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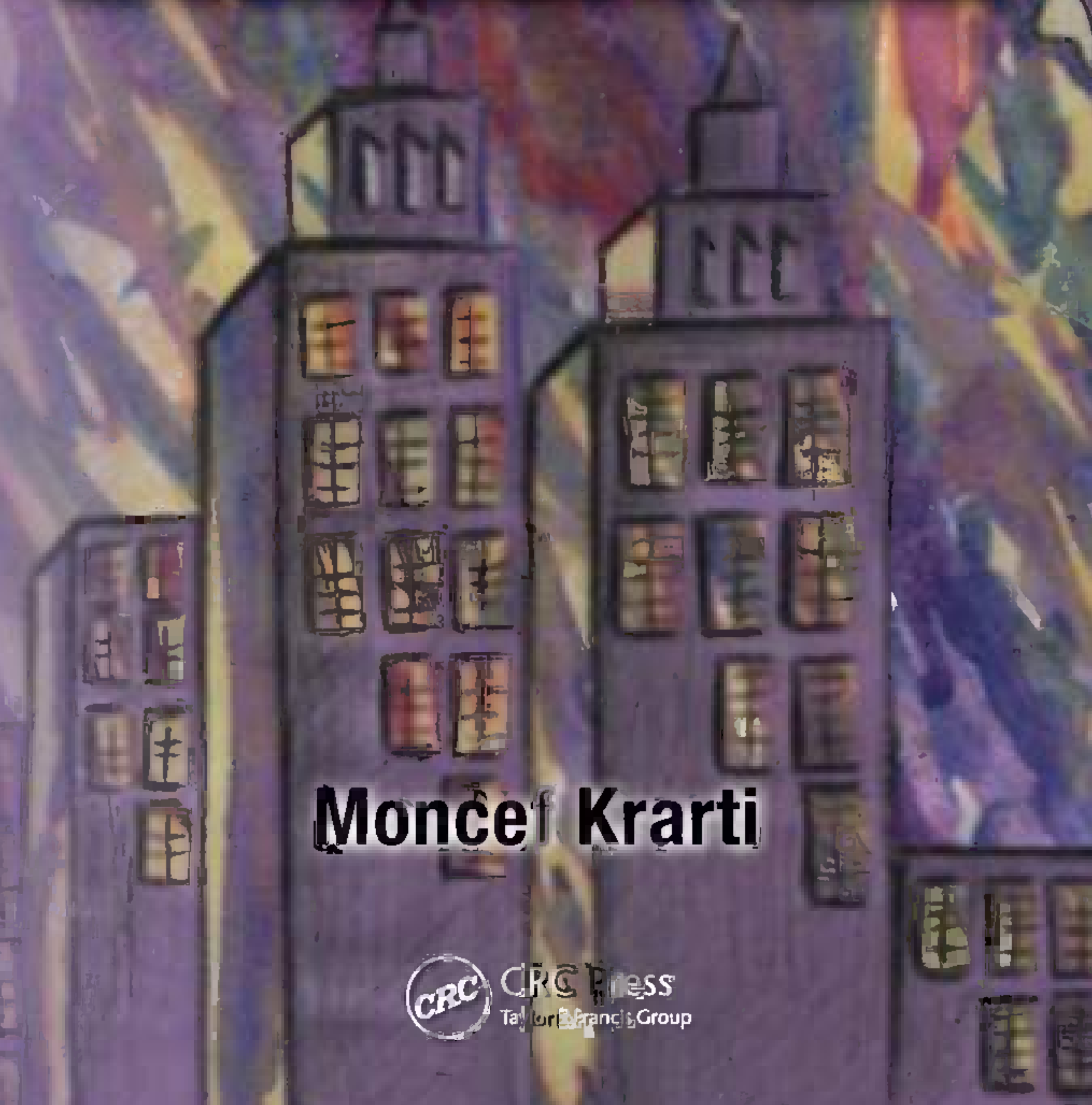
ENERGY AUDIT OF BUILDING SYSTEMS

AN ENGINEERING APPROACH



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Contents

Preface xvii

Author xix

1 Introduction to Energy Audit **1**-1

1.1 Introduction **1**-1
1.2 Types of Energy Audits **1**-2
1.2.1 Walk-Through Audit **1**-2
1.2.2 Utility Cost Analysis **1**-2
1.2.3 Standard Energy Audit **1**-2
1.2.4 Detailed Energy Audit **1**-3
1.3 General Procedure for a Detailed Energy Audit **1**-3
1.3.1 Step 1: Building and Utility Data Analysis **1**-4
1.3.2 Step 2: Walk-Through Survey **1**-4
1.3.3 Step 3: Baseline for Building Energy Use **1**-4
1.3.4 Step 4: Evaluation of Energy Savings Measures **1**-4
1.4 Common Energy Conservation Measures **1**-5
1.4.1 Building Envelope **1**-5
1.4.2 Electrical Systems **1**-7
1.4.3 HVAC Systems **1**-8
1.4.4 Compressed Air Systems **1**-9
1.4.5 Energy Management Controls **1**-9
1.4.6 Indoor Water Management **1**-9
1.4.7 New Technologies **1**-10
1.5 Case Study **1**-10
1.5.1 Step 1: Building and Utility Data Analysis **1**-11
1.5.2 Step 2: On-Site Survey **1**-13
1.5.3 Step 3: Energy Use Baseline Model **1**-14
1.5.4 Step 4: Evaluation of Energy Conservation Opportunities (ECOs) **1**-15
1.5.5 Step 5: Recommendations **1**-17
1.6 Verification Methods of Energy Savings **1**-17
1.7 Summary **1**-18

2 Energy Sources and Utility Rate Structures **2**-1

2.1 Introduction **2**-1
2.2 Energy Resources **2**-1

2.2.1 Electricity 2-2
2.2.1.1 Overall Consumption and Price 2-2
2.2.1.2 Future of U.S. Electricity Generation 2-3
2.2.1.3 Utility Deregulation Impact 2-4
2.2.2 Natural Gas 2-5
2.2.3 Petroleum Products 2-5
2.2.4 Coal 2-6
2.3 Electricity Rates 2-7
2.3.1 Common Features of Utility Rates 2-7
2.3.1.1 Billing Demand 2-8
2.3.1.2 Power Factor Clause 2-8
2.3.1.3 Ratchet Clause 2-10
2.3.1.4 Fuel Cost Adjustment 2-11
2.3.1.5 Service Level 2-12
2.3.2 Block Pricing Rates 2-12
2.3.3 Seasonal Pricing Rates 2-14
2.3.4 Innovative Rates 2-16
2.3.4.1 Time-of-Use (TOU) Rates 2-16
2.3.4.2 Real-Time-Pricing (RTP) Rates 2-16
2.3.4.3 The End-Use Rates 2-16
2.3.4.4 Specialty Rates 2-17
2.3.4.5 Financial Incentive Rates 2-17
2.3.4.6 Nonfirm Rates 2-17
2.3.4.7 Energy Purchase Rates 2-17
2.3.5 Real-Time-Pricing Rates 2-17
2.3.5.1 Category 1: Base Bill and Incremental Energy Charge Rates 2-18
2.3.5.2 Category 2: Total Energy Charge Rates 2-18
2.3.5.3 Category 3: Day-Type Rates 2-18
2.3.5.4 Category 4: Index-Type Rates 2-18
2.3.6 Case Study of RTP Rates 2-18
2.4 Natural Gas Rates 2-23
2.5 Utility Rates for Other Energy Sources 2-25
2.6 Summary 2-25

3 Economic Analysis 3-1
3.1 Introduction 3-1
3.2 Basic Concepts 3-1
3.2.1 Interest Rate 3-2
3.3 Inflation Rate 3-4
3.3.1 Tax Rate 3-5
3.3.2 Cash Flows 3-6
3.4 Compounding Factors 3-7
3.4.1 Single Payment 3-7
3.4.2 Uniform-Series Payment 3-8
3.5 Economic Evaluation Methods among Alternatives 3-9
3.5.1 Net Present Worth 3-9
3.5.2 Rate of Return 3-10
3.5.3 Benefit–Cost Ratio 3-10
3.5.4 Payback Period 3-10
3.5.5 Summary of Economic Analysis Methods 3-11

3.6 Life-Cycle Cost Analysis Method **3**-13
3.7 General Procedure for an Economic Evaluation **3**-15
3.8 Financing Options **3**-16
3.8.1 Direct Purchasing **3**-16
3.8.2 Leasing **3**-17
3.8.3 Performance Contracting **3**-17
3.9 Summary **3**-18

4 Energy Analysis Tools **4**-1
4.1 Introduction **4**-1
4.2 Ratio-Based Methods **4**-2
4.2.1 Introduction **4**-2
4.2.2 Types of Ratios **4**-3
4.2.3 Examples of Energy Ratios **4**-3
4.3 Inverse Modeling Methods **4**-4
4.3.1 Steady-State Inverse Models **4**-4
4.3.1.1 ANAGRAM Method **4**-5
4.3.1.2 PRISM Method **4**-7
4.3.2 Dynamic Models **4**-8
4.4 Forward Modeling Methods **4**-9
4.4.1 Steady-State Methods **4**-9
4.4.2 Degree-Day Methods **4**-9
4.4.3 Bin Methods **4**-10
4.4.4 Dynamic Methods **4**-11
4.5 Summary **4**-14

5 Electrical Systems **5**-1
5.1 Introduction **5**-1
5.2 Review of Basics **5**-1
5.2.1 Alternating Current Systems **5**-1
5.2.2 Power Factor Improvement **5**-4
5.3 Electrical Motors **5**-6
5.3.1 Introduction **5**-6
5.3.2 Overview of Electrical Motors **5**-6
5.3.3 Energy-Efficient Motors **5**-8
5.3.3.1 General Description **5**-8
5.3.3.2 Adjustable Speed Drives (ASDs) **5**-8
5.3.3.3 Energy Savings Calculations **5**-10
5.4 Lighting Systems **5**-13
5.4.1 Introduction **5**-13
5.4.2 Energy-Efficient Lighting Systems **5**-14
5.4.2.1 High-Efficiency Fluorescent Lamps **5**-15
5.4.2.2 Compact Fluorescent Lamps **5**-16
5.4.2.3 Compact Halogen Lamps **5**-16
5.4.2.4 Electronic Ballasts **5**-16
5.4.3 Lighting Controls **5**-17
5.4.3.1 Occupancy Sensors **5**-17
5.4.3.2 Light Dimming Systems **5**-18
5.4.3.3 Energy Savings from Daylighting Controls **5**-18

5.5 Electrical Appliances 5-20
5.5.1 Office Equipment 5-20
5.5.2 Residential Appliances 5-21
5.6 Electrical Distribution Systems 5-24
5.6.1 Introduction 5-24
5.6.2 Transformers 5-25
5.6.3 Electrical Wires 5-27
5.7 Power Quality 5-31
5.7.1 Introduction 5-31
5.7.2 Total Harmonic Distortion 5-31
5.8 Summary 5-34

6 Building Envelope 6-1
6.1 Introduction 6-1
6.2 Basic Heat Transfer Concepts 6-1
6.2.1 Heat Transfer from Walls and Roofs 6-1
6.2.2 Infiltration Heat Loss/Gain 6-3
6.2.3 Variable Base Degree-Days Method 6-9
6.3 Simplified Calculation Tools for Building Envelope Audit 6-11
6.3.1 Estimation of the Energy Use Savings 6-11
6.3.2 Estimation of the BLC for the Building 6-11
6.3.3 Estimation of the Degree Days 6-12
6.3.4 Foundation Heat Transfer Calculations 6-16
6.3.5 Simplified Calculation Method for Building Foundation Heat Loss/Gain 6-17
6.3.5.1 Calculation Example No. 1: Basement for a Residential Building 6-20
6.3.5.2 Calculation Example No. 2: Freezer Slab 6-21
6.4 Selected Retrofits for Building Envelope 6-25
6.4.1 Insulation of Poorly Insulated Building Envelope Components 6-25
6.4.2 Window Improvements 6-26
6.4.3 Reduction of Air Infiltration 6-27
6.5 Summary 6-29

7 Secondary HVAC Systems Retrofit 7-1
7.1 Introduction 7-1
7.2 Types of Secondary HVAC Systems 7-1
7.3 Ventilation Systems 7-3
7.3.1 Ventilation Air Intake 7-4
7.3.2 Air Filters 7-8
7.3.3 Air-Side Economizers 7-8
7.3.3.1 Temperature Economizer Cycle 7-8
7.3.3.2 Enthalpy Economizer Cycle 7-9
7.4 Ventilation of Parking Garages 7-9
7.4.1 Existing Codes and Standards 7-10
7.4.2 General Methodology for Estimating the Ventilation Requirements for Parking Garages 7-11
7.4.2.1 Step 1. Collect the Following Data 7-11
7.4.2.2 Step 2. 7-12
7.4.2.3 Step 3. 7-12

7.5 Indoor Temperature Controls 7-15
7.6 Upgrade of Fan Systems 7-15
7.6.1 Introduction 7-15
7.6.2 Basic Principles of Fan Operation 7-15
7.6.3 Duct Leakage 7-20
7.6.4 Damper Leakage 7-20
7.6.5 Size Adjustment 7-21
7.7 Common HVAC Retrofit Measures 7-22
7.7.1 Reduction of Outdoor Air Volume 7-22
7.7.2 Reset Hot or Cold Deck Temperatures 7-24
7.7.3 CV to VAV System Retrofit 7-25
7.8 Summary 7-26

8 Central Heating Systems 8-1
8.1 Introduction 8-1
8.2 Basic Combustion Principles 8-1
8.2.1 Fuel Types 8-1
8.2.2 Boiler Configurations and Components 8-3
8.2.2.1 Boiler Types 8-3
8.2.2.2 Firing Systems 8-4
8.2.3 Boiler Thermal Efficiency 8-5
8.3 Boiler Efficiency Improvements 8-7
8.3.1 Existing Boiler Tune-Up 8-9
8.3.2 High-Efficiency Boilers 8-11
8.3.3 Modular Boilers 8-11
8.4 Summary 8-12

9 Cooling Equipment 9-1
9.1 Introduction 9-1
9.2 Basic Cooling Principles 9-1
9.3 Types of Cooling Systems 9-5
9.3.1 Unitary AC Systems 9-5
9.3.2 Packaged AC Units 9-5
9.3.3 Heat Pumps 9-5
9.3.4 Central Chillers 9-6
9.3.4.1 Electric Chillers 9-6
9.3.4.2 Absorption Chillers 9-6
9.3.4.3 Engine-Driven Chillers 9-6
9.4 Water Distribution Systems 9-7
9.4.1 Pumps 9-7
9.4.2 Pump and System Curves 9-9
9.4.3 Analysis of Water Distribution Systems 9-10
9.5 District Cooling Systems 9-12
9.6 Multichiller Systems 9-15
9.7 Energy Conservation Measures 9-15
9.7.1 Chiller Replacement 9-17
9.7.2 Chiller Control Improvement 9-19
9.7.3 Alternative Cooling Systems 9-21
9.8 Summary 9-21

10 Energy Management Control Systems **10**-1
10.1 Introduction **10**-1
10.2 Basic Control Principles **10**-2
10.2.1 Control Modes **10**-2
10.2.2 Intelligent Control Systems **10**-6
10.2.3 Types of Control Systems **10**-7
10.3 Energy Management Systems **10**-8
10.3.1 Basic Components of an EMCS **10**-8
10.3.2 Typical Functions of EMCS **10**-9
10.3.3 Design Considerations of an EMCS **10**-10
10.3.4 Communication Protocols **10**-11
10.4 Control Applications **10**-12
10.4.1 Duty Cycling Controls **10**-14
10.4.2 Outdoor Air Intake Controls **10**-15
10.4.2.1 VAV Control Techniques for Economizer Systems **10**-17
10.4.2.2 VAV Control Techniques for Systems with a Dedicated Outside Air Duct **10**-19
10.4.2.3 Other VAV Control Techniques **10**-20
10.4.2.4 Comparative Analysis **10**-21
10.4.3 Optimum Start Controls **10**-22
10.4.4 Cooling/Heating Central Plant Optimization **10**-24
10.4.4.1 Single Chiller Control Improvement **10**-24
10.4.4.2 Controls for Multiple Chillers **10**-25
10.4.4.3 Controls for Multiple Boilers **10**-26
10.5 Summary **10**-26

11 Compressed Air Systems **11**-1
11.1 Introduction **11**-1
11.2 Review of Basic Concepts **11**-1
11.2.1 Production of Compressed Air **11**-1
11.2.1.1 Filters **11**-5
11.2.1.2 Receiving Tanks **11**-6
11.2.1.3 Dryers **11**-6
11.2.1.4 Intercoolers **11**-7
11.2.2 Distribution of Compressed Air **11**-7
11.2.2.1 Flow Pressure Drop **11**-7
11.2.2.2 Air Leaks **11**-8
11.2.3 Utilization of Compressed Air **11**-9
11.3 Common Energy Conservation Measures for Compressed Air Systems **11**-9
11.3.1 Reduction of Inlet Air Temperature **11**-10
11.3.2 Reduction of Discharge Pressure **11**-11
11.3.3 Repair of Air Leaks **11**-12
11.3.4 Other Energy Conservation Measures **11**-13
11.4 Summary **11**-13

12 Thermal Energy Storage Systems **12**-1
12.1 Introduction **12**-1
12.2 Types of TES Systems **12**-2
12.3 Principles of TES Systems **12**-4

12.4 Charging/Discharging of TES systems **12**-5
12.5 TES Control Strategies **12**-9
12.5.1 Full Storage **12**-9
12.5.2 Partial Storage **12**-9
12.5.2.1 Chiller-Priority Control **12**-9
12.5.2.2 Constant-Proportion Control **12**-10
12.5.2.3 Storage-Priority Control **12**-10
12.5.2.4 Optimal Controls **12**-10
12.5.3 Utility Rates **12**-11
12.5.3.1 TOU Rates **12**-11
12.5.3.2 RTP Rates **12**-11
12.6 Measures for Reducing Operating Costs **12**-12
12.6.1 Simplified Feasibility Analysis of TES Systems **12**-12
12.6.2 TES Control Improvement **12**-14
12.6.2.1 Effect of Plant Size **12**-14
12.6.2.2 Effect of the Cooling Load Profile **12**-15
12.7 Summary **12**-16

13 Cogeneration Systems **13**-1
13.1 Introduction **13**-1
13.2 History of Cogeneration **13**-2
13.3 Types of Cogeneration Systems **13**-3
13.3.1 Conventional Cogeneration Systems **13**-3
13.3.1.1 Bottoming Cycle **13**-5
13.3.1.2 Topping Cycle **13**-5
13.3.2 Packaged Cogeneration Systems **13**-8
13.3.3 Distributed Generation Technologies **13**-8
13.4 Evaluation of Cogeneration Systems **13**-10
13.4.1 Efficiency of Cogeneration Systems **13**-10
13.4.2 Simplified Feasibility Analysis of Cogeneration Systems **13**-12
13.4.3 Financial Options **13**-16
13.5 Case Study **13**-16
13.6 Summary **13**-18

14 Heat Recovery Systems **14**-1
14.1 Introduction **14**-1
14.2 Types of Heat Recovery Systems **14**-1
14.3 Performance of Heat Recovery Systems **14**-3
14.4 Simplified Analysis Methods **14**-6
14.5 Summary **14**-12

15 Water Management **15**-1
15.1 Introduction **15**-1
15.2 Indoor Water Management **15**-1
15.2.1 Water-Efficient Plumbing Fixtures **15**-2
15.2.1.1 Water-Saving Showerheads **15**-2
15.2.1.2 Water-Saving Toilets **15**-2
15.2.1.3 Water-Saving Faucets **15**-3
15.2.1.4 Repair Water Leaks **15**-3
15.2.1.5 Water/Energy Efficient Appliances **15**-4
15.2.2 Domestic Hot Water Usage **15**-4

15.3 Outdoor Water Management **15**-8
15.3.1 Irrigation and Landscaping **15**-8
15.3.2 Waste Water Reuse **15**-10
15.4 Swimming Pools **15**-10
15.4.1 Evaporative Losses **15**-11
15.4.2 Impact of Pool Covers **15**-13
15.5 Summary **15**-14

16 Methods for Estimating Energy Savings **16**-1
16.1 Introduction **16**-1
16.2 General Procedure **16**-2
16.3 Energy Savings Estimation Models **16**-4
16.3.1 Simplified Engineering Methods **16**-4
16.3.2 Regression Analysis Models **16**-6
16.3.2.1 Single-Variable Regression Analysis Models **16**-6
16.3.2.2 Multivariable Regression Analysis Models **16**-7
16.3.3 Dynamic Models **16**-10
16.3.4 Computer Simulation Models **16**-13
16.4 Applications **16**-17
16.5 Uncertainty Analysis **16**-18
16.6 Summary **16**-19

17 Case Studies **17**-1
17.1 Reporting Guidelines **17**-1
17.1.1 Reporting a Walk-Through Audit **17**-1
17.1.2 Reporting a Standard Audit **17**-2
17.2 Case Study 1: Walk-Through Audit of a Residence **17**-4
17.2.1 Building Description **17**-4
17.2.1.1 Building Envelope **17**-4
17.2.1.2 Building Infiltration **17**-4
17.2.1.3 HVAC System **17**-5
17.2.1.4 Water Management **17**-5
17.2.1.5 Appliances **17**-5
17.2.1.6 Thermal Comfort **17**-5
17.2.2 Energy Efficiency Measures **17**-5
17.2.2.1 Building Envelope **17**-5
17.2.2.2 Water Management **17**-6
17.2.2.3 Appliances **17**-6
17.2.3 Economic Analysis **17**-7
17.2.4 Recommendations **17**-7
17.3 Case Study 2: Standard Audit of a Residence **17**-7
17.3.1 Architectural Characteristics **17**-8
17.3.2 Utility Analysis **17**-9
17.3.3 Air Leakage Testing **17**-10
17.3.4 Energy Modeling **17**-12
17.3.5 Model Calibration **17**-13
17.3.6 Energy Conservation Measures **17**-14
17.3.7 Conclusions and Recommendations **17**-16

17.4 Case Study 3: Audit of a Museum **17**-17
17.4.1 Building Description **17**-17
17.4.1.1 HVAC Systems **17**-18
17.4.1.2 Electrical Systems **17**-19
17.4.2 Walk-Through Audit **17**-20
17.4.2.1 Lighting Systems **17**-20
17.4.2.2 Mechanical Systems **17**-21
17.4.2.3 Building Shell **17**-21
17.4.2.4 Other Issues **17**-21
17.4.3 Utility Data Analysis **17**-22
17.4.3.1 Base-Load Determination **17**-22
17.4.3.2 Building Load Characteristics **17**-23
17.4.4 Occupant Survey **17**-23
17.4.5 Field Testing and Measurements **17**-24
17.4.5.1 Lighting Quality **17**-24
17.4.5.2 Space Temperature and Humidity Profiles **17**-25
17.4.5.3 Thermal Imaging **17**-26
17.4.6 Energy Modeling **17**-27
17.4.6.1 Building Envelope, Geometry, and Thermal Zones **17**-28
17.4.6.2 HVAC Components **17**-29
17.4.6.3 Calibration of the Energy Model **17**-31
17.4.7 Analysis of Energy Conservation Measures **17**-32
17.4.7.1 Overview **17**-32
17.4.8 Energy Savings Estimation **17**-35
17.4.8.1 ECM 1: Delamping 30 Percent of Lamps **17**-35
17.4.8.2 ECM 2: Increased Roof Insulation **17**-36
17.4.8.3 ECM 3: Window Replacement **17**-36
17.4.8.4 ECM 4: Occupancy Sensors **17**-39
17.4.8.5 ECM 5: Premium Efficiency Pumps **17**-39
17.4.8.6 ECM 6: Improved Fume Hood Controls—Demand-Controlled Ventilation **17**-40
17.4.8.7 ECM 7: Improved Water Fixture Efficiency **17**-41
17.4.8.8 ECM 8: Optimized Package of ECMs **17**-42
17.4.9 Economic Analysis **17**-42
17.5 Summary and Recommendations **17**-44

Appendix A: Conversion Factors ..... **Appendix A**-1

Appendix B: Weather Data ..... **Appendix B**-1

References **References**-1

Preface

Worldwide, buildings are responsible for over 40 percent of the total primary energy use and related greenhouse emissions. Through standards and energy efficiency programs, several countries have succeeded in improving the energy performance of existing buildings. In 2005, the International Energy Agency estimated that since 1973 energy efficiency improvements have helped save over 50 percent of the energy consumed in the United States compared to the business-as-usual scenario without development and implementation of such measures (IEA, 2008). However, energy systems currently utilized in buildings are still far from achieving second law thermodynamic limits to efficiency. Even with current technologies, there is significant potential to improve energy efficiency cost-effectively for both new and existing buildings. The last few decades have seen major improvements in the efficiency of building energy systems including lighting, heating, and cooling equipment. In 2009, a study by the World Council for Sustainable Development (WBCSD) found that several energy efficiency projects were feasible with today's energy costs. Specifically, the study found that at oil prices of $60 U.S. per barrel, investments in existing building energy efficiency technologies can reduce related energy use and carbon footprints by 40 percent in five discounted payback years.

Significant investments are being made, especially in the United States and Europe, to further reduce energy consumption attributed to the existing building stock through weatherization, energy auditing, and retrofitting programs. It is a consensus among all countries that well-trained energy auditors are essential to the success of these building energy efficiency programs. It is the purpose of the second edition of this book to provide a training guide for energy auditors and energy managers outlining systematic and well-proven engineering analysis methods and techniques to reduce energy use and operating costs for both residential and commercial buildings.

The second edition of the book presents simplified analysis methods to evaluate energy conservation opportunities in buildings. These simplified methods are based on well-established engineering principles. In addition, several innovative yet proven energy efficiency technologies and strategies are presented. The book is designed to be a self-contained textbook aimed at seniors or first-year graduate students. The contents of this book can be covered in a one-semester course in energy management or building energy efficiency. The book can also be used as a reference for practitioners and as a text for continuing education short courses. Users of this book are assumed to have a basic understanding of building energy systems including the fundamentals of heat transfer and principles of heating, ventilating, and air-conditioning (HVAC). General concepts of engineering economics, building energy simulation, and building electrical systems are also recommended.

The second edition of the book is organized in 17 self-contained chapters. The first three chapters provide basic tools that are typically required to perform energy audits of buildings. Each of the following 12 chapters addresses a specific building subsystem or energy efficiency technology. The penultimate chapter provides an overview of basic engineering methods used to verify and measure actual energy savings attributed to implementation of energy efficiency projects. The final chapter is devoted to case studies. Each chapter includes some worked-out examples that illustrate the use of simplified analysis methods to evaluate the benefits of energy efficient measures or technologies. Problems are provided at the end of most chapters to serve as review or homework problems for users of the book. However, as the instructor of an energy management course at the University of Colorado, I found that the best approach for students to understand and apply the various analysis methods and tools discussed in this book is through group projects consisting of energy audits of real buildings.

When using this book as a textbook, the instructor should start at Chapter 1 and proceed through Chapter 17 in order. However, some of the chapters can be skipped or covered lightly depending on time constraints and the background of the students. First, general procedures suitable for building energy audits are presented (Chapter 1). Some of the analysis tools and techniques needed to perform building energy audits are then discussed. In particular, analysis methods are briefly provided for utility rate structures (Chapter 2), economic evaluation of energy efficiency projects (Chapter 3), and energy simulation of buildings (Chapter 4). In buildings, electrical systems consume a significant amount of energy. Several energy efficiency strategies and technologies are discussed to reduce energy use from lighting, motors, and appliances (Chapter 5). Various approaches to improve the building envelope are also outlined (Chapter 6). These approaches are particularly suitable for residential buildings characterized by skin-dominated heating/cooling loads. To maintain acceptable comfort levels, heating and cooling systems typically consume the most energy in a building. Several measures are described to improve the energy efficiency of secondary HVAC systems (Chapter 7), central heating and cooling plants (Chapters 8 and 9), and energy management control systems (Chapter 10). In addition, simple strategies are described to reduce the energy used by compressed air systems, especially in industrial facilities (Chapter 11). Selected technologies to reduce energy use and costs in buildings are discussed, including thermal energy storage systems (Chapter 12), cogeneration (Chapter 13), and heat recovery systems (Chapter 14). Cost-effective measures to improve water management inside and outside buildings are presented (Chapter 15). Analysis methods used for the measurement and verification of actual energy savings attributed to energy efficiency projects are briefly summarized (Chapter 16). Finally, general guidelines to draft reports after completing energy audits are presented with specific examples for three case studies (Chapter 17).

A special effort has been made to use metric (SI) units throughout the book. However, in several chapters English (IP) units are also used because they are still the standard set of units used in the United States. Conversion tables between the two unit systems (from English to metric and metric to English) are provided in Appendix A. Appendix B provides annual heating and cooling degree days as well as annual degree hours for various balance temperatures in alphabetic order for countries around the world. Appendix C, located on the CRC Press Web site at URL http://www.crcpress.com/product/isbn/9781439828717 provides expanded monthly weather data in both SI and IP units for over 300 sites located in both the United States and throughout the world in a searchable format.

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1

Introduction to Energy Audit

1.1 Introduction

Since the oil embargo of 1973, significant improvements have been made in the energy efficiency of new buildings. However, the vast majority of the existing building stock is more than 20 years old and does not meet current energy efficiency construction standards (IEA, 2008). Therefore, energy retrofits of existing buildings will be required for decades to come if the overall energy efficiency of the building stock is to meet the standards.

Investing to improve the energy efficiency of buildings provides an immediate and relatively predictable positive cash flow resulting from lower energy bills. In addition to the conventional financing options available to owners and building operators (such as loans and leases), other methods are available to finance energy retrofit projects for buildings. One of these methods is performance contracting, in which payment for a retrofit project is contingent upon its successful outcome. Typically, an energy services company (ESCO) assumes all the risks for a retrofit project by performing the engineering analysis and obtaining the initial capital to purchase and install equipment needed for energy efficiency improvements. Energy auditing is an important step used by energy service companies to ensure the success of their performance contracting projects.

Moreover, several large industrial and commercial buildings have established internal energy management programs based on energy audits to reduce waste in energy use or to comply with the specifications of some regulations and standards. Other building owners and operators take advantage of available financial incentives typically offered by utilities or state agencies to perform energy audits and implement energy conservation measures.

In the 1970s, building energy retrofits consisted of simple measures such as shutting off lights, turning down heating temperatures, turning up air-conditioning temperatures, and reducing hot water temperatures. Today, building energy management includes a comprehensive evaluation of almost all the energy systems within a facility. Therefore, the energy auditor should be aware of key energy issues such as the subtleties of electric utility rate structures and of the latest building energy efficiency technologies and their applications.

This chapter describes a general but systematic procedure for energy auditing suitable for both commercial buildings and industrial facilities. Some of the commonly recommended energy conservation measures are briefly discussed. A case study for an office building is presented to illustrate the various tasks involved in an energy audit. Finally, an overview is provided to outline the existing methods for measurement and verification of energy savings incurred by the implementation of energy conservation measures.

1.2 Types of Energy Audits

The term "energy audit" is widely used and may have different meanings depending on the energy service company. Energy auditing of buildings can range from a short walk-through of the facility to a detailed analysis with hourly computer simulation. Generally, four types of energy audits can be distinguished as briefly described below.

1.2.1 Walk-Through Audit

This audit consists of a short on-site visit of the facility to identify areas where simple and inexpensive actions can provide immediate energy use or operating-cost savings. Some engineers refer to these types of actions as operating and maintenance (O&M) measures. Examples of O&M measures include setting back heating set-point temperatures, replacing broken windows, insulating exposed hot water or steam pipes, and adjusting boiler fuel–air ratio. A sample of a walk-through audit for a residence is provided in Chapter 17.

1.2.2 Utility Cost Analysis

The main purpose of this type of audit is to carefully analyze the operating costs of the facility. Typically, the utility data over several years is evaluated to identify the patterns of energy use, peak demand, weather effects, and potential for energy savings. To perform this analysis, it is recommended that the energy auditor conduct a walk-through survey to get acquainted with the facility and its energy systems.

It is important that the energy auditor clearly understand the utility rate structure that applies to the facility for several reasons including:

- To check the utility charges and ensure that no mistakes were made in calculating the monthly bills. Indeed, the utility rate structures for commercial and industrial facilities can be quite complex with ratchet charges and power factor penalties.
- To determine the most dominant charges in the utility bills. For instance, peak demand charges can be a significant portion of the utility bill especially when ratchet rates are applied. Peak shaving measures can then be recommended to reduce these demand charges.
- To identify whether the facility can benefit from using other utility rate structures to purchase cheaper fuel and reduce its operating costs. This analysis can provide a significant reduction in the utility bills especially with implementation of electrical deregulation and the advent of real-time pricing (RTP) rate structures.

Moreover, the energy auditor can determine whether the facility is a candidate for energy retrofit projects by analyzing the utility data. Indeed, the energy use of the facility can be normalized and compared to indices (for instance, the energy use per unit of floor area—for commercial buildings—or per unit of a product—for industrial facilities—as discussed in Chapter 4).

1.2.3 Standard Energy Audit

The standard audit provides a comprehensive energy analysis for the energy systems of the facility. In addition to the activities described for the walk-through audit and for the utility cost analysis described above, the standard energy audit includes the development of a baseline for the energy use of the facility and the evaluation of the energy savings and the cost-effectiveness of appropriately selected energy conservation measures. The step-by-step approach of the standard energy audit is similar to that of the detailed energy audit described later on in the following section.

Typically, simplified tools are used in the standard energy audit to develop baseline energy models and to predict the energy savings of energy conservation measures. Among these tools are the degree-day methods and linear regression models (Fels, 1986). In addition, a simple payback analysis is generally performed to determine the cost-effectiveness of energy conservation measures. Examples of standard audits are provided in Chapter 17.

1.2.4 Detailed Energy Audit

This audit is the most comprehensive but also time-consuming energy audit type. Specifically, the detailed energy audit includes the use of instruments to measure energy use for the whole building or for some energy systems within the building (for instance, by end uses: lighting systems, office equipment, fans, chillers, etc.). In addition, sophisticated computer simulation programs are typically considered for detailed energy audits to evaluate and recommend energy retrofits for the facility.

The techniques available to perform measurements for an energy audit are diverse. During the on-site visit, handheld and clamp-on instruments can be used to determine the variation of some building parameters such as the indoor air temperature, luminance level, and electrical energy use. When long-term measurements are needed, sensors are typically used and connected to a data-acquisition system so measured data can be stored and be accessible remotely. Recently, nonintrusive load monitoring (NILM) techniques have been proposed (Shaw et al., 2005). The NILM technique can determine the real-time energy use of the significant electrical loads in a facility using only a single set of sensors at the facility service entrance. The minimal effort associated with using the NILM technique when compared to the traditional submetering approach (which requires a separate set of sensors to monitor energy consumption for each end-use) makes the NILM a very attractive and inexpensive load-monitoring technique for energy service companies and facility owners.

The computer simulation programs used in the detailed energy audit can typically provide the energy use distribution by load type (i.e., energy use for lighting, fans, chillers, boilers, etc.). They are often based on dynamic thermal performance of the building energy systems and typically require a high level of engineering expertise and training. These simulation programs range from those based on the bin method (Knebel, 1983) to those that provide hourly building thermal and electrical loads such as DOE-2 (LBL, 1980). The reader is referred to Chapter 4 for more detailed discussion of the energy analysis tools that can be used to estimate energy and cost savings attributed to energy conservation measures.

In the detailed energy audit, more rigorous economic evaluation of the energy conservation measures are generally performed. Specifically, the cost-effectiveness of energy retrofits may be determined based on the life-cycle cost (LCC) analysis rather than the simple payback period analysis. Life-cycle cost analysis takes into account a number of economic parameters such as interest, inflation, and tax rates. Chapter 3 describes some of the basic analytical tools that are often used to evaluate energy efficiency projects.

1.3 General Procedure for a Detailed Energy Audit

To perform an energy audit, several tasks are typically carried out depending on the type of audit and the size and function of the audited building. Some of the tasks may have to be repeated, reduced in scope, or even eliminated based on the findings of other tasks. Therefore, the execution of an energy audit is often not a linear process and is rather iterative. However, a general procedure can be outlined for most buildings.

1.3.1 Step 1: Building and Utility Data Analysis

The main purpose of this step is to evaluate the characteristics of the energy systems and the patterns of energy use for the building. The building characteristics can be collected from the architectural/mechanical/electrical drawings or from discussions with building operators. The energy use patterns can be obtained from a compilation of utility bills over several years. Analysis of the historical variation of the utility bills allows the energy auditor to determine if there are any seasonal and weather effects on the building energy use. Some of the tasks that can be performed in this step are presented below and the key results expected from each task are noted:

- Collect at least three years of utility data (to identify a historical energy use pattern).
- Identify the fuel types used (electricity, natural gas, oil, etc., to determine the fuel type that accounts for the largest energy use).
- Determine the patterns of fuel use by fuel type (to identify the peak demand for energy use by fuel type).
- Understand utility rate structure (energy and demand rates; to evaluate if the building is penalized for peak demand and if cheaper fuel can be purchased).
- Analyze the effect of weather on fuel consumption (to pinpoint any variations of energy use related to extreme weather conditions).
- Perform utility energy use analysis by building type and size (building signature can be determined including energy use per unit area: to compare against typical indices).

1.3.2 Step 2: Walk-Through Survey

From this step, potential energy savings measures should be identified. The results of this step are important because they determine if the building warrants any further energy auditing work. Some of the tasks involved in this step are

- Identify the customer concerns and needs.
- Check the current operating and maintenance procedures.
- Determine the existing operating conditions of major energy use equipment (lighting, HVAC systems, motors, etc.).
- Estimate the occupancy, equipment, and lighting (energy use density and hours of operation).

1.3.3 Step 3: Baseline for Building Energy Use

The main purpose of this step is to develop a base-case model that represents the existing energy use and operating conditions for the building. This model is to be used as a reference to estimate the energy savings incurred from appropriately selected energy conservation measures. The major tasks to be performed during this step are

- Obtain and review architectural, mechanical, electrical, and control drawings.
- Inspect, test, and evaluate building equipment for efficiency, performance, and reliability.
- Obtain all occupancy and operating schedules for equipment (including lighting and HVAC systems).
- Develop a baseline model for building energy use.
- Calibrate the baseline model using the utility data or metered data.

1.3.4 Step 4: Evaluation of Energy Savings Measures

In this step, a list of cost-effective energy conservation measures is determined using both energy savings and economic analysis. To achieve this goal, the following tasks are recommended:

- Prepare a comprehensive list of energy conservation measures (using the information collected in the walk-through survey).
- Determine the energy savings due to the various energy conservation measures pertinent to the building using the baseline energy use simulation model developed in Step 3.
- Estimate the initial costs required to implement the energy conservation measures.
- Evaluate the cost-effectiveness of each energy conservation measure using an economic analysis method (simple payback or life-cycle cost analysis).

Tables 1.1 and 1.2 provide summaries of the energy audit procedure recommended for commercial buildings and for industrial facilities, respectively. Energy audits for thermal and electrical systems are separated because they are typically subject to different utility rates.

1.4 Common Energy Conservation Measures

In this section some energy conservation measures (ECMs) commonly recommended for commercial and industrial facilities are briefly discussed. It should be noted that the list of ECMs presented below does not pretend to be exhaustive nor comprehensive. It is provided merely to indicate some of the options that the energy auditor can consider when performing an energy analysis of a commercial or industrial facility. More discussion of energy efficiency measures for various building energy systems is provided in later chapters of this book. However, it is strongly advised that the energy auditor keep abreast of any new technologies that can improve building energy efficiency. Moreover, the energy auditor should only recommend the ECMs based on a sound economic analysis for each ECM.

1.4.1 Building Envelope

For some buildings, the envelope (i.e., walls, roofs, floors, windows, and doors) can have an important impact on the energy used to condition the facility. The energy auditor should determine the actual characteristics of the building envelope. During the survey, a descriptive sheet for the building envelope should be established to include information such as construction materials (for instance, the level of insulation in walls, floors, and roofs), the area, and the number of building envelope assemblies (for instance, the type and the number of panes for the windows). In addition, comments on the repair needs and recent replacements should be noted during the survey.

Some of the commonly recommended energy conservation measures to improve the thermal performance of the building envelope are

1. *Addition of thermal insulation*. For building surfaces without any thermal insulation, this measure can be cost-effective.
2. *Replacement of windows*. When windows represent a significant portion of the exposed building surfaces, using more energy-efficient windows (high R-value, low-emissivity glazing, airtight, etc.) can be beneficial in both reducing the energy use and improving the indoor comfort level.
3. *Reduction of air leakage*. When the infiltration load is significant, leakage area of the building envelope can be reduced by simple and inexpensive weatherstripping techniques.

The energy audit of the envelope is especially important for residential buildings. Indeed, the energy use from residential buildings is dominated by weather inasmuch as heat gain or loss from direct conduction of heat or from air infiltration/exfiltration through building surfaces accounts for a major portion (50 to 80 percent) of the energy consumption. For commercial buildings, improvements to the building

TABLE 1.1 Energy Audit Summary for Residential and Commercial Buildings

Phase	Thermal Systems	Electric Systems
Utility analysis	• Thermal energy use profile (building signature) • Thermal energy use per unit area (or per student for schools or per bed for hospitals) • Thermal energy use distribution (heating, DHW, process, etc.) • Fuel types used • Weather effect on thermal energy use • Utility rate structure	• Electrical energy use profile (building signature) • Electrical energy use per unit area (or per student for schools or per bed for hospitals) • Electrical energy use distribution (cooling, lighting, equipment, fans, etc.) • Weather effect on electrical energy use • Utility rate structure (energy charges, demand charges, power factor penalty, etc.)
On-site survey	• Construction materials (thermal resistance type and thickness) • HVAC system type • DHW system • Hot water/steam use for heating • Hot water/steam for cooling • Hot water/steam for DHW • Hot water/steam for specific applications (hospitals, swimming pools, etc.)	• HVAC system type • Lighting type and density • Equipment type and density • Energy use for heating • Energy use for cooling • Energy use for lighting • Energy use for equipment • Energy use for air handling • Energy use for water distribution
Energy use baseline	• Review architectural, mechanical, and control drawings • Develop a base-case model (using any baselining method ranging from very simple to more detailed tools) • Calibrate the base-case model (using utility data or metered data)	• Review architectural, mechanical, electrical, and control drawings • Develop a base-case model (using any baselining method ranging from very simple to more detailed tools) • Calibrate the base-case model (using utility data or metered data)
Energy conservation measures	• Heat recovery system (heat exchangers) • Efficient heating system (boilers) • Temperature setback • EMCS • HVAC system retrofit • DHW use reduction • Cogeneration	• Energy efficient lighting • Energy efficient equipment (computers) • Energy efficient motors • HVAC system retrofit • EMCS • Temperature setup • Energy efficient cooling system (chiller) • Peak demand shaving • Thermal energy storage system • Cogeneration • Power factor improvement • Reduction of harmonics

envelope are often not cost-effective due to the fact that modifications to the building envelope (replacing windows, adding thermal insulation in walls) are typically considerably expensive. However, it is recommended to audit the envelope components systematically not only to determine the potential for energy savings but also to ensure the integrity of its overall condition. For instance, thermal bridges—if present—can lead to a heat transfer increase and to moisture condensation. The moisture condensation is often more damaging and costly than the increase in heat transfer because it can affect the structural integrity of the building envelope.

TABLE 1.2 Energy Audit Summary for Industrial Facilities

Phase	Thermal Systems	Electric Systems
Utility analysis	• Thermal energy use profile (building signature) • Thermal energy use per unit of a product • Thermal energy use distribution (heating, process, etc.) • Fuel types used • Analysis of the thermal energy input for specific processes used in the production line (such as drying) • Utility rate structure	• Electrical energy use profile (building signature) • Electrical energy use per unit of a product • Electrical energy use distribution (cooling, lighting, equipment, process, etc.) • Analysis of the electrical energy input for specific processes used in the production line (such as drying) • Utility rate structure (energy charges, demand charges, power factor penalty, etc.)
On-site survey	• List of equipment that uses thermal energy • Perform heat balance of the thermal energy • Monitor of thermal energy use of all or part of the equipment • Determine the by-products of thermal energy use (such as emissions and solid waste)	• List of equipment that uses electrical energy • Perform heat balance of the electrical energy • Monitor electrical energy use of all or part of the equipment • Determine the by-products of electrical energy use (such as pollutants)
Energy use baseline	• Review mechanical drawings and production flow charts • Develop a base-case model using (any baselining method) • Calibrate the base-case model (using utility data or metered data)	• Review electrical drawings and production flow charts • Develop a base-case model (using any baselining method) • Calibrate the base-case model (using utility data or metered data)
Energy conservation measures	• Heat recovery system • Efficient heating and drying system • EMCS • HVAC system retrofit • Hot water and steam use reduction • Cogeneration (possibly with solid waste from the production line)	• Energy efficient motors • Variable speed drives • Air compressors • Energy efficient lighting • HVAC system retrofit • EMCS • Cogeneration (possibly with solid waste from the production line) • Peak demand shaving • Power factor improvement • Reduction of harmonics

1.4.2 Electrical Systems

For most commercial buildings and a large number of industrial facilities, the electrical energy cost constitutes the dominant part of the utility bill. Lighting, office equipment, and motors are the electrical systems that consume the major part of energy in commercial and industrial buildings.

1. *Lighting*. Lighting for a typical office building represents on average 40 percent of the total electrical energy use. There is a variety of simple and inexpensive measures to improve the efficiency of lighting systems. These measures include the use of energy-efficient lighting lamps and ballasts, the addition of reflective devices, delamping (when the luminance levels are above the recommended levels by the standards), and the use of daylighting controls. Most lighting measures are especially cost-effective for office buildings for which payback periods are less than one year.

TABLE 1.3 Typical Efficiencies of Motors

Motor Size (Hp)	Standard Efficiency (%)	Premium Efficiency (%)
1	73.0	85.5
2	75.0	86.5
3	77.0	86.5
5	80.0	89.5
7.5	82.0	89.5
10	85.0	91.7
15	86.0	92.4
20	87.5	93.0
30	88.0	93.6
40	88.5	93.6
50	89.5	94.1

2. *Office Equipment*. Office equipment constitutes the fastest growing part of the electrical loads especially in commercial buildings. Office equipment includes computers, fax machines, printers, and copiers. Today, there are several manufacturers that provide energy-efficient office equipment (such those that comply with the U.S. EPA Energy Star specifications). For instance, energy-efficient computers automatically switch to a low-power "sleep" mode or off mode when not in use.
3. *Motors*. The energy cost to operate electric motors can be a significant part of the operating budget of any commercial or industrial building. Measures to reduce the energy cost of using motors include reducing operating time (turning off unnecessary equipment), optimizing motor systems, using controls to match motor output with demand, using variable speed drives for air and water distribution, and installing energy-efficient motors. Table 1.3 provides typical efficiencies for several motor sizes.

In addition to the reduction in total facility electrical energy use, retrofits of the electrical systems decrease space cooling loads and therefore further reduce the electrical energy use in the building. These cooling energy reductions as well as possible increases in thermal energy use (for space heating) should be accounted for when evaluating the cost-effectiveness of improvements in lighting and office equipment.

1.4.3 HVAC Systems

The energy use due to HVAC systems can represent 40 percent of the total energy consumed by a typical commercial building. The energy auditor should obtain the characteristics of major HVAC equipment to determine the condition of the equipment, operating schedule, quality of maintenance, and control procedures. A large number of measures can be considered to improve the energy performance of both primary and secondary HVAC systems. Some of these measures are listed below:

1. *Setting Up/Back Thermostat Temperatures:* When appropriate, setback of heating temperatures can be recommended during unoccupied periods. Similarly, setup of cooling temperatures can be considered.
2. *Retrofit of Constant Air Volume Systems:* For commercial buildings, variable air volume (VAV) systems should be considered when the existing HVAC systems rely on constant volume fans to condition part or the entire building.
3. *Installation of Heat Recovery Systems:* Heat can be recovered from some HVAC equipment. For instance, heat exchangers can be installed to recover heat from air handling unit (AHU) exhaust air streams and from boiler stacks.

4. *Retrofit of Central Heating Plants:* The efficiency of a boiler can be drastically improved by adjusting the fuel–air ratio for proper combustion. In addition, installation of new energy-efficient boilers can be economically justified when old boilers are to be replaced.
5. *Retrofit of Central Cooling Plants:* Currently, there are several chillers that are energy-efficient and easy to control and operate and are suitable for retrofit projects.

It should be noted that there is a strong interaction among various components of the heating and cooling system. Therefore, a whole-system analysis approach should be followed when retrofitting a building HVAC system. Optimizing the energy use of a central cooling plant (which may include chillers, pumps, and cooling towers) is one example of using a whole-system approach to reduce the energy use for heating and cooling buildings.

1.4.4 Compressed Air Systems

Compressed air has become an indispensable tool for most manufacturing facilities. Its uses range from air-powered hand tools and actuators to sophisticated pneumatic robotics. Unfortunately, staggering amounts of compressed air are currently wasted in a large number of facilities. It is estimated that only a fraction of 20 to 25 percent of input electrical energy is delivered as useful compressed air energy. Leaks are reported to account for 10 to 50 percent of the waste and misapplication accounts for 5 to 40 percent of loss in compressed air (Howe and Scales, 1998).

To improve the efficiency of compressed air systems, the auditor can consider several issues including whether compressed air is the right tool for the job (for instance, electric motors are more energy efficient than air-driven rotary devices), how compressed air is applied (for instance, lower pressures can be used to supply pneumatic tools), how it is delivered and controlled (for instance, the compressed air needs to be turned off when the process is not running), and how the compressed air system is managed (for each machine or process, the cost of compressed air needs to be known to identify energy and cost savings opportunities).

1.4.5 Energy Management Controls

With the constant decrease in the cost of computer technology, automated control of a wide range of energy systems within commercial and industrial buildings is becoming increasingly popular and cost-effective. An energy management and control system (EMCS) can be designed to control and reduce the building energy consumption within a facility by continuously monitoring the energy use of various pieces of equipment and making appropriate adjustments. For instance, an EMCS can automatically monitor and adjust indoor ambient temperatures, set fan speeds, open and close air handling unit dampers, and control lighting systems.

If an EMCS is already installed in the building, it is important to recommend a system tune-up to ensure that the controls are operating properly. For instance, the sensors should be calibrated regularly in accordance with manufacturers' specifications. Poorly calibrated sensors may cause an increase in heating and cooling loads and may reduce occupant comfort.

1.4.6 Indoor Water Management

Water and energy savings can be achieved in buildings by using water-saving fixtures instead of conventional fixtures for toilets, faucets, showerheads, dishwashers, and clothes washers. Savings can also be achieved by eliminating leaks in pipes and fixtures.

Table 1.4 provides typical water use of conventional and water-efficient fixtures for various end-uses. In addition, Table 1.4 indicates the hot water use by each fixture as a fraction of the total water. With water-efficient fixtures, savings of 50 percent of water use can be achieved for toilets, showers, and faucets.

TABLE 1.4 Usage Characteristics of Water-Using Fixtures

End-Use	Conventional Fixtures	Water-Efficient Fixtures	Usage Pattern	% Hot water
Toilets	3.5 gal/flush	1.6 gal/flush	4 flushes/pers/day	0
Showers	5.0 gal/min	2.5 gal/min	5 min./shower	60
Faucets	4.0 gal/min	2.0 gal/min	2.5 min/pers/day	50
Dishwashers	14.0 gal/load	8.5 gal/load	0.17 loads/pers/day	100
Clothes washers	55.0 gal/load	42.0 gal/load	0.3 loads/pers/day	25
Leaks	10% of total use	2% of total use	N/A	50

1.4.7 New Technologies

The energy auditor may consider the potential of implementing and integrating new technologies within the facility. It is therefore important that the energy auditor understand these new technologies and know how to apply them. Among the new technologies that can be considered for commercial and industrial buildings are

1. *Building Envelope technologies:* Recently several materials and systems have been proposed to improve the energy efficiency of the building envelope and especially windows including:
   Spectrally selective glasses that can optimize solar gains and shading effects.
   Chromogenic glazings that change their properties automatically depending on temperature or light-level conditions (similar to sunglasses that become dark in sunlight).
   Building integrated photovoltaic panels that can generate electricity while absorbing solar radiation and reducing heat gain through the building envelope (typically roofs).
2. *Light Pipe technologies:* Although the use of daylighting is straightforward for perimeter zones that are near windows, it is not usually feasible for interior spaces, particularly those without any skylights. Recent but still emerging technologies allow the engineer to "pipe" light from roof or wall-mounted collectors to interior spaces that are not close to windows or skylights.
3. *HVAC systems and controls:* Several strategies can be considered for energy retrofits including:
   Heat recovery technologies such as rotary heat wheels and heat pipes can recover 50 to 80 percent of the energy used to heat or cool ventilation air supplied to the building.
   Desiccant-based cooling systems are now available and can be used in buildings with large dehumidification loads during long periods (such as hospitals, swimming pools, and supermarket fresh produce areas).
   Geothermal heat pumps can provide an opportunity to take advantage of the heat stored underground to condition building spaces.
   Thermal energy storage (TES) systems offer a means of using less-expensive off-peak power to produce cooling or heating to condition the building during on-peak periods. Several optimal control strategies have been developed in recent years to maximize the cost savings of using TES systems.
4. *Cogeneration:* This is not really a new technology. However, recent improvements in its combined thermal and electrical efficiency have made cogeneration cost-effective in several applications including institutional buildings such as hospitals and universities.

1.5 Case Study

To illustrate the energy audit process described above, a case study is presented in this section. The activities performed for each step of the energy audit are briefly described. For more details about the

FIGURE 1.1 Front view of the audited office building.

case study, the reader is referred to Kim et al. (1998). Other case studies are presented in Chapter 17 of this book. The building analyzed in this case study is a medium-size office building located in Seoul, Korea. Figure 1.1 shows the front view of the building.

1.5.1 Step 1: Building and Utility Data Analysis

The first step in the building energy audit process is to collect all available information about the energy systems and the energy use pattern of the building. This information was collected before the field survey. In particular, from the architectural/mechanical/electrical drawings and utility bills, the following information and engineering data were gathered:

Building Characteristics: The building is a 26-story office building with 2-story penthouse and 4-story basement. It is located in downtown Seoul, Korea. The structure of the building consists of modular concrete and steel frame. The building area is 3,920 m^2 and the site area is 6,555 m^2. Single glazed windows are installed throughout the building. Figure 1.2 shows a typical floor plan of the building. Table 1.5 describes the various construction materials used throughout the building.

Energy Use: Figure 1.3 summarizes the monthly electrical energy use of the building for 1993. The monthly average dry-bulb outdoor air temperatures recorded during 1993 are also presented in Figure 1.3. It is clear that the electrical energy use increases during the summer months (June through October) when the outdoor temperatures are high. During the other months, the electrical energy use is almost constant and can mostly be attributed to lighting and office equipment. Preliminary analysis of the metered building energy use indicated that natural gas consumption is inconsistent from month to month. For example, gas consumption during January is six times higher than during December, even though the weather conditions are similar for both months. Therefore, the recorded data for the natural gas use were considered unreliable and only metered electrical energy use data were analyzed.

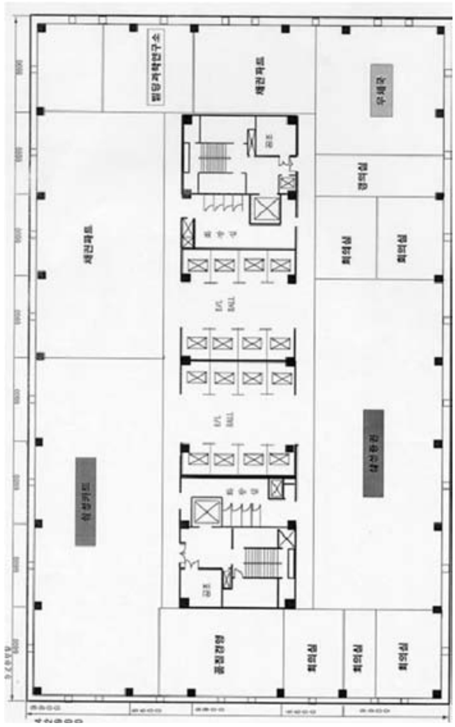


FIGURE 1.2 Typical floor plan of the audited office building.

