant of part load efficiency is whether the energy transport power reduces proportionally with load; e.g. constant vs. variable air volume. These criteria can be used to form a generalized, basic grouping of common HVAC air system types with respect to part load inefficiency traits.

HVAC Air Systems with Notably Poor Part Load Performance

Constant Volume Reheat, Double Duct, and Multi Zone are about the worst at part load. While effective at producing comfort for the various zones, part load energy use is high. The mixing used in these systems at part load creates simultaneous heating and cooling.

Constant Volume Reheat System: Provides cooling capacity to satisfy the worst case zone; the others are tempered with reheat to prevent over cooling.

Double Duct System: The temperature of the cold duct satisfies the worst case cooling zone and temperature of the hot duct satisfies the worst case heating zone; the others are tempered by mixing the two air streams

Multizone System: Schematically the same as the Double Duct system but the cold duct and hot duct are restricted to the air handler itself

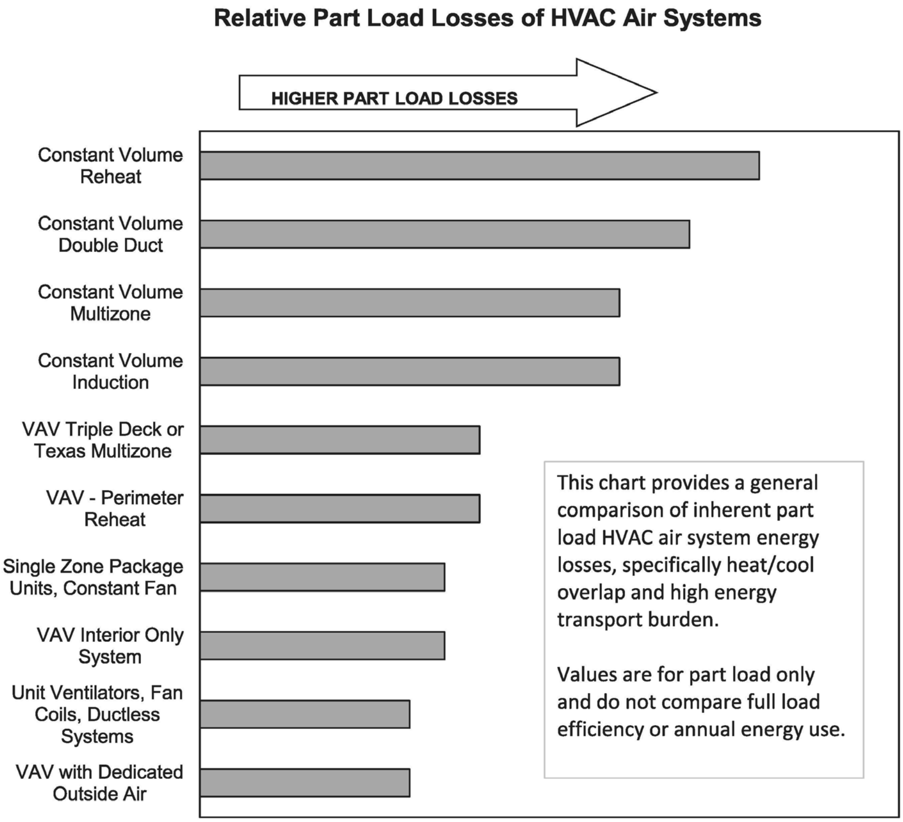


Figure 24-L6. Relative Part Load Efficiency of HVAC Air System Types

which have a cold and hot deck; blending occurs at the air handler.

Additional losses are inherent in the double duct and multi zone systems, in the shared economizer. These systems normally have a single supply fan and a single set of economizer dampers so all zones share the same mixed air stream. For cooling mode at part load, the air economizer is a good bet since it reduces mechanical cooling needs. But if there are any zones requiring heat, each degree lowering the mixed air temperature raises the heating burden for that zone by false loading the heating coil. Improvements can mitigate energy waste in these systems but fully removing the energy penalties requires a system change to one that does not have the overlapping traits. One example is a multi-zone conversion that eliminates the hot deck and uses a VAV box and reheat coil for each zone; the system is then VAV reheat. It is also possible to use single zone package equipment for each zone of a multi-zone unit; this converts the multi-zone system to multiple single zone systems. (See **Figure 24-L7**). In both examples, the existing low pressure zone ductwork is retained.

See Chapter 24—HVAC Overlapping Heating and Cooling

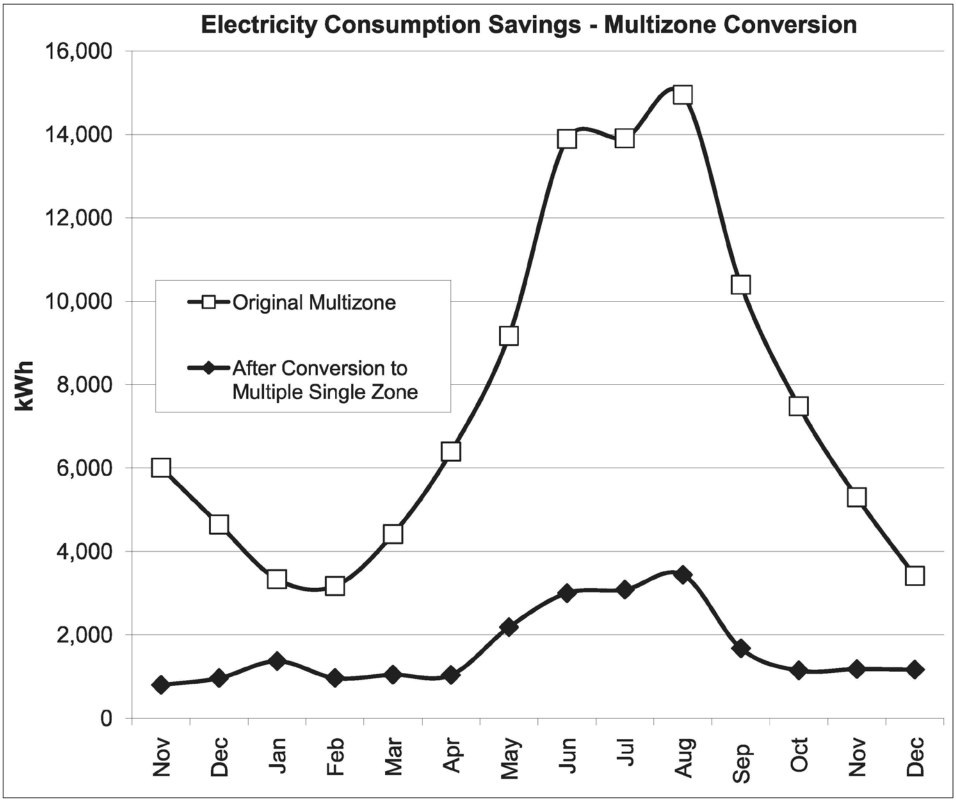


Figure 24-L7.
Case Study: Multi-Zone Conversion to Multiple Split System Control
Source: Custom Mechanical Equipment (CME), Ponca City, OK, 2008.
System change eliminated heat/cool overlap from mixing and allowed optimizing air economizer position on a per-zone basis.

System Before: 20-ton packaged DX rooftop MZ, 5 zones, gas heat.
System After: Split System Unitary Components, five 4-ton sets, gas heat.
Data is measured from utility bills, before and after.

Other system types have varying advantages or disadvantages regarding part load efficiency. Two examples are provided to illustrate basic concepts:

Single Zone Unitary Equipment: These systems usually include low pressure ductwork, small zones, and constant volume fans. Excellent protection from simultaneous cooling since the equipment runs in either heat or cooling mode. Fans are usually low efficiency type, but fan horsepower is low due to small distribution systems. Part load fan energy consumption is proportional to the system load if allowed to cycle with the equipment. However, if the fan is required to run continuously for ventilation, then the fan energy is constant and becomes a larger percentage of total energy at reduced load. Ductless split systems or PTACs render fan energy insignificant since there are no duct losses.

Variable Air Volume (VAV): Part load fan energy use for systems using VSDs is proportional to the system load, with savings that decrease exponentially (fan laws). Mechanical modulation techniques such as inlet vanes or fan shrouds are not as efficient as VSDs. Comparison of different fan modulation options are not included here but are well known.

Single duct VAV systems include ventilation air and minimum settings on the VAV boxes. The minimum air flow false loads the perimeter heating system, raising winter energy cost because the heater must heat the supply air to room temperature before any building heating occurs. The minimum air flow design also tends to over-cool interior spaces requiring supplemental heat such as space heaters. Separate systems for perimeter and interior spaces can mitigate the minimum air flow effect, but to eliminate it completely would require a system change. To operate a VAV box with zero minimums the ventilation function must be treated separately as with a dedicated outside air system (DOAS).

M— FACILITY GUIDE SPECIFICATION: SUGGESTIONS TO BUILD-IN ENERGY EFFICIENCY

Source: "Strategic Facility Guidelines for Improved Energy Efficiency in New Buildings," Strategic Planning for Energy and the Environment (SPEE) Journal, Vol. 26, No. 3, 2007.
Includes March 31, 2007 errata.
Original content has been updated.

Author's note: Design specifics will vary over time and by facility. The point of this document is to encourage the building owner to assert their intentions for energy frugality in specific and measurable ways, as an expectation given to the design team; and then to follow up in design reviews to be sure it has been provided.

A guide specification is a hand-out document given to a design team at the beginning of a project to provide general instructions and Owner preferences. The Owner handing out the guidelines has more of an effect on the end result than the same suggestions made from a lone team member, hence the term Top-Down. These instructions are then integrated into the other governing documents and codes that eventually form the design. Traditional guide specifications, used by national accounts, campuses, and large facilities, spell out preferred manufacturers, acceptable types of piping, valves, light fixtures, pavement, etc. The concept of energy efficient guidelines in the Owner's guide specification is a natural and overdue extension of an existing document. Even if the Owner does not have a guide specification to add this to, the listed items in this document can be used in standalone fashion to serve the same purpose and provide the same benefit to the Owner.

General:

- Items marked with an asterisk (*) are climate-dependent.
- Design document submittals must include detailed narrative descriptions of system functionality, features, limitations, design assumptions and parameters, for use by the Owner. The narratives will be detailed enough to provide benefit to subsequent design teams, and will be written to be informative and useful to building operations personnel. The narrative will be provided as a deliverable with the schematic design, and will be updated with each subsequent design delivery including DD and CD phases. In its final form, this document shall be placed on the first sheet of the drawing set behind the title page, so that the information is retrievable years later when all that is available to facility operations are the drawings. Design assumptions include number of people, indoor and outdoor HVAC design conditions, foot-candles of illumination, hours of operation, provisions for future expansion (if any), roof snow load, rainfall rates, etc. that all define the capabilities of the building.
- All equipment schedules, including HVAC, plumbing, lighting, glazing, and insulation shall be put onto the drawings and shall not

reside in the specification books, so that the information is retrievable years later when all that is available to facility operations are the drawings.

- Design thermal insulation values and glazing properties that affect energy use (U-Value, shading coefficient, etc.) shall be clearly noted on the drawings.
- Project commissioning that includes identifying measurable energy savings goals and monitoring the design and construction activities with these as Project Intent items, with early detection and notification of any project changes that impact energy use or demand.
- Project final payment contingent upon:
  - — Receipt of accepted accurate as-built drawings, with accuracy verified by owner and signed by the contractor.
  - — Receipt of accurate and complete O/M manuals, with certified factory performance data, repair parts data, and vendor contact information for all energy consuming equipment, including all HVAC and lighting equipment and controls.
  - — Receipt of test and balance report that demonstrates design intent is met for air, water, and ventilation quantities, showing design quantities, final adjusted quantities, and percent variance. This would include all VAV box minimum settings shown, including both heating and cooling balanced air quantities. This would also include any equipment performance testing that was specified for the project.
  - — Verification by the Owner that the test and balance settings include permanent markings so these settings can be preserved over time.
  - — Receipt of on-site factory-authorized start-up testing for primary HVAC equipment including chillers and boilers, with efficiency and heat/cool performance figures and heat exchanger approach temperatures to serve as baseline. The submitted reports would include as a minimum heating/cooling output, gas/electric energy input, heat exchanger approach temperatures, water and air flows.
  - — Receipt of control shop drawings with detailed descriptions of operation.
  - — Acceptance testing of the automatic control system using the approved sequence of operation, and verification that the sequences are fully descriptive and accurate. Acceptance testing

also includes review of the control system man-machine interface provisions to become familiar with each adjustable point in the system. Acceptance is by the Owner who will witness each sequence as part of the turnover training requirements.

- Building design must prevent negative pressure condition, unless safety considerations require it.
- Electric resistance space heating, air heating, water heating not allowed, unless there is no means to get natural gas to the site.
- Portable space heaters not allowed, unless required for an approved emergency measure.

Energy Use, Overall Performance:

- Using ASHRAE 90.1 or local Energy Code as a baseline, demonstrate through computer modeling that the building energy use will be at least 30% less than this value.

Test and Balance:

- Balance using "proportional balancing," a technique that strives to reduce throttling losses, which permanently energy transport penalties (pump and fan power).
- Any motor-driven pump or fan over 5 hp found to be throttled with a resistance element (valve or damper) more than 25% must be altered by sheave change or impeller trim to eliminate lifelong energy waste from excessive throttling losses.
- All 3-phase motor loads, including HVAC equipment, must include voltage balance verification as part of the TAB work. Voltage imbalance of more than (1) percent indicate unbalanced electrical service in the building and unacceptable efficiency losses.
- Vertical return air shafts serving multiple floors require a balancing damper at each branch outlet to proportion the return air by floor.
- Air flow performance testing for all ARI certified HVAC factory packaged unitary equipment greater than 5-tons capacity. Heating and cooling performance and efficiency verification is assumed via the ARI certification process.
- Heating efficiency, cooling efficiency, and air flow performance testing for all HVAC split system equipment greater than 5-tons capacity or 200,000 Btuh input heating capacity.
- Water flow performance testing for all ARI certified factory packaged water chillers. Cooling performance and efficiency verification

is assumed via the ARI certification process.
- Water flow and combustion efficiency testing for all boiler equipment.
- Combustion efficiency testing for all boiler equipment unless factory startup is provided on site.
- Cooling tower thermal performance verification is assumed via the CTI certification process.

Electrical Service:
- Provide separate utility metering for electric, gas, and water for the building, separate from other buildings.
- Electrical transformer conversion efficiency not less than 95% efficient at all loads from 25% to 100% capacity. Dry-type transformers NEMA TP-1 compliant.
- Locate transformers in perimeter areas that do not require air conditioning for cooling.
- Power factor correction on large motor loads, for overall building PF of 90% or better. Large mechanical equipment can be provided with the correction equipment. If motor loads are segregated, this can be done at the switchgear.
- Arrange switchgear and distribution to allow metering of the following electrical loads (requires segregating loads):
  - — Lighting.
  - — Motors and Mechanical.
  - — Plug Loads and Other.

Envelope:
- Orient buildings long dimensions E-W where possible to reduce E-W exposure and associated solar load.
- Provide building entrance vestibule large enough to close one door before the next one opens (air lock).
- Where thermal breaks are used, the thermal break material must have thermal conductivity properties an order of magnitude better than the higher conductivity material it touches
- *Minimum wall insulation 25% beyond ASHRAE 90.1 values, but not less than R-19. Insulation is generally not expensive during new construction. Incorporate exterior insulation system (outboard of the studs) for at least one half of the total R-value, to avoid thermal short circuits of standard metal stud walls, which de-rate simple batt insulation system by approximately 50%, e.g. a standard stud wall with R-19 batts between the studs yields an overall R-9.5.

- * Minimum Roof insulation R-value 25% beyond ASHRAE 90.1 values, but not less than R-30. Insulation is generally not expensive during new construction. Select insulation that will retain its thermal properties if wet, e.g. closed cell material.
- Wall glazing: double pane, with thermal breaks. U-value <=0.35, shading coefficient <=0.35, not more than 25% of gross wall area.
- Skylight/clerestory elements triple pane (layer) construction with sealed air spaces, overall U-value <=0.25, shading coefficient <0.2, not more than 5% of roof area.
- Return plenums and shafts designed with an air barrier for leakage not exceeding 0.25 cfm/square foot of building envelope surface area @ 50 Pa (EBBA Criteria). Shaft construction requires field testing and verification.
- Building envelope devoid of thermal short circuits. Provide thermal break at all structural members between outside and inside surfaces.
- Building leakage testing required (new buildings), with no more than 0.25 cfm/square foot of building envelope surface area @ 50 Pa (EBBA Criteria).
- Utilize lower ceilings to reduce necessary light input power for equivalent light levels at the work surface.
- Utilize reflective (light) color interior colors for ceilings, walls, furniture, and floors, to allow reduced lighting power for comparable illumination. It can take up to 40% more light to illuminate a dark room than a light room with a direct lighting system.
- Good reflectance parameters to use when picking interior surfaces and colors follow. If these values are used and the lighting designer is informed of it, the integrated design process will allow reduced lighting power to achieve the desired light levels.
  - — Min 80% reflective Ceiling
  - — Min 50% reflective Walls
  - — Min 25% reflective Floor and furniture
- Provide operable blinds for vision glass.

Lighting:

- Follow ASHRAE 90.1 or local Energy Code requirements for Lighting Power Budget Guidelines, and verify that designs are lower than these limits, while meeting current applicable IES lighting illumination requirements.

- Utilize task lighting and less on overhead lighting for desk work.
- Provide separate circuits for perimeter lights within 10 feet of the wall, or within 5 ft of a skylight to allow manual or automatic light harvesting.
- Use 1-2-3 switching for large open interior area spaces.
- Use ballast that will tolerate removing at least one bulb with no detriment.
- Where fluorescent lighting is used, select bulb and ballast combination for >50,000 hours life.
- Where occupancy sensors are used, provide "programmed ballast start" that will tolerate large numbers of on-off cycles without bulb or ballast life span detriment.
- Use high power factor ballast, with minimum PF of 95% at all loads.
- Outdoor lighting on photocell or time switch.

Motors and Drives:

- All motors meet or exceed federal efficiency standards.
- VFD on all HVAC motors larger than 10 hp that have variable load.
- Motor nameplate HP not more than 20% higher than actual brake horsepower served (i.e. do not grossly oversize motors).

HVAC:

- Provide HVAC calculations and demonstrate equipment is not oversized. Equipment selection should not be more than 10% greater capacity than calculated values indicate, plus allowance for pick-up load.
- HVAC calculations will include both maximum and minimum heat/cool loads and equipment shall be designed to accommodate these load swings, maintaining heat/cool efficiency equal to or better than full load efficiency at reduced loads down to 25% of maximum load, e.g. equipment capacity will track load swings and energy efficiency will be maintained at all loads.
- Design for varying HVAC airflow that tracks load. This can be VAV design, or if using packaged equipment, with 2-speed motors that reduce fan power to less than 50% on low speed.
- Provide necessary outside air, but no more than this. Excess ventilation represents a large and controllable energy use. Reduce exhaust to minimum levels and utilize variable exhaust when possible instead of continuous exhaust. Reduce 'pressurization' air commensurate with building leakage characteristics. If the building is tested

to low leakage as indicated herein, there should be little need for this extra air, or the heat/cool energy it requires. Design controls to dynamically vary outside air with occupancy.

- VAV box primary heating CFM shall be not higher than the cooling minimum CFM. This is to say the VAV box primary damper will not open up during heating mode.
- Zoning:
  - Design HVAC zoning to require heating OR cooling, not both. This will improve comfort and also reduce the inherent need for simultaneous heating and cooling.
  - Do not zone any interior areas together with any exterior areas.
  - Do not zone more than 3 private offices together.
  - Do not zone more than one exposure (N, S, E, and W) together.
- Do not heat warehouses above 60 degF.
- Do not use electric resistance heat.
- Do not use perimeter fin-tube hydronic heating.
- *In cooler climates where HVAC economizers are used, designs should normally favor air-economizers over water-economizers since the efficiency kW/ton is better for the air system. The water-side economizer 'free cooling' includes the pumping and cooling tower fan horsepower, as well as the air handler fan. If the air handler fan power is considered required regardless of cooling source, the air-side economizer is truly 'free' cooling.
- *In very dry climates, with outdoor air wet bulb temperatures consistently less than 52 degrees and dew point consistently less than 42 degrees, evaporative cooling (direct, indirect, or direct-indirect) is viable in lieu of mechanical refrigeration cooling, as long as indoor humidity of 40%rH or less can be maintained.
- *Air-side economizers for all rooftop equipment, regardless of size, for climates with design wet bulb temperatures below 65 degF.
- Insulate all outdoor ductwork to R-15 minimum.
- Use angled filters in lieu of flat filters, to reduce air friction loss.
- Reduce coil and filter velocities to a maximum of 400 fpm to lower permanent air system losses and fan power.
- Avoid series-fan-powered VAV boxes.
- For fan-powered VAV boxes, use ECM motors to achieve minimum 70% efficiency.
- Heat recovery for any 100% outside air intake point that is greater than 5000 cfm when the air is heated or cooled.

- Air filter requirements:
  - — Air handlers with 25% or less OA: 30%—MERV-7
  - — Air handlers with 25-50% OA: 45%—MERV-9
  - — Air handlers with more than 50% OA: 85%—MERV-13
  - — Provide manometers across filter banks for all air handlers over 20 tons capacity. Equip manometers with means to mark the "new-clean" filter condition, and change-out points.
- Make-up meter for all hydronic systems to log system leaks and maintain glycol mix.
- Separate systems for 24-7 loads to prevent running the whole building to serve a small load.
- Require duct leakage testing for all ducts 2 in. w.c. design pressure class or greater.
- For process exhaust and fume hoods, design for variable exhaust and make-up.
- Utilize general exhaust air as make-up for toilet exhaust and other exhaust where possible.
- Dedicated Outside Air System (DOAS) for large office facilities (over 50,000 SF) with VAV systems, allowing zero minimum settings for all VAV boxes. This will eliminate the VAV reheat penalty, and eliminate the internal zone over-cooling effect from VAV minimums which often requires running the boilers throughout the year for comfort control.
- Separate interior and exterior VAV zoning.
- Do not use grooved pipe fittings in hydronic heating or cooling piping systems to prevent operating central heating and cooling equipment year-round on account of these fittings.
- Verify that all manufacturer's recommended clearances are observed for air cooled equipment.
- Humidification:
  - — Do not humidify any general occupancy buildings such as offices, warehouses, or service centers.
  - — In data centers only, humidification should not exceed 30% rH; utilize adiabatic humidifiers.
  - — Do not locate humidifiers upstream of cooling coils, to avoid simultaneous humidification—dehumidification.
  - — Where humidification is used, provide for elevated apparatus dew point of cooling coils or other means to prevent simultaneous humidification—dehumidification.
- Dehumidification:

— Do not dehumidify below 45% rH.

- Provide performance and efficiency testing of package heating and cooling equipment over 7000 CFM or 20 tons or 500,000 Btu input heating units with factory authorized equipment representatives. Test figures to include on-site gross heat/cool output, fuel and electrical input, and efficiency, compared to advertised values.

Energy Transport Systems—Energy Budget:

- For HVAC air systems, the maximum energy transport budget will be
  - — No less than **10** Btu cooling and heating delivered to the space per Btu of fan energy spent at design conditions.

    This will generally steer the design toward generous sizing of sheet metal ducts, air handler cabinetry, coils and filters, higher efficiency fans (0.7 or better), and higher system differential temperatures to reduce air flow rates, but will result in greatly reduced lifetime energy use since it lowers the bar of system pressure.
  - — Fan Bhp limitation from:

    Cooling fan Bhp max output = Cooling Btu gross output/(10 * 3413)
  - — Air hp limitation from:

    Cooling fan Bhp max budget * fan-eff.
  - — TSP limitation from:

    TSP = (air-HP * fan-eff * 6360)/CFM

  For ***example***, a 100-ton HVAC air system would be limited to (100*12,000)/(10*3413) + <u>35.2 Bhp</u>.

  With a 70%e fan, air Hp is limited to (35.2*0.70) = 24.6 air Hp.

  At 350 cfm per ton, the static pressure TSP would be limited to (24.6*0.70*6360)/(100*350) = 3.1 in. w.c. TSP.

  Note what happens when the designer meets the fan power budget using a lower efficiency fan at 60%e:

  The fan budget is still 35.2 Bhp.

  The air HP is now 35.2*0.6 = 21.1 air Hp, and

  The TSP is now 21.1*0.6*6360/(100*350) = 2.3 in. w.c.

  NOTE: For systems with both supply and return fans, the transport energy considers both combined as the"fan."

- For HVAC water systems, the maximum energy transport budget will be:
  — No less than **50** Btu cooling and heating delivered to the space per Btu of pump energy spent, at design conditions.

  This will generally steer the design toward generous sizing of piping, strainers, coils, and heat exchangers, higher efficiency p umps (0.75 or better) and higher system differential temperatures to reduce water flow rates, but will result in reduced lifetime energy use since it lowers the bar of system pressure.

  — Pump hp limitation from:
  Cooling pump hp max input = Cooling Btu gross output / (50 * 3413 * motor-eff)

  — Water hp budget from:
  Pump max hp * pump-eff.

  — HEAD limitation from:
  HEAD = (water-HP * pump-eff * 3960)/GPM

- Heating: minimum 40 degree dT design, to reduce circulating flow rates and pump HP.
- Cooling: minimum 16 degree dT design, to reduce circulating flow rates and pump HP.

Boilers and Furnaces:
- No atmospheric burners.
- No standing pilots.
- Design hydronic system coils to return water to the boiler at or below 140 degree water with a minimum of 40 deg temperature drop. This will reduce circulating pump energy and improve boiler efficiencies.
- Minimum efficiency of 85% at all loads down to 25% load.
- For heating load turn-down greater than 4:1, provide modular boilers or a jockey boiler.
- For multiple boilers sharing multiple pumps, provide motorized valves to cause water flow to occur only through the operating boiler.

- Provide stack dampers interlocked to burner fuel valve operation.

Chillers:

- * Water-cooled centrifugal efficiency 0.5 kW/ton or less with 70 degF condenser water and 45 degF chilled water for climates with design wet bulb 65 degF and lower. 0.58kW/ton or less with 85 degF condenser water and 45 degF chilled water in climates where design wet bulb temperatures are above 75 degF.
- * Water-cooled centrifugal units able to accept 55 degree condenser water at 3 gpm per ton, all loads. Beneficial in dry climates with design wet bulb temperatures less than 65 degF and typical wet bulb temperatures less than 50 degF.
- * Water-cooled positive displacement units 0.7 kW/ton or less with 70 degF entering condenser and 45 degF chilled water for climates with design wet bulb 65 degF and lower. 0.81kW/ton or less with 85 degF condenser water and 45 degF chilled water in climates where design wet bulb temperatures are above 75 degF.
- Do not provide chilled water temperatures less than 45 degF. Select cooling coils to provide necessary cooling with 45 degF chilled water or higher.
- Air cooled chiller efficiency 1.0 kW/ton or less with 95 degF entering air.

Cooling Towers:

- Selected for 7 degree approach at design wet bulb and 0.05kW/ton or less fan input power. This will steer the design toward a larger free-breathing cooling tower box with a small fan, minimizing parasitic losses from the cooling tower fan. Cooling tower fan kW/ton should not be more than 1/10th of the chiller it serves.
- * Set condenser water temperature set point to no higher than 70 degF for climates with design wet bulb 65 degF and lower. For climates with higher wet bulb temperatures, design to 7 degrees above design wet bulb. Provide reset controls to lower the setting whenever conditions permit.

Air-Cooled Equipment and Cooling Towers in Enclosures:

- Locate to prevent air short-circuiting and associated loss of thermal performance. Rule of thumb is the height of the vertical finned surface projected horizontally. The fan discharge must be at or above the top of the enclosure and there should be amply sized inlet air openings in the enclosure walls as low as possible.

Ground Source Heat Pumps:

- COP 4.0 or higher at 40 degF entering water.
- EER 17 or higher at 80 degF entering water.
- No electric resistance heating.

Controls:

- Design OUT all simultaneous heating and cooling through the use of proper zoning, interlocks, and dead bands. This includes all constant volume systems and terminal unit systems. VAV systems inherently have an overlap which should be minimized by water and air reset in heating season, prudent use of minimum VAV box settings, and consideration of systems that separate the outside air from the supply air.
- Programmed start-stop for lighting and HVAC systems, with option for temporary user overrides. Use these controls to prevent un-necessary operating hours.
- Lock out air flows for conference rooms and intermittent occupancy rooms by interlocking VAV box to close with occupancy sensors.
- Lock out chiller operation below 50 degrees, except for data centers or humidity-sensitive areas that cannot use outside air for cooling.
- Lock out boiler operation above 60 degrees, unless space temperatures cannot be maintained within the specified ranges any other way.
- * All cooling by air-economizer below 55 degrees for climates with design wet bulb 75 degF and lower.
- Night setback for heating. Suggested temperature for unoccupied time is 60 degF.
- No night set-up for cooling—no cooling operation in unoccupied times for general occupancy buildings. If building temperature rise during unoccupied times can cause detriment, then limit off-hours cooling operation to 85 degrees indoor temperature.
- Reset boiler hot water temperature settings in mild weather.
- * Reset chilled water temperature settings in mild weather, provided outdoor air dew point is below indoor dew point levels. Refrigeration savings generally exceeds increases in pump power.
- Provide appropriate interlock for all exhaust fans to prevent infiltration of outside air from uncontrolled exhaust fans that operate in unoccupied times.
- All analog instruments—temperature, pressure, etc. other than on-

off devices—must be calibrated initially (or verified for non-adjustable devices). Merely accepting out-of-the-box performance without verification is not acceptable.

- 2-year guarantee on calibration, with 18-month re-calibration of all analog inputs. Owner re-calibrate instruments on a 5-year cycle.
- Air handler control valves with a residual positive seating mechanism for positive closure. Use of travel stops alone for this is not acceptable.
- For terminal units and heating/cooling hydronic water flow rates less than 10 gpm, use characterized ball valves for control valves instead of globe valves or flapper valves, for their inherent improved long-term close off performance. This will reduce energy use from simultaneous heating and cooling.
- Valve and damper actuator close-off rating at least 150% of max system pressure at that point, but not less than 50 psid (water) and 4 inches w.c. (air).
- Dampers at system air intake and exhaust with leakage rating not more than 10 CFM per square foot at 4" water column gage when tested in accordance with AMCA Standard 300.
- Water coil control valve wide open pressure drop sizing not to exceed the full flow coil water-side pressure drop.
- Provide main electrical energy and demand metering, and main gas metering. Establish baseline and then trend and log "kBtu/SF," "kWh/SF-yr," and "kW demand" perpetually and generate alarm if energy use exceeds baseline.
- Standardize basic building indoor temperatures, and supplement with ±2F user local adjustment. Provide deadband between heating and cooling, e.g. 74F cooling, 70F heating.
- 4 degree dead band between space heating and cooling setpoints to prevent inadvertent overlap at zone heat/cool equipment, and from adjacent zones.
- 5 degree dead band between air handler heating and cooling (or economizer) setpoints, e.g. preheat coil cannot share a single, sequenced, set point with the economizer or cooling control.
- Provide separate lighting and HVAC time schedules.
- For chillers (condenser) and hot water boilers, use temperature sensors to log heat exchanger approach values, to prompt predictive maintenance for cleaning fouled heat exchange surfaces. New-equipment approach will be the baseline value, and approach temperature

increases of 50% will prompt servicing.

- Interlock heating and cooling equipment in warehouses serving doorway areas to shut off when roll-up doors are open to reduce waste.
- Optimization routines:
  - — Automatically adjust ventilation rates for actual people count.
  - — Optimal Start to delay equipment operation as long as possible.
  - — Demand limiting control point that will limit all VFD-driven air handler fans components to a maximum of 90% max output in summer. This will cause system temperatures to drift up slightly during extreme weather, but will reduce electrical demand for this equipment (and the cooling equipment it serves) compared to full output operation, during times when utility demand is highest. Do not oversize equipment capacity to compensate for this requirement.
  - — Optimal static pressure setting based on VAV box demand, not a fixed set point. This is a polling routine.
  - — * For areas with design wet bulb temperatures below 65 degF only, optimal supply air reset that will reset the supply air temperature set point upward from 55 to 62 for VAV systems during heating season, to reduce reheat energy. This can either be from two methods.
- Method 1. Basic Optimization. When the main air handler fan is below 40% of capacity and OA temperature is below 40 degrees.
- Method 2. Fully Optimized. Polling VAV boxes (at least 80% of the boxes served are at minimum air flows)
  - — Do not reset SA temperature from return air. Do not reset SA temperature during cooling season.
  - — Reset condenser water temperature downward when outdoor conditions permit, using the lowest allowable condenser water the chiller can accept.

Plumbing:

- Max shower flow 1.5 gpm
- Max bathtub volume 35 gallons.
- Max urinal water flow 0.5 gpf, or waterless.
- Max lavatory water flow 0.5 gpf.
- Metering (self-closing) or infrared lavatory faucets.
- Avoid single lever faucets since these encourage complacency for the use of hot water.

- All domestic hot water piping insulated.
- Heat trap in domestic hot water main outlet piping.
- If a circulating system is used, provide aquastat or timer to prevent continuous operation.
- Max domestic hot water temp for hand washing 125 degrees.
- Gas water heaters in lieu of electric where natural gas is available.
- Domestic water heater equipment separate from the building boiler and heating system, to prevent year-round operation of central heating equipment.
- Water fountains instead of chilled water coolers.
- Operate the building at reduced pressure (such as 50 psig) instead of 70 psig, to reduce overall usage. Verify that design maintains at least 10 psig over the required minimum pressure at all flush valves.

Management and Maintenance Activities to Sustain Efficiency:

- Management Support
  - — Create buy-in from the building occupants. Distribute information to building occupants to raise awareness of energy consumption, especially communicating that the user's habits are an essential ingredient to overall success, and are useful and appreciated. This would be in the form of occasional friendly and encouraging reminders of how user participation is helping, fun facts, etc., along with estimated benefits from behavior changes. Provide measured results whenever available.
  - — Enforce temperature setting limitations, including the explanation of why this is helpful and also why it is reasonable. Encourage seasonal dress habits to promote comfort and conservation together.
  - — Prohibit space heaters.
  - — For offices, utilize LCD monitors and the software-driven "monitor power-off" feature, since the monitor represents 2/3 of the whole PC station energy use.
  - — Track monthly energy and water use and maintain annual graphing lines, comparing current and prior years. Establish new benchmark curves after major renovations, alterations, or energy conservation projects. Compare annual use to benchmark and verify building energy and water usage per SF is not increasing. Report results to the building occupants as an annual energy use report for their feedback.

— Escrow (save) approximately 5% of the replacement cost per year for the energy consuming equipment in the facility that has a normal life cycle, such as HVAC systems, lighting systems, control systems. This will allow 20-year replacement work without 'surprises' to sustain efficient building operations.
— For leased office space, show the tenants their utility costs to increase awareness and encourage conservation by the users. The typical industry arrangement is to build-in utilities into the lease price, and so the tenants do not see a separate utility bill. Although the customers are paying for the utilities, having those costs clearly shown will reduce the complacency in utility use.

- Chillers:
  — Owner provide annual equipment "tune up" including cooling efficiency testing and heat exchanger approach measurements.
  — Owner adjust temperature settings or clean heat exchangers or adjust water flows whenever cooling efficiency tests are less than 90% of new-equipment values. For example, if new equipment benchmark is 0.5 kW/ton, then a measurement of 0.5/0.905=0.55 kW/ton would trigger corrective action.

- Boilers:
  — Owner provide annual equipment 'tune up" including combustion efficiency testing and heat exchanger approach measurements.
  — Owner adjust temperature settings, clean heat exchangers or adjust air-fuel mixture whenever combustion efficiency tests are less than 95% of new-equipment values. For example, if new equipment benchmark is 80%e, then a measurement of 0.8*0.95=0.76 would trigger corrective action.

- HVAC air coils:
  — Owner change filters at least quarterly, and verify there are no air path short circuits allowing air to bypass the filters.
  — Owner clean HVAC coils whenever there is any sign of visible accumulation or if air pressure drop is found to be excessive.

- HVAC air-cooled condensers:
  — Owner provide location free from debris, leaves, grass, etc. and adequate spacing for free 'breathing' and no re-circulation.

— Owner clean heat exchange surfaces annually.

- Controls
  — Owner re-evaluate system occupancy times each year, to reduce un-necessary HVAC and lighting operating hours.
  — Owner re-evaluate control setpoints each year including space temperature settings, duct pressure settings, supply air temperature settings, reset schedules, heating and cooling equipment lock-out points.
  — Owner re-calibrate control instruments each two years other than on-off devices.
  — Owner cycle all motorized valves and dampers from open to closed annually, and verify tight closure.
  — Owner cycle all VAV box dampers from open to closed annually and verify the control system is responsive, since these often have a short life and can fail without the user knowing it.

N—REGRESSION FOR ENERGY MANAGEMENT

Introduction

This section provides basic terms and concepts of regression, and applications and guidelines for use. It is focused on the context of energy management tasks, and not an exhaustive treatment.

Purpose of Regression

- Looks at historical data and tries to predict future data.
- Eliminates variables to evaluate merits of improvements, such as eliminating the effects of weather.
- Utilizes variables that are predictors of energy use.

Independent variables

In energy modeling context, an independent variable is a parameter that is expected to change regularly and has a measurable effect on the energy use of a system or building. Typical independent variables that drive energy consumption that can be incorporated in regression models include outdoor temperature, other weather parameters (e.g., heating or cooling degree days), occupancy, operating hours, and other variable site conditions. Source: Measurement and Verification for Federal Energy Projects, v3.0, FEMP/EERE/DOE, 2008

Linear regression uses the form of a straight line,

y = mx + b(eq. 1)

y = dependent variable (e.g. energy usage)
m = slope (coefficient relating x to y)
x = independent variable (e.g. degree days, number of widgets, occupancy level).
b = y intercept (e.g. base usage, when x=0)

Multiple Variable Linear Regression

Linear regression can have multiple variables. One example of a linear multi-variant regression model for a weather-dependent ECM is shown. In models using weather data, it can be beneficial to define a custom temperature base for calculating HDD and CDD data based on the actual behavior of the building.

Example Equation:
Multi-variant Regression Model for a Weather-dependent ECM

Source: Measurement and Verification for Federal Energy Projects, v3.0, FEMP/EERE/DOE, 2008

$$E = B_1 + (B_2 \times (T_i - T_{i-1})) + (B_3 \times HDD_i) + (B_4 \times CDD_i) + (B_5 \times X_1) + (B_6 \times X_2) + (B_7 \times X_3) \quad \text{(eq. 2)}$$

where:

E = energy use
i = index for units of time for period
B1-7 = coefficients
T = ambient temperature
HDD = heating degree days using a base temperature of 60ºF
CDD = cooling degree days using a base temperature of 65ºF
Xn = independent steady-state variables

Regression is really curve fitting, comparing predicted to actual data

- Straight line works (most common approach)
- Exponential, polynomial or other functions
- Combinations (different functions for different regions)

What regression does:
- Identifies building/process energy use signature
- Re-plays the old building or process with current conditions
- Normalizes (equalizes/negates) uncertainties from variables like weather, occupancy, production
- ***"What the energy use would have been if nothing changed except xxx"***

Applications of regression in energy work
- Project energy use going forward, and with changes in the predictor variables
- Monitor ongoing energy goals
- Evaluate the benefits of a project
- M&V of performance contracts. Removes the effects of variables like weather, etc. to legitimize conclusions of energy savings compared to obligations
- Provide early notification when energy use is outside what is expected, to prompt investigation and corrective action. For a method to use in ongoing operations to guard against backslide of savings, see **Chapter 24o: Error Band: Using Energy Consumption Signatures as an Operational Control.**

Data

The mechanics of regression and actions taken with regression results all depend on good data that has patterns. With poor data, regression will be flawed. *Not all data is good.*
- Look at the data. Does it make sense? Is it doing what you would expect? Is there enough of it? Does it look consistent year to year? Charts are useful to visually spot patterns or anomalies. *It is possible that patterns aren't there.*
- Data that is from private sub meters should consider the quality of the meter. Has it "ever" been calibrated? Is there a way to sanity check the data?
- Monthly utility data often has more or less days than the calendar month it represents. Daily utility reads may not be exactly 24 hours. Normalizing this data to a standard time interval (24 hours, days in July, etc.) improves the data.
- Look for anomalies and search for an explanation. Construction, repair, temporary activity, etc.

- Scrub bad data, estimated reads, missed reads, etc. Where a pattern exists in the data, outliers can be reasonably eliminated >3 standard deviations. Using the regression line through the data, draw additional lines at +/– 3x standard error; data points outside the "error band" are outliers.

Regression Mechanics

- Microsoft Excel software is the source of regression examples in this section. There are other software and manual solutions.
- An x-y scatter plot in the Microsoft Excel 'chart' section provides linear regression, R^2 value, and options for trend line functions for curve fitting.
- The Excel Data Analysis add-on function provides linear regression with statistical confidence tests and allows multiple variables
- Y is the dependent variable, i.e. kWh, therms, gallons of fuel.
- X(s) is the independent variable(s), i.e. heating degree-days, cooling degree-days, production or occupancy figures (actual numbers, or percentages), number of computers, number of shifts, etc.
- Linear regression is the common, but other methods are available and all have the goal of curve-fitting historical behavior for the purpose of predicting future behavior and normalizing data.

Baseline and Signature

Regression provides the energy use signature of the building or process that is fixed. The chosen baseline (usually baseline) is the basis for comparison for other periods. Obviously, the chosen baseline needs to be a good representation of what is normal for that time. We may treat buildings like machines, but they are operated by people and are not entirely predictable. Where weather is a notable factor in energy use, a good baseline will be one where the weather for the year was normal. Sometimes it can make sense to average two or three years to make a baseline year.

The regression process evaluates variables that have a predictable and noticeable effect on energy use… and presumes nothing changes except those variables. Regression will do a good job if the variables are appropriate and the building use is static except for those variables. However, comparison between periods is void if a major change is made to the baseline condition.

- If the building or process changes (new activities, new equipment, bigger building, other changes), a new baseline or a baseline adjustment is needed.
- Provisions to identify and adapt to baseline changes is a key premise

in the field of performance contracting (guaranteed savings).

Limitations of Regression

For statistics, there is safety in numbers. Small data sets are higher risk; a large cloud of data has an automatic advantage over 12 monthly reads. Regression fits the data using the predictors it is given, and so obviously the output quality is linked to the quality of the data set and the predictors. Consider painting a house, where the preparation work before the actual painting is instrumental in the final outcome.

When multiple variables are used, they must be independent of each other, such that a change in X1 has the same effect on Y whether X2 changes or doesn't. Thus, poor quality variables or too many variables affects the quality of the regression. This is discussed more in "**Correlation**."

Using the regression output from a baseline period on another period will be valid to the extent that the variables in the regression explain usage and that nothing changes from the base period except those variables. If the chosen variables explain 80% of the change in energy use (R^2=0.8), then the predicted use of another period will be equally confident; note that the prediction is not perfect. This explains the desire to have the highest confidence levels possible; i.e. higher R^2, lower Significance F, lower P values, and lower Correlation values are better. And if the baseline conditions themselves change (building expansion, different use), the predictions will inherit errors, requiring a +/- baseline adjustment or a new regression.

Regression depends on energy use that is inherently predictable, and expecting a sound prediction from an unpredictable building is not rational. Some manufacturing facilities are very patterned and can be gauged on production; but not all. Consider a manufacturing 'job shop' that is capable of producing a wide variety of products and produces whatever the telephone brings that day. This building will use energy every day and the usage is not purely random, but predicting it will be very challenging. If it is important to quantify, it may require a large array of variables; it may also help to make basic divisions of the plant with sub meters; for example, separating heat treat from sand blast from powder coat, etc. The limitation here is the appropriate variables and available data.

Refer to "**Data**" section. Poor data limits the quality of the initial regression and also limits the results when applying the regression output to a future data set. Anomalies can sometimes be weeded out. Something transient like construction work, equipment defect, or a one-time event can be removed. Irregular uses can sometimes be separated, like weekends at a school that are randomly used for community function; or summer for a 9-month school

Complex energy uses can be difficult to predict. Consider overlapping heating and cooling and part load inefficiencies—these are inconsistent uses and non-linear.

Regression won't explain all energy use, but a good model will explain most of it

Coefficients for the Line Equation

The y-intercept defines the baseline usage that is unaffected by the dependent variable. In the example of weather dependence, it represents the residual usage of energy with no heating or cooling degree days.

The coefficient defines the relative strength of the dependent variable, or gain.

For example, in **Figure 24N-1A** the value of HDD-60 multiplied by 2.83 represents the units of energy that will change with a one HDD-60 change.

Non-Linear or Multi-Linear Data

Straight lines are convenient, but not all data is a well approximated with a straight line. Regression mechanics will determine the best fit straight line for curved data, but it will be a poor fit just the same.

Sometimes there are distinct changes visible in the data (change points), and prediction accuracy is improved when such patterns are acknowledged.

Some examples that create this need:

- Electricity used for both cooling and heating
- Air/water economizers and other systems that are active at different temperatures
- Fuel switching

Split Coefficients

See **Figure 24N-2A** Straight line regression is a poor fit for a curved data set like this.