TABLE 1.5 Building Construction Materials

Component	Materials
Exterior wall	5 cm tile
	16 cm concrete
	2.5 cm foam insulation
	0.6 cm finishing material
Roof	5 cm lightweight concrete
	15 cm concrete
	2.5 cm foam insulation
Interior wall	2 cm finishing cement mortar
	19 cm concrete block
	2 cm finishing cement mortar
Glazing	1.2 cm thick single pane glazing
Underground wall	25 cm concrete
	Asphalt shingle
	Air-space
	10 cm brick
	2 cm finishing cement mortar
Underground floor	15 cm concrete
	Asphalt shingle
	12 cm concrete
	2 cm finishing cement mortar

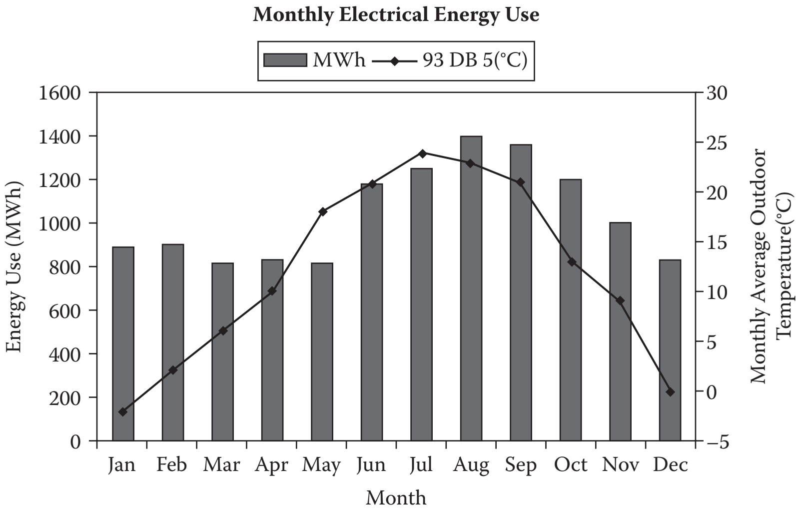


FIGURE 1.3 Monthly actual electrical energy consumption.

1.5.2 Step 2: On-Site Survey

A one-day field survey was conducted with the assistance of the building operator during the summer of 1996. During the survey, much useful and revealing information and engineering data were collected. For instance:

- It was found that the building had been retrofitted with energy-efficient lighting systems. The measurement of luminance levels throughout the working areas indicated adequate lighting. To determine an estimate of the energy use for lighting, the number and type of lighting fixtures were recorded.
- It was observed that the cooling and heating temperature set-points were set to be 25.5°C and 24.5°C, respectively. However, indoor air temperature and relative humidity measurements during the field survey revealed that during the afternoon, the thermal conditions are uncomfortable in several office spaces with average air dry-bulb temperature of 28°C and relative humidity of 65 percent. A discussion with the building operator indicated that the chillers are no longer able to meet the cooling loads after the addition of several computers in the building during the last few years. As a solution an ice storage tank was then added to reduce the peak cooling load.
- It was discovered during the survey that the building is heated and cooled simultaneously by two systems: constant air volume (CAV) and fan coil unit (FCU) systems. The CAV system is complemented by the FCU system as necessary. Two air-handling units serve the entire building, and about 58 FCUs are located on each floor.
- The heating and cooling plant consists of three boilers, six chillers, three cooling towers, and one ice storage tank. The capacity of the boilers and the chillers is provided below:
  - Boilers: 13 MBtuh (2 units) and 3.5 Mbtuh (1 unit)
  - Chillers: 215 tons (5 units) and 240 tons (1 unit)

The thermal energy storage system consists of a brine ice-on-coil tank. The charging and discharging hours are 10 and 13, respectively. The TES system is currently controlled using simple and nonpredictive storage-priority controls.

The building has relatively high internal heat gains. Some of the building internal heat gain sources are listed in Table 1.6. Operating schedules were based on the discussion with the building operators and on observations during the field survey.

TABLE 1.6 Internal Heat Gain Level for the Office Building

Internal Heat Gain	Design Load
Occupancy	17 m^2/person;
	Latent heat gain: 45 W
	Sensible heat gain: 70 W
Lighting	14 W/m^2
Equipment	16 W/m^2
Ventilation	14.7 CFM/person

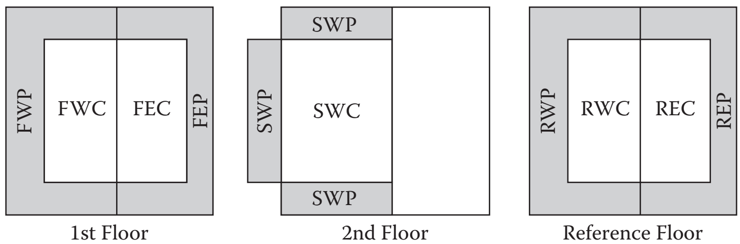


FIGURE 1.4 Building zoning configuration for DOE-2 computer simulation.

1.5.3 Step 3: Energy Use Baseline Model

To model the building using DOE-2, each floor was divided into two perimeter and two core zones. Figure 1.4 shows the zone configuration used to model the building floors. The main reason for this zoning configuration is the lack of flexibility in the DOE-2 SYSTEMS program. Although the actual building is conditioned by the combination of constant air volume and fan coil unit systems, the SYSTEM module of DOE-2.1E cannot model two different types of HVAC systems serving one zone. Therefore, a simplification has been made to simulate the actual HVAC system of the building. This simplification consists of the following. The perimeter zone is conditioned by the FCUs, whereas the core zone is conditioned by the CAV. Because all the FCUs are located at the perimeter, this simplification is consistent with the actual HVAC system operation.

Figure 1.5 shows the monthly electrical energy consumption predicted by the DOE-2 base model and the actual energy use recorded in 1993 for the building. It shows that DOE-2 predicts the actual energy use pattern of the building fairly well, except for the months of September and October. The difference between the annual metered energy use in the building and the annual predicted electricity use by the DOE-2 base-case model is about 762 MWh. DOE-2 predicts that the building consumes 6 percent more electricity than the actual metered annual energy use. To develop the DOE-2 base-case model, a TRY-type weather file of Seoul was created using the raw weather data collected for 1993. Using the DOE-2 base-case model, a number of ECOs (energy conservation opportunities) can be evaluated.

Figure 1.6 shows the distribution by end-uses of the building energy use. The electrical energy consumption of the building is dominated by lighting and equipment. The electricity consumption for lighting and office equipment represents about 75 percent of the total building electricity consumption. As mentioned earlier, a recent lighting retrofit has been performed in the building using electronic ballasts and energy-efficient fluorescent fixtures. Therefore, it was decided not to consider a lighting retrofit as an ECO for this study. The electricity consumption for cooling is about 13.1 percent. The ECOs selected for this building mostly attempt to reduce the cooling loads in order to improve indoor thermal comfort as well as save building energy cost.

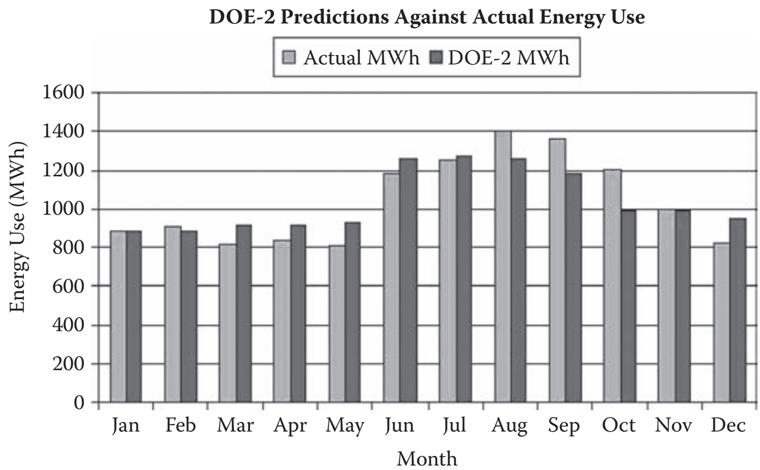


FIGURE 1.5 Comparison of DOE-2 prediction and actual building electrical energy use.

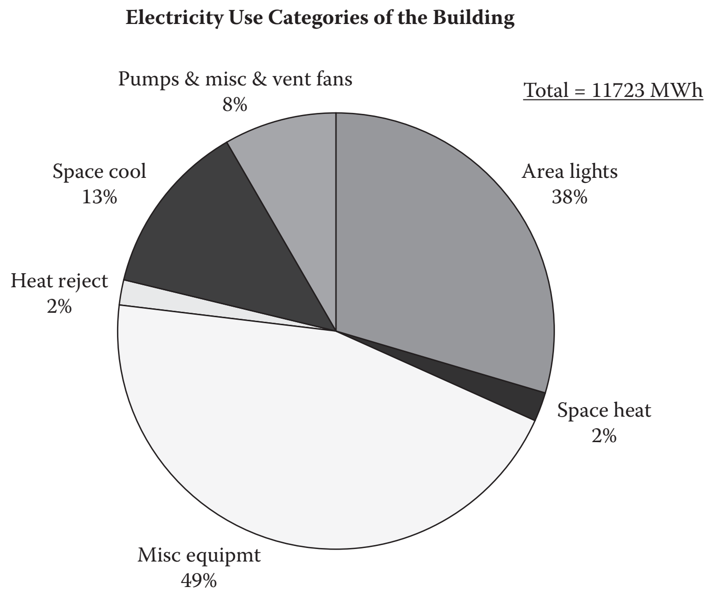


FIGURE 1.6 Electricity use distribution.

1.5.4 Step 4: Evaluation of Energy Conservation Opportunities (ECOs)

Based on the evaluation of the energy use pattern of the building, several energy conservation opportunities for the building were analyzed. Among the ECOs considered in the study, six of them successfully reduced energy consumption:

ECO #1: CAV to VAV conversion: The present AHU fans are all constant speed fans. They supply conditioned air through a constant volume air-supply system to the conditioned zones. The system is designed to supply enough air to heat or cool the building under design conditions. Under nondesign conditions, more air than needed is supplied. Changing the system to a variable-air-volume system would reduce the amount of air supplied by the AHUs and result in less energy to condition the various zones. For this ECO, the constant-volume reheat fan system assigned for the core zones in the building was changed to a VAV system. In particular, VAV boxes—controlled by the space thermostat—are proposed to vary the amount of conditioned air supplied to the building zones to control the indoor temperature. Both labor and equipment costs were included in the estimation of the payback period for this measure.

ECO #2: Optimal ice storage control: The current TES system is operated using a nonpredictive storage-priority control. To improve the benefits of the TES system, a near-optimal controller is suggested. This ECO is analyzed to determine if the cost of electrical energy consumption for the building is reduced when a near-optimal control strategy is used. To determine the savings of this option, the simulation environment developed by Henze, Krarti, and Brandemuehl (1997a) is used. This simulation environment is based on a dynamic programming technique and determines the best operating controls for the TES system given the cooling and noncooling load profiles and electrical rate structure. No DOE-2 simulation is performed for this ECO. The results of the dynamic programming simulation indicate that an energy use reduction of 5 percent can be achieved using a near-optimal control in lieu of the storage-priority control. To implement this near-optimal control, a predictor is required to determine future building cooling or noncooling loads. An example of such a predictor could be based on neural networks (Kreider et al., 1995). The labor cost and the initial cost of adding some sensors and a computer were considered to determine the payback period for this measure.

ECO #3: Glazing retrofit: For this building, low-e glazing systems are considered to reduce the internal heat gain due to solar radiation. Thus, the cooling load is reduced. In addition, the increased U-value of the glazing reduces the heating load. For this ECO, the existing single pane windows with the glass conductance of 6.17 (W/m^2-K) and the shading coefficient of 0.69 were changed to the double-pane windows. These double-pane windows reduce both the glass conductance and the shading coefficient to 1.33 (W/m^2-K) and 0.15, respectively.

ECO #4: Indoor temperature setback/setup: In this ECO, the impact of the indoor temperature setting on the building energy use is analyzed using the DOE-2 simulation program; the heating temperature was set from 24.5°C to 22.5°C and the cooling temperature was set from 25.5°C to 27.5°C. There is only the labor cost associated with this measure.

ECO #5: Motor replacement: Increasing the efficiency of the motors for fans and pumps can reduce the total electric energy consumption in the building. In this ECO, the existing efficiencies for the motors were assumed to range from 0.85 (for 10 hp motors) to 0.90 (for 50 hp motors). Energy-efficient motors have efficiencies that range from 0.91 (for 10 hp motors) to 0.95 (for 50 hp motors). Only the differential cost was considered in the economic analysis.

ECO #6: Daylighting control plan: A continuous dimming control would regulate the light level so that the luminance level inside the zones remained constant. The electricity consumption of the building can be significantly reduced, and the gas consumption can be slightly increased because of the reduced space heat gain from the lighting system. For this ECO, a daylighting system with continuous dimming control was considered for perimeter office zones.

The impact of the selected ECOs on electricity use in the building as predicted by DOE-2 simulations is shown in Table 1.7. Based on these results, "converting CAV to VAV" and "implementing daylighting control system with dimming control" reduces the total electricity consumption of the building 5.2 and 7.3 percent, respectively. These savings are significant considering that the electricity consumption for the cooling plant alone is about 13.1 percent of the total electricity consumption of the building.

The economic analysis was performed using the utility rate of Seoul, Korea. Table 1.7 presents the energy cost savings of ECOs in Korean currency (1,000 Won = $1 U.S.). In addition to the electricity cost, the natural gas cost was also included in the economic analysis. The natural gas is used only for heating the building. The economic analysis shows that the VAV conversion reduces the total building energy cost by more than 10 percent, and the daylighting control saves about 6 percent of the total energy costs.

TABLE 1.7 Economic Analysis of the ECOs

	Electricity Cost (MWon)	LNG Cost (MWon)	Total Cost (MWon)	Capital Cost (MWon)	Saving (%)	Savings (MWon)	Payback Period (Years)
Base Case	984.4	139.1	1,123.5	—	—	—	—
ECO #1	940.8	49.8	990.5	465.5	11.8	133.0	3.5
ECO #2	979.1	139.1	1,118.2	42.4	0.6	5.3	8.0
ECO #3	977.9	126.9	1,104.8	280.5	1.7	18.7	15.0
ECO #4	983.7	106.4	1,090.1	16.7	3.0	33.4	0.5
ECO #5	972.6	138.7	1,111.4	60.5	1.1	12.1	5.0
ECO #6	911.6	144.8	1,056.4	268.4	6.0	67.1	4.0

1.5.5 Step 5: Recommendations

From the results for economic analysis, the VAV conversion (ECO #1), adjustment of temperature set-point (ECO #4), and the daylighting control (ECO #6) are the recommended energy-saving opportunities to be implemented for the audited office building.

1.6 Verification Methods of Energy Savings

Energy conservation retrofits are deemed cost-effective based on predictions of energy and cost savings. However, several studies have found that large discrepancies exist between actual and predicted energy savings. Due to the significant increase in the activities of energy service companies (ESCOs), the need became evident for standardized methods for measurement and verification of energy savings. This interest led to the development of the *North American Energy Measurement and Verification Protocol* published in 1996 and later expanded and revised under the *International Performance Measurement and Verification Protocol*.

In principle, the measurement of the retrofit energy savings can be obtained by simply comparing the energy use during pre- and postretrofit periods. Unfortunately, the change in energy use between the pre- and postretrofit periods is not only due to the retrofit itself but also to other factors such as changes in weather conditions, levels of occupancy, and HVAC operating procedures. It is important to account for all these changes to determine the retrofit energy savings accurately.

Several methods have been proposed to measure and verify energy savings of implemented energy conservation measures in commercial and industrial buildings. Chapter 16 describes a number of methods suitable for measurement and verification of energy savings. Some of these techniques are briefly described below:

1. *Regression Models:* The early regression models used to measure savings adapted the variable-base degree-day (VBDD) method. Among these early regression models, the Princeton scorekeeping method (PRISM) was used to measure monthly energy consumption data and daily average temperatures to calibrate a linear regression model and determine the best values for nonweather-dependent consumption, the temperature at which the energy consumption began to increase due to heating or cooling (the change-point or base temperature), and the rate at which the energy consumption increased. Several studies have indicated that the simple linear regression model is suitable for estimating energy savings for residential buildings. However, subsequent work has shown that the PRISM model does not provide accurate estimates for energy savings for most commercial buildings (Ruch and Claridge, 1992). Single-variable (temperature) regression models require the use of at least four-parameter segmented linear or change-point regressions to be suitable for commercial buildings. Katipamula, Reddy, and Claridge (1994) proposed multiple linear regression models to include as independent variables internal gain, solar radiation, wind, and humidity ratio in addition to the outdoor temperature. For the buildings considered in

their analysis, Katipamula, Reddy, and Claridge found that wind and solar radiation have small effects on the energy consumption. They also found that internal gains have a generally modest impact on energy consumption. Katipamula, Reddy, and Claridge (1998) discuss in more detail the advantages and the limitations of multivariate regression modeling.

2. *Time-Variant Models:* There are several techniques that are proposed to include the effect of time variation of several independent variables on estimating the energy savings due to retrofits of building energy systems. Among these techniques are artificial neural networks (Krarti et al., 1998), Fourier series (Dhar, Reddy, and Claridge, 1998), and nonintrusive load monitoring (Shaw et al., 1998). These techniques are typically involved and require a high level of expertise and training.

1.7 Summary

An energy audit of commercial and industrial buildings encompasses a wide variety of tasks and requires expertise in a number of areas to determine the best energy conservation measures suitable for an existing facility. This chapter provided a description of a general but systematic approach to performing energy audits. If followed carefully, the approach helps facilitate the process of analyzing a seemingly endless array of alternatives and complex interrelationships between building and energy system components.

2

Energy Sources and Utility Rate Structures

2.1 Introduction

Energy cost is an important part of the economic viability of several energy conservation measures. Therefore, it is crucial that an energy auditor or building manager understand how energy costs are determined. Generally, a considerable number of utility rate structures do exist within the same geographical location. Each utility rate structure may include several clauses and charges that sometimes make following the energy billing procedure a complicated task. The complexity of utility rate structures is becoming even more acute with the deregulation of the electric industry. However, with new electric utility rate structures (such as real-time-pricing rates), there can be more opportunities to reduce energy cost in buildings.

At the beginning of this chapter, the primary energy sources consumed in the United States are described. The presentation emphasizes energy use and price by end-use sectors including residential, commercial, and industrial applications. In the United States, buildings and industrial facilities are responsible for 36 and 38 percent, respectively, of the total energy consumption. The transportation sector, which accounts for the remaining 26 percent of the total U.S. energy consumption, uses mostly fuel products. However, buildings and industries predominantly consume electricity and natural gas. Coal is primarily used as an energy source for electricity generation due its low price.

At the end of this chapter, the various features of utility rate structures available in the United States are outlined. More emphasis is given to the electrical rate structures because a significant part of the total energy cost in a typical facility is attributed to electricity. The price rate structures of other energy sources are discussed. The information provided in this chapter is based on recent surveys of existing utility rate structures. However, the auditor should be aware that most utilities revise their rates on a regular basis. If detailed information is required on the rates available from a specific utility, the auditor should contact the utility directly.

2.2 Energy Resources

The sources of energy used in the United States include: coal, natural gas, petroleum products, and electricity. The electricity can be generated from either power plants fueled from primary energy sources (i.e., coal, natural gas, or fuel oil) or from nuclear power plants or renewable energy sources (such as hydroelectric, geothermal, biomass, wind, photovoltaic, and solar thermal sources).

In the United States, energy consumption has fluctuated in response to significant changes in oil prices, economic growth rates, and environmental concerns especially since the oil crisis of the early

1970s. For instance, U.S. energy consumption increased from 66 quadrillion British thermal units (Btu) in 1970 to 99 quadrillion Btu in 2008 (EIA, 2009). Table 2.1 summarizes the changes in U.S. energy consumption by source from 1972 to 2008.

It is clear from the data summarized in Table 2.1 that the consumption of coal has increased significantly from 12 quadrillion Btu in 1972 to 22.5 quadrillion Btu in 2008. However, the U.S. consumption of natural gas actually declined from 22.5 quadrillion Btu in 1972 to 20.7 quadrillion in 1998 before increasingly slightly to 23.8 quadrillion Btu in 2008. This decline in natural gas consumption is due to uncertainties about supply and regulatory restrictions especially in the 1980s. Between 1972 and 2008, consumption of other energy sources generally increased. The increase is from 33.0 quadrillion Btu to 37.1 quadrillion Btu for petroleum products, from 0.6 quadrillion Btu to 8.5 quadrillion Btu for nuclear power, and from 4.5 quadrillion Btu to 7.3 quadrillion Btu for renewable energy which consists almost exclusively of hydroelectric power.

Table 2.2 provides the average nominal energy prices for each primary fuel type. Over the years, coal remains the cheapest energy source. The cost of electricity is still high relative to the other fuel types. As illustrated in Table 2.2, the prices of all energy sources have increased significantly after the energy crisis of 1973. In particular, the cost of petroleum products has increased severalfold over the last few years.

2.2.1 Electricity

2.2.1.1 Overall Consumption and Price

In the United States, coal is the fuel of choice for most existing electrical power plants as shown in Table 2.3. However, gas-fired power plants are expected to be more common in the future due to more efficient and reliable combustion turbines.

TABLE 2.1 Annual U.S. Energy Consumption by Primary Energy Sources in Quadrillion Btu

Primary Energy Source	1972	1982	1992	2002	2008[a]
Coal	12.077	15.322	19.187	21.965	22.462
Natural gas	22.469	18.505	20.714	23.558	23.838
Petroleum products	32.947	30.232	33.527	38.809	37.137
Nuclear power	0.584	3.131	6.479	7.959	8.455
Renewable energy	4.478	6.293	6.707	5.894	7.300
Total	72.758	73.442	85.559	97.858	99.304

Source: EIA Annual Energy Review, Department of Energy, Energy Information Administration, http://www.doe.eia.gov, 2009.

[a] The data for 2008 are preliminary data that may be revised.

TABLE 2.2 Consumer Price Estimates for Energy in Nominal Values in U.S. $/Million Btu

Primary Energy Source	1972	1982	1992	2002	2006
Coal	0.45	1.73	1.45	1.30	1.78
Natural Gas	—	4.23	3.83	5.27	9.62
Petroleum Products	1.78	8.35	7.07	8.82	17.89
Electricity	5.54	18.16	20.06	21.15	26.15

Source: EIA Annual Energy Review, Department of Energy, Energy Information Administration, http://www.doe.eia.gov, 2009.

The electricity sold by U.S. utilities has increased steadily for both the residential and commercial sectors as indicated by the data summarized in Table 2.4. The increase in electricity consumption could be even higher without the various energy conservation programs implemented by the federal or state governments and utilities. For instance, it is estimated that the demand-side-management (DSM) programs provided by utilities have saved about 54 billion kWh in electrical energy use during 2002 and over 69 billion kWh in 2007 (EIA, 2009).

The prices of electricity for all end-use sectors have remained stable between 1992 and 2008 after a recovery period from the 1973 energy crisis as illustrated in Table 2.5. As expected, industrial customers enjoyed the lowest electricity price over the years. Meanwhile, the cost of electricity for residential customers remained the highest.

2.2.1.2 Future of U.S. Electricity Generation

Currently, the electricity market is in the midst of a restructuring period and is becoming increasingly competitive. Several innovative technologies are being considered and tested to generate electricity. A

TABLE 2.3 Annual U.S. Electrical Energy Generated by Utilities by Primary Energy Sources in Billion kWh

Primary Energy Source	1972	1982	1992	2002	2008[a]
Coal	771	1,192	1,576	1,933	1994
Natural Gas	376	305	264	691	877
Petroleum Products	274	147	89	95	45
Nuclear Power	54	283	619	780	806
Renewable Energy	274	314	254	343	372
Total	1,749	2,241	2,797	3,859	4,110

Source: EIA Annual Energy Review, Department of Energy, Energy Information Administration, http://www.doe.eia.gov, 2009.
[a] The data for 2008 are preliminary data that may be revised.

TABLE 2.4 Annual U.S. Electrical Energy Sold by Utilities by Sector in Billion kWh

End-Use Sector	1972	1982	1992	2002	2008[a]
Residential	539	730	936	1,265	1,379
Commercial	359	526	850	1,205	1,352
Industrial	641	745	973	990	982

Source: EIA Annual Energy Review, Department of Energy, Energy Information Administration, http://www.doe.eia.gov, 2009.
[a] The data for 2008 are preliminary data that may be revised.

TABLE 2.5 Average Retail Prices of Electric Energy Sold by U.S. Utilities by Sector in 2000 Cents per kWh

End-Use Sector	1972	1982	1992	2002	2008[a]
Residential	7.9	11.0	9.5	8.1	9.3
Commercial	7.5	11.0	8.9	7.6	8.4
Industrial	4.1	8.0	5.6	4.7	5.7

Source: EIA Annual Energy Review, Department of Energy, Energy Information Administration, http://www.doe.eia.gov, 2009.
[a] The data for 2008 are preliminary data that may be revised.

relatively recent approach to producing electricity using small and modular generators is the distributed generation concept. The small generators with capacities in the range of 1 kW to 10 MW can be assembled and relocated in strategic locations (typically near customer sites) to improve power quality and reliability, and provide flexibility to meet a wide range of customer and distribution system needs.

A number of technologies have emerged in the last decade that allow the generation of electricity with reduced waste, cost, and environmental impact. It is expected that these emerging technologies will improve the viability of distribution generation in a competitive deregulated market. Among these technologies are fuel cells, microturbines, combustion turbines, gas engines, and diesel engines. Chapter 13 discusses some of the emerging technologies in electricity generation.

2.2.1.3 Utility Deregulation Impact

Following the acceptance of the Energy Policy Act of 1992, which requires open access to utility transmission lines, the U.S. Federal Energy Regulatory Commission (FERC) issued orders to allow the establishment of a wholesale power market with independent system operators. As the result of a significant increase in the quantities of bulk power sales, delivery of energy to users has become increasingly difficult especially through the existing transmission and distribution networks. The frequent power outages experienced in the last few years, especially in the western United States, illustrate the precarious stability of the transmission system.

Moreover, several states have started to implement retail access, which allows customers to choose among several electric service providers based on a competitive market that may offer a variety of customized services such as a premium power quality. Unfortunately, existing distribution networks are not designed to support multiple suppliers or to channel value-added services. Ironically, utilities have built less than half the transmission capacity between 1990 and 1995 than they built in the previous five years (1985–1990). This reduction of investment in the transmission grid is largely due to the uncertainty about ongoing electric utility deregulation and restructuring (EPRI, 1999).

In addition to adding new transmission capacity, it is believed that existing transmission and distribution networks and their control have to be upgraded using advanced and new technologies to ensure high reliability and safety of the power delivery system. Among the technologies that are being considered to upgrade the power delivery system are the following:

- The discovery in 1986 of high-temperature superconducting (HTS) materials using ceramic oxides has lowered the cost of superconducting transmission cables to a reasonable level. It is estimated that an HTS cable could carry 500 MW of electric power at voltages as low as 50 kV.
- High-voltage electronic flexible AC transmission system (FACTS) controllers are now used by several utilities to increase the capacity of transmission lines and improve overall delivery system reliability. Unlike conventional electromechanical controllers, FACTS controllers are sufficiently fast to reduce bottlenecks and transient disturbances in power flow and thus reduce transmission system congestion and improve overall delivery reliability.
- Cost-effective distributed generation and storage technologies offer flexibility to meet a wide variety of customer needs. Among distribution generation systems under development and testing are microturbines with capacities ranging from 10 to 250 kW, and fuel cells that offer clean, efficient, compact, and modular generation units.
- Diversified and integrated utility services meet the divergent needs of various market segments. For instance, innovative rate structures such as real-time-pricing rates are being offered to customers that are demanding lower rates. Moreover, some utilities are integrating electricity with other services such as Internet access, telecommunications, and cable television using fiber-optic networks. However, the move to integrate utility functions requires new hardware and software

technologies. For instance, low-cost electronic meters with two-way communications are needed to provide real-time-pricing and billing options for multiple utility services.

2.2.2 Natural Gas

As indicated in Table 2.6, the total U.S. consumption of natural gas actually declined between 1972 and 2008. The industrial sector experienced the highest reduction in natural gas use especially in the 1980s. The main reason for the decline in natural gas use is attributed to the restructuring and deregulation of several segments of the gas industry during most of the 1970s. Indeed, the regulation of natural gas markets had the effect of reducing the availability of natural gas. As indicated in Table 2.7, the prices of natural gas increased significantly between 2002 and 2008. During the 1990s, the prices of natural gas actually decreased because gas supplies became more certain and some of the regulations were removed.

In the future, it is expected that the natural gas market will continue to expand and its pricing to be competitive. In particular, the future for natural gas as a primary energy source for electricity generation is considered to be promising. Indeed, gas-fired power plants are competitive because of their high efficiencies (approaching 50 percent) and are environmentally attractive because they produce significantly lower carbon and sulfur emissions than plants powered by coal or oil.

2.2.3 Petroleum Products

Overall, the U.S. consumption of fuel oil and other petroleum products has remained stable between 1972 and 2008 as indicated in Table 2.8. However, oil prices fluctuated significantly over the last three decades after the 1973 energy crisis. In the building sector (i.e., residential and commercial applications), the U.S. consumption level of petroleum products has decreased over the years. However, the use of petroleum fuels has steadily increased in the transportation sector.

Table 2.9 clearly indicates that after a drastic increase (almost fourfold) between 1972 and 1982, crude oil prices decreased in 1998 to levels even lower than those experienced in 1972. However, another significant increase in oil prices occurred between 2002 and 2008.

TABLE 2.6 Annual U.S. Consumption of Natural Gas by Sector in Trillion Cubic Feet

End-Use Sector	1972	1982	1992	2002	2008[a]
Residential	5.13	4.63	4.69	4.90	4.87
Commercial	2.61	2.61	2.80	3.14	3.12
Industrial	9.62	6.94	8.70	8.62	7.94

Source: EIA Annual Energy Review, Department of Energy, Energy Information Administration, http://www.doe.eia.gov, 2009.

[a] The data for 2008 are preliminary data that may be revised.

TABLE 2.7 Average Retail Prices of Natural Gas by Sector in 2000 Dollars per 1,000 Cubic Feet

End-Use Sector	1972	1982	1992	2002	2008[a]
Residential	4.01	8.24	6.82	7.57	11.17
Commercial	2.92	7.68	5.65	6.36	9.79
Industrial	1.49	6.17	3.29	3.86	7.85

Source: EIA Annual Energy Review, Department of Energy, Energy Information Administration, http://www.doe.eia.gov, 2009.

[a] The data for 2008 are preliminary data that may be revised.

2.2.4 Coal

In the United States, coal is primarily used as an energy source for power generation by electric utilities as shown in Table 2.10. Indeed, the total U.S. consumption of coal increased between 1972 and 2008 due primarily to the growth in coal use by electric utilities. In all other sectors (i.e., residential, commercial, and industrial), coal consumption has decreased. These consumption trends are expected to be maintained in the near future for all sectors. However, the share of electricity generation attributed to coal will be reduced due to more reliance in the future on other generation technologies as discussed in Section 2.2.1.

The abundant coal reserve base and the lingering excess production capacity have helped maintain low coal prices especially during the last decade as indicated by Table 2.11. In the future, however, the price of coal is expected to rise slowly due to reserve depletion and slow growth in labor productivity. The higher coal prices coupled with environmental concerns may cause a future decline of coal consumption in the United States.

TABLE 2.8 U.S. Consumption of Petroleum Products by Sector in Million Barrels per Day

End-Use Sector	1972	1982	1992	2002	2008[a]
Residential/commercial	2.25	1.24	1.19	1.20	0.98
Industrial	4.19	4.06	4.52	4.93	4.58
Transportation	8.57	9.31	10.88	13.21	13.65
Electric utilities	1.36	0.69	0.43	0.43	0.21
Total	16.37	15.30	17.03	19.77	19.42

Source: EIA Annual Energy Review, Department of Energy, Energy Information Administration, http://www.doe.eia.gov, 2009.

[a] The data for 2008 are preliminary data that may be revised.

TABLE 2.9 Average Crude Oil Price in the United States in 2000 Dollars per Barrel

Year	1972	1982	1992	2002	2008[a]
Price	11.24	45.47	18.51	21.61	76.82

Source: EIA Annual Energy Review, Department of Energy, Energy Information Administration, http://www.doe.eia.gov, 2009.

[a] The data for 2008 are preliminary data that may be revised.

TABLE 2.10 Annual U.S. Consumption of Coal by Sector in Million of Short Tons

End-Use Sector	1972	1982	1992	2002	2008[a]
Residential/commercial	11.7	8.2	6.2	4.4	3.6
Industrial	160.1	103.0	106.4	84.4	76.6
Electric utilities	351.8	593.7	795.1	977.5	1,041.6
Total	524.3	706.9	907.7	1,066.4	1,121.7

Source: EIA Annual Energy Review, Department of Energy, Energy Information Administration, http://www.doe.eia.gov, 2009.

[a] The data for 2008 are preliminary data that may be revised.

TABLE 2.11 Average Coal Price in the United States in 2000 Dollars per Short Ton

Year	1972	1982	1992	2002	2008[a]
Price	25.59	43.44	24.34	17.26	26.62

Source: EIA Annual Energy Review, Department of Energy, Energy Information Administration, http://www.doe.eia.gov, 2009.

[a] The data for 2008 are preliminary data that may be revised.

2.3 Electricity Rates

To generate electricity, utilities have to consider several operating costs to determine their rates. Typically, an electric utility is faced with the following cost items:

- *Generation plant:* The cost of operating the power plant to generate electricity typically represents the highest cost category. Indeed, the power generation plants have to meet several regulations and safety requirements. For instance, several plants are required to meet strict pollution standards especially if they operate in highly populated areas. Meeting these regulations can significantly increase the generating costs.
- *Transmission/distribution systems:* To deliver the electricity from the generation plant where it is produced to areas where it is utilized, transmission lines, substations, and distribution networks have to be used. The cost of the transmission/distribution systems depends on the distances to be covered as well as transformers, capacitors, and meters to be used. Moreover, the delivery energy losses can be a significant part of the transmission/distribution costs.
- *Fuel costs:* The electricity is generated using a primary fuel source depending on the power plant. The fuel cost can be small as in the case of hydroelectric plants or significant as in conventional fuel oil or coal power plants. The cost of fuel may fluctuate depending on world markets.
- *Administrative costs:* The salaries of management, technical, and office staff as well as insurance and maintenance costs for power plant equipment are part of the administrative costs. These administrative costs can be large especially for nuclear power plants.

Other factors that affect the cost of electricity include the generating capacity of the utility, and the demand/supply conditions at a given time (i.e., on-peak and off-peak periods).

Utilities allocate the cost of electricity differently by offering various rate schedules depending on the type of customer. Three customer types are generally considered by utilities: residential, commercial, and industrial. Each utility may offer several rate structures for each customer type. It is therefore important that the auditor know the various rate structures that can be offered to the audited facility.

Utilities can tailor their rates to customer needs for electricity using several methods. Some of the common rate structures used by U.S. utilities are summarized below:

- Block pricing rates
- Seasonal pricing rates
- Innovative rates

In addition to these rates, utilities provide some riders and discounts to their customers. A rider may modify the structure of a rate based on specific qualifications of the customer. For instance, utilities can change the summer energy charges of their residential customers using an air-conditioning rider. Moreover, utilities can reduce energy or demand charges using discount provisions. For instance, several utilities offer a voltage discount when the customer is willing to receive voltages higher than the standard voltage level. To benefit from this discount, the customer may have to install and maintain a properly sized transformer.

More detailed discussion of each of the commonly available rates is provided in the following sections with some examples to illustrate how an energy auditor can apply the utility rate provisions to check the calculation of utility bills. First, common features of all electric utility rates are discussed in detail.

2.3.1 Common Features of Utility Rates

There are several utility rate features and concepts that the auditor should be familiar with to be able to interpret and analyze the utility billing procedure correctly. Some of these concepts are described in the following sections.

2.3.1.1 Billing Demand

The demand that is billed by the utility is referred to as the billing demand. The billing demand is often determined from the peak demand obtained for one month (or any billing cycle). The peak demand, also known as actual demand, is defined as the maximum demand or maximum average measured demand in any 15-minute period in the billing cycle. To better understand the concept of billing demand, consider two different monthly load profiles: a rugged profile A and a flat profile B as illustrated in Figure 2.1. It is further assumed that the average demand for profile A coincides with that of profile B (and thus the total energy use in kWh for both profiles A and B is the same). It is not equitable for the utility to bill the same charges for the two profiles. Indeed, profile A requires that the utility supply higher demand and thus increase its generation capacity for only a short period of time. Meanwhile, profile B is ideal for the utility because it does not change over time. Therefore, some utilities charge their customers for the peak demand incurred during the billing period. This demand charge may serve as an incentive to the customers to reduce or "shave" their peak demand.

In some rates, the billing demand is determined based on the utility specifications and may be different from the maximum demand actually measured during a billing period. For instance, ratchet and power factor clauses can change the determination of the billing demand. Moreover, a minimum billing demand can be specified in a contractual agreement between the utility and the customer. This minimum demand is often called the contract demand.

2.3.1.2 Power Factor Clause

The power factor is defined as the ratio of actual power used by the consumer (expressed in kW) to the total power supplied by the utility (expressed in kVA). Figure 2.2 shows the power triangle to illustrate the power factor concept. The reader is referred to Chapter 5 for more details on this concept.

For the same actual power consumed by two customers but with different power factor values, the utility has to supply higher total power to the customer with the lower power factor. To penalize customers for low power factors (generally lower than 0.85), some utilities use a power factor clause to change the

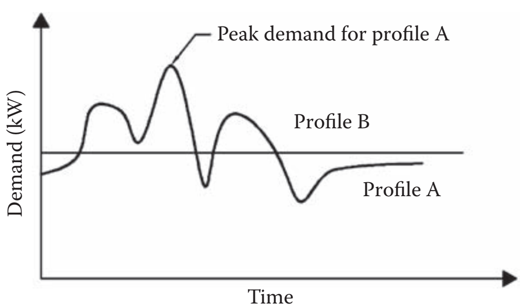


FIGURE 2.1 Peak demand for two different electric load profiles.

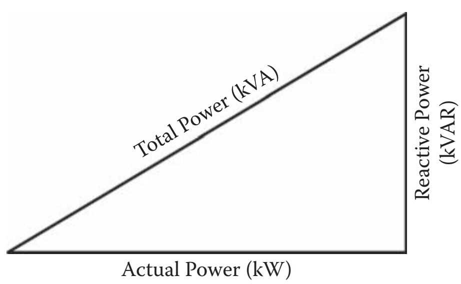


FIGURE 2.2 Power triangle diagram.

billing demand or to impose new charges (on total power demand or reactive power demand) depending on the power factor. For instance, the billing demand can be increased whenever the measured power factor of the customer is below a reference value or base power factor:

$$Billed-Demand = Actual-Demand * \left(\frac{pf_{base}}{pf_{actual}}\right) \tag{2.1}$$

$$PowerFactor(pf) = \frac{ActualPower}{TotalPower}$$

Example 2.1 illustrates how the power factor clause may affect the utility bill.

Example 2.1

A utility has the following monthly billing structure for its industrial customers:

(a) Customer charge = \$450/month
(b) Demand charge = \$20/kW
(c) Energy charge = \$0.03/kWh

A power factor clause states that the billing demand, upon which the demand charge is based, shall be the actual demand for the month corrected for the power factor by multiplying the measured actual demand by 80 and dividing the product by the actual average power factor expressed in percent. The power factor clause applies whenever the average monthly power factor is less than 80 percent.

(i) Calculate the utility bill for an industrial facility with the following energy use characteristics during a specific month:
- Actual demand: 300 kW
- Energy consumption: 50,000 kWh
- Average monthly power factor: 60 percent

(ii) Determine the cost savings achieved for the month if the facility power factor is improved to be always above 80 percent.

Solution

(i) First, the billing demand for the month is determined using the power factor clause because the average power factor is below 80 percent:

Billed demand = Actual demand * 80/pf = 300 * 80/60 = 400 kW

Then, the monthly bill can be calculated using the utility rate structure:
(a) Customer charge = \$ 450.00
(b) Demand charge = 400 kW * \$20/kW = \$ 8,000.00
(c) Energy charge = 50,000 kWh * \$0.03/Wh = \$ 1,500.00

Total monthly charges = (a) + (b) + (c) = \$ 9,950.00

The actual bill may include other components such as fuel adjustment cost and sales taxes.

(ii) The cost savings from improved power factor are due to a reduction in the billing demand. Indeed, the billing demand for the considered month will be the actual demand (i.e., 300 kW). Thus, a reduction of 100 kW in billing demand which results in a reduction of demand charges of:

$$\text{Cost Savings} = 100 \text{ kW} * \$20/\text{kW} = \$ 2{,}000/\text{month}$$

The reader is referred to Chapter 5 to determine the cost-effectiveness for power factor improvement measures.

2.3.1.3 Ratchet Clause

Typically, the utility charges are billed monthly. In particular, the demand charges are based on monthly peak demand. However, when the peak demand for one month is significantly higher than for the other months (such as the case for buildings with high cooling loads in the summer months), the utility has to supply the required peak demand and thus operate additional generators only for one or two months. For the rest of the year, the utility will have to maintain these additional generators. To recover some of this maintenance cost and to encourage demand shaving, some utilities use a ratchet clause in the determination of the billed demand. For instance, the billed demand for any given month is a fraction of the highest maximum demand of the previous 6 months (or 12 months) or the actual demand incurred in the month. Example 2.2 illustrates the calculation procedure of the utility bill with a ratchet clause.

Example 2.2

The utility has added a ratchet clause to the rate structure described in Example 2.1. Specifically, the ratchet clause states that no billing demand shall be considered as less than the 70 percent of the highest on-peak season maximum demand corrected for the power factor previously determined during the 12 months ending with the current month.

Calculate the utility bill for the industrial facility considered in Example 2.1 taking into account the ratchet clause. Assume that the previous highest demand (during the last 12 months) is 700 kW.

Solution

First, the minimum billing demand determined by the ratchet clause is calculated:

$$\text{Minimum billing demand (ratchet clause)} = 700 \text{ kW} * 0.70 = 490 \text{ kW}$$

Because the billing demand based on the power factor clause was found to be 400 kW (see Example 2.1), the billing demand for the month is:

$$\text{Billed demand} = \max (490 \text{ kW}, 400 \text{ kW}) = 490 \text{ kW}$$

Then, the monthly bill can be calculated using the utility rate structure:

(a) Customer charge = $ 450.00
(b) Demand charge = 490 kW * $20/kW = $9,800.00
(c) Energy charge = 50,000 kWh * $0.03/Wh = $1,500.00

Total monthly charges = (a) + (b) + (c) = $ 11,750.00

Thus, the ratchet clause increases the utility bill for the month.

2.3.1.4 Fuel Cost Adjustment

Most utilities have to purchase primary energy sources (fuel oil, natural gas, and coal) to generate electricity. Inasmuch as the cost of these commodities changes over time, the utilities impose an adjustment to their energy charges to account for any cost variation of their primary energy sources. Generally, utilities provide in the description of their rate structure a formula that they use to calculate the fuel cost adjustment. In addition to fuel cost adjustment, utilities may levy taxes and surcharges to recover imposts required from them by federal or state governments or agencies. In some cases, the fuel cost adjustment can be a significant proportion of the utility bill. Example 2.3 illustrates the effect of both the fuel cost adjustment and sales tax on a monthly utility bill.

Example 2.3

The utility has added a ratchet clause to the rate structure described in Example 2.2. Specifically, the ratchet clause states that no billing demand shall be considered as less than 70 percent of the highest on-peak season maximum demand corrected for the power factor previously determined during the 12 months ending with the current month.

Calculate the utility bill for the industrial facility considered in Example 2.2 taking into account both a fuel cost adjustment of 0.015/kWh and a sales tax of 7 percent.

Solution

The fuel cost adjustment should be applied to the energy use, and the sales tax should be applied to the total cost. Thus, the monthly bill for the industrial facility considered in Example 2.2 becomes:

(a) Customer charge = $ 450.00
(b) Demand charge = 490 kW * $20/kW = $9,800.00
(c) Energy charge = 50,000 kWh * [$0.03/kWh + $0.015/kW] = $2,250.00

Total monthly charges (before sales tax) = (a) + (b) + (c) = $12,500.00

Total monthly charges (after sales tax) = $12,500.00 * 1.07 = $13,375.00

The cumulative effect of both fuel cost adjustment and sales tax increased the utility bill for the month by [$13,375 – $11,750 = $1,625].

2.3.1.5 Service Level

Utilities typically offer several rate structures for a given customer depending on the type of service. For instance, utilities may have different rates depending on the voltage level provided to the customers. The higher the delivery voltage level, the cheaper the energy rate is. In particular, utilities offer reduced rates for demand or energy charges to customers that own their service transformers. This type of rate is generally described under various clauses of a utility bill including customer-owned transformer, voltage level, or service type. The owner has to determine the cost-effectiveness of owning a transformer and thus receiving a higher voltage level by comparing the utility bill savings against the cost of purchasing, leasing, and maintaining the transformer. Typically, transformer energy losses are charged to the owner. In addition, the owner should have a standby transformer or make another arrangement (such as standby utility service) in case of breakdown of the service transformer. Example 2.4 presents an estimation of savings in utility bills due to owning a service transformer.

Example 2.4

The industrial facility of Example 2.1 has an option of owning and operating its own service transformer with the advantage of reduced rate structure as described below:

- Customer charge = \$650/month
- Demand charge = \$15/kW
- Energy charge = \$0.025/kWh

Calculate the reduction in the utility bill during the month for which the energy use characteristics are given in Example 2.1. In the savings calculation, do not consider the power factor clause, ratchet clause, sales tax, or fuel cost adjustment.

Solution

First, the utility bill is determined in the case where the facility owns its service transformer:

(a) Customer charge = \$ 650.00
(b) Demand charge = 400 kW * \$15/kW = \$ 6,000.00
(c) Energy charge = 50,000 kWh * \$0.025/Wh = \$ 1,250.00

$$\text{Total monthly charges} = (a) + (b) + (c) = \$\ 7{,}900.00$$

Thus, the savings in the utility bill for the month is

$$\text{Utility bill savings} = \$9{,}950 - \$7{,}900 = \$2{,}050$$

The reader is referred to Chapter 5 to determine the cost-effectiveness of replacing or owning transformers.

2.3.2 Block Pricing Rates

In these rates, the energy price depends on the rate of electricity consumption using either inverted or descending blocks. An inverted block pricing rate structure increases the energy price as the consumption increases. On the other hand, a descending (also referred to as a declining) block rate structure reduces the price as the energy consumption increases. Typically, the rate is referred to as a "flat" rate when the energy price does not vary with the consumption level. Tables 2.12 and 2.13 show examples of block pricing rates for energy and demand charges. Examples 2.5 and 2.6 illustrate the calculation details of utility bills.

Based on a comprehensive survey of existing rate structures offered by U.S. utilities (GRI, 1993), block pricing rate structures are commonly used. Indeed, almost 60 percent of the surveyed electric utilities offer descending or inverted block pricing structure for the energy charges. For residential customers, a combination of descending, flat, and inverted rate structures is used throughout the United States. However, descending energy rate structures are used almost exclusively for commercial and industrial customers. Some electric utilities offer block pricing rates for demand as well as energy charges.

TABLE 2.12 An Electric Utility Rate with No-Demand Charges for Residential General Service (Utility Rate A)

Billing Item	Winter (Nov–Apr)	Summer (May–Oct)
Customer charge	$7.50	$7.50
Minimum charge	$7.50	$7.50
Fuel cost adjustment	0.00	0.00
Tax rate	6.544%	6.544%
No. of energy blocks	1	3
Block 1 energy size (kWh)	0	400
Block 2 energy size (kWh)	0	400
Block 3 energy size (kWh)	0	800
Block 1 energy charge ($/kWh)	0.088	0.0874
Block 2 energy charge ($/kWh)	0.000	0.1209
Block 3 energy charge ($/kWh)	0.000	0.1406

TABLE 2.13 An Electric Utility Rate with Demand Charges for Commercial General Service (Utility Rate B)

Billing Item	Winter (Oct–May)	Summer (Jun–Sep)
Customer charge	0.00	0.00
Minimum charge	$25.00	$25.0
Fuel cost adjustment	0.01605	0.01377
Tax rate	0.00	0.00
No. of energy blocks	3	3
Block 1 energy size (kWh)	40,000	40,000
Block 2 energy size (kWh)	60,000	60,000
Block 3 energy size (kWh)	>100,000	>100,000
Block 1 energy charge ($/kWh)	0.059	0.065
Block 2 energy charge ($/kWh)	0.042	0.047
Block 3 energy charge ($/kWh)	0.039	0.042
No. of demand blocks	2	2
Block 1 demand size (kW)	50	50
Block 2 demand size (kW)	>50	>50
Block 1 demand charge ($/kW)	12.39	13.71
Block 2 demand charge ($/kW)	11.29	12.52
Reactive demand charge ($/kVAR)	0.20	0.20

Example 2.5

Using utility rate structure A, calculate the utility bill for a residence during a summer month when the energy use is 800 kWh.

Solution

Considering all the charges imposed by utility rate A (see Table 2.12), the monthly bill can be calculated as follows:

(a) Customer charge = $ 7.50
(b) Energy charge = 400kWh * $0.0874/kWh + 400kWh * $0.1209/kWh = $83.32
(c) Fuel cost adjustment = $ 0.00
(d) Taxes = 6.544 percent [(a) + (b) + (c)] = $ 5.94

Total monthly charges = (a) + (b) + (c) + (d) = $96.76

Thus, the average cost of electricity for the month is $96.76/800 kWh = $0.12095/kWh.

2.3.3 Seasonal Pricing Rates

Some electric utilities offer seasonal rate structures to reflect the monthly variations in their generation capacity and energy cost differences. Generally, the utilities that provide seasonal rate structures use different energy or demand charges during winter and summer months. The summer charges are typically higher than winter charges for most electric utilities due to higher energy consumption attributed to cooling of buildings. Tables 2.14 and 2.15 show examples of seasonal pricing rates and Examples 2.5 and 2.6 illustrate the calculation procedure for monthly utility bills.

Based on a survey conducted by GRI (1993), over 55 percent of U.S. electric utilities offer seasonal pricing rates for residential customers. Only 5 percent of electric utilities have residential rates where the winter rate is actually higher than the summer rate. These utilities are located in the Northeast and the West regions of the United States. The same survey reveals that over 42 percent of U.S. utilities use seasonal pricing rates for commercial and industrial customers. Only 7 percent of the utilities surveyed offer rates with higher winter prices for their commercial and industrial customers.

TABLE 2.14 An Electric Time-of-Use Rate for Residential Service (Utility Rate C)

Billing Item	Winter (Oct–May)	Summer (Jun–Sep)
Customer charge	$9.85	$9.85
Minimum charge	$9.85	$9.85
Tax rate	3.00%	3.00%
Energy charge ($/kWh)		
On-peak hours:	0.1412	0.1500
Off-peak hours	0.0335	0.0335

Notes: On-Peak Hours are defined as:
6:00 a.m.–1:00 p.m. and 4:00 p.m.–9:00 p.m. (M–F) during Oct–Mar
10:00 a.m.–9:00 p.m. (M–F) during Apr–Sep
Off-Peak Hours are all the remaining hours including holidays.

TABLE 2.15 Day-Ahead-Price Rates from OG&E Utility for Typical Curtailment and Noncurtailment Days

Hour	Noncurtailment Rate Price ($/kWh)	Curtailment Rate Price ($/kWh)
0:00–1:00	0.01673	0.01673
1:00–2:00	0.01683	0.01683
2:00–3:00	0.01730	0.01730
3:00–4:00	0.01780	0.01780
4:00–5:00	0.01900	0.01900
5:00–6:00	0.02130	0.02130
6:00–7:00	0.02730	0.02730
7:00–8:00	0.02890	0.02890
8:00–9:00	0.02932	0.02932
9:00–10:00	0.03213	0.03213
10:00–11:00	0.03575	0.03575
11:00–12:00	0.03423	0.03423
12:00–13:00	0.03456	0.03456
13:00–14:00	0.03677	0.03677
14:00–15:00	0.03977	0.03977
15:00–16:00	0.04254	1.06350
16:00–17:00	0.04354	1.08950
17:00–18:00	0.03595	0.89875
18:00–19:00	0.02587	0.64675
19:00–20:00	0.01950	0.48750
20:00–21:00	0.01523	0.01523
21:00–22:00	0.01532	0.01532
22:00–23:00	0.01723	0.01723
23:00–24:00	0.01722	0.01722

Example 2.6

Using utility rate structure B, calculate the utility bill for a commercial facility during a winter month when the energy use is 70,000 kWh, the billing demand is 400 kW, and the average reactive demand is 150 kVAR.

Solution

Accounting for all the charges considered by utility rate B (see Table 2.13), the electric energy bill for the winter month can be calculated as follows:

(a) Customer charge = $ 0.00
(b) Energy charge = 40,000 kWh * $0.059/kWh + 30,000 kWh * $0.042/kWh = $3,620.00
(c) Demand charge = 50 kW * $12.39/kW + 350 kW * $11.29/kW = $4,571.00
(d) Fuel cost adjustments = 70,000 kWh * $0.01605 = $1,123.50
(e) Reactive demand charge = 150 kVAR * $0.20/kVAR = $ 30.00
(f) Taxes = $ 0.00

Total monthly charges = (a) + (b) + (c) + (d) + (e) + (f) = $9,344.50

The average cost of electricity for the month is then $9,344.50/70,000 kWh or $.1335/kWh.

2.3.4 Innovative Rates

Due to the increased focus on integrated resource planning, demand-side management, and the competitive energy market, several innovative rates have been implemented by utilities. These innovative rates have the main objective to profitably meet customer needs. Moreover, some utilities foster new technologies through the use of innovative rates to retain their customers.

There are several categories of rates that can be considered to be innovative rates. In the United States, innovative rates can be classified into seven categories (GRI, 1993).

2.3.4.1 Time-of-Use (TOU) Rates

The time-of-use (TOU) rates are time-differentiated rates where the cost of electricity varies during specific times of the day or year. The TOU rates, which first appeared in the 1940s, set "on-peak" and "off-peak" periods with different energy or demand charges. Generally, the on-peak periods occur during daytime hours and have a higher cost of energy and demand than the off-peak periods that occur during the night-time. Table 2.14 presents a time-of-use rate structure for residential customers. Example 2.7 illustrates the calculation approach of a utility bill for the TOU rate presented in Table 2.14.

Example 2.7

Using utility rate structure C, calculate the utility bill for a residence during a summer month when the energy use is 2,000 kWh with 55 percent of the energy consumption occurring during the on-peak period.

Solution

With all the charges considered by utility rate C (see Table 2.14), the electric energy bill for the summer month can be calculated as follows:

(a) Customer charge = $ 9.85
(b) Energy charge = 2,000kWh * [0.55 * $0.1500/kWh + 0.45 * $0.0335/kWh] = $195.15
(c) Taxes = 3 percent * [(a) + (b)] = $ 6.15

Total monthly charges = (a) + (b) + (c) = $ 211.15

The average cost of electricity for the month is then $211.15/2,000 kWh or $0.1056/kWh.

2.3.4.2 Real-Time-Pricing (RTP) Rates

The real-time-pricing (RTP) rates were first offered by Pacific Gas & Electric in 1985 (Mont and Turner, 1999). Like TOU rates, RTP rates are time-differentiated rates but the cost of electricity varies on an hourly basis. Typically, the utilities inform their customers of the hourly electricity prices only a few hours before they take effect. A more detailed description of the RTP rate structures available in the United States is provided later in Section 2.3.5.

2.3.4.3 The End-Use Rates

To encourage customers to install and operate specific energy-consuming equipment, some U.S. utilities offer end-use rates. With these rates, the utilities can impose operation periods or efficiency standards for selected and predefined equipment. For instance, the air-conditioning rate allows electric utilities to interrupt service or cycle off the air-conditioning equipment during specific times. Typically, the end-use rates require separate metering of the equipment.

2.3.4.4 Specialty Rates

The specialty rates are provided by utilities for specific purposes such as energy conservation and dispatchable customer generation. Energy conservation rates are offered by a limited number of U.S. utilities to foster the use of energy-efficient equipment or high standards of building materials. Dispatchable customer generation rates are provided to customers that have standby generators on their premises. In exchange for a reduced rate or a credit, the customers are requested to operate the generators whenever the utility needs additional generating capacity.

2.3.4.5 Financial Incentive Rates

Financial incentive rates encompass economic development rates, displacement rates, and surplus power rates. The economic development rates are typically offered to encourage new customers to locate, or existing customers to expand, in specific areas that need to be economically revitalized. The displacement rates are offered to customers that are capable of generating electricity to entice them to use utility-provided electricity. Finally, the surplus power rates are highly reduced energy rates that are offered to large commercial and industrial customers when the utility has an excess in electric capacity.

2.3.4.6 Nonfirm Rates

The nonfirm rates include interruptible rates, stand-by rates, and load-management rates. Interruptible rates are offered to customers that can reduce or even eliminate (interrupt) their electricity needs from the utility. The electricity pricing rates depend on several factors such as the capacity that can be interrupted, the length of interruption, and the notification before interruption.

Stand-by rates are intended for customers that require utility-provided electricity on an intermittent basis because they are capable of generating most of their electricity needs (by using, for instance, cogeneration systems as discussed in Chapter 13). The stand-by rates can be offered using three options:

(i) *Maintenance rates* when the customers need electricity from the utility during predefined downtime periods of the generating equipment to perform routine maintenance
(ii) *Supplementary rates* for customers that regularly use more electricity than they are capable of generating
(iii) *Back-up rates* intended for customers that prefer to have backup power from the utility in case of unexpected events (i.e., outages in the customers' generators)

Load-management rates are offered by utilities to control the usage of specific equipment such as space-conditioning systems during peak periods. Generally, the load-management rates are offered in combination with TOU rates.

2.3.4.7 Energy Purchase Rates

The energy purchase rates, also known as buy-back rates, are offered by utilities that want to purchase specific levels of energy or generating capacity from customers. The customers are nonutility electricity generators that qualify under the requirements of the Public Utility Regulatory Policies Act (PURPA) such as cogeneration facilities and independent power producers.

2.3.5 Real-Time-Pricing Rates

Based on a survey performed recently by Mont and Turner (1999), most of the existing RTP rate structures in the United States are rather experimental and are restricted to a selected number of customers with large power demands (varying from 250 kW and 10 MW depending on the utility).

The available RTP rate structures can be classified into four categories in the survey as described briefly below:

2.3.5.1 Category 1: Base Bill and Incremental Energy Charge Rates

Under these rates, the customers receive firm hourly electricity prices for the next day before a warning time. Some utilities update their prices daily (day-ahead-pricing) whereas other utilities provide hourly prices only once a week (week-ahead-pricing). Typically, the category 1 RTP rates include the following charges:

- *Base Bill Charge* so that the utility recovers its revenue requirements. This charge depends on the customer baseline load (CBL) which defines the customer's typical energy use (kWh) for each hour of the year and demand values (kW) for each month.
- *Incremental Energy Charge* (or credit) reflects the cost of the energy used by the customer above (or below) its CBL profile.

In this category of RTP, an adequate estimation of the CBL profile is an important factor in determining the final energy bill. Therefore, the customer should negotiate for the most favorable CBL values. In general, the customer would benefit from smaller CBL values because the hourly prices of the RTP rate typically fall below those of the standard rate. Moreover, flexible customers can change their electrical load profile to avoid high costs and even obtain a credit for usage below their CBL.

2.3.5.2 Category 2: Total Energy Charge Rates

The hourly electricity prices for this category are applied to the total energy consumption. Furthermore, the utility imposes an additional charge to recover its revenue requirements. Some utilities set a monthly fixed charge based on the customer's average apparent demand (kVA) and actual demand (kW) determined using the last 12-month load profile. Other utilities charge for actual demand usually ratcheted for the last 12-month period.

The category 2 RTP rates are not suitable for inflexible customers that cannot change their load profile inasmuch as they cannot protect themselves against higher electricity prices.

2.3.5.3 Category 3: Day-Type Rates

These rates are similar to the category 2 RTP rates but with several predefined day types (such as off-peak day, normal day, etc.) associated with different hourly firm electricity prices. These hourly prices are based on the total energy consumption with possibly a charge for actual demand. The utility informs the customer of the applicable day-type rate one day in advance. Again these rates are only suitable for flexible customers that can vary their energy use profile to accommodate the changes in the electricity price.

2.3.5.4 Category 4: Index-Type Rates

The electricity prices for these rates are determined based on the financial market electricity indices (Dow Jones) or the trading prices for electricity future contracts (New York Mercantile Exchange). Generally, the customers have to forecast the electricity prices (using various tools and sources) because the utility does not provide the prices in advance.

2.3.6 Case Study of RTP Rates

In this section, a detailed description is presented to calculate a monthly utility bill using RTP rate structures. The RTP rates used in this section are based on the day-ahead-pricing (DAP) pilot rates proposed by the Oklahoma Gas and Electric (OG&E) utility discussed by Mont and Turner (1999).

In the DAP rates offered by OG&E, there are two types of rates offered under either curtailment or noncurtailment services. Customers with curtailment service are allowed to buy electricity (referred to as "buy-through power" or "emergency power") above the contracted CBL profile during selected hours (or a curtailment period). The total cost of the buy-through power during a curtailment period is known as the "buy-through period." Typically, the utility notifies the customer in advance of the availability and the hourly prices of emergency power. Table 2.15 shows the hourly prices for DAP rates offered by OG&E for both noncurtailment and curtailment days (Mont and Turner, 1999). The curtailment period is set between 3:00 p.m. and 8:00 p.m. with a contracted demand of 500 kW. To illustrate the calculation procedure of utility bills using the RTP rates compared to those based on the conventional rate structures, Example 2.8 is presented using the OG&E DAP rates.

Cost savings in the utility bills based on RTP rate structures can be achieved by shifting the load during the curtailment hours to noncurtailment hours. Example 2.9 illustrates the magnitude of savings achieved by a customer under OG&E DAP rates.

Example 2.8

Calculate the utility bills for an industrial customer serviced by OG&E under two rates:

(i) Conventional rate with the following charges:
Customer charge: $150.00
Energy charge: $0.0264/kWh
Demand charge:$13.1/kW

(ii) RTP rate structure with the hourly prices given in Table 2.15. In addition, a customer charge (referred to as an administrative charge) of $300 is incurred monthly by the customer under the RTP rate.

The hourly energy use and the contracted CBL profiles (for the RTP rate) are summarized in Table 2.16 for typical week and weekend days (data provided by Mont and Turner, 1999).

Assume that the month has 21 weekdays (including two curtailment days) and 9 weekend days. In the utility bill calculations, do not account for any other charges such as taxes and credits.

Solution

(1) First, the utility bill is determined using the conventional rate structure based on the actual energy use (see Table 2.16):
  (a) Customer charge = $ 150.00
  (b) Energy charge = $0.0264/kWh * [21 * 42,461 kWh + 9 * 10,510 kWh] = $26,037.55
  (c) Demand charge = $13.1/kW * 3,015 kW = $39,496.50

Total monthly charges = (a) + (b) + (c) = $65,684.05

The average cost of electricity for the month based on the conventional rate structure is then $65,684.05/ [21 * 42,461 kWh + 9 * 10,510 kWh] or $0.06660/kWh.

(2) The utility bill based on the RTP rate is described below:
  (a) Customer charges = $ 300.00
  (b) Energy charge for week days (noncurtailment days; see Table 2.17) = $19 * 55.99
  (c) Energy charge for week days (curtailment days; see Table 2.18) = $2 * 8,196.79
  (d) Standard charge based on the CBL profile (see note below)
    i. Customer charge = $ 150.00
    ii. Energy charge = $0.0264/kWh * 958,960 (see note) = $26,037.55
    iii. Demand charge = $13.1/kW * 2,765kW = $36,221.50

$$\text{Total monthly charge} = \text{i} + \text{ii} + \text{iii} = \$62{,}409.05$$

$$\text{Total monthly charge} = (a) + (b) + (c) + (d) = \$80{,}166.44$$

The average cost of electricity for the month based on the RTP rate structure is then $80,166.44/(21 * 42,461 kWh + 9 * 10,510 kWh) = $0.08128/kWh.

Therefore, the conventional rate structure is more cost-effective for the industrial facility considered in this example (at least during the month for which the utility bills were calculated as shown above).

The total monthly energy use based on the CBL profiles is calculated as detailed below:

$$\text{kWh(CBL)} = 19 \text{ days} * 42{,}040 \text{ kWh/day} + 2 \text{ days} * 32{,}805 \text{ kWh/day} + 9 \text{ days} * 10{,}510 \text{ kWh/day},$$

or

$$\text{kWh (CBL)} = 968{,}960 \text{ kWh}$$

TABLE 2.16 Actual and CBL Hourly Energy Use Profiles for the Industrial Facility Assumed in Example 2.8

Hour	Typical Weekday Actual Energy Use (kWh)	Typical Weekend Day Actual Energy Use (kWh)	Noncurtailment Typical Day CBL Energy Use (kWh)	Curtailment Typical Day CBL Energy Use (kWh)
0:00–1:00	345	250	220	220
1:00–2:00	455	270	230	230
2:00–3:00	676	305	325	325
3:00–4:00	785	320	360	360
4:00–5:00	980	360	455	455
5:00–6:00	1,200	405	645	645
6:00–7:00	1,800	460	1,715	1,715
7:00–8:00	2,905	510	2,450	2,450
8:00–9:00	2,825	525	2,545	2,545
9:00–10:00	2,800	520	2,480	2,480
10:00–11:00	2,845	535	2,560	2,560
11:00–12:00	3,015	550	2,765	2,765
12:00–13:00	2,500	565	2,745	2,745
13:00–14:00	2,765	560	2,730	2,730
14:00–15:00	2,890	535	2,650	2,650
15:00–16:00	2,815	525	2,590	500
16:00–17:00	2,825	530	2,520	500
17:00–18:00	2,780	510	2,370	500
18:00–19:00	1,755	495	2,175	500
19:00–20:00	1,100	480	2,080	500
20:00–21:00	845	395	1,945	1,945
21:00–22:00	635	345	1,680	1,680
22:00–23:00	535	290	1,230	1,230
23:00–24:00	385	270	575	575
Total	42,461	10,510	42,040	32,805

TABLE 2.17 RTP Hourly Energy Cost During a Typical Noncurtailment Day for the Industrial Facility of Example 2.8

Hour	Typical Weekday Actual Energy Use (kWh)	Noncurtailment Typical Day CBL Energy Use (kWh)	Noncurtailment Typical Day Hourly Energy Price ($/kWh)	Noncurtailment Typical Day Hourly Energy Cost ($)
0:00–1:00	345	220	0.01673	2.09
1:00– 2:00	455	230	0.01683	3.79
2:00–3:00	676	325	0.01730	6.07
3:00–4:00	785	360	0.01780	7.57
4:00–5:00	980	455	0.01900	9.98
5:00– 6:00	1,200	645	0.02130	11.82
6:00–7:00	1,800	1,715	0.02730	2.32
7:00–8:00	2,905	2,450	0.02890	13.15
8:00–9:00	2,825	2,545	0.02932	8.21
9:00–10:00	2,800	2,480	0.03213	10.28
10:00–11:00	2,845	2,560	0.03575	10.19
11:00–12:00	3,015	2,765	0.03423	8.56
12:00–13:00	2,500	2,745	0.03456	–8.47
13:00–14:00	2,765	2,730	0.03677	1.29
14:00–15:00	2,890	2,650	0.03977	9.54
15:00–16:00	2,815	2,590	0.04254	9.57
16:00–17:00	2,825	2,520	0.04354	13.28
17:00–18:00	2,780	2,370	0.03595	14.74
18:00–19:00	1,755	2,175	0.02587	–10.87
19:00–20:00	1,100	2,080	0.01950	–19.11
20:00–21:00	845	1,945	0.01523	–16.75
21:00–22:00	635	1,680	0.01532	–16.01
22:00–23:00	535	1,230	0.01723	–11.97
23:00–24:00	385	575	0.01722	–3.27
Total	42,461	42,040		55.99

EXAMPLE 2.9

Calculate the cost savings in electric utility bills for an industrial customer serviced by the OG&E RTP rate used in Example 2.8 if the excess electrical load experienced during the curtailment hours (between 3 p.m. and 8 p.m.) is shifted uniformly to the next four hours (between 8 p.m. and midnight). With this load-shifting measure, the demand during the curtailment hours would be 500 kW and the total daily energy use for typical days would remain the same as shown in Table 2.18.

Solution

(a) The excess load during the curtailment hours is first determined as follows:

$$\text{Excess Load} = 2{,}815\ \text{kWh} + 2{,}825\ \text{kWh} + 1{,}755\ \text{kWh} + 1{,}100\ \text{kWh} - 4 * 500\ \text{kWh} = 6{,}495\ \text{kWh}$$

(b) For each hour during the next 4 hours (i.e., between 8 p.m. and midnight), the additional load will be 6,495 kWh/4 = 1,623.75 kWh. The additional energy cost during the noncurtailment hours associated with this shift is then:

$$\text{Energy Cost Increase} = 1{,}623.75\ \text{kWh} * [(0.01523 + 0.01532 + 0.01723 + 0.01722)\ \$/\text{kWh}] = \$105.54$$

(c) The shift of electrical load during the curtailment hours has the effect of eliminating any hourly energy cost during the curtailment hours from 3 p.m. to 7 p.m. as shown in Table 2.18.

$$\$2{,}533.09 + \$2{,}049.15 + \$811.67 + \$292.50 = \$5{,}686.41$$

(d) Because all the other charges remain unchanged, the reduction in cost savings is simply the energy charge avoided during the curtailment hours minus any energy cost increase during noncurtailment hours, or:

Utility cost savings = [$5,686.41/day – 105.54/day] * 2 days = $11,161.74 for the month.

TABLE 2.18 RTP Hourly Energy Cost During a Typical Curtailment Day for the Industrial Facility of Example 2.8

Hour	Typical Weekday Actual Energy Use (kWh)	Curtailment Typical Day CBL Energy Use (kWh)	Curtailment Typical Day Hourly Energy Price ($/kWh)	Curtailment Typical Day Hourly Energy Cost ($)
0:00–1:00	345	220	0.01673	2.09
1:00–2:00	455	230	0.01683	3.79
2:00–3:00	676	325	0.01730	6.07
3:00–4:00	785	360	0.01780	7.57
4:00–5:00	980	455	0.01900	9.98
5:00–6:00	1,200	645	0.02130	11.82
6:00–7:00	1,800	1,715	0.02730	2.32
7:00–8:00	2,905	2,450	0.02890	13.15
8:00–9:00	2,825	2,545	0.02932	8.21
9:00–10:00	2,800	2,480	0.03213	10.28
10:00–11:00	2,845	2,560	0.03575	10.19
11:00–12:00	3,015	2,765	0.03423	8.56
12:00–13:00	2,500	2,745	0.03456	–8.47
13:00–14:00	2,765	2,730	0.03677	1.29
14:00–15:00	2,890	2,650	0.03977	9.54
15:00–16:00	2,815	500	1.06350	2,462.00
16:00–17:00	2,825	500	1.08950	2,533.09
17:00–18:00	2,780	500	0.89875	2,049.15
18:00–19:00	1,755	500	0.64675	811.67
19:00–20:00	1,100	500	0.48750	292.50
20:00–21:00	845	1,945	0.01523	–16.75
21:00–22:00	635	1,680	0.01532	–16.01
22:00–23:00	535	1,230	0.01723	–11.97
23:00–24:00	385	575	0.01722	–3.27
Total	42,461	32,805		8,196.79

2.4 Natural Gas Rates

The rate structures for natural gas are similar to those described for electricity. However, the rates are generally easier to understand and apply. For instance, natural gas utilities rarely charge for peak demands. However, energy charges using block rates or seasonal rates are commonly offered. Examples of natural gas utility rates are illustrated in Table 2.19 for energy block pricing rates and Table 2.20 for energy and demand block pricing rates (GRI, 1993). Calculation details of monthly utility bills for natural gas are shown in Examples 2.10 and 2.11.

TABLE 2.19 A Gas Utility Rate for Residential General Service (Utility Rate D)

Billing Item	Any Month (Jan–Dec)
Customer charge	$4.50
Minimum charge	$4.50
Fuel cost adjustment	0.00
Tax rate	4.00%
No. of energy blocks	2
Block 1 energy size (MMBtu)	2.5
Block 2 energy size (MMBtu)	>2.5
Block 1 energy charge ($/MMBtu)	5.145
Block 2 energy charge ($/MMBtu)	4.033

TABLE 2.20 A Gas Utility Rate with Demand Charges for Commercial General Service (Utility Rate E)

Billing Item	Any Month (Jan–Dec)
Customer charge	0.00
Minimum charge	$300.00
Fuel cost adjustment	0.00
Tax rate	4.00%
No. of energy blocks	3
Block 1 energy size (MMBtu)	10,000
Block 2 energy size (MMBtu)	10,000
Block 3 energy size (MMBtu)	>20,000
Block 1 energy charge ($/MMBtu)	3.482
Block 2 energy charge ($/MMBtu)	3.412
Block 3 energy charge ($/MMBtu)	3.385
No. of demand blocks	2
Block 1 demand size (MMBtu/day)	10
Block 2 demand size (MMBtu/day)	>10
Block 1 demand charge ($/MMBtu/day)	5.50
Block 2 demand charge ($/MMBtu/day)	4.50

Example 2.10

Using utility rate structure D, calculate the gas utility bill for a residence during a month when the energy use is 10.4 MMBtu.

Solution

Considering all the charges imposed by utility rate D (see Table 2.19), the monthly gas bill can be calculated as shown below:

(a) Customer charge = $ 4.50
(b) Energy charge = 2.5 MMBtu * $5.145/MMBtu + 7.9 MMBtu * $4.033/MMBtu = $44.72
(c) Fuel cost adjustment = $ 0.00
(d) Taxes = 4.00 percent [(a) + (b) + (c)] = $ 8.05

Total monthly charges = (a) + (b) + (c) + (d) = $57.27

Thus, the average gas cost for the month is $57.27/10.4 MMBtu or 5.51/MMBtu.

Example 2.11

Using utility rate structure E, calculate the gas utility bill for a commercial facility during a month when the energy use is 15,000 MMBtu and the billing demand is 500 MMBtu/day.

Solution

Accounting for all the charges considered by utility rate E (see Table 2.20), the electric energy bill for the winter month can be calculated as follows:

(a) Customer charge = $ 0.00
(b) Energy charge = 10,000MMBtu * $3.482/MMBt + 5,000MMBtu * $3.412/MMBt = $51,880.00
(c) Demand charge = 10 * $5.50 + 490 * $4.50 = $ 2,260.00
(d) Taxes = 4.00 t * [(a) + (b) + (c)] = $ 2,165.60

Total monthly charges = (a) + (b) + (c) + (d) = $56,305.60

The average cost of electricity for the month is then $56,305.60/15,000 MMBtu or $3.75/MMBtu.

In addition to energy and demand charges, the price of natural gas is determined based on the interruptible priority class selected by the customer. A customer with a low priority has a cheaper rate but can be curtailed whenever a shortage in the gas supply is experienced by the utility. However, some small quantities of gas are generally supplied to prevent the pipes from freezing and to keep the pilot lights burning.

2.5 Utility Rates for Other Energy Sources

The utility rate structures for energy sources other than electricity and natural gas are generally based on a flat rate. For instance, crude oil is typically charged per gallon whereas coal is priced on a per ton basis. The prices of oil products and coal are set by market conditions but may vary within a geographical area depending on local surcharges and tax rates. Moreover, fuel oil or coal can be classified in a number of grades. The grades of fuel oil depend on the distillation process. For instance, No. 1 oil is a distillate and is used as domestic heating oil. On the other hand, No. 6 oil has a very heavy residue left after the other oils have been refined. The grades of coal depend on the sulfur content and percentage of moisture. Chapter 8 describes some of the properties of fuel oils in more detail.

In some applications, it may be possible and desirable to purchase steam or chilled water to condition buildings rather than using primary fuel to operate boilers and chillers. Steam can be available from large cogeneration plants. Chilled water and steam may also be produced based on the economics of scale in district heating/cooling systems. Generally, steam and chilled water are both charged based on either a flat rate or a block rate structure for both energy and demand. The steam is charged based on pound per hour (for demand charges) or thousand of pounds (for energy charges). Meanwhile, the chilled water is charged on the basis of tons (for demand charges) or ton-hours (for energy charges).

2.6 Summary

In this chapter, various energy sources used to operate buildings and industrial facilities in the United States are discussed. In addition, the energy price rate structures proposed by U.S. utilities are outlined with a special emphasis on electricity pricing features. Several calculation examples are presented to illustrate the effects of various components of rate structures on the monthly utility bills.

The main goal of this chapter is to help the energy auditor understand the complexities of various utility rate structures. Significant energy cost savings can be achieved by merely selecting the energy pricing rate best suited for the audited facility.

Problems

2.1 (i) Calculate the utility bill for a facility during one month characterized by the following parameters:
- Actual demand: 450 kW
- Energy consumption: 85,000 kWh
- Average power factor: 70 percent
- The facility is subject to the following rate structure:
- Customer charge: $150/month
- Billed demand charge: $10/kW
- Energy charge: $0.025/kWh
- The rate structure includes a power factor clause that states that the billing demand is determined as the actual demand multiplied by the ratio of 85 and the actual average power factor expressed in percent.

(ii) Determine the cost savings achieved for the month if the facility power factor is improved to be at least 85 percent.

2.2 A company consumed about 355,000 kWh of electrical energy during the month of April. The billed demand is estimated to be 600 kW for April. If the sales tax is 7 percent, calculate the electric utility bill for the company.
Estimate the reduction in the April electric utility bill for the company following a peak shaving measure that lowered the April billed peak demand to 450 kW.
The company is subject to the following electric rate structure:
Rate-1:
(i) Customer charge: $145.00/month
(ii) Energy charge: $0.0325/kWh/month
(iii) Demand charge applicable to the billing demand:
- On-peak season: $500 for the first 100 kW
- plus $5.50/kW for any additional kW
- Off-peak season: $300 for the first 100 kW
- plus $3.50/kW for any additional kW

The on-peak season extends from May to October and the off-peak season spans from November to April.

2.3 A manufacturing company consumes on average 250,000 kWh per month and has a billed demand of 1,200 kW during each month. The company can select between two rates: Rate-1 (described in Problem 2.2) and Rate-2 (described below). Determine the best rate suitable for the company.
Rate-2:
(i) Customer charge: $250.00/month
(ii) Energy charge: $0.0225/kWh/month
(iii) Demand charge applicable to the billing demand:
- On-peak season: $400 for the first 100 kW
- plus $6.00/kW for any additional kW
- Off-peak season: $250 for the first 100 kW
- plus $4.00/kW for any additional kW

The on-peak season extends from June to September and the off-peak season spans from October to May.

2.4 The monthly energy consumption and peak demand history of a commercial building is shown below for one year.

Month	Energy Use (kWh)	Peak Demand (kW)
Jan	125,000	455
Feb	137,500	505
Mar	155,500	610
Apr	176,000	675
May	185,500	755
Jun	201,000	920
Jul	235,500	1,150
Aug	240,000	1,200
Sep	197,500	895
Oct	184,500	650
Nov	164,000	605
Dec	141,000	550

The commercial building is on the rate structure defined in Table 2.13 (utility rate B).

(i) Estimate the annual electric utility bill for the commercial building assuming that the average power factor is 70 percent during each month.

(ii) Determine the reduction in the annual electric utility bill if the power factor is improved to 90 percent.

2.5 A utility offers the following rates to its commercial customers:

Service Level 1:
Customer charge: $500/bill/month
Energy charge: $0.025 applicable to all kWh/month
Demand charge applicable to all kW/month of billing demand:
On-peak season: $350.00 for first 75 kW or less, and
$3.75/kW for all additional kW
Off-peak season: $210.00 for first 75 kW or less, and
$2.50/kW for all additional kW

Service Level 2:
Customer charge: $200/bill/month
Energy charge: $0.027 applicable to all kWh/month
Demand charge applicable to all kW/month of billing demand:
On-peak season: $400.00 for first 75 kW or less, and
$4.50/kW for all additional kW
Off-peak season: $250.00 for first 75 kW or less, and
$2.90/kW for all additional kW

Service Level 3:
Customer charge: $110/bill/month
Energy charge: $0.028 applicable to all kWh/month
Demand charge applicable to all kW/month of billing demand:
On-peak season: $450.00 for first 75 kW or less, and
$4.90/kW for all additional kW
Off-peak season: $290.00 for first 75 kW or less, and
$3.30/kW for all additional kW

The on-peak season includes the months of June through October. The off-season includes the remainder of the months of any year.

The billing demand is defined as the maximum demand as determined above, corrected for the power factor (refer to the power factor clause defined below), provided that no billing demand shall be considered as less than a percentage of the highest on-peak season maximum demand corrected for the power factor previously determined during the 12 months ending with the current month.

The following power factor clause applies to all the utility rate structures.

Power factor clause: The consumer shall at all times use power so that the power factor shall be as close to 100 percent as possible, but when the average power factor as determined by continuous measurement of lagging reactive kilovolt-ampere hours is less than 80 percent, the billing demand shall be determined by multiplying the maximum demand, shown by the demand meter for the billing period, by 80 and dividing the product thus obtained by the actual average power factor expressed in percent.

(i) A large manufacturing company is on the level 3 rate schedule. Its peak demand for the last year is shown below. It found a way to reduce demand in the off-peak season by 100 kW, but the peak season demand will be the same (i.e., the demand in each month of November through May would be reduced by 100 kW). Assuming the company is on the 65 percent ratchet clause specified in the rate schedule, what is the dollar savings? Assume the high month was July of the previous year at 1,150 kW. If the demand reduction of 100 kW occurred in the peak season, what would be the dollar savings (i.e., the demand in June through October would be reduced by 100 kW).

(ii) Determine among the three available rate structures, the rate level that minimizes the annual utility bill for the manufacturing company before any demand reduction.

Month	Demand (kW)	Month	Demand (kW)
Jan	495	Jul	1,100
Feb	550	Aug	1,000
Mar	580	Sep	900
Apr	600	Oct	600
May	610	Nov	500
Jun	900	Dec	515

2.6 A company is on the utility rate structure defined as level 1 in Problem 2.5, In auditing a company, you find it averaged a 65 percent power factor over the past year. It is on the rate schedule given in the class handout and averaged 1,000 kW/month. Neglecting any ratchet clause and assuming the demand and power factor is constant each month, calculate the savings for correcting to an 80 percent power factor. How much capacitance would be necessary to obtain this correction? Assume level 1.

2.7 A company has contacted you regarding its rate schedule. The company is on the rate schedule defined in Problem 2.5 as service level 3 (secondary service), but is near transmission lines and so can accept service at a higher level (service level 1) if it buys its own transformers. Assuming it consumes 300,000 kWh/month and is billed for 1,000 kW each month, how much would it save by owning its own transformers? Ignore any charges other than demand and energy.

3

Economic Analysis

3.1 Introduction

In most applications, initial investments are required to implement energy conservation measures. These initial costs generally must be justified in terms of a reduction in the operating costs (due to energy cost savings). Therefore, most improvements in the efficiency of energy systems have a delayed reward; that is, expenses come at the beginning of a retrofit project and the benefits are incurred later. For an energy retrofit project to be economically worthwhile, the initial expenses have to be lower than the sum of savings obtained by the reduction in the operating costs over the lifetime of the project.

The lifetime of an energy system retrofit project typically spans several years. Therefore, it is important to compare savings and expenditures of various amounts of money properly over the lifetime of a project. Indeed, an amount of money at the beginning of a year is worth less at the end of the year and has even less buying power at the end of the second year. Consequently, the amounts of money due to expenditures or savings incurred at different times of a retrofit project cannot be simply added.

In engineering economics, savings and expenditures of amounts of money during a project are typically called *cash flows*. To compare the various cash flows over the lifetime of a project, a life-cycle cost analysis is typically used. In this chapter, the basic concepts of engineering economics are described. First, some common economic parameters are defined. In addition, data is provided to help the reader estimate relevant economic parameters. Then, the general procedure of an economic evaluation of a retrofit project is described. Finally, some of the advantages and disadvantages of the various economic analysis methods are discussed.

3.2 Basic Concepts

There are several economic parameters that affect a decision between various investment alternatives. To perform a sound economic analysis for energy retrofits, it is important that the auditor be (i) familiar with the most important economic parameters, and (ii) aware of the basic economics concepts. The parameters and the concepts that significantly affect the economic decision making include:

- The time value of money and interest rates including simple and compounded interest
- Inflation rate and composite interest rate
- Taxes including sales, local, state, and federal tax charges
- Depreciation rate and salvage value

In the following sections, the above-listed parameters are described in detail to help the reader better understand the life-cycle cost analysis procedure discussed later in this chapter.

3.2.1 Interest Rate

When money is borrowed to cover part or all the initial cost of a retrofit project, a fee is charged for the use of this borrowed money. This fee is called *interest* (I) and the amount of money borrowed is called *principal* (P). The amount of the fee depends on the value of the principal and the length of time over which the money is borrowed.

The interest charges are typically normalized to be expressed as a percentage of the total amount of money borrowed. This percentage is called the *interest rate* (i):

$$i = \frac{I}{P} \tag{3.1}$$

It is clear that an economy with low interest rates encourages money borrowing (for investment in projects or simply for buying goods) whereas an economy with high interest rates encourages money saving. Therefore, if money has to be borrowed for a retrofit project, the interest rate is a good indicator of whether the project can be cost-effective.

To calculate the total interest charges over the lifetime of a project, two alternatives are typically considered:

Simple Interest Charges: The interest fee I to be paid at the end of the life of the loan is proportional to both the interest rate i and the lifetime N:

$$I = NiP \tag{3.2}$$

The same unit of time should be used for both N and i (i.e., one year, one month, etc.). The total amount of payment F, due at the end of the loan period, includes both the principal and the interest charges:

$$F = P + I = P(1 + Ni) \tag{3.3}$$

Compounded Interest Charges: In this case, the lifetime N is divided into smaller periods, typically called interest periods (such as one month or often one year). The interest fee is charged at the end of each interest period and is allowed to accumulate from one interest period to the next. The interest charges I_k for the total amount F_k accumulated after k periods is

$$I_k = iF_k \tag{3.4}$$

Therefore, the total payment F_{k+1} at the end of period k (and thus at the beginning of period $k + 1$) is:

$$F_{k+1} = F_k + I_k = (1 + i)F_k \tag{3.5}$$

If the principal is P which is the total payment at the beginning of first period (i.e., $F_1 = P$), the total amount due at the end of lifetime N is

$$F = F_{N+1} = (1 + i)^N P \tag{3.6}$$

TABLE 3.1 Average Long-Term Interest Rates for Selected Countries

Period/Year	France	Germany	Japan	United States
Period				
1961–1973	6.9	7.2	7.0	5.3
1974–1980	11.2	8.1	8.0	8.6
1981–1990	12.0	7.8	6.5	10.3
1990–1995	8.5	7.5	5.1	7.2
Year				
1995	7.5	6.9	3.4	6.6
2000	5.4	5.3	1.7	6.0
2005	3.4	3.4	1.4	4.8
2010[a]	4.1	4.0	2.0	4.4

Source: OECD, Economic Statistics, http://www.ocde.org, 2009.
[a] Based on predictions.

It is clear that the total amount of money F increases exponentially with N. The interest charges follow the law of compounded interest.

For the economic analysis of energy-efficiency projects, the interest rate is typically assumed to be constant throughout the lifetime of the projects. Therefore, it is common to use average interest rates when an economic analysis is performed. Table 3.1 provides historical data for long-term interest rates for selected countries.

Example 3.1

A building owner has \$10,000 available and has the option to invest this money in either (i) a bank that has an annual interest rate of 7 percent or (ii) buying a new boiler for the building. If he decides to invest all the money in the bank, how much will the building owner have after ten years? Compare this amount if simple interest had been paid.

Solution

If the interest is compounded, using Eq. (3.6) with $P = \$10,000$, $N = 10$, and $i = 0.07$, the investment will accumulate to the total amount F:

$$F = \$10,000 * (1 + 0.07)^{10} = \$19,672$$

Thus, the building owner's original investment will have almost doubled over the ten-year period.

If simple interest had been paid, the total amount that would have accumulated is slightly less and is determined from Eq. (3.3):

$$F = \$10,000 * (1 + 0.07 * 10) = \$17,000$$

3.3 Inflation Rate

Inflation occurs when the cost of goods and services increases from one period to the next. The interest rate i defines the cost of money, and the inflation rate λ measures the increase in the cost of goods and services. Therefore, the future cost of a commodity FC is higher than the present cost PC of the same commodity:

$$FC = PC(1+\lambda) \tag{3.7}$$

Over a lifetime N the future cost of a commodity increases exponentially:

$$FC = PC(1+\lambda)^N \tag{3.8}$$

Similar to interest rates, inflation rates are typically assumed to be constant over the lifetime of the energy retrofit project. Table 3.2 presents historical data for inflation rates for selected countries. The escalation of energy cost is an important factor to consider in evaluating energy retrofit projects. The energy escalation rate can be considered as one form of inflation rate.

If the interest charges are compounded at the same periods during which inflation occurs, the future worth can be determined from the present value P as follows:

$$F = P\frac{(1+i)^N}{(1+\lambda)^N} \tag{3.9}$$

The expression above for F can be rearranged as follows:

$$F = P\left(\frac{1+i}{1+\lambda}\right)^N = P\left(1+\frac{i-\lambda}{1+\lambda}\right)^N \tag{3.10}$$

A composite interest rate θ can be defined to account for the fact that inflation decreases the buying power of money due to increases in the cost of commodities:

$$\theta = \frac{i-\lambda}{1+\lambda} \tag{3.11}$$

TABLE 3.2 Average Inflation Rates for Selected Countries

Period/Year	France	Germany	Japan	United States
Period				
1971–1980	9.8	5.0	8.8	7.1
1981–1990	6.2	2.5	2.1	4.7
1990–1995	1.9	2.8	1.0	2.6
Year				
1995	1.8	1.2	–0.1	2.8
2000	1.8	1.4	–0.5	3.4
2005	1.9	1.9	–0.6	3.4
2010[a]	0.7	0.4	–1.4	1.0

Source: OECD, Economic Statistics, http://www.ocde.org, 2009.
[a] Based on predictions.

It should be mentioned that theoretically the composite interest rate can be negative. In this case, the money loses its value with time.

Example 3.2

Determine the actual value of the $10,000 investment for the building owner of Example 3.1 if the economy experiences an annual inflation rate of 4 percent.

Solution

The composite interest rate can be determined using Eq. (3.11):

$$\theta = \frac{i - \lambda}{1 + \lambda} = \frac{0.07 - 0.04}{1 + 0.04} = 0.02885$$

Using Eq. (3.10) with P = $10,000 and N = 10, the investment will accumulate to the total amount F:

$$F = \$10{,}000 * (1 + 0.02885)^{10} = \$13{,}290$$

3.3.1 Tax Rate

In most economies, the interest that is received from an investment is subject to taxation. If this taxation has a rate t over a period that coincides with the interest period, then the amount of taxes T to be collected from an investment P with an interest rate i is determined as follows:

$$T = tiP \tag{3.12}$$

Therefore, the net return from the investment P to the investor after tax deductions is:

$$I' = I - T = (1 - t)iP \tag{3.13}$$

An effective interest rate io' can then be defined to account for the loss of income due to taxation:

$$i' = (1 - t)i \tag{3.14}$$

Therefore, the composite interest rate defined by Eq. (3.11) can be generalized to account for both inflation and tax rates related to present and future values:

$$\theta = \frac{(1 - t)i - \lambda}{1 + \lambda} \tag{3.15}$$

Example 3.3

If the building owner is in the 28 percent tax bracket, determine the actual value of his $10,000 investment considered in Example 3.1 if the economy experiences an annual inflation rate of 4 percent.

Solution

The composite interest rate can be determined using Eq. (3.15):

$$\theta = \frac{(1-t)i-\lambda}{1+\lambda} = \frac{(1-0.28)*0.07-0.04}{1+0.04} = 0.01$$

Using Eq. (3.10) with P = $10,000 and N = 10, the investment will accumulate to the total amount F:

$$F = \$10,000*(1+0.01)^{10} = \$11,046$$

3.3.2 Cash Flows

In evaluating energy-efficiency projects, it is important to account for the total cash receipts and disbursements due to the implementation of an energy conservation measure (such as the installation of a new boiler) for each period during the entire lifetime of the project. The difference between the total cash receipts (inflows) and total cash disbursements (outflows) for a given period of time is called a cash flow.

Over the lifetime of a project, an accurate accounting of all the cash flows should be performed. For energy-efficiency improvement projects, the cash flow accounting can be in a tabular format as illustrated in Table 3.3 that lists the cash flows attributed to the installation of a new steam boiler in a

TABLE 3.3 Cash Flows for an Installation of a New Boiler over a Lifetime of Ten Years

End of Year	Total Cash Receipts	Total Cash Disbursements	Total Cashflows	Comments
0	$ 0	$ 400,000	– $ 400,000	Installation cost of a new boiler
1	$ 40,000	$ 0	+ $ 40,000	Net cost savings through year 10
2	$ 38,000	$ 0	+ $ 38,000	
3	$ 36,000	$ 0	+ $ 36,000	
4	$ 34,000	$ 0	+ $ 34,000	
5	$ 33,000	$ 0	+ $ 33,000	
6	$ 32,000	$ 0	+ $ 32,000	
7	$ 31,000	$ 0	+ $ 31,000	
8	$ 30,500	$ 0	+ $ 30,500	
9	$ 30,000	$ 0	+ $ 30,000	
10	$ 29,500	$ 0	+ $ 29,500	

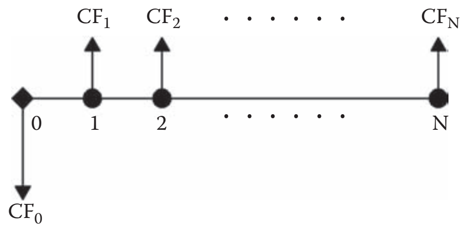


FIGURE 3.1 Typical cash flow diagram.

hospital. The table accounts for the cost related to the initial cost of a new boiler installation (counted as a disbursement for Year 0) and the cost savings due to the higher energy efficiency of the new boiler (counted as receipts in Year 1 through Year 10). The reduction in the yearly receipts is attributed to the aging of the equipment.

Note that the cash flows are positive when they represent inflows (i.e., receipts) and are negative when they are outflows (i.e., disbursements). To better visualize the evolution over time of the cash flows, a cash flow diagram as depicted in Figure 3.1 is used. Note that in this figure, the initial cash flow, C_0 = –\$400,000 (disbursement), is represented by a downward-pointing arrow. Meanwhile, the cash flows occurring later, C_1 through C_N with $N = 10$, are receipts and are represented by upward-pointing arrows.

It should be noted again that the cash flows cannot simply be added because the value of money changes from one period to the next. In the next section, various factors are defined to correlate cash flows occurring at different periods.

3.4 Compounding Factors

Two types of payment factors are considered in this section. These payment factors are useful in the economic evaluation of various energy audit projects. Without loss of generality, the interest period is assumed in the remainder of this chapter to be one year. Moreover, a nominal discount rate *d* is used throughout this chapter. This discount rate is an effective interest rate that includes the effects of several parameters such as inflation and taxation discussed above.

3.4.1 Single Payment

In this case, an initial payment is made to implement a project by borrowing an amount of money *P*. If this sum of money earns interest at a discount rate *d*, then the value of the payment *P* after *N* years is provided by Eq. (3.16). The ratio *F*/*P* is often called the single payment compound amount factor (SPCA). The SPCA factor is a function of *i* and *N* and is defined as:

$$SPCA(d,N) = F/P = (1+d)^N \tag{3.16}$$

Using the cash flow diagram of Figure 3.1, the single payment represents the case where $C_0 = P$, $C_1 = \cdots = C_N - 1 = 0$, and $C_N = F$ as illustrated in Figure 3.2.

The inverse ratio *P*/*F* allows us to determine the value of the cash flow *P* needed to attain a given amount of cash flow *F* after *N* years. The ratio *P*/*F* is called the single payment present worth (SPPW) factor and is equal to:

$$SPPW(d,N) = P/F = (1+d)^{-N} \tag{3.17}$$

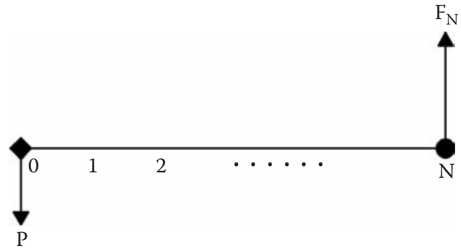


FIGURE 3.2 Cash flow diagram for single payment.

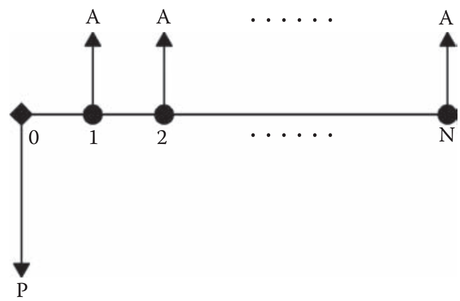


FIGURE 3.3 Cash flow diagram for uniform-series payment.

3.4.2 Uniform-Series Payment

In the vast majority of energy retrofit projects, the economic benefits are estimated annually and are obtained after a significant initial investment. It is hoped that during the lifetime of the project, the sum of all the annual benefits can surpass the initial investment.

Consider then an amount of money P that represents the initial investment, and a receipt of an amount A that is made each year and represents the cost savings due to the retrofit project. To simplify the analysis, the amount A is assumed to be the same for all the years during the lifetime of the project. Therefore, the cash flows—using the diagram of Figure 3.1—are $C0 = P$, $C1 = \cdots = CN = A$ as depicted in Figure 3.3.

To correlate between P and A, we note that for any year k, the present worth Pk of the receipt A can be determined by using Eq. (3.17):

$$P_k = A(1+d)^{-k} \tag{3.18}$$

By summing the present worth values for all the annual receipts A, the result should equal the cash flow P:

$$P = \sum_{k=1}^{N} P_k = \sum_{k=1}^{N} A(1+d)^{-k} \tag{3.19}$$

The sum can be rearranged to obtain a geometric series that can be evaluated as shown below:

$$P = \frac{A}{(1+d)^N} \sum_{k=1}^{N} (1+d)^{N-k} = \frac{A}{(1+d)^N} \sum_{k=0}^{N-1} (1+d)^k = \frac{A[(1+d)^N - 1]}{d(1+d)^N} \tag{3.20}$$

The ratio A/P is called the uniform-series capital recovery factor (USCR). This USCR factor can be determined as a function of both d and N:

$$USCR(d,N) = A/P = \frac{d(1+d)^N}{(1+d)^N - 1} = \frac{d}{1-(1+d)^{-N}} \tag{3.21}$$

The uniform-series present worth factor (USPW), which allows us to determine the value of P knowing the amount A, is the ratio P/A and can be expressed as follows:

$$USPW(d,N) = P/A = \frac{(1+d)^N - 1}{d(1+d)^N} = \frac{1-(1+d)^{-N}}{d} \tag{3.22}$$

Example 3.4

Find the various compounding factors for N = 10 years and d = 5 percent.

Solution

The values of the compounding factors for d = 0.05 and N = 10 years are summarized below:

Compounding Factor	Equation Used	Value
SPCA	Eq. (3.16)	1.629
SPPW	Eq. (3.17)	0.613
USCR	Eq. (3.21)	0.129
USPW	Eq. (3.22)	7.740

3.5 Economic Evaluation Methods among Alternatives

To evaluate the cost-effectiveness of energy retrofit projects, several evaluation tools can be considered. The basic concept of all these tools is to compare among the alternatives the net cash flow that results during the entire lifetime of the project. As discussed earlier, a simple addition of all the cash flows such as those represented in Figure 3.1 is not possible. However, by using compound factors discussed in Section 3.3, "conversion" of the cash flows from one period to another is feasible. This section provides a brief description of the common evaluation methods used in engineering projects.

3.5.1 Net Present Worth

The basic principle of this method is to evaluate the present worth of the cash flows that occur during the lifetime of the project. Referring to the cash flow diagram of Figure 3.1, the sum of all the present worth of the cash flows can be obtained by using the single payment present worth factor defined in Eq. (3.17):

$$NPW = -CF_0 + \sum_{k=1}^{N} CF_k * SPPW(d,k) \tag{3.23}$$

Note that the initial cash flow is negative (a capital cost for the project) whereas the cash flows for the other years are generally positive (revenues).

In the particular but common case of a project with constant annual revenue (due to energy operating cost savings), $CFk = A$, the net present worth is reduced to:

$$NPW = -CF_0 + A * USPW(d, N) \tag{3.24}$$

For the project to be economically viable, the net present worth has to be positive or at worst zero ($NPW \geq 0$). Obviously, the higher the NPW is, the more economically sound the project is.

The net present worth value method is often called the net savings method because the revenues are often due to the cost savings from implementing the project.

3.5.2 Rate of Return

In this method, the first step is to determine the specific value of the discount rate d', that reduces the net present worth to zero. This specific discount rate is called the rate of return (ROR). Depending on the case, the expression of NPW provided in Eq. (3.23) or Eq. (3.24) can be used. For instance, in the general case of Eq. (3.23), the rate of return d′ is solution of the following equation:

$$-CF_0 + \sum_{k=1}^{N} CF_k * SPPW(d', k) = 0 \tag{3.25}$$

To solve this equation accurately, any numerical method (such the Newton–Raphson iteration method) can be used. However, an approximate value of d' can be obtained by trial and error. This approximate value can be determined by finding the two d-values for which the NPW is slightly negative and slightly positive, and then interpolating linearly between the two values. It is important to remember that a solution for ROR may not exist.

Once the rate of return is obtained for a given alternative of the project, the actual market discount rate or the minimum acceptable rate of return is compared to the ROR value. If the value of ROR is larger ($d' > d$), the project is cost-effective.

3.5.3 Benefit–Cost Ratio

The benefit–cost ratio (BCR) method is also called the savings-to-investment ratio (SIR) and provides a measure of the net benefits (or savings) of the project relative to its net cost. The net values of both benefits (Bk) and costs (Ck) are computed relative to a base case. The present worth of all the cash flows is typically used in this method. Therefore, the benefit–cost ratio is computed as follows:

$$BCR = \frac{\sum_{k=0}^{N} B_k * SPPW(d, k)}{\sum_{k=0}^{N} C_k * SPPW(d, k)} \tag{3.26}$$

The alternative option for the project is considered economically viable relative to the base case when the benefit–cost ratio is greater than one ($BCR > 1.0$).

3.5.4 Payback Period

In this evaluation method, the period Y (typically expressed in years) required to recover an initial investment is determined. Using the cash flow diagram of Figure 3.1, the value of Y is the solution of the following equation:

$$CF_0 = \sum_{k=1}^{Y} CF_k * SPPW(d,k) \tag{3.27}$$

If the payback period Y is less than the lifetime of the project N ($Y < N$), then the project is economically viable. The value of Y obtained using Eq. (3.27) is typically called the discounted payback period (DPB) because it includes the value of money.

In the vast majority of applications, the time value of money is neglected in the payback period method. In this case, Y is called the simple payback period (SPB) and is the solution to the following equation:

$$CF_0 = \sum_{k=1}^{Y} CF_k \tag{3.28}$$

In the case where the annual net savings are constant ($CF_k = A$), the simple payback period can be easily calculated as the ratio of the initial investment over the annual net savings:

$$Y = \frac{CF_0}{A} \tag{3.29}$$

The values for the simple payback period are shorter than for the discounted payback periods because the undiscounted net savings are greater than their discounted counterparts. Therefore, acceptable values for simple payback periods are typically significantly shorter than the lifetime of the project.

3.5.5 Summary of Economic Analysis Methods

Table 3.4 summarizes the basic characteristics of the economic analysis methods used to evaluate single alternatives of an energy retrofit project.

TABLE 3.4 Summary of the Basic Criteria for the Various Economic Analysis Methods for Energy Conservation Projects

Evaluation Method	Equation	Criterion
Net present worth (*NPW*)	$NPW = -CF_0 + \sum_{k=1}^{N} CF_k * SPPW(d,k)$	NPW > 0
Rate of return (*ROR*)	$-CF_0 + \sum_{k=1}^{N} CF_k * SPPW(d',k) = 0$	$d' > d$
Benefit–cost ratio (*BCR*)	$BCR = \dfrac{\sum_{k=0}^{N} B_k * SPPW(d,k)}{\sum_{k=0}^{N} C_k * SPPW(d,k)}$	$BCR > 1$
Discounted payback period (*DPB*)	$CF_0 = \sum_{k=1}^{Y} CF_k * SPPW(d,k)$	$Y < N$
Simple payback period (*SPB*)	$Y = \dfrac{CF_0}{A}$	$Y << N$

It is important to note that the economic evaluation methods described above provide an indication of whether a single alternative of a retrofit project is cost-effective. However, these methods cannot be used or relied on to compare and rank various alternatives for a given retrofit project. Only the life-cycle cost (LCC) analysis method is appropriate for such endeavor.

Example 3.5

After finding that the old boiler has an efficiency of only 60 percent whereas a new boiler would have an efficiency of 85 percent, a building owner of Example 3.1 has decided to invest the $10,000 in getting a new boiler. Determine whether this investment is cost-effective if the lifetime of the boiler is ten years and the discount rate is 5 percent. The boiler consumes 5,000 gallons per year at a cost of $1.20 per gallon. An annual maintenance fee of $150 is required for the boiler (independently of its age). Use all five methods summarized in Table 3.4 to perform the economic analysis.

Solution

The base case for the economic analysis presented in this example is the case where the boiler is not replaced. Moreover, the salvage value of the boiler is assumed insignificant after ten years. Therefore, the only annual cash flows (A) after the initial investment on a new boiler are the net savings due to higher boiler efficiency as calculated below:

$$A = Fuel - Use_{before} * (1 - \eta_{before} / \eta_{after}) * Fuel - \mathrm{cost}/gallon$$

Thus

$$A = 5000 * (1 - 0.60 / 0.85) * \$1.20 = \$1,765$$

The cost-effectiveness of replacing the boiler is evaluated as indicated below:

1. *Net Present Worth*. For this method $CF_0 = \$10,000$ and $CF_1 = \cdots = CF_{10} = A$, $d = 0.05$, and $N = 10$ years. Using Eq. (3.24) with USPW = 7.740 (see Example 3.4):

$$NPW = \$3,682$$

Therefore, the investment in purchasing a new boiler is cost-effective.

2. *Rate of return*. For this method also $CF_0 = \$10,000$ and $CF_1 = \cdots = CF_{10} = A$, whereas SPPW($d'$,$k$) is provided by Eq. (3.17). By trial and error, it can be shown that the solution for d' is:

$$d' = 12.5\%$$

Inasmuch as $d' > d = 5\%$, the investment in replacing the boiler is cost-effective.

3. *Benefit–Cost Ratio*. In this case, $B_0 = 0$ and $B_1 = \cdots = B_{10} = A$ whereas $C_0 = \$10,000$ and $C_1 = \cdots = C_{10} = 0$.
   Note that because the maintenance fee is applicable to both the old and new boiler, this cost is not accounted for in this evaluation method (only the benefits and costs relative to the base case are considered).

Using Eq. (3.26):

$$BCR = 1.368$$

Thus, the benefit–cost ratio is greater than unity ($BCR > 1$) and the project of getting a new boiler is economically feasible.

4. *Compounded Payback Period*. For this method, $CF_0 = \$10,000$ and $CF_1 = \cdots = CF_{10} = A$. Using Eq. (3.27), Y can be solved: $Y = 6.9$ years.
   Thus the compounded payback period is shorter than the lifetime of the project ($Y > N = 10$ years) and therefore replacing the boiler is cost-effective.
5. *Simple Payback Period*. For this method, $CF_0 = \$10,000$ and $A = \$1,765$. Using Eq. (3.29), Y can be easily determined: $Y = 5.7$ years.
   Thus, the simple payback period method indicates that the boiler retrofit project can be cost-effective.

3.6 Life-Cycle Cost Analysis Method

The life-cycle cost analysis method is the most commonly accepted method to assess the economic benefits of energy conservation projects over their lifetime. Typically, the method is used to evaluate at least two alternatives of a given project (for instance, evaluate two alternatives for the installation of a new HVAC system: a VAV system or a heat pump system to condition the building). Only one alternative will be selected for implementation based on the economic analysis.

The basic procedure of the LCC method is relatively simple because it seeks to determine the relative cost-effectiveness of the various alternatives. For each alternative including the base case, the total cost is computed over the project lifetime. The cost is commonly determined using one of two approaches: the present worth or the annualized cost estimate. Then, the alternative with the lowest total cost (or LCC) is typically selected.

Using the cash flow diagram of Figure 3.1, the LCC amount for each alternative can be computed by projecting all the costs (including costs of acquisition, installation, maintenance, and operating the energy systems related to the energy-conservation project) on either:

(i) One single present value amount that can be computed as follows:

$$LCC = \sum_{k=0}^{N} CF_k * SPPW(d,k) \tag{3.30}$$

This is the most commonly used approach in calculating LCC in energy retrofit projects.

(ii) Multiple annualized costs over the lifetime of the project:

$$LCC_a = USCR(d,N)*\left[\sum_{k=1}^{N} CF_k * SPPW(d,k)\right] \tag{3.31}$$

Note that the two approaches for calculating the LCC values are equivalent.

In most energy-efficiency projects, the annual cash flow remains the same after the initial investment. In this case, LCC can be estimated based on the initial cost *IC* and the annual cost *AC* as follows:

$$LCC = IC + USPW(d,N)* AC \tag{3.32}$$

Example 3.6

The building owner of Example 3.5 has three options to invest his money as briefly described below.

(A) Replace the entire older boiler (including burner) with more efficient heating system. The old boiler/burner system has an efficiency of only 60 percent whereas a new boiler/burner system has an efficiency of 85 percent. The cost of this replacement is $10,000.
(B) Replace only the burner of the old boiler. This action can increase the efficiency of the boiler/burner system to 66 percent. The cost of the burner replacement is $2,000.
(C) Do nothing and replace neither the boiler nor the burner.

Determine the best economical option for the building owner. Assume that the lifetime of the retrofit project is ten years and the discount rate is 5 percent. The boiler consumes 5,000 gallons per year at a cost of $1.20 per gallon. An annual maintenance fee of $150 is required for the boiler (independently of its age). Use the life-cycle cost analysis method to determine the best option.

Solution

The total cost of operating the boiler/burner system is considered for the three options. In this analysis, the salvage value of the boiler or burner is neglected. Therefore, the only annual cash flows (A) after the initial investment on a new boiler are the maintenance fee and the net savings due to higher boiler efficiency. To present the calculations for LCC analysis, it is recommended to present the results in a tabular format and proceed as shown below:

Cost Item	Option A	Option B	Option C
Initial Investment			
(a) Replacement Cost ($)	10,000	2,000	0
Annual Operating Costs:			
(b) Fuel Use (gallons)	3,530	4,545	5,000
(c) Fuel Cost ($) [$1.2*(b)]	4,236	5,454	6,000
(d) Maintenance fee ($)	150	150	150
(e) Total Operating Cost ($) [(c) + (d)]	4,386	5,594	6,150
USPW factor [d = 5%, N = 10, Eq. (3.22)]	7.740	7.740	7.740
Present Worth ($) [(a) + USPW * (f)]	43,948	45,298	47,601

Therefore, the life-cycle cost for option A is the lowest. Thus, it is recommended for the building owner to replace the entire boiler/burner system.

This conclusion is different from that obtained by using the simple payback analysis [indeed, the payback period for option A, relative to the base case C, is SPB(A) = ($10,000)/($1,765) = 5.66 years; and for option B, SPB(B) = ($2,000)/ ($546) = 3.66 years].

Note that if the discount rate were d = 10 percent (which is unusually high for most markets), the USPW would be equal to USPW = 6.145 and the life-cycle cost for each option will be

$$LCC(\text{A}) = \$36{,}952 \quad LCC(\text{B}) = \$36{,}375 \quad LCC(\text{C}) = \$37{,}791$$

Therefore, Option B will become the most effective economically and will be the recommended option to the building owner.

3.7 General Procedure for an Economic Evaluation

It is important to remember that the recommendations for energy conservation projects that stem from an energy audit should be based on an economically sound analysis. In particular, before making the final recommendations the auditor should ask several questions such as:

Will project savings exceed costs?
Which design solution will be most cost-effective?
What project size will minimize overall building costs?
Which combination of interrelated projects will maximize net savings?
What priority should projects be given if the owner has limited investment capacity?

As alluded to earlier, the best suitable economic assessment method is the LCC method described in Section 3.6. Before the application of the LCC, several data are needed to perform an appropriate and meaningful economic analysis. To help the auditor in gathering the required information and in the application of the LCC method, the following systematic approach in any economic evaluation is proposed:

1. Define the problem that the proposed retrofit project is attempting to address and state the main objective of the project. (For instance, a building has an old boiler that does not provide enough steam to heat the entire building. The project is to replace the boiler with the main objective to heat all the conditioned spaces within the building).
2. Identify the constraints related to the implementation of the project. These constraints can vary in nature and include financial limitations or space requirements (for instance, the new boiler cannot be gas-fired because there is no supply of natural gas near the building).
3. Identify technically sound strategies and alternatives to meet the objective of the project. (For instance, two alternatives can be considered for the old boiler replacement: (i) a new boiler with the burner of the old boiler, (ii) a new boiler/burner system, and (iii) a new boiler/burner system with an automatic air–fuel adjustment control.)
4. Select a method of economic evaluation. There are several alternatives including the base case (which may consists of the alternative of "doing nothing"), however, the LCC method is preferred for energy projects. When a preliminary economic analysis is considered, the simple payback period method can be used. As mentioned earlier, the payback period method is not accurate and should be used with care.