Solution 1: Two lines: one above 50F, one below 50F. See **Figure 24N-2B.**

Solution 2: Curve fit from trend line options for curve fit/polynomial (Manipulate 1st, 2nd order option, and intercept). See **Figure 24N-2C.**

SUMMARY OUTPUT

Regression Statistics	
Multiple R	0.7904208
R Square	0.62
Adjusted R Sq	0.61225722
Standard Erro	786.126338
Observations	32

ANOVA

	df	*SS*	*MS*	*F*	*Significance F*
Regression	1	30868774.62	30868775	49.94991	7.38976E-08
Residual	30	18539838.56	617994.6		
Total	31	49408613.18			

	Coefficients	*Standard Error*	*t Stat*	*P-value*	*Lower 95%*	*Upper 95%*
Intercept	-67.646297	203.6069036	-0.33224	0.742019	-483.4670667	348.1745
HDD-60	2.83213381	0.400724983	7.067525	7.39E-08	2.013744212	3.650523

Figure 24N-1A One Coefficient

SUMMARY OUTPUT

Regression Statistics	
Multiple R	0.757476982
R Square	0.573771378
Adjusted R Square	0.479053906
Standard Error	7646.641805
Observations	12

ANOVA

	df	*SS*	*MS*	*F*	*Significance F*
Regression	2	708402806	3.54E+08	6.0577143	0.021547245
Residual	9	526240178	58471131		
Total	11	1234642985			

	Coefficients	*Standard Error*	*t Stat*	*P-value*	*Lower 95%*
Intercept	11,217	18225.8277	0.615472	0.5534861	-30012.1999
Production	588	218.87044	2.688357	0.0248633	93.28258876
CDD-60	35.5	13.0994078	2.706275	0.0241443	5.817682963

Figure 24N-1B Two Coefficients

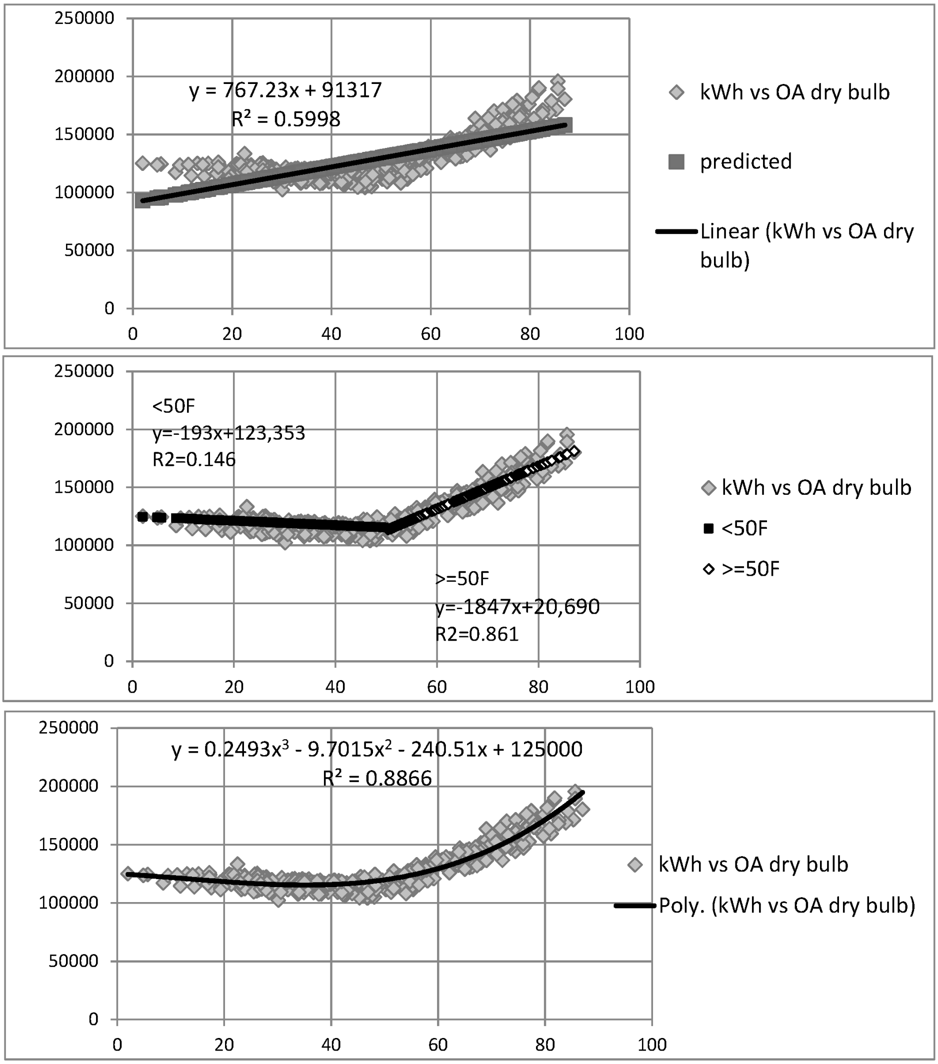


Figure 24N-2ABC Fitting Curved Data

Top chart is Figure 24N-2A: Straight Line Option (The Problem)
Middle chart is Figure 24N-2B: Split Coefficient Solution
Bottom chart is Figure 24N-2C: Curve Fit Solution

Regression Indicators

These are basic tips for basic tasks. Statistics is a complex field. Thresholds for "good" and "bad" are different depending upon text or source.

R^2 Describes the extent to which the variables predict the usage (how dependent the dependent variables are and how predictably y changes with respect to the mean value of y). This is why R^2 will be low for a horizontal line, even when fitting perfectly.

- 1.0 is perfect, 0.8 is very good, <0.5 is low confidence and not useful.

Significance F

Indicates the overall 'goodness' of the model. Typical basis of goodness is 95% confidence. Values are read as the probability the model is wrong, so smaller #s are better.

- 0.0 is perfect, <0.05 is very good, >0.10 is low confidence and not useful.

P-Value

Indicates the 'goodness' of the individual variable. Values are read as the probability the model is wrong, so smaller #s are better.

- 0.0 is perfect, <0.05 is very good, >0.20 is low confidence and not useful.
- One QC metric* says that "all variables must have P<0.2 and at least one variable <0.10" (Source: Superior Energy Performance (SEP) M&V Protocol)

Correlation (C)

A fundamental assumption in all multivariable regression is that each independent variable is fully isolated/unrelated to the other one. If variables are related, this violates the premise of the multivariable method. To the extent that there are interactions, the quality of the model is affected. In some cases, the model will even "look better" with more variables, but has been fooled by the interactions. So test for interactions.

- Correlation (C) indicates the extent of interaction between the variables. Ideally, the effect of one variable is the same on the dependent variable regardless of what the other variables are doing, so a smaller number is better.
- 0.0 is perfect, <=0.10 small correlation, >0.5 is significant correlation.
- 1.0 means both variables are measuring the same thing.

Minimum Number of Data Points

Minimum number is a function of the number of parameters in the regression model equation (# of coefficients + intercept)

- There must always be more data points than parameters; more is better.

Other Indicators (See Table 24N-1 and Figure 24N-3AB.)

Improving R^2 with Better Data

- More data: Two or three years, or daily data.
- Fine tune the balance temperature: Different sets of HDD/CDD based on balance temperatures of 60, 55, 50 degrees F, etc.
- Eliminate outliers: Bad utility reading, closed building, construction work on building, special events, other anomalies…things that are not normal.

Improving R^2 with a Better Model

- Change points.
- Curve fit.
- Add variables that are important.
- Remove variables that are not important.

Table 24N-1. Statistical Validations Guidelines

Source: Measurement and Verification for Federal Energy Projects, v3.0, FEMP/EERE/DOE, 2008

Parameter Evaluated	Abbrev-iation	Suggested Acceptable Values	Purpose
Coefficient of determination	R^2	> 0.75	Indicates model's overall ability to account for variability in the dependent variable. Lower R2 values may indicate independent variables may be missing or additional data is needed.
Coefficient of variation of root-mean squared error	CV(RS ME)	< 15%	Calculates the standard deviation of the errors, indicating overall uncertainty in the model
Mean Bias Error	MBE	+/- 7%	Overall indicator of bias in regression estimate. Positive values indicate higher than actual values; negative values indicate that regression under-predicts values.
t-statistic	t-stat	>|2.0|	Absolute value >2 indicates independent variable is significant

correlation	*CDD-60*	*Widgets*	*kWh*
CDD-60	1		
Widgets	0.0725	1	
kWh	0.321588	0.337298	1

CDD-60 has very little interaction with Widgets

correlation	*CDD-60*	*Widgets*	*kWh*
CDD-60	1		
Widgets	0.728234	1	
kWh	0.625775	0.643669	1

CDD-60 has significant interaction with Widgets

Figure 24N-3AB. Correlation Test Examples

Top chart is **Figure 24N-3A:** Low correlation between two variables
Bottom chart is **Figure 24N-3B**: High correlation between two variables

SUMMARY OUTPUT

Regression Statistics	
Multiple R	0.602419327
R Square	0.362909046
Adjusted R	0.341672681
Standard E	26795.02896
Observatio	32

Mixed Signal 1
Significance F says the model is overall very good. R^2 says it is poor. Back up.

ANOVA

	df	*SS*	*MS*	*F*	*Significance F*
Regression	1	12269477585	1.23E+10	17.08904	0.000263964
Residual	30	21539207309	7.18E+08		
Total	31	33808684894			

	Coefficients	*Standard Error*	*t Stat*	*P-value*	*Lower 95%*	*Upper 95%*
Intercept	172,794	6397.922478	27.00783	1.31E-22	159727.7226	185860.32
CDD-50	-67.4	16.31208587	-4.13389	0.000264	-100.7460749	-34.11863

Figure 24N-4. Conflicting Indicators

Conflicting Indicators

Statistic tests for regression don't always agree.

One approach:

- Focus primarily on R^2 and Significance F. If they agree, go forward; if they don't agree, back up. Question the data or a missing variable.
- If the only thing that looks bad is a P-value, question the data or re-run without that variable.

Example: An attempt is made to identify the energy use signature of a building that has variable occupancy. Data for occupancy figures and heating degree-days are used. The R^2 value and Significance F values are good, but the P-value for the 'occupancy' variable is not, indicating that occupancy in this facility is not a good predictor of energy use. See **Figure 24N-5A.**

In **Figure 24N-5B** the regression has been re-run without the occupancy variable. Significance F test improves markedly.

Dummy Variables (a.k.a. binary variables)

Binary (dummy) variables can be useful to improve prediction quality by adding granularity to the data set. These take on the value of 1 or zero, which is to say the condition is either true or it isn't... Examples:

- Does the school have a kitchen or not? (Yes or no? 1 or 0?)
- Is there a pool or not?
- Is this record a weekend or not? (need daily data)
- Does this day of the week have a third shift?

Example Use of Dummy Variables

Daily electric use against average temperature shows two curves. This building has a strict weekend practice and it shows. See **Figure 24N-6A**.

Regressing the data set with only outside air temperature as the independent variable naturally produces a single line between the two clouds of data. See **Figure 24N-6B.**

Adding a variable for 'weekday' (**Figure 24N-6D)**, identifies each record as either a weekday (value=1) or not a weekday (value=0). Re-plotting predicted vs. actual acknowledges two curves for energy use considering weather, and whether it is a weekday or not. Electric use is regressed against HDD, CDD, *and "weekend."* See **Figure 24N-6C.**

SUMMARY OUTPUT

Regression Statistics	
Multiple R	0.910518987
R Square	0.829044826
Adjusted R Square	0.791054787
Standard Error	21804.59309
Observations	12

Part A
Significance F and R^2 says it is good, P-value says 'occupancy' variable is poor. Try re-running without it. (See **Part B**)

ANOVA

	df	SS	MS	F	Significance F
Regression	2	20750771410	1.04E+10	21.82269	0.000353159
Residual	9	4278962517	4.75E+08		
Total	11	25029733927			

	Coefficients	Standard Error	t Stat	P-value	Lower 95%	Upper 95%
Intercept	78,551	82012.54519	0.957794	0.363187	-106974.1456	264076.39
occupancy	390	920.2791095	0.423284	0.682023	-1692.276337	2471.3556
HDD-60	135	46.78994583	2.879502	0.018193	28.88552617	240.57795

SUMMARY OUTPUT

Regression Statistics	
Multiple R	0.90864817
R Square	0.825641497
Adjusted R Square	0.808205647
Standard Error	20890.54077
Observations	12

Part B
Occupancy was not a good indicator. More variables are not always better.

ANOVA

	df	SS	MS	F	Significance F
Regression	1	20665586992	2.07E+10	47.3531	4.29082E-05
Residual	10	4364146935	4.36E+08		
Total	11	25029733927			

	Coefficients	Standard Error	t Stat	P-value	Lower 95%	Upper 95%
Intercept	113036.4554	9015.879201	12.53749	1.93E-07	92947.82471	133125.09
HDD-60	116.3900951	16.91382498	6.881359	4.29E-05	78.7037447	154.07645

Figure 24N-5AB. Conflicting Indicators Example and Solution

Top chart is **Figure 24N-5A:** Regression with heating degree-days and occupancy
Bottom chart is **Figure 24N-5B:** Regression with heating degree-days only

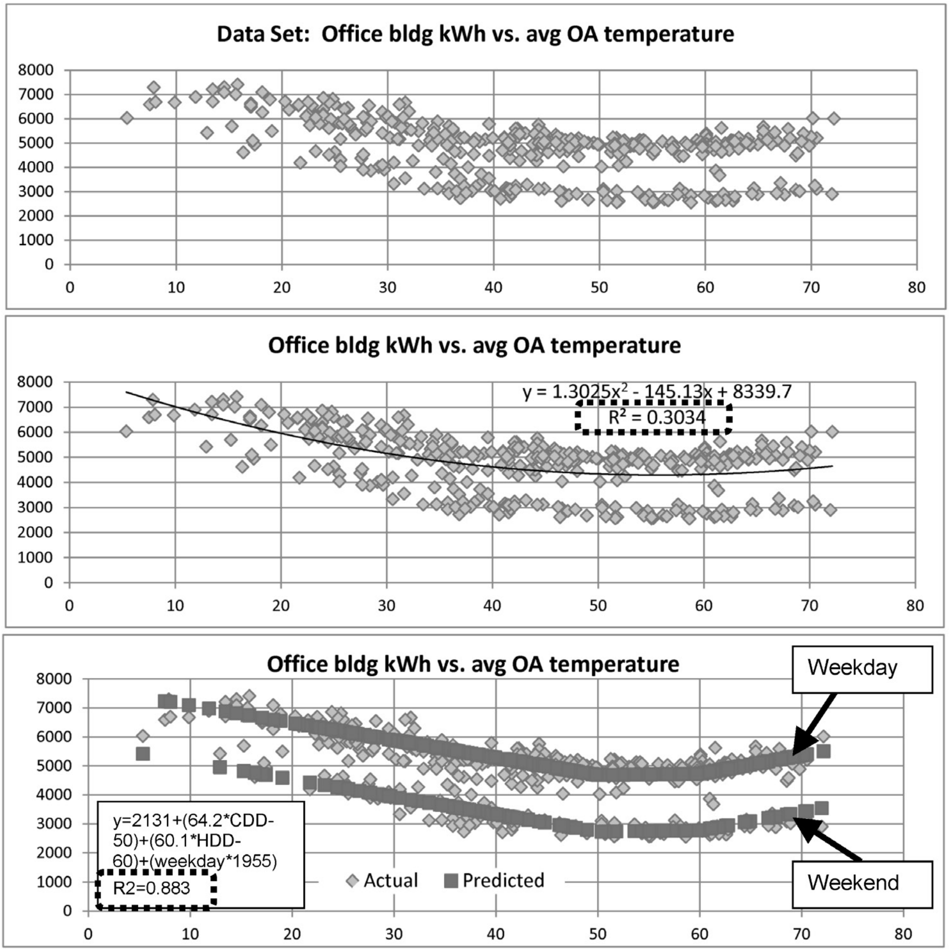


Figure 24N-6ABC. Dummy Variable (Binary Variable) Example

Top chart is	Figure 24N-6A:	Data Set with Two Distinct Patterns
Middle chart is	Figure 24N-6B:	Conventional Regression Fit
Bottom chart is	Figure 24N-6C:	Regression Fit Using Dummy Variable

day	avg temp F	kWh	CDD-50	HDD-60	dummy 1=weekday	
1	27.5	4378	0	33	0	Sun
2	30.2	5227	0	30	1	Mon
3	35.96	5080	0	24	1	Tue
4	31.1	5164	0	29	1	Wed
5	36.86	5195	0	23	1	Thu
6	32.9	5492	0	27	1	Fri
7	24.98	4633	0	35	0	Sat
8	17.42	4946	0	43	0	Sun
9	21.56	6558	0	38	1	Mon
10	30.38	5941	0	30	1	Tue
11	11.84	6901	0	48	1	Wed
12	13.46	7213	0	47	1	Thu
13	24.98	6611	0	35	1	Fri
14	35.96	4150	0	24	0	Sat
15	34.52	3808	0	25	0	Sun
16	29.48	6283	0	31	1	Mon
17	15.8	7413	0	44	1	Tue
18	31.64	6674	0	28	1	Wed
19	44.24	5765	0	16	1	Thu
20	37.22	5229	0	23	1	Fri

Figure 24N-6D. Adding a Dummy Variable to a Data Set

When Regression Just Doesn't Work

If the regression model is poor:

- Question the variables.
- Question the data.
- Learn more about the building and find a way to increase the confidence.
- If guaranteed savings is involved, either pad the estimates so there is wiggle room to float the guarantee or use a different M&V method like Method A or Method B .

Multi-Variable Regression-final notes

- Statistics experts may experiment with a dozen factors to see which ones are influential. Test by trying a variable and seeing if it helps ('multi-channel' concept). Many tests for dependence and validity
- Additional variables can improve prediction; too many can weaken it
- Correlated variables can give false positive
- Different influences are mixed together like a large soup pot.

- Where patterns exist and are identified and separated, model accuracy is improved.
- Not all buildings have a highly predictable building signature. If the model explains most of the energy use…70-80% of it, feel good.

Additional Resources

- IPMVP International Performance Measurement and Verification Protocol: Concepts and Options for Determining Energy and Water Savings, Volume I
- Excel Multiple Variable Regression and Correlation Tools
- ASHRAE Guideline 14 Measurement of Energy and Demand Savings
- ASHRAE Inverse Linear Modeling Toolkit

Regression Application 1:
Weather Normalization for Annual Energy Goal and Progress Monitoring

Energy use tracking is to begin on a building using monthly utility data, and the base year is established. Data is shown in **Figure 24N-7A** that contrasts energy use with degree days. It is required to remove the effects of weather for year-to-year comparison to evaluate progress toward a goal.

The baseline year data is scatter plotted and regressed to find a linear relationship between weather and energy use (energy use signature). **Figure 24N-7B**.

The coefficients produced from linear regression represent "m" and "b" for a line equation y=mx+b. A quality check at this point is to use these values to prove they will reproduce the annual total energy of the base year. **Figure 24N-7C**.

The building energy signature line equation is then used to normalize energy consumption for year+1 and subsequent years. **Figure 24N-7D**. The values from this step are interpreted as 'what the energy use would have been for the base building with current weather, i.e. *if nothing changed except weather (what year+1 usage should have been if nothing changed but weather)*. Then, usage can be directly compared to see if went up or down between years.

The apparent difference in electric use is

Apparent energy use change = Year+1 actual**—**Base year actual

In this example, the apparent electric use is nearly the same as the base year

The weather normalized difference in electric use is

Normalized energy use change = Year+1 actual – Year+1 normalized

With weather normalized, electric usage increased for this example.

Baseline	Raw Data, normalized to billing days			
Year	379,099	1,357	13,701	4,597
	kWh	CDD-60	therm	HDD-60
Dec	25,420	0	2991	786
Nov	27,000	0	1674	651
Oct	25,142	17	417	204
Sep	36,014	218	111	1
Aug	39,995	345	66	0
Jul	42,560	401	195	0
Jun	38,690	310	379	13
May	30,056	62	746	197
Apr	28,000	1	1050	354
Mar	29,439	3	1521	594
Feb	26,775	0	2146	896
Jan	30,008	0	2407	901

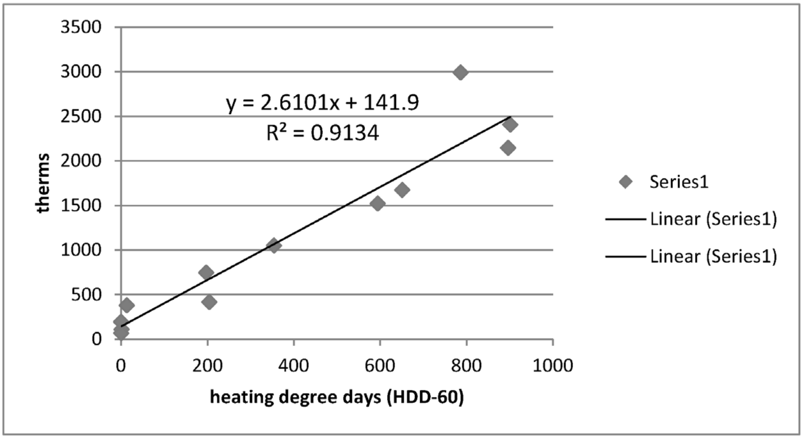
Year +1	Raw Data, normalized to billing days			
	374,900	1,065	13,532	4,864
	kWh	CDD-60	therm	HDD-60
Dec	26,300	0	3050	750
Nov	27,900	0	1458	770
Oct	26,500	15	410	233
Sep	35,211	175	88	40
Aug	36,480	288	98	0
Jul	39,357	302	202	0
Jun	36,200	225	405	8
May	29,995	56	701	250
Apr	27,500	1	1190	288
Mar	30,500	3	1480	630
Feb	28,962	0	2200	905
Jan	29,995	0	2250	990

Figure 24N-7A. Weather Normalization Data Set

Regression Application 2:
Weather Normalization for Equipment Energy Use

A school building had the boiler replaced and a 12-15% heating savings was estimated based on before/after heating efficiency. After the first year of service, the bills were reviewed and the customer has complained that savings are less than anticipated, as the total consumption is only 4% lower than last year.

This is a different application, but essentially the same series of steps shown in **Regression Application 1**. In this case, weather normalization identifies what the original boiler would have used if nothing had changed but the weather; then, that normalized data is compared to what actually was used in year +1. Old boiler running on new weather vs. new boiler running on new weather.



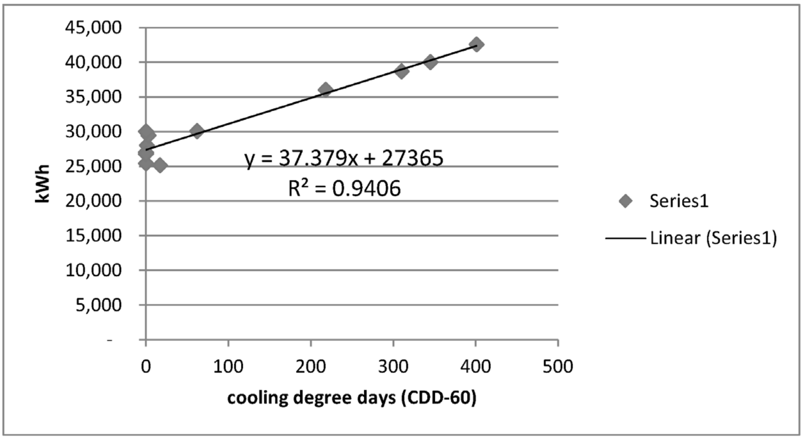


Figure 24N-7B. Weather Normalization Scatter and Linear Regression

Step 1: normalize utility data reads to calendar months for baseline and year+1, and identify the heating degree days corresponding to each month.

Step 2: scatter plot the data and create a trend line and equation for the line in y=mx+b form

Step 3: double check that the slope and intercept will accurately reproduce the baseline consumption with baseline degree days (QC, for annual total)

Step 4: normalize the baseline consumption using the year+1 data, to see what the building (with the old boiler) ***would have used*** in this weather.

Step 5: Calculate the apparent and actual percent energy savings.

Apparent % savings: **(old-new)/old**

Actual % savings: **$(old_{norm}\text{-}new)/old_{norm}$**

	sanity check heating			sanity check cooling		
	m	b	therms	m	b	kWh
Dec	2.6101	141.9	2,193	37.38	27365	27,365
Nov			1,841			27,365
Oct			674			28,000
Sep			145			35,514
Aug			142			40,261
Jul			142			42,354
Jun			176			38,952
May			656			29,682
Apr			1,066			27,402
Mar			1,692			27,477
Feb			2,481			27,365
Jan			2,494			27,365
		predict	13,701		predict	379,103
		actual	13,701		actual	379,099
			OK			OK

Figure 24N-7C. Weather Normalization Sanity Check for Building Signature

Weather normalized

	year+1 kWh	year+1 therm
Dec	27,365	2,099
Nov	27,365	2,152
Oct	27,937	750
Sep	33,906	246
Aug	38,130	142
Jul	38,653	142
Jun	35,775	163
May	29,451	794
Apr	27,399	894
Mar	27,466	1,786
Feb	27,365	2,504
Jan	27,365	2,726

379,099	13,701	base year
374,900	13,532	yr+1 actual
368,177	14,398	yr+1 normalized

Evaluation

apparent change

Year+1 actual – Base year actual

kWh	therm
(4,199)	(169)

normalized change

Year+1 actual – Year+1 normalized

kWh	therm
6,723	(866)

Figure 24N-7D. Weather Normalization Applied to Year+1 and Future Years

See **Figure 24N-8 for solution.** In this case, the boiler did save as much as intended. Simply comparing utility bills without weather normalization concealed the fact that the subsequent year was much colder. I.e. had it not been for the higher boiler efficiency, there would have been considerably more gas usage.

	Year 0 (baseline) Raw Data, normalized to billing days		Year +1 Raw Data, normalized to billing days		Normalizing
	therm	HDD-60	therm	HDD-60	therm
Dec	3591	786	3412	1022	3396
Nov	2224	651	2188	846	2936
Oct	992	204	1057	245	1359
Sep	736	1	754	1	720
Aug	666	0	600	0	717
Jul	785	0	590	0	717
Jun	929	13	883	13	751
May	1221	197	1132	197	1234
Apr	1660	354	1626	407	1784
Mar	2121	594	2089	743	2663
Feb	2776	896	2644	1075	3536
Jan	2957	901	2778	1081	3551
totals	20656	4597	19751	5630	23364

Evaluation

old use	20656	**Apparent change**
new use	19751	(old-new)/old
difference	905	4%

old use, normalized	23364	**Actual change, after weather normalized**
new use	19751	$(old_{norm}-new)/old_{norm}$
difference	3613	15%

Figure 24N-8. Weather Normalization Example 2 Solution

O— ERROR BAND: USING ENERGY CONSUMPTION SIGNATURES AS AN OPERATIONAL CONTROL

Method and Case Study
This section contributed by
Mark Imel, PE, CEM, CP EnMS

Introduction

This section will describe a method for monitoring energy performance and providing notification when energy use is outside what has been defined as normal. This can be of particular value when time and money have been expended to bring energy consuming systems into an efficient state. Without proper operational control there is a tendency to backslide and erode savings. Actions taken by facility operators can be instrumental in maintaining the efficient operations and creating savings equal to the avoided losses that would otherwise tend to occur. The method described provides a tool for operators to receive early notification that 'something is amiss', prompting action to make corrections. The method relies upon being in a desirable state of efficiency and having a reliable energy consumption signature from regression. The tool identifies what the energy consumption should be, compares that to actual consumption and then calculates an energy performance indicator (EnPI) which is a measure of how well the actual consumption compares to predicted consumption. The method monitors the EnPI over time and establishes boundaries which identify proper operation. When the EnPI exceeds the established error band, action is prompted to find out why and make a correction. The simplicity of this method makes it easier to implement in more facilities, rather than rely on a high degree of automation and technical expertise which can be expensive. The method will be demonstrated by way of an actual example.

The method presented here is relatively simple and effective when the consumption of a building or system can be well represented through a regression model. It incorporates the techniques used by industrial energy managers by looking at the way a building consumes energy as a process. The method can also be used as part of **the Plan-Do-Check-Act**, continuous improvement model incorporated in the international standard ***ISO 50001, Energy Management Systems.*** In the following example one will see how the monitoring tool was used to continually ***Check*** performance and that once improper operation was noted, facility personnel were able to ***Act*** to correct the problem. Throughout this section terminology that is part of *ISO 50001* will be bold and in italics.

Project Background

This example involves an energy retrofit project in southern Florida which included four office type buildings and a central chilled water plant that served three of these buildings. The first task was to conduct an ***energy review***. Electricity was identified as the only ***energy source.*** There are a total of five electric meters, one for each building and the chilled water plant. Review of the annual ***consumption*** for the five meters showed that the chilled water plant accounted for over 30% of the total electric consumption of the campus. The chilled water plant (e.g., cooling) was identified as a ***significant energy use (SEU)***. Once an SEU is identified the standard has requirements such as operational control of the SEU, monitoring and improving performance. A number of energy conservation measures (ECM) were implemented during the project. Some ECMs were in the plant, while others reduced the cooling load in the buildings, thus resulting in energy savings at the plant.

Baseline Development

Data for the ***baseline*** included one year of monthly electric bills following the retrofit project. The ***relevant variables*** chosen for the regression model included number of days (NOD) in the billing cycle and the cooling degree-days (CDD) associated with those days. The linear regression did not include a constant but rather the coefficients associated with the two input variables. This was done to make the model more flexible so that it could be used for various periods of time (e.g., monthly, weekly, bi-weekly). The R^2 for the resulting model was 0.9. The form of the regression equations follows.

$$\text{Consumption (kWh)} = (41.65\ \text{kWh/CDD}) * (\text{CDD}) + (1{,}062\ \text{kWh/day}) * (\text{NOD})$$

Where:

Consumption kWh = the electrical usage for this period
kWh/CDD = the regression coefficient for the variable CDD
CDD = the cooling degree-days for the period
NOD = number of days in the period

Note: This equation is in a slightly different form then the more common y = mX + b with X being the input variable (e.g., CDD), m the coefficient applied to the *input variable and b the constant or intercept. Using the monthly data the resulting regression equation in this form becomes y = (43.61 kWh/CDD)*CDD + 31,477 kWh. Note the value of m is similar in the 2 variable and 1 variable models which are 41.65 kWh/CDD and 43.61 kWh/CDD respectively. The NOD coefficient in the 2 variable regression equation is 1,062 kWh/day. On average there are 30 days in a month, so using a value of 30 for NOD multiplied by the NOD coefficient results in a value of 31,860 kWh which is very close to the constant of 31,477 kWh for the 1 variable regression equation. In other words the two models give similar results when applied to monthly data. However, by using the 2 variable with no constant approach the model now becomes more flexible and could be used for weekly, bi weekly, monthly or any other time frame.*

Figure 24o-1 is a trend graph which compares the predicted consumption to the actual consumption. One can see the actual and predicted consumption values compare well. **Figure 24o-1** also introduces another value which is an ***energy performance indicator (EnPI)***. *ISO 50001* defines an EnPI as a "**quantitative measure of energy performance**." In this case the EnPI is simply the ratio of the actual to predicted consumption. A value of 1.0 means that the plant consumed exactly what the model predicted. A value of 1.1 would mean the plant consumed 10% more than the model predicted.

Introduction of the Control Chart

The model has shown that consumption can be predicted reliably. An industrial energy engineer would call this a stable process. One industrial technique to monitor the stability of a process is to use a control chart. In this case the control chart is created by taking the 12 monthly values of the EnPI from the baseline period and calculating the **mean as well as plus or minus three standard deviations**. These three values are plotted as straight lines and then the individual EnPI values plotted as shown in **Figure 24o-2.**

There are a number of rules when using this type of chart which indicate whether the process is, or is not in control. The most basic one is that the **values do not fall outside of the plus or minus three standard deviation range. Figure 24o-2** shows that all 12 EnPI values are within the band thus indicating the plant is in control.

Using the Monitoring Tool to *Check* Performance and *Act* when a Fault is Detected

Figure 24o-2 shows the plant performed well during the first 12 month after the retrofit. **Figure 24o-3** continues the control chart for six

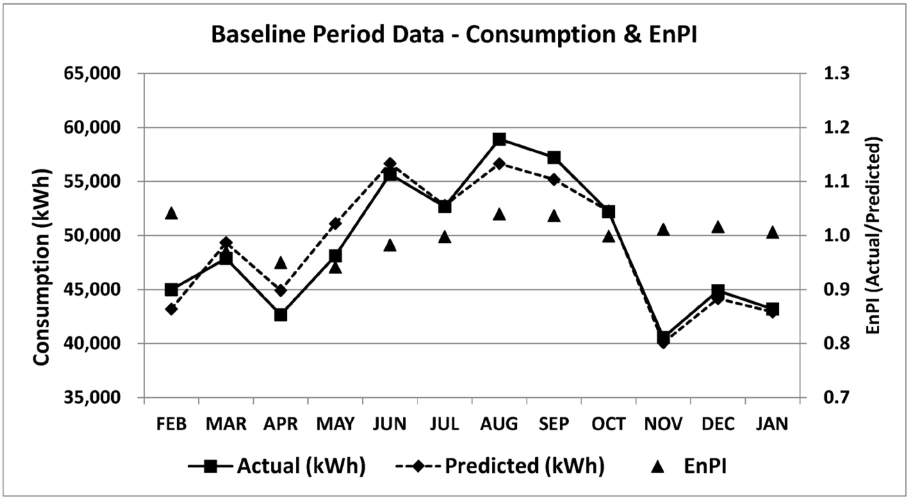


Figure 24o-1. Baseline Period Data

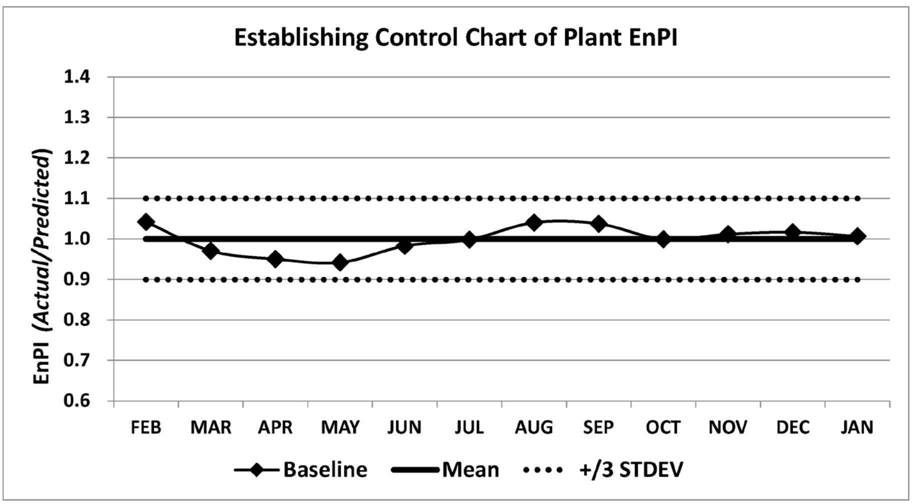


Figure 24o-2. Baseline Period Control Chart

months beyond the baseline period. The EnPI value for February jumped to 1.19 indicating the plant consumed approximately 19% more energy than the regression signature predicted it would. With a good regression model, this is interpreted as the building or process using more energy than it should be, which is a notification or alert. The control chart served

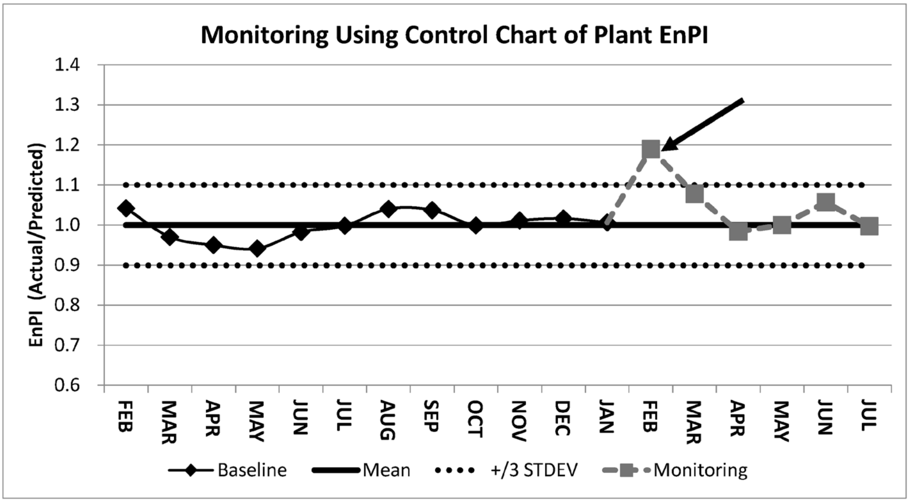


Figure 24o-3. Monitoring Performance with Control Chart

as a tool to *Check* performance.

Facility personnel were now able to *Act* and try and resolve the problem indicated by the jump in the February EnPI. Review of the control system uncovered an inaccurate temperature sensor that was causing the chilled water pumps to run at 100% all of the time. The error was corrected on 14 March. The March EnPI in **Figure 24o-3** was 1.08 meaning it had dropped by 0.11 *(over halfway)* from the February value back towards a stable value of 1.0. This is consistent as proper operation occurred for slightly over half of the month of March. During the following four months the EnPI returned to values close to 1.0 indicating the plant was back under stable operation.

Shortening the Monitoring Period

The regression model used to develop the EnPI and control chart was created to be flexible and allow for varying time periods. The owner of the facility saw the value of performance monitoring and wished to shorten the period to one week. Each Monday morning facility personnel log in to the building control system which displays the readings for all 5 meters. These values are entered into an Excel spreadsheet along with the date. The spreadsheet automatically calculates the NOD value and has a link to a weather website where the CDD value is found and then entered into the spreadsheet. Once again, the model used for this one week performance monitoring period is the same one used for the monthly tracking discussed above. **Figure 24o-4** shows an example of the one week period EnPI control chart.

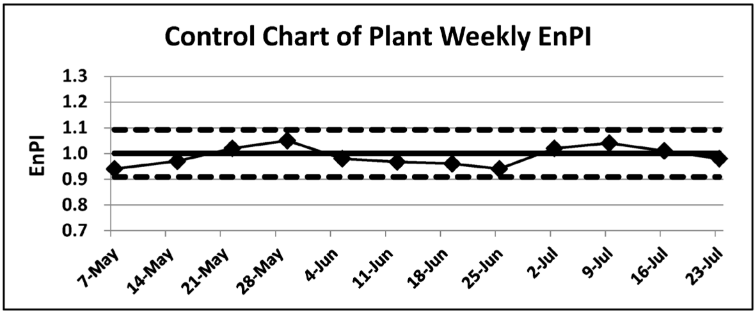


Figure 24o-4. Control Chart of Weekly Plant EnPI

DETAILS—CONCLUSIONS—OTHER APPLICATIONS

Developing the Case Study Tracking Tool

Data required for development of the baseline included actual monthly energy consumption, the NOD associated with each consumption value and the CDD realized during the period of time. Monthly utility bills provided the consumption and meter read dates associated with the bill. These dates were used to determine the NOD. The dates were also used as input to a weather website to gather the CDD data. All of the data were entered into an Excel spreadsheet and used as input to the program's *Data/Data Analysis/Regression* built in functions. The consumption represents the Y-axis data or dependent variable. The NOD and CDD represent the X-axis or independent variables. The equation constant was set to zero. The R^2 for the resulting model was 0.9. The form of the regression equations follows.

Predicted Consumption (kWh) = (41.65 kWh/CDD) * (CDD) + (1,062 kWh/day) * (NOD)

This equation was used in an additional column of the spreadsheet table to calculate the predicted consumption for each month. The final column in the table included the monthly EnPI which is the ratio of the actual to predicted consumption. The mean and +/- 3 standard deviations of the 12 baseline EnPI values were plotted to form the control chart which identifies the range of values within which future EnPI values should fall.

Using the Case Study Tracking Tool

In the case study the tool was used to monitor performance on a monthly and weekly basis. **Table 24o-1** is the table from the spreadsheet tool used to input monthly data, calculate results and generate all graphs associated with the case study. This version includes 24 months, the initial 12 post-retrofit months used to develop the baseline and EnPI control chart, and an additional 12 months which begins the performance monitoring period.

Each month the user inputs the NOD and actual consumption from the utility bill. The user then takes the dates from the utility bill and gathers the corresponding CDD for that period from the internet. After entering the CDD value the predicted consumption and EnPI are calculated by the tool. The process takes approximately 10 minutes. In this case six months of monitoring have occurred using the control chart. Improper operation was flagged for the first month during the monitoring period when the EnPI reached 1.19.

A similar process was used for the weekly tracking of the EnPI. In this case there is not utility bill. Instead the user reads the electric meter from the building automation computer station. The user enters the date and meter reading. The NOD value is calculated by the tool. The user gathers the CDD value from the internet and enters into the tool.

Maintaining the Case Study Tracking Tool

The tool is easy to maintain as long as the building or system continues to follow the same consumption pattern captured in the baseline regression analysis. The primary reason for making a change is when the consumption pattern develops a significant and continuous deviation from the baseline consumption pattern. In other words, say an office building is monitored with this tool. Because of an increase in work load a second shift of workers is added for a two month period. The control chart would indicate an increase in the EnPI for this period that may exceed the upper limit. However, if it is known that the second shift will end as scheduled, and normal operation will return, there is no need to change the tracking tool. It has simply fulfilled its purpose and indicated that consumption was higher than normal. The operators note this, document the reason (e.g., the temporary second shift), and continue to use the original tracking tool equations and control chart.

If on the other hand changes effecting the consumption pattern are significant and permanent, the tool should be modified. This is accom-

Table 24o-1. EnPI Tool Input/Output Table

	USER INPUTS			CALCULATED		
Month	**NOD**	**CDD**	**Actual (kWh)**	**Predicted (kWh)**	**EnPI**	
FEB	30	272	45,000	43,193	1.04	BASELINE PERIOD
MAR	32	369	47,880	49,346	0.97	
APR	29	339	42,670	44,916	0.95	
MAY	30	462	48,120	51,105	0.94	
JUN	32	545	55,680	56,672	0.98	
JUL	29	528	52,680	52,778	1.00	
AUG	31	570	58,920	56,651	1.04	
SEP	31	535	57,240	55,193	1.04	
OCT	30	490	52,200	52,257	1.00	
NOV	30	198	40,560	40,097	1.01	
DEC	31	270	44,880	44,157	1.02	
JAN	30	266	43,200	42,929	1.01	
FEB	30	236	49,560	41,663	***1.19***	MONITORING PERIOD
MAR	31	170	43,080	39,980	1.08	
APR	30	416	48,360	49,176	0.98	
MAY	30	434	49,920	49,925	1.00	
JUN	32	550	60,120	56,879	1.06	
JUL	32	525	55,680	55,838	1.00	
AUG						
SEP						
OCT						
NOV						
DEC						
JAN						
To Date	***548***	***7,175***	***895,750***	***882,753***	***1.01***	

plished by adjusting the baseline (regression model/energy signature) and control chart operating boundaries. This is consistent with ISO 50001 which requires that baselines be adjusted when *"EnPIs no longer reflect organizational energy use or consumption,"* or *"when there are major changes to the process, organizational patterns or systems."*

Automating the Tracking Tool

Examples given were made using data that was manually input into Microsoft Excel. Once built, the tool is sustained with periodic doses of new data. For monthly data, this involves only a few data points and is a minimal time investment. The more granular the data and the more frequent the input, the earlier discovery of actionable information, however the additional effort would likely be a barrier to ongoing usage of the tool. Automatic data input updating of the chart is a natural goal; things that are easier to use get used more and used longer. The visual format of the tool as presented is very similar to standard trend charts in energy management and control systems, which standardly allow trending of hardware points as well as virtual (calculated) points against time. The challenge with incorporating the EnPI/error band tool within an energy management system is automatic input of the additional parameters beyond hard wired sensors, such as energy use and degree-days. Energy use is already available in many EMS systems from sub meters or utility meter data feeds, and degree-days can be calculated for each site by the same method weather stations do it: measuring the high and low values for the day with good quality instruments and averaging them. These are not the only two data points for all EnPI charts, but the point is that with a little ingenuity the value potential of the tool is not out of reach of automation.

Control Charts with Volatile EnPI Patterns

In many cases the baseline and resulting control charts are much more volatile than shown in the case study. This can occur if the time frame is very small or if the building or system has many interactions that are difficult to capture with the regression. As an example we will consider an office building which has both electric cooling and heating. A daily regression model was developed using CDD, heating degree days (HDD) and a dummy variable indicating whether it was a weekday or weekend.

Figure 24o-5 (top) is the control chart for this model over a year long period. Note that due to the volatile nature of the daily values the control boundaries range from +/- 0.25 from the mean. Even though this is a broad range, a number of days are either near, or outside the boundaries. Using a ***moving average*** of the daily EnPI values can help reduce volatility of the plot and tighten the control band. **Figure 24o-5 (middle)** is variation of the control chart using the same data. Starting with day 7, a seven day moving average of the EnPI is used. The control band has tightened to +/- 0.134 about the mean.

Taking the process further a 30 day-moving average can be used. **Figure 24o-5 (bottom)** shows the same data charted using a 30-day moving average of the EnPI. The control band has tightened further to +/- 0.106 about the mean and the curve is even smoother.

Limitations of Control Chart/Error Band Concept

The most significant limitation when using this methodology is cases where a good regression model cannot be established. A model that cannot represent the consumption pattern well will result in such a wide band of what is considered an *"in control"* EnPI that it will have little functionality as a monitoring tool. For instance if the resulting control range was +/- 0.4 about the mean, that would indicate a building or system could have an EnPI that ranged from 0.6 to 1.4 and still be *"in control."* In other words the actual consumption could range from 60% to 140% of the predicted value, thus providing little value for assessing performance.

Another limitation or challenge in using this method is determining

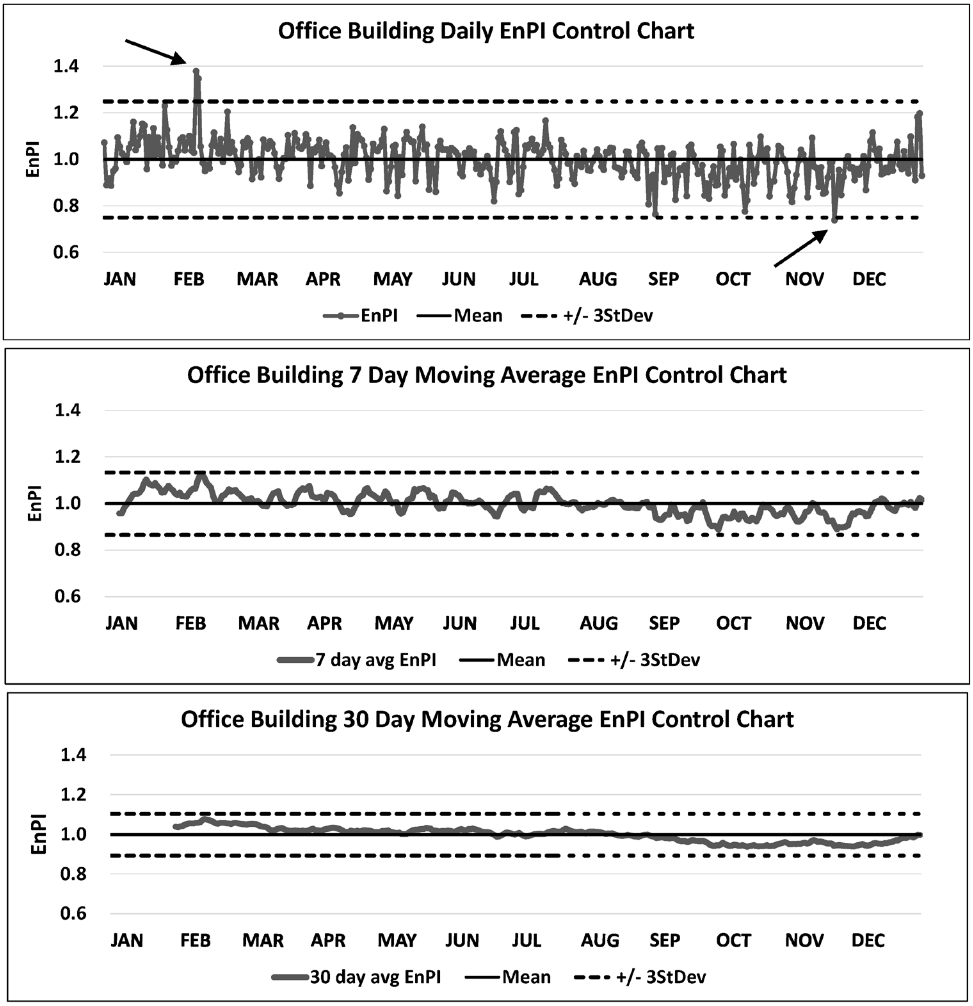


Figure 24o-5. Control Chart Options for Volatile Data (Office Building Example)

the relevant variables that effect energy consumption and being able to reliably gather past and future data for these variables. Weather data are readily available from numerous websites and other sources, and can be measured locally. Building or system specific data may be harder to gather. Examples include hours of operation, occupancy, production quantities, etc.

Additional Topics and Resources

Superior Energy Performance (SEP)—This is an ANSI certification for energy performance of industrial facilities. SEP requires the organization to be certified to ISO 50001 as well as meet strict, quantifiable improvements in energy performance. The SEP has an M&V protocol with specific requirements on how baselines and EnPIs are developed and used. The Department of Energy Advanced Manufacturing Office has available a free, spreadsheet based EnPI tool that facilitates developing and using regression models.