5. Compile data and establish assumptions. The data includes the discount rates and the energy, installation, operating, and maintenance costs. Some of these data are difficult to acquire and some assumptions or estimations are required. For instance, an average discount rate over the life cycle of the project may be assumed based on a historical data.
6. Calculate indicators of economic performance. These indicators depend on the economic evaluation method selected. The indicators are the life-cycle costs for the LCC method.
7. Evaluate the alternatives. This evaluation can be performed by simply comparing the values of the LCC obtained for various alternatives.
8. Perform sensitivity analysis. Because the economic evaluation performed in Step 6 is typically on some assumed values (for instance, the annual discount rate), it is important to determine whether the results of the evaluation performed in Step 7 depend on some of these assumptions. For this purpose, the economic evaluation is repeated for all alternatives using different but plausible assumptions.
9. Take into account unqualified effects. Some of the alternatives may have effects that cannot be included in the economic analysis but may be determining factors in decision making. For instance, the environmental impact (emission of pollutants) can be important to disqualify an otherwise economically sound alternative.
10. Make recommendations. The final selection will be based on the findings of the three previous steps (i.e., Steps 7, 8, and 9). Typically, the alternative with the lowest LCC value will be recommended.

Once the project for energy retrofit is selected based on an economic analysis, it is important to decide on the financing options to actually carry out the project and implement the measures that allow a reduction in the energy cost of operating the facility. The next section discusses the common payment and financing options typically available for energy retrofit projects.

3.8 Financing Options

There are several alternatives that the owner or the facility manager can use to finance an energy retrofit project. These alternatives can be found under three main categories:

Direct purchasing
Leasing
Performance contracting

Each of the above-listed financing options is briefly described in the following sections with some indications of their advantages and disadvantages.

3.8.1 Direct Purchasing

This category includes all the financing alternatives where the facility (through its representatives) purchases the equipment and the services required to implement the energy retrofit projects either using its own money or through a loan.

Typically, the facility uses its own money to purchase the equipment and services when it has a strong cash reserves or when the measures to be implemented are simple and inexpensive with a payback period less than one year. The advantages of using its own funds are that (i) the organization benefits directly from any cost savings realized from the retrofit project, and (ii) the depreciation value of the purchased equipment can be deducted from taxes. The main disadvantage is a loss of capital available for other investment opportunities.

The facility can take out a loan to finance the retrofit project. Generally, the lending companies provide only part of the funds required for the implementation of the project. Therefore, the facility has to finance the other part of the project initial cost through its own capital and has to provide assets as security for the loan. The loan payments are generally structured to be lower than estimated energy cost savings. However, the facility bears all the risks associated with the project such as lower energy savings than predicted. Therefore, only projects with low payback periods (less than one year) are considered for this financing option.

If the management of the operation and maintenance are performed by an external organization, the financing of the energy profit can be carried out by this external organization. In this case, the facility purchases and gets the ownership of the equipment as soon as it is implemented.

3.8.2 Leasing

Instead of purchasing, the facility can lease the equipment required for the energy retrofit project. The laws and regulations for equipment leasing are generally complex and change frequently. It is therefore recommended that a financial expert or an attorney be consulted before finalizing any lease agreement. The advantage of leasing is that the payments are typically lower than loan payments. Two types of leasing are commonly available: (i) capital leasing and (ii) operating leasing.

Capital leasing meets one or more of the following criteria defined by the Financial Accounting Standard (FASB):

- The lease transfers ownership of the property to the customer at end of lease term.
- The lease includes a bargain purchase option (i.e., the purchase amount is less than fair market value).
- The lease term covers 75 percent or more of the estimated economic life of the leased equipment.
- The present value of the minimum lease equals or exceeds 90 percent of the fair market value of the leased equipment.

Therefore, the capital lease does not require an initial capital payment from the facility. Moreover, the capital lease is recorded on the facility's balance sheet as an asset and a liability.

If a lease does not meet any of the criteria listed above for capital leases, it is considered as an operating lease. Thus, the equipment is leased to the facility for a fixed monthly or annual fee during the contract period. At the end of the contract, the facility has only three options: remove the equipment, renegotiate the lease, or purchase the equipment but only at fair market value. The operating lease is recorded only as a periodic expense for the facility and is not booked on the facility's balance sheet.

3.8.3 Performance Contracting

Performance contracting has become an increasingly common financing option for energy retrofit projects during the last five years. Typically, a third party such as an energy services company (ESCO) obtains financing and assumes the performance risks associated with the energy retrofit project. Generally, the facility does not assume or provide any up-front investment but rather is guaranteed a certain amount from the energy cost savings. The financing organization owns the equipment during the term of the contract. Thus, the equipment asset and the associated debt do not appear on the facility's balance sheet.

The performance contracts are sometimes referred to as "shared savings" or "paid from savings" contracts because the incurred savings due to the energy retrofit project are distributed between the ESCO and the facility based on an agreement that is documented in the performance contract. At the end of the contract, ownership of the equipment is transferred to the facility based on terms specified in the contract. In general, performance contracts are inherently complex and take a long time to negotiate. Indeed, these contracts typically involve:

- Detailed specifications of the work to be performed for each facility
- Large sums of capital to implement the agreed-on energy retrofit measures

- Long periods inasmuch as the contracts may span over five years
- A wide range of contingencies
- High level of expertise in various disciplines including engineering, finance, and law

Therefore, the energy retrofit project has to have high potential for energy savings for the ESCO and the financial institution to commit to performance contracting. Therefore, the ESCO typically spends a significant amount of time, effort, and resources to perform the energy audit and establish a sound baseline from which energy savings can be estimated.

3.9 Summary

Energy audits of commercial and industrial buildings encompass a wide variety of tasks and require expertise in a number of areas to determine the best energy conservation measures suitable for an existing facility. This chapter described a general but systematic approach to perform the economic evaluation of various alternatives of energy retrofit projects.

Several analysis methods described in this chapter (such as present net worth, benefit–cost ratio, return, and payback period) are suitable for evaluating single alternatives but may not be used to rank among alternatives. Only the life-cycle cost analysis method should be used to select the most economical option among several alternatives of the same project. Under certain market conditions, the simple payback method can lead to erroneous conclusions and thus should be used only to provide an indication of the cost-effectiveness of an energy retrofit project. For sound economic analysis, the simple payback analysis method should not be used.

Problems

3.1 A company is considering the purchase of a new machine that will last eight years and cost $90,000 the first year, decreasing by $1,000 each year to $2,000 the eighth year. Determine how much money should the company set aside to pay for this machine:
(a) If the interest rate is 4 percent per year, compounded annually
(b) If the interest rate is 7 percent per year, compounded annually

3.2 You want to buy a new machine that will last 20 years and cost $200,000. Determine how much money you should pay for this machine each year based on equal annual payments:
(a) If the interest rate is 5 percent per year, compounded annually
(b) If the interest rate is 10 percent per year, compounded annually

3.3 An energy audit of a residential heating system reveals that the boiler-burner efficiency is only 60 percent. In addition, the energy audit showed that an electric water heater is used. The house uses 1,500 gallons of oil annually at a cost of $1.40 per gallon. The total electric bill averages $84.50 per month for an average monthly consumption of 818 kWh. Of this total, about 35 percent is for domestic water heating. It is suggested to the owner of the house to equip the boiler with a tankless domestic water heater replacing the existing electric water heater. The existing water heater is 12 years old and has no resale value and little expected life. The owner of the house is expected to spend $300 within the coming year to replace the electric water heater. The cost of the tankless heater is $400. To improve heating efficiency, it is suggested to

(i) Replace the existing burner with a new one (this burner replacement will improve efficiency to 65%) costing \$520.
(ii) Replace the entire heating plant with higher efficiency (85 percent) with a cost of \$2,000.

For two discount rates (5 percent and 10 percent), provide the LCC analysis of the following options and make the appropriate recommendations:

- Keep the boiler-burner and replace the electric water heater with like system.
- Replace the burner and electric water heater with like systems.
- Replace the boiler-burner with an efficient boiler and replace the electric heater with a like system.
- Replace the existing boiler-burner and electric water heater with new boiler/tankless domestic water heater.

Note: State all the assumptions made in your calculations.

3.4 An electrical energy audit indicates that the motor control center consumption is 8×10^6 kWh per year. By using high-efficiency motors, a savings of 15 percent can be achieved. The additional cost for these motors is about \$80,000. Assuming that the average energy charge is \$0.08 per kWh, is the expenditure justified based on a minimum rate of return of 18 percent before taxes? Assume a 20-year life cycle and use the present worth, annual cost, and rate of return methods.

3.5 An electrical energy audit indicates that lighting consumes 12.5 10^6 kWh per year. By using more efficient lighting fixtures, electrical energy savings of 8 percent can be achieved. The additional cost for these energy-efficient lighting fixtures is about \$90,000 (due mostly to the need for new ballasts). Assuming that the average energy charge is \$ 0.06 per kWh, is the expenditure justified based on a minimum rate of return (i.e., a discount rate) of 7.0 percent? Assume a 10-year life cycle and use the present worth, annual cost, and rate of return methods.

3.6 A residential building is heated with a 78 percent efficient gas-fired furnace (costing \$1,600). The annual heating load is estimated at 160 MMBtu. The cost of gas is \$6.00/MMBtu. For the discount rate of 5 percent and a 10-year cycle, determine:
(a) The life cycle of the existing heating system
(b) If it is worth considering a 90 percent efficient gas-fired furnace that costs \$2,800
(c) The variation of the life-cycle cost savings between the two systems [the existing system and that proposed in (b)] versus the life of the system for life cycles ranging from 3 to 20 years

Conclude.

3.7 A chiller consumes 9.5 × 105 kWh annually with an overall efficiency of 0.90 kW/ton. If this chiller is replaced by a more energy-efficient chiller (0.75 kW/ton) at a cost of \$95,000, determine:
(a) The simple payback period of the chiller replacement.
(b) If the expected life of the old chiller is 15 years, is it cost-effective to replace the chiller in an economy with a discount rate of 7 percent?

3.8 Two chillers are proposed to cool an office space. Each chiller has a rated capacity of 300 tons and is expected to operate 650 full-load equivalent hours per year. Chiller A has a standard efficiency with a COP of 3.0 and costs \$160,000 whereas chiller B is more efficient with a COP of 3.5 at a cost of \$210,000. The electricity cost is estimated to be \$0.08/kWh. For a lifetime of 20 years and a discount rate of 5 percent,

(a) Estimate the payback time for using chiller B instead of chiller A.
(b) Calculate the life-cycle costs for both chillers. Conclude.
(c) Estimate the rate of return for using chiller B instead of chiller A.
(d) Determine the highest cost for chiller B at which chiller A is more competitive.
(e) Determine the cut-off electricity price at which both chillers A and B have the same life-cycle cost.

Assume that the average cost of electricity is \$0.06/kWh.

4

Energy Analysis Tools

4.1 Introduction

To analyze energy consumption and estimate the cost-effectiveness of energy conservation measures, an auditor can use myriad calculation methods and simulation tools. The existing energy analysis methods vary widely in complexity and accuracy. To select the appropriate energy analysis method, the auditor should consider several factors including speed, cost, versatility, reproducibility, sensitivity, accuracy, and ease of use (Sonderegger, 1985). There are hundreds of energy analysis tools and methods that are used worldwide to predict the potential savings of energy conservation measures. In the United States, the DOE provides an up-to-date listing of selected building energy software (DOE, 2009).

Generally, the existing energy analysis tools can be classified into either forward or inverse methods. In the forward approach as depicted in Figure 4.1, the energy predictions are based on a physical description of the building systems such as geometry, location, construction details, and HVAC system type and operation. Most of the existing detailed energy simulation tools such as DOE-2, TRNSYS, and EnergyPlus follow the forward modeling approach. In the inverse approach illustrated in Figure 4.2, the energy analysis model attempts to deduce representative building parameters [such as the building load coefficient (BLC), the building base-load, or the building time constant] using existing energy use, weather, and relevant performance data. In general, the inverse models are less complex to formulate than the forward models. However, the flexibility of inverse models is typically limited by the formulation of the representative building parameters and the accuracy of the building performance data. Most of the existing inverse models rely on regression analysis [such as the variable-base degree-day models (Fels, 1986), the change-point model (Kissock, Reddy, and Claridge, 1998)] tools, or the connectionist approach (Kreider et al., 1997) to identify the building parameters.

It should be noted that tools based on the forward or inverse approaches are suitable for other applications. Among the common applications are verification of energy savings actually incurred from energy conservation measures (for more details about this application, the reader is referred to Chapter 16), diagnosis of equipment malfunctions, and efficiency testing of building energy systems.

Energy analysis tools can also be classified based on their ability to capture the dynamic behavior of building energy systems. Thus, energy analysis tools can use either steady-state or dynamic modeling approaches. In general, the steady-state models are sufficient to analyze seasonal or annual building energy performance. However, dynamic models may be required to assess the transient effects of building energy systems such as those encountered for thermal energy storage systems and optimal start controls.

In this chapter, selected energy analysis tools commonly used in the United States and Europe are described. These tools are grouped into three categories:

- *Ratio-based methods,* which are preaudit analysis approaches that rely on building energy/cost densities to quickly evaluate building performance.

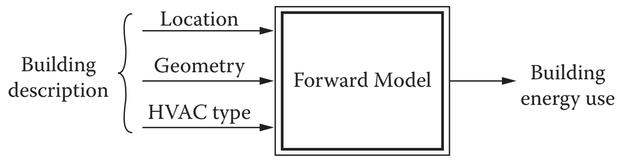


FIGURE 4.1 Basic approach of a typical forward energy analysis model.

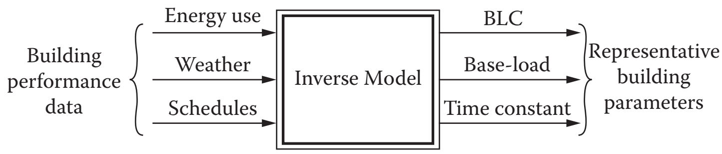


FIGURE 4.2 Basic approach of a typical inverse energy analysis model.

- *Inverse methods,* which use both steady-state and dynamic modeling approaches and include variable-base degree-day methods.
- *Forward methods,* which include either steady-state or dynamic modeling approaches and are often the bases of detailed energy simulation computer programs.

4.2 Ratio-Based Methods

4.2.1 Introduction

The ratio-based methods are not really energy analysis tools but rather preaudit analysis approaches to determine a specific energy or cost indicators for the building. These energy/cost building indicators are then compared to reference performance indices obtained from several other buildings with the same attributes. The end-use energy or consumption ratios can provide useful insights on some potential problems within the building such as leaky steam pipes, inefficient cooling systems, or high water usage. In particular, the building energy densities or ratios can be useful:

- To determine if the building has high energy consumption and to assess if an energy audit of the building would be beneficial.
- To assess if a preset energy performance target has been achieved for the building. If not, the energy ratio can be used to determine the magnitude of the required energy use reduction to reach the target.
- To estimate typical consumption levels for fuel, electricity, and water to be expected for new buildings.
- To monitor the evolution of energy consumption of buildings and estimate the effectiveness and profitability of any energy management program carried out following an audit.

To estimate meaningful reference ratios, large databases have to be collected. Typically, data for thousands of similar buildings and facilities is required to estimate reference indicators. This data has to be screened in order to eliminate any erroneous or implausible data points. A number of statistical analysis procedures can be carried out to screen the databases. Common statistical methods include:

(i) Elimination of any data point that has a value which is not part of an interval bounded by 2.5 times the standard deviation centered on the average of the entire data set
(ii) Elimination of any data point that has a value below the 10th percentile or above the 90th percentile

When new data points are added to the database, the screening process of the data should be repeated. In particular, any data points previously eliminated should be reinstated in the database. To obtain up-to-date reference ratios, the databases have to be regularly screened and updated. It is recommended that the databases be updated at least once every five years for energy ratios, and once every two years for the cost ratios.

The ratio-based methods are approaches used only for preaudit energy diagnostic or energy screening. These methods do not provide sufficient information to conduct a complete and detailed energy analysis of the building and its systems. However, energy screening tools using ratio-based methods can be useful, if properly employed, to assess the energy efficiency of buildings quickly. Examples of screening tools developed in the United States include BEST (Building Energy Screening Tool) and Scheduler (developed for the Energy Star Buildings program).

4.2.2 Types of Ratios

Energy or cost ratios are typically computed as a fraction made up of a numerator and a denominator. A set of variables is used in the numerator. Another set of specific parameters is suitable for the denominator. For energy ratios, the variables that are typically used for the numerator include:

- Total building energy use (i.e., including all end-uses). This total energy use can be expressed in kWh or MMBtu.
- Building energy consumption by end-use (i.e., heating, cooling, and lighting).
- Energy demand (in kW).

For the cost ratios, a monetary value (specifically for the energy expenditure or for the overall building operation) is typically used for the numerator.

For the denominator, several variables can be used depending on the building type and the main goal intended for the computed ratios. Some of the common variables, used in the denominator of energy and cost ratios, are:

- Surface area or space volume (such as heating area or conditioned volume in offices)
- Building users (in collective buildings such as hotels or schools)
- Degree-day [generally with 65°F (18°C) as a base temperature]
- Units of productions (especially for manufacturing facilities or restaurants)

In general, annual or seasonal values are used to obtain the energy or cost ratios. However, daily or monthly ratios can be considered. The monthly variations of energy ratios are often referred to as building signatures.

4.2.3 Examples of Energy Ratios

Generally, meaningful energy ratios require careful analysis and screening of the data. For instance, it is important to consider effects such the climate and building function when estimating energy ratios. Sources of building energy data are typically difficult to obtain. In the United States, the Energy Information Administration (EIA) annually provides statistical data about the energy use of various building types. In other countries, it is generally very difficult to obtain such data even through governmental agencies. Table 4.1 provides some energy ratios for selected commercial and institutional building types in the United States and France.

It should be stressed that energy ratios such as those provided in Table 4.1 should be used only as generic indicators of typical energy use for the listed buildings or facilities. Energy ratios specified by climate zone, type of HVAC system, or building size may be required for an adequate energy screening or preaudit energy analysis.

TABLE 4.1 Energy Ratio (or Energy Use Intensity) by Principal Building Activity in kWh/m²

Major Building Activity	France[a]	United States[b]
Office	296	300
Education	160	270
Health care	283	610
Lodging	230	325
Food service	416	649
Mercantile and service	264	250
Sports	278	NA[c]
Public assembly	NA	305
Warehouse and storage	NA	147

[a] *Source*: ADEME based on 2001 energy consumption (ADEME, 2005).
[b] *Source:* EIA using 2003 data (EIA, 2006).
[c] Not available.

4.3 Inverse Modeling Methods

As discussed in the introduction, methods using the inverse modeling approach rely on existing building performance data to identify a set of building parameters. Inverse modeling methods can be valuable tools in improving building energy efficiency. In particular, the inverse models can be used to:

- Help detect malfunctions by identifying time periods or specific systems with abnormally high energy consumption.
- Provide estimates of expected savings from a defined set of energy conservation measures.
- Verify the savings achieved by energy retrofits.

Typically, regression analyses are used to estimate the representative parameters for the building or its systems (such as building load coefficient or heating system efficiency) using measured data. In general, steady-state inverse models are based on monthly or daily data and include one or more independent variables. Dynamic inverse models are usually developed using hourly or subhourly data to capture any significant transient effect such as the case where the building has a high thermal mass to delay cooling or heating loads.

4.3.1 Steady-State Inverse Models

These models generally attempt to identify the relationship between the building energy consumption and selected weather-dependent parameters such as monthly or daily average outdoor temperatures, degree-hours, or degree-days. As mentioned earlier, the relationship is identified using statistical methods (based on linear regression analysis). The main advantages of the steady-state inverse models are:

- *Simplicity:* Steady-state inverse models can be developed based on a small dataset such as energy data obtained from utility bills.
- *Flexibility:* Steady-state inverse models have a wide range of applications. They are particularly valuable in predicting the heating and cooling energy end-uses for both residential and small commercial buildings.

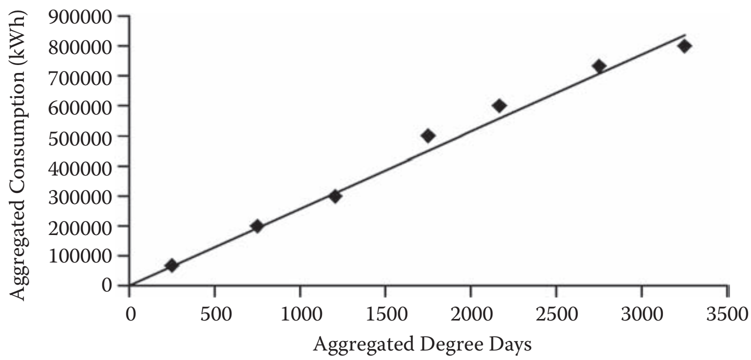


FIGURE 4.3 Typical application of cumulative (or aggregated) degree-day method.

However, steady-state inverse models have some limitations because they cannot be used to analyze transient effects such as thermal mass effects and seasonal changes in the efficiency of the HVAC system.

Steady-state inverse models are especially suitable for measurement and verification (M&V) of energy savings accrued from energy retrofits. Chapter 16 provides a more detailed discussion about the inverse models that are commonly used for M&V applications. In this section, only simplified methods based on steady-inverse modeling are briefly presented. These simplified models have been used to determine the energy impact of selected energy-efficiency measures and are based on the degree-day methods. Two simplified energy analysis approaches are briefly outlined:

1. The cumulative (or aggregated) degree-day method consists in correlating (using a linear regression analysis) the cumulative building energy use to the cumulative degree-days (using a reference temperature of 18°C [65°F]). Figure 4.3 illustrates the basic concept of the cumulative degree-day method. This method is used in some European countries to monitor the variation of building energy use throughout the heating season. In particular, the cumulative degree-day approach helps us to visualize easily any changes in the building energy use pattern attributed to energy retrofit measure through the slope of the regression line. Any improvement in the building thermal performance (such as the addition of thermal insulation or increase in the efficiency of the heating system) will reduce the slope.
2. The variable-base degree-day method uses a linear regression analysis to estimate the building balance temperature. Chapter 6 discusses the details of the variable-base degree-day method. Several energy analysis tools and software have been developed using one form of the variable-base degree-day method. Among these tools are PRISM (Princeton Scorekeeping Method) used in the United States (Fels, 1986) to analyze energy use for residential and small commercial buildings, and the ANAGRAM (*ANalyse GRAphique Mensuelle des consommations*) software developed in France by GDF (1985) specifically to estimate monthly heating energy use for buildings. Both of these programs are briefly described below:

4.3.1.1 ANAGRAM Method

Using the ANAGRAM approach, the annual building heating energy use is first calculated as follows:

$$E_H = 24 \times \frac{\mathrm{BLC_V}}{\eta_\mathrm{H}} \times \mathrm{V_B} \times \mathrm{DD_H} \times \mathrm{I} \tag{4.1}$$

where

E_H is the annual building heating energy consumption (in kWh).

$\mathrm{BLC_V}$ is the building loss coefficient based on the building volume (in kW/m^3°C).

η_H is the average seasonal energy efficiency of the heating system.
V_B is the heated building volume (m^3).
DD_H is the heating degree-days (based on 18°C).
I is a correction factor to account for the effects of night setback coefficient in reducing the building heating load (if there is no night setback, $I = 1$).

Then, ANAGRAM proceeds to calculate the monthly heating energy use using the following expression:

$$E_{H,m} = 24 \times \frac{BLC_V}{\eta_H} \times V_B \times I \times [DD_{H,m} - (18 - T_b) \times 30] \tag{4.2}$$

where
$E_{H,m}$ is the monthly building heating energy use (in kWh).
$DD_{H,m}$ is the heating degree-days (based on 18°C).
T_b is the building balance temperature. T_b is defined as the outdoor temperature for which the building does not need any heating.
BLC_V, I, V_B, η_H have the same definition as in Eq. (4.1).
30 is the number of days in a month (which corresponds to the number of days included in the GDF utility bills).

Thus, a linear regression is carried out to correlate the monthly building energy use to the monthly degree-days (based on 18°C using only heating season data). This regression analysis provides an estimation of the building balance temperature T_b and the ratio of BLC_V/η_H and is the average seasonal energy efficiency of the heating system. Example 4.1 illustrates how the ANAGRAM approach is used to analyze monthly building energy use.

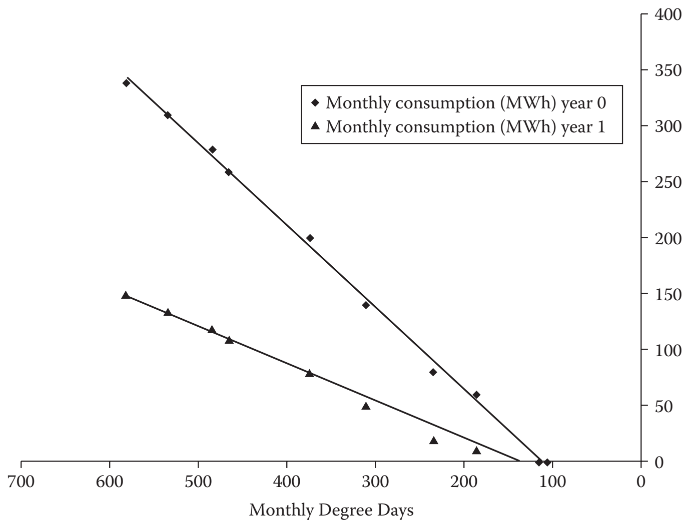
Example 4.1

ILLUSTRATION OF THE ANAGRAM METHOD

Using the ANAGRAM approach, analyze the heating energy use data for a building having a heated volume of 15,000 m^3. Monthly energy consumption and degree-day data are provided below for two years, 0 and 1.

	Monthly Degree Days	Monthly Consumption (MWh) Year 0	Monthly Consumption (MWh) Year 1
September (S)	115	0	0
October (O)	235	80	20
November (N)	375	200	80
May (My)	185	60	10
March (M)	465	260	110
June (Ju)	105	0	0
January (J)	580	340	150
February (F)	535	310	135
December (D)	485	280	120
April (A)	310	140	50

The results of the linear regression analysis of the monthly heating energy consumption with degree days are provided in the graph below. The regression analysis is based on the model described by Eq. (4.2) with $I = 1$(no temperature setback).



For year 0, the regression line intersects the *x*-axis at degree-days $DD_{H.,m} = 120°C$-days. Thus, the balance temperature T_b can be estimated as follows: $[DD_{H,m} - (18 - T_b) \times 30] = 0$. Therefore, for year 0, the balance temperature is:

$$T_b = 18 - 120/30 = 14°C$$

Using the slope of the regression line (which is 570 kWh/°C-days), the ratio BLC_V/η_H can be estimated using the fact that:

$$570 \text{ kWh/°C-days} = 24 * BLC_V/\eta_H * V_B,$$

Because $V_B = 15{,}00 \text{ m}^3$, it is estimated that:

$$BLC_V/\eta_H = 1.\ 58 \text{ W/m}^3\text{°C}$$

A similar analysis can be carried out for year 1. It is found that between year 0 and 1, the balance temperature has decreased from 14°C to 13.5°C, and the ratio BLC_V/η_H is reduced from 1.58 W/m³°C to 0.70 W/m³°C. It is most likely that this reduction in both T_b and BLC_V/η_H is attributed to improvement in the energy efficiency of the building envelope.

4.3.1.2 PRISM Method

PRISM correlates the building energy use per billing period to heating or cooling degree-days (obtained for the billing period). Thus the energy consumption is estimated for each billing period using the following expression:

$$E_{H/C} = 24 \times \frac{BLC}{\eta_{H/C}} \times DD_{H/C}(T_{b,\,H/C}) + E_{base,\,H/C} \tag{4.3}$$

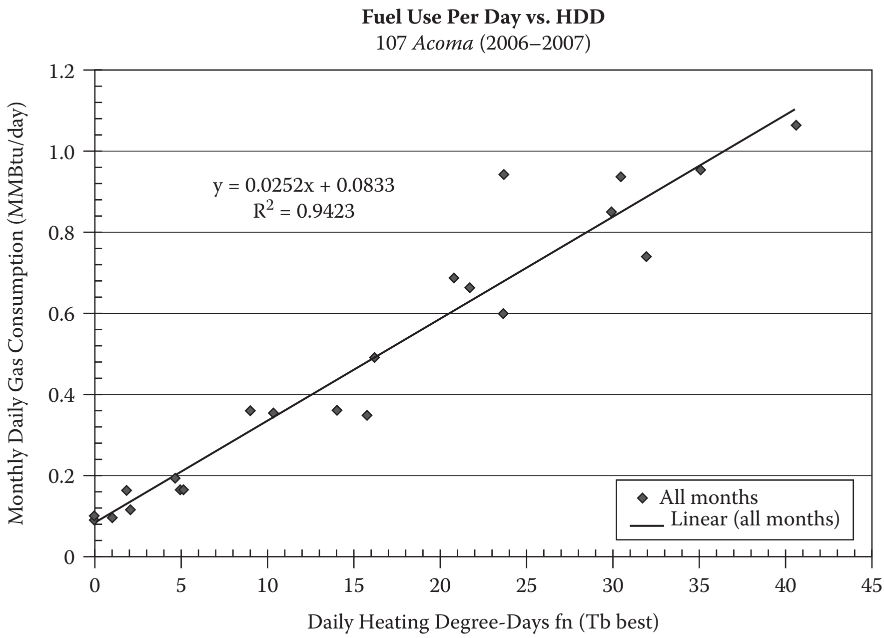


FIGURE 4.4 Analysis of gas consumption as a function of monthly heating degree-days. (Kalinic, 2009.)

where

EH/C represents the annual building energy use during heating or cooling season.

BLC is the building loss coefficient.

$\eta_{H/C}$ is the average seasonal energy efficiency of the heating or the cooling system.

T_b, H/C is the building balance temperature for heating or cooling energy use.

DDH is the heating or cooling degree-days (based on the balance temperature). Chapter 6 discusses in more detail how variable base degree-days are defined. For heating, for instance, the heating degree-days are estimated as follows: $DD_H(T_b) = \Sigma_j[T_b - T_{o,j}]^+$ with $T_{o,j}$ the outdoor air temperature at day *j*.

E base, H/C is the base-load for building energy use. It represents nonheating or noncooling energy use.

Through regression analysis, the balance temperature and the building load coefficient can be determined (assuming the heating or cooling system efficiency is known). With these two parameters determined, the PRISM tool can be used to establish an energy use model for the building and to determine any energy savings attributed to measures that affect one of the three parameters, balance temperature, building load coefficient, or heating/cooling system efficiency. Figure 4.4 illustrates the PRISM method for the analysis of natural gas usage as a function of the heating degree-day for a residential building. One variation of the PRISM approach represented by Eq. (4.3) is to use the average outdoor temperature instead of the variable-base degree-day. This method is described in Chapter 6.

4.3.2 Dynamic Models

Steady-state inverse models are only suitable for predicting long-term building energy use. Therefore, energy use data is collected for a relatively long time period (at least one season or one year) to carry out the regression analysis. On the other hand, dynamic inverse models can be used to predict short-term building energy use variations using data collected for a short period of time such as one week. Generally, a dynamic inverse model is based on a building thermal model that uses a specific set of parameters. These building model parameters are typically identified using some form of regression analysis. Chapter 16 discusses other approaches to developing dynamic inverse models suitable for predicting building energy use.

An example of a dynamic model relating building cooling energy use to the outdoor air temperatures at various time steps (typically hours) is presented by Eq. (4.4):

$$E_C^n + b_1 E_C^{n-1} + \ldots + b_N E_C^{n-N} = a_0 T_o^n + a_1 T_o^{n-1} + \ldots + a_M T_0^{\,n-M} \tag{4.4}$$

Other examples of dynamic inverse models include equivalent thermal network analysis, Fourier series models, and artificial neural networks. These models are capable of capturing dynamic effects such as building thermal mass dynamics. The main advantages of the dynamic inverse models include the ability to model complex systems that depend on several independent parameters. Their disadvantages include their complexity and the need for more detailed measurements to fine-tune the model. Unlike steady-state inverse models, dynamic inverse models usually require a high degree of user interaction and knowledge of the modeled building or system.

4.4 Forward Modeling Methods

Forward modeling methods are generally based on a physical description of the building energy systems. Typically, forward models can be used to determine the energy end-uses as well as predict any energy savings incurred from energy conservation measures. Selected existing U.S. energy analysis tools that use the forward modeling approach are described in the following sections. For a more detailed discussion, the reader is referred to the *ASHRAE Handbook of Fundamentals* (ASHRAE, 2009).

4.4.1 Steady-State Methods

Steady-state energy analysis methods that use the forward modeling approach are generally easy to use because most of the calculations can be performed by hand or using spreadsheet programs. Two types of steady-state forward tools can be distinguished: degree-day methods and bin methods.

4.4.2 Degree-Day Methods

The degree-day methods use seasonal degree-days computed at a specific set-point temperature (or balance temperature) to predict the energy use for building heating. Typically, these degree-day methods are not suitable for predicting building cooling loads. In the United States, the traditional degree-day method using a base temperature of 65°F has been replaced by the variable-base degree-day method and is applied mostly to residential buildings. In Europe, heating degree-day methods using 18°C as the base temperature are still used for both residential and commercial buildings.

The variable-base degree-day methods predict seasonal building energy used for heating with one variation of the following formulation:

$$FU = \frac{24.BLC.f.\ DD_H(T_b)}{\nu\eta_H} \tag{4.5}$$

where

FU represents the fuel use (gas, fuel oil, or electricity depending on the heating system).

BLC is the building loss coefficient including transmission and infiltration losses through the building envelope. Chapter 6 indicates how BLC can be computed.

f is a correction factor to include various effects such as the part-load performance of the heating system, night setback effects, and free heat gains.

T_b is the building heating balance temperature. Again, the reader is referred to Chapter 6 which describes the procedure to calculate the balance temperature based on the building description and indoor temperature settings

DDH(T_b) is the heating degree-days calculated at the balance temperature T_b. Chapter 6 provides two simplified methods to estimate degree-days at any balance temperature. Monthly and annual degree-days for selected balance temperatures are provided in the appendices.

Variable-base degree-day methods generally provide good predictions of the fuel use for buildings dominated by transmission loads (i.e., low-rise buildings). However, they are not recommended for buildings dominated by internal loads or with involved HVAC system operation strategies.

4.4.3 Bin Methods

Another energy analysis method that uses the forward approach but is based on steady-state modeling of building energy systems is the bin method (Knebel et al., 1983). The bin method is similar to the variable-base degree-day method but relies on bin weather data to estimate total building heating or cooling energy consumption. In the United States, a number of HVAC engineers use the bin method to perform a variety of energy analyses. Moreover, computer energy simulation tools based on the bin methods have been developed (DOE, 2010). The simulation tools based on the bin method are typically appropriate for residential or small commercial buildings.

In the classical bin method, only the outdoor temperatures are grouped into bins of equal size, typically 5°F (2.8°C) bins. The number of hours of occurrence is determined for each bin. For other weather variables, only average values coincident to each temperature bin are determined. The resulting weather data from the classical bin method is often referred to as one-dimensional bin weather data. Table 4.2 illustrates one-dimensional weather data obtained for Atlanta, Georgia. In addition to the outdoor dry-

TABLE 4.2 Classical (or One-Dimensional) Bin Weather Data for Atlanta, Georgia

Average of Outdoor Dry-Bulb Temperature Bin (°F)	Number of Hours of Occurrence	Average Coincident Humidity Ratio (lb/lb)
15	1	0.0020
20	42	0.0020
25	154	0.0020
30	291	0.0025
35	354	0.0031
40	641	0.0038
45	623	0.0044
50	665	0.0053
55	741	0.0065
60	882	0.0083
65	905	0.0100
70	1225	0.0128
75	1000	0.0133
80	672	0.0137
85	421	0.0143
90	133	0.0156
95	19	0.0170

TABLE 4.3 Partial Listing of Two-Dimensional Bin Weather Data for Atlanta, Georgia

Average of Humidity Ratio Bin (lb/lb)	Average of Dry-Bulb Temperature Bin (oF)								
	50	55	60	65	70	75	80	85	90
0	0	0	0	0	0	0	0	0	0
0.0020	69	48	29	14	0	0	0	0	0
0.0040	229	183	123	54	35	11	1	0	0
0.0060	238	178	91	75	39	37	10	0	0
0.0080	129	210	196	122	71	85	28	14	0
0.0100	0	122	335	228	140	111	72	40	3
0.0120	0	0	108	312	202	131	156	53	5
0.0140	0	0	0	100	486	245	156	131	42
0.0160	0	0	0	0	237	289	184	142	57
0.0180	0	0	0	0	15	89	62	37	19
0.0200	0	0	0	0	0	2	3	4	7
0.0220	0	0	0	0	0	0	0	0	0

bulb temperature bins, Table 4.3 provides the average values for coincident humidity ratio. The current version of ASEAM software uses one-dimensional bin weather data.

The accuracy of the classical bin method is adequate only for buildings dominated by sensible heat loads and with no significant thermal mass effects. However, the classical bin method may not provide accurate energy predictions for buildings with high latent heat loads as reported by Harriman et al. (1999) and Cohen and Kosar (2000). To improve the accuracy of the bin method especially for buildings with significant latent loads, two-dimensional weather data bins were introduced by ASHRAE (1997). The two-dimensional (also referred to as joint-frequency) weather data bins are generated based on bins obtained for two variables (such as the dry-bulb temperature and humidity ratio) as presented in Table 4.3 using a partial dataset for Atlanta, Georgia.

The two-dimensional weather bin data can be created using hourly data such as TMY-2 files. A number of software are available to create these bins including the ASHRAE Weather Data Viewer developed by Colliver et al. (1998).

4.4.4 Dynamic Methods

Dynamic analytical models use numerical or analytical methods to determine energy transfer among various building systems. These models generally consist of simulation computer programs with hourly or subhourly) time steps to estimate adequately the effects of thermal inertia, due for instance to energy storage in the building envelope or its heating system. The important characteristic of the simulation programs is their capability to account for several parameters that are crucial for accurate energy use especially for buildings with significant thermal mass, thermostat setbacks or setups, explicit energy storage, or predictive control strategies. A typical calculation flowchart of detailed simulation programs is presented in Figure 4.5.

Detailed computer programs require a high level of expertise and are generally suitable to simulate large buildings with complex HVAC systems and involved control strategies that are difficult to model using simplified energy analysis tools.

In general, an energy simulation program requires a detailed physical description of the building (including building geometry, building envelope construction details, HVAC equipment type and operation, and occupancy schedules). Thermal load calculations are based on a wide range of algorithms depending on the complexity and the flexibility of the simulation program. To adequately estimate energy savings from energy-efficiency measures, energy simulation tools have to be calibrated using

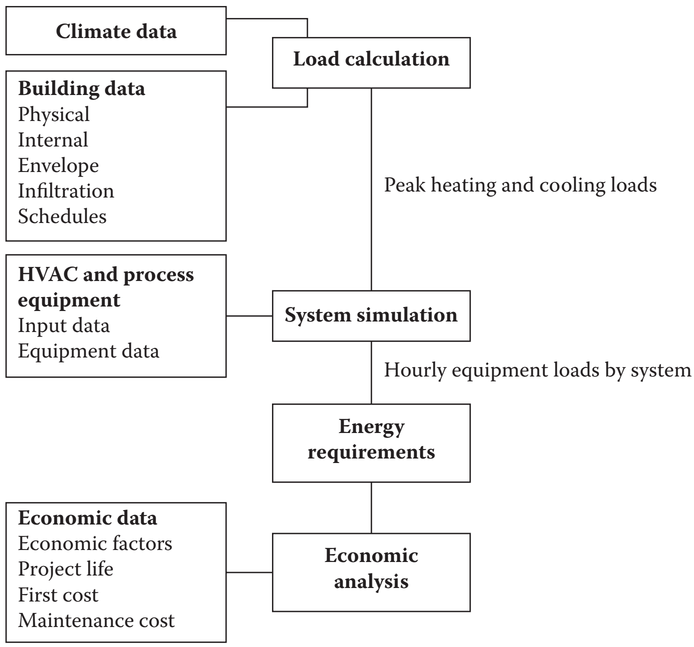


FIGURE 4.5 Flowchart of complete building model.

existing measured energy data (utility bills, for instance). A basic calibration procedure is discussed in detail in Chapter 16.

Although energy simulation programs are generally capable of modeling most of the building energy systems, they are often not sufficiently flexible and have inherent limitations. To select the appropriate energy simulation program, it is important that the user be aware of the capabilities of each simulation available. Some of the well-known simulation programs are briefly presented below:

- DOE-2 (version DOE-2.1). DOE-2 was developed at the Lawrence Berkeley National Laboratory (LBNL) by the U.S. Department of Energy and is widely used because of its comprehensiveness. It can predict hourly, daily, monthly, or annual building energy use. DOE-2 is often used to simulate complex buildings. Figure 4.6 illustrates a typical zoning scheme used to model the office building of the case study in Chapter 1 with DOE-2. Typically, significant efforts are required to create DOE-2 input files using a programming language called the Building Description Language (BDL). Several tools are currently available to facilitate the process of developing DOE-2 input files. Among energy engineers and professionals, DOE-2 has become a standard building energy simulation tool in the United States and several other countries using interfaces such eQUEST and VisualDOE. Figure 4.7 shows a 3-D rendering of a residential building modeled using eQUEST.
- BLAST (building loads analysis and systems thermodynamics). This program enables the user to predict the energy use of the whole building under design conditions or for long-term periods. The heating/cooling load calculations implemented in BLAST are based on a heat balance approach (instead of the transfer function technique adopted by DOE-2). Therefore, BLAST can be used to analyze systems such as radiant heating or cooling panels that cannot be adequately modeled by DOE-2.
- EnergyPlus. This builds on the features and capabilities of both DOE-2 and BLAST. EnergyPlus uses new integrated solution techniques to correct one of the deficiencies of both BLAST and DOE-2: the inaccurate prediction of space temperature variations. Accurate prediction of space temperatures is crucial to properly analyze energy-efficient systems. For instance, HVAC system

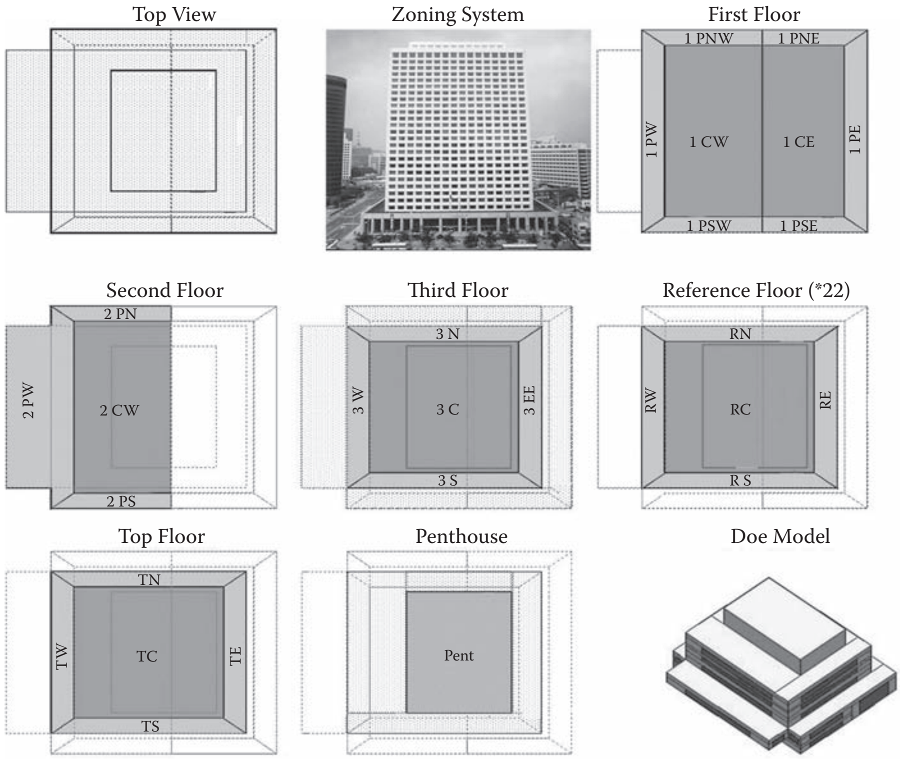


FIGURE 4.6 Space zoning used to model an office building using DOE-2.

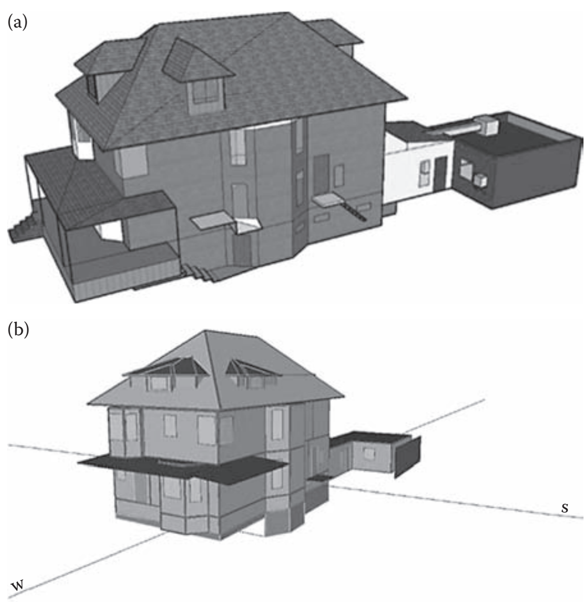


FIGURE 4.7 Three-dimensional models for a residential building (a) CAD rendering and (b) eQUEST rendering.

performance and occupant comfort are directly affected by space temperature fluctuations. Moreover, EnergyPlus has several features that should aid engineers and architects in evaluating a number of innovative energy-efficiency measures that cannot be simulated adequately with either DOE-2 or BLAST. These features include:
  - Free cooling operation strategies using outdoor air
  - Realistic HVAC systems controls
  - Effects of moisture adsorption in building elements
  - Indoor air quality with a better modeling of contaminant and air flows within the building
- Several interfaces for EnergyPlus have been developed over the last few years. A complete list of the interfaces is periodically updated on the EnergyPlus website (EnergyPlus, 2009).
- HAP (1.17 or 3.05) is a computer tool developed by Carrier and is currently used to calculate cooling and heating loads in order to design air-conditioning systems in buildings with multiple zones (up to 20 zones). It can also perform seasonal and annual energy analyses and evaluate potential energy savings from several energy conservation measures.
- TRNSYS provides a flexible energy analysis tool to simulate a number of energy systems utilizing user-defined modules. A good knowledge of computer programming (Fortran) is required to properly use the TRNSYS simulation tool.

4.5 Summary

In this chapter, selected energy analysis tools are described with a brief discussion of the general analysis procedures used by these tools. The advantages as well as the limitations of the presented modeling approaches are outlined. The auditor should select the proper tool to carry out the energy analysis of the building and to estimate the potential energy and cost savings for retrofit measures. Throughout the following chapters, simplified analysis methods are presented to estimate energy savings.

5

Electrical Systems

5.1 Introduction

In most buildings and industrial facilities, electrical systems consume a significant part of the total energy use. Table 5.1 compares the part of electricity consumption in three sectors (residential, commercial, and industrial) for both the United States and France which is representative of most western European countries. It is clear that in the United States, electrical energy is used more significantly in commercial and residential buildings than in industrial facilities where fossil fuels (such as coal, oil, and natural gas) are predominantly used.

For residential buildings, lighting and heating, ventilating, and air conditioning (HVAC) each account for approximately 20 percent of the total U.S. electricity use. Refrigerators represent another important energy end-use in the residential sector with about 16 percent of electricity. For the commercial sector as a whole, lighting accounts for over 40 percent whereas HVAC accounts for 11 percent of the total electricity use. However, for commercial buildings with space conditioning, HVAC is one of the major electricity end-uses and can be more energy intensive than lighting. Moreover, computers and other office equipment (such as printers, copiers, and facsimile machines) are becoming an important electric energy end-use in office buildings.

In this chapter, a brief description is given of electric energy end-uses such as motors with focus on some measures to reduce electrical energy use. First, a brief review of basic characteristics of an electrical system is provided to highlight the major issues that should be considered when designing, analyzing, or retrofitting an electrical system.

5.2 Review of Basics

5.2.1 Alternating Current Systems

For a linear electrical system subject to an alternating current (AC), the time variation of the voltage and current can be represented as a sine function:

$$V(t) = V_m \cos\omega t \tag{5.1}$$

$$i(t) = I_m \cos(\omega t - \phi) \tag{5.2}$$

where

V_m and I_m are the maximum instantaneous values of voltage and current, respectively. These maximum values are related to the effective or root mean square (rms) values as follows:

$$V_m = \sqrt{2} * V_{rms} = 1.41 * V_{rms}$$

TABLE 5.1 Percentage Share of Electricity in the Total Energy Use in Three Sectors for the United States[a] and France[b]

Sector	United States (%)	France (%)
Residential buildings	41	27
Commercial buildings	55	40
Industrial facilities	15	30

[a] *Source:* EIA. Annual Energy Review, Department of Energy, Energy Information Administration, http://www.doe.eia.gov, 2009.
[b] *Source:* ADEME (2007)

$$I_m = \sqrt{2} * I_{rms} = 1.41 * I_{rms}$$

In the United States, the values of V_{rms} are typically 120 V for residential buildings or plug-load in the commercial buildings, 277 V for lighting systems in commercial buildings, and 480 V for motor loads in commercial and industrial buildings. Higher voltages can be used for some power-intensive industrial applications.

ω is the angular frequency of the alternating current and is related to the frequency f as follows:

$$\omega = 2\pi f$$

In the United States, the frequency f is 60 Hz, that is, 60 pulsations or oscillations in one second. In other countries, the frequency of the alternating current is $f = 50$ Hz.

ϕ is the phase lag between the current and the voltage. In the case where the electrical system is a resistance (such as an incandescent lamp), the phase lag is zero and the current is on phase with the voltage. If the electrical system consists of a capacitance load (such as a capacitor or a synchronous motor), the phase lag is negative and the current is in advance relative to the voltage. Finally, when the electrical system is dominated by an inductive load (such as a fluorescent fixture or an induction motor), the phase lag is positive and the current lags the voltage.

Figure 5.1 illustrates the time variation of the voltage for a typical electric system. The concept of root mean square (also called effective value) for the voltage V_{rms} is also indicated in Figure 5.1. It should be noted that in the United States the cycle for the voltage waveform repeats itself every 1/60 s (because the frequency is 60 Hz).

The instantaneous power *p(t)* consumed by the electrical system operated on a one-phase AC power supply can be calculated using Ohm's law:

$$p(t) = v(t).i(t) = V_m I_m \cos\omega t.\cos(\omega t - \phi) \tag{5.3}$$

The above equation can be rearranged using some basic trigonometry and the definition of the rms values for voltage and current:

$$p(t) = V_{rms}.I_{rms}\left(\cos\phi.(1+\cos 2\omega t) + \sin\phi.\sin 2\omega t\right) \tag{5.4}$$

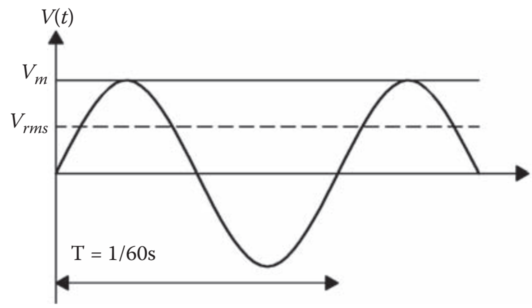


FIGURE 5.1 The voltage waveform and the concept of V_{rms}.

Two types of power can be introduced as a function of the phase lag angle ϕ, the real power P_{R}, and the reactive power P_{X}, as defined below:

$$P_R = V_{rms}.I_{rms}\cos\phi \tag{5.5}$$

$$P_X = V_{rms}.I_{rms}\sin\phi \tag{5.6}$$

Note that both types of power are constant and are not functions of time. To help understand the meaning of each power, it is useful to note that the average of the instantaneous power actually consumed by the electrical system over one period is equal to P_{R}:

$$\bar{p} = \frac{1}{T}\int_0^T p(t)dt = P_R \tag{5.7}$$

Therefore, P_{R} is the actual or real power consumed by the electrical system over its operation period (which consists typically of a large number of periods $T = 1/2\pi f$). P_{R} is typically called real power and is measured in kW. Meanwhile, P_{X} is the power required to produce a magnetic field to operate the electrical system (such as induction motors). P_{X} is stored and then released by the electrical system; this power is typically called reactive power and is measured in kVAR. A schematic diagram is provided in Figure 5.2 to help illustrate the meaning of each type of power.

Although the user of the electrical system actually consumes only the real power, the utility or the electricity provider has to make available to the user both the real power P_{R} and the reactive power P_{X}. The algebraic sum of P_{R} and P_{X} constitutes the total power P_{T}. Therefore, the utility has to know in addition to the real power needed by the customer, the magnitude of the reactive power, and thus the total power.

As mentioned earlier, for a resistive electrical system, the phase lag is zero and thus the reactive power is also zero [see Eq. (5.6)]. Unfortunately, for commercial buildings and industrial facilities, the electrical systems are not often resistive and the reactive power can be significant. In fact, the higher the phase lag angle ϕ, the larger is the reactive power P_{X}. To illustrate the importance of the reactive power relative to the real power P_{R} and the total power P_{T} consumed by the electrical system, a power triangle is typically used to represent the power flow as shown in Figure 5.3.

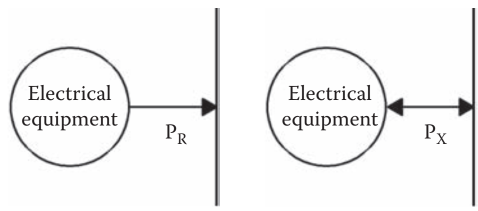


FIGURE 5.2 The direction of electricity flow for (a) real power, and (b) reactive power.

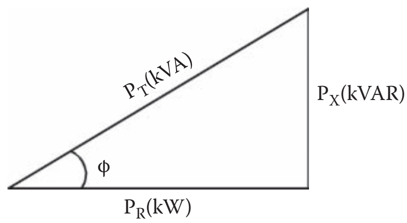


FIGURE 5.3 Power triangle for an electrical system

In Figure 5.3, it is clear that the ratio of the real power to the total power represents the cosine of the phase lag. This ratio is widely known is the power factor *pf* of the electrical system:

$$pf = \frac{P_R}{P_T} = \cos\phi \tag{5.8}$$

Ideally, the power factor has to be as close to unity as possible (i.e., $pf = 1.0$). Typically, however, power factors above 90 percent are considered to be acceptable. If the power factor is low, that is, if the electrical system has a high inductive load, capacitors can be added in parallel to reduce the reactive power as illustrated in Figure 5.4.

5.2.2 Power Factor Improvement

As mentioned in the previous section, the reactive power has to be supplied by the utility even though it is not actually recorded by the power meter (as real power used). The magnitude of this reactive power increases as the power factor decreases. To account for the loss of energy due to the reactive power, most utilities have established rate structures that penalize any user that has a low power factor. The penalty is imposed through a power factor clause set in the utility rate structures as discussed in Chapter 2. Significant savings in the utility costs can be achieved by improving the power factor. As illustrated in Figure 5.4, this power factor improvement can be obtained by adding a set of capacitors connected in parallel to the electrical system. The size of these capacitors P_C is typically measured in kVAR (the same unit as the reactive power) and can be determined as indicated in Figure 5.5 using the power triangle analysis:

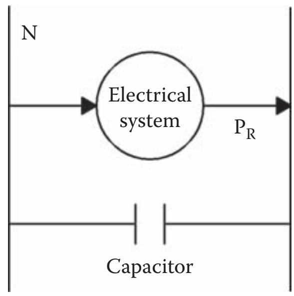


FIGURE 5.4 The addition of a capacitor can improve the power factor of an electrical system.

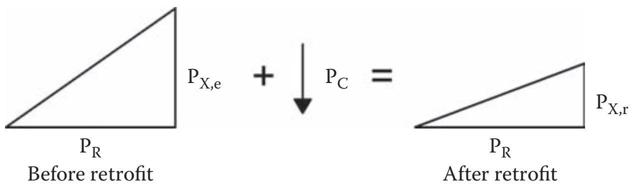


FIGURE 5.5 Effect of adding capacitors on the power triangle of the electrical system.

$$P_C = P_{Xe} - P_{Xr} = P_R.(\tan\phi_e - \tan\phi_r) \tag{5.9}$$

where

P_{Xe} and P_{Xr} are the reactive power before retrofit (existing conditions) and after retrofit (retrofitted conditions), respectively.

P_C is the reactive power of the capacitor to be added.

ϕ_e and ϕ_r are phase lag angles before retrofit (existing conditions) and after retrofit (retrofitted conditions), respectively.

Using the values of power factor before and after the retrofit, the size of the capacitors can be determined:

$$P_C = P_R.\left[\tan(\cos^{-1} pf_e) - \tan(\cos^{-1} pf_r)\right] \tag{5.10}$$

The calculations of the cost savings incurred from power factor improvement depend on the utility rate structure. In most all the rate structures, one of three options is used to assess the penalty for low power factors. These three options are summarized below. Basic calculation procedures are typically performed to estimate the annual cost savings in the utility bills:

1. *Modified Billing Demand:* In this case, the demand charges are increased in proportion to a fraction by which the power factor is less than a threshold value. The size for the capacitors should be selected so the system power factor reaches at least the defined threshold value.
2. *Reactive Power Charges:* In this case, charges for reactive power demand are included as part of the utility bills. In this option, the size of the capacitors should be ideally determined to eliminate this reactive power (so that the power factor is unity).