Institute for Energy Management Professionals (IEnMP)—This is an organization responsible for granting and maintaining certifications relative to the SEP program. These include Certified Practitioner in Energy Management Systems (CP EnMS), SEP Performance Verifier (SEP PV) and SEP Lead Auditor (SEP LA). Information on training and certification is available from the IEnMP.

References:

ISO 50001:2011 *Energy Management Systems—Requirements and Guidance for Use*, International Organization for Standardization, 2011

P—INFORMATION FROM INTERVAL DATA

Source: Information from Interval Data, paper presented at WEEC (World Energy Engineering Congress), Washington, DC, 2013, Association of Energy Engineers

Interval data is the term used to describe energy use snapshots taken more frequently than traditional monthly billing reads. Interval data records are commonly 15 minutes apart, but can be any increment such as 5 minutes, 60 minutes, or daily reads. The shorter the interval, the more the data approaches true analog; the longer the interval, the more assumptions are applied to the blank spaces in between reads. Interval data is usually applied to electricity flows, but can be applied to natural gas, other fuels, and water. The granularity of interval data gives energy professionals new tools and insight into how and when energy is used, leading to new ways to manage it. The advantage of interval data over conventional monthly utility reads is fewer blind spots in the data and a more accurate view of how and when energy or other flows are used. The beauty of interval data is the prompting of new questions that would have otherwise gone undetected. The how/when leads to "why," "what can we do about it," and savings. Viewing actual usage patterns allows comparison of actual to expectations, schedules, nighttime operation, and so on. Persistent usage during zero occupancy or zero production times can be viewed and challenged as a controllable cost.

Interval data is time/date stamped, and typically comes in blocks of time after the fact, displayed in a long list of records. Although not real-time, interval data shows very accurate load profiles that are not possible with monthly reads—this means knowing the usage at 3pm *and 3am*, rather than guessing. Peaks, ghost loads, and anomalies—usage that shouldn't be there—become apparent.

Evaluating interval data can lead to opportunities that may otherwise go unnoticed. Examples and case studies are presented to show ways to turn the data into useful information.

Information is power, and interval data holds the promise of better information. Some ways that energy professionals can use interval data to advantage are:

- Clarity of usage profile
- Identify when peak usage occurs
- Confirm when energy end uses occur
- Validate building use schedules
- Identify anomalies
- Detect automatic control dysfunction
- Improve load factor by distributing loads
- Commissioning for energy management
- Measurement and verification

Interval data has become available to more customers, who can then use it to advantage. Using the utility reads as the source of the data, there is no need to purchase or maintain any equipment, making it easy to start using interval data.

Comparison to Real-Time Data

- Interval data is in data logger format; granular and historical. Real-time is "right now," no delays, instant gratification. There are things real time data can do that interval data cannot do, especially when applied as sub meters nearer to the load. Even the best utility meter on the main feed of a building is only good for Method C in IPMVP (1).

- Automated processes like demand shedding, automatic notifications, etc. require real time data, so they can make a choice and act "now"—interval data that is after the fact would not be suitable for this.
- Real-time measurements can be obtained from utility meters by the addition of a 'pulse output', a fee-added service. Here, the customer receives a stream of pulses, or some other output, that represents rate of consumption. Some advanced metering systems allow the rate-of-use data to be streamed to the customer over IP (internet) rather than hard wired.
- Real time data can also come from privately owned meters. The advantages are putting them wherever you want, and eliminating any limitations of the data inherent in utility-provided data. The disadvantages are going into the meter business: cost, ongoing calibration (yes), and data management.

Data to Information

The real challenge is putting interval data to work to operate smarter and save money. The term "interval data" is most commonly associated with periods of minutes or hours. This allows daily load profiles to be seen—when and how much usage characteristically occurs. **Figure 24P-1** shows examples of load profiles over a 24 hour period. The x-axis shows the number of records.

Figure 24P-2 shows typical interval data format in a spreadsheet. A year's worth of data in 15 minute intervals is over 35,000 records. Because of the volume of data, some different approaches are needed to express visually. Monthly readings are easily expressed with a graph for total kWh or maximum kW demand per month. **Figures 24P-3 and 24P-4** show the information advantage of interval data compared to monthly reads; also pointing out that the monthly utility read represents the *end* of the period. For sequential monthly reads, the annual pattern is established and useful; interval data fills in the gaps in between for a closer look.

Figure 24P-4 shows a 1-month period comparing interval and monthly maximum kW readings. No surprise that a single event pushes up the

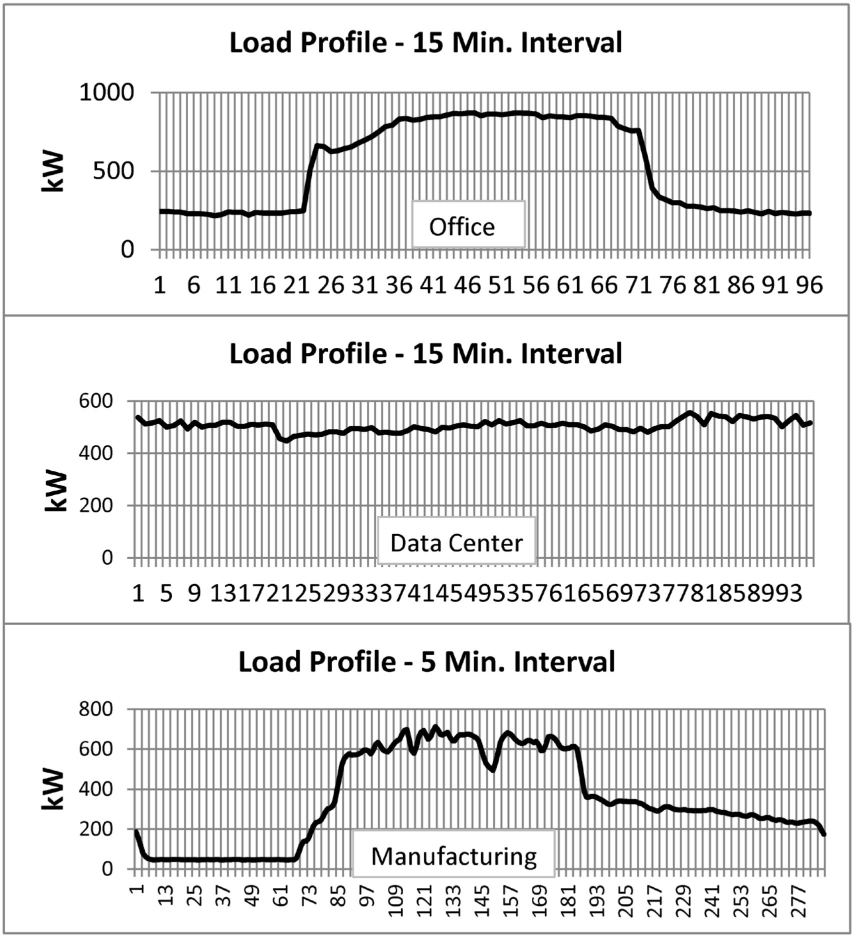


Figure 24P-1. Load Profile from Interval Data

maximum billed demand for the month. However, the interval data tells when it occurred which is very useful when you go looking for it.

Figure 24P-5 shows a 3-month period of water use, comparing daily readings to calculated daily average use (monthly use divided by # of days), and showing that errors from presumed steady use between any two points can be large. The beauty of interval data is eliminating blind spots in the data—which eliminates guesswork.

At intervals of one hour, numerical values of kW and kWh are the same (1 kW over 1 hour = 1 kWh) and are converted according to the time

DATE	HOUR	IN	UN	KW	KVAR	kWh	PF
12/14/10	0015	15	KW	262.80	165.66	65.70	0.8460
12/14/10	0030	15	KW	260.82	158.76	65.21	0.8542
12/14/10	0045	15	KW	252.00	169.74	63.00	0.8294
12/14/10	0100	15	KW	233.16	152.70	58.29	0.8366
12/14/10	0115	15	KW	231.48	166.26	57.87	0.8122
12/14/10	0130	15	KW	226.20	166.02	56.55	0.8062
12/14/10	0145	15	KW	194.04	144.84	48.51	0.8014
12/14/10	0200	15	KW	197.04	165.66	49.26	0.7654
12/14/10	0215	15	KW	183.24	157.50	45.81	0.7584
12/14/10	0230	15	KW	178.20	154.32	44.55	0.7559
14/10	0245	15	KW	188.82	168.36	47.21	0.7464
		15	KW	178.26	150.18	44.57	0.7648
				184.80	160.74	46.20	0.7545
					34	46.71	0.7509
							0.7612

Figure 24P-2. Sample Interval Data Format

increment in use (1 kW over 15 minutes = 0.25 kWh, 1 kW over a full day = 24 kWh). When increments of time are 1 hour and less, power (kW) and energy (kWh) units are close and viewing only kW is common; at this level of granularity, insight into usage patterns is evident, such as daily peaks, occupancy schedules, shift changes, large batch processes, weather effects, and residual energy use when the building is unoccupied. This level of data can also be used to measure benefits of before/after 'tests' such as an experimental process or automatic control routine. Measurement and verification activities can benefit from interval data and can use any combination of hourly, daily, or monthly data as needed.

Unfortunately, even a well constructed building with prudent occupant habits can still consume large amounts of energy when the building is closed or production has stopped. A guiding light for energy management is using energy only when it is needed, and interval data allows viewing of the hours when the energy use is presumed to be "off." **Figure 24P-6** illustrates how energy use can be curbed by focusing on hours when the facility is closed. Sometimes the unoccupied energy use is unavoidable and becomes an overhead cost; sometimes not. Interval data will at least point to the issue and prompt the question "why." Note that most commercial/industrial electric rates have much lower cost in off peak periods such as nights and weekends, and may lead to indifference; it is inexpensive to waste during these times.

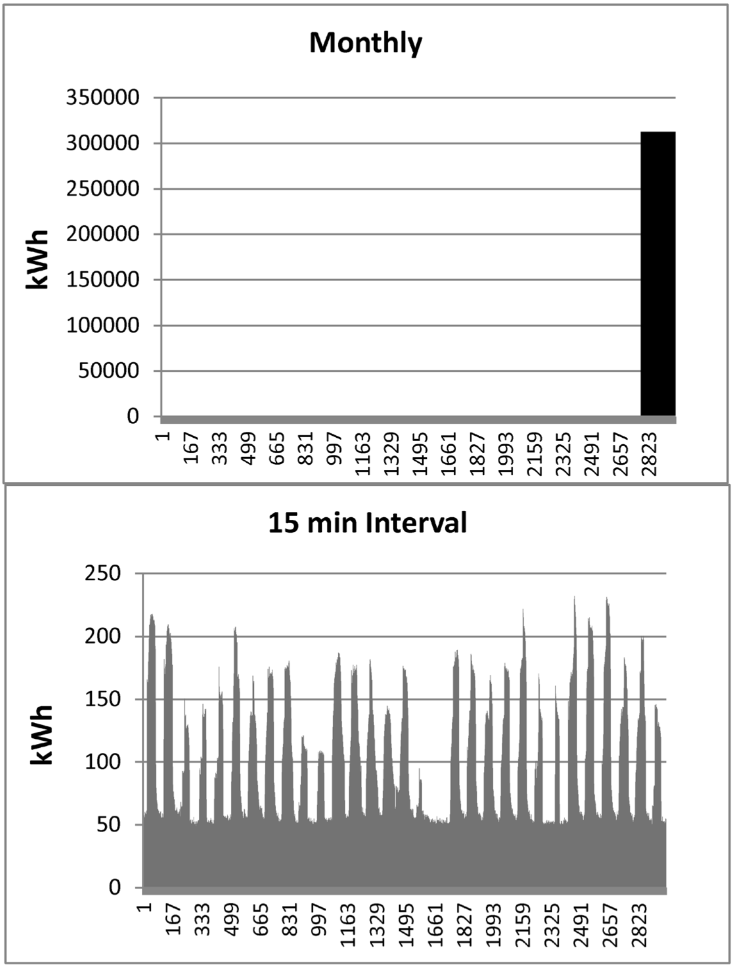


Figure 24P-3. Monthly vs. Interval Data (kWh)

Graphing Interval Data

Graphs are a good way to convert data to information. The main consideration is the interval of time that is of interest, i.e. patterns of use within a day, week, etc. See **Figure 24P-7**. Other considerations:

- Line or column charts both work.
- For small intervals (one hour or less) kW and kWh are nearly synonymous, so graphing kW is reasonable.
- For long periods of time, graphs will get wide (time on the x-axis);

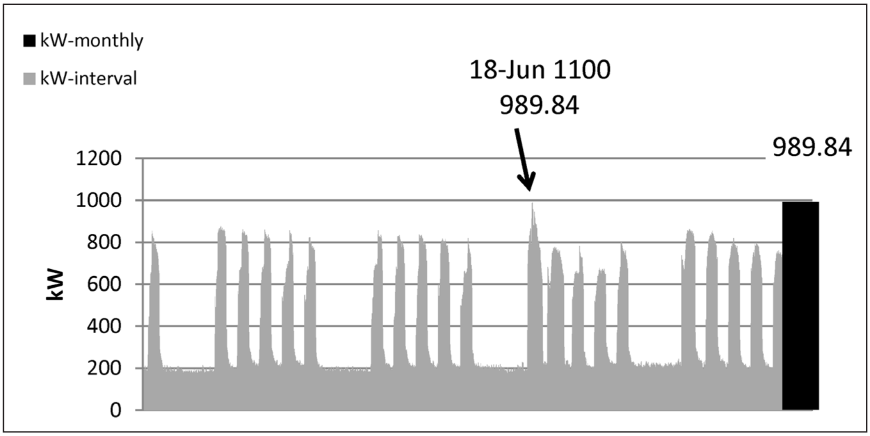


Figure 24P-4. Monthly vs. Interval Data (kW)

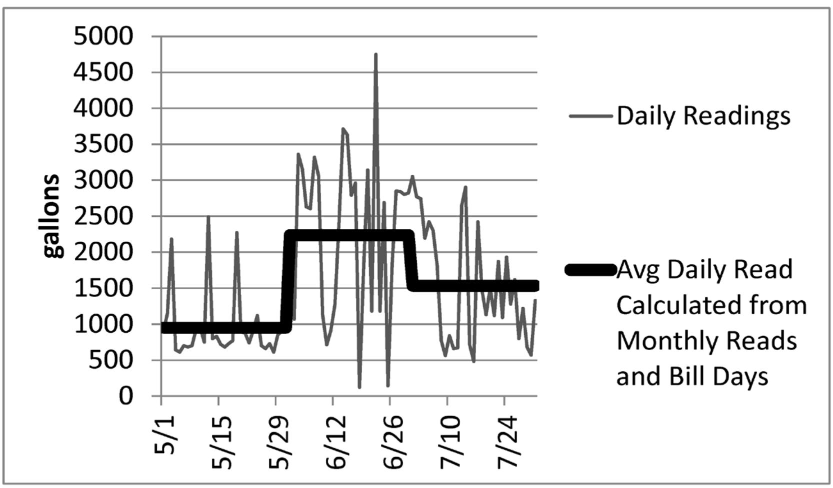


Figure 24P-5. Daily Avg. vs. Daily Readings

fine for personal review, not fine for presentations or reports. One solution is a sequence of progressively narrower time bands so the detail appears; e.g. a month, then a week, then a day. Another solution is a typical week(s).

- When too compressed, the data is a mess. An annual graph formatted to fit on a standard sheet of paper obscures some information.

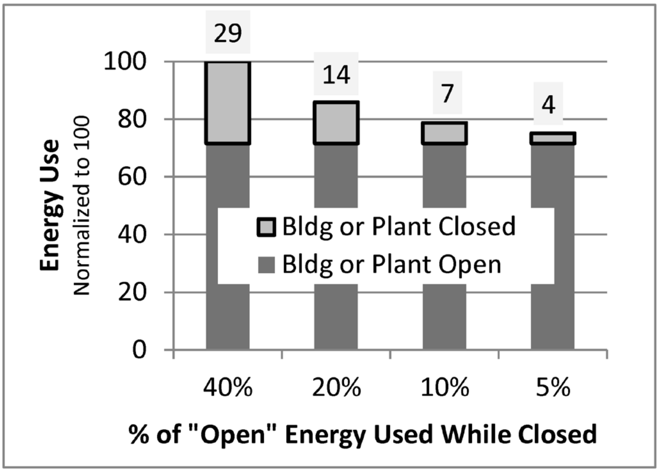


Figure 24P-6. Curbing Off-Hours Energy Use

- Dividing lines for days are useful when the time band is a week. This can be done by adding a new record beside the time slot of midnight for each day—this is shown in the lower left portion of **Figure 24P-7**. Days or months can be named if desired.
- For most graphs, the time/date stamp of the record will be illegible on a graph.

BASIC USES OF INTERVAL DATA:

Key for all interpretation of interval data is comparing what is seen to what is expected. Energy uses in most facilities follow some kind of pattern. This may include weather, controls, or production. Anything that doesn't fit the expected pattern prompts the question.

- Quick checks. For non-manufacturing customers, data snips can show two winter and two summer weeks with weekends but no holidays. This provides a glimpse of the building usage profile (two successive weeks should look the same) and other functions evidenced from energy use. See **Figure 24P-8**. For manufacturing customers, blocks of time can provide a visual display of typical weeks, shift changes, seasonal production changes, product line changes, and idle times.
- Schedules, setbacks, etc. Again, compare what is supposed to be happening to what is actually happening. Overlay actual usage with building occupancy patterns to see if they match, since you can lit-

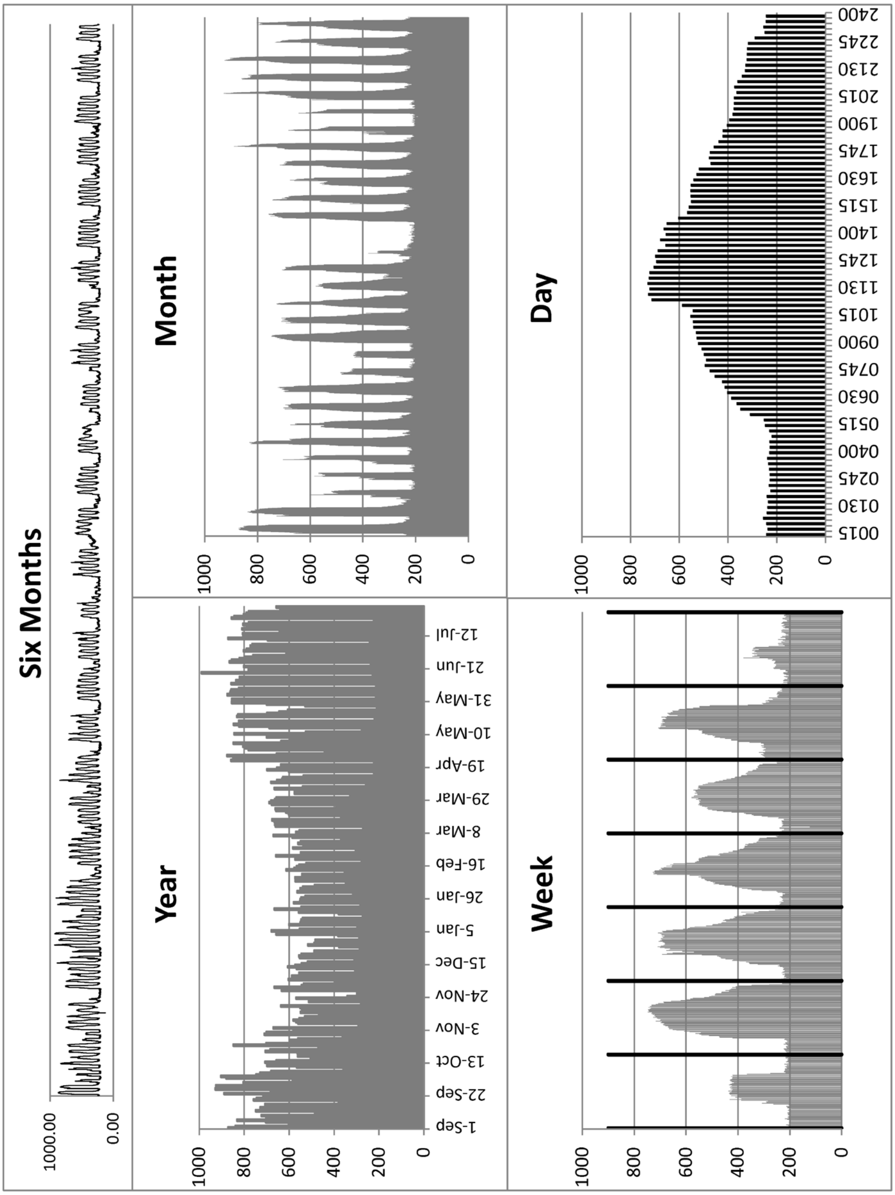


Figure 24P-7. Time Segment Choices Graphing Interval Data

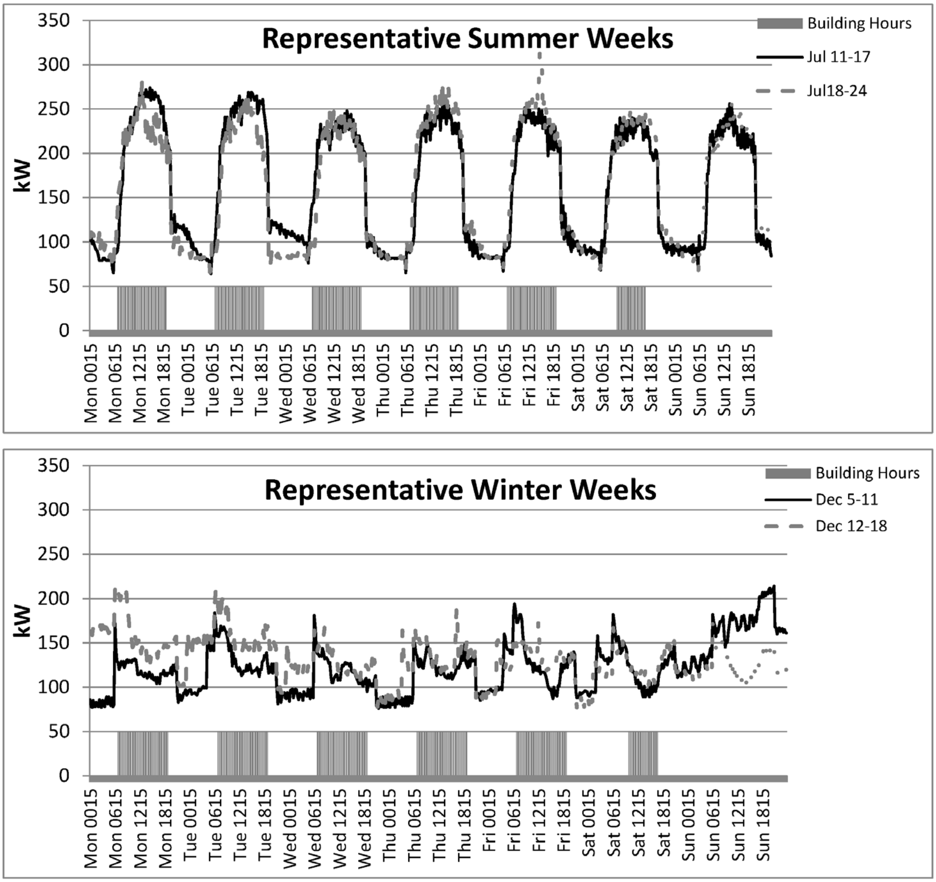


Figure 24P-8. Representative Summer/Winter Weeks, Non-Manufacturing

erally see large loads start and stop. In many cases, control system schedules can be tightened.

- Anomalies. If errant usage is discovered at certain times of day (say a weekend or middle of the night), interval data can prompt the question "what is that?" Once corrected, follow-up review of additional interval data can verify the benefit of the change, and ongoing monitoring can be the 'watch dog' to make sure it doesn't creep back.
- Residual loads. Identifying usage during periods that the building or manufacturing facility is closed is the first step in reducing it. Viewing energy as an ingredient, all energy use that occurs with no productivity benefit is waste, and directly affects product cost and

profit. Imagine cutting off 30% of the raw steel stock at a factory receiving dock and putting it directly into the dumpster. The question is whether zero productivity overhead energy use is unavoidable, or just unnoticed. The analysis of idling or ghost loads can teach us a lot.

Additional metrics and patterns made possible by interval data:

- Usage for time periods coinciding with specific manufacturing processes or batches
- Comparative usage for same period last year
- Usage pattern compared to daily temperatures
- Load factor patterns
- KVAR patterns for power factor correction
- Typical days of the week (typical Monday, etc.)
- Utility burn rates in dollars

Other Uses for Interval Data

- Energy management measurements involving 'before and after' can use interval data. For example, a '*Wednesday is turn off your lights day*' campaign can graphically show the results and encourage occupants to participate.
- Measurement and verification activities can make use of interval data as any other data logger, both to verify results and to establish baseline usage.
- Utilities use interval data for a clearer view of customer use patterns, contributing to rate designs that distribute costs more fairly. Usage patterns and price signals help optimize assets and delay costly upgrades.

Interval Data Limitations

- Comparing usage between intervals requires the same span of time for each one, and poor conclusions can result when this is not the case. For example, if a 'daily' read is really 40 hours (instead of 24), usage on that day will look like a 'spike'. Unless the uneven time interval is noticed, conclusions and investigations and changes can ensue, chasing something that isn't there. When consumption data time intervals are inconsistent and usage is reasonably steady, the data can be normalized with the ratio of nominal to actual time span. For the example in **Figure 24P-9**, daily usage record #3 would be multiplied by the factor (24/43.8).

Repeating, normalizing in this manner is inappropriate when usage is inconsistent. For example, if reading #2 (**Figure 24P-9**) is water usage and the 4-hour captured block of time occurred during an irrigation event, normalizing to 24 hours will overestimate usage presuming steady use for 24 hours. Time normalizing is only applicable to cumulative data like kWh, cubic feet, therms, gallons, pounds (kG), Btus; and not applicable to instantaneous values such as kW, kVA, kVAR, power factor, or rate-of-flow data.

Granularity of data is very insightful but can also be misleading. A single anomaly may be a reading error or something meaningless. The saying 'don't fall in love with data' applies (2).

Reading	Date	Reading	time span hours	usage
	Jun 19 11:43 PM	156087		
1	Jun 20 11:57 PM	156401	24.2	314
2	Jun 21 04:03 AM	156492	4.1	91
3	Jun 22 11:52 PM	157110	43.8	618
4	Jun 23 11:52 PM	157401	24.0	291
5	Jun 24 11:32 PM	157755	23.7	354

Figure 24P-9. Effect of Variable Time Intervals

Interval Data vs. Monthly Utility Data

- Monthly readings are the basis of monthly billings and interval data does not replace them. Basic metrics of utility cost can be derived from interval data with some data manipulation, such as on/off peak, maximum demand, and coincident power factor. Where ratchet charges exist, interval data may be the only way to identify when the event occurred that created the lingering charge.
- Interval data easily aligns with the first and last day of each month, whereas monthly utility read dates can be any day of the month. If the bill date is May 15, does that block of usage represent May or April?
- Interval data allows aligning calendar and weather data, for evaluating weather influence. Instead of degree-days for the month (…was that May or April?), average temperatures can be compared directly with daily usage.

Using Interval Data to Estimate Dollar Savings

The granularity of interval data is not directly comparable to utility costs derived from traditional monthly bills. Monthly utility bills consist of multiple charges including fixed charges regardless of use, consumption, demand, time of use, and power factor. Provisions like block rates and ratchet charges complicate matters further. Reproducing a monthly bill from interval data is complicated. Blended rates (monthly total dollars divided by monthly total consumption) are convenient, but not always appropriate—for example, dollar savings from large energy reductions in off-peak times will be over-stated using blended rates, and reducing off-peak demand may not reduce the utility bill at all.

References

(1) International Performance Measurement and Verification Protocol, Concepts and Options for Determining Energy and Water Savings—Volume 1, 2012, www.ipmvp.org
(2) Five Simple Principles of Modelling, Pidd, M. Simulation Conference, 1996, winter proceedings, pp.721-728

CASE STUDIES

Case Study #1. Office Building

Office space. Lighting: T8 fluorescent, approx 1.25 W/SF. HVAC: VAV, DX cooling, air-cooled packaged rooftop, natural gas-fired morning warm-up, and electric resistance heaters in VAV boxes. Controls: DDC for HVAC, interior lights manually controlled, parking lot lights on a timer. Building hours: 7:30am-5:30pm, M-F, closed on weekends.

Review of Interval Data

Ref. **Fig. 24P-Appx 1.1** Data is from mid-December, cold weather.
Major equipment is being turned off and on consistent with building occupancy times, represented by light colored boxes.

Observations:

A. Building is being operated on Saturday. Not sure if this a pattern or anomaly/tenant request

B. Appears that electric heat spikes at the beginning of the occupied period, and then settles down. More aggressive use of morning warm-up (gas heat), with a few degrees of overshoot or balancing the warm-up air flows to favor the perimeter areas, will probably eliminate this high spot.

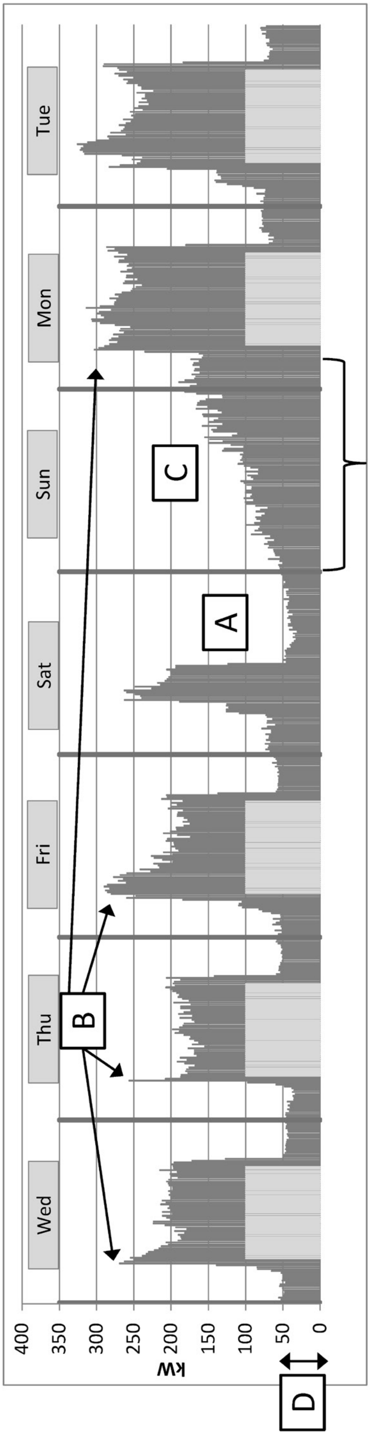


Figure 24P-Appx 1.1. Interval Data and Inferences for Case #1

C. Appears that "off" command stopped at the main HVAC units but the VAV boxes with fans and electric heaters took over, with energy use approaching occupied status much of Sunday and until Monday opening time. Any heating required in unoccupied periods should be handled by the gas heat unit since it is uses a less expensive energy source.

D. Unoccupied period electric use is consistently ~50kW, about 15% of max usage. This occurs with the building completely unoccupied and dark. There is no data center.

Recommendations:

1. Adjust scheduling controls for Saturday "off."
2. Fuel switching. Adjust temperature controls so that only gas heat is used in unoccupied periods.
3. Adjust morning warm-up controls to exceed occupied temperature by a few degrees to avoid the spike at the beginning of occupied periods.

Case Study #2. Large Office Building

Office space, with no data center. Lighting: T8 fluorescent, approx 1.1 W/SF. HVAC: VAV, chilled water, cooling tower, natural gas-fired morning warm-up, electric resistance heaters in VAV boxes. Controls: DDC for HVAC, interior lights manually controlled, parking lot lights on a photo cell. Space temperatures are set back 10-degF in unoccupied mode, and all electric heat is locked out so all unoccupied heating comes from gas heat. Operating hours: 6:00am-7:00pm M-F, 8:00am-5:00pm Saturday, 9:00am-3:00pm Sunday.

Review of Interval Data

Fig. 24P-Appx 2.1 shows interval data for summer and winter, for contrast. An additional winter graph (**Figure 24P-Appx 2.2**) shows electric data overlaid with daily natural gas usage.

Observations:

A. Load profile shows unoccupied usage ~300 kW during summer (no electric heat), about 30% of maximum demand. There is no data center, so the source of residual load is a question.

B. Summer power demand follows occupancy schedule closely, but this is not the case in winter. It appears that the unoccupied electric heat lockout function is not working.

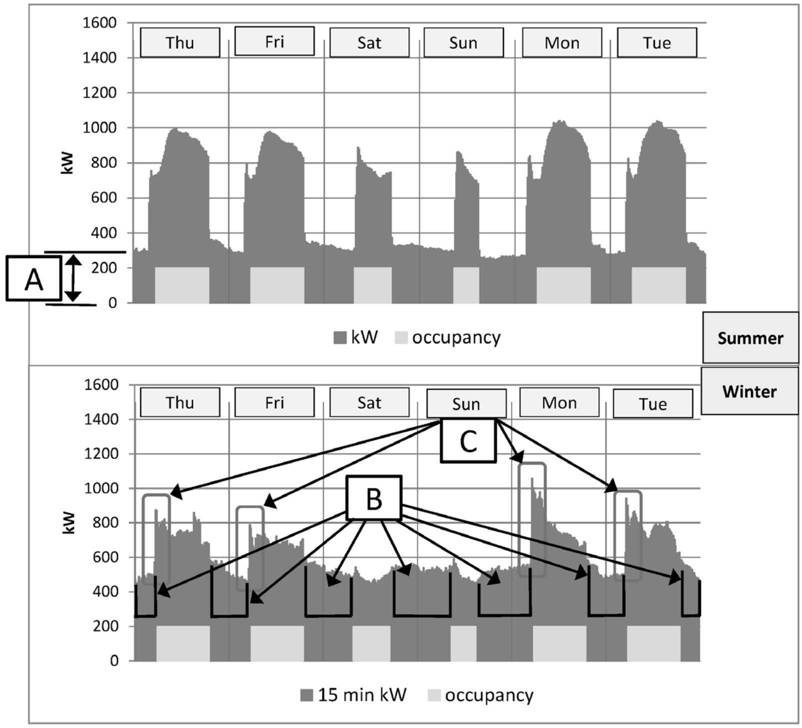


Figure 24P-Appx 2.1 Summer and Winter Interval Data—Case #2

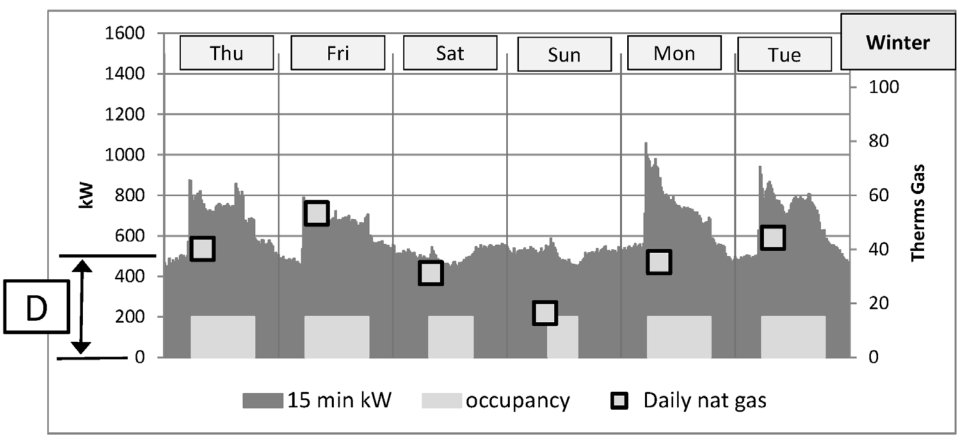


Figure 24P-Appx 2.2. Winter Interval Data with Gas Usage Overlaid—Case #2
Time placement for daily natural gas usage is 8:00am, an approximation of when the morning warm-up would end.

C. There is a large spike Monday morning in winter which suggests the gas preheat is turning off prematurely. This may be the highest demand moment for the year; certainly for the month.

D. Based on summer residual demand compared to typical weekend winter demand, an average of 200 kW of electric heat is in use on the weekend (500kW-300kW). Calculations showed that gas heat accounted for only 17% of the unoccupied heating burden; thus more than 80% of the heating work is by electric resistance and gas heating is underutilized (Avg. 40 therms of gas vs. 200 kW*24 hours*3413/100,000 = 40/164 therms=17%).

Recommendations:

1. Take steps to reduce the 300 kW residual load that is currently a third of daytime peak demand. Reducing this value by 100 kW over 3600 annual unoccupied hours represents 360,000 kWh annually.

2. Fuel switching. Adjust controls to utilize natural gas heating for unoccupied periods instead of electric resistance. For the same temperature achieved, the cost of operation will be reduced according to the cost ratio of natural gas/electric resistance delivered heat, which may be 50%.

3. Adjust morning warm-up controls to exceed occupied setting by a few degrees before reverting to daytime electric heat, to avoid the initial spike that is setting the demand for the month. At $15/kW-month, reducing electric demand by 100 kW would reduce demand charges for that month by $1500.

Case Study #3. Manufacturing Facility

Light manufacturing. Metal machining and assembly, heat treat, plating, painting, welding. Lighting: T5HO fluorescent in the plant, T8 fluorescent in the front engineering and sales area. HVAC: Evaporative cooling, natural gas radiant tube heaters, packaged rooftop units for front area, gas-fired make-up air units for process exhaust. Controls: Manual control for lighting, conventional thermostats for HVAC, exterior lighting on a photocell. Process shared support systems: Steam boiler for plating lines. Most heat treat ovens are natural gas, but some are electric. Air compressors: Production hours: 5:00am-11:00pm (two shifts) M-F, 5:00am-1:00pm (one shift) Saturday, closed Sunday. "Off" (plant idle) = 3200 hours per year.

Review of Interval Data

Fig. 24P-Appx 3.1 shows interval data for production and idle plant for contrast. Natural gas is used for heating, and cooling is evaporative, so electricity use is mostly process-related regardless of season.

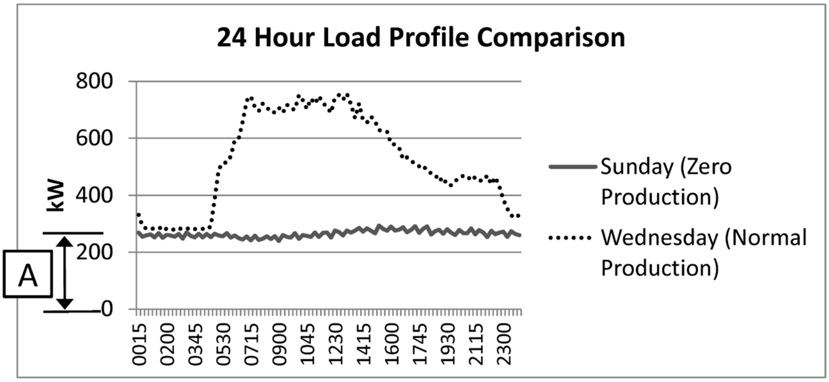


Figure 24P-Appx 3.1. Interval Data and Inference for Case #3

Observations:

A. Load profile shows idle plant power usage steady at 250 kW, which constitutes 33% of active demand levels and 23% of annual electric usage. Energy use in zero production periods is higher than expected and begs the question "why." The pattern of persistent electrical use is found regardless of season.

B. Ref. **Fig. A3.2**. Equipment electric loads were measured at individual equipment with zero production/no employees/lights out condition. Interval data was gathered the next day for the same period of time, to verify that most of the usage points had been identified. In this case, about 70% of load was accounted for.

C. Ref. **Fig. A3.3**. Idle plant usages were grouped and a pie chart shows "where it all goes." Lights were mostly turned off. Minor ghost loads were found in many units (e.g. PLCs idling at CNC machines), which is expected and of minor concern. Significant residual loads were discovered for the plating area, and also the air compressor which was serving only leaks and throttling losses. Personal computers were mostly off, but servers and UPS units were idling and consuming energy.

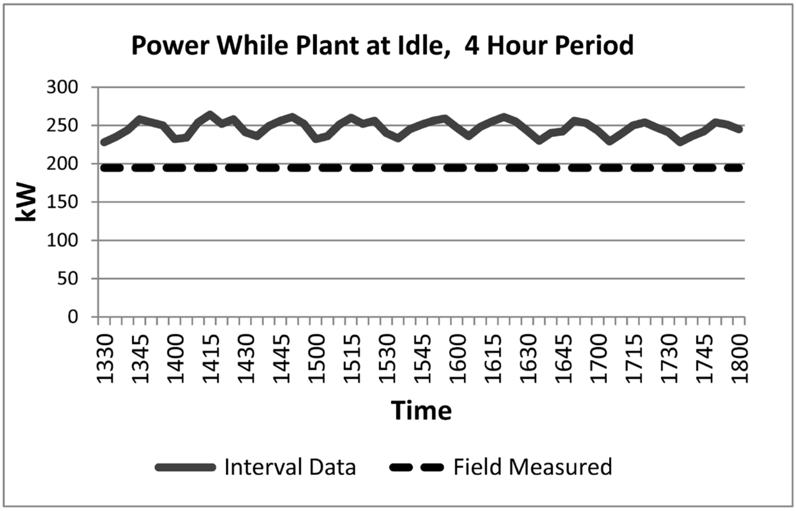


Figure 24P-Appx 3.2. Interval Data Sanity Check for Field Measurements, Case #3

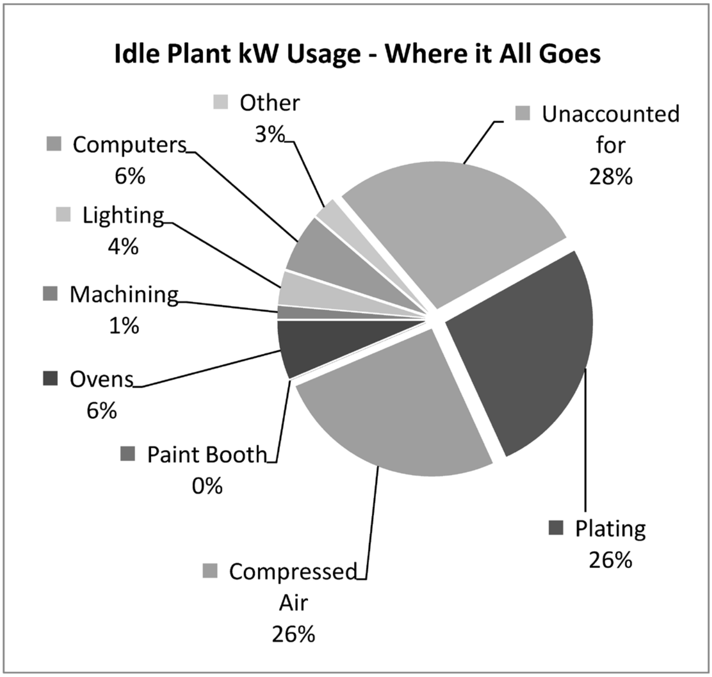


Figure 24P-Appx 3.3. End Use Breakdown for Case #3

Recommendations:

There is resistance to changes that may impact production timing, product quality, and safety, and these must be answered as a higher priority than energy cost. However, expenses without revenue are appropriate to pursue, since they dilute profits. Sometimes things are the way they are for excellent reason, and sometimes just habit. Machining equipment is not a source of high standby loss and no idling hydraulic motors or flywheels were observed.

1. One critical item needing compressed air was identified, which were air-powered emergency pumps for chemical spill containment. These pumps, off except for emergency, were connected to the main air system, complete with leaks, causing continuous compressor operation. Providing a packaged tank-mounted 'shop' air compressor, piped directly to these pumps, would accommodate the emergency spill scenario without plant air, allowing the main compressors to be stopped. This small dedicated compressor would not run unless the pumps started. **60 kW and 190,000 kWh annual reduction**.
2. Plating chemistry is complex but predictable and continuous operation of circulators, aeration pumps, and stirrers are probably not required once production has stopped. Conveyors and rectifiers are already being turned off at shift end. It is reasonable to keep plating tanks at temperature for the 6 hours between weekday shifts, and simply cover them at the end of the day; weekend off-shift is 40 hours long and warrants a cool-down. With covered tanks it may be possible to lower ventilation to half in unoccupied times, which would reduce load for the scrubber and make-up air unit; however safety regulations would govern. Chilled brine pumps are integral to portions of the plating line—two pumps run continuously where one would be plenty during off shift. **40 kW and 120,000 kWh annual reduction**.

Section III

Appendix

Appendix

GLOSSARY OF TERMS

absorption—A type of cooling system that utilizes heat (often waste heat) to generate the cooling effect. The process is a series of repeating chemical reactions, in contrast to mechanical refrigeration compression cycle cooling.

AEE—Association of Energy Engineers

approach—Heat exchanger context. Since heat exchangers work to bring the temperatures of two fluids closer together, the "approach" is simply how close they can get. The approach value is affected by the amount of surface area of the heat exchanger, the time the two fluids are in contact with each other, and the degree of fouling on the surfaces. As surfaces foul, approach increases. Purchasing extra heat exchange surface area (larger unit) reduces the approach.

ARI—Air-Conditioning and Refrigeration Institute

ASD—Adjustable Speed Drive

ASHRAE—American Society of Heating, Refrigerating and Air-Conditioning Engineers

Balance Temperature—For a given building, the balance temperature is that point where envelope heat loss matches internal heat gains. At this value of outside air temperature, the building inside temperature is constant with no heating or cooling energy being applied. Buildings with high internal loads have low balance temperatures.

Bin Weather—A tabulation method of categorizing weather patterns. The method counts the number of hours that weather in a given area meets a criteria (dry bulb temperature, wet bulb temperature, humidity ratio, etc.) and they form a collective total hours for that 'bin'. The larger the value of the bin, the more hours per period (e.g. per year) that weather will match the criteria.

The bin concept applies to other values such as wind or solar incidence by area.

Bhp—Brake hp. Generally used to represent the hp required to drive a machine at the shaft. This value is less than motor horsepower which adds motor efficiency losses

BLC—Building Load Coefficient. A collective of energy use factors that are dependent upon the differential temperature between indoors and outdoors (dT). For example, the weighted average value of envelope components yields an overall heat transfer coefficient UA for the envelope from insulation R-values, and is (∑1/Rroof*Area of roof, 1/Rwall*Area of wall… / Area total); this is the overall UA value of the building envelope, the main component of the BLC. The overall UA is used to predict heat transfer from inside-outside dT for any given hour according to Q (in Btu/hour) = Overall UA * dT. The BLC includes envelope overall UA, perimeter and ground conduction loss, ventilation and infiltration, each with energy use as a function of inside-outside temperature difference. Once collected, the BLC forms a basic energy use signature that can be used to predict annual energy use from degree-days in the form Q (in Btu per year) = BLC*Degree-days/year*24 hours/day. Accuracy depends upon knowing the balance temperature of the building since this affects the value of degree-days. The BLC method is valid for winter heating loss approximations for thermally light buildings. BLC is not applicable to thermally heavy buildings (where internal gains dominate envelope-related energy use) or summer heat gains (solar and dehumidification loads are not accounted for in BLC).

Btu—British thermal unit. A measure of energy in Inch-Pound units. The amount of heat required to raise one pound of water one degree Fahrenheit.

Btuh—Btus per hour

CD—Construction documents. The final phase of completion of design documents. **CFL—**Compact fluorescent. A type of lighting.

cfm—Cubic feet per minute. An air flow rate

cfm/SF—Cubic feet per minute air flow per square foot.

COP—Coefficient of performance. Heat out divided by work in, same units. A measure of heating and cooling apparatus efficiency.

CTI—Cooling Tower Institute

Cv—Flow coefficient for a valve, commonly a control valve

CV—Constant volume. An HVAC air system that maintains steady air flow rates at all times, and controls comfort by varying temperatures.

db or dry bulb—Air temperature parameter. The temperature as indicated on an ordinary thermometer, contrasted to “wet bulb” temperature.

DD—Design development. An interim phase of completion for design documents that is after schematic phase and before construction document phase.

DDC—Direct digital control. Refers to microprocessor-based controls that are used to monitor and control equipment and processes, often associated with building systems.

Degree-Days—A compound unit of degrees * days. The value of degrees used in degree-days compares the average outside air temperature Tavg to the building balance temperature Tbalance. In cooling season the value of degrees is (Tavg – Tbalance) and in heating season the value of degrees is (Tbalance – Tavg). For example, for a 60F balance temperature, a daily high temperature of 40F and a daily low temperature of 10F, the average temperature for this day is (40+10)/2 = 25F, and there are 60-25 = 35F heating degree-days for this day. Since Tavg is below the building balance temperature the cooling degree-days for this day are zero. Degree-day values are tabulated for time periods (months or years) for different areas. On an hourly basis, degree-hours represent the relative demand for heating and cooling compared to other times, i.e. the colder it is in a given hour the harder the heater runs. On an annual basis, degree-days represents the demand for energy associated with outside air temperatures for the year. Comparing degree-days between years indicates relative severity of cold and hot seasons and implies relative heating and cooling costs.

delta T or dT—Differential temperature. Parameter for heat transfer, the difference between inlet and outlet, different sides of a wall, etc.

dP—Differential pressure. A pressure measurement, used in measuring or describing water systems and the energy they consume by pump energy

Dp or Dew Point—Dew Point Temperature

Psychrometrics. That dry bulb temperature where condensation (dew) will form.

DSM—Demand Side Management

DX—Direct Expansion

Air conditioning cooling with refrigerant metering control valve directly at the point of air cooling, e.g. inside the tubes that are in the air stream.

ECM—Energy conservation measure

ECM Motor—Electrically Commutated Motor. An electronic variation of conventional squirrel cage motor design that increases motor efficiency, emulating DC motor technology.

ECO—Energy conservation opportunity

EER—A parameter used to evaluate HVAC equipment efficiency

Numerically it is the Btu output divided by the watts input. Watts input includes auxiliary equipment such as indoor and outdoor fans.

efficacy—A parameter used to evaluate different types of lighting. Numerically it is the lumen output divided by the watts input

eff—Efficiency. Generally quantified by useful output divided by energy input, expressed as a fraction or percent, less than or equal to 1 or 100%.

EMS—Energy management system. Refers to digital control systems that monitor and control building HVAC systems, lighting systems, and other energy consuming systems.

ESCO—Energy Services Company

Specialists in energy engineering and projects, often utilizing shared savings or guaranteed savings contracts

ft. head or ft. w.c.—A unit of liquid pressure measurements. Refers to the measured vertical rise in a manometer. 1 psi = 2.31 ft. w.c..

fpm—Feet per minute. Velocity, usually air.

fps—Feet per second. Velocity, usually water.

gpm—Gallons per minute. A water flow rate.

GSHP—Ground source heat pump. A refrigeration system that provides HVAC heating and cooling, and utilizes the earth mass as a heat sink and heat source.

HID—High intensity discharge. A type of lighting.

hydronic—HVAC system that transports the heating and cooling energy to the points of use by circulating fluid, usually water

HVAC—Heating, ventilating and air conditioning. The general term used to describe systems used to regulate indoor air comfort

hp—Horsepower

IESNA—Illumination Engineering Society of North America

in. w.c.—Inches water column. A low range pressure measurement, used in measuring or describing HVAC air systems and the energy they consume by fan energy. Refers to the vertical rise of a manometer

Inverter—A device that changes direct current to alternating current.

IPLV—Integrated Part Load Value

A standardized measurement system for cooling equipment part load efficiency.

IRR—Internal rate of return. Economic analysis term

KBtu—Thousands of Btus

kSF—Thousands of Square Feet

kW—Kilowatt. A measure of power or capacity: a thousand watts.

kWh—Kilowatt-hours. A measure of energy

kW/ton—A measure of cooling apparatus efficiency. Watts in divided by cooling tons output.

LCCA—Life cycle cost analysis

LED—Light emitting diode

LPB—Lighting power budget

Lum—Lumens

MA—Mixed air

MMBtu—Millions of Btus. A measure of energy.

MERV—Minimum efficiency reporting value. A numerical system of rating filters based on a minimum particle size efficiency. A rating of 1 is least efficient; a 16 is the most efficient. See also ASHRAE Standard 52.2.

M&V—Measurement and verification

MW—Megawatt. A measure of power or capacity. A million watts.

NEMA—National Electrical Manufacturers Association

OA—Outside air

O&M or O/M—Operations and maintenance. Manuals.

Pa—Pascals. A unit of pressure.

PC—Performance contract

PF—Power factor.

Electrical. The ratio of real power to apparent power.

PTAC—Package terminal air conditioner. A through-the-wall unit, as in a hotel.

Rectifier—A device that converts alternating current to direct current.

rH—Relative humidity

RA—Return air

ROI—Return on investment.

RTU—Rooftop unit. Refers to packaged HVAC equipment located upon the roof of a building

SD—Schematic design. The initial phase of completion for design documents that is before the design development phase

SEER—Seasonally adjusted EER. Based on improved HVAC performance in mild weather

Service Factor (motor)—A multiplier that indicates allowable motor overloading, e.g. a 10 Hp motor with SF=1.15 can be operated at 11.5 Hp. Operating in the service factor for extended periods will likely shorten the motor life.

SF—Square feet

SPP—Simple payback period. Economic analysis term

TAB—Test and balance

Therm—100,000 Btus. A standard measure of natural gas heating systems, but can apply to any heating system.

Thermally Light—Describes the relative proportion of internal heat loads to envelope transmission losses. A thermally light building has relatively low internal loads compared to envelope loads and a higher balance temperature.

Thermally Heavy—Describes the relative proportion of internal heat loads to envelope transmission losses. A thermally heavy building has relatively high internal loads compared to envelope loads and a lower balance temperature.

Ton (of cooling)—Cooling capacity, e.g. rate of cooling. Convention defines this as the amount of heat required to melt one ton of ice in a 24-hour period. 12,000 Btu/hr cooling rate.

Ton-hour—The factor of tons cooling capacity and hours of duration. A cooling energy unit. 12,000 Btu. Can be equated to kWh if kW/ton is known.

TSP—Total static pressure. Air pressure quantity used to evaluate fan power requirements.

T-5—A type of fluorescent lighting tube. "5-eighths" of an inch diameter.

T-8—A type of fluorescent lighting tube. "8-eighths" of an inch diameter, or 1" diameter tubes.

T-12—A type of fluorescent lighting tube. "12-eighths" of an inch diameter, or 1.5-inch diameter tubes

VAR or VARs—Volt-amp-reactive. Refers to the power use that is non-resistive and does not show up on a watt-meter. This is the symptom of low power factor. The additional current from this "apparent power" (contrasted to "real power) is the source of increased copper losses and voltage drop in distribution systems, i.e. the conductors carry additional amps for no "real" purpose, but suffer the losses just the same.

VAV—Variable air volume. An HVAC air system that maintains a steady air temperature and controls comfort by reducing the system air flow proportionally as cooling demand decreases

VFD—Variable Frequency Drive. An electronic form of Variable Speed Drive (VSD) or Adjustable Speed Drive (ASD) that re-forms the standard AC power and combines varying frequency, volts, and amps to control motor speed.

VSD—Variable Speed Drive

W—Watt. A measure of power or capacity. The basic unit of electrical power measure.

W/SF—Watts per square foot

Wb or wet bulb—Wet bulb temperature. A property of air, measured by covering a standard temperature element (the "dry" bulb) with a moist sock and exposing it to the air stream. The amount that wet bulb temperature is less than the 'dry bulb' temperature is a function of the relative humidity of the air.

XA or EA—Exhaust air

CONFLICTING ECMS AND 'WATCH OUTS'

Any engineer's nightmare is to solve one problem and create new ones at the same time. Here are a few of the watch-outs for this field. There will always be more, so tread lightly and seek peer review when you are in new territory. The general answer to all of these is that energy measures are usually system changes and usually more than one thing is affected.

Action	**Reaction**
Multiple measure savings	Measures affect each other, such that savings are not fully additive. Aggregate savings overstated.
Lighting retrofit savings	Heating energy increases, eroding savings and possibly resulting in insufficient heating capacity if the heating system is marginally sized.
Lighting retrofit	Existing heating system was marginally sized and now is inadequate.
Lighting—Occupancy Sensors	Savings assume lights are always left on, but occupants were actually turning their lights off much of the time, so savings are overstated.
Lighting—Daylight Harvesting	Sunlight in abundance will provide good lighting but will also increase heat load in cooling season. Skylights will increase heat loss in heating season.

(*Continued*)

Action	Reaction
Refrigeration retrofit for high efficiency	Waste heat was used for heating, and now auxiliary heating elements are needed, increasing energy use.
Chiller savings from reduced head pressure, via lower condenser water temperature	Low efficiency cooling tower requires large increase in tower fan kW, eroding most of the chiller savings. Too much reset can create operational problems for the chiller. Some chillers cannot be reset below 75 degrees. 65-70 deg F is usually safe, and some can accept colder water with excellent reductions in kW.
Chiller savings from increased suction pressure, from increased chilled water temperature set point.	Higher chilled water temperature creates higher apparatus dew point temperature in air handlers, and loss of dehumidification.
Condenser water or primary chilled water pump energy savings from variable flow pumping to track chiller load profile.	Changes in average chiller condensing or evaporating temperatures and onset of laminar flow will erode system savings by increasing chiller kW/ton. Still a net gain, but less than calculated by pump energy savings alone.
Evaporative cooling, including cooling towers, evaporative condensers, etc. for compressor savings.	Cost of water and sewer forgotten. This can be a third or half of the energy savings sometimes; It's still worth doing, but remember to subtract those costs from the energy savings.
Condensing boiler retrofit for 90+ efficiency.	High efficiency not achieved because boiler rating depends on low return water temperature, and existing hydronic design is not set up for this. May require different coils with more surface area to accept reduced water temperature and achieve the same heat transfer with less approach.
Power factor correction	Facility with high levels of harmonics can cause premature failure of the capacitors and possible interruption of service.
Air Economizer	Building problems: doors standing open is a common complaint.

(*Continued*)

Action	Reaction
	For buildings with very low balance points, using very cold air for cooling encourages stratification and nuisance freeze stat tripping. Relative humidity swings inside the building for humidity sensitive activities, including electronics, books, and artifacts.
Water Economizer	Operating the cooling tower at very low temperatures during winter, for economizer mode, creates a narrow differential between condenser water basin temperature and the setting of the basin heater, if so equipped. If the basin heaters are active, the cooling tower fan simply rejects the additional heat, but the savings of the water economizer operation are eroded in the process.
VFD	Savings overestimated using standard cube rule. Prevent this by using square instead of cube and ignoring savings for speeds below 20%. VFD applied to standard duty motor. Premature failure is likely. Prevent this with Inverter Duty or VFD Grade motors. Damage potential for any splash-lubricated equipment. Prevent this with a minimum speed setting. Damage potential for rotating equipment with a critical frequency. Prevent this with "skip frequency" setting. VFD located in hot conditions, near hot equipment or outdoors. Prevent this with remote location or special cooling provisions. VFD located upstream of an equipment disconnect. The open circuit scenario will overload the VFD. Prevent this with an interlock kit on the electrical disconnect switch.
Constant volume HVAC converted to VAV	Constant Volume HVAC low pressure ductwork may not be suitable. Keep converted duct pressures low and protect with static pressure switches.

(*Continued*)

Action	Reaction
Demand reduction measure savings	Utility 'ratchet' clause can negate savings unless the measure is consistent. A one-time high demand can set the minimum demand charge for up to a year.
Heat recovery savings	Parasitic losses of auxiliary pumps or fans, and system friction added from the equipment in the fluid stream, filters, etc.
Flow savings from increasing delta-T on hydronic systems	The 'signature' of the delta-T is largely fixed by the building loads, e.g. the air handlers and coils, and often the delta-T is limited by the existing building, unless building modifications are also made. Unless coils are replaced to gain more surface area, the system dT can only be increased if both the lower and upper temperatures are extended by the same amount, unless other air system parameters are also changed such as supply air temperature, space temperature, or space relative humidity. For ***example***, changing a 45-55 chilled water coil to 44-56 or changing a 180-160 hot water coil to 185-155, but this impacts the cooling and heating efficiency.
Insulate the floor to reduce winter heat loss	Frozen pipes in the crawl space that were heated from the losses through the floor.
Thermostats turned off in unoccupied areas	Frozen pipes at the perimeter
Reduce minimum outside air damper position OR Reduce VAV box minimum position (single path VAV) OR Mixed air low limit control overrides ventilation damper position in cold weather.	Ventilation provisions in the HVAC design are reduced or defeated.

TOP 10 IAQ MISTAKES A CEM SHOULD AVOID

Source: *Energy Management Handbook*, 7th ed, Chap. 17, Halliwell, J., Fairmont Press

10) Do not underestimate or under-respond to IAQ complaints in your building.

9) Do not delay investigating IAQ complaints, their cause and origin.

8) Do not forget to conduct an IAQ baseline survey before you implement your energy management measures.

7) Do not fail to review your energy management program for its potential to elevate indoor airborne contaminants.

6) Do not implement energy management measures that would reduce ventilation, increase humidity, depressurize the building, or reduce filtration without carefully reviewing their potential impacts on IAQ.

5) Do not be intimidated by IAQ. All problems have a common denominator cause: increased airborne contaminants.

4) Understand that many IAQ problems occur not from their energy management programs, but from operator error.

3) Do not assume that solving the physical IAQ problem will automatically solve the occupants' perceptions of the problem.

2) Do not perform air sampling as an initial IAQ investigative strategy.

1) Do not panic. IAQ problems can be solved quickly and effectively when they are responded to early.

ENERGY AUDIT LEVELS

Levels are designed to be built-upon. A level 2 audit includes the scope of Level 1 and a Level 3 audit includes the scope of Levels 1 and 2. Preliminary analysis is included for all audits. Checklist audits and online audits are not described and considered Level 0.

Preliminary Analysis

Review step prior to field work. Basic metrics help gage potential for savings, and provide a baseline. Energy Use Intensity (EUI, kBtu/SF-yr) compared to available similar buildings or statistical data norms. Other metrics include Energy Cost Intensity (ECI, $/SF-yr), monthly patterns of usage for one or more years prior. Unusual patterns sometimes appear which can prompt questions and additional reviews when on site. Metrics in visual form (charts) help engage the customer and serve the educational component of the audit process. Additional metrics that may/may not apply:

- Interval data (daily, hourly, etc.) is useful for opportunities such as equipment left on in unoccupied times (ghost loads), automatic control dysfunction, and other unexplained uses.
- When the customer is on a time of use electric rate, additional metrics may be appropriate such as on/off peak usage, load factor, and power factor.
- Usage overlaid with occupancy, weather, production, etc. to establish dependencies.
- Regression for building energy signature and multi-year normalization

Level 1 Audit (A.k.a. Scoping Audit, Walk-Through Audit)

Can be applied a lead-in audit to spark interest for more detailed work, or to assess economic potential for performance contracting. Does not include detailed recommendations or analysis. Looks for quick and easy payback items. Includes basic metrics like a year's usage, energy per SF, and a benchmark comparison. Includes applicable rebates or incentive items. When this is the extent of energy surveying the 'cherry picking' effect can result in missed opportunities that remain stranded once the easy opportunities are implemented. Level of rigor is more cursory than detailed.

Level 2 Audit (Most common)
Looks in detail at systems and not just equipment. Review of envelope, lighting, HVAC, domestic hot water, plug loads, compressed air, process uses. Review of end uses to identify where energy goes. Significant attention given to operations and maintenance (O/M), automatic controls. Includes low cost measures as well as capital measures. Evaluates related considerations such as comfort, lighting levels and quality, ventilation and indoor air quality, acknowledging that these things interact with energy measures. Provides basic rough cost and savings calculations for capital measures that allow prioritizing of measures. Scope of suggestions is wider than Level 1 and not limited to low hanging fruit.

Level 3 Audit (Drill-down high rigor for selected Level 2 items)
Adds drill-down information for selected Level 2 capital projects (not all Level 2 measures). Added tasks can include computer modeling or spreadsheet calculations, data logging, and other labor intensive activities. Process analysis beyond conceptual falls in this category. Iterations or options within a given design are possible. The added rigor is justified for large projects where the expense represents a business risk. There is a clear expectation for higher accuracy cost/savings estimates compared to Level 2 which requires skill in analysis, and usually entails preliminary design as an enabler for more detailed cost estimating. Complex analysis such as heat pumps, co-generation, heat recovery, or fuel switching can be at this level. Time demands increase exponentially and could take months for a large facility.

System-Specific Audits
Applies only to systems of interest. Conventional approaches for 'whole buildings' will not apply. Evaluation for opportunities may/ may not have the benefit of industry averages, although baseline usage, demand and usage patterns, and load profiles are viable. Examples include lighting-only surveys, central or district heating and cooling, and process review.

Levels of Energy Audits
Source: *A Guide to Energy Audits*, EERE, DOE Building Technologies Program, PNNL-20956, 2012

The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) defines three levels of audits.

Each audit level builds on the previous level. As audit complexity increases, so does thoroughness of the site assessment, the amount of data collected and the detail provided in the final audit report.

Level I	Site Assessment or Preliminary Audits identify no-cost and low-cost energy saving opportunities, and a general view of potential capital improvements. Activities include an assessment of energy bills and a brief site inspection of your building.
Level II	Energy Survey and Engineering Analysis Audits identify no-cost and low-cost opportunities, and also provide EEM recommendations in line with your financial plans and potential capital-intensive energy savings opportunities. Level II audits include an in-depth analysis of energy costs, energy usage and building characteristics and a more refined survey of how energy is used in your building.
Level III	Detailed Analysis of Capital-Intensive Modification Audits (sometimes referred to as an "investment grade" audit) provide solid recommendations and financial analysis for major capital investments. In addition to Level I and Level II activities, Level III audits include monitoring, data collection and engineering analysis.

REPRESENTATIVE TASKS AND BACKGROUND KNOWLEDGE FOR ENERGY MANAGEMENT AND ENERGY ENGINEERING

This table shows the basic separation between energy engineer and energy manager tasks in industry. Most practitioners are more or less in one or the other group, but will work in the other group when needed, depending on their skills. **The center column describes the pool of knowledge both job titles share, which represent core knowledge for the energy industry overall**. The outer columns help differentiate energy manager vs. energy engineer. Items are not correlated horizontally between columns.

Representative Tasks Energy Engineer	Pool of Background Knowledge and Skills	Representative Tasks Energy Manager
• Review historical consumption usage and cost • Conduct field work for energy audit or process review • Identify energy opportunities • Apply recording devices / measurements / control system trend log reports • Analyze energy demand, consumption, time of use • Incorporate occupancy census, operating schedules, production data, weather data with energy metrics to identify dependencies	• Mechanical and electrical engineering concepts • Mechanical and electrical system types and characteristics • Basic physics incl. thermodynamics, heat transfer, fluids, boundary conditions, control volumes, power cycles, refrigeration cycle • Engineering economics • Basic architecture • Energy conservation principles and best practices • Building codes, energy codes, ventilation standards • Building and system design practices	• Obtain management buy-in • Identify mission of organization and organizational structure • Create goals and timeline for energy reduction • Create policy for comfort and equipment scheduling • Create policy and timeline for alternative energy • Promote a culture of conservation attitude • Create feedback mechanisms for energy end users, departments, divisions, for accountability
• Data normalizing and regression to identify dependencies • Allocate consumption by system, building, end use • Interview facility staff • Determine if envelope loads or internal loads dominate energy usage • Evaluate building envelope components • Quantify energy savings • Energy modeling for base case and alternatives • Model calibration • Fuel switching analysis • Calculate give and take from system interactions • Calculate cost savings and avoidance	• Building systems/interactions • Cost estimating • Energy accounting and analysis • Emerging energy technologies • Building construction practices • Indoor environmental quality concepts • Indoor air quality concepts • Equipment capabilities/lifecycles/operations • Plumbing equipment operations • Utility metering and rate design practices • Modeling techniques • Instrumentation capabilities • System interactions/controls/operations • Load profiles and part load equipment behavior	• Equate energy expense to behaviors and choices so energy is seen as a controllable cost • Identify agreed-upon energy program metrics for energy program success/failure • Identify required company rate of return • Identify future rate projections • Identify upcoming utility rebate programs • Determine appropriate benchmarks and metrics for building type or industry • Collect utility consumption data • Identify what is metered and identify additional metering requirements to enable metrics and accountability
• Calculate demand reduction • Verify targeted maintenance practices with energy benefit • Provide financial analysis and evaluating options vs. existing • Evaluate impacts of changes on occupant comfort and indoor air quality • Evaluate equipment sizing for right size / over sizing • Document and evaluate manufacturing process flows • Optimize sequence of operations and controls • Create guide specifications to evaluate designs, guide replacement equipment purchases, and steer future work	• Energy usage intensity • Steam tables • Automatic control systems, control theory, devices, and practices • Psychrometrics of air • Test and balance procedures • Equipment ratings and specifications • O&M best practices • Commissioning principles and procedures • Measurement and verification concepts and procedures • Drawing and specification interpretation • Common manufacturing processes • Compressed air systems and equipment • Carbon cycle principles and practices	• Track utility usage and cost by facility, process, area, etc. • Reconcile utility bill to sub meter readings • Determine load factor, power factor, unit energy costs • Assess competitive trends • Assess regulatory trends • Create indoor air quality guidelines • Create initiatives to meet goals • Develop timelines for implementation • Identify responsible parties for implementation • Identify funding sources • Identify annual utility budget • Education for stakeholders, management, O/M personnel

(*Continued*)

Representative Tasks Energy Engineer	Pool of Background Knowledge and Skills	Representative Tasks Energy Manager
• Specify equipment, system, scope of work • Review equipment performance against specifications • Verify proper operation of equipment components • Identify fundamental energy use signatures of systems and processes • Oversee or perform commissioning process • Oversee or perform measurement and verification • Oversee performance contracting details • Research emerging technologies	• Energy purchase agreements • Utility units, bills, and billing rates • Energy units and conversions • Data logging methods • Utility rate structure and cost breakdowns • Industry scorecards and dashboards • Project Management • Performance contracting concepts • Business acumen • Marketing techniques	• Give presentations and updates • Forecast / project usage and rates • Utilize utility rate structure for cost avoidance • Solicit energy conservation opportunities ideas • Educate stakeholders and O&M staff regarding energy practices • Request proposals or quotations • Make recommendations for changes during design for improved energy performance • Carbon counting / avoided emissions • Prepare financial summaries • Share best practices with peers, including case studies • Review O&M practices

NET ZERO DEFINITIONS

These are the four generally accepted definitions. Two measure energy units (site or source), one measures cost, and one measures emissions.

Source: (Torcellini et al. 2006a), summarized in Getting to Net Zero, NREL Journal Article NREL/JA-550-46382, September 2009

Net-Zero Energy Building Definitions

- Net Zero Site Energy: A site NZEB produces at least as much renewable energy as it uses in a year, when accounted for at the site.

- Net Zero Source Energy: A source NZEB produces (or purchases) at least as much renewable energy as it uses in a year, when accounted for at the source. Source energy refers to the primary energy used to extract, process, generate, and deliver the energy to the site. To calculate a building's total source energy, imported and exported energy is multiplied by the appropriate site-to-source conversion multipliers based on the utility's source energy type.

- Net Zero Energy Costs: In a cost NZEB, the amount of money the utility pays the building owner for the renewable energy the building exports to the grid is at least equal to the amount the owner pays the utility for the energy services and energy used over the year.

- Net Zero Emissions: A net zero emissions building produces (or purchases) enough emissions-free renewable energy to offset emissions from all energy used in the building annually. Carbon, nitrogen oxides, and sulfur oxides are common emissions that ZEBs offset. To calculate a building's total emissions, imported and exported energy is multiplied by the appropriate emission multipliers based on the utility's emissions and on-site generation emissions (if there are any).

COST ESTIMATING—ACCURACY LEVELS DEFINED

Source: AACE International Recommended Practice No. 18R-97 "Cost Estimating Classification System—as Applied in Engineering, Procurement, and Construction for the Process Industries," 1997. Reprinted with the permission of AACE International, 209 Prairie Ave., Suite 100, Morgantown, WV 25601 USA. http://www.aacei.org, copyright 2007 © by AACE International; all rights reserved.

INCREASING PROJECT DEFINITION

<table>
<tr><th>AACE Classification Standard</th><th>ANSI Standard Z94.0</th><th>AACE Pre-1972</th><th>Association of Cost Engineers (UK) ACostE</th><th>Norwegian Project Management Association (NFP)</th><th>American Society of Professional Estimators (ASPE)</th></tr>
<tr><td rowspan="3">Class 5</td><td rowspan="3">Order of Magnitude Estimate -30/+50</td><td rowspan="3">Order of Magnitude Estimate</td><td rowspan="3">Order of Magnitude Estimate Class IV -30/+30</td><td>Concession Estimate</td><td rowspan="4">Level 1</td></tr>
<tr><td>Exploration Estimate</td></tr>
<tr><td>Feasibility Estimate</td></tr>
<tr><td rowspan="2">Class 4</td><td rowspan="4">Budget Estimate -15/+30</td><td rowspan="2">Study Estimate</td><td rowspan="3">Study Estimate Class III -20/+20</td><td rowspan="2">Authorization Estimate</td></tr>
<tr><td rowspan="2">Level 2</td></tr>
<tr><td rowspan="2">Class 3</td><td rowspan="2">Preliminary Estimate</td><td rowspan="2">Master Control Estimate</td></tr>
<tr><td>Budget Estimate Class II -10/+10</td><td>Level 3</td></tr>
<tr><td rowspan="2">Class 2</td><td rowspan="4">Definitive Estimate -5/+15</td><td rowspan="2">Definitive Estimate</td><td rowspan="4">Definitive Estimate Class I -5/+5</td><td rowspan="4">Current Control Estimate</td><td>Level 4</td></tr>
<tr><td rowspan="2">Level 5</td></tr>
<tr><td rowspan="2">Class 1</td><td rowspan="2">Detailed Estimate</td></tr>
<tr><td>Level 6</td></tr>
</table>

(*Continued*)

Source: AACE International Recommended Practice No. 17R-97 "Cost Estimate Classification System," 1997. Reprinted with the permission of AACE International, 209 Prairie Ave., Suite 100, Morgantown, WV 25601 USA. http://www.aacei.org, copyright 2007 © by AACE International; all rights reserved.

ESTIMATE CLASS	LEVEL OF PROJECT DEFINITION Expressed as % of complete definition	END USAGE Typical purpose of estimate of estimate	EXPECTED ACCURACY RANGE Typical +/- range relative to best index of 1 [a]
Class 5	0% to 2%	Screening or Feasibility	4 to 20
Class 4	1% to 15%	Concept Study or Feasibility	3 to 12
Class 3	10% to 40%	Budget, Authorization, or Control	2 to 6
Class 2	30% to 70%	Control or Bid/ Tender	1 to 3
Class 1	50% to 100%	Check Estimate or Bid/ Tender	1

[a] If the range index value of "1" represents +10/-5%, then an index value of 10 represents +100/-50%.

SIMPLE PAYBACK VS. INTERNAL RATE OF RETURN (IRR)

Alternate Method to Calculate Internal Rate of Return (IRR)

Internal rate of return (IRR) is that interest rate that produces a net present worth of zero for a series of cash flows. It can also be defined as *the interest rate where the present worth of the project cost equals the present worth of the savings,* or *the interest rate where P/A=1.*

The IRR is compared to the cost of borrowed money (the hurdle rate), and measures with an IRR are normally not pursued unless the IRR is greater than the hurdle rate. Simple Payback Period is a more common representation of project merit, although IRR is more accurate since it includes the time value of money. Simple payback can be equated to IRR if the payback period and the measure life are known.

Example *IRR Calculation*

Investment $200,000
Life of measure 10 years
Annual savings $40,000
5-year simple payback period (SPP)

First Cost = present worth of the savings at some interest value
P = A (P/A, i?, 10 yrs)
200,000 = 40,000 (P/A, i?, 10 yrs)
factor P/A (simple payback)
200,000/40,000 = (P/A, i?, 10 yrs)
5 = (P/A, i?, 10 yrs)
now, pick through the interest tables for (P/A, i?, 10 yrs) = 5.0, and that is the equivalent interest rate or IRR (internal rate of return). In this case, IRR = 15%

Simple payback period (SPP) is the cost/benefit ratio of a project. SPP and IRR can be equated, provided the life of the measure is known.

Derivation: Internal Rate of Return (IRR) is that interest rate where the present worth of the savings is equal to the initial investment.

P = A * (P/A, i, n)

$$P = A * \frac{(1 + i)^n - 1}{i(1 + i)^n}$$

So, for some value of i,

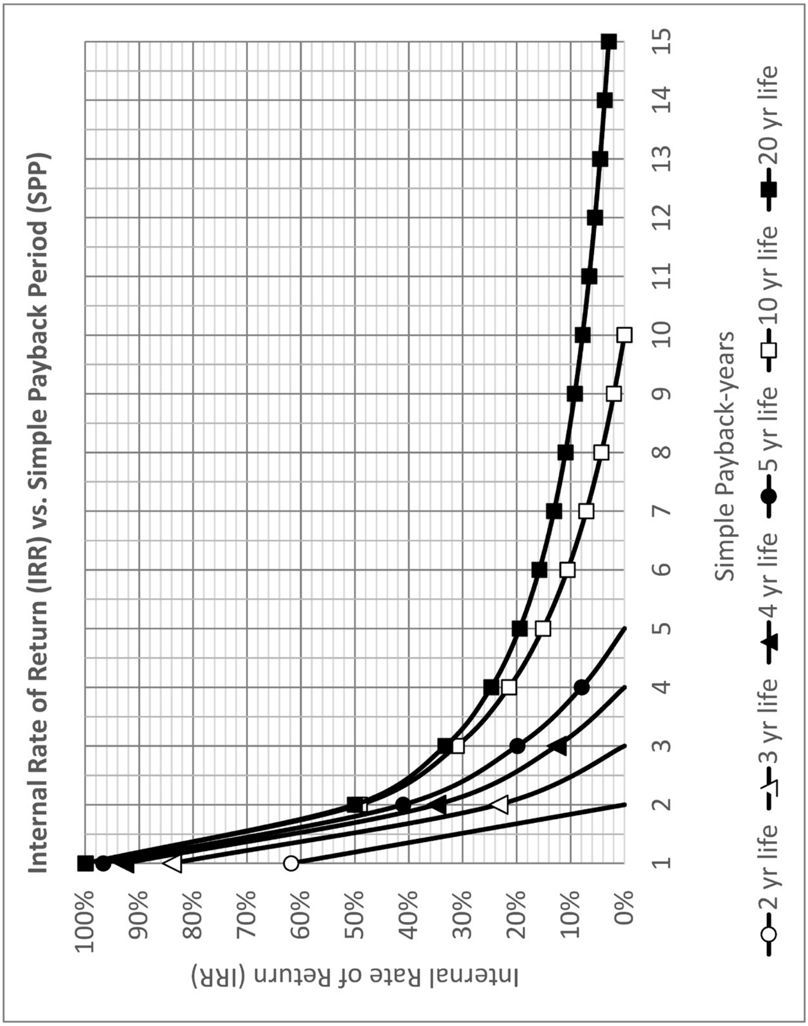
$$P/A \text{ (simple payback)} = \frac{(1 + i)^n - 1}{i(1 + i)^n}$$

The true benefit of this relationship is shown in the graph comparing SPP and IRR. While SPP is the common metric, it is the true time value of money that is most important for viable economic decisions, which is a balance of payback period and project lifespan.

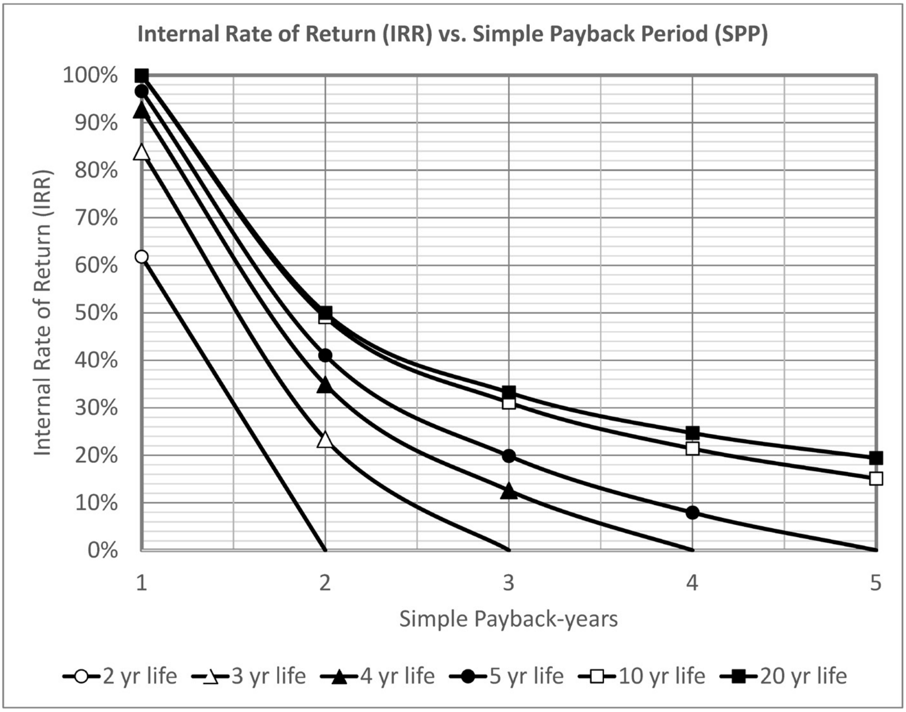
Internal Rate of Return from Simple Payback and Measure Life Tabular

Simple Payback Period, years	Measure Life, years: 2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
1	62%	84%	93%	97%	98%	99%	100%	100%	→										
2	0%	23%	35%	41%	45%	47%	48%	49%	49%	49%	50%	50%	→						
3		0%	13%	20%	24%	27%	29%	30%	31%	32%	32%	32%	33%	33%	→				
4			0%	7.9%	13%	16%	19%	20%	21%	22%	23%	23%	24%	24%	24%	24%	25%	25%	25%
5				0%	5.5%	9.2%	12%	14%	15%	16%	17%	18%	18%	18%	19%	19%	19%	19%	19%
6					0%	4.0%	6.9%	9.0%	11%	12%	13%	13%	14%	14%	15%	15%	15%	16%	16%
7						0%	3.1%	5.3%	7.1%	8.4%	9.5%	10%	11%	11%	12%	12%	13%	13%	13%
8							0%	2%	4.3%	5.7%	6.9%	7.8%	8.5%	9.1%	10%	10%	10%	11%	11%
9								0%	2.0%	3.5%	4.7%	5.7%	6.5%	7.2%	7.7%	8.2%	8.6%	8.9%	9.2%
10									0%	1.6%	2.9%	4.0%	4.8%	5.6%	6.2%	6.7%	7.1%	7.4%	7.8%
11										0%	1.4%	2.5%	3.4%	4.2%	4.8%	5.3%	5.8%	6.2%	6.5%
12											0%	1.2%	2.1%	2.9%	3.6%	4.2%	4.7%	5.1%	5.4%
13												0%	1.0%	1.8%	2.6%	3.2%	3.7%	4.1%	4.5%
14													0%	0.9%	1.6%	2.2%	2.8%	3.3%	3.7%
15														0%	0.8%	1.4%	2.0%	2.5%	2.9%

Internal Rate of Return from Simple Payback and Measure Life Graphical, 1-20 years SPP



Internal Rate of Return from Simple Payback and Measure Life Graphical, 1-5 years SPP



The greater the ratio of measure life to simple payback, the greater the IRR. The maximum IRR (at very long measure life) is 1/SPP (1/simple payback period).

This can be shown with an example:

A measure has a $50,000 investment cost and produces $10,000 per year savings. SPP=5 years

IRR is the interest rate where the present value of the savings is equal to the investment. So: $50,000 = $10,000 * (P/A, i, 50)

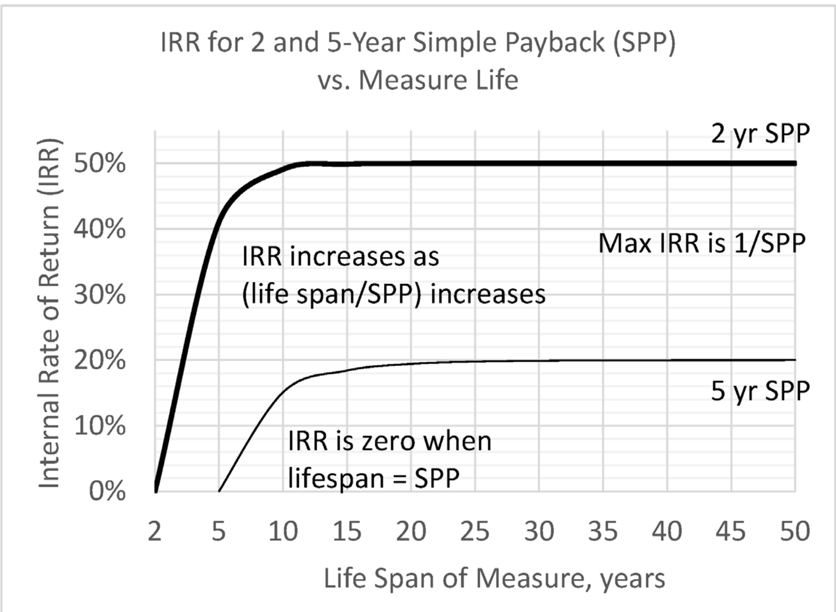
[50 years represents a very long lived project]

For equality, the factor (P/A, i, 50) must equal 5, which is the SPP value

The value of the factor, from standard interest tables, is 20% which is also 1/SPP

Maximum values of IRR for various values of simple payback period
Reference to 'factor' is standard economic interest tables.

SPP	Factor for (P/A, i, 50)	Max IRR% Where (P/A, i, 50) = SPP	Max IRR% from 1/SPP
1	1	100%	100%
2	2	50%	50%
3	3	33%	33%
4	4	25%	25%
5	5	20%	20%
10	10	10%	10%
20	20	5%	5%



IRR vs. Life Span for 2 and 5-year Simple Payback Period

When life span is equal to SPP, IRR is zero.

IRR increases as life span increases in relation to SPP, and reaches a maximum value at 1/SPP.

When life span is less than SPP, IRR is negative.

DSM PROGRAM COST EFFECTIVENESS TESTS

Source: *Energy Management Handbook*, Chap. 16, Baker,S., 7th Ed, The Fairmont Press

The contents of this appendix are based on the California Standard Practice Manual: Economic Analysis of Demand-side Programs and Projects (SPM) This provides an overview of various economic tests that utilities and policy-makers use to evaluate the cost-effectiveness of programs to support alternative energy. The formulae are abbreviations of those that appear in the SPM and should only be used for evaluation of conservation and load management (not fuel switching) programs.

Overview of economic test to evaluate cost-effectiveness of DSM programs

Cost-Effectiveness Test	Perspective	Benefit-Cost Ratio	Notes
Participant Cost	Participant	$\frac{\text{Bill Reduction + Incentive + Tax Credits}}{\text{Participant Cost}}$	1,2
Ratepayer Impact Measure	Ratepayer (customers who do <u>not</u> participate)	$\frac{\text{Avoided Supply Cost}}{\text{Program Cost + Incentive Cost + Revenue Loss}}$	1
Program Administrator Test (A.k.a. utility cost test)	Program Administrator (utility)	$\frac{\text{Avoided Supply Cost}}{\text{Program Cost + Incentive Cost}}$	1,3
Total Resource Cost (TRC)	Society	$\frac{\text{Avoided Supply Cost}}{\text{Program Cost + Net Participant Cost}}$	1,4
Societal Cost	Environment	$\frac{\text{Avoided Supply Cost + Avoided Environmental Impact}}{\text{Program Cost + Net Participant Cost}}$	1,5

Notes:

1. For the purpose of the cost effectiveness tests, the "cost" of the rebate measure has different meanings:
   - Cost for a **new project** = incremental cost, which is the total measure cost less a "base measure" (code required / needed anyway) cost. i.e. the efficiency uptick cost above basic measure.
   - Cost for a **retrofit that is presumed to be replacement upon failure** (key is needed anyway) = incremental cost like new project.
   - Cost for **retrofit that is early replacement**, i.e. discarding something that is still working = full project cost (not incremental)
   - Cost for a **fuel switching** project = full project cost (not incremental)
2. Strictly from the business case viewpoint of the customer. Ignores impact on utility rates, non-participating customers, etc.
3. Same as RIM test, but ignores lost revenue which affects rates.
4. Includes both the participant and utility costs. Ignores lost revenue effect on rates.
5. Same as TRC test, but includes cost of externalities (environmental, health costs from pollution, etc.).

Participant Test

The Participant Test is the measure of the quantifiable benefits and costs to the customer due to participation in a program. The benefits of participation include the reduction in the customer's utility bill(s), any incentive paid by the utility or other third parties, and any federal, state, or local tax credit received. The costs to a customer are all out-of-pocket expenses incurred as a result of participating in the program, such as the cost of any equipment or materials purchased as well as any ongoing operation and maintenance costs. The benefit-cost ratio (BCR_P) for the Participant Test is as follows:

$$BCR_p = B_p / C_p$$

If BCRP is greater than one, then the program is considered cost effective.

$$B_p = \sum_{t=1}^{n} \frac{BRt + TCt + INCt}{(1+d)^{t-1}}$$

$$C_p = \sum_{t=1}^{n} \frac{PCt}{(1+d)^{t-1}}$$

Where:

B_p = Net present value (NPV) of benefits to participants
C_p = Net present value of costs to participants
BRt = Bill reductions in year t
TCt = Tax credits in year t
INCt = Incentives paid to participant by utility
d = Discount rate
t = year
n = Lifecycle of the DSM measure
PCt = Participant costs in year t

Ratepayer Impact Measure Test

The Ratepayer Impact Measure (RIM) Test measures what happens to customer rates due to changes in utility (or other program administrator) revenues and operating costs caused by the program. The test indicates the direction and magnitude of the expected change in customer rate levels. If DSM causes utility rates to go up, non-participants will see increases in their bills. Participants, on the other hand, encountering the same rate increases may see still their total utility bills go down since they will consume less energy.

The benefits calculated in the RIM test are the savings from avoided supply costs. These *avoided* costs include the reduction of transmission, distribution, generation and capacity costs for during periods of load reduction. The costs calculated in the RIM test include program costs (incentives paid to the participants, program administrative costs) and decreases in utility revenues.

For a program to be cost-effective using the RIM Test, utility rates must not increase as a result of the program, i.e., non-participants will see no increase in their utility bills. The benefit-cost ratio (BCR_{RIM}) for the RIM Test is as follows:

$$BCR_{RIM} = B_{RIM}/C_{RIM}$$

If BCRRIM is greater than one, then the program is considered cost effective.

$$B_{RIM} = \sum_{t=1}^{n} \frac{UACt + RGt}{(1+d)^{t-1}}$$

$$C_{RIM} = \sum_{t=1}^{n} \frac{RLt + PRCt + INCt}{(1+d)^{t-1}}$$

Where:

B_{RIM}	=	NPV of benefits to rate levels
C_{RIM}	=	NPV of costs to rate levels
UACt	=	Utility avoided supply costs in year t
RGt	=	Revenue gain from increased sales in year t
RLt	=	Revenue loss from reduced sales in year t
PRCt	=	Program costs in year t
INCt	=	Incentives paid to participant by utility
d	=	Discount rate
t	=	Year
n	=	Lifecycle of the DSM measure

Program Administrator Test

The Program Administrator Test measures the net costs of a program as a resource option based on the costs incurred by the program administrator (usually a utility) and excluding any net costs incurred by the participant. The benefits of the Program Administrator Test are the avoided supply costs. The costs are program administration costs associated with running the program (rebates and administrative costs). When benefits exceed costs, the Program Administrator Test is satisfied, indicating a reduction in the total revenue requirements of the utility and resulting in a lower customer bill on average. Even though total utility revenues drop, the addition of program costs may result in higher rates ($/kWh); thus, non-participants' bills may go up even if the average customer bill goes down. The benefit-cost ratio (BCR_{PA}) for the Program Administrator Test is as follows:

$$BCR_{PA} = B_{PA}/C_{PA}$$

If BCRPA is greater than one, then the program is considered cost-effective.

$$B_{PA} = \sum_{t=1}^{n} \frac{UACt}{(1+d)^{t-1}}$$

$$C_{PA} = \sum_{t=1}^{n} \frac{PRCt + INCt}{(1+d)^{t-1}}$$

Where:

B_{PA}	=	NPV of program benefits
C_{PA}	=	NPV of program costs
UACt	=	Utility avoided supply costs in year t
PRCt	=	Program costs in year t
INCt	=	Incentives paid to participant by utility
d	=	Discount rate
t	=	Year
n	=	Lifecycle of the DSM measure

Total Resource Cost Test

The Total Resource Cost (TRC) Test measures the net costs of a program as a resource option based on the total costs of the program, including both the participants' and the utility costs. The TRC Test is the most commonly used measure of cost effectiveness since it provides an indication of whether the totality of costs, to utility and ratepayer, is being reduced. The benefits calculated in the TRC Test are the avoided supply costs. The costs are the program costs paid by both the utility and the participants. The benefit-cost ratio (BCR_{TRC}) for the TRC Test is as follows:

$$BCR_{TRC} = B_{TRC} / C_{TRC}$$

If BCRTRC is greater than one, then the program is considered cost effective.

$$B_{TRC} = \sum_{t=1}^{n} \frac{UACt}{(1 + d)^{t-1}}$$

$$C_{TRC} = \sum_{t=1}^{n} \frac{PRCt + PCt}{(1 + d)^{t-1}}$$

Where:

B_{TRC} = NPV of benefits to total resources
C_{TRC} = NPV of costs to total resources
UACt = Utility avoided supply costs in year t
PRCt = Program costs in year t
PCt = Participant costs in year t
d = Discount rate
t = Year
n = Lifecycle of the DSM measure

Societal Cost Test

The Societal Cost Test (SC) is a variant of the TRC Test, the difference being that it includes quantified effects of externalities (such as environmental impacts) in the measure of costs and benefits. The benefits calculated in the SC Test are the avoided supply costs plus the avoided environmental costs. The costs are the program costs paid by both the utility and the participants. The benefit-cost ratio (BCR_{SC}) for the SC Test is as follows:

$$BCR_{SC} = B_{SC} \;/\; CSC$$

If BCR_{SC} is greater than one, then the program is considered cost effective.

$$B_{SC} = \sum_{t=1}^{n} \frac{UACt + UECt}{(1+d)^{t-1}}$$

$$C_{SC} = \sum_{t=1}^{n} \frac{PRCt + PCNt}{(1+d)^{t-1}}$$

Where:

BSC	=	NPV of benefits to total resources including environmental effects
CSC	=	NPV of costs to total resources including environmental effects
UACt	=	Utility avoided supply costs in year t
UECt	=	Avoided environmental costs in year t
PRCt	=	Program costs in year t
PCNt	=	Net participant costs in year t (net off rebates)
D	=	Discount rate
T	=	Year
N	=	Lifecycle of the DSM measure

Heat Loss From Uninsulated Hot Piping And Surfaces

Source for piping table and characteristic curve: 3EPlus software, V4.1
Based on rigid fiberglass insulation, ASTM C547 type I and II, steel pipe,
All service jacket, K=0.25 Btu/hr-in-degF at 75F and k=0.65 at 500F,
80F ambient temperature
Source for flat surface table: Armstrong Machine Works, Chart No. 1111, 1983
Values are for surfaces 4 square feet area and larger.