3. *Total Power Charges:* The penalty charges are imposed based on the total power required by the building/facility. Again, capacitors should be sized so the power factor is equal to unity.

The calculations of the cost savings due to power factor improvement are illustrated in Example 5.1.

Example 5.1

Consider a building with a total real power demand of 500 kW with a power factor of pf_e = 0.70. Determine the required size of a set of capacitors to be installed in parallel with the building service entrance so that the power factor becomes at least pf_r = 0.90

Solution

The size in kVAR of the capacitor is determined using Eq. (5.10):

$$P_C = 500.\left[\tan(\cos^{-1} 0.70) - \tan(\cos^{-1} 0.90)\right] = 268$$

Thus a capacitor rated at 275 kVAR can be selected to ensure a power factor for the building electrical system to be higher than 0.90.

5.3 Electrical Motors

5.3.1 Introduction

Motors consume over 50 percent of the electricity generated in the United States. In large industrial facilities, motors can account for as much as 90 percent of the total electrical energy use. In commercial buildings, motors can account for more than 50 percent of the building electrical load.

Motors convert electrical energy to mechanical energy and are typically used to drive machines. The driven machines can serve myriad purposes in the building including moving air (supply and exhaust fans), moving liquids (pumps), moving objects or people (conveyors, elevators), compressing gases (air compressors, refrigerators), and producing materials (production equipment). To select the type of motor to be used for a particular application, several factors have to be considered including:

(a) The form of the electrical energy that can be delivered to the motor: direct current (DC) or alternating current (AC), single or three phase
(b) The requirements of the driven machine such as motor speed and load cycles
(c) The environment in which the motor is to operate: normal (where a motor with an open-type ventilated enclosure can be used), hostile (where a totally enclosed motor must be used to prevent outdoor air from infiltrating the motor), or hazardous (where a motor with an explosion-proof enclosure must be used to prevent fires and explosions)

The basic operation and the general characteristics of AC motors are discussed in the following sections. In addition, simple measures are described to improve the energy efficiency of existing motors.

5.3.2 Overview of Electrical Motors

There are basically two types of electric motors used in buildings and industrial facilities: (i) induction motors and (ii) synchronous motors. Induction motors are the more common type, accounting for about 90 percent of the existing motor horsepower. Both types use a motionless stator and a spinning rotor to convert electrical energy into mechanical power. The operation of both types of motor is relatively simple and is briefly described below.

Alternating current is applied to the stator, which produces a rotating magnetic field in the stator. A magnetic field is also created in the rotor. This magnetic field causes the rotor to spin in trying to align with the rotating stator magnetic field. The rotation of the magnetic field of the stator has an angular speed that is a function of both the number of poles *NP* and the frequency *f* of the AC current as expressed in Eq. (5.11):

$$\omega_{mag} = \frac{4\pi.f}{N_P} \tag{5.11}$$

The above expression is especially useful in explaining the operation of variable frequency drives for motors with variable loads as discussed in this chapter.

One main difference between the two motor types (synchronous vs. induction) is the mechanism by which the rotor magnetic field is created. In an induction motor, the rotating stator magnetic field induces a current, and thus a magnetic field, in the rotor windings which are typically of the squirrel-cage type. In a magnetic motor, the rotor cannot rotate at the same speed as the magnetic field (if the rotor spins with the same speed as the magnetic field, no current can be induced in the rotor because effectively the stator magnetic field remains at the same position relative to the rotor). The difference between the rotor speed and the stator magnetic field rotation is called the slip factor.

In a synchronous motor, the magnetic field is produced by application of direct current through the rotor windings. Therefore, the rotor spins at the same speed as the rotating magnetic field of the stator and thus the rotor and the stator magnetic fields are synchronous in their speed.

Because of their construction characteristics, the induction motor is basically an inductive load and thus has a lagging power factor whereas the synchronous motor can be set so it has a leading power factor (i.e., acts like a capacitor). Therefore, it is important to remember that a synchronous motor can be installed to both provide mechanical power and improve the power factor for a set of induction motors. This option may be more cost-effective than just adding a bank of capacitors.

Three parameters are typically used to characterize an electric motor during full-load operation. These parameters include:

(a) The mechanical power output of the motor P_M. This power can be expressed in kW or horsepower (1 hp = 0.746 kW). The mechanical power is generally the most important parameter in selecting a motor.
(b) The energy conversion efficiency of the motor η_M. This efficiency expresses the mechanical power as a fraction of the real electric power consumed by the motor. Due to various losses (such as friction, core losses due to the alternating of the magnetic field, and resistive losses through the windings), the motor efficiency is always less than 100 percent. Typical motor efficiencies range from 75 to 95 percent depending on the size of the motor.
(c) The power factor of the motor pf_M. As indicated earlier in this chapter, the power factor is a measure of the magnitude of the reactive power needed by the motor.

Using the schematic diagram of Figure 5.6, the real power used by the motor can be calculated as follows:

$$P_R = \frac{1}{\eta_M}.P_M \tag{5.12}$$

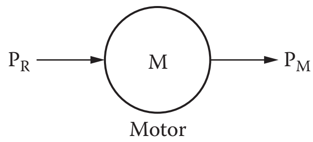


FIGURE 5.6 Definition of the efficiency of a motor.

Therefore, the total power and the reactive power needed to operate the motor are, respectively:

$$P_T = \frac{P_R}{pf_M} = \frac{1}{pf_M . \eta_M} . P_M \tag{5.13}$$

$$P_X = P_R \tan\phi = \frac{1}{\eta_M} . P_M . \tan(\cos^{-1} pf) \tag{5.14}$$

5.3.3 Energy-Efficient Motors

5.3.3.1 General Description

Based on their efficiency, motors can be classified into two categories: (i) standard-efficiency motors and (ii) high or premium-efficiency (i.e., energy-efficient) motors. The energy-efficient motors are 2 to 10 percentage points more efficient than standard-efficiency motors depending on the size. Table 5.2 summarizes the average efficiencies for both standard and energy-efficient motors that are currently available commercially. The improved efficiency for the high or premium-motors is mainly due to better design and use of better materials to reduce losses. However, this efficiency improvement comes with a higher price of about 10 to 30 percent more than standard-efficiency motors. These higher prices may be the main reason that only one-fifth of the motors sold in the United States are energy efficient.

However, the installation of premium-efficiency motors is becoming a common method of improving the overall energy efficiency of buildings. The potential for energy savings from premium-efficiency motor retrofits is significant. In the United States alone, in 1991 there were about 125 million operating motors which consumed approximately 55 percent of the electrical energy generated in the country (Andreas, 1992). It was estimated that replacing all these motors with premium-efficiency models would save approximately 60 Thw of energy per year (Nadel et al., 1991).

To determine the cost-effectiveness of motor retrofits, there are several tools available including the MotorMaster developed by the Washington State Energy Office (WSEO, 1992) and currently available from the DOE website (DOE, 2009). These tools have the advantage of providing large databases for cost and performance information for various motor types and sizes.

5.3.3.2 Adjustable Speed Drives (ASDs)

With more emphasis on energy efficiency, an increasing number of designers and engineers are recommending the use of variable speed motors for various HVAC systems. Indeed, the use of adjustable speed drives (ASDs) is now becoming common especially for supply and return fans in variable-air-volume (VAV) systems and for hot and chilled water pumps in central heating and cooling plants.

Electronic ASDs convert the fixed-frequency AC power supply (50 or 60 Hz) first to a DC power and then to a variable frequency AC power as illustrated in Figure 5.7. Therefore, the ASDs can change the speed of AC motors with no moving parts presenting high reliability and low maintenance requirements.

TABLE 5.2 Typical Motor Efficiencies

Motor Mechanical Power Output kW (hp)	Average Nominal Efficiency for Standard-Efficiency Motors	Nominal Efficiency for Premium-Efficiency Motors
0.75 (1.0)	0.730	0.855
1.12 (1.5)	0.750	0.865
1.50 (2.0)	0.770	0.865
2.25 (3.0)	0.800	0.895
3.73 (5.0)	0.820	0.895
5.60 (7.5)	0.840	0.917
7.46 (10)	0.850	0.917
11.20 (15)	0.860	0.924
14.92 (20)	0.875	0.930
18.65 (25)	0.880	0.936
22.38 (30)	0.885	0.936
29.84 (40)	0.895	0.941
37.30 (50)	0.900	0.945
44.76 (60)	0.905	0.950
55.95 (75)	0.910	0.954
74.60 (100)	0.915	0.954
93.25 (125)	0.920	0.954
111.9 (150)	0.925	0.958
149.2 (200)	0.930	0.962

Source: Adapted from Hoshide, Electric motor do's and don'ts, *Energy Engineering*, 1(1), 6–24, 1994 for standard motors and NEMA Standard Publications MG-1, NEMA Premium Efficiency Electric Motor Program, National Electrical Manufacturers Association, Rosslyn, VA. (2006) for premium motors.

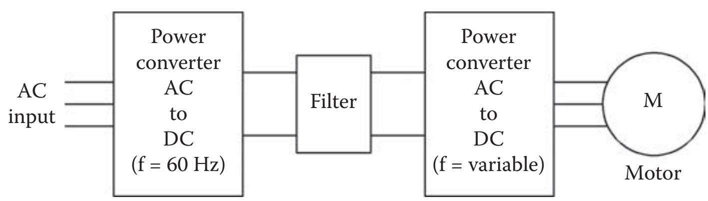


FIGURE 5.7 Basic concept of ASD power inverter.

In order to achieve the energy savings potential for any HVAC application, the engineer needs to know the actual efficiency of the motors. For ASD applications, it is important to separate the losses between the drive and the motor to achieve an optimized HVAC system with the lowest operating cost. Specifically, the engineer would need to know the loss distribution including the iron, copper, friction, and windage losses, and the distribution of losses between stator and rotor. To determine these losses and thus the motor efficiency, accurate measurements are needed. Unfortunately, existing power measurement instruments are more suitable for sinusoidal rather than distorted waveforms (which are typical for ASD applications).

The use of low-cost solid-state power devices and integrated circuits to control the speed has made most of the commercially available ASDs draw power with extremely high harmonic content. Some investigators (Dominijan et al., 1995; Czarkowski and Domijan, 1997) have studied the behavior of commercially available power measurement instruments subjected to voltage and current waveforms typical of ASD-motor connections. In particular, three drive technologies used in the HVAC industry were investigated by Czarkowski and Domijan (1997) and by Domijan et al. (1996) and were used namely by PWM induction, switched reluctance, and brushless DC drives. The main finding of their investigation was that existing power instruments failed to accurately measure power losses (and thus motor efficiency) due to the high harmonic content in the voltage spectra especially for brushless DC and switched reluctance motors which represent a substantial portion of the HVAC market.

Typically, the motor losses were measured indirectly by monitoring the power input and the power output and then taking the difference. This traditional approach requires an extremely high accuracy for the power measurements to achieve a "reasonable" estimation of motor losses especially for premium-efficiency motors (with low losses). However, a new approach to measure the ASD-motor losses directly and with better accuracy has been proposed (Fuchs and Fei, 1996).

5.3.3.3 Energy Savings Calculations

There are three methods to calculate the energy savings due to energy-efficient motor replacement. These three methods are outlined below.

5.3.3.3.1 Method 1: Simplified Method

This method has been and is still being used by most energy engineers to determine the energy and cost savings incurred by motor replacement. Inherent to this method, two assumptions are made: (i) the motor is fully loaded and (ii) the change in motor speed is neglected.

The electric power savings due to the motor replacement is first computed as follows:

$$\Delta P_R = P_M \cdot \left(\frac{1}{\eta_e} - \frac{1}{\eta_r} \right) \tag{5.15}$$

where

P_M = the mechanical power output of the motor.
η_e = the design (i.e., full-load) efficiency of the existing motor (e.g., before retrofit).
η_r = the design (i.e., full-load) efficiency of the energy-efficient motor (e.g., after retrofit).

The electric energy savings incurred from the motor replacement is thus:

$$\Delta kWh = \Delta P_R . N_h . LF_M \tag{5.16}$$

where

N_h = the number of hours per year during which the motor is operating.
LF_M = the load factor of the motor's operation during one year.

Example 5.2 indicates the potential energy use, cost savings, and payback period when a standard-efficiency 10-hp (7.46-kW) motor is replaced with a premium-efficiency motor.

Example 5.2

Determine the cost-effectiveness of replacing a 10-hp motor with an efficiency of 85 percent with a premium-efficiency motor with a rated full-load efficiency of 91.70 percent. Assume that:

- The cost of electricity is $0.10/kWh.
- The differential cost of a premium versus standard motor is $300.

- The average load factor of the motor is 0.80.
- The average full-load operating hours of the motor are 5,000 hours/year.

Solution

To determine the cost-effectiveness of installing a premium-efficiency motor instead of a standard-efficiency motor, a simplified economic analysis is used to estimate the simple payback period. The savings in energy use in kWh for the premium-efficiency motor can be calculated using Eqs. (5.15) and (5.16):

$$kWh_{saved} = N_h.HP * 0.746.LF_M.\left(\frac{1}{\eta_{std}} - \frac{1}{\eta_{eff}}\right)$$

where

N_s = the total number of hours (per year) during which the motor is operating at full load is 5,000 hours/yr.

hp = the rated motor power output [10 hp].

LF_M = the annual average load factor of the motor [LF_M = 0.80].

η_{std} and η_{eff} = the efficiency of the standard transformer and the efficient transformer [0.850 and 0.917], respectively.

Thus, the energy saving in kWh is calculated as follows:

$$kWh_{saved} = 5000 * 10 * 0.746 * 0.80 * \left(\frac{1}{0.850} - \frac{1}{0.917}\right) = 2565 kWh/yr$$

Therefore, the simple payback period *SPB* for investing in the premium-efficiency motor is:

$$SPB = \frac{\$300}{2565 kWh * \$0.10 / kWh} = 1.2 years$$

5.3.3.3.2 Method 2: Mechanical Power Rating Method

In this method, the electrical peak demand of the existing motor is assumed to be proportional to its average mechanical power output:

$$P_{R,e} = \frac{P_M}{\eta_{op,e}}.LF_{M,e}.PDF_{M,e} \tag{5.17}$$

where

$\eta_{op,e}$ = the motor efficiency at the operating average part-load conditions. To obtain this value, the efficiency curve for the motor can be used. If the efficiency curve for the specific existing motor is not available, a generic curve can be used.

$LF_{M,e}$ = the load factor of the existing motor and is the ratio between the average operating load of the existing motor and its rated mechanical power. In most applications, the motor is oversized and operates at less than its capacity.

$PDF_{M,e}$ = the peak demand factor and represents the fraction of the typical motor load that occurs at the time of the building peak demand. In most applications, $PDF_{M,e}$ can be assumed to be unity because the motors often contribute to the total peak demand of the building.

The mechanical load does not change after installing an energy-efficient motor, thus it is possible to consider a smaller motor with a capacity $P_{M,r}$ if the existing motor is oversized with a rating of $P_{M,e}$. In this case, the smaller energy-efficient motor can operate at a higher load factor than the existing motor. The new load factor LF_r of the energy-efficient motor can be calculated as follows:

$$LF_r = LF_e . \frac{P_{M,r}}{P_{M,e}} \tag{5.18}$$

Moreover, the energy-efficient motors often operate at a higher speed than the standard motors they replace inasmuch as they have lower internal losses. This higher speed actually has a negative impact because it reduces the effective efficiency of the energy-efficient motor by a factor called the slip penalty. The slip penalty factor SLIPP is defined as shown in Eq. (5.19):

$$SLIP_P = \left(\frac{\omega_{M,r}}{\omega_{M,e}} \right)^3 \tag{5.19}$$

where

$\omega_{M,e}$ = the rotation speed of the existing motor.
$\omega_{M,r}$ = the rotation speed of the energy-efficient motor.

Using a similar equation to Eq. (5.17), the peak electrical demand for the retrofitted motor (e.g., energy-efficient motor) can be determined:

$$P_{R,r} = \frac{P_{M,r}}{\eta_{op,r}} .LF_{M,r} .PDF_{M,r} .SLIP_P \tag{5.20}$$

The electrical power savings due to the motor replacement can thus be estimated:

$$\Delta P_R = P_{R,e} - P_{R,r} \tag{5.21}$$

The electric energy savings can therefore be calculated using Eq. (5.16).

5.3.3.3.3 Method 3: Field Measurement Method

In this method, the motor electrical power demand is measured directly on site. Typically, current *IM*, voltage *VM*, and power factor *pfM*, readings are recorded for the existing motor to be retrofitted. For three-phase motors (which are common in industrial facilities and in most HVAC systems for commercial buildings), the electrical power used by the existing motor can be either directly measured or calculated from current, voltage, and power factor readings as follows:

$$P_{R,E} = \sqrt{3} .V_M .I_M .pf_M \tag{5.22}$$

The load factor of the existing motor can be estimated by taking the ratio of the measured current over the nameplate full-load current *IFL* as expressed by Eq. (5.23):

$$LF_{M,E} = \frac{I_M}{I_{FL}} \tag{5.23}$$

A study by Biesemeyer and Jowett (1996) has indicated that Eq. (5.23) has much higher accuracy to estimate the motor load ratio than an approach based on the ratio of the motor speeds (i.e., measured speed over nominally rated speed) used by BPA (1990) and Lobodovsky (1994). It should be noted that Eq. (5.23) is recommended for load ratios that are above 50 percent because for these load ratios, a typical motor draws electrical current that is proportional to the imposed load.

The methodology for the calculation of the electrical power and energy savings is the same as described for the mechanical power rating method using Eqs. (5.18) through (5.23).

5.4 Lighting Systems

5.4.1 Introduction

Lighting accounts for a significant portion of the energy use in commercial buildings. For instance, in office buildings, 30 to 50 percent of the electricity consumption is used to provide lighting. In addition, heat generated by lighting contributes to additional thermal loads that need to be removed by the cooling equipment. Typically, energy retrofits of lighting equipment are very cost-effective with payback periods of less than two years in most applications.

In the United States, lighting energy efficiency features are the most often considered strategies to reduce energy costs in commercial buildings as shown in Table 5.3. The data for Table 5.3 is based on the results of a survey (EIA, 1997) to determine the participation level of commercial buildings in a variety of specific types of conservation programs and energy technologies.

To better understand the retrofit measures that need to be considered in order to improve the energy efficiency of lighting systems, a simple estimation of the total electrical energy use due to lighting is first considered:

$$Kwh_{Lit} = \sum_{j=1}^{J} N_{Lum,j}.WR_{Lum,j}.N_{h,j} \tag{5.24}$$

TABLE 5.3 Level of Participation in Lighting Conservation Programs by U.S. Commercial Buildings

Lighting Retrofit	Percent Participation in Number of Buildings	Percent Participation in Floor Area of Spaces
Energy-efficient lamps and ballasts	31	49
Specular reflectors	18	32
Time clock	10	23
Manual dimmer switches	10	23
Natural lighting control sensors	7	13
Occupancy sensors	5	11

Source: EIA (1997).

where

$N_{Lum,j}$ = the number of lighting luminaires of type j in the building to be retrofitted. Recall that a luminaire consists of the complete set of a ballast, electric wiring, housing, and lamps.

$WR_{Lum,j}$ = the wattage rating for each luminaire of type j. The energy use due to both the lamp and ballast should be accounted for in this rating.

$N_{h,j}$ = the number of hours per year when the luminaires of type j are operating.

J = the number of luminaire types in the building.

It is clear from Eq. (5.24) that there are three options to reduce the energy use attributed to lighting systems as briefly discussed below:

(a) Reduce the wattage rating for the luminaires including both the lighting sources (e.g., lamps) and the power transforming devices (e.g., ballasts) [thereby decreasing the term $WR_{Lum,j}$ in Eq. (5.24)]. In the last decade, technological advances such as compact fluorescent lamps and electronic ballasts have increased the energy efficiency of lighting systems.

(b) Reduce the time of use of the lighting systems through lighting controls [thereby, decrease the term $N_{h,j}$ in Eq. (5.24)]. Automatic controls have been developed to decrease the use of a lighting system so illumination is provided only during times when it is actually needed. Energy-efficient lighting controls include the occupancy sensing systems and light dimming controls through the use of daylighting.

(c) Reduce the number of luminaires [thereby decreasing the term $N_{Lum,j}$ in Eq. (5.24)]. This goal can be achieved only in cases where delamping is possible due to overillumination.

In this section, only measures related to the general actions described in items (a) and (b) are discussed. To estimate the energy savings due to any retrofit measure for the lighting system, Eq. (5.24) can be used. The energy use due to lighting has to be calculated before and after the retrofit and the difference between the two estimated energy uses represents the energy savings.

5.4.2 Energy-Efficient Lighting Systems

Improvements in the energy efficiency of lighting systems have provided several opportunities to reduce electrical energy use in buildings. In this section, the energy savings calculations for the following technologies are discussed:

- High-efficiency fluorescent lamps
- Compact fluorescent lamps
- Compact halogen lamps
- Electronic ballasts

First, a brief description is provided for the factors that an auditor should consider in order to achieve and maintain an acceptable quality and level of comfort for the lighting system. Second, the design and the operation concepts are summarized for each available lighting technology. Then, the energy savings that can be expected from retrofitting existing lighting systems using any of the new technologies are estimated and discussed.

Typically, three factors determine the proper level of light for a particular space. These factors include: age of the occupants, speed and accuracy requirements, and background contrast (depending on the task being performed). It is a common misconception to consider that overlighting a space provides higher visual quality. Indeed, it has been shown that overlighting can actually reduce the illuminance quality and the visual comfort level within a space in addition to wasting energy. Therefore, it is important when upgrading a lighting system to determine and maintain the adequate illuminance level as recommended by the appropriate authorities. Table 5.4 summarizes the lighting levels recommended for various activities and applications in selected countries including the United States based on the most recent illuminance standards.

TABLE 5.4 Recommended Lighting Levels for Various Applications in Selected Countries

Application	France AEF	Germany DIN5035	Japan JIS	United States/ Canada IESNA
Offices				
General	425	500	300–750	200–500
Reading tasks	425	500	300–750	200–500
Drafting (detailed)	850	750	750–1,500	1,000–2,000
Classrooms				
General	325	300–500	200–750	200–500
Chalkboards	425	300–500	300–1,500	500–1,000
Retail Stores				
General	100–1,000	300	150–750	200–500
Tasks/till areas	425	500	750–1,000	200–500
Hospitals				
Common areas	100	100–300	—	—
Patient rooms	50–100	1,000	150–300	100–200
Manufacturing				
Fine knitting	850	750	750–1,500	1,000–2,000
Electronics	625–1,750	100–1,500	1,500–300	1,000–2,000

Note: In Lux maintained on horizontal surfaces.

5.4.2.1 High-Efficiency Fluorescent Lamps

Fluorescent lamps are the most commonly used lighting systems in commercial buildings. In the United States, fluorescent lamps illuminate 71 percent of the commercial space. Their relatively high efficacy, diffuse light distribution, and long operating life are the main reasons for their popularity.

A fluorescent lamp generally consists of a glass tube with a pair of electrodes at each end. The tube is filled at very low pressure with a mixture of inert gases (primarily argon) and liquid mercury. When the lamp is turned on, an electric arc is established between the electrodes. The mercury vaporizes and radiates in the ultraviolet spectrum. This ultraviolet radiation excites a phosphorous coating on the inner surface of the tube that emits visible light. High-efficiency fluorescent lamps use a krypton–argon mixture that increases the efficacy output by 10 to 20 percent from a typical efficacy of 70 lumens/watt to about 80 lumens/watt. Improvements in the phosphorous coating can further increase the efficacy to 100 lumens/watt.

It should be mentioned that the handling and disposal of fluorescent lamps is highly controversial due to the fact that the mercury inside the lamps can be toxic and hazardous to the environment. A new technology is being tested to replace the mercury with sulfur to generate the radiation that excites the phosphorous coating of the fluorescent lamps. The sulfur lamps are not hazardous and would present an environmental advantage to the mercury-containing fluorescent lamps.

Fluorescent lamps come in various shapes, diameters, lengths, and ratings. A common labeling used for fluorescent lamps is

$$F \cdot S \cdot W \cdot C - T \cdot D$$

where

F stands for the fluorescent lamp.

S refers to the style of the lamp. If the glass tube is circular, then the letter *C* is used. If the tube is straight, no letter is provided.

W is the nominal wattage rating of the lamp (it can be 4, 5, 8, 12, 15, 30, 32, 34, 40, etc.).

C indicates the color of the light emitted by the lamp: *W* for white, *CW* for cool white, *BL* for black light.

T refers to tubular bulb.

D indicates the diameter of the tube in eighths of one inch (1/8 in. = 3.15 mm) and can be, for instance, 12 (*D* = 1.5 in. = 38 mm) for the older and less energy-efficient lamps and 8 (D = 1.0 in. = 31.5 mm) for more recent and energy-efficient lamps.

Thus, F40CW-12 designates a fluorescent lamp that has a straight tube, uses 40 W electric power, provides cool white color, and is tubular with 38 mm (1.5 inches) in diameter.

Among the most common retrofit in lighting systems is the upgrade of the conventional 40 W T12 fluorescent lamps to more energy-efficient lamps such 32 W T8 lamps. For a lighting retrofit, it is recommended that a series of tests be conducted to determine the characteristics of the existing lighting system. For instance, it is important to determine the illuminance level at various locations within the space especially in working areas such as benches or desks.

5.4.2.2 Compact Fluorescent Lamps

These lamps are miniaturized fluorescent lamps with small diameter and shorter length. The compact lamps are less efficient that full-size fluorescent lamps with only 35 to 55 lumens/watt. However, they are more energy efficient and have longer life than incandescent lamps. Currently, compact fluorescent lamps are being heavily promoted as energy-saving alternatives to incandescent lamps even though they may have some drawbacks. In addition to their high cost, compact fluorescent lamps are cooler and thus provide less pleasing contrast than incandescent lamps.

5.4.2.3 Compact Halogen Lamps

Compact halogen lamps are adapted for use as direct replacements for standard incandescent lamps. Halogen lamps are more energy-efficient, produce whiter light, and last longer than incandescent lamps. Indeed, incandescent lamps typically convert only 15 percent of their electrical energy input into visible light because 75 percent is emitted as infrared radiation and 10 percent is used by the filament as it burns off. In halogen lamps, the filament is encased inside a quartz tube which is contained in a glass bulb. A selective coating on the exterior surface of the quartz tube allows visible radiation to pass through but reflects the infrared radiation back to the filament. This recycled infrared radiation permits the filament to maintain its operating temperatures with 30 percent less electrical power input.

Halogen lamps can be dimmed and present no power quality or compatibility concerns as can be the case for compact fluorescent lamps.

5.4.2.4 Electronic Ballasts

Ballasts are integral parts to fluorescent luminaires because they provide the voltage level required to start the electric arc and regulate the intensity of the arc. Before the development of electronic ballasts in the early 1980s, only magnetic or "core and coil" ballasts were used to operate fluorescent lamps. Although the frequency of the electrical current is kept at 60 Hz (in countries other than the United States, the frequency is set at 50 Hz) by the magnetic ballasts, electronic ballasts use solid-state technology to produce high-frequency (20–60 MHz) current. The use of high-frequency current increases the energy efficiency of the fluorescent luminaires because light cycles more quickly and appears brighter. When used with high-efficiency lamps (T8, for instance), electronic ballasts can achieve 95 lumens/watts as opposed to 70 lumens/watts for conventional magnetic ballasts. It should be mentioned, however, that efficient magnetic ballasts can achieve similar lumen/watt ratios as electronic ballasts.

Other advantages that electronic ballasts have relative to their magnetic counterparts include:

- Higher power factor. The power factor of electronic ballasts is typically in the 0.90 to 0.98 range. Meanwhile, the conventional magnetic ballasts have a low power factor (less than 0.80) unless a capacitor is added as discussed in Section 5.2.

- Fewer flicker problems. Because the magnetic ballasts operate at 60 Hz current, they cycle the electric arc about 120 times per second. As a result, flicker may be perceptible during normal operation especially if the lamp is old, or when the lamp is dimmed to less than 50 percent capacity. However, electronic ballasts cycle the electric arc several thousands of times per second and flicker problems are avoided even when the lamps are dimmed to as low as 5 percent of capacity.
- Fewer noise problems. The magnetic ballasts use electric coils and generate audible hum which can increase with age. Such noise is eliminated by the solid-state components of the electronic ballasts.

5.4.3 Lighting Controls

As illustrated by Eq. (5.24), energy savings can be achieved by not operating the lighting system in cases when illumination becomes unnecessary. Control of the lighting system operation can be achieved by several means including manual on–off and dimming switches, occupancy sensing systems, and automatic dimming systems using daylighting controls.

Energy savings can be achieved by manual switching and manual dimming, however, the results are typically unpredictable inasmuch as they depend on occupant behavior. Scheduled lighting controls provide a more efficient approach to energy savings but can also be affected by the frequent adjustments by occupants. Only automatic light switching and dimming systems can respond in real-time to changes in occupancy and climatic changes. Some of the automatic controls available for lighting systems are briefly discussed below.

5.4.3.1 Occupancy Sensors

Occupancy sensors save energy by automatically turning off the lights in spaces that are not occupied. Generally, occupancy sensors are suitable for most lighting control applications and should be considered for lighting retrofits. It is important to properly specify and install the occupancy sensors to provide reliable lighting during periods of occupancy. Indeed, most failed occupancy sensor installations result from inadequate product selection and improper placement. In particular, the auditor should select the proper motion-sensing technology used in occupancy sensors. Two types of motion-sensing technologies are currently available on the market:

(i) *Infrared sensors*, which register the infrared radiation emitted by various surfaces in the space including the human body. When the controller connected to the infrared sensors receives a sustained change in the thermal signature of the environment (as is the case when an occupant moves), it turns the lights on. The lights are kept on until the recorded changes in temperature are not significant. The infrared sensors operate adequately only if they are in direct line-of-sight with the occupants and thus must be used in smaller enclosed spaces with regular shapes and without partitions.

(ii) *Ultrasound sensors* operate on a sonar principle as do submarines and airport radars. A device emits a high frequency sound (25–40 KHz) so it is beyond the hearing range of humans. This sound is reflected by the surfaces inside a space (including furniture and occupants) and is sensed by a receiver. When people move inside the space, the sound wave pattern changes. The lights remain on until no movement is detected for a preset period of time (after five minutes). Unlike infrared radiation, sound waves are not easily blocked by obstacles such as wall partitions. However, the ultrasound sensors may not operate properly in large spaces which tend to produce weak echoes.

Based on a study by EPRI, Table 5.5 illustrates the typical energy savings to be expected from occupancy sensor retrofits. As shown in Table 5.5, significant energy savings can be achieved in spaces where occupancy is intermittent, such as conference rooms, rest rooms, storage areas, and warehouses.

TABLE 5.5 Energy Savings Potential with Occupancy Sensor Retrofits

Space Application	Range of Energy Savings (%)
Offices (private)	25–50
Offices (open space)	20–25
Rest rooms	30–75
Conference rooms	45–65
Corridors	30–40
Storage areas	45–65
Warehouses	50–75

5.4.3.2 Light Dimming Systems

Dimming controls allow the variation of the intensity of lighting system output based on natural light level, manual adjustments, and occupancy. A smooth and uninterrupted decrease in the light output is defined as a continuous dimming as opposed to stepped dimming in which the lamp output is decreased in stages by preset amounts.

To accurately estimate the energy savings from dimming systems that use natural light controls (e.g., daylighting), computer software exists such as RADIANCE (LBL, 1991). With this computer tool, an engineer can predict the percentage of time when natural light is sufficient to meet all lighting needs.

Example 5.3 provides a simple calculation procedure to estimate the energy savings from lighting retrofit project.

Example 5.3

Consider a building with total 500 luminaires of four 40-watt lamps/luminaire. Determine the energy saving after replacing those with two 40-watt high-efficiency lamps/luminaire. This building is operated 8 hours/day, 5 days/week, 50 weeks/year.

Solution

The energy saving in KWh is

$$\Delta KWh = 500 \cdot (4 * 40 - 4 * 32) \cdot 8.5.50 \cdot \frac{1}{1000} = 32{,}000 kWh/yr$$

Thus, the energy saving is 32,000 kWh/year.

5.4.3.3 Energy Savings from Daylighting Controls

Several studies indicate that daylighting can offer a cost-effective alternative to electrical lighting for commercial and institutional buildings. Through sensors and controllers, daylighting can reduce and even eliminate the use of electrical lighting required to provide sufficient illuminance levels inside office spaces. Recently, a simplified calculation method was developed by Krarti, Erickson, and Hillman (2005) to estimate the reduction in the total lighting energy use due to daylighting with dimming controls for office buildings. The method has been shown to apply for office buildings in the United States

TABLE 5.6 Coefficients *a* and *b* of Eq. (5.25) for Various Locations Throughout the World

Location	a	b	Location	a	b
Atlanta	19.63	74.34	Casper	19.24	72.66
Chicago	18.39	71.66	Portland	17.79	70.93
Denver	19.36	72.86	Montreal	18.79	69.83
Phoenix	22.31	74.75	Quebec	19.07	70.61
New York City	18.73	66.96	Vancouver	16.93	68.69
Washington, DC	18.69	70.75	Regina	20.00	70.54
Boston	18.69	67.14	Toronto	19.30	70.48
Miami	25.13	74.82	Winnipeg	19.56	70.85
San Francisco	20.58	73.95	Shanghai	19.40	67.29
Seattle	16.60	69.23	Kuala Lumpur	20.15	72.37
Los Angeles	21.96	74.15	Singapore	23.27	73.68
Madison	18.79	70.03	Cairo	26.98	74.23
Houston	21.64	74.68	Alexandria	36.88	74.74
Fort Worth	19.70	72.91	Tunis	25.17	74.08
Bangor	17.86	70.73	Sao Paulo	29.36	71.19
Dodge City	18.77	72.62	Mexico	28.62	73.63
Nashville	20.02	70.35	Melbourne	19.96	67.72
Oklahoma City	20.20	74.43	Rome	16.03	72.44
Columbus	18.60	72.28	Frankfurt	15.22	69.69
Bismarck	17.91	71.50	Kuwait	21.98	65.31
Minneapolis	18.16	71.98	Riyadh	21.17	72.69
Omaha	18.94	72.30			

as well as in Egypt (El-Mohimen et al., 2005). The simplified calculation method is easy to use and can be employed as a predesign tool to assess the potential of daylighting in saving electricity use associated with artificial lighting for office buildings.

To determine the percent savings *fd* in annual use of artificial lighting due to implementing daylighting using daylighting controls in office buildings, Krarti, Erickson, and Hillman (2005) found that the following equation can be used:

$$f_d = b\left[1-\exp\left(-a\tau_w A_w / A_p\right)\right]\frac{A_p}{A_f} \tag{5.25}$$

where

A_w/A_p = window to perimeter floor area. This parameter provides a good indicator of the window size relative to the daylit floor area.

A_p/A_f = perimeter to total floor area. This parameter indicates the extent of the daylit area relative to the total building floor area. Thus, when $A_p/A_f = 1$, the whole building can benefit from daylighting.

a and b = coefficients that depend only on the building location and are shown in Table 5.6 for various sites throughout the world.

τ_w = the visible transmittance of the glazing.

Example 5.4 outlines the calculation procedures to estimate the annual energy savings associated with installing daylighting dimming controls for an office building.

Example 5.4

Consider a three-story office building 50 m by 30 m located in Denver, Colorado with a window-to-wall ratio (WWR) of 0.25 with low-E glazing which has a visible transmittance of 0.41. The building has a lighting density of 10 W/m^2. Determine the energy and cost savings implementing a dimming daylighting control system through the perimeter zones (5-m wide) of the building. Assume that without daylighting controls, the lighting system is operated 8 hours/day, 5 days/week, 50 weeks/year. The cost of electricity is 0.10/kWh. Assume the typical wall height is 3 m per floor.

Solution

First, the various areas used in Eq. (5.25) are computed on a per floor basis:

$$A_f = 50mx30m = 1500\,m^2$$

$$A_p = 2x[50 + 30 - 10]x5m^2 = 700\,m^2$$

$$A_w = 0.25x2.x[50 + 30]x3m^2 = 120\,m^2$$

Based on Eq. (5.25) using the coefficients a = 19.36 and b = 72.8f, the percent reduction in electrical lighting associated with dimming daylighting controls installed in all perimeter zones for one floor is:

$$f_d = 72.86x\left[1 - \exp\left(-19.36x(0.41)\ \ 120/700\right)\right]\frac{700}{1500} = 25.3\%$$

Therefore, the savings in annual electrical energy for the entire building associated with the lighting system are computed as follows:

$$\Delta kWh = 25.3\% * 3 * [1500x10x \cdot 8.5.50 \cdot \frac{1}{1000}] = 22{,}753 kWh/yr$$

Thus, the annual energy cost saving is $2,275/year.

5.5 Electrical Appliances

5.5.1 Office Equipment

Over the last decade, the energy used by office equipment has increased significantly and currently accounts for more than 7 percent of the total commercial sector electricity use. Recognizing this problem, the U.S. Environmental Protection Agency (EPA) in cooperation with the U.S. Department of Energy (DOE) has launched the Energy Star Office Program to increase the energy efficiency of commonly used office equipment such as computers, fax machines, printers, and scanners. To help consumers in

TABLE 5.7 Summary of the Energy Star Specifications for Copiers, Scanners, and Fax Machines

Product	Typical Electrical Consumption (TEC) in kWh/week	Power Management Preset Default Times (min)
Color copiers, scanners, and fax machines	0.1 ipm + 2.8 (ipm < 32)	5–30
(Specifications depend on the ipm: images per minute)	0.35 ipm –5.2 (32 < ipm < 58)	5–60
	0.70 ipm – 26.0 (ipm > 58)	5–60
Monochrome printers, scanners, and fax machines	1.0 (ipm < 15)	5–15
(Specifications depend on the ppm: pages per minute)	0.1 ipm – 0.5 (15 < ipm < 40)	5–60
	0.35 ipm – 10.3 (40 < ipm < 82)	5–60
	0.70 ipm – 39.0 (ipm > 82)	5–60

Source: EPA (2009).

identifying energy-efficient office equipment, Energy Star labels are provided to indicate the energy-saving features of the products.

It is estimated that Energy Star-labeled products can save as much as 75 percent of total electricity use depending on the type and the usage pattern of the office equipment. Almost all office equipment manufacturers currently integrate power management features in their products. For instance, computers can enter a low-power "sleep" mode when idled for a specific period of time. Similarly, copiers can go into a low-power mode of only 15–45 watts after 30–90 minutes of activity. Table 5.7 summarizes the Energy Star features of common office equipment.

A recent study has estimated that the Energy Star-labeled products have reduced significant carbon emissions through voluntary labeling efforts (Sanchez et al., 2008). Specifically and since its inception in 1992 through 2006, the program has saved 4.8 exajoules of primary energy (1 exajoule = 10^{18} joules = 1 quadrillion Btu = 10^{15} Btu). The same study indicates that the program will save 12.8 exajoules over the period 2007–2010 (Sanchez et al., 2008).

5.5.2 Residential Appliances

Appliances account for a significant part of the energy consumption in buildings and used about 41 percent of electricity generated worldwide in 1990 (IPCC, 1996). Moreover, the operating cost of appliances during their lifetime (typically 10 to 15 years) far exceeds their initial purchase price. However, consumers—especially in the developing countries where no labeling programs for appliances are enacted—do not generally consider energy efficiency and operating cost when making purchases inasmuch as they are not well informed.

Recognizing the significance and impact of appliances on national energy requirements, a number of countries have established energy efficiency programs. In particular, some of these programs target improvements of energy efficiency for residential appliances. Methods to achieve these improvements include energy efficiency standards and labeling programs.

Minimum efficiency standards for residential appliances have been implemented in some countries for a number of residential end-uses. The energy savings associated with the implementation of these standards are found to be substantial. For instance, studies have indicated that in the United Kingdom, the energy consumption of average new refrigerators and freezers in 1993 was about 60 percent of the consumption in 1970. Similar improvements have been obtained in Germany (Waide, Lebot, and

Hinnells, 1997). In the United States, the savings due to the standards are estimated to be about 0.7 exajoules per year during the period extending from 1990 to 2010.

Energy standards for appliances in the residential sector have been highly cost-effective. In the United States, it is estimated that the average benefit–cost ratios for promoting energy-efficient appliances are about 3.5. In other terms, each U.S. dollar of federal expenditure implementing the standards is expected to contribute $165 of net present-valued savings to the economy over the period of 1990 to 2010. In addition to energy and cost savings, minimum efficiency standards reduce pollution with a significant reduction in carbon emissions. In the period of 2000 to 2010, it is estimated that energy efficiency standards will result in an annual carbon reduction of 4 percent (corresponding to 9 million metric tons of carbon/year) relative to the 1990 level.

Currently, energy efficiency standards are utilized with various degrees of comprehensiveness, enforcement, and adoption in a limited number of countries as summarized in Table 5.8. However, several other countries are in the process of developing national standards or labeling programs to promote energy-efficient residential appliances.

Minimum efficiency standards have been in use in the United Sates and Canada for more than two decades and cover a wide range of products. In the last few years, standards programs have spread to other countries including Brazil, China, Korea, Mexico, and the Philippines. Most of these countries currently have one or two products subject to standards such as the European countries, Korea, and Japan. Other countries are in the process of implementing or considering energy efficiency standards for household appliances. Among these countries are Colombia, Denmark, Egypt, Indonesia, Malaysia, Pakistan, Singapore, and Thailand.

Many of the currently existing standards are mandatory and prohibit the manufacture or sale of noncomplying products. However, there other standards that are voluntary and thus are not mandatory such as the case of product quality standards established in India.

As indicated in Table 5.8, most countries have established minimum efficiency standards for refrigerators and freezers because this product type has one of the highest growth rates both in terms of sales value and volume. The existing international energy efficiency standards for refrigerators and freezers set a limit on the energy use over a specific period of time (generally, one month or one year). This

TABLE 5.8 Status of Selected International Residential Appliance Energy Efficiency Standards

Country/Region	Compliance Status	Products[a]
Australia	Mandatory	R, FR, WH
Brazil	Voluntary	R, FR
Canada	Mandatory	All
China	Mandatory	R, CW, RAC
European Union	Mandatory	R, FR
India	Voluntary	R, RAC, A/C
Japan	Voluntary	A/C
Korea	Mandatory	R, A/C
Mexico	Mandatory	R, FR, RAC
Philippines	Mandatory	A/C
United States	Mandatory	All

Source: Turiel, Present status of residential appliance energy efficiency standards, an international review, *Energy and Buildings*, 26(1), 5, 1997.

[a] Products are: Refrigerators (R), Freezers (FR), Clothes Washers (CW), Dishwashers (DW), Clothes Dryers (CD), Water Heaters (WH), Room Air-Conditioners (RAC), and Central Air-Conditioning (A/C).

TABLE 5.9 Maximum Allowable Annual Energy Use (Kwh/Yr) for Refrigerators and Freezers Sold and Manufactured in the United States

Product Category	2001 Standard	2008 Standard
Manual defrost R/FR	248.4 + 8.82AV	198.72.4 + 7.056AV
Partial auto-defrost R/FR	248.4 + 8.82AV	198.72.4 + 7.056AV
Top-mount auto-defrost R/FR	276 + 9.8AV	220.8 + 7.84AV
Top-mount auto-defrost with through the door features R/FR	356 + 10.20AV	284.8 + 8.16AV
Side-mount auto-defrost R/FR	507.5 + 4.91AV	406 + 3.928AV
Side-mount auto-defrost with through the door features R/FR	406 + 10.10AV	324.8 + 8.08AV
Bottom-mount auto-defrost R/FR	459 + 4.6AV	367.2 + 3.68AV
Upright manual FR	258.3 + 7.55AV	232.47 + 8.08AV
Upright auto-defrost FR	326.1 + 12.43AV	293.49 + 11.87AV
Chest RF	143.7 + 9.88AV	129.33 + 8.892AV

Source: Energy Star, U.S. Environmental Protection Agency, information provided on the Energy Star Web site: http://www.energystar.gov, 2009.

Notes: R = Refrigerators; FR = Freezers; AV = adjusted volume = Volume of R + 1.63 × Volume of FR (for R/FR) = 1.73 × Volume of FR (for R).

energy use limit may vary depending on the size and the configuration of the product. Table 5.9 shows the maximum limits for the allowable annual energy use for U.S. refrigerators and freezers. Two standards are shown in Table 5.9: the standards effective in 2001 and the updated standards effective in April 2008. It should be noted that models with higher energy efficiencies than those listed in Table 5.9 do exist and are sold in the U.S. market. To keep up with the technology advances, U.S. standards are typically amended or changed periodically. However, any new or amended U.S. standard for energy efficiency has to be based on improvements that are technologically feasible and economically justified. Typically, energy-efficient designs with payback periods of less than three years can be incorporated into new U.S. standards.

In addition to standards, labeling programs have been developed to inform consumers about the benefits of energy efficiency. There is a wide range of labels used in various countries to promote energy efficiency for appliances. These labels can be grouped into three categories:

1. Efficiency-type labels used to allow consumers to compare the performance of different models for a particular product type. For instance, a common label used for refrigerators indicates the energy use and operating cost over a specific period such as one month or one year. Other labels such as labels for clothes washers show the efficiency expressed in kWh of energy used per pound of clothes washed. Another feature of efficiency labels is the ability to provide consumers with a comparative evaluation of the product models by showing the energy consumption or efficiency of a particular model on a scale of the lowest (or highest) energy use (or efficiency) models.
2. Eco-labels provide information on more than one aspect (i.e., energy efficiency) of the product. Other aspects include noise level, waste disposal, and emissions. Green Seal in the United States is an example of an eco-label program that certifies the products are designed and manufactured in an environmentally responsible manner (Green Seal, 1993). Certification standards have been established for refrigerators, freezers, clothes washers, clothes dryers, dishwashers, and cooktops/ovens.
3. Efficiency seals of approval, such as the Energy Star program in the United States, are labels that indicate a product has met a set of energy efficiency criteria but do not quantify the degree

by which the criteria were met. The Energy Star label, established by the U.S. Environmental Protection Agency, indicates, for instance, that a computer monitor is capable of reducing its standby power level when not in use for some time period (Johnson and Zoi, 1992).

In recent years, labeling of appliances has become a popular approach around the world in order to inform consumers about the energy use and energy cost of purchasing different models of the same product. Presently, Australia, the United States, and Canada have the most comprehensive and extensive labeling programs. Other countries such as the European Union, Japan, Korea, Brazil, the Philippines, and Thailand have developed labels for a few products.

In addition to energy efficiency, standards have been developed to improve the performance of some appliances in conserving water. For instance, water-efficient plumbing fixtures and equipment have been developed in the United States to promote water conservation.

The reduction of water use by some household appliances can also increase their energy efficiency. Indeed, a large fraction of the electrical energy used by both clothes washers and dishwashers is attributed to heating the water (85 percent for clothes washers and 80 percent for dishwashers). Chapter 15 provides more detailed discussion on water and energy performance of conventional and energy/water-efficient models currently available for residential clothes washers and dishwashers.

5.6 Electrical Distribution Systems

5.6.1 Introduction

All electrical systems have to be designed in order to provide electrical energy to the utilization equipment as safely and reliably as economically possible. Figure 5.8 shows a typical one-line diagram of an electrical system for a commercial building. The main distribution panel includes the switchgear-breakers to distribute the electric power and the unit substation to step down the voltage. The unit substation consists typically of a high voltage disconnect switch, a transformer, and a set of low-voltage breakers. The circuit breakers for lighting and plug-connected loads are housed in lighting panelboards and the protective devices for motors are typically assembled into motor control centers (MCCs). Specifically, a MCC consists generally of the following equipment:

- Overload relays to prevent the current from the motor to exceed any dangerous level
- Fuse disconnect switches or breakers to protect the motor from short-circuit currents

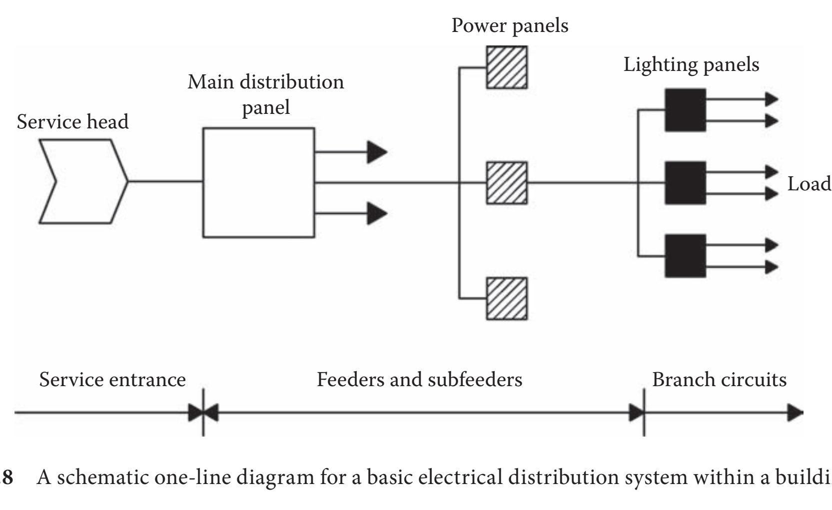


FIGURE 5.8 A schematic one-line diagram for a basic electrical distribution system within a building.

An important part of any electrical system is the electrical wiring that connects all the system components. Three types of connecting wires can be identified:

1. Service entrance conductors are those electrical wires that deliver electricity from the supply system to the facility. For large facilities, electricity is typically supplied by an electric utility at a relatively high voltage (13.8 kV) requiring a transformer (part of a unit substation) to step down the voltage to the utilization level.
2. Feeders are the conductors that deliver electricity from the service entrance equipment location to the branch circuits. Two types of feeders are generally distinguished: the main feeders that originate at the service entrance (or main distribution panel) and the subfeeders that originate at distribution centers (lighting panelboards or motor control centers).
3. Branch circuits are the conductors that deliver electricity to the utilization equipment from the point of the final over-current device.

5.6.2 Transformers

The transformer is the device that changes the voltage level of an alternating current. In particular, it is common to use transformers at generating stations to increase the transmission voltages to high levels (13,800 volts) and near or inside buildings to reduce the distribution voltages to low levels for utilization (480 or 208 volts).

A typical transformer consists of two windings: primary and secondary windings. The primary winding is connected to the power source whereas the secondary winding is connected to the load. Between the primary and secondary windings, there is no electrical connection. Instead, the electric energy is transferred by induction within the core which is generally made up of laminated steel. Therefore, transformers operate only on alternating current.

There are basically two types of transformers (i) liquid-filled transformers and (ii) dry-type transformers. In liquid-filled transformers, the liquid acts as a coolant and as insulation dielectric. Dry-type transformers are constructed so that the core and coils are open to allow for cooling by free movement of air. In some cases, fans may be installed to increase the cooling effect. The dry-type transformers are widely used because of their lighter weight and simpler installation compared to liquid-filled transformers.

A schematic diagram for a single-phase transformer is illustrated in Figure 5.9. A three-phase transformer can be constructed from a set of three single-phase transformers electrically connected so that the primary and the secondary windings can be either wye or delta configurations. For buildings, a delta-connected primary and wye-connected secondary is the most common arrangement for transformers. It can be shown that the primary and the secondary voltages V_p and V_s are directly proportional to the respective number of turns N_p and N_s in the windings:

$$\frac{V_p}{V_s} = \frac{N_p}{N_s} = a \tag{5.26}$$

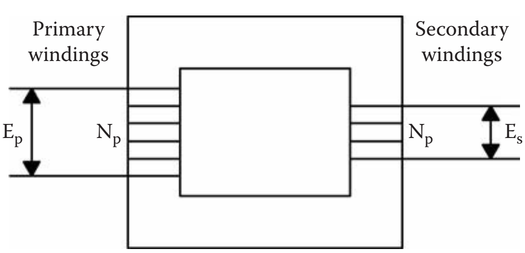


FIGURE 5.9 Simplified model for a single-phase transformer.

where a is the turns ratio of the transformer. As indicated by Eq. (5.25), the turns ratio can be determined directly from the voltages without needing to know the actual number of turns on the transformer windings.

Transformers are rated by their volt-ampere capacity from the secondary windings. For large transformers, the power output in kilovolt-ampere, or kVA, rating is generally used as expressed by Eq. (5.27):

$$kVA = \frac{\sqrt{3}.V_s.I_s}{1000} \tag{5.27}$$

where V_s and I_s are, respectively, the rated line-to-line voltage and the rated line current of the secondary.

Transformers are typically very efficient with energy losses (in the core and windings) representing only 1–2 percent of the transformer capacity. It may be cost-effective to invest in more energy-efficient transformers especially if they are used continuously at their rated capacity as illustrated in Example 5.5.

Example 5.5

Determine the cost-effectiveness of selecting a unit with an efficiency of 99.95 percent rather than 99.90 percent for a 500-kVA rated transformer. Assume that:

- The cost of electricity is \$0.10/kWh.
- The installed costs of 99.0 percent and 99.5 percent efficient transformers are, respectively, \$7,000 and \$9,000.
- The average power factor of the load 0.90.
- The no-load losses are the same for both transformers.

For the analysis consider two cases for the length of time during which the transformer is used at its rated capacity:

(a) 10 hours/day and 250 days/year
(b) 16 hours/day and 300 days/year

Solution

To determine the cost-effectiveness of installing an energy-efficient transformer, a simplified economic analysis is used to estimate the simple payback period. The savings in energy losses in kWh for the high-efficiency transformer can be calculated as follows:

$$kWh_{saved} = N_h.kVA.pf.\left(\frac{1}{\eta_{std}} - \frac{1}{\eta_{eff}}\right)$$

where

- N_s is the total number of hours (per year) during which the transformer is operating at full-load [for case (a) N_s = 10 * 250 = 2,500 hrs/yr; for case (b) N_s =16 * 300 = 4800 hrs/yr].
- kVA is the rated transformer power output [500 kVA].
- pf is the annual average power factor of the load [pf = 0.90].
- η_{std} and η_{eff} are the efficiency of the standard transformer and the efficient transformer (0.990 and 0.995), respectively.

The energy savings and the simple payback period for each case are presented above.
For case (a) $N_s = 2{,}500$ hrs/yr.
The energy savings in kWh is calculated as follows:

$$kWh_{saved} = 2500 * 500 * 0.90 * \left(\frac{1}{0.990} - \frac{1}{0.995} \right) = 5710 kWh / yr$$

Therefore, the simple payback period, SPB, for investing on the efficient transformer is:

$$SPB = \frac{\$9{,}000 - \$7{,}000}{5710 kWh * \$0.07 / kWh} = 5.0 years$$

For case (b) $N_s = 4{,}800$ hrs/yr.
The energy savings in kWh is calculated as follows:

$$kWh_{saved} = 4800 * 500 * 0.90 * \left(\frac{1}{0.990} - \frac{1}{0.995} \right) = 10964 kWh/yr$$

Thus, the payback period SPB, for investing in the efficient transformer, is:

$$SPB = \frac{\$9{,}000 - \$7{,}000}{10964 kWh * \$0.07 / kWh} = 2.6 years$$

It is clear that it can be cost-effective to consider investing in a more energy-efficient transformer especially when the load is supplied during longer periods of time. It should be noted that additional energy savings can be expected during no-load conditions for the energy-efficient transformer.

5.6.3 Electrical Wires

The term electrical wire is actually generic and refers typically to both a conductor and a cable. The conductor is the copper or aluminum wire that actually carries electrical current. The cable generally refers to the complete wire assembly including the conductor, the insulation, and any shielding or protective covering. A cable can have more than one conductor, each with its own insulation.

The size of an electrical conductor represents its cross-sectional area. In the United States, two methods are used to indicate the size of a conductor: the American Wire Gauge (AWG) for small sizes and thousands of circular mils (MCM) for larger sizes. For the AWG method, the available sizes are from number 18 to number 4/0 with the higher number representing the smaller conductor size. For buildings, the smallest size of copper conductor that can be used is number 14 which is rated for a maximum loading of 15 amperes. The AWG size designation soon became inadequate after its implementation in the early 1900s due to the ever-increasing electrical load in buildings. For larger conductors, the cross-sectional area is measured in circular mils. A circular mil corresponds to the area of a circle that has diameter of 1 mil or 1/1,000th of an inch. For instance, a conductor with a half-inch (500 mils) diameter has a circular mil area of 250,000 which is designated by 250 MCM.