Pipe

Btu/hr per foot — Differential Temperature, Pipe vs. Ambient, degF

Pipe Size inches	Insulation Thickness inches	50	100	200	300	500	750	1000
0.5	Bare	25	56	136	240	535	1,134	2,104
	1	5.0	11	24	41	91	192	348
	2	3.6	7.6	17	30	65	136	247
1	Bare	36	83	202	358	805	1,723	3,217
	1	6.2	13	30	52	114	240	435
	2	4.4	9.3	21	36	80	167	304
2	Bare	60	140	346	616	1,399	3,021	5,678
	1	9.4	20	46	79	174	366	666
	2	6.2	13	30	51	112	235	428
	3	5.0	11	24	41	90	189	343
2.5	Bare	72	167	413	736	1,676	3,625	6,817
	1	11	23	53	91	201	422	767
	2	6.5	14	31	54	118	248	450
	3	5.3	11	26	44	96	202	366
3	Bare	86	200	496	885	2,020	4,377	8,235
	1	13	27	62	108	238	501	910
	2	8.0	17	39	67	147	309	560
	3	6.3	13	30	52	115	240	436
4	Bare	109	253	628	1,123	2,569	5,579	10,510
	1	15	33	75	129	285	599	1,089
	2	10	20	46	78	176	368	669
	3	7.3	16	35	61	133	279	505
6	Bare	156	365	906	1,624	3,727	8,117	15,310
	1	23	48	111	191	423	890	1,619
	2	13	28	63	108	238	499	906
	3	10	21	47	81	178	373	676
8	Bare	200	469	1,167	2,092	4,811	10,490	19,780
	1	27	58	132	228	505	1,064	1,935
	2	16	34	78	133	294	617	1,120
	3	12	25	56	96	210	440	799
10	Bare	247	579	1,442	2,588	5,956	12,990	24,500
	1	35	74	170	293	649	1,368	2,489
	2	19	40	91	156	345	724	1,314
	3	14	30	67	115	253	531	964
12	Bare	292	683	1,701	3,053	7,035	15,360	28,980
	1	37	79	180	311	688	1,450	2,637
	2	22	46	105	181	400	839	1,524
	3	16	34	77	132	291	611	1,108

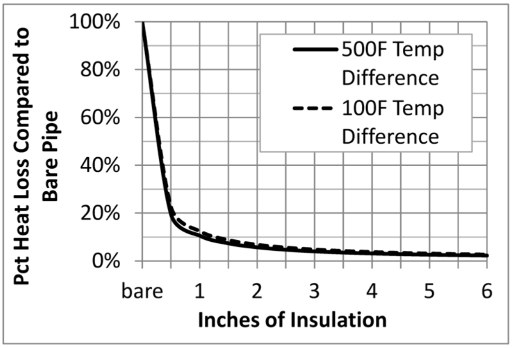
Flat Surface

Differential Temperature, Surface vs. Ambient, degF

Btu/hr per sq. ft	50	100	200	300	500	750	1000
Vertical	92	214	540	990	2,395	5,445	10,620
Horiz. Face Up	102	237	594	1,077	2,560	5,708	10,990
Horiz. Face Down	81	186	472	879	2,185	5,100	10,140

Characteristic Curve For Insulation Benefit

1-1.5 inches of insulation on a bare hot pipe reduces the heat loss by a factor of 10. Additional insulation benefits see diminishing returns.



DUCT FITTING LOSS COEFFICIENTS

This table shows a few common fittings with high transport energy losses. For a complete listing of duct fittings, refer to the sources at the end of the chart, and to *ASHRAE Fundamentals Handbook*.

Adapted from Carrier System Design Manual,1974 and
SMACNA HVAC Systems Duct Design Manual,1990

Notes:

1. Values marked with (*) were calculated with the approximate relationship L/D=50C
2. L/D = equivalent diameters, equal units, where

D = hydraulic diameter = $D = 1.3 * \left(\frac{(w*h)^{0.625}}{(w+h)^{0.25}}\right)$

3. Pressure Loss Formula: Duct fitting and system pressure losses, using "C" Factor

Pressure Loss = $\left({}^{V}/_{4005}\right)^2 * C$
where:
V = velocity in fpm
C=loss coefficient

Duct and fitting pressure losses using equivalent diameters (L/D)

Pressure Loss = f(L/D * $\left({}^{V}/_{4005}\right)^2$
where:
f = friction factor
L/D = equivalent diameters

4. For fittings with a change in velocity, use the difference of the inlet and outlet velocity head.
5. Fan discharge velocity profile is non-uniform and the fan system will incur loss unless duct geometry allows sufficient length of straight duct to establish a uniform velocity profile before take-offs or bends.

To calculate "100% effective duct length", assume a minimum of 1-1/2 duct diameters for each 2500 fpm or less, and add 1 duct diameter for each 1000 fpm over that.

Type	Description	Equiv. duct diam. (L/D)	Velocity Heads (C)
REDUCERS	Blunt 90 deg reduction (25% reduction)	***12 ****	0.25
	Blunt 90 deg reduction (75% reduction)	***25 ****	0.5
	Reduction 45 degrees per side	***10 ****	0.2
	Reduction 30 degrees per side	***5 ****	0.1
	Reduction 15 degrees per side	***2.5 ****	0.05
	Blunt 90 deg expansion (V1/V2=0.4)	***25 ****	0.5
	Blunt 90 deg expansion (V1/V2=0.8)	***35 ****	0.7
	Expansion 10 degrees per side	***30 ****	0.6
	Expansion 15 degrees per side	***50 ****	1.0
	Expansion 45 degrees per side	***75 ****	1.5
DAMPERS	**Fire Dampers**		
	Type "A" (retracted curtain is obstruction in the airstream)	***75 ****	1.5
	Type "B" (retracted curtain pocket out of the airstream)	***10 ****	0.2
	Control Dampers or Balancing Dampers (full open)		
	flat, stamped blade shape	***40 ****	0.8
	air foil blade shape	***15 ****	0.3
ENTRANCES and EXITS	Entrance - abrupt	***18 ****	0.35
	Entrance - bell mouth	***1.5 ****	0.03
	Exit - all	all pressure lost	all pressure lost
FAN DISCHARGE	Fan blast outlet area 60% of fan connection size, FC fan		
	Discharge Transition		
	12% of ideal effective duct (Note 5)		0.7
	25% of ideal effective duct		0.3
	50% of ideal effective duct		0.1
	100% of ideal effective duct		0.0
	Elbow Too Close to Discharge		
	12% of ideal effective duct (Note 5)		1.6
	25% of ideal effective duct		1.2
	50% of ideal effective duct		0.5
	100% of ideal effective duct		0.0
FAN INLET	**No Vanes**		
	Elbow @ fan inlet, no straight section		2.0
	Elbow @ fan inlet, 2D straight section		1.2
	Elbow @ fan inlet, 5D straight section		0.5
	With Vanes		
	Elbow @ fan inlet, no straight section		0.8
	Elbow @ fan inlet, 2D straight section		0.5
	Elbow @ fan inlet, 5D straight section		0.2
	0% obstructed fan inlet		0.0
	25% obstructed fan inlet		0.8
	50% obstructed fan inlet		1.6
	Fan inlet too close to cabinet wall, 0.75x fan diameter to wall		0.2
	Fan inlet too close to cabinet wall, 0.5x fan diameter to wall		0.4

Type	Description	Equiv. duct diam. (L/D)	Velocity Heads (C)
ELBOWS	**Radius Rectangular Elbow** (R/D=1, no vanes, W/D 2.0 - 3.0, average bend)		
	90 deg ell	15	***0.3 ****
	45 deg ell	7.5	***0.15 ****
	Rectangular Square Elbow (equal dimensions in/out, with vanes. Increase 4x if no vanes)		
	90 degree ell, vanes	15	0.7
	90 degree ell, no vanes	60	1.2
	90 degree offset "Z" - 2 ells, vanes	45	***0.9 ****
	90 degree offset - 2 ells, no vanes	***130 ****	2.6
	90 degree offset - 2 ells, different planes	***170 ****	3.4
	45 degree ell, no vanes	***15 ****	0.3
	Round Elbows (R/D=1, no vanes)		
	90 degree ell, smooth	10	0.2
	90 degree ell, 3-piece or 5-piece	25	0.4
	90 offset - 2 ells	75	***1.5 ****
	45 degree ell, smooth	5	0.13
	45 degree ell, 3-piece	15	0.24
	90 degree offset - 2 ells	75	***1.5 ****
	45 degree offset, 3-piece	45	***0.9 ****
	Round mitered ell (no vanes)		
	90 degree ell	65	1.2
	90 degree offset	195	***3.9 ****
	45 degree ell	35	0.34
	45 degree offset	105	***2.1 ****

EVAPORATION LOSS FROM WATER IN HEATED TANKS
Table units are Btu/hr-SF

Source, "Energy Tips," Office of Industrial Technologies (ITP), US DOE Office of Energy Efficiency and Renewable Energy. Original data from "Steam Efficiency Improvement" by the Boiler Efficiency Institute at Auburn University

Liquid Temp, degF	65 degF ambient air temp	75 degF ambient air temp	85 degF ambient air temp	95 degF ambient air temp	105 degF ambient air temp
110	244	222	200	177	152
130	479	452	425	397	369
150	889	856	822	788	754
170	1608	1566	1524	1482	1440
190	2900	2845	2790	2737	2684

Notes:
Heat loss table assumes dry and still air. Losses will be reduced when ambient air is humid; losses will increase with air velocity.

Approximate Adjustment for Air Velocity:

m/s	mph	fpm	Factor A
0.0	**0.0**	**0**	**1.0**
0.4	1.0	88	**1.4**
0.9	2.0	176	**1.8**
1.3	3.0	264	**2.2**
1.8	4.0	352	**2.6**
2.2	5.0	440	**3.0**
2.7	6.0	528	**3.4**
3.1	7.0	616	**3.8**
3.6	8.0	704	**4.1**
4.0	9.0	792	**4.5**
4.5	10.0	880	**4.9**

Approximate Adjustment for Relative Humidity:

RH	Factor B
0%	**1.00**
10%	**0.97**
20%	**0.97**
30%	**0.95**
40%	**0.93**
50%	**0.91**
60%	**0.89**
70%	**0.88**
80%	**0.86**
90%	**0.84**
100%	**0.82**

Table of factors based on: Wp=(A/Y)*(pw-pa)*(95+0.425V)
Where:
wp = evaporation of water, lb/hr
A = area of pool surface, ft2
pa = saturation pressure at room air dew point, in. Hg
pw = saturation vapor pressure taken at surface water temperature, in. Hg
V=velocity over the water, fpm
Source: ASHRAE Applications Handbook, 2011 © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org.

Example: a 100 SF open tank operates at 150F in an 85F ambient room. There is a push-pull fume control system that maintains 175 fpm across the open tank, and the room relative humidity is 50% RH.
Heat loss = Base table value * Factor A * Factor B * Square feet.
Heat loss = 822*1.8*0.91*100SF = 13,464 Btuh.

When evaporated water is replaced with make-up water that is colder than tank temperature, additional energy use is required, equal to the pounds of lost water (= pounds of make-up water) and the differential temperature between make-up temperature and tank temperature.

Tank **agitation** from air injection or circulation is sometimes used to assist process work, but also increases evaporation substantially. One study comparing a still water heated tank to a frothing tank from aeration measured a **29% increase** in heat load in response to higher evaporation rate.
Source: Heat Loss and Evaporation Rate from an Agitated Water Tank, Huynh, B, Huynh,B., 2009

COOLING TOWER COLD WATER BASIN HEAT LOSS

Courtesy: SPX Corporation

Cooling Tower Cold Water Basin Heat Loss
(Btu/Hr-SF)

	Water Surface	Exposed Basin Sides and Bottom				Concrete Sides and Bottom Below Grade		
Ambient Temp degF		Plain Steel	Insulated Steel	Wood	Concrete 8 inch thick	2 ft Depth	4 ft Depth	6 ft Depth
+30	48	20	2	8	12	0.5	0.3	0.2
+20	78	40	4	16	23	1.0	0.5	0.3
+10	106	60	6	24	35	2.0	1.3	0.8
0	133	80	8	32	47	3.0	2.0	1.3
-10	159	100	10	40	58	4.0	3.0	2.0
-20	185	120	12	48	70	5.0	4.0	3.0
-30	210	140	14	56	82	6.0	5.0	4.0

NOTES

1. Water surface loss is based on 5 mph wind over water surface under fill. Exposed side and bottom losses assume a 15 mph wind, and are weighted by orientation of basin surfaces and test experience.
2. Losses for insulated steel assume at least 1 inch thick outdoor grade insulation with K=0.30 Btu/Hr-SF-in.
3. Water temperature assumed to be 40 degF.

CLEAN ROOM PARTICLES AND AIR CHANGES BY CLASS

Source: Modular Clean Rooms, Denver CO

Clean Room Particles and Air Changes by Class

FED Std 209	Max Particles per ft3	Max Particles per ft3	Typ Air Changes/Hr
Class	0.1 µm	0.5 µm	
1	35	1	540-600+
10	345	10	540-600
100	3,450	100	400-480
1,000	34,500	1,000	120-150
10,000	345,000	10,000	45-60
100,000	3,450,000	100,000	20-30

Maximum Allowable Particles by Class
ISO 14644-1 per m3

ISO	0.1 µm	0.5 µm
CLASS 3	1,000	35
CLASS 4	10,000	352
CLASS 5	100,000	3,520
CLASS 6	1,000,000	35,200

BIN WEATHER DATA FOR 5 CITIES

TMY3 Data

Source: Bin Maker Pro 3.0 software, Gas Technology Institute and Psych Pro software, Linric Co.

Wet bulb, humidity ratio, and dew point values are coincident to dry bulb.

Work Week 1 = 6a-8p M-F, and 10a-5p Sa,Su, 4375 hours.

Work Week 2 = 6a-6p M-F, Sat 10a-4p, Sun off all day, 3432 hours.

		Climate Zone 1						Climate Zone 2					
		Anchorage AK						Colorado Springs, CO					
		24x7	Wk 1	Wk 2				24x7	Wk 1	Wk 2			
Mid-pts	DB (F)	Total Hrs	Total Hrs	Total Hrs	HR (gr /lb)	WB (F)	Dew pt (F)	Total Hrs	Total Hrs	Total Hrs	HR (gr /lb)	WB (F)	Dew pt (F)
117.5	115 to 120												
112.5	110 to 115												
107.5	105 to 110												
102.5	100 to 105												
97.5	95 to 100							1	1		46	61.6	45.8
92.5	90 to 95							28	28	20	45.8	60	45.7
87.5	85 to 90							151	136	97	42.6	57.8	43.8
82.5	80 to 85							336	300	243	45.2	56.7	45.4
77.5	75 to 80	3	3	1	57.2	61.7	51.6	370	297	221	49	55.8	47.5
72.5	70 to 75	15	14	12	55.4	59.5	50.7	409	286	235	54.3	55.2	50.2
67.5	65 to 70	171	145	107	50.2	56.3	48.1	616	363	280	52.5	53	49.3
62.5	60 to 65	569	436	305	48.3	53.8	47.1	935	475	371	50.4	50.7	48.2
57.5	55 to 60	1102	539	417	46.9	51.4	46.3	838	383	303	45.3	47.5	45.4
52.5	50 to 55	1086	504	431	42.6	47.9	43.8	767	385	293	37.1	43.2	40.3
47.5	45 to 50	533	226	179	33.1	42.7	37.4	591	237	183	32.4	39.5	36.9
42.5	40 to 45	540	246	204	28.3	38.5	33.5	721	313	255	25.9	35	31.4
37.5	35 to 40	826	398	313	23.6	34.1	29.4	874	359	298	22.6	31.3	28.4
32.5	30 to 35	808	388	303	20.2	29.9	26.0	675	269	202	19.7	27.7	25.5
27.5	25 to 30	639	300	230	16.5	25.5	21.8	390	158	122	16.8	23.8	22.2
22.5	20 to 25	703	350	278	13.3	20.9	17.3	466	202	170	14.5	19.8	19.1
17.5	15 to 20	838	420	347	10.5	16.1	12.5	360	113	95	12.1	15.5	15.4
12.5	10 to 15	374	169	132	8.1	11.3	7.4	113	46	37	8.9	10.6	9.2
7.5	5 to 10	199	101	76	6.3	6.6	2.5	80	35	27	6.7	5.8	3.7
2.5	0 to 5	143	69	58	4.5	1.5	-3.9	35	10	10	5.5	1.3	-0.1
-2.5	-5 to 0	142	61	45	3.7	-3.2	-7.6	4			5.2	-3	-1.2
-7.5	-10 to -5	61	18	16	2.8	-8	-12.7						
-12.5	-15 to -10	8	2	2	2	-13	-18.6						
-17.5	-20 to -15												

		Climate Zone 3 Portland OR						Climate Zone 4 Atlanta GA					
		24x7	Wk 1	Wk 2				24x7	Wk 1	Wk 2			
Mid-pts	DB (F)	Total Hrs	Total Hrs	Total Hrs	HR (gr /lb)	WB (F)	Dew pt (F)	Total Hrs	Total Hrs	Total Hrs	HR (gr /lb)	WB (F)	Dew pt (F)
117.5	115 to 120												
112.5	110 to 115												
107.5	105 to 110												
102.5	100 to 105	2	2	2	101	77.3	67.2						
97.5	95 to 100	9	9	7	75.3	71.6	59.0	13	13	13	101	75.1	67.3
92.5	90 to 95	12	10	6	65.5	68.3	55.2	127	121	98	102	74	67.6
87.5	85 to 90	62	50	34	68.8	67.4	56.6	424	363	261	103	72.8	67.9
82.5	80 to 85	194	171	124	66.4	65.4	55.6	703	533	395	94.9	70.1	65.6
77.5	75 to 80	284	237	156	64.9	63.4	55.0	879	487	382	94.3	68.5	65.4
72.5	70 to 75	337	251	180	61.7	60.9	53.6	977	432	342	92.3	66.6	64.8
67.5	65 to 70	597	369	280	60	58.7	52.9	944	418	315	75.7	61.6	59.3
62.5	60 to 65	1193	604	498	58	56.3	52.0	1066	468	368	60.9	56.4	53.4
57.5	55 to 60	1351	629	518	53	53	49.6	795	354	278	48.3	51.3	47.2
52.5	50 to 55	1309	660	524	45.4	48.8	45.5	751	327	257	39.3	46.5	41.9
47.5	45 to 50	1068	463	359	40.5	45	42.5	472	155	121	33.6	42.4	37.9
42.5	40 to 45	1137	501	410	34.9	40.8	38.7	594	259	218	28.4	38.2	33.7
37.5	35 to 40	891	344	285	28.7	36	33.8	491	205	162	22.6	33.4	28.5
32.5	30 to 35	278	93	80	25.1	31.8	30.7	221	96	77	15.8	28	21.0
27.5	25 to 30	31	13	13	21.2	27.4	27.1	171	57	53	13.9	24.2	18.3
22.5	20 to 25	5	4	4	17.7	22.5	23.3	103	50	46	11.9	20.1	15.1
17.5	15 to 20							20	12	12	8.7	15.2	8.8
12.5	10 to 15							8	3	3	5.8	10	0.9
7.5	5 to 10							1	1	1	5.7	6.1	0.6
2.5	0 to 5												
-2.5	-5 to 0												
-7.5	-10 to -5												
-12.5	-15 to -10												
-17.5	-20 to -15												

		Climate Zone 5 Tampa FL					
		24x7	Wk 1	Wk 2			
Mid-pts	DB (F)	Total Hrs	Total Hrs	Total Hrs	HR (gr /lb)	WB (F)	Dew pt (F)
117.5	115 to 120						
112.5	110 to 115						
107.5	105 to 110						
102.5	100 to 105						
97.5	95 to 100						
92.5	90 to 95	157	153	124	121	77.7	72.5
87.5	85 to 90	778	740	588	119	76.2	72.0
82.5	80 to 85	1523	1064	822	113	74	70.5
77.5	75 to 80	1780	719	535	112	72.4	70.3
72.5	70 to 75	1139	426	337	97.8	68.3	66.4
67.5	65 to 70	1190	458	379	83.7	63.9	62.1
62.5	60 to 65	893	365	271	67.4	58.6	56.1
57.5	55 to 60	523	174	151	54.2	53.4	50.2
52.5	50 to 55	387	147	120	43.5	48.3	44.5
47.5	45 to 50	167	49	37	36.2	43.7	39.8
42.5	40 to 45	139	52	47	28.4	38.6	33.7
37.5	35 to 40	69	20	19	23.8	34.2	29.6
32.5	30 to 35	12	5	5	14.5	27.8	19.2
27.5	25 to 30	3	3	3	12.7	24	16.4
22.5	20 to 25						
17.5	15 to 20						
12.5	10 to 15						
7.5	5 to 10						
2.5	0 to 5						
-2.5	-5 to 0						
-7.5	-10 to -5						
-12.5	-15 to -10						
-17.5	-20 to -15						

Hours Per Year Below Outside Dry Bulb and Wet Bulb Temperatures

Representative U.S. cities in defined climate zones

Climate zone convention: ASHRAE 90.1-2010, IECC-2009

Hours from TMY3 weather data, **8760 hour bins**

	Zone	1A	2A	2B	3A	3B	3C	4A	4B	4C	5A	5B	5B	6A	6B	7	8
		Miami, Florida	Houston, Texas	Phoenix, Arizona	Memphis, Tennessee	El Paso, Texas	San Francisco, California	Baltimore, Maryland	Albuquerque, New Mexico	Salem, Oregon	Chicago, Illinois	Boise, Idaho	Colorado Springs, Colorado	Burlington, Vermont	Helena, Montana	Duluth, Minnesota	Fairbanks, Alaska
Dry	**< 65F**	934	3244	2997	4453	4410	7775	5721	5740	7222	6309	6718	6852	6995	7384	7710	8109
Bulb	**< 60F**	454	2294	2068	3472	3500	6136	4818	4773	6334	5461	5887	5917	6087	6636	6874	7441
Hours	**< 55F**	178	1670	1338	2868	2741	3366	4097	4028	5076	4784	5031	5079	5382	5860	6101	6727
Per	**< 50F**	32	1100	668	2256	1912	1076	3284	3302	3685	4143	4139	4312	4702	5125	5407	6018
Year	**< 45F**	8	778	339	1716	1320	346	2701	2739	2577	3735	3414	3721	4189	4540	4898	5609
	< 40F	0	406	71	1154	756	56	2069	2018	1394	3140	2577	3000	3478	3785	4347	4634
	< 35F	0	139	0	678	267	0	1313	1133	554	2275	1581	2126	2709	2792	3681	4220
Wet	**< 65F**	1819	4186	6838	5724	7931	8740	6696	8729	8570	7345	8717	8759	8066	8760	8447	8757
Bulb	**< 60F**	1149	3190	5600	4778	6350	8357	5903	7760	7850	6427	8364	8481	7253	8603	7812	8672
Hours	**< 55F**	597	2419	4128	3932	5336	6252	5130	6418	6531	5591	7552	7357	6392	7916	6840	7998
Per	**< 50F**	296	1747	2777	3169	4470	2807	4369	5555	4825	4869	6276	6296	5551	6836	6037	7097
Year	**< 45F**	62	1111	1530	2351	3354	865	3562	4792	3143	4197	4840	5354	4729	5798	5416	6256
	< 40F	7	654	441	1701	2098	124	2797	3837	1634	3549	3629	4377	4075	4822	4784	5623
	< 35F	0	343	84	1070	1042	7	1943	2606	736	2923	2375	3378	3317	3763	4169	5064
	< 30F	0	93	4	634	396	0	1233	1343	298	2011	1266	2372	2603	2749	3461	4512

Hours Per Year Below Outside Dry Bulb and Wet Bulb Temperatures

Representative U.S. cities in defined climate zones

Climate zone convention: ASHRAE 90.1-2010, IECC-2009

Hours from TMY3 weather data, **Mon-Sat 6a-6p bins**

	Zone	1A	2A	2B	3A	3B	3C	4A	4B	4C	5A	5B	5B	6A	6B	7	8
		Miami, Florida	Houston, Texas	Phoenix, Arizona	Memphis, Tennessee	El Paso, Texas	San Francisco, California	Baltimore, Maryland	Albuquerque, New Mexico	Salem, Oregon	Chicago, Illinois	Boise, Idaho	Colorado Springs, Colorado	Burlington, Vermont	Helena, Montana	Duluth, Minnesota	Fairbanks, Alaska
Dry	**< 65F**	308	1159	1127	1668	1689	2991	2258	2242	2781	2446	2656	2623	2806	2971	3112	3409
Bulb	**< 60F**	155	801	755	1270	1305	1987	1905	1855	2329	2120	2312	2222	2416	2647	2752	3073
Hours	**< 55F**	61	548	491	1056	974	991	1602	1526	1853	1861	1978	1887	2142	2353	2463	2740
Per	**< 50F**	14	337	262	814	654	378	1257	1178	1301	1615	1625	1563	1865	2038	2171	2456
Year	**< 45F**	5	244	143	630	441	132	1035	932	901	1470	1314	1358	1664	1790	1973	2313
	< 40F	0	48	0	247	107	0	457	360	182	883	555	752	1098	1481	1764	2123
	< 35F	0	12	0	124	36	0	267	187	76	577	275	525	803	1109	1509	1949
Wet	**< 65F**	740	1676	2831	2361	3271	3753	2775	3752	3630	3019	3727	3767	3395	3744	3612	3777
Bulb	**< 60F**	432	1285	2290	1943	2595	3460	2460	3177	3177	2630	3502	3560	3016	3627	3299	3731
Hours	**< 55F**	229	924	1665	1550	2174	2276	2145	2599	2547	2236	3090	2977	2684	3245	2835	3398
Per	**< 50F**	113	644	1077	1209	1810	939	1792	2304	1849	1971	2519	2595	2283	2804	2507	3009
Year	**< 45F**	26	391	588	880	1253	319	1450	1953	1169	1706	1942	2153	1930	2374	2247	2629
	< 40F	4	234	173	656	714	41	1118	1475	555	1434	1455	1719	1670	1977	1972	2332
	< 35F	0	110	33	397	369	0	737	900	257	1163	907	1291	1352	1510	1728	2117
	< 30F	0	36	3	232	146	0	444	438	102	795	473	863	1059	1089	1417	1915

ALTITUDE CORRECTION FACTORS AT DIFFERENT TEMPERATURES (FA)

Source of data: Greenheck Corporation, www. greenheck.com
Original data inverted (1/x) for use in this text as a multiplying factor (Fa)

The term "Fa" in this text is a collective density correction factor for air, from the effects of altitude and temperature. Industry convention is a factor of 1.0 at sea level and 70F. "Fa" is a multiplying factor, in the context of density.
Table values here have been inverted for use as multiplying factors, e.g. (cfm*TSP*Fa)/6356= Air Hp.

Temp (F)	Altitude, feet										
	0	**1,000**	**2,000**	**3,000**	**4,000**	**5,000**	**6,000**	**7,000**	**8,000**	**9,000**	**10,00**
0	1.149	1.111	1.064	1.031	0.990	0.952	0.926	0.885	0.855	0.820	0.794
50	1.042	1.000	0.962	0.926	0.901	0.870	0.833	0.806	0.769	0.746	0.714
70	**1.000**	0.962	0.926	0.893	0.862	0.820	0.800	0.769	0.741	0.714	0.690
100	0.943	0.909	0.877	0.847	0.820	0.787	0.758	0.730	0.704	0.676	0.649
200	0.800	0.775	0.746	0.714	0.694	0.667	0.641	0.621	0.595	0.571	0.552
300	0.699	0.671	0.649	0.625	0.602	0.581	0.559	0.538	0.518	0.498	0.481
400	0.617	0.595	0.571	0.552	0.532	0.515	0.493	0.478	0.457	0.441	0.422
500	0.552	0.532	0.513	0.495	0.476	0.459	0.442	0.426	0.410	0.394	0.380
600	0.500	0.483	0.465	0.448	0.433	0.417	0.400	0.386	0.372	0.352	0.344
700	0.457	0.441	0.426	0.410	0.395	0.380	0.366	0.353	0.340	0.326	0.315
800	0.420	0.403	0.389	0.375	0.362	0.350	0.336	0.324	0.312	0.300	0.290
900	0.391	0.373	0.360	0.348	0.337	0.326	0.313	0.300	0.289	0.279	0.270
1,000	0.362	0.348	0.334	0.324	0.313	0.302	0.290	0.279	0.268	0.260	0.250

PARSING CBECS DATA

A great deal of effort is put forth by the Energy Information Administration (EIA) to collect data on energy use patterns in the United States and make available to the public. Of particular interest for this handbook is the Commercial Building Energy Consumption Survey (CBECS) data. The CBECS data is presented in established reporting formats that are consistent from year to year, and the information is also available in raw form. The raw data can be culled and sorted for particular interests and can be useful, but requires a little thought.

Examples Data Parsing Considerations:

Major categories include sub-categories. Focusing on a particular sub-category is a common desire, e.g. nursing homes specifically instead of "healthcare-outpatient"	Sorting on **PBA Plus** will allow separation of the sub-categories provided.
"Small" vs. "large" facilities of a given type behave differently and the standard tables only give a single category, e.g. "office."	There are several different size designations. One suggestion is to define "small" as 0-50 kSF, and "large" as >50kSF.
Buildings behave differently in different climates, instead of national averages.	Two methods of sorting are provided: "census region" and "climate zone" which are categorized by heating and cooling degree days. However, using this has produced some questionable results, so use caution.
Overall averages include all submitted data including facilities that report zero energy usage and vacant buildings.	These have no use for energy auditing and are valid to remove. They are identified as "vacant" in PBA, and also "MFUSED" (any major fuel used?) in the raw data set.

Other notes:

1. CBECS staff recommends caution with calculations with less than 20 records. The more the data is drilled down, the less records there are and the less confidence in the results due to small sample size. Checking the number of records for each custom search is strongly recommended. Whether you choose to take stock in an average of less than 20 records is a judgment call.
2. End use data is not generated from customer surveys like the other

```
(Climate Zone / Degree Days)

VALUE $CLIMAT
  '1'='<2000 CDD,>7000 HDD'
  '2'='<2000 CDD,5500-7000 HDD'
  '3'='<2000 CDD,4000-5499 HDD'
  '4'='<2000 CDD,<4000 HDD'
  '5'='>=2000 CDD,<4000 HDD'
```

```
(PBA = Primary Business Activity)

VALUE $PBA
  '01'='Vacant'
  '02'='Office'
  '04'='Laboratory'
  '05'='Non refrigerated warehouse'
  '06'='Food sales'
  '07'='Public order and safety'
  '08'='Outpatient health care'
  '11'='Refrigerated warehouse'
  '12'='Religious worship'
  '13'='Public assembly'
  '14'='Education'
  '15'='Food service'
  '16'='Inpatient health care'
  '17'='Nursing'
  '18'='Lodging'
  '25'='Retail other than mall'
  '26'='Service'
  '91'='Other'
```

data—this data is from modeling.

3. The more finely the data is cut, the smaller the sample size and the more the data anomalies will affect the arithmetic result.

CBECS Format Codes

CBECS provides instructions on how to work with their data set. These include formats, which are the codes to subdivide the data. The codes may change for different survey years, so always check.

Basic Procedure to Parse CBECS Raw Data

Using the raw data, arrive at a EUI benchmark value for a particular category (kBtu/SF-yr) with the following steps:

Sort the rows as needed, such as for "libraries," then
For each row, calculate energy use as (MFBTU * ADJWT), and
For each row, calculate square footage as (SQFT * ADJWT)

Where:

MFBTU	=	annual major fuel consumption (kBtu)
SQFT	=	square footage (SF)
ADJWT	=	sample building weight factor

For the group of rows involved, create two subtotal values:

(A) Sum the weighted energy, e.g. sum all (MFBTU * ADJWT)
(B) Sum the weighted square footage, e.g. sum all (SQFT * ADJWT)

Divide A/B.

NOTE: It may seem that the ADJWT factor cancels and is not needed in this method, however it is required and will yield different results if not used, so use it.

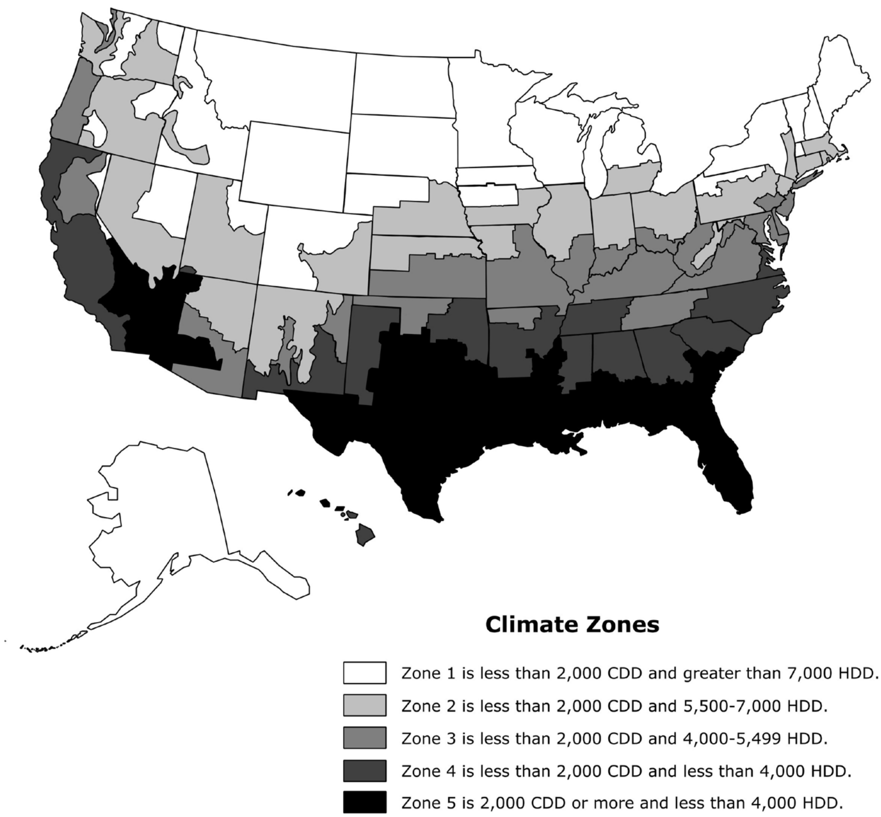
CBECS Categories of PBA-Plus that are within a given PBA (2003)

PBA=Primary Business Activity

"Vacant" and "Other" categories omitted

PBA	Name	PBA-Plus	Name
2	Office	2	Administrative/professional office
		3	Bank/other financial
		4	Government office
		5	Medical office (non-diagnostic)
		6	Mixed-use office
		7	Other office
4	Laboratory	8	Laboratory
5	Non-refrigerated warehouse	9	Distribution/shipping center
		10	Non-refrigerated warehouse
		11	Self-storage
6	Food sales	12	Convenience store
		13	Convenience store with gas station
		14	Grocery store/food market
		15	Other food sales
7	Public order and safety	16	Fire station/police station
		17	Other public order and safety
8	Outpatient health care	18	Medical office (diagnostic)
		19	Clinic/other outpatient health
11	Refrigerated warehouse	20	Refrigerated warehouse
12	Religious worship	21	Religious worship
13	Public assembly	22	Entertainment/culture
		23	Library
		24	Recreation
		25	Social/meeting
		26	Other public assembly
14	Education	27	College/university
		28	Elementary/middle school
		29	High school
		30	Preschool/daycare
		31	Other classroom education
15	Food service	32	Fast food
		33	Restaurant/cafeteria
		34	Other food service
16	Inpatient health care	35	Hospital/inpatient health
17	Nursing	36	Nursing home/assisted living
18	Lodging	37	Dormitory/fraternity/sorority
		38	Hotel
		39	Motel or inn
		40	Other lodging
25	Retail other than mall	41	Vehicle dealership/showroom
		42	Retail store
		43	Other retail
26	Service	44	Post office/postal center
		45	Repair shop
		46	Vehicle service/repair shop
		47	Vehicle storage/maintenance
		48	Other service

CBECS CLIMATE ZONE MAP



BUILDING USE CATEGORIES DEFINED (CBECS)

Source: EIA/CBECS 2003

Education

Buildings used for academic or technical classroom instruction, such as elementary, middle, or high schools, and classroom buildings on college or university campuses. Buildings on education campuses for which the main use is not classroom are included in the category relating to their use. For example, administration buildings are part of "Office," dormitories are "Lodging," and libraries are "Public Assembly."

- elementary or middle school
- high school
- college or university
- preschool or daycare
- adult education
- career or vocational training
- religious education

Food Sales

Buildings used for retail or wholesale of food.

- grocery store or food market
- gas station with a convenience store
- convenience store

Food Service

Buildings used for preparation and sale of food and beverages for consumption.

- fast food
- restaurant or cafeteria

Health Care—Inpatient

Buildings used as diagnostic and treatment facilities for inpatient care.

- hospital
- inpatient rehabilitation

Health Care—Outpatient

Buildings used as diagnostic and treatment facilities for outpatient care. Medical offices are included here if they use any type of diagnostic medical equipment -if they do not, they are categorized as an office building.

- medical office clinic or other outpatient health care
- outpatient rehabilitation
- veterinarian

Laboratories

Buildings whose use is primarily scientific in nature, and that use a lot of specialized equipment. Computer labs, training facilities, and research and development buildings are considered "Office." A high school science or biology lab would most likely be part of a larger classroom building which would be considered "Education," but if it did happen to be a separate building, it would be classified as a lab. Lab departments within a hospital would be considered inpatient health care, but if it did happen to be a separate building, it would be classified as a lab.

- pharmaceutical labs
- water chemistry labs
- blood work labs
- test labs and other labs that are in separate buildings

Lodging

Buildings used to offer multiple accommodations for short-term or long-term residents

- motel or inn
- hotel
- dormitory, fraternity, or sorority
- convent or monastery
- shelter, orphanage, or children's home
- halfway house

Nursing

- retirement home
- nursing home, assisted living, or other residential care
- skilled nursing and other residential care buildings

Office

Buildings used for general office space, professional office, or administrative offices. Medical offices are included here if they do not use any type of diagnostic medical equipment (if they do, they are categorized as an outpatient health care building).

- administrative or professional office
- government office
- mixed-use office
- bank or other financial institution
- medical office (see previous column)
- sales office
- contractor's office (e.g. construction, plumbing, HVAC)
- non-profit or social services
- research and development

- city hall or city center
- religious office
- call center

Retail—Other Than Mall

Buildings used for the sale and display of goods other than food.

retail store

- beer, wine, or liquor store
- rental center
- dealership or showroom for vehicles or boats
- studio/gallery

Public Assembly

Buildings in which people gather for social or recreational activities, whether in private or non-private meeting halls.

- social or meeting (e.g. community center, lodge, meeting hall, convention center, senior center)
- recreation (e.g. gymnasium, health club, bowling alley, ice rink, field house, indoor racquet sports)
- entertainment or culture (e.g. museum, theater, cinema, sports arena, casino, night club)
- library
- funeral home
- student activities center
- armory
- exhibition hall
- broadcasting studio
- transportation terminal

Public Order and Safety

Buildings used for the preservation of law and order or public safety.

- police station
- fire station
- jail, reformatory, or penitentiary
- courthouse or probation office

Religious Worship

Buildings in which people gather for religious activities, (such as chapels, churches, mosques, synagogues, and temples).

Service

Service buildings in which some type of service is provided, other than food service or retail sales of goods

- vehicle service or vehicle repair shop
- vehicle storage/maintenance (car barn)
- repair shop
- dry cleaner or laundromat
- post office or postal center
- car wash
- gas station
- photo processing shop
- beauty parlor or barber shop
- tanning salon
- copy center or printing shop
- kennel

Warehouse and Storage

Buildings used to store goods, manufactured products, merchandise, raw materials, or personal belongings (such as self-storage).

- refrigerated warehouse
- non-refrigerated warehouse
- distribution or shipping center

OPERATING EXPENSES: PERCENT THAT ARE FROM UTILITY COSTS (Cont'd)

Business Type	Percent	Source	Remarks
Colleges	3.2	American School and University (ASU) survey, 2007	Purchased utilities cost divided by Total operating expenses, including payroll (includes teaching staff payroll)
Day Care	2.1	U.S. Census Bureau, Business Expenses: 2002 (NAICS 6244)	Purchased utilities cost divided by Total operating expenses, including payroll
Healthcare - Hospitals	1.3	U.S. Census Bureau, Business Expenses: 2002 (NAICS 622)	Purchased utilities cost divided by Total operating expenses, including payroll
Healthcare - Outpatient	0.8	U.S. Census Bureau, Business Expenses: 2002 (NAICS 6214)	Purchased utilities cost divided by Total operating expenses, including payroll
Hotels and Motels	5.3	U.S. Census Bureau, Business Expenses: 2002 (NAICS 7211)	Purchased utilities cost divided by Total operating expenses, including payroll
K-12 Schools	2.1	American School and University (ASU) survey, 2007	Purchased utilities cost divided by Total operating expenses, including payroll (includes teaching staff payroll)
Linen and Uniform Supply	4.4	U.S. Census Bureau, Business Expenses: 2002 (NAICS 81233)	Purchased utilities cost divided by Total operating expenses, including payroll
Medical and Diagnostic Lab	0.6	U.S. Census Bureau, Business Expenses: 2002 (NAICS 6215)	Purchased utilities cost divided by Total operating expenses, including payroll
Museums, Historical Sites, and Similar Institutions	3.6	U.S. Census Bureau, Business Expenses: 2002 (NAICS 712)	Purchased utilities cost divided by Total operating expenses, including payroll
Nursing Care Facilities	2.1	U.S. Census Bureau, Business Expenses: 2002 (NAICS 6231)	Purchased utilities cost divided by Total operating expenses, including payroll
Office Buildings – Leased Building - tenant salaries not included	20	"Experience Exchange Report", 2006, used with permission from BOMA International. All rights reserved.	Weighted average of Class A and Class B buildings that reported. Percentage value is based on $2.00/SF-yr cost for all utilities. Percentage includes variable and fixed expenses for the buildings, but does not include payroll expenses of the building tenants.

OPERATING EXPENSES:
PERCENT THAT ARE FROM UTILITY COSTS (*Concluded*)

Business Type	Percent	Source	Remarks
Office Buildings – Leased Building - tenant salaries not included			This would be from the viewpoint of the building leasing entity, since the tenant salaries are not included.
Office Buildings - Including payroll (approximate)	1.25	BOMA figure for leased space is 20% of total O/M expenses less payroll, and $2.00/SF-yr. "Experience Exchange Report", 2006, used with permission from BOMA International. All rights reserved. If $2.00/SF-yr is 20% of total, the total is $10/SF-yr This figure was adjusted using an approximate value of $150/SF-yr for salaries in an office building, for a revised total cost of $160/SF-yr including payroll. The same $2/SF-yr for utilities is then 1.2% Source of salary burden approximation: Derived from gathered data in the following document: NEMI, National Energy Management Institute, Productivity Benefits Due to Improved Indoor Air Quality, August 1995, pp 4-8, 4-11.	Approximate value of purchased utilities cost divided by Total operating expenses, including payroll. See Chapter 8 – Building Operations and Maintenance **"Productivity Value"** for this and other categories of building use. "Office" value of $97/SF-yr was adjusted from 1995 dollars at 4% inflation.
Restaurant – Full Service	5.1	U.S. Census Bureau, Business Expenses: 2002 (NAICS 7221)	Purchased utilities cost divided by Total operating expenses, including payroll
Restaurant – Limited Service (Fast Food)	4.6	U.S. Census Bureau, Business Expenses: 2002 (NAICS 7222)	Purchased utilities cost divided by Total operating expenses, including payroll
Retail	2.7	U.S. Census Bureau, Business Expenses: 2002 (NAICS 44-45)	Purchased utilities cost divided by Total operating expenses, including payroll
Supermarkets	4.9	U.S. Census Bureau, Business Expenses: 2002 (NAICS 445)	Purchased utilities cost divided by Total operating expenses, including payroll
Warehouse	4.9	U.S. Census Bureau, Business Expenses: 2002 (NAICS 493)	Purchased utilities cost divided by Total operating expenses, including payroll

OPERATING EXPENSES: PERCENT THAT ARE FROM UTILITY COSTS (MANUFACTURING)

Source: 2007 U.S. Economic Census, EC0731SG2: Manufacturing: Summary Series: General Summary: Detailed Statistics by Subsectors and Industries
Pct operating cost from energy calculated from (energy cost / total operating expenses).
Total operating expenses included payroll, benefits, materials, purchased fuels, purchased electricity, contract work, rental payments, and 'other expenses'.

NAICS Code	Description	% Oper. Cost from Energy
311211	Flour milling	2.4%
311230	Breakfast cereal manufacturing	3.5%
311421	Fruit and vegetable canning	2.8%
311511	Fluid milk manufacturing	1.7%
311615	Poultry processing	2.0%
311812	Commercial bakeries	2.5%
312113	Ice manufacturing	7.2%
312120	Breweries	3.0%
313	Textile mills	4.2%
314	Textile product mills	1.7%
321113	Sawmills	3.0%
322	Paper manufacturing	6.3%
322110	Pulp mills	9.8%
322121	Paper (except newsprint) mills	10.4%
323117	Books printing	2.2%
324121	Asphalt paving mixture and block manufacturing	6.4%
325211	Plastics material and resin manufacturing	5.0%
325311	Nitrogenous fertilizer manufacturing	13.3%
325510	Paint and coating manufacturing	1.1%
326160	Plastics bottle manufacturing	4.9%
326211	Tire manufacturing (except retreading)	2.6%
326220	Rubber and plastics hoses and belting manufacturing	2.1%
327111	Vitreous china plumbing fixture and china and earthenware bathroom accessories manufacturing	5.9%
327320	Ready-mix concrete manufacturing	1.9%
331311	Alumina refining	17.0%
332111	Iron and steel forging	4.2%
332116	Metal stamping	1.5%

NAICS Code	Description	% Oper. Cost from Energy
332322	Sheet metal work manufacturing	1.0%
332811	Metal heat treating	7.6%
332813	Electroplating, plating, polishing, anodizing, and coloring	4.1%
332991	Ball and roller bearing manufacturing	2.8%
334413	Semiconductor and related device manufacturing	2.3%
334418	Printed circuit assembly (electronic assembly) manufacturing	0.6%
335122	Commercial, industrial, and institutional electric lighting fixture manufacturing	0.8%
335312	Motor and generator manufacturing	1.0%
335911	Storage battery manufacturing	3.5%
336111	Automobile manufacturing	0.5%
336214	Travel trailer and camper manufacturing	0.6%
336360	Motor vehicle seating and interior trim manufacturing	0.6%
337	Furniture and related product manufacturing	1.1%

SERVICE LIFE OF VARIOUS SYSTEM COMPONENTS

Note: The figures given are the end of life period when the equipment is scrapped. This does not mean there are no repairs along the way. For example, it is almost certain that an air conditioner or heat pump will get several repairs including a new compressor during the advertised 'life'.

Sources:

1. ASHRAE Applications Handbook, 2003. © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org.
2. Iowa Department of Natural Resources, 2002, "Life Cycle Cost Analysis Guidelines—2002"
3. "Service Life of Energy Conservation Measures" Bonneville Power Administration (July 14, 1987)
4. Stoffer Inspections, L.C. Home Page, "Life Cycles and Approximate Costs to Repair/Replace/Upgrade," 2004 (home inspector)
5. Author estimate
6. Manufacturer's Literature
7. Author's field experience

8. Measure Life Report, Residential and Commercial/Industrial Lighting and HVAC Measures, The New England State Program Working Group (SPWG), GDS Associates Inc, 2007
9. LBNL 2007

Equipment	Normal Replacement Life	Source
Hot Water Boiler	25 years	1
Steam Boiler	30 years	1
Steam Traps	7 years	3
Tank-Type Domestic Water Heater, Gas or Electric	15 years	8
Heat Pump Water Heater	10 years	3
Solar Water Heater	15 years	3
Point of Use Water Heater	10 years	8
Low Flow Shower Head	10 years	8
Centrifugal Chillers	23 years	1
Reciprocating Chiller	12-14 years	2
Screw Chiller	20 years	3
Galvanized Cooling Towers	20 years	1
Package Rooftop A/C Unit	15 years	1
Water Cooled Package Unit	15 years	1
Split System A/C	15 years	1
Fan Coil	20 years	2
VAV Boxes	20 years	1
Hot Water Unit Heaters	20 years	1
Electric Unit Heaters	13	1
PTAC (Packaged Terminal Air Conditioner)	10-15 years	2,1
Computer Room Air Conditioner	10-15 years	2,3
Air Curtain	15 years	8
Gas Furnace	18 years	1
Gas Fired Radiant Tube Heater	10 years	3
Electric Spot Radiant Heater	10 years	8
Air Source Heat Pump	12 years	9
Water Source or Ground Source Heat Pump (closed loop)	19 years	1
Ground Source Heat Pump Bore Field (HDPE pipe material life is 50 years. System life may be less and may be limited by the bond between the grout, the pipe, and the earth and the degradation of heat transfer interface to the earth)	30+ years (estimated)	5

(*Continued*)

Equipment	Normal Replacement Life	Source
Indoor Air Handler	20-25 years	2
Air-Side Economizers	10 years	3
Water-Side Economizers	11 years	3
Electric Baseboard Heat	10-15 years	4
Electric Duct Heater	15 years	1
Hot Water Baseboard Heat	25 years	1
Base Mounted Pump	20 years	1
Sump Pump	10 years	1
Utility Fans	20 years	2
Ductwork (metal)	30 years	1
Air Curtain	10 years	3
Polyethylene Strip Curtain	3 years	3
Kitchen Exhaust Hood Make-Up Air Tempering Unit	10 years	3
De-Stratification Fan	10 years	8
Shell and Tube Heat Exchanger	24 years	1
Heat Pipe Heat Recovery	14 years	3
Rotary Wheel Heat Recovery	11 years	3
Heat Recovery from Refrigeration Condensers	11 years	3
Thermal Energy Storage System (TES) - Ice	19 years	3
Thermal Energy Storage System (TES) - Water	20 years	3
Direct Evaporative Cooling	7-10 years	5
Evaporative Pre-Cooling	8-12 years	5
Indirect-Direct Evaporative Cooling	15-20 years	5
Evaporative Cooling Cellulose Media	5 years	6
Evaporative Cooling Felt Pads	2 years	6
Air Washer	17 years	1
Motors	15-17 years	3
Fractional HP PSC and ECM motors	15 years	8
Variable Speed Drive (VSD / VFD)	13 years	8
Motor Starter	17 years	3
Air Compressor	20	2

(*Continued*)

Equipment	Normal Replacement Life	Source
Lighting Fixture – all types (new)	15 years	8
Ballast – all types	12 years	3
Bulbs, including CFL. Rated in hours	Varies-mfg data	
Occupancy/Motion Sensor	10 years	3
Occupancy Sensor – Plug Loads	10 years	8
Daylight Dimming / Harvesting Systems	10	8
Double Pane Windows	12-20 years	6
Solar Shade Film	10 years	8
Molded Insulation	20 years	1
Blanket Insulation	24 years	1
Control Valves	20 years	1
Dampers	20 years	1
Valve/Damper Actuator - pneumatic	20 years	1
Valve/Damper Actuator – hydraulic	15 years	1
Valve/Damper Actuator – mini hydraulic (for terminal units)	5 years	7
Valve/Damper Actuator – electric – oil filled	10-15 years	7
Valve/Damper Actuator – electric – open air	5-7 years	7
Valve/Damper Actuator – self contained (system powered)	10 years	1
Valve/Damper Actuator – Residential style "clock motor" terminal valves	5 years	7
"Active" control sensors and transmitters (powered-type)	5 years	7
Pneumatic Controls – General	20 years	1
Air Compressor - Controls	15 years	8
Refrigerated Air Drier	15 years	8
Analog Electronic Controls - General	7-10 years	7
EMS / DDC Controls (may be obsolete by technology advances in 7-10)	15 years	8
EMS Software Optimizing	5 years	8
Programmable Thermostat	8 years	8
Retro-commissioning	7 years	8
Energy Audit	3 years	8

EQUATING ENERGY SAVINGS TO PROFIT INCREASE

Energy is an operating expense and energy savings go to the bottom line, providing an increase in overall profitability. The following tables illustrate this effect. The largest profit boost will occur when the businesses profit margin is small and when the percent of operating expense that is from energy is large.

5% Energy Savings

Table shows revised profit value

Original Profit Margin	Energy Cost % of Total Operating Cost										
	1%	**2%**	**3%**	**4%**	**5%**	**6%**	**7%**	**8%**	**9%**	**10%**	**20%**
1%	1.1%	1.1%	1.2%	1.2%	1.3%	1.3%	1.4%	1.4%	1.5%	1.5%	2.0%
2%	2.1%	2.1%	2.2%	2.2%	2.3%	2.3%	2.4%	2.4%	2.5%	2.5%	3.0%
3%	3.1%	3.1%	3.2%	3.2%	3.3%	3.3%	3.4%	3.4%	3.5%	3.5%	4.0%
5%	5.1%	5.1%	5.2%	5.2%	5.3%	5.3%	5.4%	5.4%	5.5%	5.5%	6.1%
10%	10.1%	10.1%	10.2%	10.2%	10.3%	10.3%	10.4%	10.4%	10.5%	10.6%	11.1%
20%	20.1%	20.1%	20.2%	20.2%	20.3%	20.4%	20.4%	20.5%	20.5%	20.6%	21.2%
30%	30.1%	30.1%	30.2%	30.3%	30.3%	30.4%	30.5%	30.5%	30.6%	30.7%	31.3%

10% Energy Savings

Table shows revised profit value

Original Profit Margin	Energy Cost % of Total Operating Cost										
	1%	**2%**	**3%**	**4%**	**5%**	**6%**	**7%**	**8%**	**9%**	**10%**	**20%**
1%	1.1%	1.2%	1.3%	1.4%	1.5%	1.6%	1.7%	1.8%	1.9%	2.0%	3.1%
2%	2.1%	2.2%	2.3%	2.4%	2.5%	2.6%	2.7%	2.8%	2.9%	3.0%	4.1%
3%	3.1%	3.2%	3.3%	3.4%	3.5%	3.6%	3.7%	3.8%	3.9%	4.0%	5.1%
5%	5.1%	5.2%	5.3%	5.4%	5.5%	5.6%	5.7%	5.8%	6.0%	6.1%	7.1%
10%	10.1%	10.2%	10.3%	10.4%	10.6%	10.7%	10.8%	10.9%	11.0%	11.1%	12.2%
20%	20.1%	20.2%	20.4%	20.5%	20.6%	20.7%	20.8%	21.0%	21.1%	21.2%	22.4%
30%	30.1%	30.3%	30.4%	30.5%	30.7%	30.8%	30.9%	31.0%	31.2%	31.3%	32.7%

15% Energy Savings

Table shows revised profit value

Original Profit Margin	Energy Cost % of Total Operating Cost										
	1%	**2%**	**3%**	**4%**	**5%**	**6%**	**7%**	**8%**	**9%**	**10%**	**20%**
1%	1.2%	1.3%	1.5%	1.6%	1.8%	1.9%	2.1%	2.2%	2.4%	2.5%	4.1%
2%	2.2%	2.3%	2.5%	2.6%	2.8%	2.9%	3.1%	3.2%	3.4%	3.6%	5.2%
3%	3.2%	3.3%	3.5%	3.6%	3.8%	3.9%	4.1%	4.3%	4.4%	4.6%	6.2%
5%	5.2%	5.3%	5.5%	5.6%	5.8%	6.0%	6.1%	6.3%	6.4%	6.6%	8.2%
10%	10.2%	10.3%	10.5%	10.7%	10.8%	11.0%	11.2%	11.3%	11.5%	11.7%	13.4%
20%	20.2%	20.4%	20.5%	20.7%	20.9%	21.1%	21.3%	21.5%	21.6%	21.8%	23.7%
30%	30.2%	30.4%	30.6%	30.8%	31.0%	31.2%	31.4%	31.6%	31.8%	32.0%	34.0%

20% Energy Savings
Table shows revised profit value

Original Profit Margin	Energy Cost % of Total Operating Cost										
	1%	**2%**	**3%**	**4%**	**5%**	**6%**	**7%**	**8%**	**9%**	**10%**	**20%**
1%	1.2%	1.4%	1.6%	1.8%	2.0%	2.2%	2.4%	2.6%	2.9%	3.1%	5.2%
2%	2.2%	2.4%	2.6%	2.8%	3.0%	3.2%	3.4%	3.7%	3.9%	4.1%	6.3%
3%	3.2%	3.4%	3.6%	3.8%	4.0%	4.3%	4.5%	4.7%	4.9%	5.1%	7.3%
5%	5.2%	5.4%	5.6%	5.8%	6.1%	6.3%	6.5%	6.7%	6.9%	7.1%	9.4%
10%	10.2%	10.4%	10.7%	10.9%	11.1%	11.3%	11.6%	11.8%	12.0%	12.2%	14.6%
20%	20.2%	20.5%	20.7%	21.0%	21.2%	21.5%	21.7%	22.0%	22.2%	22.4%	25.0%
30%	30.3%	30.5%	30.8%	31.0%	31.3%	31.6%	31.8%	32.1%	32.4%	32.7%	35.4%

25% Energy Savings
Table shows revised profit value

Original Profit Margin	Energy Cost % of Total Operating Cost										
	1%	**2%**	**3%**	**4%**	**5%**	**6%**	**7%**	**8%**	**9%**	**10%**	**20%**
1%	1.3%	1.5%	1.8%	2.0%	2.3%	2.5%	2.8%	3.1%	3.3%	3.6%	6.3%
2%	2.3%	2.5%	2.8%	3.0%	3.3%	3.6%	3.8%	4.1%	4.3%	4.6%	7.4%
3%	3.3%	3.5%	3.8%	4.0%	4.3%	4.6%	4.8%	5.1%	5.4%	5.6%	8.4%
5%	5.3%	5.5%	5.8%	6.1%	6.3%	6.6%	6.9%	7.1%	7.4%	7.7%	10.5%
10%	10.3%	10.6%	10.8%	11.1%	11.4%	11.7%	12.0%	12.2%	12.5%	12.8%	15.8%
20%	20.3%	20.6%	20.9%	21.2%	21.5%	21.8%	22.1%	22.4%	22.8%	23.1%	26.3%
30%	30.3%	30.7%	31.0%	31.3%	31.6%	32.0%	32.3%	32.7%	33.0%	33.3%	36.8%

30% Energy Savings
Table shows revised profit value

Original Profit Margin	Energy Cost % of Total Operating Cost										
	1%	**2%**	**3%**	**4%**	**5%**	**6%**	**7%**	**8%**	**9%**	**10%**	**20%**
1%	1.3%	1.6%	1.9%	2.2%	2.5%	2.9%	3.2%	3.5%	3.8%	4.1%	7.4%
2%	2.3%	2.6%	2.9%	3.2%	3.6%	3.9%	4.2%	4.5%	4.8%	5.2%	8.5%
3%	3.3%	3.6%	3.9%	4.3%	4.6%	4.9%	5.2%	5.5%	5.9%	6.2%	9.6%
5%	5.3%	5.6%	6.0%	6.3%	6.6%	6.9%	7.3%	7.6%	7.9%	8.2%	11.7%
10%	10.3%	10.7%	11.0%	11.3%	11.7%	12.0%	12.4%	12.7%	13.1%	13.4%	17.0%
20%	20.4%	20.7%	21.1%	21.5%	21.8%	22.2%	22.6%	23.0%	23.3%	23.7%	27.7%
30%	30.4%	30.8%	31.2%	31.6%	32.0%	32.4%	32.8%	33.2%	33.6%	34.0%	38.3%

35% Energy Savings
Table shows revised profit value

Original Profit Margin	Energy Cost % of Total Operating Cost										
	1%	**2%**	**3%**	**4%**	**5%**	**6%**	**7%**	**8%**	**9%**	**10%**	**20%**
1%	1.4%	1.7%	2.1%	2.4%	2.8%	3.2%	3.5%	3.9%	4.3%	4.7%	8.6%
2%	2.4%	2.7%	3.1%	3.4%	3.8%	4.2%	4.6%	4.9%	5.3%	5.7%	9.7%
3%	3.4%	3.7%	4.1%	4.5%	4.8%	5.2%	5.6%	6.0%	6.4%	6.7%	10.8%
5%	5.4%	5.7%	6.1%	6.5%	6.9%	7.3%	7.6%	8.0%	8.4%	8.8%	12.9%
10%	10.4%	10.8%	11.2%	11.6%	12.0%	12.4%	12.8%	13.2%	13.6%	14.0%	18.3%
20%	20.4%	20.8%	21.3%	21.7%	22.1%	22.6%	23.0%	23.5%	23.9%	24.4%	29.0%
30%	30.5%	30.9%	31.4%	31.8%	32.3%	32.8%	33.3%	33.7%	34.2%	34.7%	39.8%

40% Energy Savings
Table shows revised profit value

Original Profit Margin	Energy Cost % of Total Operating Cost										
	1%	**2%**	**3%**	**4%**	**5%**	**6%**	**7%**	**8%**	**9%**	**10%**	**20%**
1%	1.4%	1.8%	2.2%	2.6%	3.1%	3.5%	3.9%	4.3%	4.8%	5.2%	9.8%
2%	2.4%	2.8%	3.2%	3.7%	4.1%	4.5%	4.9%	5.4%	5.8%	6.3%	10.9%
3%	3.4%	3.8%	4.3%	4.7%	5.1%	5.5%	6.0%	6.4%	6.8%	7.3%	12.0%
5%	5.4%	5.8%	6.3%	6.7%	7.1%	7.6%	8.0%	8.5%	8.9%	9.4%	14.1%
10%	10.4%	10.9%	11.3%	11.8%	12.2%	12.7%	13.2%	13.6%	14.1%	14.6%	19.6%
20%	20.5%	21.0%	21.5%	22.0%	22.4%	23.0%	23.5%	24.0%	24.5%	25.0%	30.4%
30%	30.5%	31.0%	31.6%	32.1%	32.7%	33.2%	33.7%	34.3%	34.9%	35.4%	41.3%

50% Energy Savings
Table shows revised profit value

Original Profit Margin	Energy Cost % of Total Operating Cost										
	1%	**2%**	**3%**	**4%**	**5%**	**6%**	**7%**	**8%**	**9%**	**10%**	**20%**
1%	1.5%	2.0%	2.5%	3.1%	3.6%	4.1%	4.7%	5.2%	5.8%	6.3%	12.2%
2%	2.5%	3.0%	3.6%	4.1%	4.6%	5.2%	5.7%	6.3%	6.8%	7.4%	13.3%
3%	3.5%	4.0%	4.6%	5.1%	5.6%	6.2%	6.7%	7.3%	7.9%	8.4%	14.4%
5%	5.5%	6.1%	6.6%	7.1%	7.7%	8.2%	8.8%	9.4%	9.9%	10.5%	16.7%
10%	10.6%	11.1%	11.7%	12.2%	12.8%	13.4%	14.0%	14.6%	15.2%	15.8%	22.2%
20%	20.6%	21.2%	21.8%	22.4%	23.1%	23.7%	24.4%	25.0%	25.7%	26.3%	33.3%
30%	30.7%	31.3%	32.0%	32.7%	33.3%	34.0%	34.7%	35.4%	36.1%	36.8%	44.4%

INTEGRATED DESIGN EXAMPLES

An important concept in economically successful energy efficiency designs is that of Integrated Design, whereby energy reduction in one element results in a reduction in size of a related supporting element, thereby subsidizing or even paying for the measure. Examples are:

- The added cost of high performance suspended film windows can be offset in some climates by the elimination of perimeter fin-tube heating.
- Down-sizing cooling systems in conjunction with reduced lighting power design.
- Down-sizing electrical systems in conjunction with reduced HVAC equipment load and sizes.
- Down-sizing electrical systems in conjunction with reduced transport energy costs (smaller fans and pumps from amply sized ducts and pipes).
- Down-sizing cooling systems in conjunction with improved window shading coefficients, films, or exterior shading systems.
- Down-sizing heating and cooling systems by using 1% or 2% ASHRAE design outdoor weather conditions instead of 0.4%, allowing the temperature to drift up a few degrees a few hours of the year.
- Down-sizing a boiler or hot water unit by virtue of selecting higher efficiency equipment. Since output is the design driver, it is often possible to utilize the next smaller size unit, but at higher efficiency, to achieve the same result.
- Down-sizing primary heating cooling equipment in conjunction with upgrades in envelope elements like insulation or, especially, window shading.
- Down-sizing primary heating and cooling equipment in conjunction with heat recovery systems.
- Down-sizing fan and pump motors, and primary cooling equipment, by increasing duct and pipe sizes, filter areas, coil areas, etc., as the tradeoff for using less transport energy. Note that the extra heat of transport energy elements, especially fans, often drives the equipment size up a notch.
- Down-sizing overhead lighting and HVAC cooling via an owner commitment to a greater use of task lighting.
- Switching a data center cooling system from air-cooled to water-cooled allows the incorporation of a water-side economizer and thousands of hours of free cooling in some climates.

ENERGY AUDIT APPROACH FOR COMMERCIAL BUILDINGS

Note: These are the basics. Depth of data gathering, analysis, and detail will vary.

Preliminary

- Identify the business principal building activity (PBA), and understand basic common drivers of energy use.
- Know the end uses that constitute the majority of the overall energy use, and focus on those.
- Know the percentage of annual operating cost that is from energy expense, which will indicate how important energy cost and energy savings are. The relative proportions of pct energy expense to total, and percent net profit, will indicate the impact of energy savings on bottom line business profit.
- Identify any leasing arrangements that exist, and the extent that they deter incentives for energy improvements. This requires knowing who pays for what, and how long is the term of the lease.

Utility Rate Review

- Identify unit cost of energy for gas and electric, and on and off peak times.
- Identify unit cost of demand kW, as well as any ratchet clause provisions and power factor charges.
- Determine the cost of water/ sewer, to be able to evaluate the give-and-take of proposals that change air-cooled to water-cooled technologies.

Utility History Review

- Identify patterns and anomalies. These may be unexplained high or low uses associated with particular months of the year.
- Establish the energy use index (kBtu/SF-yr) and compare to benchmark data if available.
- Benchmark data will gage whether the customer's use seems reasonable, or unusually high or low. This helps the customer make a general comparison to peer competition.
- Benchmark data will also provide early indication of relative potential savings. If the benchmark is valid, then a customer with energy use per SF above that can easily see what percent of reduction is

needed to fall in line with their peers.

- For buildings with multiple 'business" use, determine proportions and create an "adjusted" benchmark. This will provide insight as to where the energy use should be
- In most cases, use kBtu/SF-yr as the unit, not $/SF-yr since this is more stable (utility rates change).
- Determine load profile by month for insight into end use patterns. For example, boilers left running all summer are usually readily apparent in such graphs if the only other source of gas use is domestic hot water.
- If available obtain load profile by hour for even better insight (interval data). Few customers have time of use meters, but it is good to ask.
- Determine load factor and fraction of bill that is demand. Poor load factors are sometimes an opportunity to spread out load.
- Review energy use with respect to weather to understand the extent to which energy use is weather dependent.
- Determine the overall 'blended' rate of electric cost in $/kW and $/therm. Note that the blended rate is applicable to buildings that do not have demand charges, or where demand charges are not calculated separately. If demand savings are to be calculated, the blended rate cannot be used.

Questionnaire (See Appendix for Sample Questionnaire)

- Inquire about usage habits, hours of occupancy, number of people, when equipment is turned on and off, etc.
- Inquire about basic energy using systems including lights, HVAC, process equipment.
- Inquire about computer rooms, size, and cooling load. Cooling load is a good indicator of computer equipment actual load, since they balance. Cooling load is a good indicator of computer equipment actual load, since they balance. Sub metered data for computer rooms is always preferable.
- Ask questions that lead to the understand of the primary energy use points.
- Determine if process equipment dominates the usage, and to what extent process equipment loads can be influenced. In some cases, energy use for process equipment cannot be reduced.
- Prepare the customer to have a person to escort you through the facility for 4-8 hours Large facilities will take more time.

Analysis and Report

- Estimate energy use from various sources. Compare sum total of expected energy use to actual utility bills and typical end-use break down where applicable benchmarks and "pie charts" exist. This step is a vital sanity check.
- Take great care not to over-estimate savings. Not only would owner confidence be lost, but their budget may depend upon the accuracy of the estimates. Take extra care for savings estimates that depend on a change in user behavior, and document these.
- Review measures and calculations for overlapping effects, where savings do not add or where benefits complement and amplify.
- Review measures for 'watch-outs' where an energy saving measure has unintended negative consequences.
- Estimate demand savings, in relation to local rate structure. For example, demand may only be considered by the utility at certain times, such as a peak time. There are many subtle differences in utility rates.
- Look hard at control routines being used. Controls are underutilized in most facilities.
- Look at ventilation. Often ventilation is less than it is supposed to be and correcting this will add energy costs.
- Recommendations should begin with points of use and work back to the source
- For equipment side, larger and less efficiency equipment should be targeted
- Group the recommendations
- Strategic—smart choices to make for long term energy frugality
- Low-Cost—relatively easy, quick items
- Capital—investment required, cost identified, payback or ROI. It may be helpful to sub-group capital measures into medium and very large size projects.
- For cost estimates, include design fees, escalation, and contingency. To the customer, all of the associated costs are part of "the cost."
- Remember that retrofit costs are higher than new construction costs.
- For equipment that is in need of replacing anyway, the "cost" should not be normal replacement, it should be only for the efficiency upgrade or early replacement portion. Energy conservation payback should not be saddled with normal replacement costs; these should be budgeted anyway for normal expected life spans.

- Identify system issues noted, safety issues noted, etc. that provide additional value to the customer
- Non-energy benefits, such as labor savings and pollution avoidance, may or may not be included.
- Low cost or hard-to-quantify measures need not be excluded. These can be value-add features of the audit, and also help provide a safety margin for savings estimates.
- Remember the focus of the business—profit.

ENERGY AUDIT LOOK-FOR ITEMS

General

- Visit major equipment and large points of energy use
- Walk the interior spaces and spot-measure temperatures and lighting levels
- If utility graphs show usage is temperature dependent, look hard at building envelope items and HVAC systems
- Large exhaust flow points that are adjacent to coincident large make-up air points
- Negative pressure building
- The expectation for energy savings to pay for normal replacement work

"Look For" Items – Utilities

- Customers that do not see the bills for purchased energy and/or are not accountable for it. This leads to complacency
- Advertisements for 'free utilities' in housing. This leads to complacency
- Equipment run concurrently that could be spread out, and reduce demand charges
- Electric space heating that could be staged through the control system, esp. when it all comes on for a few hours straight after a cold weekend.
- Oversized equipment with low power factor, aggravating demand-related charges
- Electric heating equipment that could be "fuel switched" to combustion heating

"Look For" Items – Maintenance

- Significant deferred maintenance and normal replacement work. Equipment abandoned in place
- Good documentation, including O/M, drawings, control sequences?
- Maintenance outsourced (and the institutional knowledge that goes with it)?
- Systems not well understood by operators
- Heat exchangers kept clean? Fouling is most likely in open systems or if no chemical treatment. Note if end plate bolt caps have had wrenches on them frequently.
- Sealed hydronic systems with automatic fill that is plain water; if leaks over the years without replenishing chemicals, can be very fouled inside.
- Approach temperatures unreasonable for heat exchangers and equipment
- How does the cooling tower basin water look? That's what the condenser tubes see
- Look at the filters and also the coils inside – often the filters are changed just before the audit starts
- Filter bypassing.
- Very strong negative pressure on air handler casing, indicating blockage.
- Air economizers on package units that do not lock out compressors. Easy to test by forcing the controls to call for cooling on a day that is below the cutout setting (e.g. 50 degF when set to cut out at 55 degF) and touring the roof to see what is humming.
- Air handler coils in series. No cleaning provision means they have not been cleaned
- Energy savings achieved by eliminating outside air intake
- Comfort improved by zeroing VAV box minimums, eliminating outside air to the zone

"Look For" items – Envelope

- Single Pane Glass in cold climates
- No wall insulation
- Minimal (1-2 inch) roof insulation. A good place to view this is the cut-out at a roof scuttle.
- Leaks around doors, window frames, and operable windows, often with complaints of cold drafts in winter.

- Large east and west glass exposures that aren't shaded or Low-E coated.
- Return plenum not sealed at building perimeter, coupled to the envelope.
- Reports of very cold spots (complaints) in winter which can be from envelope leaks into return plenums. Best to search for these in cold weather, near edges. Not uncommon to find construction defects that allow air into the plenum, and freezing temperatures above the ceiling.
- Building elements and geometry that form chimneys and high infiltration loads.
- Poor, missing, or wet insulation, evidenced by temperature measurements. For example, if the inside surface temperature of the roof is lower than the floor temperature, the roof is pulling heat from the building rapidly. Missing insulation in walls is easily found with a thermometer.
- "Holes" in the insulation envelope, from poorly sealed windows, skylights, window frames with no thermal break, single pane glass, exhaust openings with no dampers, rooftop units with no dampers or back draft dampers, un-insulated curbs, un-insulated doors, un-insulated crawlspaces.
- Missing insulation, detected on hot or cold days from infrared thermometer.

"Look For" Items – Domestic Hot Water

- Domestic water heater circulating pump left on all the time
- Excessive temperature of water
- Missing mechanical insulation
- Electric resistance heat when combustion heating is available
- Domestic water heater "side arm" from the boiler, requiring the boiler to be active year-round, or short cycle in summer
- 180F rinse water in dishwashers, usually with an electric booster heater

"Look For" items – Other

- "Tail wagging the dog" conditions, where an anomaly or small area is driving energy use in a large or common system. For example, one room in a hospital driving the entire chiller plant, one cold complaint starting a boiler in summer, a small data closet keeping a large air

handler or cooling unit running, etc.
- Grossly oversized equipment; short cycling or throttling a lot

"Look For" Items – Lighting
- Over lit areas
- Inefficient lighting
- Excessive hours of operation
- Lights on in unoccupied areas, such as offices and shared spaces (meeting rooms, break rooms, copy rooms)
- Lights on adjacent to large glass areas or beneath skylights
- Outdoor lighting on during the day
- Lights on in unoccupied areas, and also outdoor lights on during the day. Lights are a double-effect energy item with regard to cooling
- Lights on at night or weekend from cleaning crew
- Over-lit areas

"Look For" Items – HVAC and Mechanical
- Inherently low efficient HVAC system types including constant volume reheat, double duct, multi zone, and induction
- 100% outside air units
- Low efficiency cooling equipment.
- Low efficiency heating equipment, esp. gravity burner systems
- Low efficiency equipment inherent from small heat exchange surface areas, evidenced from high approach temperatures, high stack temperatures, etc. even when clean and with proper flow
- HVAC fan horsepower in relation to cooling horsepower. Common (but not great) practice ends up with fan HP being half of the cooling HP, i.e. 100 ton cooling unit with a 50 Hp fan. Usually paired with tight ductwork, tough to change after the fact
- Zoning issues, restrictive ducts and other systemic issues that will hamper any system, regardless of efficiency
- Air cooled equipment or cooling towers located too close to each other or near a screen wall enclosure – recirculating and ingesting hot discharge air.
- High velocity sections, usually accompanied by noise, that indicate restriction and high delivery loss
- Loose fins on air cooled equipment
- Short cycling equipment indicating oversized equipment, lack of modulating control, or control staging error

- Part load parasitic losses such as distribution losses. An example is a condenser pump that continues to run when a chiller is off.
- Electric humidifiers
- Humidified areas with heavy exhaust or coupled to HVAC return air (becomes constant operation for the humidifier)
- Year-round cooling loads not served by economizer equipment

"Look For" items – Central Mechanical Systems

- Boilers with too much excess air (combustion test required)
- Some very old chillers can consume 50 percent more power than modern units
- Cooling towers undersized in proportion to equipment served. It is common to undersize a cooling tower to save money and compensate with a large fan motor and/or higher approach. Rules of thumb for good energy efficiency from amply sized cooling towers are 0.05 kW/ton and 7 degree approach at full load using local design wet bulb temperature.
- Blocked plate frame (no strainers) that is not doing much
- Oversized pumps that were never trimmed – evidenced by discharge balance valves significantly closed, acting as head dissipation devices and constant power loss
- Missing mechanical insulation
- High stack temperature

"Look For" Items – Automatic Temperature Control

- Continuous operation of equipment
- DDC schedule times empty or 00:00 to 00:00 or 00:01 to 23:59 which results in constant operation
- Constant set points
- Programmable thermostats accessible to occupants – usually the programming varies widely
- Overlap fighting from heating/cooling, humidify/dehumidify, pressurize/dissipate
- Extended schedules for the sake of convenience and no phone calls, but which increases operational time unnecessarily, i.e. building occupancy 7a-6p for most people and control schedule starts at 3am and runs to 11pm
- Baseboard heat zoning not matching HVAC air zoning and controls fight

- Areas kept too warm or too cold, when obvious. For example if 74F summer and 70F winter are normal for a given area, 70F summer and 74F winter is an opportunity.
- Control valves and dampers that don't fully close. Can cause false loading.
- Primary heating and cooling equipment running concurrently.
- Duct static pressure set too high
- Supply air reset on VAV air handlers that is from return air (return air temperature is constant, so this does practically nothing)
- Zone resets based on zone temperature rather than zone demand.
- Primary and zone level controls not coordinated. i.e. air handler controls without feedback from zone demands. This leads to throttling losses.
- Zone control that remains active when air handlers shut off. This can keep unoccupied space temperatures at occupied temperatures levels (no reset).
- DDC control that does not extend to zone level. This greatly limits opportunities for optimization that uses zone level demands as the basis for reset (enough but just enough). Zone level controls that include local 2-hour override button allow basic scheduling to be compressed to reduce run time.
- Space temperature adjustments greater than 2F allowed from local tenant
- Double duct terminal unit controls with single actuator/drag link control (heat/cool overlap)
- Chiller condenser water temperature settings too high
- Control overlap in air handlers for sequenced items
- Chillers left on in winter
- Boilers left on in summer
- Flat plate water economizer running 5F delta T or less with full pump flows
- VAV minimums set too high
- VAV box minimum air flow setting for heating is higher than for minimum cooling
- Parallel VAV box not using the fan as stage 1 of heat
- Series VAV box with fan cfm greater than primary air flow at wide open air valve
- Exhaust fans with no control, left to run continuously.
- Underutilized morning warm-up sequence that relies too much on

electric heat – evidenced from very small gas usage with very large gas burners on packaged rooftop units

- VAV systems with electric heat in the boxes and gas preheat capability can operate as gas heating in unoccupied setback mode and lock out electric heat. Same amount of heat, but less cost
- Windows open in cold weather (drive-around), indicating defective heating controls and overheated interior spaces
- Pneumatic controls that are not responsive. Oil fouled, or just broken and abandoned

"Look For" Items – Pools

- Pools with large difference between water temperature and surrounding air temperature humidity, causing (Pw-Pa) and evaporation rates to be high

"Look For" Items – Restaurant/Kitchen

- High heat loss around oven doors indicating poor seals
- Multiple kitchen hoods on a single switch when their required times of use are greatly different
- Walk-in cooler doors propped open
- Blocked condensers from moist kitchen air
- Kitchen hoods left on all the time
- Kitchen cooking equipment left on all the time, or started early and not used for hours later
- Hoods without make-up air, taking building air

"Look For" Items – Laundry

- High remaining moisture content in commercial laundry, from too short spin cycle
- Missing mechanical insulation

"Look For" Items – Steam and Condensate

- Missing mechanical insulation
- Capped/abandoned condensate piping
- Steam traps that have not been regularly dismantled for cleaning and repairing
- "Off" boilers with high water alarm or manual blow down, condensing steam from "on" boilers

"Look For" Items – Data Centers
See **Chapter 24 – Special Topics,** ***Data Center Efficiency***

"Look For" Items – Compressed Air
- **See Chapter 14, Compressed Air**

"Look For" Items – Manufacturing and Process
- High heat loss around oven doors indicating poor seals
- Un-covered heated tanks
- New energy used for heating when adjacent to a similar grade of waste heat
- Equipment running all the time instead of only when needed
- Standby losses from idling equipment
- Exhaust without direct make-up replacement air
- For process equipment, determine the relationship between production and energy use. E.g. when equipment is idle, is it still consuming energy?

ENERGY AUDIT - SAMPLE QUESTIONNAIRE / CHECKLIST

Note: not all sections apply.
This list is intended as a reminder of some things to discuss in the initial interview and things to be watchful for in the field. Most of the items are system-based questions where the answers lead to a potential opportunity. Specific equipment-level data is needed in addition to this information. In all cases, the most important tool in energy auditing is the skill of the person. For water survey concepts, see **Chapter 22 "Water Efficiency."**

Customer Objectives

What do you hope to get out of this energy audit?
How will you define success of this energy audit?
What portion of your monthly operating costs are from utilities (energy)?
Do you plan to implement recommended low cost measures? Capital measures?
For capital projects, is there a hurdle rate for financing? Payback period or IRR? Bear in mind that short paybacks requirements restrict opportunities.
Past and Future: Describe any prior or planned energy saving projects.
From an energy usage standpoint, which area(s) of your operation are you most concerned with? If multiple, please rank in order of priority
Are you able to fund normal replacements separate from upgrades and improvements? For example, 20 year old rooftop AC units are a normal replacement item (needed anyway). Energy savings will not pay for normal replacement work and it is best to fund those projects separately, or combine finding for the normal replacement portion of the energy project.

Documentation Requested

If you have them, please provide the following (copies) for us to review: ○ Automatic control shop drawings and sequences of operation. This would include DDC and conventional controls (pneumatic, etc) ○ A facility map showing the different areas, that we can have to write on? This could be an 'evacuation' or 'life safety' 11x17 map. ○ Architectural drawings ○ Mechanical and electrical design drawings and equipment schedules ○ Test and balance reports Submittal data O/M data
Additional data: ○ Sub meter records ○ Long term trends of central plant cooling and heating output (tons, Mbh, pct load, run time, flows) ○ CO2 trends in areas where CO2 sensors are used ○ Boiler test slips (Long term trends over time show load profiles, occupancy profiles)

Utilities

Do you see utility bills?
Is energy use monitored, tracked, logged, and benchmarked by month and year including usage and demand? Interested in starting to record and track energy use?
Are there any measures in place to monitor and control building electric demand? If monthly or annual usage or demand increases past the benchmark, what happens?
If any areas are sub-metered, a 12-month history of electrical or other energy use as applicable would be helpful. Example: leased areas, data centers, large equipment, etc.
Please note each utility meter for energy use, and the meter ID.
Any buildings *not* separately metered to identify its specific energy use?
Are there heating processes (water heating, HVAC, or process) that use electric resistance for heating? ○ Water heating VAV boxes or duct heaters ○ Process tank heaters Ovens, steam generators Please indicate these so we can see them.

General

Overall building square footage.
Percent that the building is occupied / utilized. Any areas closed off? Vacancy?
Indicate the building SF that is heated and cooled. Or, areas not conditioned.
Indicate the building SF that is heated and cooled by the primary equipment (chillers and boilers) vs. unitary packaged HVAC equipment.
What is the building vintage? ○ Different ages of different additions? ○ Please describe each major addition, when it occurred, and what was done.
What are the principal activities in the building? For each use that applies, indicate the name and areas in SF. ○ Assembly Data Center Education ○ Food Service Laboratory Laundry ○ Library / Museum Warehouse ○ Lodging Multi-Family Office ○ Worship Retail Service ○ Health Care – Inpatient Health Care - Outpatient ○ Other ○ Manufacturing (Please provide SIC or NAICS code)
Building occupied (open/close) times for occupants, by day.
HVAC start/stop times, by day
Number of people that normally occupy the building, their hours of occupancy.

Maintenance

Is there a lot of deferred maintenance from under-funding or lack of staff?
Please describe any equipment that is known to be broken, defective, or need replacing.
How often are filters changed? What types/efficiencies of filters are used?
How often are air handler / furnace coils cleaned? (Applies to both heating and cooling coils, indoor and outdoor coils).
How often are boilers and gas appliances checked for operating efficiency? This requires flue gas testing – can you provide printouts of the results?
How often are boiler heat exchanger surfaces (fire side and water side) cleaned?
How often are chiller condenser tubes cleaned?
How often are building temperature controls calibrated?
How often are VAV box controls checked to see if they are still 'alive'?
How often are control valves checked for positive seating to avoid internal leak-by?
For water-cooled equipment (cooling towers, evaporative cooling, air washers, fluid coolers, etc.) do you have records of water used for a 12-mo. period?
Kitchen - How often are refrigeration evaporator (cold) coils and condenser (heat rejection) cleaned?
Compressed Air - How often are compressed air pipes, hoses checked for leaks?
Steam and Condensate - How often are steam traps checked for internal leak-by?
Data Center – how often are water cooled shell/tube heat exchangers cleaned? How often are dry cooler coils cleaned?
Other heat exchangers on a cooling loop (such as ice machines, refrigeration machines, dedicated cooling units, process cooling loop) – how often are water cooled heat exchangers cleaned?
Fluid Cooler – heat exchanger coil pack ever cleaned?
Plate-Frame heat exchanger – ever cleaned? Are there strainers at inlet connection?
Evaporative cooling / air washers – what is the procedure and frequency for cleaning and sanitizing? Is the sump automatically drained and filled with fresh water? How often?

Envelope

Has any insulation been added or replaced as part of a roofing project since the building was built?
Is the building glass single or double pane?
Are there areas where cold draft complaints exist, to indicate loose or leaky construction? If yes, which areas are these?
Is the building pressure positive, negative, or neutral? Does this vary by season? Do you notice air pushing out the front entrance doors or air being sucked in at the front entrance doors? Any other places where air movements are noticeable, such as rear doors, elevators, or corridors?
Is there a building entrance vestibule?
Any chronically cold areas over ceilings near the edges of the building in winter? Areas where there are common complaints for being too hot or cold?

Domestic Water Heating

How is domestic water heated?
Does domestic water heating depend upon central boiler operation?
Any point-of-use water heaters?
If showers: o What is the flow rate of the shower heads? o Average # of showers per week?
Are there stack dampers on the water heaters?
What temperature hot water is supplied? o Tempering valve? Circulating pump? If yes, how is it controlled?

Other

Where do *you* think most of your energy is going? Sources of waste?
Any other major equipment pieces that affect energy use?
Anything that runs all the time that could be turned off sometimes?
Anything you know to be defective and not working properly?
Anything that is kept at a constant temperature or pressure all the time that could be turned off sometimes?
Any equipment (pumps, fans, compressors, cooling equipment, heating equipment) known to be way oversized? There can be large throttling losses if this is the case.

Lighting

What types of interior lighting are used and where? ○ Fluorescent T-5? Fluorescent T-8? Fluorescent T-12 ○ Compact fluorescent Metal Halide or other HID lights ○ Incandescent or halogen lights Track lights ○ LED
Is task lighting used? Spot lighting / accent lighting used?
Lighting problems: ○ Are any areas you think are over-lit? Under-lit ○ Other lighting issues? (glare, flicker)
How is the building interior lighting controlled? Indicate where each is used ○ Always on Manually controlled Occupancy sensor ○ Computer control Bi-level control Daylight harvesting
Exterior lighting: What type of lights? ○ Metal Halide Mercury Vapor Incandescent or halogen ○ LED
How are exterior lights controlled? ○ Photocell? Dusk-to-dawn timer Dusk=on / timer=off ○ Bi-level control based on time Always on Could exterior lights be run for fewer hours? Or do security concerns prevent this?
How are enclosed parking lights controlled? Perimeter lighting separately circuited?
What are the scheduled hours of interior lighting operation?
What are the scheduled days and hours of after-hour cleaning crews that extend hours of operation?

HVAC / Mechanical

Describe the end-use HVAC systems – what the zone level thermostats are controlling. ○ Single zone package units? VAV box, series fan powered? ○ VAV box, parallel fan powered? VAV box, no fans ○ VAV box, no heat Fan coil ○ Multizone Reheat coil ○ Baseboard PTAC ○ Ductless split Other
Describe the air handlers ○ Single zone heating and cooling Cooling only constant volume reheat ○ Cooling only variable volume 100% outside air ○ Multi-zone Dual duct
Thermal Break Even Temperatures: For this building, is there a pattern where you can predict: ○ Below what OA temperature is heat will be used? ○ Above what OA temperature cooling will be used?
What mechanical equipment and areas are noisiest? Hottest?
Are there any areas that are humidity sensitive? e.g. clean rooms, etc.
Are there any areas that have year-round 24x7 cooling requirements? Do these spaces cause the entire main HVAC system to run to cool a small area?
Is spot-cooling or spot-heating used?
If there are any areas with routine heating cooling problems (cold complaints, hot complaints) note them so we can see these areas.
Are there any areas with indoor air quality (IAQ) complaints? Stuffy areas?
Demand Controlled Ventilation (DCV) Is CO2 control used to reduce outside air during low occupancy? If yes, is it measured by individual zone or is it measured in the return plenum for 'averaging'?
Are there any areas that have requirements for simultaneous heating and cooling such as a dehumidification cycle or process control?

HVAC / Mechanical (cont'd)

Do you have any dual duct, multizone, or constant volume terminal reheat HVAC systems? If yes, please note the locations so we can see these.
For areas served by air conditioning (cooling), which have air-side economizers? Which do not?
Are there any air handlers that have been retrofitted with VFD's where the original inlet vanes are still there?
If humidifiers: ○ Electric? Gas? Steam? Used for: ○ Winter comfort? Static electricity? Data center? Process?
If evaporative cooling: ○ Swamp coolers? Air washers? Used for: ○ Kitchen make up? Warehouse? Factory?
If fume hoods: ○ Run all the time? Turned off when not in use? ○ Sash doors to reduce air flow?
Are any of the air handlers using 100% Outside Air? If yes, note where they are so we can see them
Are there transfer ducts or transfer fans used as make-up for exhaust only areas, such as restrooms, kitchen, etc?
For VAV air handlers, how is the fan capacity accomplished? ○ inlet vanes ○ VFD ○ other
Are there any dedicated heating or cooling equipment for certain areas that are separate from the primary heat/cool systems, such as computer rooms units and other de-centralized equipment?
Are there any heat recovery systems? If yes, please indicate where. ○ Pre heating/ pre cooling outside air? Pre heating boiler feed water? ○ Pre heating domestic hot water? Process?

Central Mechanical Systems

Do you have a 'two pipe' systems? This is where the distributed piping carries either heating water or cooling water but not both.
Electric chillers: ○ Age / condition? Tons capacity? Air-cooled or water cooled? ○ kW/ton or data sheets that show amps at full load cooling. ○ Is capacity modulation method by inlet vanes or VFD? ○ Are redundant units or do all the units run together during summer? ○ Do the units ever short-cycle during low load? ○ Any retrofits for refrigerant replacement? Units run in winter?
If absorption chillers: ○ Age / condition? Steam or hot water or direct fired? ○ Do they operate with waste heat or new-energy heating?
Boilers: ○ Age / condition? Input Btuh capacity? Fuel? Dual fuel? ○ Forced draft or natural draft (draft hood)? ○ Capacity control: Start-stop? Hi/Lo Fire? Modulating? ○ Are redundant units or do they all run together during winter? ○ Do the units ever short-cycle during low load? ○ When a boiler is off (not firing), is the system hot water still being pumped through it? (hot water boiler only) ○ Economizers for inlet air? Feed water (steam)? Return water (HW boiler)? ○ Stack dampers used? Units run in summer?
Cooling Towers and Fluid Coolers: ○ Tons capacity? Fan horsepower? Typical outlet temperature in summer? ○ Fan type (propeller or centrifugal)? Drive (direct, gear, belt)? ○ Modulation method (on-off? two-speed? VFD?) ○ Type of water treatment? Cycles of concentration?
Plate Frame heat exchanger: ○ Capacity? Used for?
Building differential temperature (delta-T): Chilled water? Heating water?
How are mild weather / low loads met with chillers/boilers: ○ Stages of equipment? Jockey (pony) small units? ○ Throttling? Cycling on/off? False loading?
Hydronic pumping: ○ CHW: Constant or variable flow? ○ HHW: Constant or variable flow? Mixing valve used for building water supply?

Automatic Controls

Air-Side Economizer: ○ Control sequence? When are compressors locked out? Any low limit?
Water-Side Economizer ○ Control sequence? Activated based on wet bulb temperature? Other? ○ Is water-economizer used where air economizer could be used instead? ○ Is there variable speed control of pumps during water economizer operation?
Scheduled Operation of Equipment ○ Indicate weekday control times vs. weekend control times, and 'special' schedules ○ Indoor lighting Outdoor lighting Air handlers Exhaust fans ○ Chillers Boilers Pumps Other
When main HVAC equipment is off or on setback, are terminal units able to provide heating? I.e., are they also controlled by nighttime control schedule or are the terminal unit controls independent?
Control System Set Points: If the control system is DDC, we can learn a lot from screen shots (pictures and lists of settings) either saved to a file or put onto a thumb drive, if that is acceptable. ○ Zone temperature settings for heating and cooling. Dead band between? ○ Unoccupied set back temperature values? ○ Variable pumping and fan control pressure set points? ○ Fan tracking sequences of operation? Describe. ○ Cooling tower leaving water temperature set point and reset method, if any? ○ Temperature > chiller is locked out? Temp < boiler locked out? ○ Supply air temperature set point and reset schedule, if any. Reset from outside air? Return air? Space temperature? Or not at all? Describe. ○ Hot water temperature set point and reset schedule, if any? Describe. ○ Chilled water temperature set point and reset schedule, if any? Describe. ○ Demand Controlled Ventilation CO2 set point? ○ VFD-controlled pump control pressure setting and reset, if any?
Do air handler controls allow any overlap? Or are their firm dead bands to prevent it? ○ Heating-cooling overlap? Mixed air-mechanical cooling overlap? ○ Evaporative cooling-mechanical cooling overlap?
For VAV boxes, what are the following settings for five percent of the typical boxes: ○ Max cooling CFM Min cooling CFM ○ Min heating CFM (if different than min cooling CFM) ○ Parallel VAV boxes, fan sequenced as heating stage 1 before heating?