To determine the correct size of conductors to be used for feeders and branch circuits in buildings, three criteria generally need to be considered:

1. The rating of the continuous current under normal operating conditions. The National Electric Code (NEC, 1996) refers to the continuous current rating as the ampacity of the conductor. The main parameters that affect the ampacity of a conductor include the physical characteristics of the wire such as its cross-sectional area (or size) and its material and the conditions under which the wire operates such as the ambient temperature and the number of conductors installed in the same cable. Table 5.10 indicates the ampacity rating of copper and aluminum conductors with various sizes. Various derating and correction factors may need to be applied to the ampacity of the conductor to select its size.
2. The rating of short-circuit current under fault conditions. Indeed, high short-circuit currents can impose significant thermal or magnetic stresses not only on the conductor but also on all the components of the electrical system. The conductor has to withstand these relatively high short-circuit

TABLE 5.10 Ampacity of Selected Insulated Conductors Used in Buildings

Conductor Size (AWG or MCM)	THW (Copper)	THHN (Copper)	THW (Aluminum)	THHN (Aluminum)
18	–	14	–	–
16	–	18	–	–
14	20	25	–	–
12	25	30	20	25
10	35	40	30	35
8	50	55	40	45
6	65	75	50	60
4	85	95	65	75
3	100	110	75	85
2	115	130	90	100
1	130	150	100	115
1/0	150	170	120	135
2/0	175	195	135	150
3/0	200	225	155	175
4/0	230	260	180	205
250	255	290	205	230
300	285	320	230	255
350	310	350	250	280
400	335	380	270	305
500	380	430	310	350
600	420	475	340	385
700	460	520	375	420
750	475	535	385	435
800	490	555	395	450
900	520	585	425	480
1,000	545	615	445	500
1,250	590	665	485	545
1,500	625	705	520	585
1,750	650	735	545	615
2,000	665	750	560	630

Source: Adapted from NEC Table 310-16.

currents because the protective device requires some finite time before detecting and interrupting the fault current.

3. The maximum allowable voltage drop across the length of the conductor. Most electrical utilization equipment is sensitive to the voltage applied to it. It is therefore important to reduce the voltage drop that occurs across the feeders and the branch circuits. The NEC recommends a maximum voltage drop of 3 percent for any one feeder or branch circuit with a maximum voltage drop from the service entrance to the utilization outlet of 5 percent.

For more details, the reader is referred to Section 220 of the NEC that covers the design calculations of both feeders and branch circuits.

Two conductor materials are commonly used for building electrical systems: copper and aluminum. Because of its highly desirable electrical and mechanical properties, copper is the preferred material used for conductors of insulated cables. Aluminum has some undesirable properties and its use is restricted. Indeed, an oxide film, which is not a good conductor, can develop on the surface of aluminum and can cause poor electrical contact especially at the wire connections. It should be noted that aluminum can be considered in cases when cost and weight are important criteria for the selection of conductors. However, it is highly recommended even in these cases to use copper conductors for the connections and the equipment terminals to eliminate poor electrical contact.

To protect the conductor, several types of insulation materials are used. The cable (which is the assembly that includes the conductor, insulation, and any other covering) is identified by letter designations depending on the type of insulation material and the conditions of use. In buildings, the following letter designations are used:

- For the insulation material type: A (asbestos), MI (mineral insulation), R (rubber), SA (silicone asbestos), T (thermoplastic), V (varnished cambric), and X (cross-linked synthetic polymer)
- For the conditions of use: H (heat up to 75°C), HH (heat up to 90°C), UF (suitable for underground), and W (moisture resistant)

Thus, the letter designation THW refers to a cable that has a thermoplastic insulation rated for maximum operating temperature of 75°C and suitable for use in dry as well as wet locations).

Moreover, some types of electrical cables have outer coverings that provide mechanical/corrosion protection such as lead sheath (L), nylon jacket (N), armored cable (AC), metal-clad cable (MC), and NM (nonmetallic sheath cable).

For a full description and all types of insulated conductors, their letter designations, and their uses, the reader is referred to the NEC, Article 310 and Table 310-13.

In general, the electrical cables are housed inside conduits for additional protection and safety. The types of conduit commonly used in buildings are listed below:

- Rigid metal conduit (RMC) can be of either steel or aluminum and has the thickest wall of all types of conduit. Rigid metal conduit is used in hazardous locations such as high exposure to chemicals.
- Intermediate metal conduit (IMC) has a thinner wall than the rigid metal conduit but can be used in the same applications.
- Electrical metallic tubing (EMT) is a metal conduit but with very thin walls. The NEC restricts the use of EMT to locations where it is not subjected to severe physical damage during or after installation.
- Electrical nonmetallic conduit (ENC) is made of nonmetallic material such as fiber or rigid PVC (polyvinyl chloride). Generally, rigid nonmetallic conduit cannot be used where subject to physical damage.
- Electrical nonmetallic tubing (ENT) is a pliable corrugated conduit that can be bent by hand. Electrical nonmetallic tubing can be concealed within walls, floors, and ceilings.
- Flexible conduit can be readily flexed and thus is not affected by vibration. Therefore, a common application of the flexible conduit is for the final connection to motors or recessed lighting fixtures.

It should be noted that the number of electrical conductors that can be installed in any one conduit is restricted to avoid any damage to cables (especially when the cables are pulled through the conduit). The NEC restricts the percentage fill to 40 percent for three or more conductors. The percentage fill is defined as the fraction of the total cross-sectional area of the conductors—including the insulation—over the cross-sectional area of the inside of the conduit.

When selecting the size of the conductor, the operating costs and not only the initial costs should be considered. As illustrated in Example 5.6, the cost of energy encourages the installation of larger conductors than are typically required by the NEC especially when smaller-size conductors are involved (i.e., numbers 14, 12, 10, and 8). Unfortunately, most designers do not consider the operating costs in their design due to several reasons including interest in lower first costs and uncertainty in electricity prices.

Example 5.6

Determine if it is economically feasible to install number 10 (AWG) copper conductors instead of number 12 (AWG) on a 400-foot branch circuit that feeds a load of 16 amperes. Assume that

- The load is used 10 hours/day and 250 days/year.
- The cost of electricity is \$0.10/kWh.
- The installed costs of No. 12 and No. 10 conductors are, respectively, \$60.00 and \$90.00 per 1,000-ft long cable.

Solution

In addition to the electric energy used to meet the load, there is an energy loss in the form of heat generated by the flow of current *I* through the resistance of the conductor *R*. The heat loss in watts can be calculated as follows:

$$Watts = R \cdot I$$

Using the information from the NEC (Table 8), the resistance of both conductors Nos. 12 and 10 can be determined to be, respectively, 0.193 ohm and 0.121 ohm per 100 feet. Thus, the heat loss for the 400-ft branch circuit if No. 12 conductor is used can be estimated as follows:

$$Watts_{12} = 0.193 * 400/100.(16)^2 = 197.6W$$

Similarly the heat loss for the 400-ft branch circuit when No. 10 conductor is used is found to be:

$$Watts_{10} = 0.121 * 400/100.(16)^2 = 123.9W$$

The annual cost of copper losses for both cases can be easily calculated:

$$Cost_{12} = 197.6W * 250days/yr * 10hrs/Day * 1kW/1000W * \$0.10/kWh = \$49.4/yr$$

$$Cost_{10} = 123.9W * 250days/yr * 10hrs/Day * 1kW/1000W * \$0.10/kWh = \$31.0/yr$$

Therefore, if No. 10 is used instead of No. 12, the simple payback period SPB for the higher initial cost for the branch circuit conductor is:

$$SPB = \frac{(\$90/1000\,ft - \$60/1000\,ft) * 400\,ft}{(\$49.4 - \$31.0)} = 0.68\,yr = 8 \quad months$$

The savings in energy consumption through the use of larger conductors can thus be cost-effective. Moreover, it should be noted that the larger size conductors reduce the voltage drop across the branch circuit which permits the connected electrical utilization equipment to operate more efficiently. However, the applicable code has to be carefully consulted to determine if a larger size conduit is required when larger size conductors are used.

5.7 Power Quality

5.7.1 Introduction

Under ideal operating conditions, the electrical current and voltage vary as a sine function of time. However, problems due to a utility generator or distribution system such as voltage drops, spikes, or transients can cause fluctuations in the electricity which can reduce the life of electrical equipment including motors and lighting systems. Moreover, an increasing number of electrical devices operating on the system can cause distortion of the sine waveform of the current or voltage. This distortion leads to poor power quality which can waste energy and harm both electrical distribution and devices operating on the systems.

5.7.2 Total Harmonic Distortion

Power quality can be defined as the extent to which an electrical system distorts the voltage or current sine waveform. The voltage and current for an electrical system with ideal power quality vary as a simple sine function of time, often referred to as the fundamental harmonic, and are expressed by Eqs. (5.1) and (5.2), respectively. When the power is distorted due, for instance, to electronic ballasts (which change the frequency of the electricity supplied to the lighting systems), several harmonics need to be considered in addition to the fundamental harmonic to represent the voltage or current time variation as shown in Eqs. (5.28) and (5.29):

$$v(t) = \sum_{k=1}^{N_V} V_k \cos(k\omega - \theta_k) \tag{5.28}$$

$$i(t) = \sum_{k=1}^{N_I} I_k \cos(k\omega - \phi_k) \tag{5.29}$$

Highly distorted waveforms contain numerous harmonics. Although the even harmonics (i.e., second, fourth, etc.) tend to cancel each other's effects, the odd harmonics (i.e., third, fifth, etc.) have their peaks coincide and significantly increase the distortion effects. To quantify the level of distortion for both voltage and current, a dimensionless number referred to as the total harmonic distortion (THD) is determined through a Fourier series analysis of the voltage and current waveforms. The THD for voltage and current are, respectively, defined as follows:

$$THD_V = \sqrt{\frac{\sum_{k=2}^{N_V} V_k^2}{V_1^2}} \tag{5.30}$$

$$THD_I = \sqrt{\frac{\sum_{k=2}^{N_V} I_k^2}{I_1^2}} \tag{5.31}$$

Table 5.11 provides the current THD for selected but specific lighting and office equipment loads (NLPIP, 1995). Generally, it is found that devices with high-current THD contribute to voltage THD in proportion to their share of the total building electrical load. Therefore, the engineer should consider higher-wattage devices before lower devices to reduce the voltage THD for the entire building or facility. Example 5.7 shows a simple calculation procedure that can be used to assess the impact of an electrical device on the current THD. Thus, the engineer can determine which devices need to be corrected first to improve the power quality of the overall electric system. Typically, harmonic filters are added to electrical devices to reduce the current THD values.

Example 5.7

Assess the impact of two devices on the current THD of a building: 13-W compact fluorescent lamp (CFL) with electronic ballasts and a laser printer while printing. Use the data provided in Table 5.11.

Solution

Both devices have an rms voltage of 120 V (i.e., $V_{\text{rms}} = 120$ V); their rms current can be determined using the real power used and the power factor given in Table 5.11 and Eq. (5.5):

$$I_{rms} = \frac{P_R}{V_{rms}.pf}$$

The above equation gives an rms current of 0.22 A for the CFL and 6.79 A for the printer. These values correspond actually to the rms of each device's fundamental current waveform and can be used in the THD equation, Eq. (5.29), to estimate the total harmonic current of each device:

$$I_{tot} = I_{rms}.THD_I$$

The resultant values of 0.33 A for the CFL and 1.02 A for the printer show that although the printer has a relatively low current THD (15 percent), the actual distortion current produced by the printer is more than three times that of the CFL because the printer uses more power.

TABLE 5.11 Typical Power Quality Characteristics (Power Factor and Current THD) for Selected Electrical Loads

Electrical Load	Real Power used (W)	Power factor	Current THD (%)
Incandescent lighting systems			
100-W incandescent lamp	101	1.0	1
Compact fluorescent lighting systems			
13-W lamp w/ magnetic ballast	16	0.54	13
13-W lamp w/electronic ballast	13	0.50	153
Full-size fluorescent lighting systems (2 lamps per ballast)			
T12 40-W lamp w/ magnetic ballast	87	0.98	17
T12 40-W lamp w/ electronic ballast	72	0.99	5
T10 40-W lamp w/ magnetic ballast	93	0.98	22
T10 40-W lamp w/ electronic ballast	75	0.99	5
T8 32-W lamp w/ electronic ballast	63	0.98	6
T5 28-W lamp w/ electronic ballast	62	0.95	15
High-intensity discharge lighting systems			
400-W high-pressure sodium lamp w/ magnetic ballast	425	0.99	14
400-W metal halide lamp w/ magnetic ballast	450	0.94	19
Office equipment			
Desktop computer w/o monitor	33	0.56	139
Color monitor for desktop computer	49	0.56	138
Laser printer (in standby mode)	29	0.40	224
Laser printer (printing)	799	0.98	15
External fax/modem	5	0.73	47

Source: Adapted from NLPIP, Power quality. Lighting answers. *Newsletter from the National Lighting Product Information Program,* 2(2), 5, 1995 and 2002.

IEEE (1992) recommends a maximum allowable voltage THD of 5 percent at the building service entrance (i.e., point where the utility distribution system is connected to the building electrical system). Based on a study by Verderber, Morse, and Alling (1993), the voltage THD reaches the 5 percent limit when about 50 percent of the building electrical load has a current THD of 55 percent or when 25 percent of the building electrical load has a current THD of 115 percent.

It should be noted that when the electrical device has a power factor of unity (i.e., $pf = 1$), there is little or no current THD (i.e., $THD_I = 0$ percent) because the device has only a resistive load and effectively converts input current and voltage into useful electric power. As shown in Table 5.11, the power factor and the current THD are interrelated and both define the characteristics of the power quality. In particular Table 5.11 indicates that lighting systems with electronic ballasts typically have a high power factor and low current THD. This good power quality is achieved using capacitors to reduce the phase lag between the current and voltage (thus improving the power factor as discussed in Section 5.2) and filters to reduce harmonics (and therefore increase the current THD value).

The possible problems that have been reported due to poor power quality include:

1. Overload of neutral conductors in three-phase with four wires. In a system with no THD, the neutral wire carries no current if the system is well balanced. However, when the current THD becomes significant, the currents due to the odd harmonics do not cancel each other and rather add up on the neutral wire which can overheat and cause a fire hazard.
2. Reduction in the life of transformers and capacitors. This effect is mostly caused by distortion in the voltage.
3. Interference with communication systems. Electrical devices that operate with high frequencies such as electronic ballasts (that operate at frequencies ranging from 20 to 40 kHz) can interfere and disturb the normal operation of communication systems such as radios, phones, and energy management systems (EMS).

5.8 Summary

In this chapter, an overview is provided for the basic characteristics of electrical systems in HVAC applications for buildings. In particular, the operating principles of motors are emphasized. Throughout the chapter, several measures are described to improve the energy performance of existing or new electrical installations. Moreover, illustrative examples are presented to evaluate the cost-effectiveness of selected energy efficiency measures. For instance, it was shown that the use of larger conductors for branch circuits can be justified based on the reduction of energy losses and thus operating costs. Moreover, the chapter provided suggestions to improve the power quality, increase the power factor, and reduce lighting energy use in buildings. These suggestions are presented to illustrate the wide range of issues that an engineer should address when retrofitting electrical systems for buildings.

Problems

5.1 Provide a simple payback period analysis of lighting controls in a 10,000-square-foot office area comprised of 7,200 square feet of open landscaped office space and 20 enclosed perimeter offices (2,800 square feet). The total lighting load used for the space is 17 kW and the office is operated 260 days per year. The open space is monitored by 24 ceiling-mounted motion sensors. The perimeter offices are monitored with wall-mounted sensors (one for each office). The installed cost of a ceiling sensor is $110 with a $64 utility rebate. The wall sensor costs $75 with a $32 utility rebate.
Determine the simple payback periods with and without rebate. Assume the additional off-time (hours/day) is 1, 2, 4, 6, and 8. Two scenarios for electricity cost: $0.07/kWh and $0.14/kWh. Show your results in a tabular format.

5.2 Two motors operate 5,000 h/hr at full load. One is 40 hp with an efficiency of 0.75 and a power factor of 0.65 and the other is 100 hp with an efficiency of 0.935 and a power factor of 0.85. Determine:
(a) The overall power factor.
(b) The simple payback period of replacing each motor by and energy-efficient motor. Assume that the cost of electricity is $0.07/kWh.

5.3 Determine the capacitor ratings (in kVAR) to add to an 80 hp motor with an efficiency of 0.85 to increase its power factor from 0.80 to 0.85, 0.90, and 0.95, respectively.

5.4 Consider an aluminum fabrication plant with 400 employees that runs on three shifts for a total of 8,760 hours per year. The company spent $1.1 million in electric energy bills last year. The plant uses numerous motors to drive process-related equipment such as shell presses and compound presses. Determine if it is cost-effective to replace the existing motors with high-efficient motors under the following three rate structures:

For all the rates, the utility company charges the facility an average cost of electricity, without demand, of $0.05/kWh. The demand charges are:

(a) Rate-1: 7.02/kVA

(b) Rate-2: $5.00/kW (billed demand)

(c) Rate-3: $5.00/kW plus $0.75 for excess KVAR above 6 percent of real demand

The motors that will be replaced are listed below.

hp	N	LF[a]	eff-s	eff-e	pf-s	pf-e
100	4	0.75	0.919	0.950	0.872	0.905
60	2	0.75	0.916	0.940	0.854	0.861
40	13	0.50	0.908	0.934	0.797	0.805
20	2	0.50	0.886	0.923	0.759	0.833
15	12	0.50	0.875	0.916	0.681	0.774
10	5	0.50	0.864	0.910	0.714	0.770
7.5	57	0.50	0.846	0.902	0.683	0.731
5	43	0.50	0.839	0.890	0.687	0.714
2	14	0.50	0.791	0.864	0.516	0.540
1.5	2	0.50	0.780	0.852	0.546	0.580

[a] LF indicates the load factor of the motor.

5.5 A daylighting system using dimming controls is being considered for a five-story office building located in Denver, Colorado. A typical floor is 200 ft by 50 ft with perimeter offices extending 12 ft from the vertical windows. The electrical lighting has a density of 1.2 W/ft^2 and is operated (without daylighting controls) 5,500 hours/year. The cost of the daylighting system is estimated at $40,000. The energy cost is $0.08/kWh.

(i) Assess the cost-effectiveness of the daylighting system using a 20-yr cycle and 5 percent discount rate if double clear glazing is used. Consider three WWR values 10, 20, and 30 percent.

(ii) Repeat (i) for double reflective tinted glazing.

5.6 Determine the cost-effectiveness of adding a dimming daylighting control system to a thin two-story office building (each floor is 24 ft by 1,000 ft) with a window-to-floor area ratio of 30 percent. The building has clear windows (i.e., visible transmittance = 0.78) and is located in Denver, Colorado. The lighting system has a density of 1.1 W/sq ft and operates 3,000 hours per year.

The cost of installing the daylighting system (sensors, controls, and dimming ballasts) is estimated at $45,000. Assume that the cost of electricity is $0.10/kwh and that the interest rate is 8 percent with an inflation rate of 3 percent.

5.7 The nameplate of a motor provides the following information:

Full load hp	100
Volts	440/220
Amperes	123/246
Full load RPM	1,775
Efficiency	0.92

During an audit, measurements on the motor indicated that:

Average volts	440
Amperes	116
Full load RPM	1,779

(a) Calculate the following parameters: the slip RPM and the power factor of the motor

(b) Determine if it is cost-effective to replace this motor with a more efficient motor (You need to document the reference for the motor cost). Perform the calculations with and without adjusting the speed of the energy efficient motor.

5.8 Consider a three-phase 1,500 kVA transformer to step-down the voltage from 13.8 kV to 480Y/277 volt. The transformer is old and needs to be replaced. Two options are available: (i) replace it by the same transformer with an efficiency of 98.6 percent and an installed cost of $45,000 and (ii) replace it by a more energy-efficient transformer with an efficiency of 99.0 percent and an installed cost of $60,000.

(a) Determine the best option for the transformer replacement considering a lifetime of 25 years, and electricity price of $0.06, a discount rate of 4.1 percent, and an average load factor of 50 percent.

(b) Determine the electricity price for which the energy-efficient motor is not cost-effective using the same assumptions of question (a).

(c) Determine the size of the main secondary feeders for both the standard and the energy-efficient transformers. Calculate the energy savings if the next higher feeder size is used.

6 Building Envelope

6.1 Introduction

Generally, the envelope of a structure is designed by architects to respond to many considerations including structural and aesthetic. Before the oil crisis of 1973, the energy efficiency of the envelope components was rarely considered as an important factor in the design of a building. However, since 1973 several standards and regulations have been developed and implemented to improve the energy efficiency of various components of building envelopes. For energy retrofit analysis, it is helpful to determine if the building was constructed or modified to meet certain energy efficiency standards. If it is the case, retrofitting of the building envelope may not be cost-effective especially for high-rise commercial buildings. However, improvements to the building envelope can be cost-effective if the building or industrial facility was built without any concern for energy efficiency such as the case with structures constructed with no insulation provided in the walls or roofs.

Moreover, the building envelope retrofit should be performed after careful assessment of the building thermal loads. For instance in low-rise buildings such as residential and small commercial buildings or warehouses, the envelope transmission losses and infiltration loads are dominant and the internal loads within these facilities are typically low. Meanwhile in high-rise commercial, industrial, and institutional facilities, the internal heat gains due to equipment, lighting, and people are typically dominant and the transmission loads affect only the perimeter spaces.

The accurate assessment of the energy savings incurred by building envelope retrofits generally requires detailed hourly simulation programs because the heat transfer in buildings is complex and involves several mechanisms. In this chapter, only simplified calculation methods are presented to estimate the energy savings for selected building envelope improvements commonly proposed to improve not only the energy efficiency of the building but also the thermal comfort of its occupants and the structural integrity of its shell.

6.2 Basic Heat Transfer Concepts

The heat transfer from the building envelope can occur by various mechanisms including conduction, convection, and radiation. In this section, various fundamental concepts and parameters are briefly reviewed. These concepts and parameters are typically used to characterize the thermal performance of various components of the building envelope and are useful to estimate the energy use savings accrued by retrofits of building envelope.

6.2.1 Heat Transfer from Walls and Roofs

In buildings, the heat transfer through walls and roofs is dominated by conduction and convection. Typically, one-dimensional heat conduction is considered to be adequate for above-grade building components unless significant thermal bridges exist such as at the wall corners or at the slab edges. Specifically, the heat transfer from a homogeneous wall or roof layer illustrated in Figure 6.1 can be calculated as follows using the Fourier law:

$$\dot{q} = \frac{k}{d}.A.(T_i - T_o) \tag{6.1}$$

where
 A = the area of the wall.
 T_i = inside wall surface temperature.
 T_o = outside wall surface temperature.
 k = thermal conductivity of the wall.
 d = the thickness of the wall.

To characterize the heat transfer presented by Eq. (6.1), a thermal resistance R-value or a U-value is defined for the layer as shown below:

$$R = \frac{d}{k} = \frac{1}{U} \tag{6.2}$$

where
 h = the convective heat transfer coefficient of the surface.

The concept of thermal resistance can be extended to convection heat transfer that occurs at the outer or inner surfaces of the building envelope:

$$R_{conv} = \frac{1}{h} \tag{6.3}$$

In buildings, a wall or a roof consists of several layers of homogeneous materials as illustrated in Figure 6.2.

The heat transfer from a multilayered wall or roof can be found by determining first its overall R-value:

$$R_T = \sum_{j=1}^{N_L} R_j \tag{6.4}$$

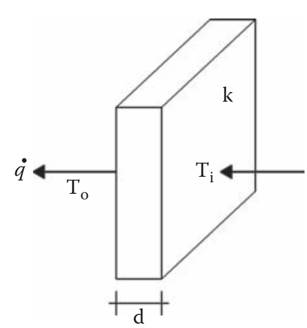


FIGURE 6.1 Conduction heat transfer through one-layer wall.

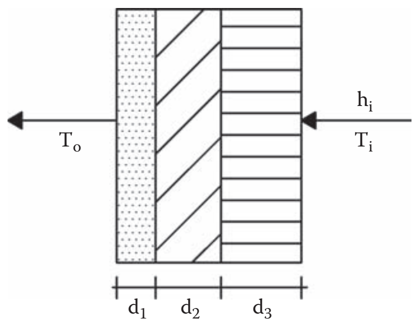


FIGURE 6.2 Heat transfer from a multilayered wall.

where

R_j = the R-value of each homogeneous layer part of the construction of the wall or roof assembly. It includes the R-value due to convection at both inner and outer surfaces of the wall or roof obtained by Eq. (6.3).

N_L = the number of layers (including the convection boundary layers) that are part of the wall or roof assembly. For instance, in the wall assembly presented in Figure 6.2, $N_L = 5$ (3 conductive layers and 2 convective layers).

The overall U-value of the wall or roof can be defined simply as the inverse of the overall R-value:

$$U_T = \frac{1}{R_T} \tag{6.5}$$

It should be noted that practitioners usually prefer to use R-values rather than U-values inasmuch as the U-values are small especially when insulation is added to the wall or roof assembly. For doors and windows, the use of U-values is more common because these components have low R-values.

From Eq. (6.1), it is clear that in order to reduce the heat transfer from the above-grade building envelope components, its R-value should be increased or its U-value decreased. To achieve this objective, thermal insulation can be added to the building envelope. In the next section, calculation methods of the energy savings due to addition of insulation are presented to determine the cost-effectiveness of such a measure.

To characterize the total heat transmission of the entire building, a building load coefficient (BLC) is defined to account for all the above-grade building envelope components (roofs, walls, doors, and windows):

$$BLC = \sum_{i=1}^{N_E} A_i . U_{T,i} = \sum_{i=1}^{N_E} \frac{A_i}{R_{T,i}} \tag{6.6}$$

where A_i is the area of each element of the above-grade building envelope including walls, roofs, windows, and doors.

6.2.2 Infiltration Heat Loss/Gain

Air can flow in or out of the building envelope through leaks. This process is often referred to as air infiltration or exfiltration. Thus, infiltration (and exfiltration) is rather an uncontrolled flow of air unlike ventilation (and exhaust) for which air is moved by mechanical systems. Generally, air infiltration occurs in all buildings but is more important for smaller buildings such as detached residential buildings. In larger buildings, air infiltration is typically less significant for two reasons:

1. The volume over the envelope surface area (from which air leakage occurs) is small for larger buildings.
2. The indoor pressure is generally maintained higher than outdoor pressure by mechanical systems in larger buildings.

Typically, infiltration is considered significant for low-rise buildings and can affect energy use, thermal comfort, and especially structural damage through rusting and rotting of the building envelope materials due to the humidity transported by infiltrating or exfiltrating air. Without direct measurement, it is difficult to estimate the leakage air flow through the building envelope. There are two basic measurement techniques that allow estimation of the infiltration characteristics for a building. These measurement techniques include fan pressurization or depressurization techniques and tracer gas techniques.

Fan pressurization/depressurization techniques are commonly known as blower door tests and allow the estimation of the volumetric air flow rate variation with the pressure difference between the outdoors and indoors of a building. Several pressure-differential values are typically considered and a correlation is found in the form of:

$$\dot{V} = C.\Delta P^n \tag{6.7}$$

where C and n are correlation coefficients determined by fitting the measured data of pressure differentials and air volumetric rates. Using the correlation of Eq. (6.7), an effective leakage area (ELA) can be determined as follows:

$$ELA = \dot{V}_{ref}.\sqrt{\rho/_{2.\Delta P}} \tag{6.8a}$$

Using English units, the effective leakage area (in inches squared) can be estimated using a modified Eq. (6.8a) as follows:

$$ELA = 0.186\dot{V}_{ref}.\sqrt{\rho/_{2.\Delta P}} \tag{6.8b}$$

where $\dot{V}_{ref}$ is the reference volume air rate through the building at a reference pressure difference (between indoors and outdoors) of typically 4 Pa and obtained by extrapolation from Eq. (6.7). The ELA provides an estimate of the equivalent area of holes in the building envelope through which air leaks can occur.

To determine the building air infiltration rate under normal climatic conditions (due to wind and temperature effects), the LBL infiltration model developed by Sherman and Grimsrud (1980) is commonly used:

$$\dot{V} = ELA.\left(f_s.\Delta T + f_w.v_w^2\right)^{1/2} \tag{6.9}$$

where ΔT is the indoor–outdoor temperature difference, v_w is the period-average wind speed, and f_s and f_w are the stack and wind coefficients, respectively. Table 6.1 provides the crack coefficients for three levels of building heights. Table 6.2 lists the wind coefficients for various shielding classes and building heights.

Blower door tests are still being used to find and repair leaks in low-rise buildings. Typically, the leaks are found by holding a smoke source and watching where the smoke exits the house. Several weatherstripping methods are available to reduce air infiltration through the building envelope

TABLE 6.1 Stack Coefficient, f_s

	IP Units[a] House Height (Stories)			SI Units[b] House Height (Stories)		
	One	Two	Three	One	Two	Three
Stack Coefficient	0.0150	0.0299	0.0449	0.000139	0.000278	0.000417

Source: ASHRAE, *Handbook of Fundamentals*, Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 2009.

[a] IP Units for f_s: $(ft^3/min)^2/in^4 \times °F$

[b] SI Units for f_s: $(L/sft)^2/cm^4 \times °C$

TABLE 6.2 Wind Coefficient, f_w

Shielding Class[c]	IP Units[a] House Height (Stories)			SI Units[b] House Height (Stories)		
	One	Two	Three	One	Two	Three
1	0.0119	0.0157	0.0184	0.000319	0.000420	0.000494
2	0.0092	0.0121	0.0143	0.000246	0.000325	0.000382
3	0.0065	0.0086	0.0101	0.000174	0.000231	0.000271
4	0.0039	0.0051	0.0060	0.000104	0.000137	0.000161
5	0.0012	0.0016	0.0018	0.000032	0.000042	0.000049

Source: ASHRAE, *Handbook of Fundamentals*, Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 2009.

[a] IP Units for f_w: $(ft^3/min)^2/in^4 \times mph$

[b] SI Units for f_w: $(L/sft)^2/cm^4 \times (m/s)^2$

[c] Description of shielding classes: 1–no obstructions or local shielding; 2–light local shielding: few obstructions, few trees, or small shed; 3– moderate local shielding: some obstructions within two house height, thick hedge, solid fence, or one neighboring house; 4– heavy shielding: obstructions around most of perimeter, buildings or trees within 30 ft (10 m) in most directions; typical suburban shielding; 5–very heavy shielding: Large obstructions surrounding perimeter within two house heights; typical downtown shielding.

including caulking, weatherstripping, landscaping around the building to reduce the wind effects, and installing air barriers to tighten the building envelope.

It should be mentioned, however, that the blower door technique cannot be used to determine accurately the amount of fresh air supplied to the building through either infiltration or ventilation. For this purpose, it is recommended to use the tracer gas techniques described below.

In a typical blower test, the house should first be prepared. In particular, windows are closed, interior doors that are normally open are kept open, and the fireplace ash is cleaned. The main entrance door is generally used to place the blower fan to either introduce air (for the pressurization test) or extract air (for the depressurization test). The airflow rate is generally measured using a pressure gauge attached to the blower setup. The pressure gauge should first be checked to make sure that it reads zero with the fan set to off. An additional pressure gauge is used to measure the differential in pressure between the inside and outside of the house. Figure 6.3 shows the setup for both the depressurization and pressurization tests. Example 6.1 illustrates how the results of blower door tests can be used to determine the infiltration rate in a house. The results and the analysis presented in Example 6.1 are based on actual tests performed by Azerbegi, Hunsberger, and Zhou (2000).

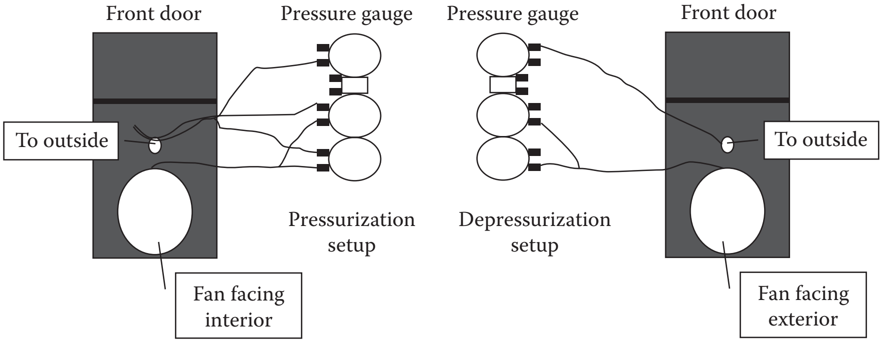


FIGURE 6.3 Typical blower door setup for both pressurization and depressurization tests.

Example 6.1

A blower door test has been performed in a house located in Evergreen, Colorado. The results of the pressurization and depressurization tests are summarized in Figure 6.4 based on the results of the blower door. Determine the leakage areas and ACHs for both pressurization and depressurization tests.

Solution

To determine the air leakage characteristics of a house using blower door tests, the following procedure is used:

(i) First, the data consisting of pressure differential (ΔP [Pa]) and airflow rate ($\dot{V}$ [CFM]) presented in Figure 6.3 is plotted in a log–log scale as illustrated in Figure 6.5.

(ii) Then, a linear regression analysis is used to determine the coefficients C and n of Eq.(6.7):

$$\dot{V} = C.\Delta P^n$$

- For the pressurization test, the coefficients C and n are found to be $C = 552.25$ and $n = 0.679$.
- For the depressurization test, the coefficients C and n are found to be $C = 556.13$ and $n = 0.694$.

It should be noted that the values of C and n provided above are valid only if the airflow rate $\dot{V}$ and pressure differential ΔP are expressed in cfm (cubic feet per minute) and Pa, respectively.

(iii) Based on the correlation and the coefficients provided above, the infiltration and exfiltration rates under a normal pressure differential ($\Delta P_{ref} = 4$ Pa) can be calculated:

- For pressurization: $\dot{V}_{ref} = 552.25 * (4)^{0.679} = 1455$ cfm
- For depressurization: $\dot{V}_{ref} = 556.13 * (4)^{0.694} = 1416$ cfm

The leakage areas for both pressurization and depressurization tests can be obtained using Eq. (6.8b):

$$ELA = 0.186\dot{V}_{ref}.\sqrt{\rho/2.\Delta P}$$

Pressurization Test		Depressurization Test	
Pressure [Pa]	Airflow [ft3/min]	Pressure [Pa]	Airflow [ft3/min]
0	0	0	0
7	2000	6.5	2100
7.5	2250	8	2250
8.5	2450	9	2700
11	2850	12	3100
10.5	2525	13	3200
12.5	3150	14.5	3500
13	3200	16.2	3625
14.2	3326	17	4000
16	3700	19	4400
18.5	3950	21	4575
20.5	4250	23	4975
24.5	4850	24.5	5225
28.5	5300	25	5250
32.5	6000	27.5	5500

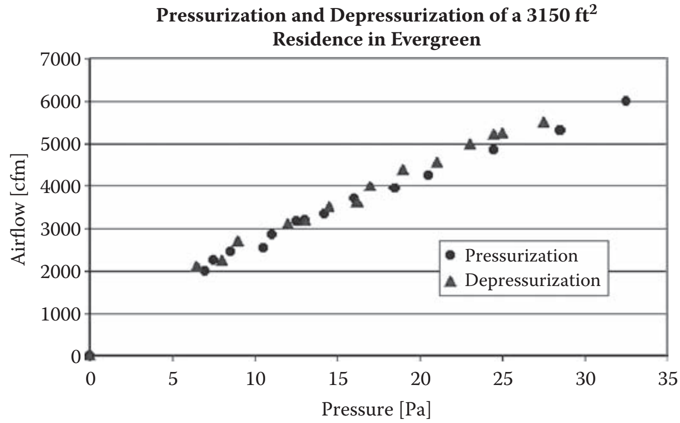


FIGURE 6.4 Summary of the blower door results for both pressurization and depressurization tests.

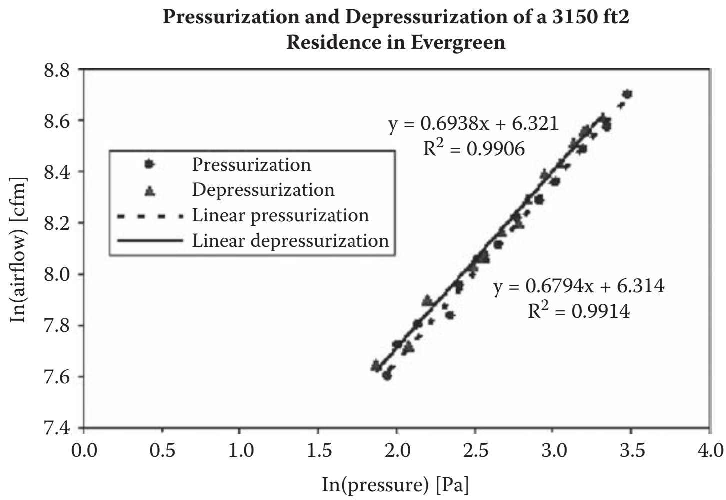


FIGURE 6.5 Summary of the blower door results for both pressurization and depressurization tests.

For Evergreen, Colorado, the air density should be adjusted for altitude and is found to be: _ = 0.063 lbm/ft^3.

- For pressurization: ELA = 369.6 in.^{2}
- For depressurization ELA = 379.7 in.^{2}

Therefore, the average leakage area for the house based on both pressurization and depressurization tests is:

$$ELA(\text{average}) = 2.60 \text{ ft}^2 = 374.4 \text{ in.}^2$$

(iv) The annual average leakage air flow rate expressed in air change per hour (ACH) can be determined using the LBL model expressed by Eq. (6.9):

$$\dot{V} = ELA.\left(f_s.\Delta T + f_w.v_w^2\right)^{1/2}$$

To determine the average ACH for the house, the average ELA of 374.4 in.^{2} is used. Moreover, the coefficients $f_s = 0.0266$ and $f_w = 0.0051$ need to be used for a two-story building for a shielding class of 4 (refer to Tables 6.1 and 6.2). The annual average wind speed is V_w (for Denver) = 8.5 mi/hr, and the annual average outdoor temperature $t$out = 46.17°F (for Evergreen). To account for the effect of indoor temperature setback (during the winter season), an annual average indoor temperature of 60°F is considered as the annual average leakage air flow rate $\dot{V}$:

$$\dot{V} = 374.9 * \left(0.026 * [60 - 46.2) + 0.0051 * (8.5)^2\right)^{1/2} = 331 cfm$$

Then, the volume flow rate $\dot{V}$ is divided by the conditioned volume of the house (in this case, 28,350 ft^3) to obtain the annual average ACH:

$$ACH = \frac{331 cfm * 60 \min/hr}{28{,}350 ft^3} = 0.70$$

It should be noted that ASHRAE recommends a leakage of 0.35 ACH for proper ventilation of a house with minimum heat loss.

Tracer gas techniques are commonly used to measure the ventilation rates in buildings. By monitoring the injection and the concentration of a tracer gas (a gas that is inert, safe, and mixes well with air), the exchange of air through the building can be estimated. For instance, in the decay method, the injection of a tracer gas is performed for a short time and then stopped. The concentration of the decaying tracer gas is then monitored over time. The ventilation is measured by the air change rate within the building and is determined from the time variation of the tracer gas concentration:

$$c(t) = c_O . e^{-ACH.t} \tag{6.10}$$

or

$$ACH = \frac{\dot{V}}{V_{bldg}} = \frac{1}{t} . Ln[\frac{c_O}{c(t)}] \tag{6.11}$$

where C_o is the initial concentration of the tracer gas, and V_{bldg} is the volume of the building.

The thermal load of air infiltration is rather difficult to assess. It has been traditionally thought that the sensible thermal load due to infiltration air is simply calculated as follows:

$$E_{\inf} = \rho . c_{p,a} . \dot{V}_{\inf} . (T_i - T_o) = \dot{m}_{\inf} . c_{p,a} . (T_i - T_o) \tag{6.12}$$

Equation (6.12) assumes that infiltrating air entering the condition space (kept at temperature T_i) has the same temperature as the outdoor temperature T_o. However, various recent studies showed that actually infiltrating air can warm up through the building envelope before entering a condition space. This heat exchange occurs especially when the air leakage occurs in a diffuse manner (through long airflow channels inside the building envelope). As a consequence the actual thermal load due to air infiltration is lower than that determined by Eq. (6.12) by a fraction that depends on the heat exchange rate between air infiltration and heat conduction through building envelope.

When the building envelope is not airtight, it can be assumed that the airflow by infiltration occurs through direct and short paths from outdoors to the indoors without significant heat recovery and thus Eq. (6.12) can be used to estimate the thermal load due to air infiltration.

6.2.3 Variable Base Degree-Days Method

The degree-days method provides an estimation of the heating and cooling loads of a building due to transmission losses through the envelope and any solar and internal heat gains. The degree-days method is based on steady-state analysis of the heat balance across the boundaries of the building. A building is typically subject to several heat flows including conduction, infiltration, solar gains, and internal gains as illustrated in Figure 6.6. The net heat loss or heat gain at any instant is determined by applying a heat balance (i.e., the first law of thermodynamics) to the building. For instance, for heating load calculation, the instantaneous heat balance provides:

$$\dot{q}_H = BLC.(T_i - T_o) - \dot{q}_g \tag{6.13}$$

where

BLC = the building load coefficient as defined in Eq.(6.6) but modified to include the effects of both transmission and infiltration losses. Thus, the BLC for any building can be calculated as follows:

$$BLC = \sum_{j=1}^{N_E} U_{T,j}.A_j + \dot{m}_{\text{inf}}.c_{p,a} \tag{6.14}$$

q_g = the net heat gains due solar radiation q_{sol}; internal gains (people, lights, and equipment) q_{int}; and in some cases the ground losses q_{grd}, if they are significant:

$$\dot{q}_g = \dot{q}_{sol} + \dot{q}_{\text{int}} - \dot{q}_{grd}$$

This equation can be rearranged to introduce the balance temperature T_b for the building

$$q_H = BLC.\left[(T_i - \frac{q_g}{BLC}) - T_o\right] = BLC.(T_b - T_o) \tag{6.15}$$

with the balance temperature defined as

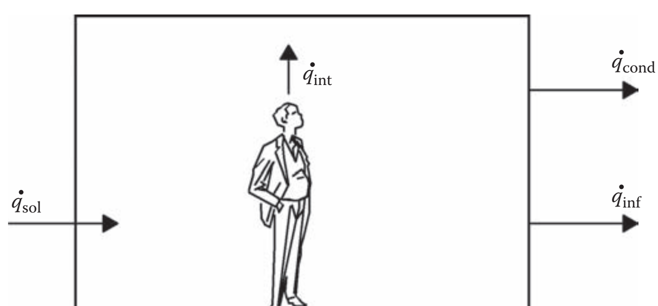


FIGURE 6.6 Flow rate as a function of the pressure difference in a log–log scale.

$$T_b = T_i - \frac{q_g}{BLC} \tag{6.16}$$

Therefore, the balance temperature adjusts the interior temperature set-point by the amount of temperature increase due to a reduction in the building heating load resulting from the internal gains. Before the oil crisis, the transmission and the infiltration losses were significant (and thus the BLC value was high relative to the internal gains). It is estimated that the net internal gains contribute to about 3°C (or 5°F) in most buildings. Therefore, the balance temperature was assumed to be 18°C (or 65°F) for all the buildings. However, with the increase in thermal efficiency of the building envelope and the use of more equipment within the buildings, the internal heat gains are greater and thus can contribute in significantly reducing the heating load of the buildings.

By integrating the instantaneous heating load over the heating season, the total building heating load can be determined. Note only the positive values of q_H are considered in the integration. In practice, the integration is approximated by the sum of the heating loads averaged over short time intervals (one hour or one day). If daily averages are used, the seasonal total building heating load is estimated as:

$$Q_H = 24.\sum_{i=1}^{N_H} \dot{q}_{H,i}^{+} = 24.BLC.\sum_{i=1}^{N_H} (T_b - T_{o,i})^+ \tag{6.17}$$

The sum is performed over the number N_H of days in the heating season. From Eq. (6.17), a parameter that characterizes the heating load of the building can be defined as the heating degree-days (DD_H) which are a function of only the outdoor temperatures and the balance temperature which varies with the building heating set-point temperature and the building internal gains:

$$DD_H(T_b) = \sum_{i=1}^{N_H} (T_b - T_{o,i})^+ \tag{6.18}$$

The total energy use E_H to meet the heating load of the building can be estimated by assuming a constant efficiency of the heating equipment over the heating season (for instance, several heating equipment manufacturers provide the annual fuel use efficiency rating or AFUE for their boilers or furnaces):

$$E_H = \frac{Q_H}{\eta_H} = \frac{24.BLC.DD_H(T_b)}{\eta_H} \tag{6.19}$$

The variable base degree-days method stated by Eq.(6.19) can also be applied to determine the cooling load by estimating the cooling season degree days (DD_C) using an equation similar to Eq. (6.17):

$$DD_C(T_b) = \sum_{i=1}^{N_C} (T_{o,i} - T_b)^+ \tag{6.20}$$

where N_c is the number of days in the cooling season.

It should be noted that the variable base degree-days method can provide a remarkably accurate estimation of the annual energy use due to heating especially for buildings dominated by losses through the building envelope including infiltration. Unfortunately, the degree-days method is not as accurate for calculating the cooling loads (Claridge, Krarti, and Bida, 1987) due to several factors including effects of building thermal mass that delays the action of internal gains, mild outdoor temperatures in summer resulting in large errors in the estimation of the cooling degree-days, and the large variation in infiltration or ventilation rates as occupants open windows or economizer cycles are used.

6.3 Simplified Calculation Tools for Building Envelope Audit

To determine the cost-effectiveness of any energy conservation measure for the building envelope, the energy use savings has to be estimated. In this section, a general calculation procedure based on the variable-base degree-days method is provided with some recommendations to determine the values of the parameters required to estimate the energy use savings.

6.3.1 Estimation of the Energy Use Savings

When an energy conservation measure is performed to improve the efficiency of the building envelope (for instance, by adding thermal insulation to a roof or by reducing the air leakage area for the building envelope), the building load coefficient is reduced. Assuming no change in the indoor temperature setpoint and in the internal gains within the building, the heating balance temperature actually decreases due to the envelope retrofit as can be concluded from the definition of the heating balance temperature illustrated by Eq. (6.15). Therefore, the envelope retrofit reduces the heating load and thus the energy use because both the *BLC* and the $DD(T_b)$ are reduced. The energy use savings due to the retrofit can be generally calculated as follows:

$$DE_{H,R} = E_{H,E} - E_{H,R} = \frac{24.\left(BLC_E.DD_H(T_{b,E}) - BLC_R.DD_H(T_{b,R})\right)}{h_H} \tag{6.21}$$

The efficiency of the heating system is assumed to remain the same before and after the retrofit. It is generally the case unless the heating system is replaced or retrofitted. In many applications, the variation caused by the retrofit of the balance temperature is rather small. In these instances, the degree-days can be considered constant before and after the retrofit so that the energy use savings can be estimated more easily with the following equation:

$$\Delta E_{H,R} = \frac{24.\left(BLC_E - BLC_R\right).DD_H(T_{b,E})}{\eta_H} \tag{6.22}$$

Note than when only one element of the building envelope is retrofitted (for instance, the roof), the difference ($BLC_E - BLC_R$) is equivalent to the difference in the roof *UA* values before and after the retrofit (i.e., $UA_{roof,E} - UA_{roof,R}$).

To use either Eq. (6.21) or Eq. (6.22), it is clear that the auditor needs to estimate the heating degree-days and the existing overall building load. Some recommendations on how to calculate these two parameters are summarized below.

6.3.2 Estimation of the BLC for the Building

The building load coefficient can be estimated using two approaches as briefly described below. Depending on the data available, the auditor should select the appropriate approach.

1. *Direct Calculation:* The auditor should have all the data (either through the architectural drawings or from observation during a site walk-through) needed to estimate the *R*-value or *U*-value of all the components of the building envelope and their associated surface areas. Several references are available to provide the *R*-value of various construction layers commonly used in buildings (ASHRAE, 2009). In addition, the auditor should estimate the infiltration/ventilation rates either by rules of thumb or by direct measurement as discussed in Section 6.2.2. With this data, the building load coefficient can be calculated using Eq. (6.14).
2. *Indirect Estimation:* In this method, the auditor can rely on the utility energy use (even monthly data would be sufficient for this purpose) and its correlation with the outdoor temperature to

provide an accurate estimation of the BLC. This method is similar to the Princeton Scorekeeping Methods (PRISM) described in detail by Fels (1986). Examples are illustrated by Figures 6.7 and 6.8 to determine the BLC for the heating and the cooling mode, respectively. In both figures, the BLC is determined by the slope of the regression line correlating the building energy use to the outdoor air temperature. It should be noted that the outdoor air temperature should be averaged over the same periods for which the utility data Is available.

6.3.3 Estimation of the Degree Days

Data for heating degree-days can be found in several sources for various values of balance temperature. Table 6.3 provides heating degree-days for selected U.S. cities for various balance temperatures: 65°F (18°C). Additional degree-days data published in the *ASHRAE Handbook* (ASHRAE, 2009) can be found in Appendix B.

Table 6.4 presents the DD_H values (18°C) calculated over one year and over the heating season of eight months (the period from October 1 through May 21) for 15 locations representative of various climate zones in France.

Rarely is the balance temperature for an audited building adequately estimated to be equal to 65°F (18°C). Several simplified methods are available to estimate the degree-days for any balance point temperature from limited climatic data. Two of these simplified methods are described in this section.

The first simplified method, proposed by Erbs, Klein, and Beckman (1983), is based on the assumption that the outdoor ambient temperature for each month follows a probability distribution with a

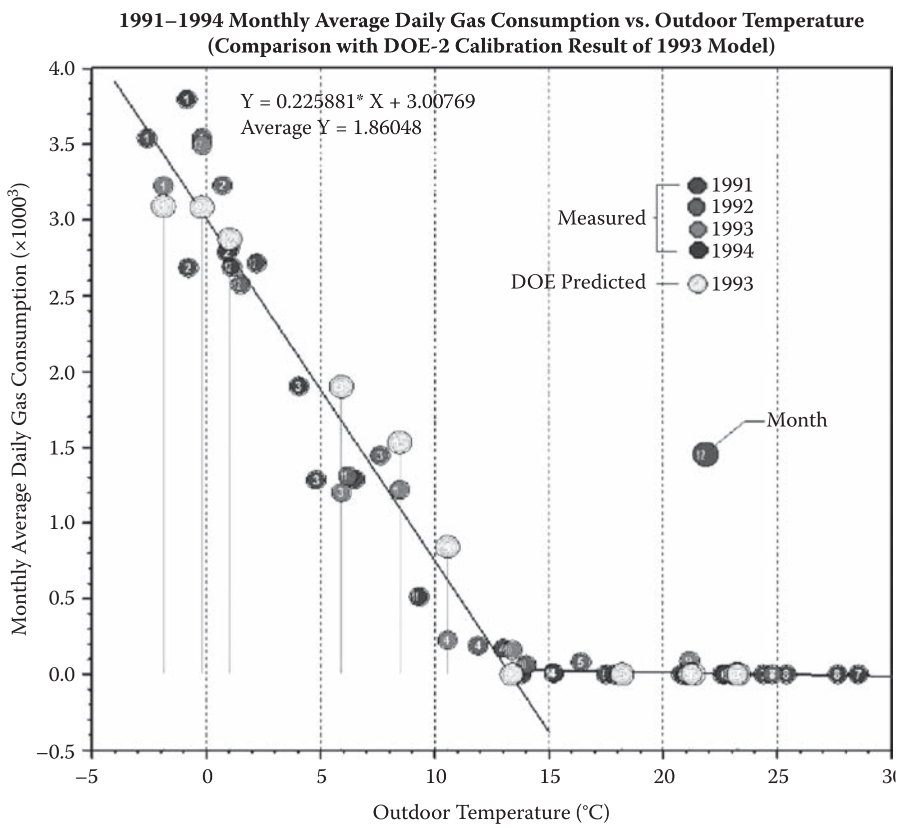


FIGURE 6.7 Determination of the BLC for the heating season based on the gas consumption. (Courtesy of Yoon et al., *Building Energy Audit: Samsung Building*, KIER Report, Korea, 1997).

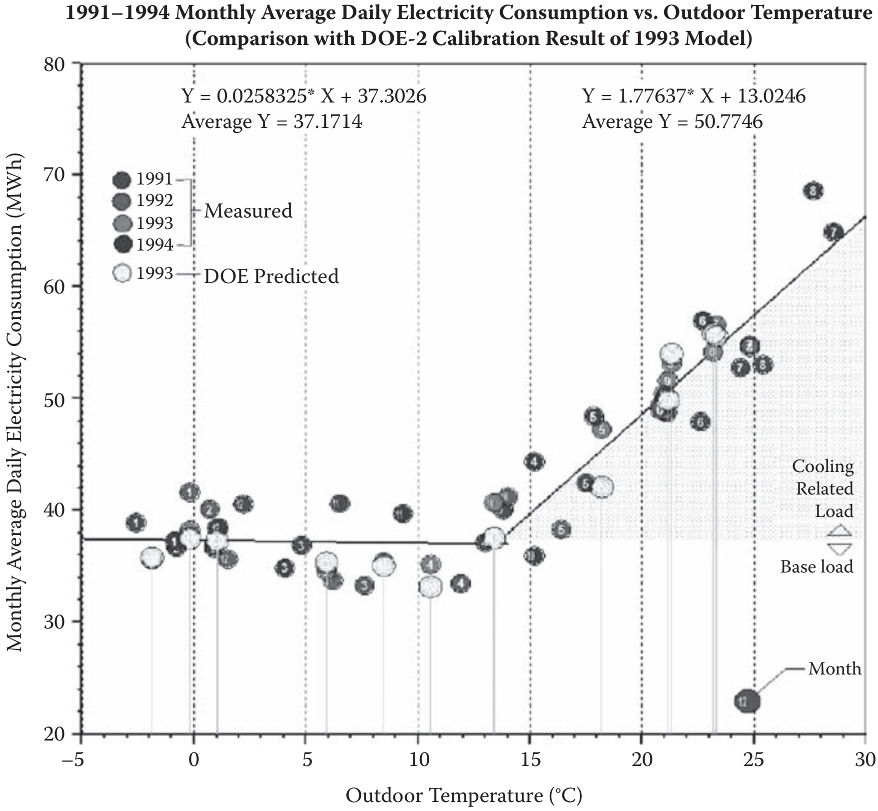


FIGURE 6.8 Determination of the BLC for the cooling season based on the electricity consumption. (Courtesy of Yoon et al., Building Energy Audit: Samsung Building, KIER Report, Korea, 1997.)

standard deviation σ_m, an average temperature $\overline{T}_{o,m}$, and a frequency distribution F function of the balance temperature T_b:

$$F(T_b) = 1/\left(1 + e^{-2.a.\theta_m}\right) \tag{6.23}$$

where θ_m is the normalized average outdoor temperature and is defined as:

$$\theta_m = \frac{T_b - \overline{T}_{o,m}}{\sigma_m.N_m^{1/2}} \tag{6.24}$$

with N_m the number of days in the month considered.

Using the variation of the frequency distribution F the heating degree-days can be obtained as a function of T_b:

$$DD_H(T_b) = \sigma_m.N_m^{3/2}.\left[\frac{\theta}{2} + \frac{Ln(e^{-a\theta} + e^{a\theta})}{2.a}\right] \tag{6.25}$$

For locations spanning most climates in the United States and Canada, Erbs, Klein, and Beckman (1983) found that the coefficients a and σ_m can be estimated using the following expressions:

TABLE 6.3 Heating Degree-Days for Base Temperature 65°F (18°C), 55°F (13°C), and 45°F (7°C) for Selected Locations in the United States

Location	DD_H (65°F)	DD_H (55°F)	DD_H (45°F)
Albuquerque, NM	4,292	2,330	963
Bismarck, ND	9,044	6,425	4,374
Chicago, IL	6,127	3,912	2,219
Dallas/Ft. Worth, TX	2,290	949	250
Denver, CO	6,016	3,601	1,852
Los Angeles, CA	1,245	158	0
Miami, FL	206	8	0
Nashville, TN	3,696	1,964	1,338
New York, NY	4,909	2,806	1,311
Seattle, WA	4,727	2,091	602

TABLE 6.4 Heating Degree-Days for Base Temperature 18°C for Selected Locations in France

Location	DD_H (18°C) over 1 Year	DD_H(18°C) for 8 Months
Embrun	3,087	2,875
Bourg-St. Maurice	3,426	3,135
Besançon	2,995	3,093
St. Quentin	3,085	2,777
Le Bourget (Paris)	2,758	2,549
Lyon	2,656	2,529
Marignane (Marseille)	1,760	1,744
Bordeaux	2,205	2,082
Toulouse	2,205	2,123
Toulon	1,376	1,367
La Rochelle	2,179	2,073
Nantes	2,413	2,244
Deauville	2,961	2,604
Ouessant	2,314	1,954

$$a = 1.698\sqrt{N_m} \tag{6.26}$$

and

$$\sigma_m = 3.54 - 0.029 * \overline{T}_{o,m} + 0.0664 * \sigma_{yr} \tag{6.27}$$

where σ_{yr} is the standard deviation of the monthly temperatures relative to the annual average temperature, $\overline{T}_{o,yr}$:

$$\sigma_{yr} = \sqrt{\frac{\sum_{m=1}^{12} (\overline{T}_{o,m} - \overline{T}_{o,yr})^2}{12}} \tag{6.28}$$

For some other countries such as France, only daily (rather than hourly) average temperature can typically be obtained. Therefore, the values for a and σ_m to be used for these countries are different from those proposed by Erbs, Klein, and Beckman (1983) which are based on hourly average outdoor temperature. For France, Bourges (1987) determined that the following parameters should be used:

- $a = 2.1$.
- A standard deviation based on the maximum and minimum quintals (F_{max} and F_{min}) of the monthly outdoor temperatures corresponding to the frequency values of 80 and 20 percent, respectively:

$$\sigma_m = \frac{F_{max} - F_{min}}{1.683} \tag{6.29}$$

The second simplified method, established by Schoenau and Kehrig (1990), uses the cumulative normal probability function:

$$F(Z) = \int_{-\infty}^{Z} f(z)dz \tag{6.30}$$

where $f(z)$ is the normal probability density function defined as

$$f(z) = \frac{1}{\sqrt{2\pi}} \exp(-\frac{z^2}{2}) \tag{6.31}$$

Both $f(z)$ and $F(Z)$ can be readily computed using built-in functions available in several scientific calculators or spreadsheet programs,

The monthly degree-days based on any balance temperature T_b are then estimated as follows:

$$DD_H(T_b) = \sigma_m.N_m.\left[Z_m F(Z_m) + f(Z_m)\right] \tag{6.32}$$

where Z_m is the normalized outdoor temperature relative to the balance temperature and is defined as

$$Z_m = \frac{T_b - T_{o,m}}{\sigma_m} \tag{6.33}$$

With this simplified method, the monthly cooling degree days can also be determined using a similar expression of Eq. (6.32):

$$DD_C(T_b) = \sigma_m.N_m.\left[Z_m F(Z_m) + f(Z_m)\right] \tag{6.34}$$

with Z_m defined instead as follows:

$$Z_m = \frac{T_{o,m} - T_b}{\sigma_m} \tag{6.35}$$

Annual degree-days can be obtained by simply adding the monthly degree-days over the 12 months of the year. Example 6.2 shows the calculation procedure for estimating the heating and cooling degree-days for one month using the Schoenau and Kehrig method.