Automatic Controls (cont'd)

What other optimization routines are being used? ○ Optimal start? Summer morning cool-down using outside air? ○ Winter morning warm-up? (And does it keep the outside air damper closed?) ○ Reset: Condenser water? Chilled water? Heating water? Supply air? ○ Fan static pressure reset from OA? ○ DCV (Demand Controlled Ventilation) (reducing outside air proportionally when there are less people)? ○ Occupancy control of VAV box minimums and daytime setbacks? ○ Optimal chiller/boiler sequencing from efficiency curves / mapping? ○ Fan/pump control pressure reset from greatest individual demand? ○ Demand limiting? ○ Approach temperature monitoring for heat exchanger equipment cleaning?
Zone override capabilities (by the user) ○ Lights? Space temperature? How long is the override, once pushed? ○ Temperature adjustment limited?
Multi-zone or double duct systems only ○ Hot deck / cold deck temperature setting and reset schedule, if any.
Constant volume reheat systems ○ Supply air temperature set point and reset method, if any. Reset from outside air? Reset from zone demand? Reset from zone temperature?
Operational issues with controls: ○ Temperature swings? Hunting (oscillating back and forth) control? ○ When last were the instruments calibrated?
Where are two-way and three-way control valves used?
Are control valves tested to assure they are closing tightly?
Control valves and dampers ○ Condition? Stuck dampers? Valves that leak by internally? ○ Any barometric relief dampers? Any failed actuators (esp. terminal units)? ○ Inlet vane dampers left in place from VSD project?
Outside air damper position in occupied and unoccupied modes?
At night when air handlers shut down, do the associated exhaust fans shut down too? Any exhaust fans not on automatic control that run all the time?
Outside air damper position in occupied and unoccupied modes?
Identify/ list control set points that are constants
How closely to on/off schedules match actual occupancy patterns? Can schedules be compressed? Could they be compressed by integrating zone level override buttons?

Special Use: Swimming Pools

Hours of operation for each pool?
Square foot surface area, volume, and water temperature for the pool?
What are the temperature and humidity settings for the air surrounding the pool?
How is humidity controlled in the pool area?
Does the pool air connect to any other parts of the building besides the pool?
Do you find a lot of condensation on the walls or roof?
Do you cover the pool?
Filtration: ○ If sand filters, what is the blow down rate (gpm), times per week, and duration per event?
Pool Heating: ○ How is the pool heated? ○ Any condensing gas-fired heaters used? ○ Any electric heaters?
Is the make-up water sub metered?
Does the pool leak?
Does air move from the lockers toward the pool, or the other way?
Is there heat recovery for pool area exhaust to preheat incoming make-up air?
What are the air change rates for the pool area, occupied and unoccupied?
What are the water change rates for the pool, occupied and unoccupied?

Special Use: Restaurant / Kitchen

Square feet of kitchen?
Serving Breakfast / Lunch / Dinner?
Meals per Day?
Is the kitchen cooking equipment used to capacity? I.e. underutilized cooking equipment (oversized equipment) being used?
Hours of Operation: ○ Restaurant Hours for Customers? ○ Extended Store Hours for Staff?
How many hoods?
Hood Control – Exhaust and Make-up Interlocked to Run Together? Or, is there a single make up air unit for multiple hoods?
For multiple hoods, do they all start/stop from a single switch?
Exhaust over Dishwasher? Controlled with the dishwasher? Or always on?
For each hood: ○ Hood used for? ○ Hood Control – When On / When Off? ○ Are Hoods Run Only When Cooking? ○ Ceiling temperature just outside of hood discolored or hot?
Hood Make-Up Air ○ Cooling Tempered with DX or evaporative cooling or neither? ○ Heating Tempered with electric or gas heat or neither? How warm?
Is the kitchen comfortable or uncomfortable during cooking periods?
Indicate Gas or Electric ○ Grille ○ Fryer ○ Ovens
Cooking Equipment ○ Cooking equipment turned off or down during slow periods? ○ Cooking equipment turned off at night? ○ Hot equipment adjacent to and warming surfaces of refrigerated equipment? ○ Food Heater Lamps Used? Do the lamps stay on even without food under them?
Are oven door seals tight fitting? Any discoloration around seals indicating heat loss? Food warmer box door seals tight fitting?

Special Use: Restaurant / Kitchen (cont'd)

Stand By Losses: ○ Is heating equipment turned off when not being used, or left running? ○ Other equipment turned off when not in use?
Indicate operating temperatures: ○ Walk-in cooler(s) ○ Reach-in cooler(s) ○ Walk-in freezer(s) ○ Reach-in freezer(s)
Walk-in cooler/freezers: ○ Latches hold door closed tightly? ○ Door gaskets in good condition? ○ Doorway plastic curtains used to keep cold air inside? ○ Door-bottom sweep in good condition? ○ Walk-in door for freezer located inside walk-in cooler?
Any equipment discharging warm air into the kitchen? ○ Air cooled ice machines ○ Air cooled reach-in cooler ○ Air cooled walk-in coolers or freezers
Thermostats: ○ Thermostat programmable? ○ Settings set back at night? Temperature settings: ○ Kitchen-summer ○ Kitchen-winter ○ Is the kitchen area heated? ○ Dining-summer ○ Dining-winter ○ Hood make-up air – summer ○ Hood Make-up air - winter
Does the kitchen A/C unit have an air economizer? Does the kitchen A/C unit have a heater?
Kitchen Water Heating ○ Water Heater – Gas or Elec? ○ Stack damper? ○ Water temperature? ○ Does the dishwasher / pot washer use 180 degree hot water or chemicals for sanitizing rinse?

Special Use: Laundry

Extraction: ○ Centrifugal Press ○ Percent extraction of washed clothes before drying ○ Remaining moisture content (RMC)?
Waste water heat recovery to incoming water?
Equipment turned off when not in use?
Temperature of hot water as low as possible?
Cooling system: ○ Air conditioning ○ Swamp cooler ○ Just exhaust
Is space heating provided in the laundry?

Special Use: Steam and Condensate

What is the steam pressure at the point of use?
Steam Pressure: ○ Is steam being used at the lowest possible pressure? ○ Has an attempt been made to operate at reduced pressure? ○ Are there large uses that must be pressure-reduced from the main system supply that could otherwise be generated at a lower pressure?
What is the steam used for? Comfort heating? Process heating / cooking? Humidification?
What percent of the condensate is returned to the system? (E.g. condensate piping leaks?)
Any heat recovery economizers for preheating boiler air or water?
Is the boiler system turned off during unoccupied times?
Stack dampers used?

Special Use: Computer Room / Data Centers

Square footage of data center?
Overhead lights on continuously or on an occupancy sensor?
Room temperature and humidity settings: ○ What are the temp/humidity settings? ○ Could relative humidity be lowered (to 25-30%)? ○ Could space temperature be raised (to 78-80F)?
Humidifiers: ○ Any humidification equipment other than what is in the CRAC/CRAH units? ○ What separates the humidified areas and non-humidified areas? ○ Is there a tight vapor barrier to outdoors, adjacent floors and adjacent un-humidified spaces? (Room envelope air-tight, including walls, doors, above ceilings, below raised floors)
For chilled water CRAC units, what is the chilled water supply temperature? Could chilled water set point be raised to 50 degrees?
For chilled water systems, is the chiller dedicated to the data center or does it also provide comfort cooling to another space such as office area?
What is the supply and return air temperature?
How the room is ventilated (fresh air)?
Any electric heat in the CRAC/CRAH units? Could it be removed?
Unintentional dehumidification ○ Is condensate visible in the drains or on the cooling coils? ○ Are the coil faces wet to touch? ○ What is the supply air temperature? (DX units)
Are heating and cooling mode lights ever on in opposite modes for adjacent equipment?
Are humidification and dehumidification mode lights ever on in opposite modes for adjacent equipment?
Computer Equipment ○ Is the total computer equipment load known? ○ An equipment list may be useful, but sum of data plate values is more than actual load. Operator logs for a main UPS may provide this information if it serves all the computer equipment.
Describe any cooling economizers in use for data center cooling, either at the unitary CRAC/CRAH level or central cooling

Special Use: Computer Room / Data Centers (cont'd)

Is there any connection between this space and the building HVAC system? I.e. do the HVAC air streams mix at all or are they fully isolated? Please describe any mixing of the two systems.
Is the data center power sub metered? If yes, it will be helpful to have historical data for the power use profile.
Is the IT power (UPS output) measured/recorded separately from total data center power?
Type of CRAC/CRAH units: ○ Air-cooled split with compressor indoors and condenser outdoors. ○ Glycol fluid heat exchanger, compressor indoors and dry cooler outdoors. ○ Glycol fluid heat exchanger, compressor indoors and fluid cooler (or heat exchanger + cooling tower) outdoors. ○ Chilled water: No individual compressors, all cooling from air cooled chiller outdoors. ○ Chilled water: No individual compressors, all cooling from water-cooled chiller ○ Rear door ○ Direct liquid-cooled ○ Other (describe)
HVAC enhancements: ○ Hot/cold aisle ○ Pre-cooling ○ Heat recovery ○ Glycol fluid heat exchanger, compressor indoors and dry cooler outdoors. ○ Glycol fluid heat exchanger, compressor indoors and fluid cooler (or heat exchanger + cooling tower) outdoors.

Special Use: Compressed Air

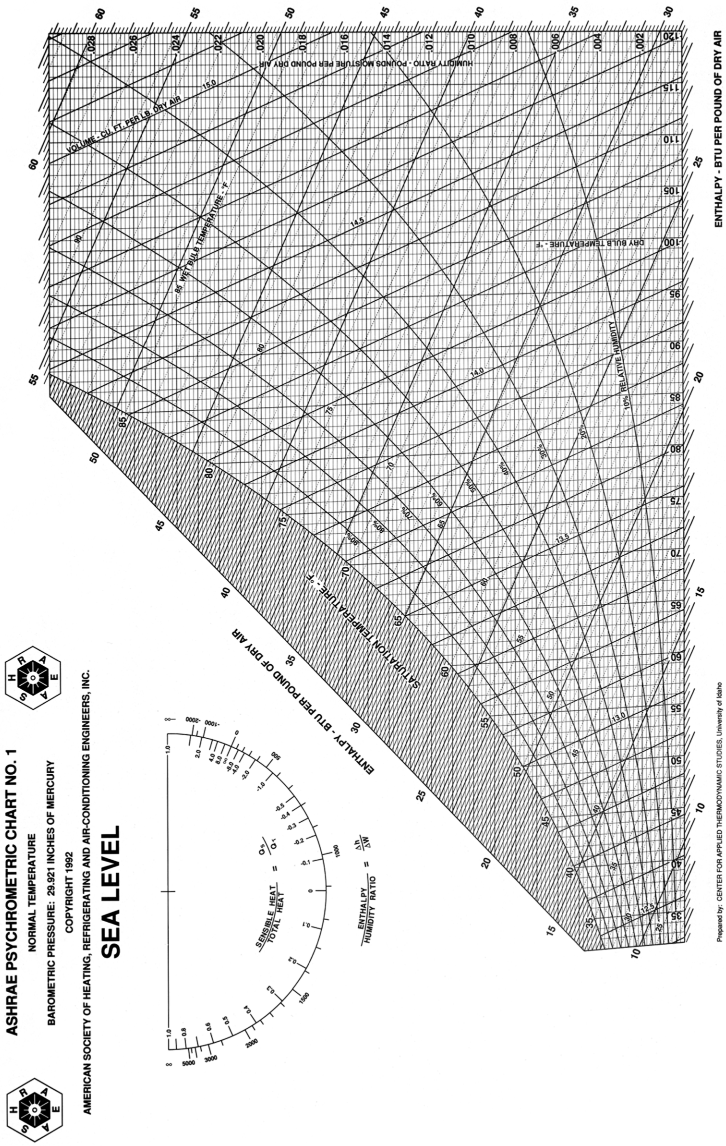
Air Pressure: ○ What is the compressed air pressure at the point of use? ○ Are there large uses that must be pressure-reduced from the main system supply that could otherwise be generated at a lower pressure? ○ Is the system delivering air at the lowest possible pressure? ○ Is pressure boosted to accommodate an occasional high demand by a particular piece of equipment? ○ Has an attempt been made to operate at reduced pressure?
Anything using air that shouldn't be?
Anything using air constantly that could be used only when needed?
What tasks that use air pressure could be served from low pressure blowers? ○ Blow off? ○ Other low pressure use?
Is the outside air taken from the building or from a separate duct?
Any heat recovery from the compressed air system? E.g. utilizing warm air from the intercooler?
Is CA turned off when not in use? Is the compressed air system turned off during unoccupied times?
Drier ○ Refrigerated? What dew point temperature? ○ Desiccant? What dew point temperature? ○ Air drier than it needs to be? If desiccant: ○ Purge based on time? ○ Purge based on air dew point? ○ Purge air heated? ○ Other purge strategy?
Compressor operating modes? ○ On/off? ○ Load-unload? ○ Modulate? ○ Blow off?
How are compressors staged on? Staged off?
How and how often do you check and correct air leaks?
Is CA flow metered? Records to show load profile over time?

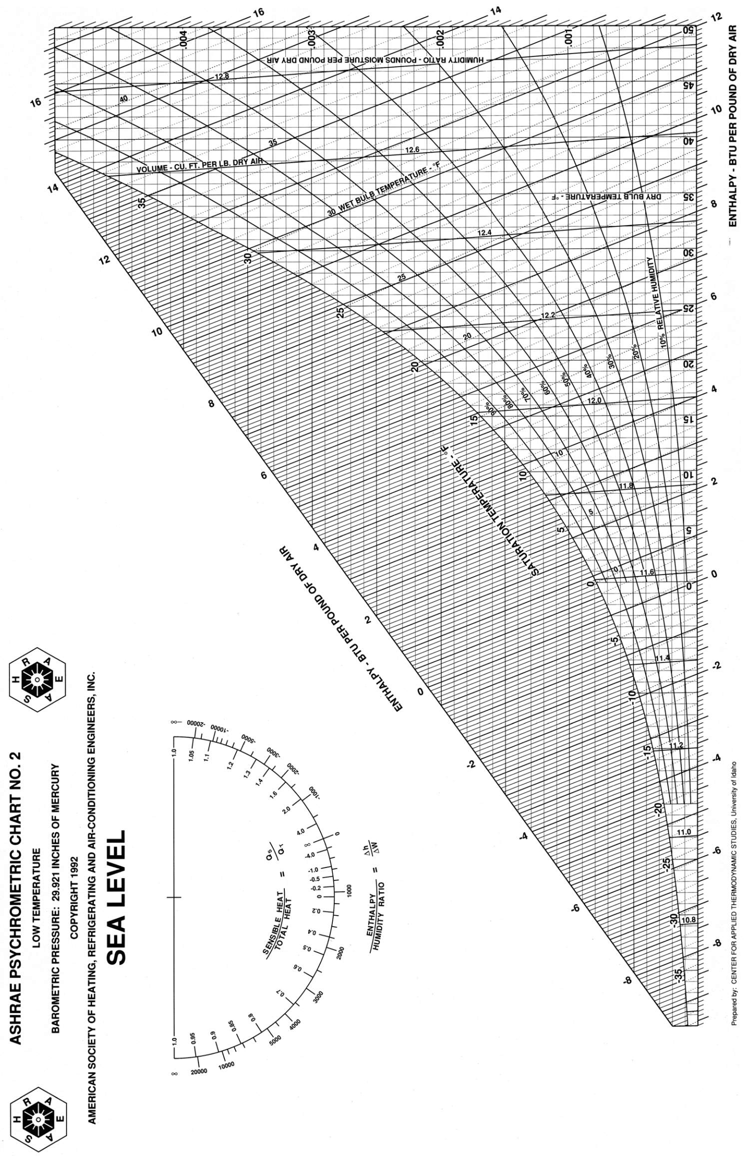
Special Use: Manufacturing / Process

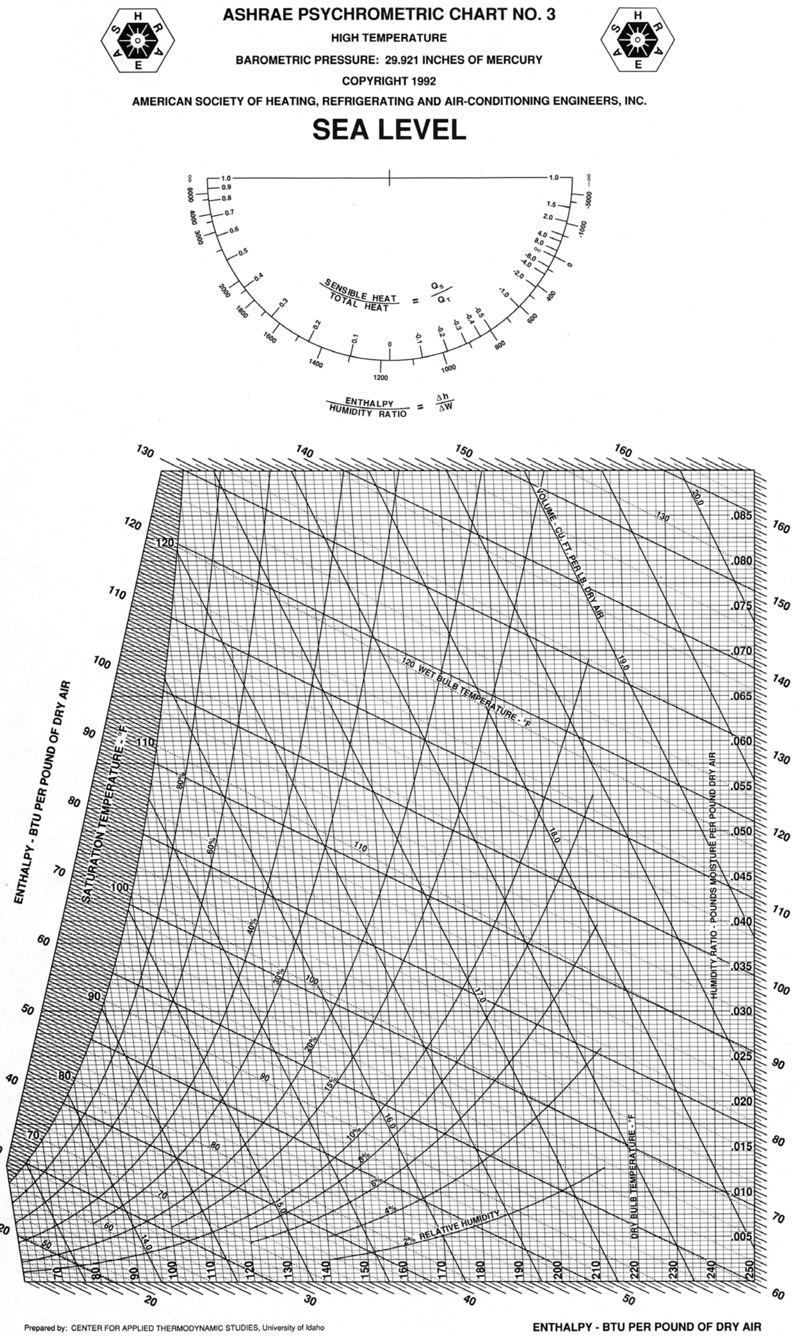
Explain the process, from start to finish. ○ Ideally, map the steps, and identify the significant energy in/out points. ○ Is the process repeating? So that energy use can be a function of production? Or, random / no pattern / different products?
Of shifts? Hours for each shift? Days per week, by shift?
Do utility costs track production levels? I.e. at half production capacity, do utility costs lower to around half? If not, why not?
What percent of operating cost is the expense of utilities?
Are utility costs attributed to the manufactured product? I.e. the energy cost viewed as an ingredient in each part?
Load factor: ○ What portion of the electric bill comes from demand? ○ Are you able to spread out / coordinate times of use of electric-intensive activities, to avoid low load factors and high demand charges? ○ Do you see any energy cost impact from Just-In-Time production methods?
Are oven door seals tight fitting? Any discoloration around seals indicating heat loss?
Stand By Losses: ○ Any equipment that runs all the time, even when no production? Why? What would be needed for those things to turn off when production stops? (i.e. workarounds) ○ Is heating equipment turned off when not being used, or left running? ○ Any machines have a 'standby mode' (like a sleep mode)
Cooling in the manufacturing area: ○ Air conditioned? Swamp coolers? Spot cooling? Just exhaust
Heating in the manufacturing area: ○ Is heat being lost to high bay areas instead of at the floor where workers are? ○ Spot heating? Dock doors habitually left open in winter? Which areas are hottest? Are they hot even in winter?
Are heated tanks covered or open? Can they be covered? Electric heat? Steam? Hot water?
Process exhaust fans: ○ Turned off when process is not active? ○ Any that run continuously for fume hoods, paint booths, dust collection, etc? ○ How are the individual exhaust drops controlled when the point of use is not active? Examples: a branch shut off damper, sash door, on-off switch, etc. ○ Any dust/oil removal exhausters with filtration and recirculation of exhaust air?
Any heat recovery? Any opportunities for heat recovery you know of?
Any idea what the process energy use per unit of production should be?
Where do *you* think most of the energy use goes?

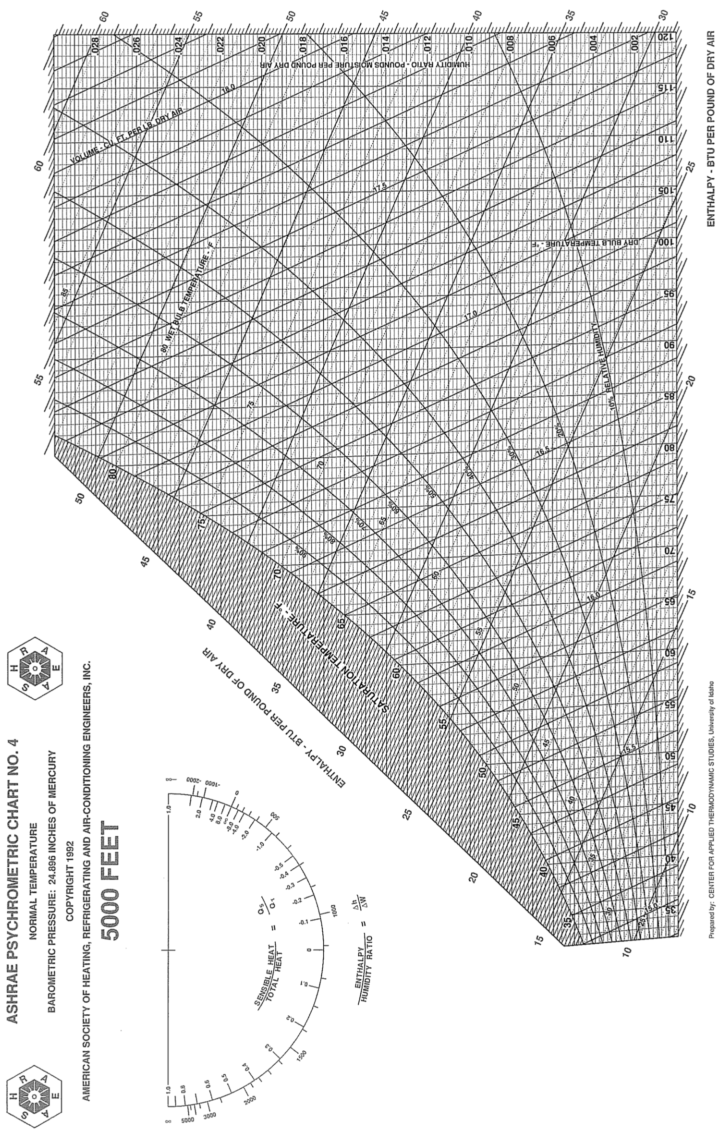
ASHRAE PSYCHROMETRIC CHARTS 1-5

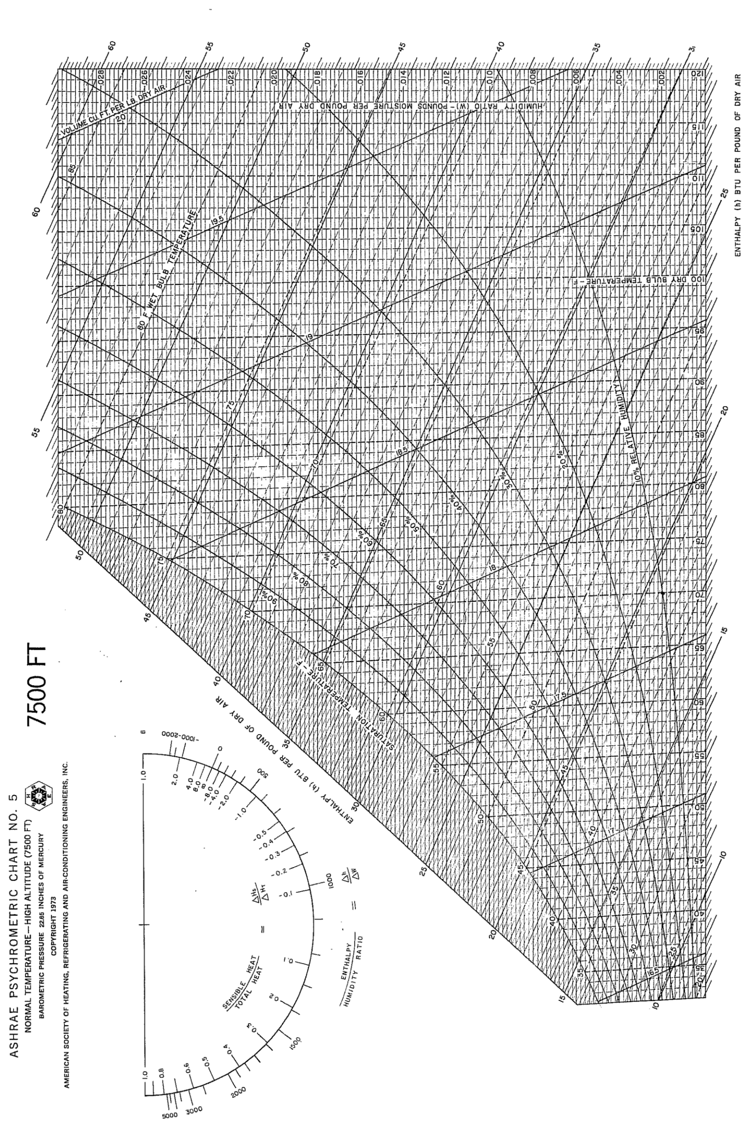
Source: ASHRAE, © American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., www.ashrae.org.




ASHRAE PSYCHROMETRIC CHART NO. 2
LOW TEMPERATURE
BAROMETRIC PRESSURE: 29.921 INCHES OF MERCURY
COPYRIGHT 1992
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
SEA LEVEL
SENSIBLE HEAT / TOTAL HEAT = Qs / Qt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY - BTU PER POUND OF DRY AIR
SATURATION TEMPERATURE - °F
WET BULB TEMPERATURE - °F
VOLUME - CU. FT. PER LB. DRY AIR
HUMIDITY RATIO - POUNDS MOISTURE PER POUND DRY AIR
DRY BULB TEMPERATURE - °F
RELATIVE HUMIDITY
Prepared by: CENTER FOR APPLIED THERMODYNAMIC STUDIES, University of Idaho


ASHRAE PSYCHROMETRIC CHART NO. 3
HIGH TEMPERATURE
BAROMETRIC PRESSURE: 29.921 INCHES OF MERCURY
COPYRIGHT 1992
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
SEA LEVEL
SENSIBLE HEAT / TOTAL HEAT = Q_S / Q_T
ENTHALPY / HUMIDITY RATIO = $\Delta h / \Delta W$
ENTHALPY - BTU PER POUND OF DRY AIR
SATURATION TEMPERATURE - °F
120 WET BULB TEMPERATURE - °F
VOLUME - CU. FT. PER LB. DRY AIR
HUMIDITY RATIO - POUNDS MOISTURE PER POUND DRY AIR
DRY BULB TEMPERATURE - °F
2% RELATIVE HUMIDITY
Prepared by: CENTER FOR APPLIED THERMODYNAMIC STUDIES, University of Idaho
ENTHALPY - BTU PER POUND OF DRY AIR


ASHRAE PSYCHROMETRIC CHART NO. 4
NORMAL TEMPERATURE
BAROMETRIC PRESSURE: 24.896 INCHES OF MERCURY
COPYRIGHT 1992
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
5000 FEET
ENTHALPY - BTU PER POUND OF DRY AIR
SATURATION TEMPERATURE - °F
HUMIDITY RATIO - POUNDS MOISTURE PER POUND DRY AIR
DRY BULB TEMPERATURE - °F
WET BULB TEMPERATURE - °F
VOLUME - CU. FT. PER LB. DRY AIR
RELATIVE HUMIDITY
SENSIBLE HEAT / TOTAL HEAT = Q_S / Q_T
ENTHALPY / HUMIDITY RATIO = $\Delta h / \Delta W$
Prepared by: CENTER FOR APPLIED THERMODYNAMIC STUDIES, University of Idaho


ASHRAE PSYCHROMETRIC CHART NO. 5
NORMAL TEMPERATURE—HIGH ALTITUDE (7500 FT)
BAROMETRIC PRESSURE 22.65 INCHES OF MERCURY
COPYRIGHT 1973
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
7500 FT
ENTHALPY (h) BTU PER POUND OF DRY AIR
HUMIDITY RATIO (W) - POUNDS MOISTURE PER POUND DRY AIR
VOLUME CU FT PER LB DRY AIR
WET BULB TEMPERATURE
SATURATION TEMPERATURE
DRY BULB TEMPERATURE
10% RELATIVE HUMIDITY
SENSIBLE HEAT / TOTAL HEAT
ENTHALPY / HUMIDITY RATIO

REFRIGERANT REPLACEMENT MATRIX

A few common refrigerant replacements are noted. The list changes over time.

Any consideration of replacing a refrigerant in an existing machine should be done with great caution and with the approval of the manufacturer.

Refrigeration application categories of high, medium, and low temperature are in reference to the evaporator refrigerant temperatures. Generally:

	Refrigerant Temp (degF)	Process temp (degF)
High	25-40	45-60
Medium	10-32	30-45
Low	<12	Below Freezing

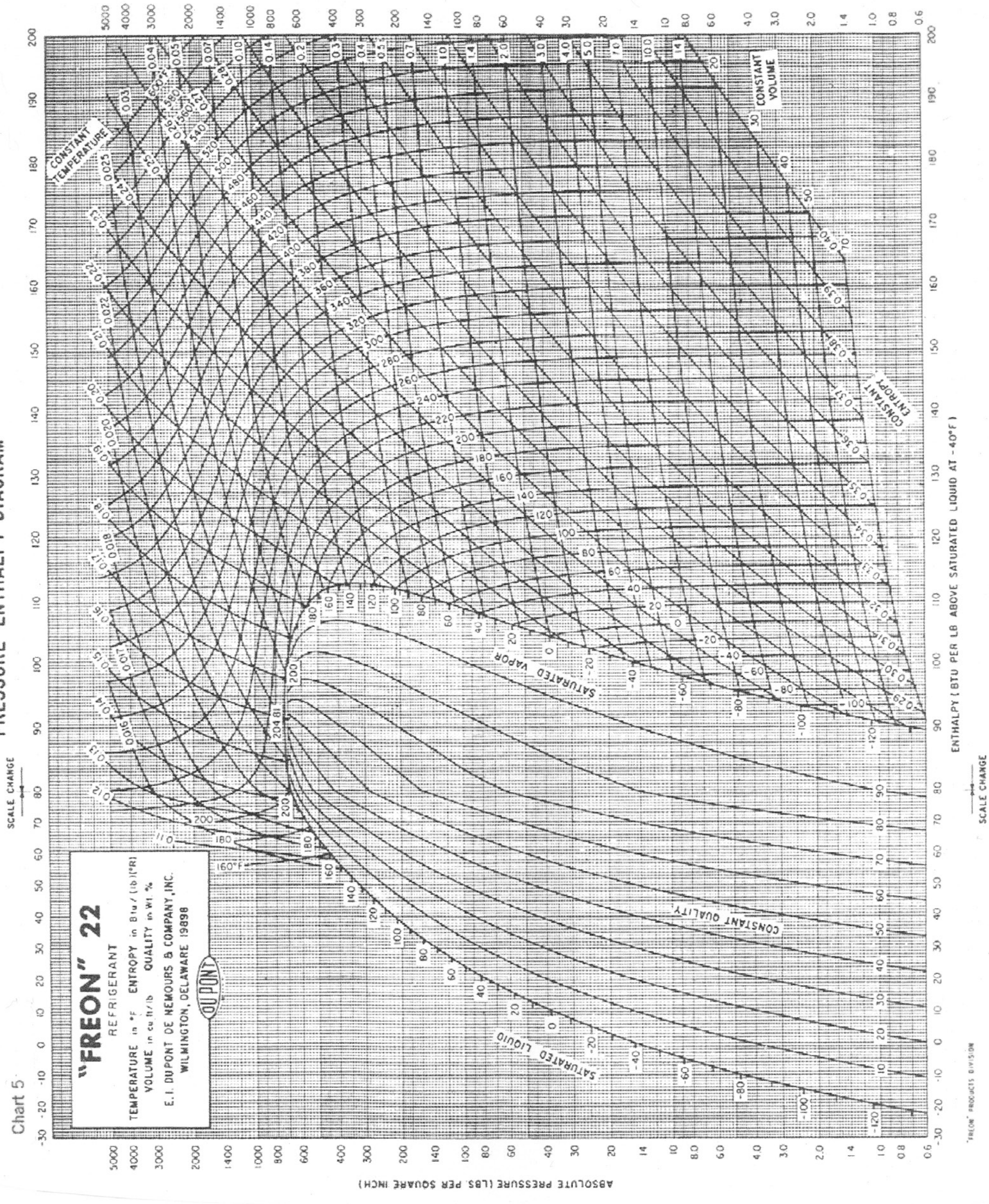
Refrige rant	Replace ment	HFC	HCFC	High Temp	Med Temp	Low Temp	New	Retro-fit	
R-11	R-123		X	X	X		X	X	Centrifugal Water Chillers
R-12	R-134A	X		X	X		X	X	Auto A/C, Commercial Refrigeration, Appliances, Water Chillers
	R-423A	X		X	X			X	Centrifugal Water Chillers
R-22	R-410A	X		X	X		X		Commercial/ Residential AC and Heat Pumps
	R-407C	X		X			X	X	Commercial Refrigeration, Commercial/ Residential AC
	R-422D		X		X	X		X	Commercial/ Industrial Refrigeration
R-409A	R-134A	X		X	X		X	X	Commercial Refrigeration
R-502	R-404A	X			X	X	X		Commercial/ Industrial Refrigeration
	R-507	X			X	X	X		Commercial/ Industrial Refrigeration
	R-422A	X			X	X		X	Commercial/ Industrial Refrigeration

REFRIGERANT PRESSURE-ENTHALPY (MOLLIER) DIAGRAMS

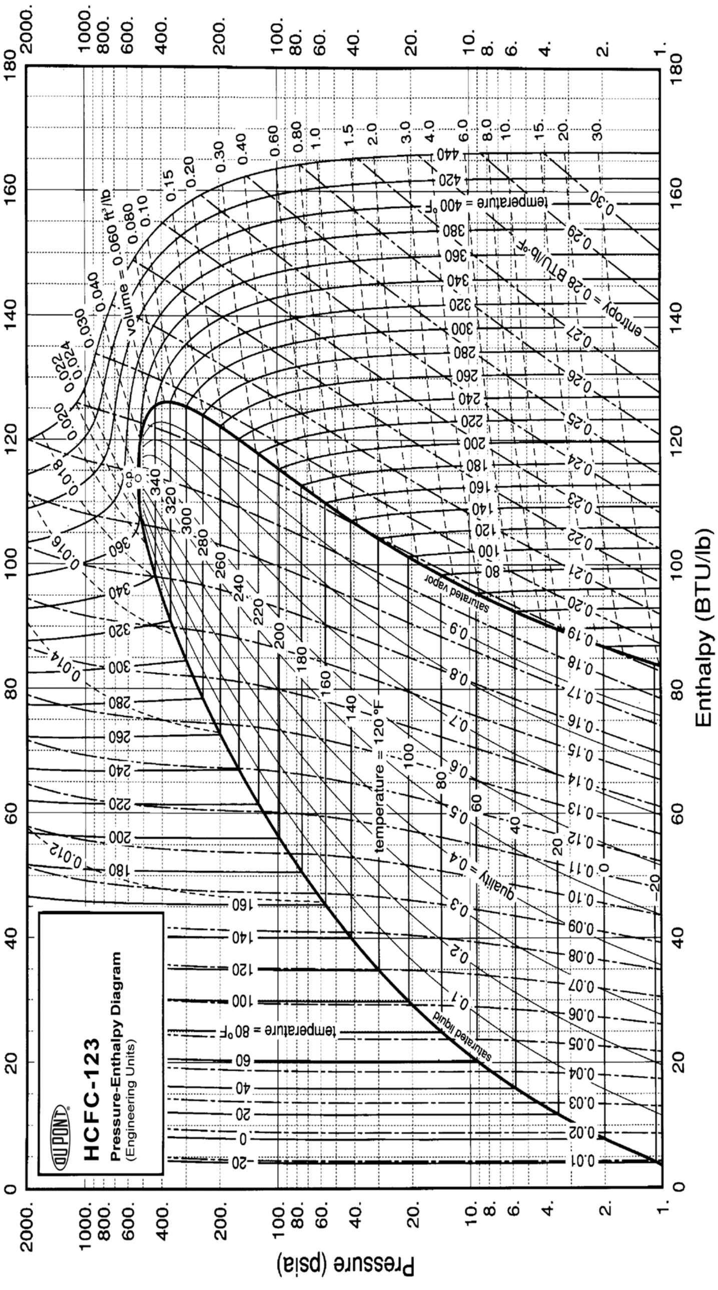
Source (except R-717): DuPont Refrigerants

See also **Chapter 11—Refrigeration Cycle** for generic Mollier Diagram.

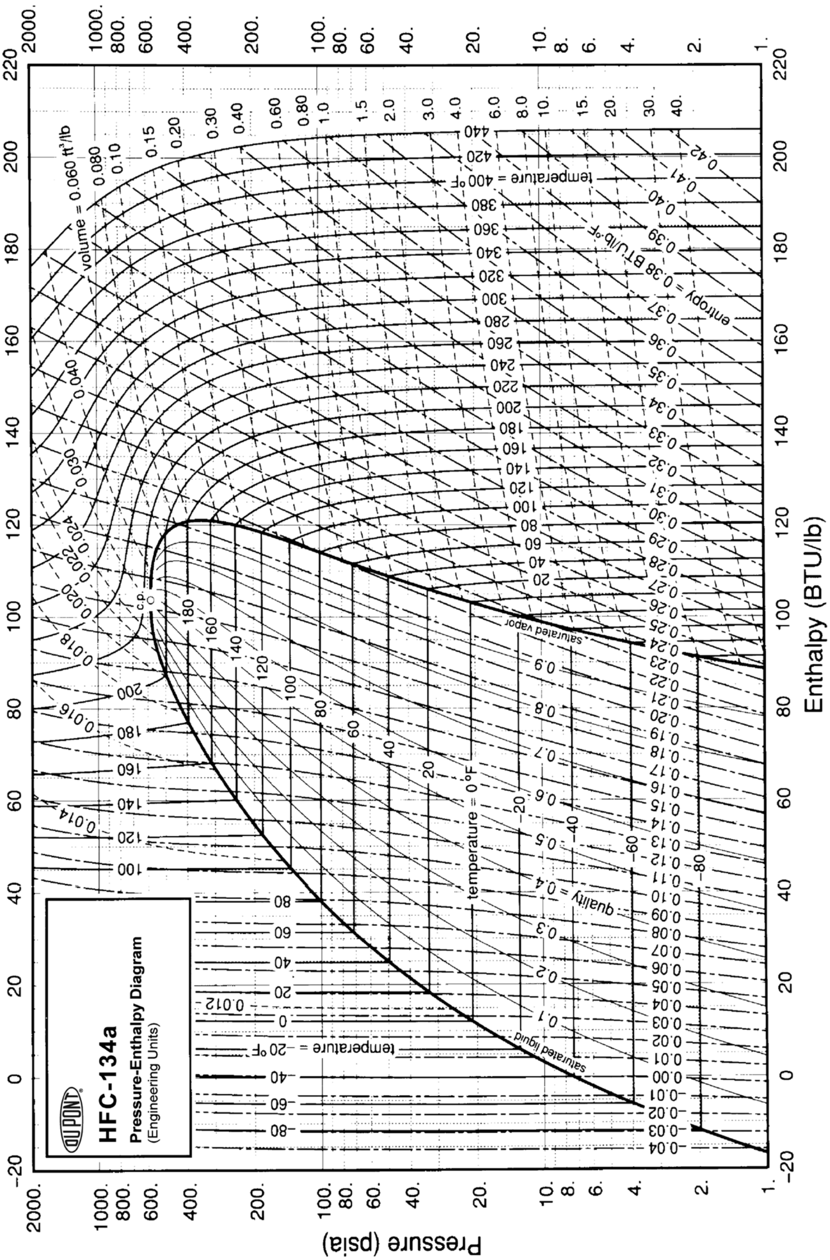
R-22



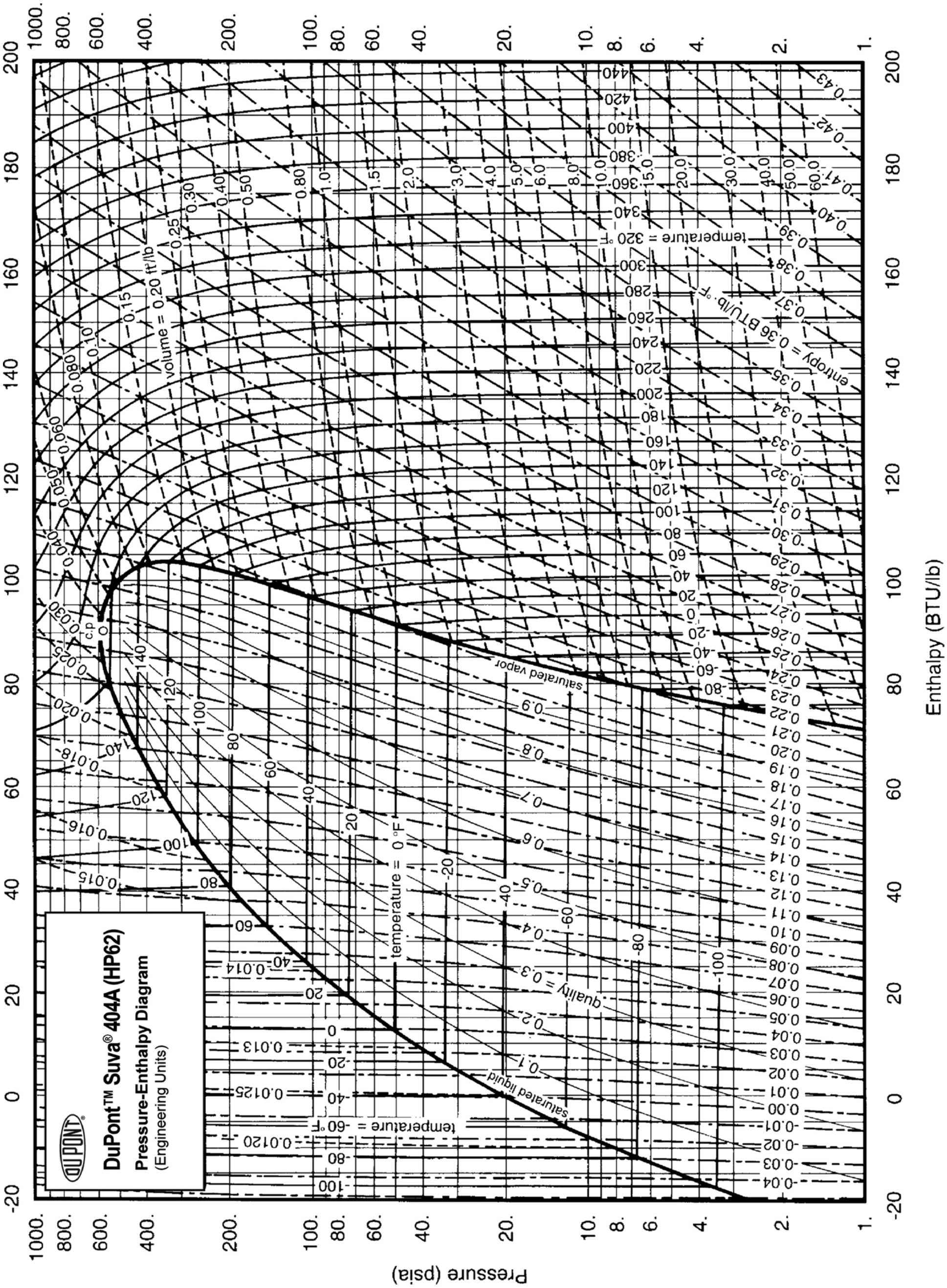
R-123


HCFC-123
Pressure-Enthalpy Diagram
(Engineering Units)
Pressure (psia)
Enthalpy (BTU/lb)
saturated liquid
saturated vapor
temperature = 80 °F
temperature = 120 °F
temperature = 400 °F
quality = 0.4
entropy = 0.28 BTU/lb·°F
volume = 0.060 ft³/lb

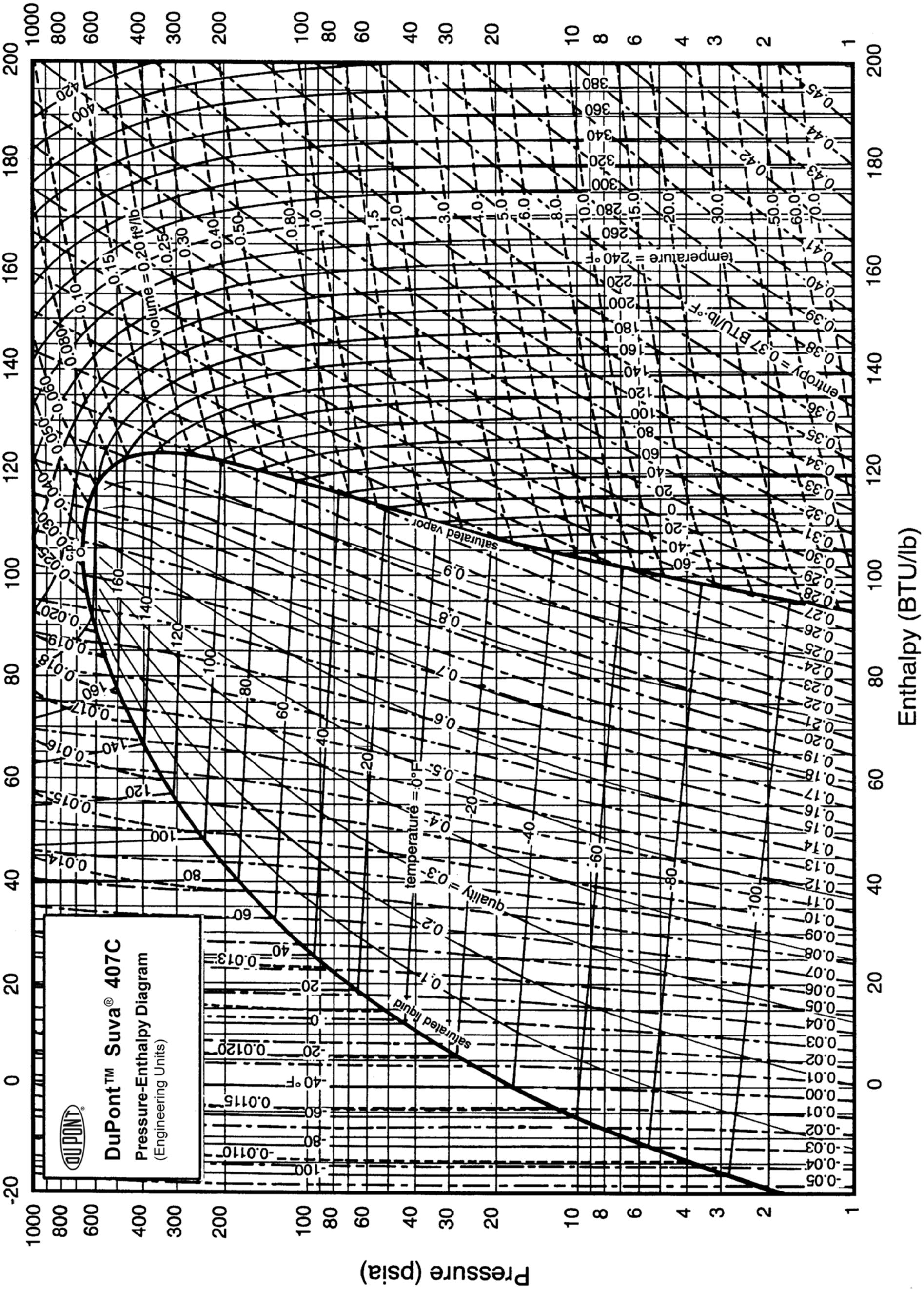
R-134A


HFC-134a
Pressure-Enthalpy Diagram
(Engineering Units)
Enthalpy (BTU/lb)
Pressure (psia)
saturated liquid
saturated vapor
c.p.
volume = 0.060 ft³/lb
entropy = 0.38 BTU/lb-°F
temperature = 400°F
temperature = 0°F
temperature = −20°F
quality = 0.4