Example 6.2

Estimate the heating and cooling degree-days for the base temperature, T_{tb} = 60°F, for Denver, Colorado for the month of May. Use the data provided in Appendix B

Solution

Based on the weather data of Appendix C, available at the CRC Web site www.CRCpress.com/product/isbn/9781439828717 the average temperature and standard deviation in Denver for the month of May are, respectively, $T_{o,m\,=}$ 57.2°F and σ_{yr} = 8.7°F.

For heating degree-days, Eq. (6.33) is used and provides Z_m = 0.321. From normal distribution tables, $f(Z_m)$ = 0.379 and $F(Z_m)$ = 0.626. Then, Eq. (6.32) gives:

$$DD_H\,(\,T_b\,) = (8.70).(31).[(\,0.321)(0.626) + 0.379)] = 156.5°\text{F-days}$$

Similarly, for cooling degree-days, Eq. (6.35) is used and provides Z_m = −0.321. From normal distribution tables, $f(Z_m)$ = 0.379 and $F(Z_m)$ = 0.374. Then, Eq.(6.32) gives:

$$DD_C\,(\,T_b\,) = (8.70).(31).[(\,0.321)(0.374) + 0.379)] = 69.7°\text{F-days}$$

6.3.4 Foundation Heat Transfer Calculations

The practice of insulating building foundations has become more common over the last few decades. However, the vast majority of existing residential buildings are not insulated. It was estimated that in 1985 less than 5 percent of the existing building stock had insulated foundations. Earth-contact heat transfer appears to be responsible for 1 to 3 quadrillion kJ of annual energy use in the United States. This energy use is similar to the impact due to infiltration on annual cooling and heating loads in residential buildings (Claridge, 1988). In addition to the energy-saving potential, insulating building foundations can improve the thermal comfort especially for occupants of buildings with basements or earth-sheltered foundations.

Typically, the foundation heat transfer is a major part of heating/cooling loads for low-rise buildings including single-family dwellings, small commercial and institutional buildings, refrigerated structures, and large warehouses. A detailed discussion of the insulation configurations for various building types as well as various calculation techniques to estimate foundation heat transfer can be found in Krarti (1999). In this section, only a simplified calculation method is provided for annual and seasonal foundation heat loss or gain from residential foundations.

It should be noted that in the United States, there are three common foundation types for residential buildings: slab-on-grade floors, basements, and crawlspaces. The basement foundations can be either deep or shallow. Typically, shallow basements and crawlspaces are unconditioned spaces. Figure 6.9 shows the three common building foundation types. In some applications, the building foundation can include any combination of the three foundation types such as a basement with a slab-on grade floor. Among the factors that affect the selection of the foundation type include the geographical location and the speculative real estate market.

A recent report from the U.S. Census Bureau indicates that the share of houses built with crawlspaces remained constant at about 20 percent over the last seven years (Krarti, 1999). However, the percentage of houses with slab foundations has increased from 38 percent in 1991 to 45 percent in 1997. Meanwhile, the share of houses built with basements has declined from a peak of 42 percent in 1992 to

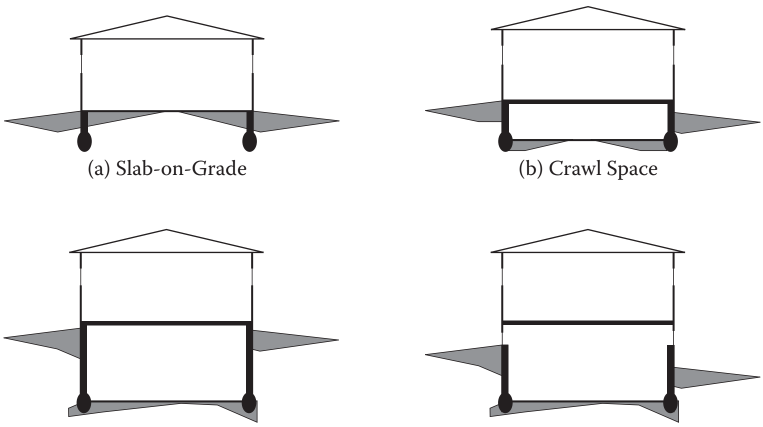


FIGURE 6.9 Foundation types for the buildings.

37 percent in 1997. In 1993, houses were built with almost an equal number of basement and slab foundations. Moreover, data from the U.S. Census Bureau clearly indicates that the foundation type selection depends on the geographical location. In the Northeast and Midwest regions, the basement foundation is the most common with a share of about 80 percent during the period between 1991 and 1997, whereas the slab foundation is more dominant in the South and the West.

6.3.5 Simplified Calculation Method for Building Foundation Heat Loss/Gain

A simplified design tool for calculating heat loss for slabs and basements has been developed by Krarti and Chuangchid (1999). This design tool is easy to use and requires straightforward input parameters with continuously variable values including foundation size, insulation *R*-values, soil thermal properties, and indoor and outdoor temperatures. The simplified method provides a set of equations suitable for estimating the design and the annual total heat loss for both slab and basement foundations as a function of a wide range of variables.

Specifically, the simplified calculation method can calculate the annual or seasonal foundation heat transfer using two equations to estimate, respectively, the mean Q_{m} and the amplitude Q_{a} of the annual foundation heat loss.

For the annual mean foundation heat loss:

$$Q_m = U_{eff,m} \cdot A \cdot (T_a - T_r) \tag{6.36}$$

where

$$U_{eff,m} = m \cdot U_o \cdot D$$

For the annual amplitude foundation heat loss:

$$Q_a = U_{eff,a} \cdot A \cdot T_a \tag{6.37}$$

where

$$U_{eff,a} = a \cdot U_o \cdot D^{0.16} \cdot G^{-0.6}$$

The coefficients a and m depend on the insulation placement configurations and are provided in Table 6.5.

The normalized parameters used by both Eqs. (6.36) and 6.37) are defined below

$$U_o = \frac{k_s}{(A/P)_{eff,b}}; \qquad G = k_s.R_{eq} \cdot \sqrt{\frac{\omega}{\alpha_s}}$$

$$D = \ln\left[(1+H)\left(1+\frac{1}{H}\right)^H\right]; \qquad H = \frac{(A/P)_{eff,b}}{k_s.R_{eq}}$$

For partial insulation configurations (for slab foundations, the partial insulation can be placed either horizontally extending beyond the foundation, or vertically along the foundation walls):

TABLE 6.5 Coefficients m and a to Be Used in Equations (6.36) and (6.37) for Foundation Heat Gain Calculations

Insulation Placement	m	A
Uniform–Horizontal	0.40	0.25
Partial–Horizontal	0.34	0.20
Partial–Vertical	0.28	0.13

TABLE 6.6 Comparison of the Results Between the Simplified and the ITPE Solution[a]

Method	Mean (Q_m)	Amplitude (Q_a)
Simplified	699	208
ITPE solution[b]	658	212

[a] The heat loss per unit area are provided in W/m^2.

[b] For more details about the ITPE (interzone temperature profile estimation) solution technique for foundation heat transfer problems, the reader is referred to Krarti, M., Foundation Heat Transfer, Chapter in *Advances in Solar Energy*, Edited by Y. Goswami and K. Boer, ASES, Boulder, CO, 1999.

TABLE 6.7 Comparison of the Simplified Design Tool and the ITPE Solution Predictions[a]

Method	No Insulation	Uniform Insulation
Simplified	16.91	5.85
ITPE solution[b]	17.28	5.69

Source: Courtesy of Krarti, M., Foundation Heat Transfer, Chapter in *Advances in Solar Energy*, Edited by Y. Goswami and K. Boer, ASES, Boulder, CO, 1999.

[a] The heat gain per unit area is provided in W/m^2.

$$R_{eq} = R_f \times \frac{1}{\left[1-\left(\frac{c}{A/P}\times\frac{R_i}{(R_i+R_f)}\right)\right]}$$

For uniform insulation configurations: $R_{eq} = R_f + R_i$, where

$$(A/P)_{eff,b,mean} = \left[1 + b_{eff} \times \left(-0.4 + e^{-H_b}\right)\right] \times (A/P)_b$$

$$(A/P)_{eff,b,amp} = \left[1 + b_{eff} \times e^{-H_b}\right] \times (A/P)_b$$

$$H_b = \frac{(A/P)_b}{k_s . R_{eq}} \text{ and } b_{eff} = \frac{B}{(A/P)_b}$$

In the equations above, the following parameters are used:

A	Basement/slab area (total of floor and wall) [m^2 or ft^2]
B	Basement depth [m or ft]
b_{eff}	Term defined in Eqs. (6.30) and (6.31)
C_p	Soil specific heat [J/kg°C or Btu/lbm. °F]
c	Insulation length of basement/slab [m or ft]
D	Term defined in Eqs. (6.30) and (6.31)
G	Term defined in Eq. (6.31)
H	Term defined in Eqs. (6.30) and (6.31)
k_s	Soil thermal conductivity [Wm^{-1}.°C^{-1} or Btu/hr.ft.°F]
P	Perimeter of basement/slab [m or ft]
Q	Total heat loss [W or Btu/hr]
Q_m	The annual mean of the total heat loss [W or Btu/hr]
Q_a	The annual amplitude of the total heat loss [W or Btu/hr]
R_{eq}	Equivalent thermal resistance *R*-value of entire foundation [m^{2}K/W or ft^2.°F.hr/Btu]
R_f	Thermal resistance *R*-value of floor [m^{2}K/W or ft^2.°F.hr/Btu]
R_i	Thermal resistance *R*-value of insulation [m^{2}K/W or ft^2.°F.hr/Btu]
T_a	Ambient or outdoor air temperature [°C or °F]
T_r	Room or indoor air temperature [°C or °F]
$U_{eff,m}$	Effective *U*-value for the annual mean [Wm^{-2}.°C^{-1} or Btu/hr.ft^2.°F] defined in Eq (6.36)
$U_{eff,a}$	Effective *U*-value for the annual amplitude [Wm^{-2}.°C^{-1} or Btu/hr.ft^2.°F] defined in Eq (6.37)
U_o	*U*-value [Wm^{-2}.°C^{-1} or Btu/hr.ft^2.°F] defined in Eqs, (6.36) and (6.37)
ρ	Soil density [kg/m^3 or lbm/ ft^3]
ω	Annual angular frequency [rad/s or rad/hr]
α_s	Thermal diffusivity [m^2/s or ft^2/hr]

It should be noted that the simplified model provides accurate predictions when A/P is larger than 0.5 meter. The annual average heat flux (heat loss or gain) from the building foundation is simply Q_m. The highest foundation heat flux under design conditions can be obtained as $Q_{de\ s} = Q_m + Q_a$.

To illustrate the use of the simplified models, two calculation examples are presented for a basement structure insulated with uniform insulation.

6.3.5.1 Calculation Example No. 1: Basement for a Residential Building

Determine the annual mean and annual amplitude of total basement heat loss for a house. The basic geometry and construction details of the basement are provided below (see data provided in Step 1). The house is located in Denver, Colorado.

Solution

Step 1. Provide the required input data (from ASHRAE, 2009):

Dimensions

Basement width = 10.0 m (32.81 ft)
Basement length = 15.0 m (49.22 ft)
Basement wall height = 1.5 m (4.92 ft)
Basement total area = 225.0 m^2 (2422.0 ft^2)
Ratio of basement area to basement perimeter: $(A/P)_b$ = 3.629 m (11.91 ft)
Four inch thick reinforced concrete basement, thermal resistance R-value:
= 0.5 m^{2}K/W (2.84 h.ft^{2}F/Btu)

Soil Thermal Properties

Soil thermal conductivity: k_s = 1.21 W/m.K (0.70 Btu/h.ft.F)
Soil thermal diffusivity: $\alpha_s = 4.47 \times 10^{-7}$ m^2/s (48.12×10^{-7} ft^2/s)

Insulation

Uniform insulation R-value = 1.152 m^{2}K/W (6.54 h.ft^{2}F/Btu)

Temperatures

Indoor temperature: T_r = 22 °C (71.6 °F)
Annual average ambient temperature: T_a = 10 °C (50 °F)
Annual amplitude ambient temperature: T_{amp} = 12.7 °K (23 °R)
Annual angular frequency: $\omega = 1.992 \times 10^{-7}$ rad/s

Step 2. Calculate Q_m, and Q_a values:

Using Equations (6.36) and (6.37), the various normalized parameters are first calculated. Then the annual mean and amplitude of the basement heat loss are determined.

$$H_b = \frac{(A/P)_b}{k_s.R_{eq}} = \frac{3.629}{1.21 \times (0.5 + 1.152)} = 1.8155$$

$$b_{eff} = \frac{B}{(A/P)_b} = \frac{1.5}{3.629} = 0.4133$$

$$(A/P)_{eff,b,mean} = \left[1 + 0.4133 \times \left(-0.4 + e^{-1.8155}\right)\right] \times 3.629 = 3.2731$$

$$(A/P)_{eff,b,amp} = \left[1 + 0.4133 \times e^{-1.8155}\right] \times 3.629 = 3.8731$$

$$U_{o,m} = \frac{k_s}{(A/P)_{eff,b,mean}} = \frac{1.21}{3.2731} = 0.3697$$

$$U_{o,a} = \frac{k_s}{(A/P)_{eff,b,mean}} = \frac{1.21}{3.8731} = 0.3124$$

$$H_{mean} = \frac{(A/P)_{eff,b,mean}}{k_s.R_{eq}} = \frac{3.2731}{1.21 \times (0.5 + 1.152)} = 1.6374$$

$$H_{amp} = \frac{(A/P)_{eff,b,amp}}{k_s.R_{eq}} = \frac{3.8731}{1.21 \times (0.5 + 1.152)} = 1.9376$$

$$D_{mean} = \ln\left[(1+H)\left(1+\frac{1}{H}\right)^H\right] = 1.7503$$

$$D_{amp} = \ln\left[(1+H)\left(1+\frac{1}{H}\right)^H\right] = 1.8839$$

$$G = k_s.R_{eq} \cdot \sqrt{\frac{\omega}{\alpha_s}} = 1.21 \times (0.5 + 1.152) \times \sqrt{\frac{1.992 \times 10^{-7}}{4.47 \times 10^{-7}}} = 1.3344$$

Therefore

$$Q_m = U_{eff,m} A(T_a - T_r) = 0.4 \times 0.3697 \times 1.7503 \times 225 \times (22.0 - 10.0) = 698.85 \text{ W } (2384.48 \text{ Btu/h})$$

and

$$Q_a = U_{eff,a} A T_a = 0.25 \times 0.3124 \times 1.8839^{0.16} \times 1.3344^{-0.6} \times 225 \times 12.7 = 207.72 \text{ W } (708.74 \text{ Btu/h})$$

6.3.5.2 Calculation Example No. 2: Freezer Slab

For a freezer warehouse depicted in Sketch 6.1 determine the total freezer heat gain under design conditions. The warehouse is located in Denver, Colorado. Estimate the cost-effectiveness of uniformly insulating the freezer foundation slab using four inches of extruded polystyrene insulation.

Step 1. Provide the required input data:

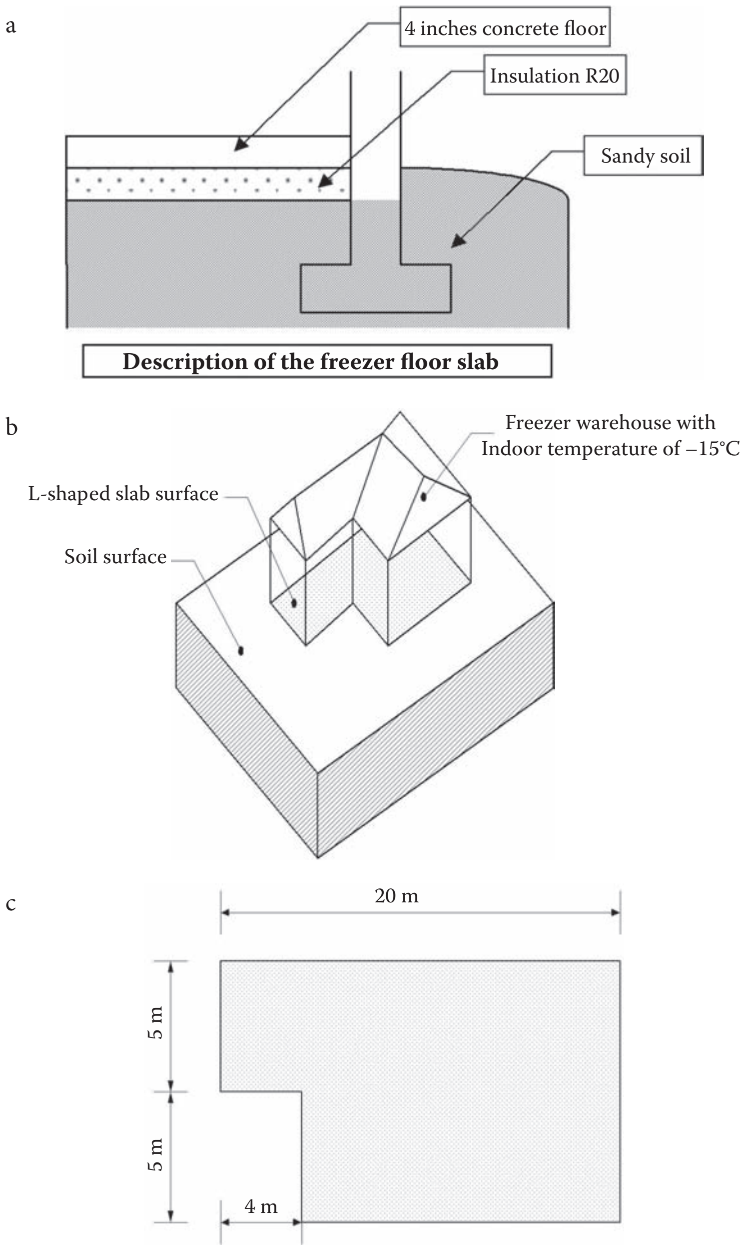
Dimensions

Slab width = 10.0 m (32.81 ft)
Slab length = 20.0 m (49.22 ft)
Ratio of slab area to slab perimeter: A/P = 3.0 m (9.84 ft)
100 mm (4 inches) lightweight concrete slab with thermal resistance
R-value = 0.587 m^2K/W (3.33 $h.ft^2F/Btu$)*

Soil Thermal Properties

Type of soil : Sandy soils
Soil thermal conductivity: k_s = 1.51 W/m.K (0.87 Btu/h.ft.F)†

* Source: ASHRAE (2009, Table 4, p. 26.8).
† Source: ASHRAE (2009, Table 5, p. 26.13).



SKETCH 6.1

Soil density: ρ_s = 2740 kg/m3 (171.06 lb_m/ft^3)*
Soil heat capacity: c_s = 774.0 J/kg.C (0.18 $Btu/lb_m.F$)
Soil thermal difussivity: $\alpha_s = k_s/\rho_s c_s = 7.12 \times 10^{-7}$ m^2/s (76.6×10^{-7} ft^2/s)

Insulation

100 mm (4 inches) extruded polystyrene: *R*-value = 3.52 m^2K/W (20.0 $h.ft^2F/Btu$)*

* Source: Kersten, M.S. *Thermal Properties in Soils*, Bulletin from University of Minnesota, Institute of Technology and Engineering, Experimental Station Bulletin, No. 28, 1949.

Temperatures

Indoor temperature: T_r = –15°C (5°F) (for freezer storage)
Annual average ambient temperature: T_m = 6.3°C (43°F)*
Annual amplitude ambient temperature: T_a = 30°C (54°F)*
Annual angular frequency: ω = 1.992 × 10^{-7} rad/s

Cases of insulation configurations

Case 1: No insulation
Case 2: Uniform insulation

Step 2. Calculate Q_{des}:

Using Eqs. (6.36) and (6.37), the various normalized parameters are first calculated. Then the design heat gain for freezer slab is determined.
Case 1: No insulation

$$R_{eq} = 0.587 + 0.0 = 0.587 \text{ m}^2\text{K/W}$$

$$U_o = \frac{1.51}{3.0} = 0.5033$$

$$H = \frac{3.0}{1.51 \times 0.587} = 3.3846$$

$$D = \ln\left[(4.3846)(1.2954)^{3.3846}\right] = 2.3541$$

$$G = (1.51)(0.587) \times \sqrt{\frac{1.992 \times 10^{-7}}{7.12 \times 10^{-7}}} = 0.4688$$

$$U_{eff,m} = 0.4 \times 0.5033 \times 2.3541 = 0.4739$$

$$U_{eff,a} = 0.25 \times 0.5033 \times (2.3541)^{0.16} (0.4688)^{-0.6} = 0.2273$$

Therefore, the design total heat gain for freezer slab with no insulation is:

$$\frac{Q_{des}}{A} = (0.4739)(6.3 - (-15.0)) + (0.2273)(30.0) = 16.91 \text{ W/m}^2 \text{ (5.36 Btu/h.ft}^2\text{)}$$

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* Source: ASHRAE (*Proposed ASHRAE Guideline 14P, Measurement of Energy and Demand Savings*, Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1997. Table 1A, p.26.8).

Case 2: Uniform insulation

$$R_{eq} = 0.587 + 3.52 = 4.107 \text{ m}^2\text{K/W}$$

$$U_o = \frac{1.51}{3.0} = 0.5033$$

$$H = \frac{3.0}{1.51 \times 4.107} = 0.4837$$

$$D = \ln\left[(1.4837)(3.0674)^{0.4837}\right] = 0.9367$$

$$G = (1.51)(4.107) \times \sqrt{\frac{1.992 \times 10^{-7}}{7.12 \times 10^{-7}}} = 3.2802$$

$$U_{eff,m} = 0.4 \times 0.5033 \times 0.9367 = 0.1886$$

$$U_{eff,a} = 0.25 \times 0.5033 \times (0.9367)^{0.16}(3.2802)^{-0.6} = 0.0610$$

Therefore, the design total heat gain for freezer slab with uniform insulation is:

$$\frac{Q_{des}}{A} = (0.1886)(6.3 - (-15.0)) + (0.0610)(30.0) = 5.85 \text{ W/m}^2 \text{ (1.85 Btu/h.ft}^2\text{)}$$

Step 3. Perform an economic analysis:

Assume that warehouse is operated 24 hours a day for whole year.

kW/ton for freezer is: 2.35 kW/ton*

Electricity cost in Denver is: $0.08/kWh

Cost of extruded polystyrene is: $3.00/m².in. (material cost) and $2.50/m² (labor cost; cost data are specific to Denver, Colorado).

To evaluate the annual performance of the insulation, only annual mean heat gains are considered (for both cases: with and without four inches of insulation along the foundation slab) in the economical analysis outlined below.

First, the savings on the annual average heat gain (expressed in watts) are estimated:

$$\begin{aligned}\text{Heat Gain Saving} &= \text{(No insulation heat gain} - \text{Uniform insulation heat gain)} * \text{Slab Ar} \\ &= (16.91 - 5.85) \times 200 = 2{,}212 \text{ Watt}\end{aligned}$$

Thus, the annual total heat gain savings expressed in kWh/yr are calculated as follows:

$$\text{Annual Heat Gain Saving} = \frac{Watt \times 24 \times 7 \times 52}{1000} = 19{,}324 \text{ kWh/yr}$$

The electrical energy savings for the refrigeration equipment are then determined to be:

$$\text{Electrical Energy Saving } = \frac{kWh \times 2.35\text{kW/ton}}{3.517 \text{ kW/ton.refrig}} = 12{,}912 \text{ kWh/yr}$$

Therefore, the annual cost savings attributed to the addition of the foundation insulation are:

$$\text{Annual Cost Saving } = \text{kWh/yr} \times \$0.08 / \text{kWh} = \$1{,}033/\text{yr}$$

The cost of the foundation insulation is estimated to be:

$$\text{Investment Cost } = (4 \text{ inches } \times \ \$3.00/\text{m}^2\text{in } + \ \$2.50/\text{m}^2) \times 180 = \$2{,}900$$

Therefore, the payback period for the addition of the foundation insulation is:

$$\text{Payback Period } = \frac{\$2{,}900}{\$1{,}033 / \text{yr}} = 2.8 \text{ years}$$

To prevent heaving problems, the freezer foundation has to be insulated in any case, unless a floor heating system is installed under the slab foundation.

6.4 Selected Retrofits for Building Envelope

Generally, improvements in the energy efficiency of the building envelope are expensive because labor-intensive modifications are typically involved (such as addition of thermal insulation and replacement of windows). As a consequence, the payback periods of most building envelope retrofits are rather long. In these instances, the building envelope retrofits can still be justified for reasons other than energy efficiency such as increase in occupant thermal comfort or reduction of moisture condensation to avoid structural damage. However, there are cases where retrofits of building envelope can be justified based solely on improvement in energy efficiency. Some of these retrofit measures are discussed in this section with some examples to illustrate how the energy savings and the payback periods are calculated.

6.4.1 Insulation of Poorly Insulated Building Envelope Components

When an element of a building envelope is not insulated or poorly insulated, it may be cost-effective to add insulation in order to reduce transmission losses. Although the calculation of the energy savings due to such a retrofit may require a detailed simulation tool to account for effects of the building thermal mass or the building HVAC systems, Eq. (6.21) or (6.22) can be used to determine the energy savings during the heating season. If the building is heated and cooled, the total energy savings due to adding insulation to the building envelope can be estimated by summing the energy savings obtained from a reduction in heating loads and those obtained from a decrease (or increase) in cooling loads as outlined in Example 6.3.

In most cases where the building envelope is already adequately insulated, the addition of thermal insulation is not cost-effective based only on energy cost savings.

Example 6.3

A machine shop has a 500 m^2 metal frame roof that is uninsulated. Determine the payback period of adding insulation (R = 2.0°C.m^2/W). The building is electrically heated. The cost of electricity is \$0.07/kWh. The machine shop is located in Paris (Le Bourget, France) and operates 24 hours/day, 7 days/week throughout the heating season. Assume that the installed cost of the insulation is \$15/m^2.

Solution

Based on the *ASHRAE Handbook*, the existing *U*-value for a metal frame roof is about 1.44 W/m^2×°C. To determine the energy savings due to the addition of insulation, we assume that the annual heating degree-days before and after the retrofit remained unchanged and are close to 18°C. Using Eq. (6.21) with the retrofitted roof *U*-value to be 0.37 W/m^2×°C and heating system efficiency set to be unity (electrical system), the energy savings are calculated to be:

$$\Delta E = 24.500 m^2 * [(1.44 - 0.37) W/m^2 {}^\circ C] * 2758^\circ C.day/yr = 35{,}413 kWh/yr$$

Thus, the payback period for adding insulation on the roof can be estimated to be:

$$Payback = \frac{500 m^2 * 15\$/m^2}{35{,}413 kWh/yr * 0.07\$/kWh} = 3.0\, years$$

Therefore, the addition of insulation seems to be cost-effective. Further analysis is warranted to determine the cost-effectiveness of this measure more precisely.

6.4.2 Window Improvements

Window improvements such as installation of high-performance windows, window films and coatings, or storm windows can save energy through reductions in the building heating and cooling thermal loads. Improvements in windows can affect both the thermal transmission and solar heat gains. In addition, energy-efficient windows create more comfortable environments with evenly distributed temperatures and quality lighting. Energy-efficient improvements can be made to all the components of a window assembly including:

- Insulating the spacers between glass panes can reduce conduction heat transfer.
- Installing multiple coating or film layers can reduce heat transfer by radiation.
- Inserting argon or krypton gas in the space between the panes can decrease the convection heat transfer.
- Providing exterior shading devices can reduce the solar radiation transmission to the occupied space.

To determine accurately the annual energy performance of window retrofits, dynamic hourly modeling techniques are generally needed because fenestration can have an impact on the building thermal loads through several mechanisms. However, the simplified calculation method based on Eq. (6.21) to account for both heating and cooling savings can be used to provide a preliminary assessment of the cost-effectiveness of window retrofits as outlined in Example 6.4.

Example 6.4

A window upgrade is considered for an apartment building from double-pane metal frame windows (U_E = 4.61 W/m^2.°C) to double-pane with low-e film and wood frame windows (U_R = 2.02 W/m^2.°C). The total window area to be retrofitted is 200 m^2. The building is located in Nantes (France) and is conditioned 24 hours/day, 7 days/week throughout the heating season. An electric baseboard provides heating and a window AC provides cooling (EER = 8.0). Assume that the cost of electricity is $0.10/kWh.

Solution

To determine the energy savings due to the addition of insulation, we assume that the annual heating and cooling degree-days before and after the retrofit remained unchanged (this assumption is justified by the fact that the window contribution to the BLC is relatively small) and are, respectively, DD_H = 2,244°C-day/yr and DDC = 255°C-day/yr. The energy savings during heating (assume that the system efficiency is 1.0 for electric heating) are calculated to be:

$$\Delta E = 24.200 m^2 * [(4.61 - 2.02) W/m^{2\circ}\ C] * 2244^\circ\ C.day/yr = 27{,}897 kWh/yr$$

If an EER (energy efficiency ratio) value of 8.0 is assumed for the AC system, the energy savings during cooling are estimated as follows:

$$\Delta E = 24.200 m^2 * [(4.61 - 2.02) W/m^2] * 255^\circ\ C.day/yr * 1/8.0 = 396 kWh/yr$$

Therefore the total energy savings due to upgrading the windows is 28,293 kWh which corresponds to about $2,829 when the electricity cost is $.10/kWh. The cost of replacing the windows is rather high (it is estimated to be $150/m^2 for this project). The payback period of the window retrofit can be estimated to be:

$$Payback = \frac{200 m^2 * 150\$/m^2}{28{,}293 kWh/yr * 0.10\$/kWh} = 10.4\, years$$

Therefore, the window upgrade is not cost-effective based solely on thermal performance. The investment on new windows may, however, be justifiable based on other factors such as increased comfort within the space.

6.4.3 Reduction of Air Infiltration

In several low-rise facilities, the thermal loads due to air infiltration can be significant. It is estimated that for a well-insulated residential building, the infiltration can contribute up to 40 percent of the total building heating load. Tuluca et al. (1997) reported that measurements in eight U.S. office buildings found average air leakage rates of 0.1 to 0.5 air changes per hour (ACH). This air infiltration accounted for an estimated 10 to 25 percent of the peak heating load. Sherman and Matson (1993) have shown that the housing stock in the United States is significantly overventilated from air infiltration and that there are 2 exajoules of potential annual savings that could be captured. As described in Section 6.2.2, two measurement techniques can be used to evaluate the existing amount of infiltrating air. The blower technique is relatively cheap and quick to set up and can be useful for small buildings to locate air leaks, whereas the tracer gas technique is more expensive and time-consuming and is appropriate to measure the outdoor air flow rate from both ventilation and infiltration entering large commercial or institutional buildings.

Several studies exist to evaluate the leakage distribution for residential buildings (Diekerhoff, Grimsrud, and Lipschutz, 1982; Harrje and Born, 1982), although very little work is available for U.S. commercial and industrial buildings. However, some results indicate that the envelope air tightness levels for commercial buildings are similar to those in typical U.S. houses. In particular, it was found that leaks in walls (frames of windows, electrical outlets, plumbing penetrations) constitute the major sources of air leakage of both residential and commercial buildings. For instance, for office buildings, Tumura and Shaw (1976) found that typical air leakage values per unit wall area at 75 Pa (pressure differential between indoors and outdoors) are 500, 1,500, and 300 $cm^3/(s.m^2)$ for, respectively, tight, average, and leaky walls. Other sources of air leakage identified for large commercial buildings are through internal partitions (such as elevator and service shafts), and exterior doors (especially for retail stores).

To improve the air tightness of the building envelope several methods and techniques are available including:

1. *Caulking:* Several types of caulking (urethane, latex, and polyvinyl) can be applied to seal various leaks such as those around the window and door frames, and any wall penetrations such as holes for water pipes.
2. *Weatherstripping:* By applying foam rubber with adhesive backing, windows and doors can be air sealed.
3. *Landscaping:* This is a rather long-term project and consists of planting shrubs or trees around the building to reduce wind effects and air infiltration.
4. *Air Retarders:* These systems consist of one or more air-impermeable components that can be applied around the building exterior shell to form a continuous wrap around the building walls. There are several air retarder (AR) types such as liquid-applied bituminous, liquid-applied rubber, sheet bituminous, and sheet plastic. The AR membranes can be applied to impede the vapor movement through the building envelope and thus act as vapor retarders. Unless they are part of an overall building envelope retrofit, these systems are typically expensive to install for existing buildings.

To assess the energy savings due to a reduction in air infiltration, Eq. (6.21) or (6.22) can be used as illustrated by Example 6.5. Whenever available, the degree-days IDD_H (T_b) determined specifically to calculate infiltration loads can be used instead of the conventional temperature-based degree-days DD_H (T_b). The infiltration heating degree-days for the balance temperature is defined as follows:

$$IDD_H(T_b) = \sum_{i=1}^{N_H} \frac{\dot{V}}{\dot{V}_{ref}} (T_b - T_{o,i})^+ \tag{6.38}$$

where $\dot{V}$ is calculated as shown in Eq. (6.9) to account for the climatic data and $\dot{V}_{ref}$ is the reference volume rate defined for Eq. (6.8). However, the conventional variable-base degree-days method (which basically ignores the effects of weather on the variation of the infiltration rate) provides a generally good estimation of the energy savings incurred from a reduction in air infiltration.

Example 6.5

Consider a heated manufacturing shop with a total conditioned volume of 1,000 m^3. A measurement of the air leakage characteristics of the shop showed an infiltration rate of 1.5 ACH. Determine the energy savings due to caulking and weatherstripping improvements of the exterior envelope of the facility to reduce air infiltration by half. Assume the shop is located in Seattle, Washington and is heated by a gas-fired boiler with a seasonal efficiency of 80 percent.

Solution

To determine the energy savings due to the addition of insulation, we assume that the annual heating degree-days before and after the retrofit remain unchanged and are close to 18°C. For Seattle, the degree-days (with base 65°F or 18°C) are about 2,656°C-days.

The existing air infiltration has an equivalent *UA*-value of $UA_{inf} = mc_{p,a} = 500$ W/°C. From Eq. (6.21) with the new air infiltration the equivalent *UA*-value 250 W/°C and the heating system efficiency set to be 80 percent (gas-fired boiler), the energy savings are calculated to be:

$$\Delta E = \frac{24}{0.80}.[(500 - 250)W/^\circ C] * 2656^\circ C.day/yr = 19{,}920kWh/yr$$

The cost of caulking and weatherstripping is estimated to be about $1,500 (if only material costs are included). For a gas price of $0.05/kWh, the payback period for reducing the infiltration rate can be estimated to be:

$$Payback = \frac{\$1{,}500}{19{,}920kWh/yr * 0.05\$/kWh} = 1.5\,years$$

Therefore, the caulking and weatherstripping can be justified based only on energy cost savings. Additional benefits of reducing infiltration are improved thermal comfort.

6.5 Summary

Energy efficiency improvements of building envelope systems are generally expensive and are not cost-effective especially for large commercial buildings. However, increasing the energy performance of a building shell can be justified for low-rise and small buildings based on energy cost savings but also based on improvement in indoor thermal comfort and integrity of the building structure. For residential buildings, weatherstripping to reduce infiltration losses is almost always economically justifiable.

Problems

6.1 For the case of a balance temperature of 65°F, compare your results to the heating degree-days for the three cities directly computed from hourly weather data (these degree-days are widely reported in the existing literature. Appendix B provides the degree-days for selected U.S. locations). Comment on the results of your comparative analysis.

Month	Denver	New York City	Los Angeles
Jan	29.9	32.2	54.5
Feb	32.8	33.4	55.6
Mar	37.0	41.1	56.5
Apr	47.5	52.1	58.8
May	57.0	62.3	61.9
Jun	66.0	71.6	64.5
Jul	73.0	76.6	68.5
Aug	71.6	74.9	69.6
Sep	62.8	68.4	68.7
Oct	52.0	58.7	65.2
Nov	39.4	47.4	60.5
Dec	32.6	35.5	56.9

6.2 Repeat Problem 6.1 using the simplified calculation method of Schoenau and Kehrig. Use the required statistical data for the various U.S. locations provided in the appendix. In addition to the heating degree-days, find the cooling degree-days based on 60°F, 65°F, and 70°F.

6.3 Determine the heating base-load, the balance temperature, and the building load coefficient (BLC) of a house located in Boulder, Colorado based on the monthly utility bills provided in the table below. For the analysis assume that the house is heated 24 hours/day and that the seasonal efficiency of the gas furnace is about 78 percent. It should be noted that the house is generally unoccupied during the period of May 25 to July 24 (for both 2004 and 2003).

Provide any pertinent comments on the thermal performance of this house.

Date of Bill	Billed Days	Therms Used	Outdoor Temp. (°F)
Dec. 20, 2004	34	230	33
Nov. 21, 2004	30	71	51
Oct. 17, 2004	29	50	63
Sep. 19, 2004	31	21	72
Aug. 20, 2004	28	18	77
Jul. 24, 2004	35	6	79
Jun. 20, 2004	30	10	65
May 22, 2004	30	61	56
Apr. 18, 2004	30	118	45
Mar. 22, 2004	30	163	38
Feb. 16, 2004	27	186	26
Jan. 23, 2004	35	230	32
Jan. 02, 2004	30	199	32
Nov. 17, 2003	29	132	41
Sep. 19, 2003	29	16	73
Sep. 5, 2003	31	4	78
Jun. 20, 2003	30	9	67
May 24, 2003	31	53	56
Apr. 25, 2003	30	87	49
Mar. 22, 2003	27	107	41
Feb. 24, 2003	32	159	36

6.4 An energy audit of the roof indicates the following:

Roof area:	10,000 sq.ft.
Existing roof *R*-value:	5
Degree-days (winter):	6,000
Degrees-hours (summer):	17,000
Fuel cost:	$4/MMbtu
Boiler efficiency:	0.70
Electric rate:	$0.07/kWh
Air-condition requirement:	0.7 kW/ton

It is proposed to add R-19 insulation in the roof.

(a) Comment on the potential energy savings of adding insulation.

(b) If the insulation costs $0.35 per sq.ft, determine the payback of adding insulation.

(c) Is the expenditure on the additional insulation justified based on a minimum rate of return of 12 percent? Assume a 30-year life cycle. Solve the problem using both the present worth and annual cost methods.

6.5 Determine the annual energy savings due to replacing a single-pane window (R-1) by a double-pane window (R-2). The total area of the glazing is 200 sq.ft. Assume the following:

Heating degree-days:	6,000
Cooling degree-days:	500
Fuel cost:	$8/MMBtu
Boiler efficiency:	0.80
Electric rate:	$0.12/kWh
Refrigeration requirement:	0.70 kW/ton
Insulation cost:	$0.225 per sq.ft

6.6 Do you recommend adding R-20 insulation to a 25,000 sq.ft. uninsulated roof (R-5) when you consider a 25-year life cycle with a rate of return of 8 percent? Assume the following:

Heating degree-days:	5,000
Cooling degree-hours:	10,000
Fuel cost:	$5/MMBtu
Boiler efficiency:	0.70
Electric rate:	$0.10/kWh
Refrigeration requirement:	0.75kW/ton
Insulation cost:	$0.225 per sq.ft

6.7 A blower door test on a 30 ft × 50 ft × 9 ft house located in Denver, Colorado, revealed that under a 4 Pa pressure differential between indoors and outdoors, the leakage area is about 200 in.^{2}.

(a) Determine in air changes per hour, the annual average infiltration rate for the house.
(b) It was decided to weather-strip the house so that the infiltration is reduced to just 0.25 ACH (for a 4 Pa pressure differential). Determine the payback period of weatherstripping the house given the following parameters:

Heating degree-days:	6,000
Cooling degrees-hours:	5,000
Fuel cost:	$5/MMBtu
Boiler efficiency:	0.70
Electric rate:	$0.10/kWh
Refrigeration requirement:	0.75kW/ton
Cost of weatherstripping:	$150

6.8 Both pressurization and depressurization tests were conducted on a residence. The residence has 950 ft^2 (95 m^2) of floor area with an average ceiling height of 8 ft. The results of these tests are shown in the table below:

(a) Determine the leakage area of the house.
(b) Determine the effect of shielding level on the infiltration rate. Assume the house is located in Denver, Colorado.

Pressure (Pa)	Pressurization Test (cfm)	Depressurization Test (cfm)
10	177	237
20	324	384
30	411	471
40	531	561
50	590	650
60	676	710
70	735	765
80	795	825
90	885	887
100	942	942

6.9 For a building with a 200 ft by 100 ft slab on grade floor made up of 4-in concrete floor,

(a) Estimate the annual energy ground-coupled loss/gain from the building located in Denver, Colorado.

(b) Estimate the annual heating energy use savings if the slab is insulated uniformly with R-10 rigid insulation. Assume a furnace efficiency of 85 percent.

(c) Implement the calculation procedure for slab-on-grade floor heat loss in a spreadsheet and determine if there is an optimal cost-effective insulation level (i.e., R-value) for the foundation. Assume that the fuel cost is \$10.0/MMBtu, the efficiency of the heating system is 0.80, the cost of adding insulation is 0.5 per ft^2 of R-5 rigid insulation, and the discount rate is 5 percent.

7

Secondary HVAC Systems Retrofit

7.1 Introduction

The heating, ventilating, and air-conditioning (HVAC) system maintains and controls temperature and humidity levels to provide an adequate indoor environment for human activity or for processing goods. The cost of operating an HVAC system can be significant in commercial buildings and in some industrial facilities. In the United States, it is estimated that the energy used to operate the HVAC systems can represent about 30 percent of the total electrical energy use in a typical commercial building (EIA, 2006). It is therefore important that the auditor recognize some of the characteristics of the HVAC systems and determine if any retrofits can be recommended to improve the system's energy.

7.2 Types of Secondary HVAC Systems

A basic HVAC air distribution system consists of an air-handling unit (AHU; with the following components as shown in Figure 7.1:

- Dampers to control the amount of air to be distributed by the HVAC system including: outside air (OA) damper, return air (RA) damper, exhaust air (EA) damper, and supply air (SA) damper
- Preheat coil in case the outside air is too cold to avoid any freezing problems
- Filter to clear the air of any dirt
- Cooling coils to condition the supply air to meet the cooling load of the conditioned spaces
- Humidifiers to add moisture to the supply air in the case where a humidity control is provided to the conditioned spaces
- A distribution system (i.e., ducts) where the air is channeled to various locations and spaces

Each of the above-listed components can come in several types and styles. The integration of all the components constitutes the secondary HVAC system for the sole purpose of conditioned air distribution. Two main categories of secondary HVAC central air systems can be distinguished:

1. *Constant-Air-Volume (CAV) Systems:* These systems provide a constant amount of supply air conditioned at the proper temperature to meet the thermal loads in each space based on a thermostat setting. Typically, the supply air temperature is controlled by either mixing cooled air with heated or bypassed air or by directly reheating cooled air. Therefore, these systems waste energy because of the mixing or reheating especially under partial thermal load conditions. Among the constant air volume systems commonly used to condition existing buildings are:

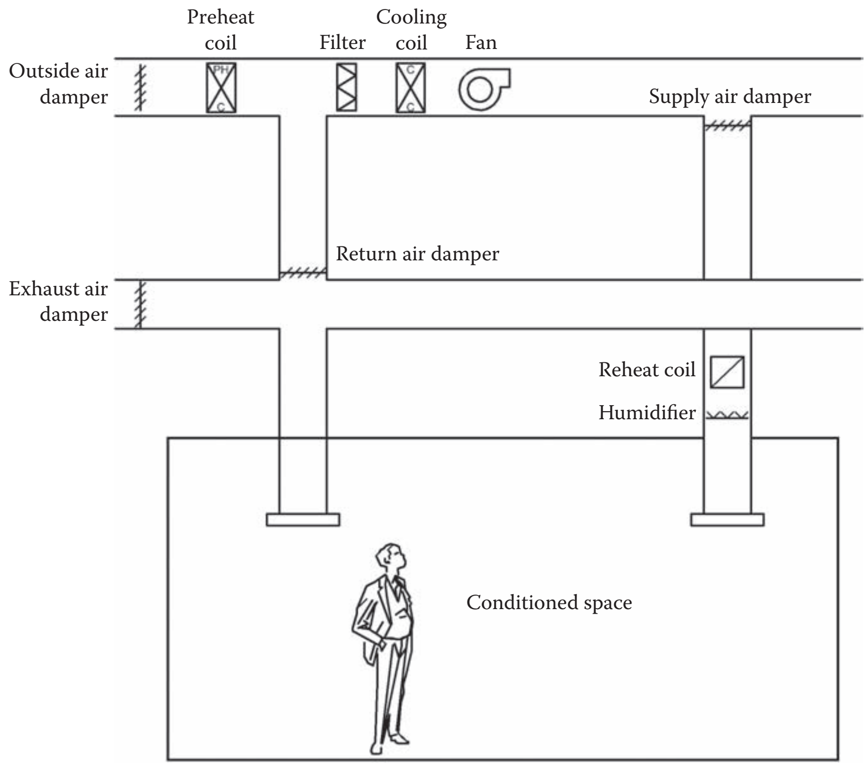


FIGURE 7.1 Typical air-handling unit for an air HVAC system.

(a) Constant-air-volume with terminal reheat systems. These systems require that circulated air be cooled to meet design thermal loads. If partial thermal load conditions occur, reheating of precooled air is required.
(b) Constant-air-volume systems with terminal reheat in interior spaces and perimeter induction or fan-coil units. For these systems, the energy waste is not significant for the perimeter spaces. Indeed, a large portion of the air supplied to the perimeter spaces is recirculated within each perimeter space by either induction or fan-coil units.
(c) All-air induction systems with perimeter reheat. The system attempts to capture heat from lights by returning air through the light fixtures. The induction units accept varying amounts of warm return air to mix with primary air for temperature control. The energy waste due to reheat is small for these systems. However, extensive static pressure control is required at the terminals.
(d) Constant-air-volume dual duct systems. These systems have a cold air duct and a hot air duct. The supply air temperature is controlled by mixing cold air with hot air proportionally to meet the thermal load of the space. Energy waste occurs during partial thermal load conditions when mixing is needed.

2. *Variable-Air-Volume (VAV) Systems:* These systems provide a variable amount of supply air conditioned at a constant temperature to meet thermal loads in all spaces based on thermostat settings. The supply air volume can be controlled and modulated using various techniques such as outlet dampers, inlet vanes, and variable speed drives. Typically, only cooled air is supplied at the central air-handling unit. In each space, reheat is provided depending on the space thermal load. These systems waste significantly less energy than constant air volume. Retrofitting existing constant

volume systems to variable air volume systems constitutes a common and generally cost-effective energy conservation measure for secondary HVAC systems. Among the variable air volume systems commonly used are:

(a) Variable-air-volume with terminal reheat systems. These systems reduce the amount of air supplied as the cooling load reduces until a preset minimum volume is reached. At this minimum volume, reheat is provided to the supply air to meet the thermal load. Because of this volume reduction, reheat energy waste is significantly reduced relative to the constant air volume systems with reheat terminal.

(b) Variable-air-volume systems with perimeter heating systems. These systems provide cooling only and heating is performed by other auxiliary systems such as hot water baseboard units. The baseboard heating units are often controlled by outside air temperature because the perimeter heating load is a function of the transmission losses.

(c) Variable air volume dual duct systems. These systems have a cold air duct and a hot air duct and operate in a similar manner to the variable-air-volume systems with terminal reheat. As the cooling load decreases, only the cold air is supplied until a preset minimum volume is reached. At this minimum volume, the hot air is mixed with the cold air stream.

To summarize, variable-air-volume systems are more energy efficient than the constant air volume (CAV or CV) because they minimize reheat energy waste. However, several energy conservation opportunities can be considered even if the existing HVAC system is a VAV system. The potential for energy savings in the secondary HVAC system depends on several factors including the system design, the method of operation, and the maintenance of the system. Generally, energy can be conserved in the HVAC system by following one or several principles listed below:

- Operate the HVAC systems only when needed. For instance, there is no need to provide ventilation during unoccupied periods.
- Eliminate overcooling and overheating of the conditioned spaces to improve comfort levels and avoid energy waste.
- Reduce reheat because it wastes energy.
- Provide free cooling and heating whenever possible by using economizer cycles or heat recovery systems to eliminate the need for mechanical air conditioning.
- Reduce the amount of air delivered by the HVAC systems by reducing the supply air and especially makeup and exhaust air.

In the following sections, selected energy conservation measures are described starting from measures specific to each component of an air-handling unit to other measures related to conversion of a constant volume system to a variable air volume system.

7.3 Ventilation Systems

The energy required to condition ventilation air can be significant in both commercial buildings and industrial facilities especially in locations with extreme weather conditions. Although ventilation is used to provide fresh air to occupants in commercial buildings, it is also used to control the level of dust, gases, fumes, or vapors in several industrial applications. The auditor should estimate the existing volume of fresh air and compare this estimated amount of ventilation air with that required by the appropriate standards and codes. Excess air ventilation should be reduced if it can lead to increases in heating or cooling loads. Some energy conservation measures related to ventilation are described in this section. However, in some climates and periods of the year or the day, providing more air ventilation can be beneficial and may actually reduce cooling and heating loads through the use of air-side economizer cycles.

7.3.1 Ventilation Air Intake

The auditor should first estimate the existing level of ventilation air brought by the mechanical system (rather than by natural means such as infiltration through the building envelope). As described in Chapter 6, the tracer gas technique can be used to determine the amount of fresh air entering a facility. However, this technique does not differentiate between the outside air coming from the mechanical ventilation system to that from infiltration. Currently, several measurement techniques are available to measure the flow of air through a duct. Some of these techniques are summarized in Table 7.1 which provides the range and accuracy of each listed technique.

It should be noted that all the techniques listed in Table 7.1 provide direct measurement of ventilation air. However, these techniques are relatively expensive and are generally difficult to set up in existing systems. To have an estimation of ventilation air provided by the mechanical system, an enthalpy balance technique can be used. In this technique, the temperature is measured at three locations in the duct system as shown in Figure 7.2: before the outdoor air damper (to measure the outdoor air temperature T_{oa}), in the return duct (to measure the return air temperature T_{ra}), and in the mixing plenum area (to measure the mixing air temperature, T_{ma}). The outside air fraction X_{oa} (fraction of the ventilation air over the total supply air) is then determined using the following equation (based on the first law of thermodynamics):

$$T_{ma} = X_{oa}.T_{oa} + (1 - X_{oa}).T_{ra} \tag{7.1}$$

Thus, the amount of ventilation V_{oa} can be determined under design conditions using the capacity of the air-handling unit V_{des} as indicated in Eq. (7.2):

TABLE 7.1 Accuracy and Range of Airflow Meansurement Techniques

Technique	Range (m/s)	Accuracy (%)	Comments
Pitot-tube	1–45	1–5	For low flows (1–3 m/s), high accuracy DP is needed.
Thermal anemometer	>0.005	2–5	Sensitive to turbulence. Needs frequent calibration
Rotating vanes anemometer	0.5–15	2–5	Susceptible to changes in flow rates. Needs periodic calibration.
Swinging vanes anemometer	0.25–50	10	Not sufficiently accurate for OA measurements.
Vortex shedding meter	>2.5	1–5	Not accurate for low flow rates.
Integrated damper/ measuring device	1– 45	1–5	Same errors and limitation as Pitot-tube.
Laser Doppler anemometer	0.005– 25	1–3	Accurate at low rates. Too costly for field applications.
Orifice meter	>0.1	1–5	Accuracy is affected by installation conditions.

Source: Adapted from Krarti et al., Final report for ASHRAE RP-980, Atlanta, GA: American Society of Heating, Refrigerating, and Air Conditioning Engineers, 1999b.

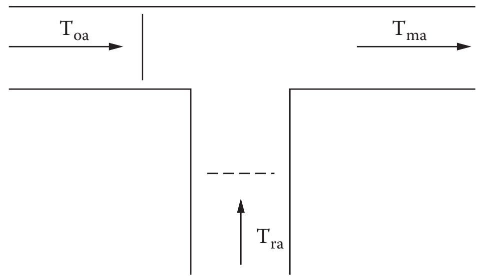


FIGURE 7.2 Location of the temperature sensors to measure the ventilation air fraction in an air handling unit.

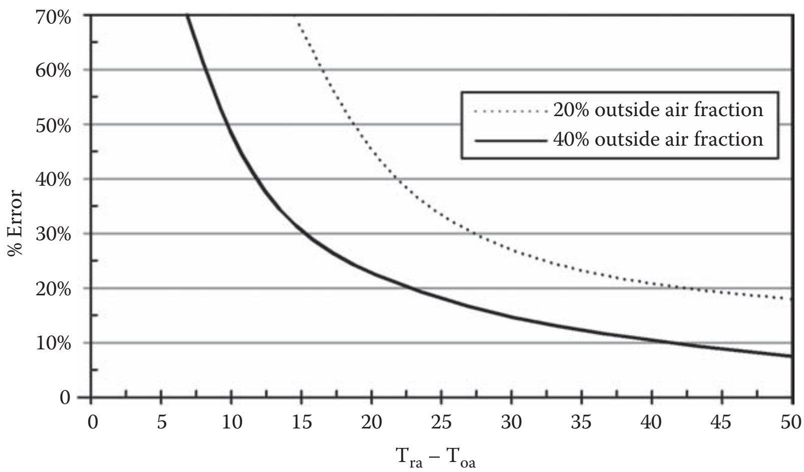


FIGURE 7.3 Predicted errors of temperature balance.

$$\dot{V}_{oa} = X_{oa}.\dot{V}_{des} = \left(\frac{T_{ra} - T_{ma}}{T_{ra} - T_{oa}}\right).\dot{V}_{des} \tag{7.2}$$

It should be noted that the accuracy of the estimation for the ventilation air using Eq. (7.2) is reduced as the difference between the return air and outside air temperature is small. Figure 7.3 indicates the accuracy of the temperature balance for two outside air fractions 20 percent and 40 percent as a function of the temperature difference between return air and outside air (Krarti et al., 1999b). Thus, it is recommended that the auditor perform temperature measurements when the outdoor temperatures are extreme (i.e., during heating or cooling seasons).

Once the existing ventilation air is estimated, it has to be compared to the ventilation requirements by the applicable standards. Table 7.2 summarizes some of the minimum outdoor air requirements for selected spaces in commercial buildings.

If excess ventilation air is found, the outside air damper setting can be adjusted to supply the ventilation to meet the minimum outside requirements as listed in Table 7.2. Further reductions in outdoor air can be obtained by using demand ventilation controls by supplying outside air only during periods when there is a need for fresh air. A popular approach for demand ventilation is the monitoring of the

TABLE 7.2 Minimum Ventilation Rate Requirements for Selected Spaces in Commercial Buildings

Space or Application	Minimum Outside Air Requirements	Reference
Office space	9.5 L/s (20 cfm) per person	ASHRAE Standard 62-2004
Corridor	0.25 L/s per m^2 (0.05 cfm/ft^2)	
Restroom	24 L/s (50 cfm) per toilet	
Smoking lounge	28.5 L/s (60 cfm) per person	
Parking garage	7.5 L/s (1.5 cfm/ft^2)	

CO_2 concentration level within the spaces. CO_2 is considered a good indicator of pollutants generated by occupants and other construction materials. The outside air damper position is controlled to maintain a CO_2 set-point within the space.