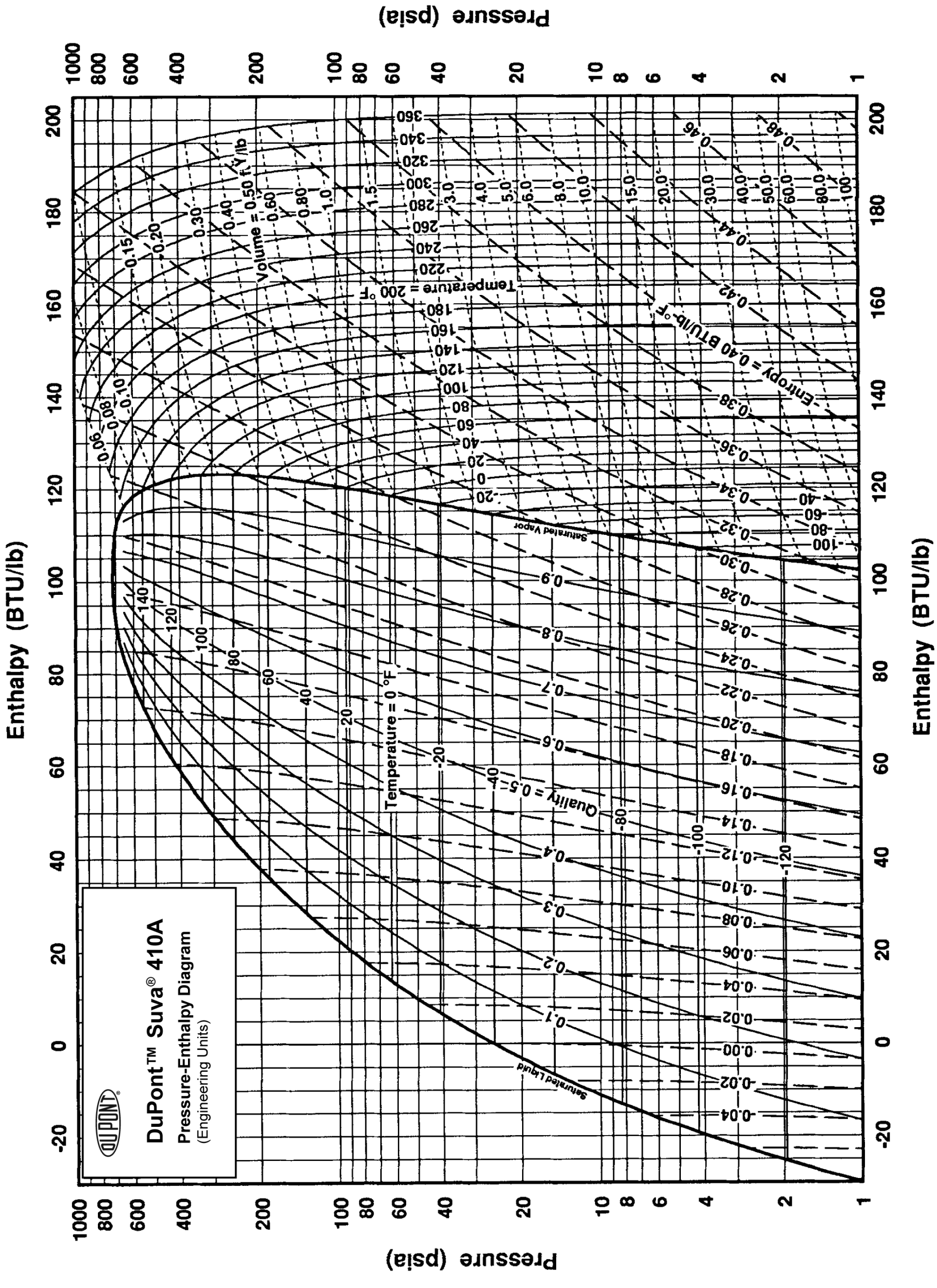
R-404A


DuPont™ Suva® 404A (HP62)
Pressure-Enthalpy Diagram
(Engineering Units)
Enthalpy (BTU/lb)
Pressure (psia)
saturated liquid
saturated vapor
c.p.
temperature = 0 °F
temperature = 320 °F
temperature = -60 °F
volume = 0.20 ft³/lb
entropy = 0.36 BTU/lb · °F
quality = 0.3

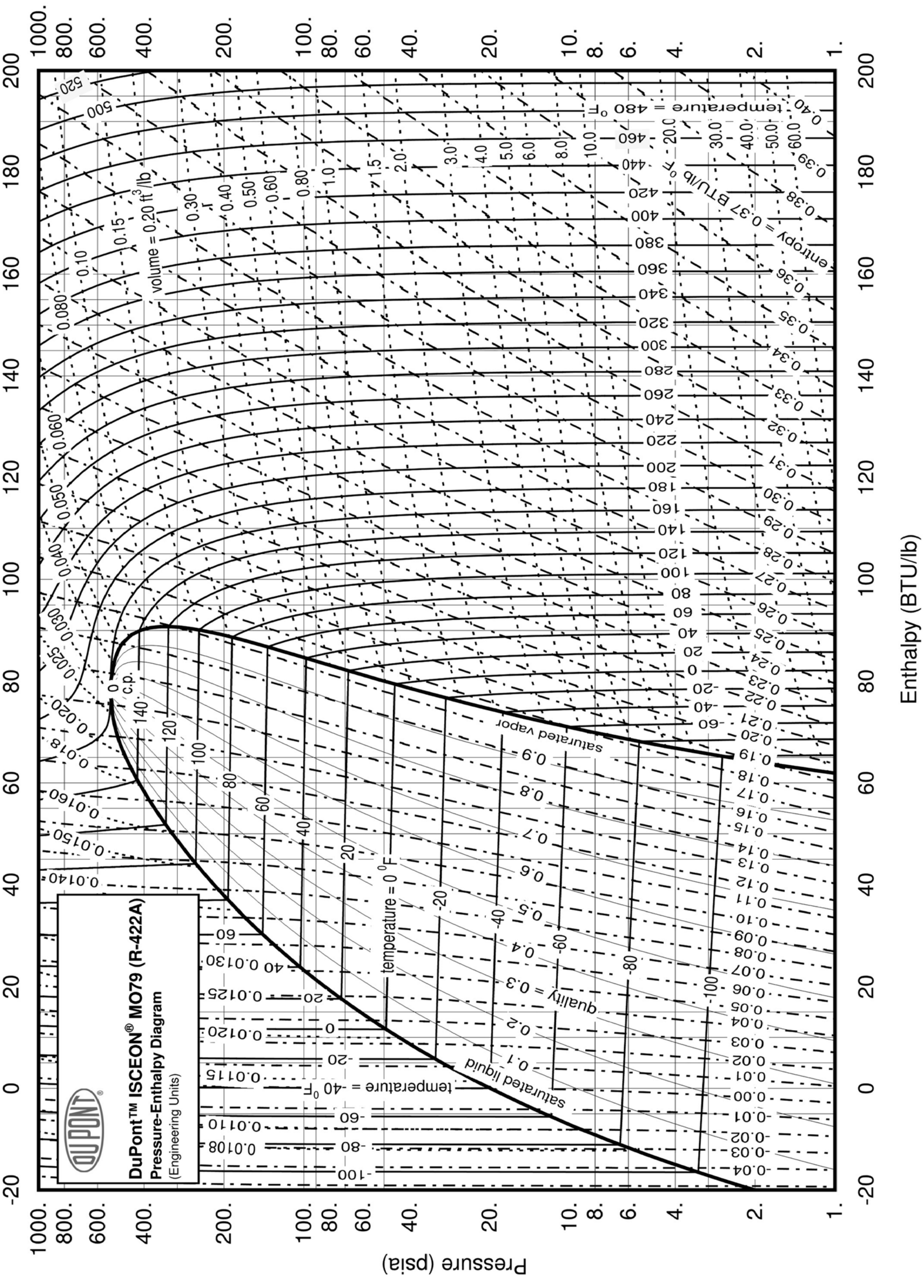
R-407C


DuPont™ Suva® 407C
Pressure-Enthalpy Diagram
(Engineering Units)
Enthalpy (BTU/lb)
Pressure (psia)
saturated liquid
saturated vapor

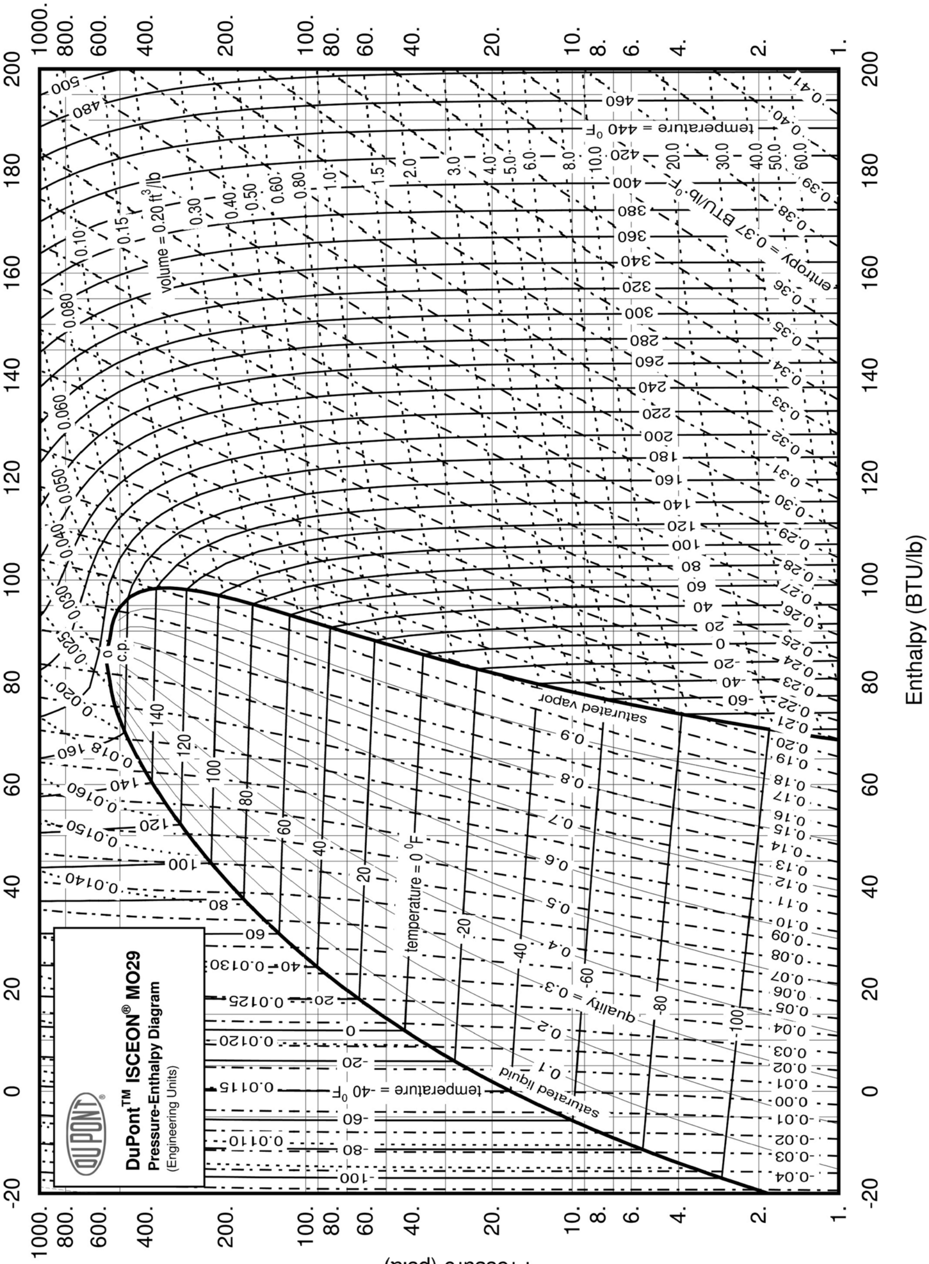
R-410A


Pressure (psia)
Enthalpy (BTU/lb)
DuPont™ Suva® 410A
Pressure-Enthalpy Diagram
(Engineering Units)
Temperature = 200 °F
Temperature = 0 °F
Volume = 0.50 ft³/lb
Entropy = 0.40 BTU/lb·°F
Quality = 0.5
Saturated Vapor
Saturated Liquid

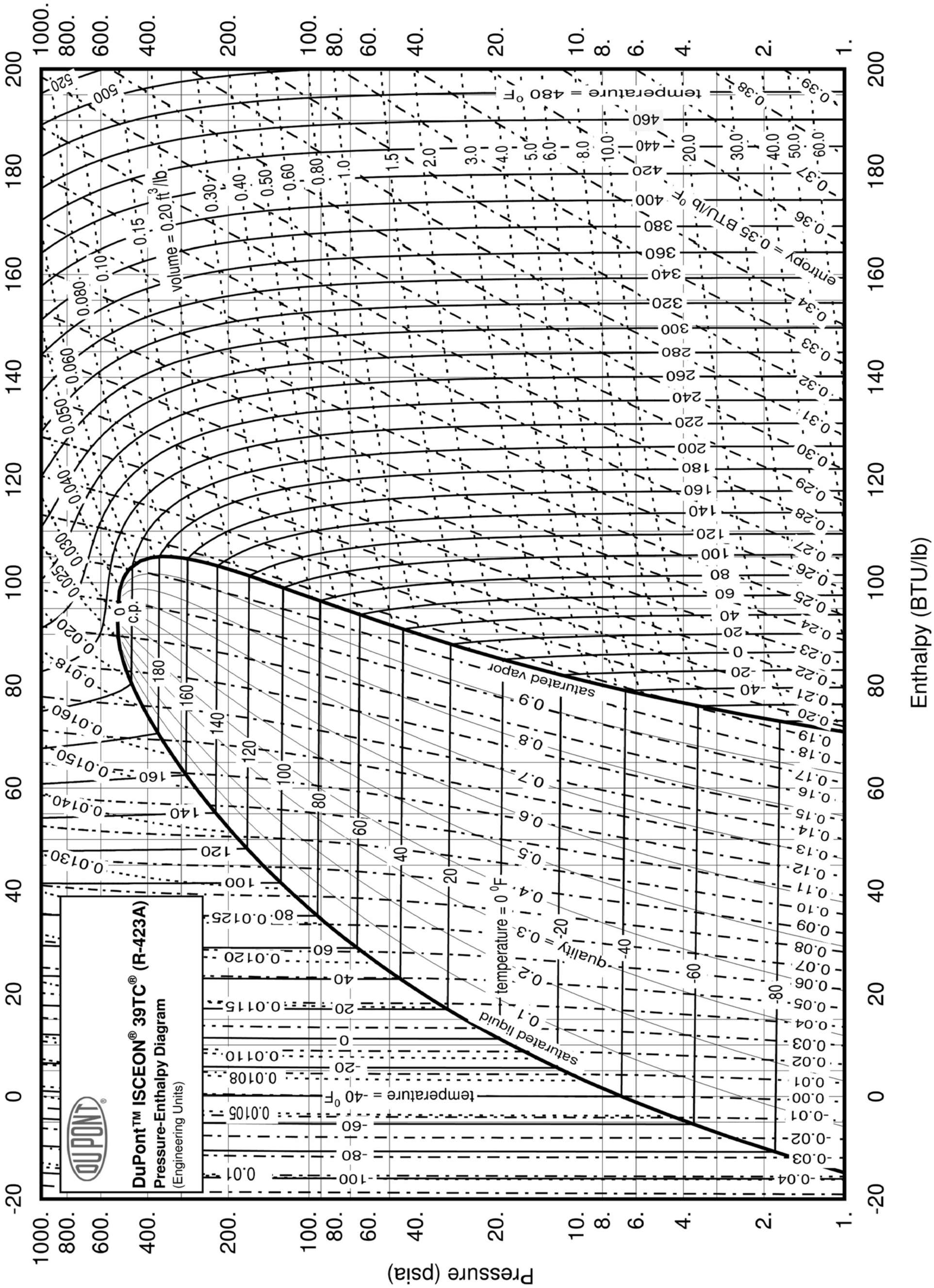
R-422A


DuPont™ ISCEON® MO79 (R-422A)
Pressure-Enthalpy Diagram
(Engineering Units)
Enthalpy (BTU/lb)
Pressure (psia)
saturated liquid
saturated vapor
c.p.

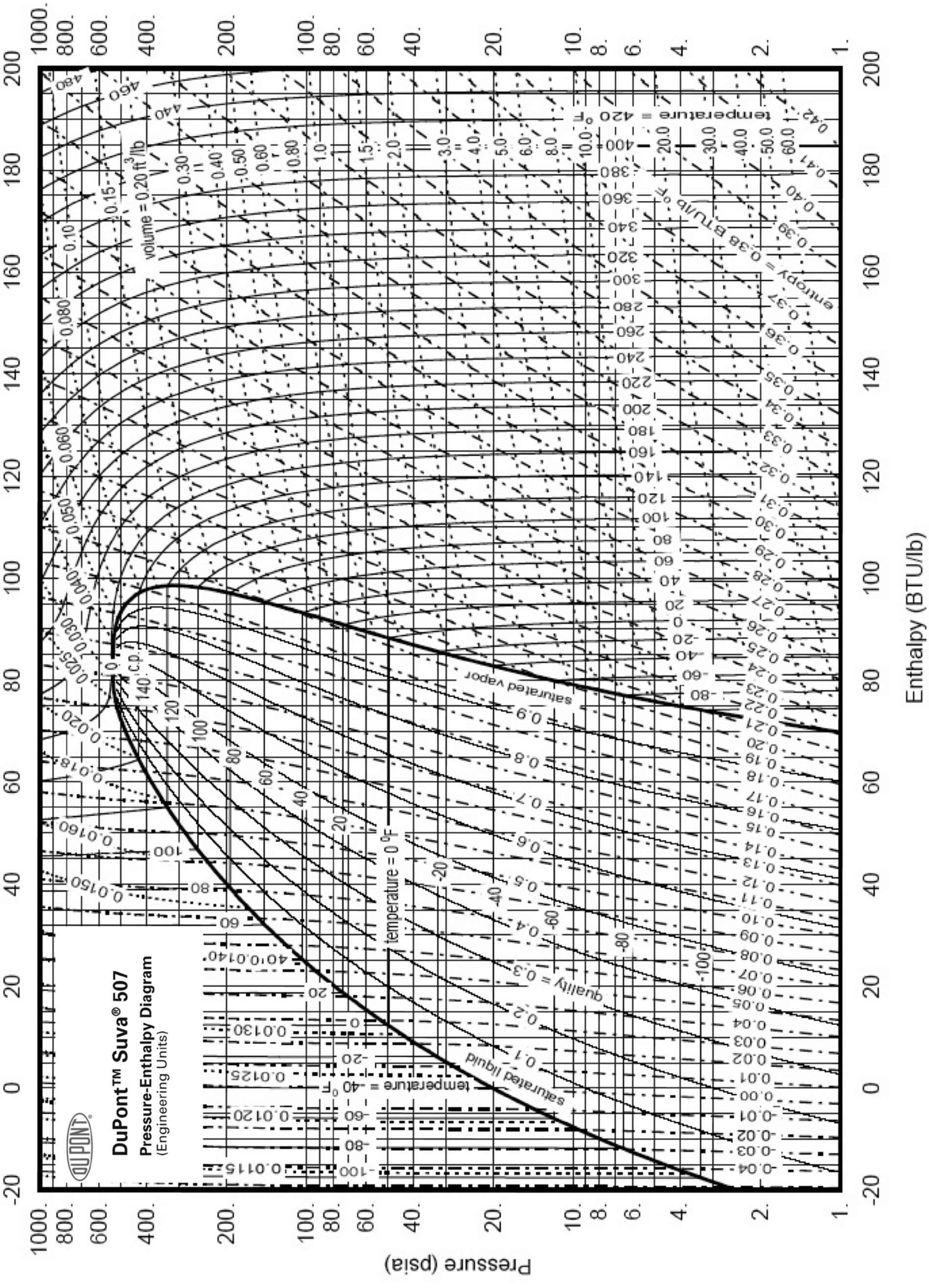
R-422D


DuPont™ ISCEON® MO29
Pressure-Enthalpy Diagram
(Engineering Units)
Enthalpy (BTU/lb)
Pressure (psia)
saturated liquid
saturated vapor
c.p.

R-423A

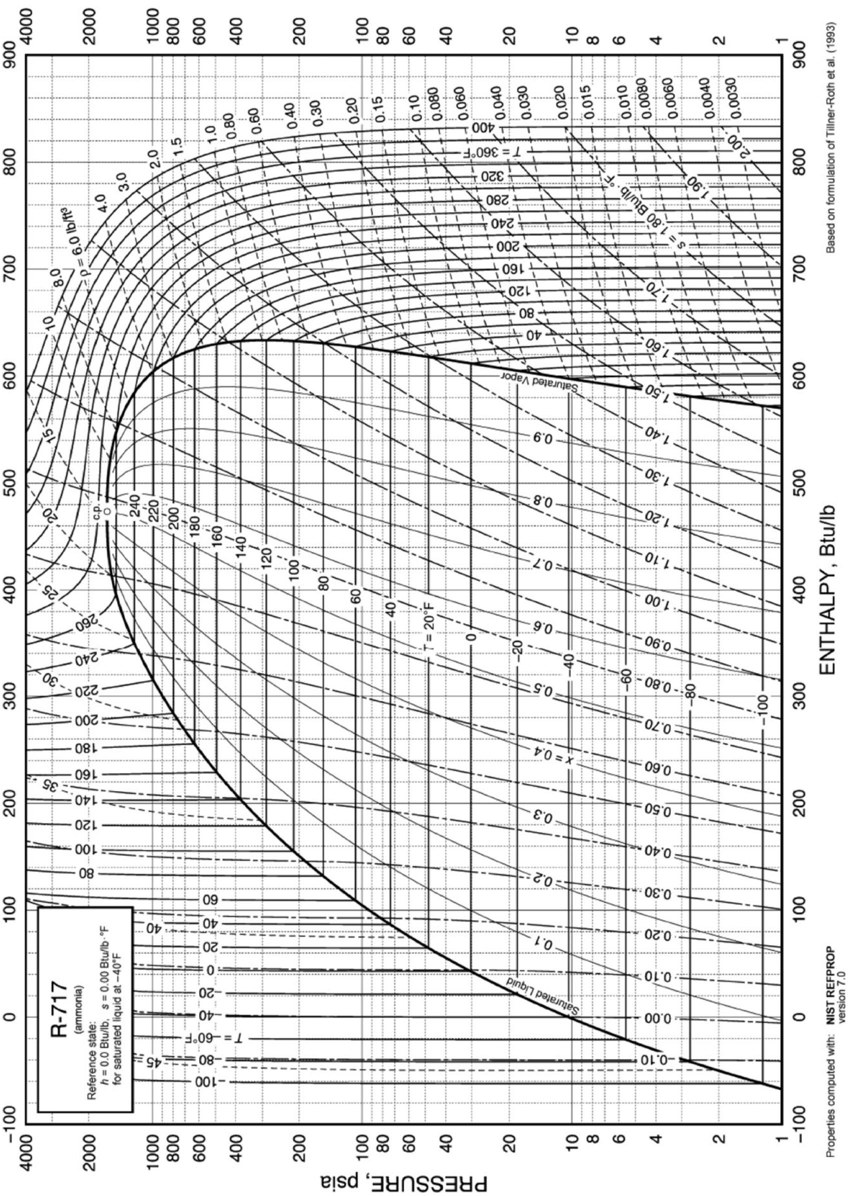

DuPont™ ISCEON® 39TC® (R-423A)
Pressure-Enthalpy Diagram
(Engineering Units)
Enthalpy (BTU/lb)
Pressure (psia)
saturated liquid
saturated vapor
c.p.

R-507


DuPont™ Suva® 507
Pressure-Enthalpy Diagram
(Engineering Units)
Enthalpy (BTU/lb)
Pressure (psia)

R-717 (Ammonia)

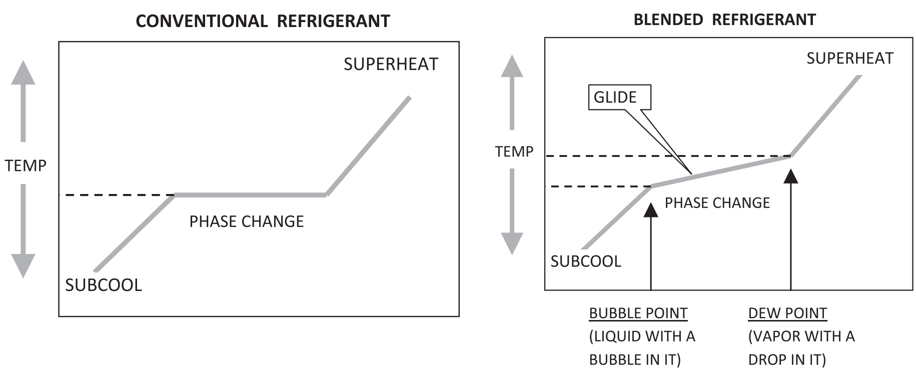
Source: ASHRAE Fundamentals Handbook, 2005, © American Society of Heating, Refrigerating and Air-Conditioning



BLENDED REFRIGERANTS

Properties of refrigerant blends are different than the traditional refrigerants. Blends shift in composition as they are boiling or condensing, e.g. one of the constituent parts will evaporate or condense faster than the other, known as fractionation. The changing composition causes the boiling point to shift as well, giving rise to a phenomena for blended refrigerants called "glide." For blended refrigerants, the pressure/temperature tables include "bubble point" and "dew point" values which demark the beginning and ending of the glide range. Rather than a single saturation temperature, there is a range.

For blends with a glide region, the proportion of one part of the refrigerant vs. another will vary in during phase changes. This can create operational issues if, for example, a leak occurs in a condensing or evaporating heat exchanger—more of one refrigerant could escape, thus changing the mixture of the remaining refrigerant. As the proportions change, the refrigerant properties change and it may be necessary to replace the full refrigerant charge in some cases to restore proper operation. Blends that exhibit no glide are called azeotropic (distillation occurs equally for the individual ingredients), while those with glide are zeotropic blends.



PRESSURE-TEMPERATURE CHARTS FOR REFRIGERANTS

Source: National Refrigerants
Figures inside dotted lines are in. hg. vacuum

Pressure-Temperature Charts for Refrigerants

Source: National Refrigerants
Figures inside dotted lines are in. hg. vacuum

Temperature °F	°C	R409A Liquid Press/	R409A Vapor Press.	R414B Liquid Press/	R414B Vapor Press.	R416A Liquid Press/	R416A Vapor Press.
-30	-34.4	0.2	9.9	0.0	9.7	12.1	13.4
-25	-31.7	1.8	7	1.9	6.8	9.6	11.0
-20	-28.9	3.9	3.8	4.0	3.6	6.7	8.3
-15	-26.1	6.2	0.3	6.3	0.0	3.5	5.3
-10	-23.3	8.7	1.7	8.8	2.0	0.0	2.0
-5	-20.6	11.4	3.8	11.5	4.1	1.9	0.8
0	-17.8	14.4	6.1	14.5	6.5	4.0	2.8
5	-15.0	17.6	8.6	17.7	9.0	6.3	5.0
10	-12.2	21.1	11.4	21.2	11.9	8.9	7.4
15	-9.4	24.9	14.4	25.0	14.9	11.6	10.0
20	-6.7	29.0	17.6	29.0	18.3	14.6	12.8
25	-3.9	33.4	21.2	33.4	21.9	17.8	15.9
30	-1.1	38.1	25.0	38.1	25.8	21.4	19.3
35	1.7	43.2	29.2	43.1	30.0	25.2	22.9
40	4.4	48.6	33.6	48.5	34.6	29.3	26.8
45	7.2	54.4	38.5	54.3	39.5	33.7	31.1
50	10.0	60.6	43.6	60.4	44.8	38.4	35.6
55	12.8	67.2	49.2	67.0	50.4	43.5	40.5
60	15.6	74.2	55.2	73.9	56.5	49.0	45.7
65	18.3	81.7	61.5	81.3	62.9	54.8	51.3
70	21.1	89.6	68.4	89.1	69.8	61.1	57.3
75	23.9	98.0	75.6	97.4	77.1	67.7	63.7
80	26.7	107	83.4	106	85.0	74.8	70.6
85	29.4	116	91.6	116	93.3	82.3	77.8
90	32.2	126	100	125	102	90.3	85.5
95	35.0	137	110	136	111	98.8	93.7
100	37.8	148	120	146	121	108	102
105	40.6	159	130	158	132	117	112
110	43.3	172	141	170	143	127	121
115	46.1	184	153	183	155	138	132
120	48.9	198	165	196	167	149	143
125	51.7	212	178	210	180	161	154
130	54.4	227	192	224	193	173	166
135	57.2	242	207	239	208	186	179
140	60.0	258	222	255	223	200	192
145	62.8			272	239	214	206
150	65.6			289	255	229	221

Temperature °F	°C	R404A	R507	R502	R402A	R402B	R408A
-40	-40.0	4.3	5.5	4.1	6.3	3.6	2.8
-35	-37.2	6.8	8.2	6.5	9.1	6.0	5.1
-30	-34.4	9.5	11.1	9.2	12.1	9.0	7.6
-25	-31.7	12.5	14.3	12.1	15.4	12.0	10.4
-20	-28.9	15.7	17.8	15.3	18.9	15.4	13.5
-15	-26.1	19.3	21.7	18.8	22.9	18.6	16.8
-10	-23.3	23.2	25.8	22.6	27.1	22.6	20.4
-5	-20.6	27.5	30.3	26.7	31.7	27.0	24.4
0	-17.8	32.1	35.2	31.1	36.7	31.0	28.7
5	-15.0	37.0	40.5	35.9	42.1	36.0	33.3
10	-12.2	42.4	46.1	41.0	48.0	42.0	38.3
15	-9.4	48.2	52.2	46.5	54.2	47.0	43.7
20	-6.7	54.5	58.8	52.4	60.9	54.0	49.5
25	-3.9	61.2	65.8	58.8	68.1	60.0	55.8
30	-1.1	68.4	73.3	65.6	75.8	67.0	62.5
35	1.7	76.1	81.3	72.8	84.0	75.0	69.7
40	4.4	84.4	89.8	80.5	92.8	83.4	77.4
45	7.2	93.2	98.9	88.7	102	91.6	85.6
50	10.0	103	109	97.4	112	100	84.3
55	12.8	113	119	107	123	110	104
60	15.6	123	130	116	134	120	114
65	18.3	135	141	127	146	133	124
70	21.1	147	154	138	158	143	135
75	23.9	159	167	149	171	155	147
80	26.7	173	180	161	185	170	159
85	29.4	187	195	174	200	183	173
90	32.2	202	210	187	215	198	186
95	35.0	218	226	201	232	213	201
100	37.8	234	244	216	249	230	217
105	40.6	252	262	232	267	247	233
110	43.3	270	281	248	286	262	250
115	46.1	289	301	265	305	283	268
120	48.9	310	322	283	326	303	287
125	51.7	331	344	301	347	323	307
130	54.4	353	368	321	370	345	327
135	57.2	377	393	341	393	-	349
140	60.0	401	419	363	418	-	372
145	62.8	426	446	-	443	-	-
150	65.6	453	475	-	470	-	-

Temperature °F	°C	R422A Liquid	R422A Vapor	R422B Liquid	R422B Vapor	R422D Liquid	R422D Vapor
-40	-40.0	5.2	3.2	0.9	2.7	2.4	1.2
-35	-37.2	7.8	5.6	3.0	0.9	4.6	0.8
-30	-34.4	10.7	8.3	5.4	1.1	7.1	3.0
-25	-31.7	13.9	11.3	7.9	3.2	9.9	5.4
-20	-28.9	17.3	14.6	10.7	5.7	12.9	8.1
-15	-26.1	21.1	18.2	13.8	8.3	16.2	11.0
-10	-23.3	25.2	22.1	17.1	11.3	19.8	14.3
-5	-20.6	29.6	26.3	20.7	14.5	23.7	17.8
0	-17.8	34.4	30.9	24.7	18.0	27.9	21.7
5	-15.0	39.6	35.6	29.0	21.9	32.5	25.8
10	-12.2	45.2	41.4	33.6	26.1	37.5	30.4
15	-9.4	51.3	47.2	38.6	30.6	42.8	35.3
20	-6.7	57.8	53.5	43.9	35.5	48.5	40.7
25	-3.9	64.7	60.2	49.7	40.8	54.7	46.4
30	-1.1	72.2	67.5	55.9	46.6	61.3	52.6
35	1.7	80.1	75.2	62.5	52.7	68.4	59.3
40	4.4	88.6	83.5	69.6	59.4	75.9	66.4
45	7.2	97.6	92.3	77.2	66.5	84.0	74.0
50	10.0	107	102	85.3	74.1	92.6	82.2
55	12.8	117	112	93.9	82.2	102	90.9
60	15.6	128	122	103	90.9	111	100
65	18.3	140	134	113	100	122	110
70	21.1	152	146	123	110	133	121
75	23.9	165	158	134	120	144	132
80	26.7	179	172	145	132	156	144
85	29.4	193	186	158	143	169	156
90	32.2	208	201	170	156	183	170
95	35.0	224	217	184	169	197	184
100	37.8	241	234	198	183	212	198
105	40.6	258	251	213	198	228	214
110	43.3	277	270	229	213	245	231
115	46.1	296	289	246	230	262	248
120	48.9	317	310	263	247	281	266
125	51.7	338	331	281	265	300	286
130	54.4	361	354	301	284	320	306
135	57.2	385	378	321	304	341	327
140	60.0	410	403	342	326	364	350
145	62.8	-	-	364	348	387	373
150	65.6	-	-	387	371	411	398

Temperature °F	°C	R11	R113	R114	R123	R124	R500
-20	-28.9	27.0	29.1	22.9	27.8	16.1	3.2
-15	-26.1	26.5	28.9	21.8	27.4	14.1	5.4
-10	-23.3	26.0	28.7	20.6	26.9	12.0	7.8
-5	-20.6	25.4	28.5	19.3	26.4	9.6	10.4
0	-17.8	24.7	28.2	17.8	25.9	6.9	13.3
5	-15.0	23.9	27.9	16.2	25.2	3.9	16.4
10	-12.2	23.1	27.6	14.4	24.5	0.6	19.7
15	-9.4	22.1	27.2	12.4	23.8	1.6	23.4
20	-6.7	21.1	26.8	10.2	22.8	3.5	27.3
25	-3.9	19.9	26.3	7.8	21.8	5.7	31.5
30	-1.1	18.6	25.8	5.2	20.7	8.1	36.0
35	1.7	17.2	25.2	2.3	19.5	10.5	40.9
40	4.4	15.6	24.5	0.4	18.1	13.2	46.1
45	7.2	13.9	23.8	2.0	16.6	16.1	51.6
50	10.0	12.0	22.9	3.8	14.9	19.2	57.6
55	12.8	10.0	22.2	5.8	13.0	22.6	63.9
60	15.6	7.8	21.0	7.9	11.2	26.3	70.6
65	18.3	5.4	19.9	10.1	8.9	30.2	77.8
70	21.1	2.8	18.7	12.6	6.5	34.4	85.4
75	23.9	0.0	17.3	15.2	4.1	38.9	93.5
80	26.7	1.5	15.9	18.0	1.2	43.7	102
85	29.4	3.2	14.3	20.9	0.9	48.8	111
90	32.2	4.9	12.5	24.1	2.5	54.2	121
95	35.0	6.8	10.6	27.5	4.3	60.0	131
100	37.8	8.8	8.6	31.2	6.1	66.1	141
105	40.6	10.9	6.4	35.0	8.1	72.6	152
110	43.3	13.2	4.0	39.1	10.3	79.5	164
115	46.1	15.6	1.4	43.4	12.6	86.8	177
120	48.9	18.2	0.7	48.0	15.1	94.5	189
125	51.7	21.0	2.2	52.8	17.8	103	203
130	54.4	24.0	3.7	58.0	20.6	111	217
135	57.2	27.1	5.4	63.4	23.6	120	232
140	60.0	30.4	7.2	69.1	26.8	130	248
145	62.8	34.0	9.2	75.1	30.2	140	-
150	65.6	37.7	11.2	81.4	33.9	150	-

Temperature °F	°C	R13	R23	R503	R508B	R403B
-120	-84.4	4.5	4.0	3.1	3.1	25.4
-115	-81.7	0.3	0.3	6.0	6.0	24.5
-110	-78.9	2.1	2.9	9.3	9.3	23.4
-105	-76.1	4.7	5.8	12.9	12.9	22.2
-100	-73.3	7.6	9.0	16.9	16.9	20.9
-95	-70.6	10.8	12.7	21.4	21.4	19.3
-90	-67.8	14.3	16.7	26.3	26.4	17.5
-85	-65.0	18.2	21.3	31.8	31.8	15.5
-80	-62.2	22.5	26.3	37.7	37.8	13.2
-75	-59.4	27.2	31.8	44.2	44.4	10.6
-70	-56.7	32.3	37.9	51.3	51.5	7.8
-65	-53.9	37.8	44.6	59.0	59.3	4.6
-60	-51.1	43.9	52.0	67.3	67.8	1.0
-55	-48.3	50.4	60.0	76.4	76.9	1.5
-50	-45.6	57.5	68.7	86.1	86.8	3.6
-45	-42.8	65.1	78.1	96.6	97.5	6.0
-40	-40.0	73.3	88.3	108	109	8.6
-35	-37.2	82.1	99.4	120	121	11.4
-30	-34.4	91.6	111	133	135	14.5
-25	-31.7	102	124	147	149	17.9
-20	-28.9	113	138	161	164	21.6

Temperature °F	°C	R22	R407A Liquid Press/	R407A Vapor Press.	R407C Liquid Press/	R407C Vapor Press.	R410A
-40	-40.0	0.5	3.9	1.0	3.0	4.4	11.6
-35	-37.2	2.6	6.4	1.0	5.4	0.6	14.9
-30	-34.4	4.9	9.2	3.3	8.0	1.8	18.5
-25	-31.7	7.4	12.2	5.8	10.9	4.1	22.5
-20	-28.9	10.1	15.6	8.5	14.1	6.6	26.9
-15	-26.1	13.2	19.2	11.5	17.6	9.4	31.7
-10	-23.3	16.5	23.2	14.9	21.3	12.5	36.8
-5	-20.6	20.1	27.5	18.5	25.4	15.9	42.5
0	-17.8	24.0	32.2	22.5	29.9	19.6	48.6
5	-15.0	28.2	37.3	26.9	34.7	23.6	55.2
10	-12.2	32.8	42.8	31.6	39.9	28.0	62.3
15	-9.4	37.7	48.7	36.7	45.6	32.8	70.0
20	-6.7	43.0	55.1	42.3	51.6	38.0	78.3
25	-3.9	48.8	62.0	48.3	58.2	43.6	87.3
30	-1.1	54.9	69.3	54.8	65.2	49.6	96.8
35	1.7	61.5	77.2	61.8	72.6	56.1	107
40	4.4	68.5	85.6	69.4	80.7	63.1	118
45	7.2	76.0	94.6	77.4	89.2	70.6	130
50	10.0	84.0	104	86.1	98.3	78.7	142
55	12.8	92.6	114	95.3	108	87.3	155
60	15.6	102	125	105	118	96.8	170
65	18.3	111	137	116	129	106	185
70	21.1	121	149	127	141	117	201
75	23.9	132	162	139	153	128	217
80	26.7	144	175	152	166	140	235
85	29.4	156	190	165	180	153	254
90	32.2	168	205	179	195	166	274
95	35.0	182	221	194	210	181	295
100	37.8	196	238	210	226	196	317
105	40.6	211	255	227	243	211	340
110	43.3	226	274	245	261	229	365
115	46.1	243	293	264	280	247	391
120	48.9	260	314	284	300	266	418
125	51.7	278	335	305	321	286	446
130	54.4	297	358	327	342	307	476
135	57.2	317	382	350	365	329	507
140	60.0	337	406	375	389	353	539
145	62.8	359	432	401	-	-	573
150	65.6	382	459	428	-	-	608

Temperature °F	Temperature °C	R12	R134a	R401A Liquid Press/	R401A Vapor Press.	R401B Liquid Press/	R401B Vapor Press.
-40	-40.0	11.0	14.8	8.1	13.2	6.5	11.8
-35	-37.2	8.4	12.5	5.1	10.7	3.3	9.1
-30	-34.4	5.5	9.9	1.7	7.9	0.2	6.1
-25	-31.7	2.3	6.9	1.0	4.8	2.1	2.8
-20	-28.9	0.6	3.7	3.0	1.4	4.3	0.5
-15	-26.1	2.4	0.6	5.2	1.2	6.6	2.5
-10	-23.3	4.5	1.9	7.7	3.3	9.2	4.7
-5	-20.6	6.7	4.0	10.3	5.5	12.0	7.1
0	-17.8	9.2	6.5	13.2	8.0	15.1	9.7
5	-15.0	11.8	9.1	16.3	10.7	18.4	12.6
10	-12.2	14.6	11.9	19.7	13.7	22.0	15.8
15	-9.4	17.7	15.0	23.4	16.9	25.9	19.2
20	-6.7	21.0	18.4	27.4	20.4	30.1	23.0
25	-3.9	24.6	22.1	31.7	24.2	34.6	27.0
30	-1.1	28.5	26.1	36.4	28.3	39.5	31.4
35	1.7	32.6	30.4	41.3	32.8	44.8	36.1
40	4.4	37.0	35.0	46.6	37.6	50.4	41.1
45	7.2	41.7	40.1	52.4	42.7	56.4	46.6
50	10.0	46.7	45.5	58.5	48.2	62.8	52.4
55	12.8	52.0	51.3	65.0	54.1	69.6	58.7
60	15.6	57.7	57.5	71.9	60.4	76.9	65.4
65	18.3	63.8	64.1	79.3	67.2	84.7	72.5
70	21.1	70.2	71.2	87.1	74.4	92.9	80.1
75	23.9	77.0	78.8	95.4	82.1	102	88.2
80	26.7	84.2	86.8	104	90.2	111	96.8
85	29.4	91.8	95.4	114	98.9	121	106
90	32.2	99.8	104	123	108	131	116
95	35.0	108	114	134	118	142	126
100	37.8	117	124	145	128	153	137
105	40.6	127	135	156	139	166	148
110	43.3	136	147	169	151	178	160
115	46.1	147	159	181	163	192	173
120	48.9	158	171	195	176	206	187
125	51.7	169	185	209	189	220	201
130	54.4	181	199	224	203	236	216
135	57.2	194	214	239	218	252	231
140	60.0	207	229	255	234	269	248
145	62.8	220	246	272	250	287	265
150	65.6	234	263	290	267	305	283

REFRIGERANT 717—AMMONIA

Source: Extol of Ohio, Inc.

Values in Bold Italic are in. hg.

TEMP.	PRESS	TEMP.	PRESS	TEMP.	PRESS	TEMP.	PRESS
°F	PSIG	°F	PSIG	°F	PSIG	°F	PSIG
-60	***18.6***	-14	6.7	31	46.3	76	128.3
-58	***17.8***	-13	7.3	32	47.6	77	130.7
-57	***17.4***	-12	7.9	33	48.9	78	133.2
-56	***17***	-11	8.4	34	50.2	79	135.8
-55	***16.6***	-10	9	35	51.6	80	138.3
-54	***16.2***	-9	9.6	36	52.9	81	140.9
-53	***15.7***	-8	10.3	37	54.3	82	143.6
-52	***15.3***	-7	10.9	38	55.7	83	146.3
-51	***14.8***	-6	11.6	39	57.2	84	149
-50	***14.3***	-5	12.2	40	58.6	85	151.7
-49	***13.8***	-4	12.9	41	60.1	86	154.5
-48	***13.3***	-3	13.6	42	61.6	87	157.3
-47	***12.8***	-2	14.3	43	63.1	88	160.1
-46	***12.2***	-1	15	44	64.7	89	163
-45	***11.7***	0	15.7	45	66.3	90	165.9
-44	***11.1***	1	16.5	46	67.8	91	168.9
-43	***10.6***	2	17.2	47	69.5	92	171.9
-42	***10***	3	18	48	71.1	93	174.9
-41	***9.3***	4	18.8	49	72.8	94	178
-40	***8.7***	5	19.6	50	74.5	95	181.1
-39	***8.1***	6	20.4	51	76.2	96	184.2
-38	***7.4***	7	21.2	52	78	97	187.4
-37	***6.8***	8	22.1	53	79.7	98	190.6
-36	***6.1***	9	22.9	54	81.5	99	193.9
-35	***5.4***	10	23.8	55	83.4	100	197.2
-34	***4.7***	11	24.7	56	85.2	101	200.5
-33	***3.9***	12	25.6	57	87.1	102	203.9
-32	***3.2***	13	26.5	58	89	103	207.3
-31	***2.4***	14	27.5	59	90.9	104	210.7
-30	***1.6***	15	28.4	60	92.9	105	214.2
-29	***0.8***	16	29.4	61	94.9	106	217.8
-28	***0***	17	30.4	62	96.9	107	221.3
-27	0.4	18	31.4	63	98.9	108	225
-26	0.8	19	32.5	64	101	109	228.6
-25	1.3	20	33.5	65	103.1	110	232.3
-24	1.7	21	34.6	66	105.3	111	236.1
-23	2.2	22	35.7	67	107.4	112	239.8
-22	2.6	23	36.8	68	109.6	113	243.7
-21	3.1	24	37.9	69	111.8	114	247.5
-20	3.6	25	39	70	114.1	115	251.5
-19	4.1	26	40.2	71	116.4	116	255.4
-18	4.6	27	41.4	72	118.7	117	259.4
-17	5.1	28	42.6	73	121	118	263.5
-16	5.6	29	43.8	74	123.4	119	267.6
-15	6.2	30	45	75	125.8	120	271.7

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