The energy savings due to reduction in the ventilation air can be attributed to lower heating and cooling loads required to condition outdoor air. The instantaneous heating and cooling savings can be estimated using, respectively, Eq. (7.3) and Eq. (7.4):

$$\Delta e_H = \rho_a . c_{P,a} . (\dot{V}_{oa,E} - \dot{V}_{oa,R}) . (T_i - T_o) \tag{7.3}$$

and

$$\Delta e_C = \rho_a . (\dot{V}_{oa,E} - \dot{V}_{oa,R}) . (h_o - h_i) \tag{7.4}$$

where

$\dot{V}_{oa,E}$, and $\dot{V}_{oa,R}$ are, respectively the ventilation air rate before and after retrofit.

$\rho_{a,}$ and $c_{p,a}$ are, respectively, the density and the specific heat of ventilation air.
T_i and T_o are the air temperatures of, respectively, the indoor space and outdoor ambient during winter.
h_i and h_o are the air enthalpies of, respectively, the indoor space and the outdoor ambient during summer.

It should be noted that the humidity control is not typically performed during the winter and thus the latent energy is neglected as indicated in Eq. (7.3).

To determine the total energy use savings attributed to ventilation air reduction, the annual savings in heating and cooling loads has to be estimated. Without using detailed energy simulation, the savings in heating and cooling loads can be calculated using Eqs. (7.3) and (7.4) for various bin temperatures and summing the changes in the thermal loads over all the bins. By taking into account the energy efficiency of the heating and cooling equipment, the energy use savings due to a reduction in the ventilation air can be estimated for both winter and summer as illustrated in Eqs. (7.5) and (7.6), respectively:

$$\Delta kWh_H = \frac{3.6 * \sum_{k=1}^{N_{bin}} N_{h,k} . \Delta e_{H,k}}{\eta_H} \tag{7.5}$$

and

$$\Delta kWh_C = \frac{3.6 * \sum_{k=1}^{N_{bin}} N_{h,k} . \Delta e_{C,k}}{EER_C} \tag{7.6}$$

where

$N_{h,}k$ = the number of hours in bin k.

EERC = the average seasonal efficiency ratio for the cooling system.

η_H = the average seasonal efficiency of the heating system.

When an air-side economizer is present, the summation in both Eqs. (7.5) and (7.6) should be performed for only the bin temperatures corresponding to time periods when the outside air damper is set at its minimum position.

The calculations of the energy use savings can be further simplified if the air density is assumed to be constant (i.e., independent of temperature) and if the HVAC system has a year-round operation. Under these conditions, the energy savings due to heating can be estimated as follows:

$$\Delta kWh_H = \frac{3.6 * \rho_a.c_{p,a}.N_h.(\dot{V}_{oa,E} - \dot{V}_{oa,R}).(T_i - \overline{T}_o)}{\eta_H} \tag{7.7}$$

where

N_h = the total number of hours in the heating season. When the ventilation air is not provided during all hours, N_h can be adjusted to include only occupied hours (i.e., periods when ventilation is provided) assuming that the average outdoor air temperature does not vary significantly with this adjustment.

$\overline{T}_o$ = the average outdoor air temperature during the heating season.

For the energy use savings due to cooling, a simplified form of Eq.(7.5) can be obtained with the introduction of a seasonal cooling load to condition a reference amount (such as 1,000 m^3/hr or 1,000 cfm) of outdoor air ΔHc which depends on the climate and the indoor temperature setting:

$$\Delta kWh_C = \frac{3.6 * \rho_a.N_h.(\dot{V}_{oa,E} - \dot{V}_{oa,R}).\Delta H_C}{EER_C} \tag{7.8}$$

Other measures that reduce the ventilation air through the HVAC system include the following actions:

1. Reduce leakage through the outside air damper especially when this damper is set to be closed. The change in the ventilation air can be calculated using the leakage percentage. The low leakage dampers can restrict leakage to less than 1 percent and the standards can allow 5 percent up to 10 percent leakage when closed.
2. Eliminate the ventilation during unoccupied periods or when ventilation is not needed.

The calculations of the energy savings for the above-listed measures follow the same methods illustrated by Eq. (7.3) through Eq. (7.8). Some examples are provided at the end of this chapter to show how the energy use savings can be calculated for these measures.

It should be noted that when the ventilation air is reduced, the amount of exhaust air should also be adjusted. Otherwise, a negative static pressure can be obtained in the building (because more air is exhausted than introduced to the building). Several problems can occur because of the negative pressure within the building including:

- Difficulty in opening exterior doors and windows.
- Draft can be felt at the perimeter of the building because outside cold air is drawn near the windows and doors.
- Accumulation of fumes, odors, dirt, and dust is increased because exhaust fans cannot operate at rated capacity under negative pressure.
- Combustion efficiency of boilers and ovens can decrease if these systems depend on natural draft to operate properly.

7.3.2 Air Filters

To remove dust and other unwanted particulates from air supplied to conditioned spaces, air filters are typically placed in air-handling units or packaged HVAC systems. There are two main types of air filters commonly used in building HVAC systems:

1. Dry-type extended surface filters that consist of bats or blankets made up of fibrous materials such as bonded glass fiber, cellulose, wool felt, and synthetic materials
2. Viscous impingement filters that include filter media coated with viscous substances (such as oil) to catch particles in the air stream

The efficiency of an air filter to remove particulates from air streams can be measured using standard testing procedures such as ASHRAE Standard 52.1. Table 7.3 outlines typical efficiencies and rated air flow velocities for commonly available air filters. The flat filters are the least efficient and the least expensive. The HEPA (high-efficiency particulate air) filters are the most efficient but are generally expensive. For general HVAC applications such as office buildings and schools, pleated filters are typically used. Bag filters are suitable for most hospital spaces. HEPA filters are appropriate for clean-room applications. To extend the life of the more expensive filters, it is a common practice to add inexpensive prefilters which are placed upstream of the more expensive and effective filters in order to reduce dirt loading.

Air filters affect the pressure pattern across the ducts and can increase the electric power requirement for supply fans. Table 7.3 indicates typical pressure drops created by clean and dirty air filters. As expected, the pressure drop is higher for dirty air filters. Moreover, the pressure drop increases as the efficiency of the filter increases.

7.3.3 Air-Side Economizers

When the outdoor air conditions are favorable, excess ventilation air can actually be used to condition the building and thus reduce the cooling energy use for the HVAC system. There are typically two control strategies to determine the switchover point and decide when it is better to use more than the minimum required amount of the outdoor air to cool a building: one strategy is based on dry-bulb temperatures and the other on enthalpies. These control strategies are known as temperature and enthalpy air-side economizers, respectively. For both strategies, the operation of the HVAC system is said to be on an economizer cycle. Several existing HVAC systems do not have an economizer cycle and thus do not take advantage of its potential energy use savings. The two economizer cycles are briefly described below.

7.3.3.1 Temperature Economizer Cycle

For this cycle, the outside air intake damper is opened beyond the minimum position whenever the outside air temperature is colder than the return air temperature. However, when the outdoor air temperature is either too cold or too hot, the outside air intake damper is set back to its minimum position. Therefore, there are outdoor air temperature limits beyond which the economizer cycle should

TABLE 7.3 Typical Efficiencies and Pressure Drops for Air Filters

Filter Type	Average Efficiency (%)	Rated Air Face Velocity (fpm)	Pressure Drop for Clean Filter (In. Water)	Pressure Drop for Dirty Filter (In. Water)
Flat	85	500	0.10–0.20	1.00
Pleated	90	500	0.15–0.40	1.00
Bag	90	625	0.25–0.40	1.00
HEPA	99.9	250–500	0.65–1.35	1.00

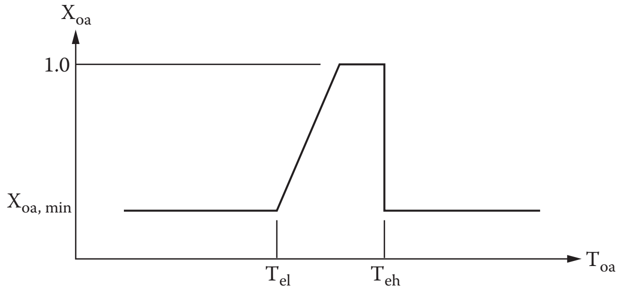


FIGURE 7.4 Economizer temperature limits.

not operate. These temperature limits are called, respectively, the economizer low temperature limit T_{el}, and economizer high temperature limit T_{eh}, as illustrated in Figure 7.4. Beyond the two economizer temperature limits, the outside air intake damper is set to its minimum position.

Although the economizer high temperature limit is typically difficult to define and is often set to be the same as the return air temperature, the economizer low limit temperature can be determined as a function of the conditions of both the return air and supply air using a basic principle. There is no need to introduce more than the required amount (i.e., for ventilation purpose) of outdoor air if it has more heat content than the return air.

7.3.3.2 Enthalpy Economizer Cycle

This cycle is similar to the temperature-based economizer cycle except that enthalpy of the air streams is used instead of the temperature. Therefore two parameters are typically measured for each air stream in order to estimate its enthalpy (dry- and wet-bulb temperatures, for instance). Because of this requirement, the enthalpy economizers are less common inasmuch as they are more expensive to implement and less robust to use even though they may achieve greater savings if properly operated.

7.4 Ventilation of Parking Garages

Automobile parking garages can be partially open or fully enclosed. Partially open garages are typically above-grade with open sides and generally do not need mechanical ventilation. However, fully enclosed parking garages are usually underground and require mechanical ventilation. Indeed, in the absence of ventilation, enclosed parking facilities present several indoor air quality problems. The most serious is the emission of high levels of carbon monoxide (CO) by cars within the parking garages. Other concerns related to enclosed garages are the presence of oil and gasoline fumes, and other contaminants such as oxides of nitrogen (NO_x) and smoke haze from diesel engines.

To determine the adequate ventilation rate for garages, two factors are typically considered: the number of cars in operation and the emission quantities. The number of cars in operation depends on the type of the facility served by the parking garage and may vary from 3 percent (in shopping areas) up to 20 percent (in sports stadiums) of the total vehicle capacity (ASHRAE, 2007). The emission of carbon monoxide depends on individual cars including such factors as the age of the car, engine power, and level of car maintenance.

For enclosed parking facilities, ASHRAE Standard 62 specifies a fixed ventilation rate of below 7.62 L/s.m^2 (1.5 cfm/ft^2) of gross floor area (ASHRAE, 2004). Therefore, a ventilation flow of about 11.25 air changes per hour is required for garages with 2.5-m ceiling height. However, some of the model code authorities specify an air change rate of 4 to 6 air changes per hour. Some of the model code authorities allow the ventilation rate to vary and be reduced to save fan energy if CO-demand controlled ventilation is implemented; that is, a continuous monitoring of CO concentrations is conducted, with the monitoring system being interlocked with the mechanical exhaust equipment. The acceptable level of contaminant concentrations varies significantly from code to code as outlined in the following section.

7.4.1 Existing Codes and Standards

Table 7.4 provides a summary of existing codes and standards for ventilating enclosed parking garages in the United States, and other selected countries.

As shown in Table 7.4, the recommendations for the CO exposure limits are not consistent among various regulations within the United States and between countries. However, the recommendations offer an indication of risks from exposure to CO in parking garages. A limit level of 25 ppm for long-term CO exposure would meet almost all the codes and standards listed in Table 7.4. As part of an ASHRAE-sponsored project (945-RP), field measurements for the seven tested parking facilities were performed. For a more detailed description of the field measurements, refer to Krarti et al. (1999a). In particular, the following selected results were obtained from the field study (Ayari and Krarti, 2000):

1. All the tested enclosed parking garages had contaminant levels that were significantly lower than those required by even the most stringent regulations (i.e., 25 ppm of 8-hr weighted average of CO concentration).
2. The actual ventilation rates supplied to the tested garages were generally well below those recommended by ASHRAE Standard 62-1989 [i.e., below 7.6 L/s.m^2 (1.5 cfm/ft^2)].
3. When it was used, demand controlled ventilation was able to maintain acceptable indoor air quality within the tested enclosed parking facilities.

TABLE 7.4 Summary of U.S. and International Standards for Ventilation Requirements of Enclosed Parking Garages

	Time (Hrs)	Ppm	Ventilation
ASHRAE	8	9	7.6 L/s.m^2
	1	35	(1.5 cfm/ft^2)
ICBO	8	50	7.6 L/s.m^2
	1	200	(1.5 cfm/ft^2)
NIOSH/OSHA	8	35	
	ceiling	200	
BOCA			6 ACH
SBCCI			6–7 ACH
NFPA			6 ACH
ACGIH	8	25	
Canada	8	11/13	
	1	25/30	
Finland	8	30	2.7 L/s.m^2
	15 min	75	(0.53 cfm/ft^2)
France	ceiling	200	165 L/s.car
	20 min	100	(350 cfm/car)
Germany			3.3 L/s.m^2
			(0.66 cfm/ft^2)
Japan/South Korea			6.35–7.62 L/s.m^2
			(1.25–1.5 cfm/ft^2)
Netherlands	0.5	200	
Sweden			0.91 L/s.m^2
			(0.18 cfm/ft^2)
United Kingdom	8	50	6–10 ACH
	15 min	300	

4. The location of the supply and exhaust vents, the traffic flow pattern, the number of moving cars, and the travel time were important factors that affected the effectiveness of the ventilation system in maintaining acceptable CO (or NO_x) levels within enclosed parking garages. Any design guidelines should account for these factors to determine the ventilation requirements for enclosed parking facilities.

It is clear from the results of the field study that the current ventilation rate specified in the ASHRAE standard 62 for enclosed parking garages can lead to significant energy waste. To prevent this waste, ventilation requirements for enclosed parking garages can be estimated using a new design method developed by Krarti et al. (1999a). The new method allows the estimation of the minimum ventilation rate required to maintain contaminant concentrations within parking facilities at the acceptable levels set by the relevant health authorities without large penalties in fan energy use. Moreover, the new method accounts for variability in parking garage traffic flow, car emissions, travel time, and number of moving cars.

7.4.2 General Methodology for Estimating the Ventilation Requirements for Parking Garages

Based on the results of several parametric analyses (Krarti et al., 1999a), a simple design method was developed to determine the ventilation flow rate required to maintain an acceptable CO level within enclosed parking facilities. Ventilation rates for enclosed parking garages can be expressed in terms of either flow rate per unit floor area (L/s.m^2 or cfm/ft^2) or air volume changes per unit time (ACH). The design ventilation rate required for an enclosed parking facility depends on four factors:

1. Contaminant level acceptable within the parking facility
2. Number of cars in operation during peak conditions
3. Length of travel and operation time of cars in the parking garage
4. Emission rate of a typical car under various conditions

Data for the above-listed factors should be available to determine accurately the design ventilation rate for enclosed parking garages. A simple design approach is presented in the following section to determine the required ventilation rate for both existing and newly constructed enclosed parking garages. This design approach is also described in the *ASHRAE Handbook* (ASHRAE, 2007).

To determine the required design flow rate to ventilate an enclosed parking garage, the following procedure is recommended:

7.4.2.1 Step 1. Collect the Following Data

(i) Number of cars in operation during the hour of peak use, N (# of cars). The *ITE Trip Generation Handbook* (ITE, 2003) is a good source to estimate the value of N.

(ii) Average CO emission rate for a typical car per hr ER (gr/hr). The CO emission rate for a car depends on several factors such as vehicle characteristics, fuel types, vehicle operation conditions, and environment conditions. Data provided in the *ASHRAE Handbook* (ASHRAE, 2007) and reproduced in Table 7.5 can be used to estimate CO emission rates for a typical car. Typically, hot starts are common in facilities where cars are parked for short periods such as shopping malls. On the other hand, cold starts characterize facilities where cars park for long periods such as office buildings.

(iii) Average length of operation and travel time for a typical car T (seconds). The *ASHRAE Handbook* gives average entrance/exit times for vehicles. However, higher values may be used for worst-case scenarios such as during rush hours or special events.

(iv) The level of CO concentration acceptable within the garage, CO_{max} (ppm). This level can be defined based on the recommendations of the applicable standards (typically, CO_{max} = 25 ppm).

(v) Total floor area of the parking area A_f (ft^2 or m^2).

TABLE 7.5 Typical CO Emissions Within Parking Garages

	Hot Emissions (Stabilized) grams/min		Cold Emissions grams/min	
Season	1991	1996	1991	1996
Summer [32°C (90°F)]	2.54	1.89	4.27	3.66
Winter [0°C (32°F)]	3.61	3.38	20.74	18.96

Source: ASHRAE, 2007, *Handbook of HVAC Applications*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. Atlanta, GA.

7.4.2.2 Step 2.

(i) Determine the peak generation rate, GR [gr/hr.m^2 (gr/hr.ft^2)], for the parking garage per unit floor area using Eq. (7.9):

$$GR = \frac{N^* \, ER}{A_f} \tag{7.9}$$

(ii) Normalize the value of generation rate using a reference value GR_o = 26.8 gr/hr.m2 (GR_o = 2.48 gr/hr.ft2). This reference value was obtained using the worst emission conditions (cold emissions in winter season) for an actual enclosed parking facility (Krarti et al., 2000):

$$f = \frac{GR}{GR_o} * 100 \tag{7.10}$$

7.4.2.3 Step 3.

Determine the required ventilation rate per unit floor area (L/s.m^2 or cfm.ft^2), the correlation presented by Eq. (7.11) depending on the maximum level of CO concentration CO_{max}:

$$L/s.m^2 = C.\ f.\ T \tag{7.11}$$

where the correlation coefficient *C* is given below:

$$1.204 \times 10^{-3}\ L/m^2.s^2\ (2.370 \times 10^{-4}\ cfm/ft^2.s) \text{ for } CO_{max} = 15 \text{ ppm}$$

$$C = 0.692 \times 10^{-3}\ L/m^2.s^2\ (1.363 \times 10^{-4}\ cfm/ft^2.s) \text{ for } CO_{max} = 25 \text{ ppm}$$

$$0.482 \times 10^{-3}\ L/m^2.s^2\ (0.948 \times 10^{-4}\ cfm/ft^2.s) \text{ for } CO_{max} = 35 \text{ ppm}$$

and *T* is the average travel time of cars within the garage in seconds.

Example 7.1

An office building has a two-level enclosed parking garage with a total capacity of 450 cars, a total floor area of 8,300 m^2 (89,290 ft^2), and an average height of 2.75 m (9.0 ft). The total length of time for a typical car operation within the garage is 2 minutes (120 s).

(a) Determine the required ventilation rate for the enclosed parking garage in L/s.m^2 (or cfm/ft^2) and in ACH so that CO levels never exceed 25 ppm. Assume that the number of cars in operation is 40 percent of the total vehicle capacity (a shopping mall facility).

(b) Determine the annual fan energy savings if the ventilation found in (a) is used instead of 1.5 cfm/ ft^2 recommended by ASHRAE 62-1989. The ventilation system is operated 16 hours/ day over 365 days/year. The cost of electricity is \$0.07/kWh. Assume the size of the initial ventilation fan is 30 hp (22.9 kW).

Solution

(a) The procedure for the new design methodology can be applied to estimate the minimum ventilation requirements for the garage defined in this example:

Step 1. Garage data: $N = 450 * 0.4 = 180$ cars, ER = 11.66 gr/min (average emission rate for a winter day using the data from Table 7.4), $T = 120$ s, $CO_{max} = 25$ ppm.
Step 2. Calculate CO generation rate:
Step 3. Determine the ventilation requirement. Using the correlation of Eq. (7.11) for $CO_{max} = 25$ ppm, the design ventilation rate in L/s.m^2 can be calculated:

$$\text{L/s.m}^2 = 0.692 \times 10^{-3} * 56.6 * 120 \text{ s} = 4.7$$

This ventilation rate corresponds to 0.91 cfm/ft^2. In terms of air change per hour, the ventilation rate required for the enclosed parking garage is:

$$\text{ACH} = \frac{4.7 \ \ \text{L/m.s}^2 \text{ x } 10^{-3} \text{ L/m}^3 \text{ x } 3600 \ \ \text{s/hr}}{2.75 \text{ m}} = 6.1$$

(a) Due to the reduced ventilation rate (from 1.5 cfm/ft^2 to 0.91 cfm/ft^2), the fan energy use can be reduced either by installing smaller fans or by reducing the fan speeds (for fans equipped with variable speed drives). In either case, the theoretical reduction in fan energy use can be estimated using the fan laws (in particular, these laws state that fan power consumption is proportional to the cube of the ventilation flow rate):

$$\Delta kW_{e,fan} = \frac{kW_{exist}}{\eta_m} .(\frac{\dot{m}_{new}}{\dot{m}_{exist}})^3 .N_h$$

Because N_h = 16 hrs/day * 365 days/yr = 5,840 hrs/yr and kW_{exist} =22.9 kW with $\dot{m}_{new} / \dot{m}_{exist}$ = 0.91/1.5 = 0.61, the annual fan energy savings can be estimated to be (assuming 90 percent motor efficiency):

$$\Delta kW_{e,fan} = \frac{22.9kW}{0.90} .(0.61)^3 .5{,}840 \ \ hrs/yr = 33{,}728 kWh/yr$$

Based on an electricity rate of \$0.07/kWh, the cost savings attributed to savings in fan energy use is:

$$\Delta C_{e,fan} = \$0.07/kWh * 33{,}728 \ \ kWh/yr = \$2{,}361/yr$$

To further conserve energy, fan systems can be controlled by CO meters to vary the amount of air supplied, if permitted by local codes. For example, fan systems could consist of multiple fans with single-or variable-speed motors or variable pitch blades. In multilevel parking garages or single-level structures of extensive area, independent fan systems, each under individual control are preferred. Figure 7.5

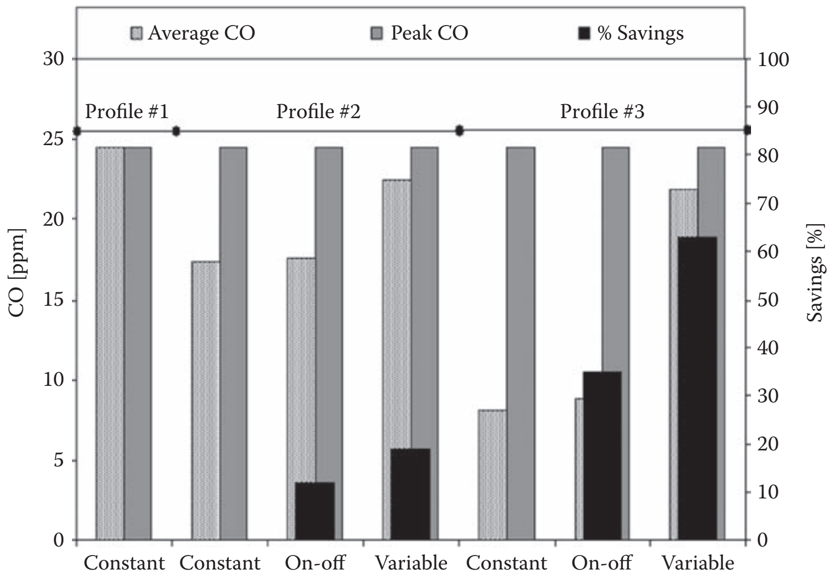


FIGURE 7.5 Typical energy savings and maximum CO level obtained for demand CO-ventilation controls.

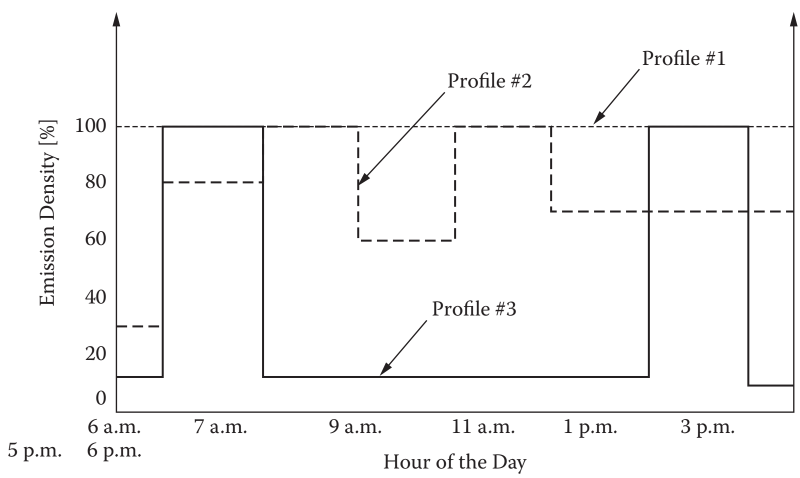


FIGURE 7.6 Car movement profiles used in the analysis conducted by Krarti et al. Final report for ASHRAE RP-945, Atlanta, GA: American Society of Heating, Refrigerating, and Air Conditioning Engineers, (1999a).

provides the maximum CO level in a tested garage (Krarti et al., 1999a) for three car movement profiles (as illustrated in Figure 7.6) and three ventilation control strategies:

(i) CV where the ventilation is on during the entire occupancy period
(ii) On–off with the fans are operated based on CO-sensors, (either on or off)
(iii) VAV with variable volume fans that are adjusted depending on the CO level within the garage

Figure 7. 5 also indicates the fan energy savings achieved by the on–off and VAV systems (relative to the fan energy use by the CV system). As illustrated in Figure 7.5, significant fan energy savings can be obtained when a demand CO-ventilation control strategy is used to operate the ventilation system while maintaining acceptable CO levels within the enclosed parking facility.

7.5 Indoor Temperature Controls

The indoor temperature settings during both heating and cooling seasons have significant effects on the thermal comfort within occupied spaces and on the energy use of the HVAC systems. It is therefore important for the auditor to assess the existing indoor air temperature controls within the facility to evaluate the potential for reducing energy use or improving indoor thermal comfort without any substantial initial investment. There are four options for adjustments of the indoor temperature setting that can save heating and cooling energy:

1. Eliminating overcooling by increasing the cooling set-point during the summer.
2. Eliminating overheating by reducing the heating set-point during the winter.
3. Preventing simultaneous heating and cooling operations for the HVAC system by separating heating and cooling set-points.
4. Reducing heating or cooling requirements during unoccupied hours by setting back the set-point temperature during heating and setting up the set-point temperature (or letting the indoor temperature float) during cooling.

The calculations of the energy use savings for these measures can be estimated based on degree-days methods as outlined in Chapter 6. However, even more simplified calculation methods can be considered to estimate the magnitude of the energy savings. Some examples are provided to illustrate the energy use savings due to adjustments in indoor temperature settings.

It should be noted that some of the above-listed measures could actually increase energy use if they are not adequately implemented. For instance, when the indoor temperature is set lower during the winter, the interior spaces may require more energy because they need to be cooled rather than heated. Similarly, setting the indoor temperature higher can lead to an increase in the reheat energy use for the zones with reheat systems.

7.6 Upgrade of Fan Systems

7.6.1 Introduction

Fans are used in several HVAC systems to distribute air throughout the building. In particular, fans are used to move conditioned air from central air-handling units to heat or cool various zones within a building. According to a survey reported by the U.S. Energy Information Administration (EIA, 2006), the energy use for fans represents about 15 percent of the total electrical energy use of a typical office building. Thus, improvements in the operation of fan systems can provide significant energy savings.

In a typical air-handling unit, fans create the pressure required to move air through ducts, heating or cooling coils, filters, and any other obstacles within the duct system. Two types of fans are used in HVAC systems: *centrifugal* and *vane-axial* fans. The centrifugal fan consists of a rotating wheel, generally referred to as an impeller, mounted in the center of a round housing. The impeller is driven by an electric motor through a belt drive. The vane-axial fan includes a cylindrical housing with the impeller mounted inside along the axis of the cylindrical housing. The impeller of an axial fan has blades mounted around a central hub similar to an airplane propeller. Typically, axial fans are more efficient than centrifugal fans but are more expensive because they are difficult to construct. Currently, the centrifugal fans are significantly more common in existing HVAC systems.

There are several energy conservation measures that help reduce the energy use of fan systems. Some of these measures are described in this section. A brief review of basic laws that characterize the fan operation is provided first.

7.6.2 Basic Principles of Fan Operation

Fans generally fall into two categories as illustrated in Figure 7.7. Centrifugal fans use a rotating drum to “throw” air into a duct. Axial fans are propellers that “sweep” air into a duct. The performance of a

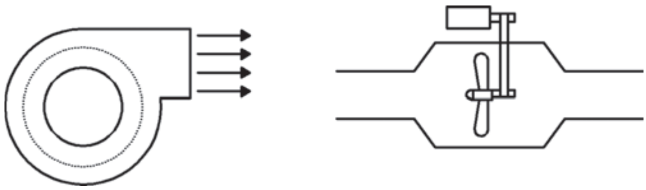


FIGURE 7.7 Fan types commonly used in air-handling units.

fan depends on the power of the motor, the shape and orientation of the fan blades, and the inlet and outlet duct configurations.

To characterize the operation of a fan, several parameters need to be determined including the electrical energy input required in kW (or hp), the maximum amount of air it can move in L/s (or cfm) for a total pressure differential (ΔP_T) or a static pressure differential (ΔP_s), and the fan efficiency. A simple relationship exists that allows calculation of the electrical energy input required for a fan as a function of the air flow amount, the pressure differential, and its fan efficiency. If the total pressure is used, Eq. (7.12a) provides the electrical energy input using metric units.

$$kW_{fan} = \frac{\dot{V}_f . \Delta P_T}{\eta_{f,t}} \tag{7.12a}$$

Using English units, the horsepower of the fan can be determined as follows:

$$Hp_{fan} = \frac{\dot{V}_f . \Delta P_T}{6{,}356 * \eta_{f,t}} \tag{7.12b}$$

If static pressure is considered instead of total pressure, Eqs. (7.13a) and (7.13b) should be used. Note that the static fan efficiency is needed.

For SI units:

$$kW_{fan} = \frac{\dot{V}_f . \Delta P_s}{\eta_{f,s}} \tag{7.13a}$$

For IP units:

$$kHP_{fan} = \frac{\dot{V}_f . \Delta P_s}{6{,}356 * \eta_{f,s}} \tag{7.13b}$$

To measure the total pressure of the fan, a Pitot-tube can be used in two locations within the duct that houses the fan as illustrated in Figure 7.8.

As shown in Figure 7.7, the total pressures $P_{t,i}$ and $P_{t,o}$ at, respectively, the inlet and outlet of the fan are first measured. Then, the fan total pressure is simply found by taking the difference:

$$\Delta P_T = P_{t,o} - P_{t,i} \tag{7.14}$$

When any of the fan parameters (such as total pressure ΔP_T , static pressure ΔP_s , or air flow rate $\dot{V}_f$) vary, they are still related through certain laws, often referred to as the fan laws. Fan laws can be very useful to estimate the energy savings from any change in fan operation. These fan laws are summarized in the following three equations.

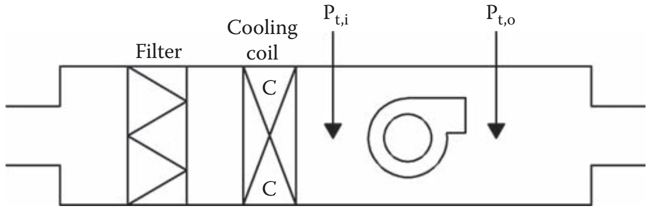


FIGURE 7.8 Measurement of total pressure of the fan.

Equation (7.15) states that the air flow rate $\dot{V}_f$ is proportional to the fan speed ω:

$$\frac{\dot{V}_{f,1}}{\dot{V}_{f,2}} = \frac{\omega_1}{\omega_2} \tag{7.15}$$

Equation (7.16) indicates that the fan static pressure ΔP_s varies as the fan air flow rate $\dot{V}_f$ or the square of the fan speed ω:

$$\frac{\Delta P_{s,1}}{\Delta P_{s,2}} = \left(\frac{\dot{V}_{f,1}}{\dot{V}_{f,2}}\right)^2 = \left(\frac{\omega_1}{\omega_2}\right)^2 \tag{7.16}$$

Finally Eq. (7.17) states that the power used by the fan kW_{fan} varies as the cube of the fan air flow rate $\dot{V}_f$, or the square of the fan speed ω:

$$\frac{kW_{fan,1}}{kW_{fan,2}} = \left(\frac{\dot{V}_{f,1}}{\dot{V}_{f,2}}\right)^3 = \left(\frac{\omega_1}{\omega_2}\right)^3 \tag{7.17}$$

Generally, fans are rated based on standard air density. However, when the air density changes, the fan static pressure and the power required to drive the fan will change. Both fan static pressure and fan power vary in direct proportion to the change in air density.

There are several applications to the fan laws. In particular, when the volume of air needs to be reduced (as in the case of CV to VAV retrofits) two options can be considered:

1. Static pressure can be increased by closing dampers (for instance, by using outlet dampers or variable inlet vanes).
2. Speed can be reduced by using variable speed drives (through the use of variable frequency drives).

It can be noted from the fan laws that by reducing the amount of air to be moved by the fan, the electrical energy input required is reduced significantly. For instance, a 50 percent reduction in the volume of air results in an 87.5 percent reduction in fan energy use. This fact hints clearly at the advantage of using variable-air-volume fan systems compared to constant-volume fans.

In HVAC systems, there are three common approaches to reduce the air volume including

1. *Outlet dampers*: located at the outlet of the fan. To reduce the air volume, the dampers are controlled to the desired position. Effectively, the outlet dampers increase air flow resistance.
2. *Variable inlet vanes:* to change the characteristics of the fan. The position of the fan inlet vanes can be changed to allow a variable volume of air.
3. *Variable speed drives:* to change the speed of the fan. Using variable frequency drives (see Chapter 5), it is possible to control fan speed and thus supplied air volume. Theoretically and according to the fan laws, fan power consumption can be reduced substantially when fan speed is reduced. However, due to back pressure effects, the actual reduction in fan power consumption is less significant (see Example 7.5 for more details).

TABLE 7.6 Percent Reduction of Fan Power as a Function of the Percent Reduction in Air Volume for the Three Control Options

Fan Control Option	Fan Air Volume (in % of Maximum Flow) 100%	80%	60%	40%
Outlet dampers	100	96	88	77
Variable inlet vanes	100	78	61	51
Variable speed drives:				
Actual performance	100	64	36	16
Theoretical performance	100	51	22	6

Table 7.6 compares the reduction of fan power as a function of the reduction in air volume for the three control options. For the variable speed drives, the fan performance curve is given for both actual and theoretical conditions.

To determine the best control approach to vary the air volume (for VAV systems, for instance) an economic analysis is recommended. However, the use of variable frequency drives is often more cost-effective than the other control options. Example 7.2 illustrates a simplified energy and economic analysis to determine the best control option for VAV fan system.

Example 7.2

A 50,000-cfm CV air-handling unit is to be converted to a VAV system. The motor for the existing supply fan is rated at 35 hp with 90 percent efficiency. The fan is operated for 3,500 hours/year according to the following load profile:

- 100 percent load, 10 percent of the time
- 80 percent load, 40 percent of the time
- 60 perent load, 40 percent of the time
- 40 percent load, 10 percent of the time

Three options for variable air volume controls have been considered:

(a) Outlet dampers at a cost of $,3,500
(b) Variable inlet vanes at a cost of $8,000
(c) Variable speed drive at a cost of 9,000

Determine the best option for the fan control using simple payback period analysis. Assume that electricity costs $0.05/kWh.

Solution

The electrical energy input of the supply fan motor is first determined for each control option by calculating a weighted average fan horse power rating using the fan performance curves given in Table 7.6.

For the outlet dampers:

$$k\bar{W}_{e,fan}^{OD} = 35 \quad Hp \quad * (1.0 * 0.1 + 0.96 * 0.4 + 0.88 * 0.40 + 0.77 * 0.1) = 31.95 \quad Hp$$

For the variable inlet vanes:

$$k\bar{W}_{e,fan}^{VIV} = 35 \quad Hp \quad * (1.0 * 0.1 + 0.78 * 0.4 + 0.61 * 0.40 + 0.51 * 0.1) = 24.76 \quad Hp$$

Solution

The electrical energy input of the supply fan motor is first determined for each control option by calculating a weighted average fan horse power rating using the fan performance curves given in Table 7.6.

For the outlet dampers:

$$k\bar{W}^{OD}_{e,fan} = 35 \quad Hp \quad * (1.0 * 0.1 + 0.96 * 0.4 + 0.88 * 0.40 + 0.77 * 0.1) = 31.95 \quad Hp$$

For the variable inlet vanes:

$$k\bar{W}^{VIV}_{e,fan} = 35 \quad Hp \quad * (1.0 * 0.1 + 0.78 * 0.4 + 0.61 * 0.40 + 0.51 * 0.1) = 24.76 \quad Hp$$

For the variable speed drives, both the actual and the theoretical fan performance curves are considered. The actual fan performance leads to:

$$k\bar{W}^{VSD}_{e,fan} = 35 \quad Hp \quad * (1.0 * 0.1 + 0.64 * 0.4 + 0.36 * 0.40 + 0.16 * 0.1) = 18.06 \quad Hp$$

and the theoretical fan performance gives:

$$k\bar{W}^{VSD}_{e,fan} = 35 \quad Hp \quad * (1.0 * 0.1 + 0.51 * 0.4 + 0.22 * 0.40 + 0.064 * 0.1) = 13.99 \quad Hp$$

The total number of hours for operating the fan for one year is $N_{f,h}$ = 3,500 hours for all fan types including the constant volume fan (used as a base case for the payback analysis). First, the electrical energy use for each fan control option is estimated.

For the existing constant volume fan:

$$kWh^{CV}_{e,fan} = \left(\frac{35Hp * 0.746kW/Hp}{0.90}\right) * 3{,}500hrs = 101{,}539 \quad kWh/yr$$

For the outlet dampers (OD):

$$kWh^{OD}_{e,fan} = \left(\frac{31.95Hp * 0.746kW/Hp}{0.90}\right) * 3{,}500hrs = 92{,}690 \quad kWh/yr$$

For the variable inlet vanes (VIV):

$$kWh^{OD}_{e,fan} = \left(\frac{24.76Hp * 0.746kW/Hp}{0.90}\right) * 3{,}500hrs = 71{,}832 \quad kWh/yr$$

For the actual performance of the variable speed drives (VSD):

$$kWh^{OD}_{e,fan} = \left(\frac{18.06Hp * 0.746kW/Hp}{0.90}\right) * 3{,}500hrs = 52{,}394 \quad kWh/yr$$

For the theoretical performance of the variable speed drives:

$$kWh^{OD}_{e,fan} = \left(\frac{13.99Hp * 0.746kW/Hp}{0.90}\right) * 3{,}500hrs = 40{,}587 \quad kWh/yr$$

It is clear from the results of the economic analysis presented above, that the fan equipped with a variable speed drive is the most cost-effective option.

Fan Control Option	Annual Energy Use (kWh/yr)	Annual Energy Cost ($/yr)	Annual Cost Savings ($/yr)	Simple Payback Period (yrs)
CV	101,539	5,077	0	—
OD	92,690	4,635	442	7.9
VIV	71,832	3,592	1,485	5.4
VSD (act.)	52,394	2,620	2,457	3.7
VSD (th.)	40,587	2,029	3,048	3.0

7.6.3 Duct Leakage

Ducts are generally used to distribute conditioned air to various spaces within buildings. Ducts do not consume energy, however, they can be a significant source of air leakage and energy waste decreasing the overall performance of the HVAC system. Several field studies showed that up to 30 percent of energy input required for a typical HVAC system (including fan electrical power, heating or cooling energy) can be lost in the ducts.

Air leakage in ducts can be measured using a pressurization test similar to the blower door test and tracer gas technique used to estimate air infiltration through the building envelope as outlined in Chapter 6. Duct leakage can be estimated knowing the pressure in the ducts P_{in} and the pressure surrounding the ducts P_{out} using the following equation:

$$\dot{V}_{leak} = C(P_{in} - P_{out})^n \tag{7.18}$$

where C and n are correlation coefficients that are determined based on the pressurization test. If there are relatively large leaks (i.e., holes) in the ducts, the coefficient n is close to 0.5.

Field tests performed on residential and small commercial buildings by Florida Solar Energy Center (FSEC, 2004) showed that duct leaks represent about 10–20 percent of the total fan air flow on each side of the fan.

A significant fraction of heating or cooling energy can be lost through ducts. Indeed, uninsulated ducts experience a high rate of heat transfer when placed in unconditioned areas especially in extreme weather conditions. Studies by Lawrence Berkeley National Laboratory indicated that 20 to 30 percent of thermal energy can be lost from ducts in residential and commercial buildings (LBNL, 1997).

7.6.4 Damper Leakage

Dampers can be divided into two different types, opposed-blade and parallel-blade, as illustrated in Figure 7.9. The pressure drop across a damper is given by the loss coefficient C_d (a function of the blade angle):

$$C_d = \frac{\Delta P_t}{P_v} \tag{7.19}$$

where DP_t is the total pressure loss across the damper and P_v is the velocity pressure. The loss coefficient depends on the damper type. Figure 7.10 illustrates a typical variation of the loss coefficient C_d as a function of the damper position for both parallel-blade and opposed-blade dampers.

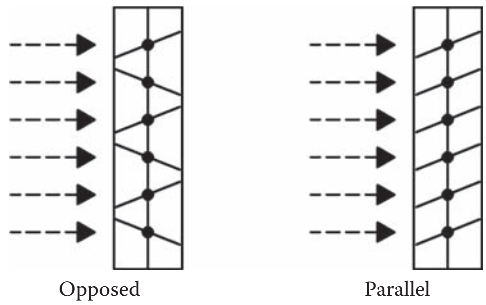


FIGURE 7.9 Types of dampers commonly used in air-handling units.

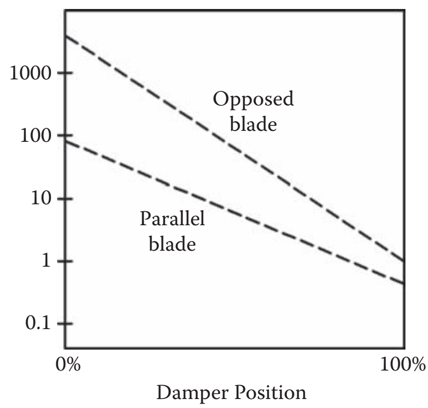


FIGURE 7.10 Typical variation of loss coefficient as a function of damper position.

Opposed blades are preferred for modulating service from noise and energy-loss standpoints. Typical leakage amounts are about 50 cfm/ft^2 at a pressure drop across the damper of DP = 1.5-in. wc. Outside air dampers, however, need to be much tighter to prevent excessive infiltration and coil freezing. Tight dampers can have leakage rates as low as 10 cfm/ft^2 at a pressure drop across the damper of DP = 4-in. wc.

7.6.5 Size Adjustment

A recent EPA study found that 60 percent of building fan systems were oversized by at least 10 percent. By reducing the size of these fans to the required capacity, it is estimated that average savings of 50 percent can be achieved in energy use of fan systems. Even more savings can be expected if the fan size adjustment is implemented with other measures related to fan systems such as energy-efficient motors, energy-efficient belts, and variable speed drives.

The size adjustment of fans can be implemented for both constant volume and variable air volume systems. In addition to energy savings, the benefits of using the proper fan size include:

- Better comfort: If the fan system is oversized, more air than needed may be supplied to the zones which may reduce the comfort of the occupants in addition to wasting energy.
- Longer equipment life: An oversized fan equipped with a variable speed drive operates at low capacities. This mode of operation can reduce the useful life of motors and other equipment.

To determine if a fan system is oversized, some in situ measurements can be made depending on the type of the HVAC system: constant volume or variable air volume.

For constant volume systems, the measurement of supply fan static pressure is generally sufficient to assess whether the fan is properly sized. To ensure that static pressure for the main supply fan is measured when the HVAC system is operating close to its design capacity, the testing should be done on a hot and humid day. Moreover, all dampers and fan vanes should be fully open during the tests. If the measured static pressure is larger than the design pressure which is typically provided in the building mechanical drawings (otherwise, design value should be determined), the fan is supplying too much air and is most probably oversized.

For variable-air-volume systems, three methods can be used to determine if the fan system is oversized including:

(a) Measurement of the electrical current drawn by the fan motor. If this current is lower than 75 percent of the nameplate full-load amperage rating (this value can be obtained directly on the motor's nameplate or from the operations and maintenance manual), then the fan is oversized.
(b) Checking the position of fan control vanes and dampers. If the vanes or dampers are closed more than 20 percent, the fan is oversized.
(c) Measurement of the static pressure for the main supply fan. If the measured static pressure is larger than the set-point, the fan system is oversized.

It should be noted that all the diagnostic methods need to be performed when the HVAC system is operating under peak load such as during a hot and humid day.

When it is clear that the fan system is oversized, the adjustment of its size can be achieved by one or a combination of these measures:

1. Installation of a larger pulley to reduce the speed of the existing fan. By reducing the speed of the fan, not only the flow of air moved through the duct is proportionally reduced, but also the energy use of the fan system is reduced significantly. For instance, a 20 percent reduction in the fan's speed will reduce its energy consumption by 50 percent.
2. Replacement of the existing oversized motor by a smaller energy-efficient motor that matches the peak load. The use of smaller motor will obviously reduce the energy use of the fan system. For instance, replacing a 50-kW standard motor with a 35-kW energy-efficient motor will reduce the energy use of the fan system by about one third (i.e., 33 percent).
3. Adjustment of the static pressure setting (for VAV systems, only). By reducing the static pressure set-point to a level sufficient to maintain indoor comfort, the energy use by the fan system will be reduced. For instance, a VAV system operating at a static pressure of 6 inches of water can be reduced in some cases to 4 inches with loss of thermal comfort. This 33 percent reduction of static pressure will achieve about 45 percent of energy savings in the fan system operation.

It is important to analyze the performance of the entire HVAC system with any of the measures described above to ensure effective adjustment of the fan size. Indeed, changes to the fan system can affect the operation or control of other components of the HVAC system.

7.7 Common HVAC Retrofit Measures

In this section, simplified analysis methods are provided to estimate the energy savings potential for retrofit measures related to the operation of HVAC systems.

7.7.1 Reduction of Outdoor Air Volume

This measure can be implemented by replacing or appropriately controlling the outside air dampers as depicted in Figure 7.11 using several approaches including:

(a) Use of low leakage dampers to reduce the amount of unwanted ventilation air when outdoor air dampers are closed. Standard dampers can allow up to 15 percent air leakage when closed.

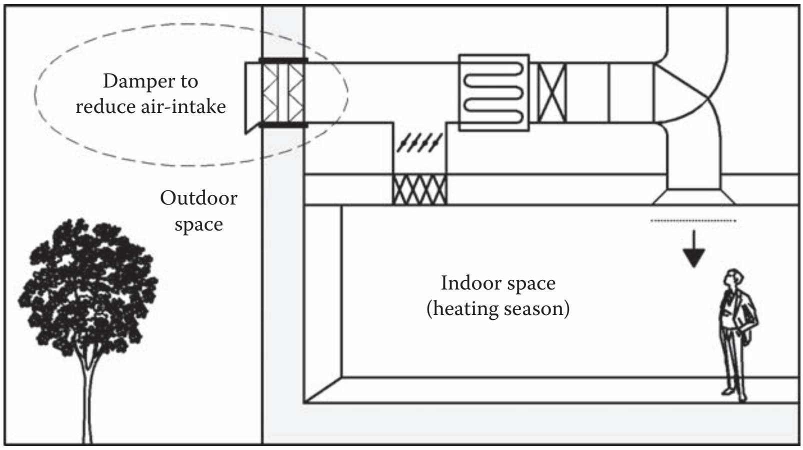


FIGURE 7.11 Outdoor air dampers can be a source of significant energy waste.

Low-leakage dampers restrict air leakage to less than 1 percent.

(b) Reduce the ventilation air requirements to the levels recommended by ASHRAE Standard 62. In addition, ventilation air requirements may be reduced by continuously monitoring indoor contaminant levels (for instance, CO_2 for offices and schools and CO for parking garages).

(c) Eliminate or reduce the ventilation air during unoccupied periods. Some HVAC systems are operated during unoccupied hours to maintain specified temperature set-points (even for systems operated with night temperature setback) or to start conditioning the space before occupancy (optimum start controls). In these cases, the ventilation requirements can be safely reduced as long as there is no occupant.

To estimate the energy savings obtained from a reduction of outdoor air volume, Eqs. (7.3) through (7.6) can be used. Example 7.3 illustrates the simplified calculation procedure for a retrofit of an outside air duct with low-leakage dampers.

Example 7.3

Estimate the fuel savings obtained by using low-leakage dampers for a 40,000 cfm air-handling unit which serves an office building located in Chicago, Illinois. The office is conditioned during the entire heating season to 72°F (even during unoccupied hours). It was found that the existing dampers have 15 percent leakage when closed. The outdoor air damper system is scheduled to be closed 3,500 hours during the heating season. For the analysis, assume that the cost of fuel is \$6/MMBtu and the boiler efficiency is 80 percent. The average heating season outdoor temperature in Chicago is 34.2°F. The low-leakage dampers have a 1 percent leakage.

Solution

The annual fuel savings is estimated based on the energy saved due to a reduction in the amount of outdoor air volume to be heated from $T_{out} = 34.2^o F$ to $T_{in} = 72^o F$:

$$\Delta FU_H = \frac{\Delta \dot{m}_a . c_p . (T_{out} - T_{in})}{\eta_b} . N_{h,H}$$

Therefore:

$$\Delta FU_H = \frac{40{,}000 cfm * (0.15 - 0.01) * (1.08 Btu/cfm.hr.^oF) * (72^oF - 34.2^oF)}{(0.80) * (10^6 Btu/MMBtu)} * 3{,}500 hrs/yr$$

or

$$\Delta FU_H = 1{,}000\, MMBtu/yr$$

In terms of cost savings, the use of low-leakage dampers can reduce fuel costs by $6,000/yr.

7.7.2 Reset Hot or Cold Deck Temperatures

In several HVAC systems with dual ducts, a fixed temperature set-point is used throughout the year for the hot deck or the cold deck. This operation strategy can waste significant heating and cooling energy by mixing air that can be too hot with air that can be too cold. Simple controls can be set to allow the temperature set-point for both the hot and cold decks to be changed according to the zone loads.

Again, Eqs. (7.3) through (7.6) can be used to estimate the energy savings for a hot or cold deck temperature setting. Example 7.4 illustrates the simplified calculation procedure for hot-deck temperature reset.

Example 7.4

Estimate the fuel savings obtained by resetting the hot deck of a 60,000 cfm dual duct system by 2°F during the winter and 1°F during the summer. The HVAC system is operated 80 hours per week and serves a building in Chicago, Illinois. For the analysis, assume that the cost of fuel is $6/MMBtu and the boiler efficiency is 80 percent. The average heating season outdoor temperature in Chicago is 34.2°F. It should be noted that for Chicago, the number of weeks is 30 for the heating season (winter) and 21 for the cooling season (summer).

Solution

The annual fuel savings is estimated based on the energy saved through a change in the temperature difference: $\Delta T_H = 2^oF$ in the winter and $\Delta T_H = 2^oF$ in the summer:

$$\Delta FU_H = \frac{\dot{m}_a . c_p . (\Delta T_H . N_{h,H} - \Delta T_C . N_{h,C})}{\eta_b}$$

Therefore,

$$\Delta FU_H = \frac{60{,}000 cfm * (1.08 Btu/cfm.hr.^oF) * [2^oF * (30 * 80 hrs) + 1^oF * (21 * 80 hrs)]}{(0.80) * (10^6 Btu/MMBtu)}$$

or

$$\Delta FU_H = 525\, MMBtu/yr$$

Thus, the hot deck temperature reset can reduce the fuel cost by $3,150/yr.

7.7.3 CV to VAV System Retrofit

The conversion of a constant-volume system to a variable-air-volume system can provide significant energy savings for three reasons: (i) reduction in fuel use for heating requirements, (ii) reduction in electrical energy use for cooling requirements, and (iii) reduction in fan electrical power use for air circulation.

Two types of constant volume systems are good candidates for VAV conversion. These systems are:

1. Single-duct reheat systems that provide a constant volume of cold air continuously to satisfy the peak cooling load requirements for critical zones. For other zones or under other operating conditions, the cold air is reheated using hot water or electrical terminal heaters. This reheat of cold air represents a waste of energy. The conversion from CV to VAV systems can be easily implemented for single-duct reheat systems by adding a variable-air-volume sensor/controller to the terminal units. Reheat may be retained in the case where heating and cooling occur simultaneously.
2. Dual-duct systems that have both hot and cold air streams that are continually mixed to meet the zone load. In peak heating or cooling conditions, mixing may not occur and only hot or cold air is provided to meet zone loads.

The energy analysis of CV to VAV retrofit measures requires detailed simulation programs. Simplified methods can be used but provide only an order of magnitude of the total energy savings. Example 7.5 presents a simplified analysis to estimate the energy and cost savings attributed to the conversion of a single duct reheat system to a VAV system. Example 7.2 compares the fan energy savings for various fan systems that can be used to deliver variable air flow volumes.

Example 7.5

Estimate the heating and cooling energy and cost savings incurred by converting a 20,000-cfm single duct reheat system to a VAV system in an office building located in Chicago, Illinois. For the CV system, the return air is measured to be 76°F and the cold duct air temperature is set to be 58°F. The HVAC system is operated 80 hours per week. For the analysis, consider only the sensible loads. The cost of fuel is \$5/MMBtu and the boiler efficiency is 80 percent. The cost of electricity is \$0.05/kWh and the chiller efficiency is 0.9 kW/ton. For Chicago, the number of weeks is 30 for the heating season (winter) and 21 for the cooling season (summer).

Solution

(a) First, fuel savings are determined. These fuel savings are the results of the heating load reduction incurred from the CV to VAV conversion of the 20,000-cfm air-handling unit. Assuming that the average heating load X is 50 percent over the entire year, the annual fuel savings are attributed to the energy saved through the reduction of the air volume (by a fraction of 1 – X). This air volume has to be heated from: the cold air temperature T_{return} = 8° to the return air temperature T_{return} = 76° Fduring the entire year (52 weeks):

$$\Delta FU_H = \frac{\dot{m}_a . c_p .(1-X)(T_{return} - T_{coldC}).N_{h,yr}}{\eta_b}$$

Therefore,

$$\Delta FU_H = \frac{20{,}000cfm * (1.08Btu/cfm.hr.^oF) * (1-0.5) * (76^oF - 58^oF) * (52 * 80hrs)]}{(0.80) * (10^6 Btu/MMBtu)}$$

(b) The electricity savings are incurred from the cooling load reduction. Again, assuming that the average cooling load X is 50 percent over the cooling season, the annual cooling energy savings are estimated based on the energy saved through the reduction of the supplied air volume (by a fraction of 1 – X). This air volume has to be cooled from: the return air temperature $T_{return} = 76^\circ F$ to the cold air temperature $T_{cold} = 58^\circ F$ during the cooling season (21 weeks):

$$\Delta kWh_C = \frac{\dot{m}_a.c_p.(1-X)(T_{return} - T_{coldC}).N_{h,C}}{12{,}000\,Btu/ton.hr}.kW/ton$$

Thus:

$$\Delta kWh_C = \frac{20{,}000cfm * (1.08Btu/cfm.hr.^oF) * (1-0.5) * (76^oF - 58^oF) * (21 * 80hrs)]}{12{,}000\ \ Btu/ton.hr)} * 0.90kW/ton$$

or

$$\Delta kWh_C = 24{,}494\,kWh/yr$$

The electricity cost savings amounts to \$1,225/yr. Therefore, savings of \$6,279 are achieved for both heating and cooling energy costs as a result of the CV to VAV conversion. Additional savings can be obtained from the reduction in fan power as illustrated in Example 7.2.

7.8 Summary

In commercial buildings, significant energy savings can be obtained by improving the energy efficiency of secondary HVAC systems. These improvements can be achieved with simple operating and maintenance measures with little or no investment. Better operation and control of HVAC systems provide not only energy and cost savings but also improved thermal comfort. Finally, and in most cases, conversion of constant-volume systems to variable-air-volume systems is cost-effective and should be considered for existing commercial and institutional buildings.

Problems

7.1 A 50-hp constant volume fan is to be retrofitted to a variable volume fan. Three options are considered:

OPTION A: Install outlet dampers at a cost of \$3,100.
OPTION B: Use inlet vanes fan at a cost of \$6,950.
OPTION C: Use variable speed drive at a cost of \$8,500.

Percent Load	Percent of Occurrence Load Profile 1	Percent of Occurrence Load Profile 2
100	5	20
90	5	30
80	10	25
70	15	20
60	25	5
50	20	0
40	10	0
30	5	0
20	5	0

The fan is operated 5,500 hours per year. If the cost of electricity is \$0.08/kWh and the discount rate is 6 percent, determine for the following load profiles, the best option for retrofitting the fan.

Use both a simple payback analysis and LCC analysis (assume a 15-year life cycle). For the performance of various fan control strategies, you can use and interpolate the values given in Table 7.5. Typically, at a 20 percent load, the fan uses 59 percent of power for outlet dampers and 41 percent of power for inlet vanes. Conclude.

7.2 It is proposed to retrofit the supply fan of a 60,000 cfm air-handling-unit. The fan has a 25-hp motor with a total efficiency of 85.5 percent. If the fan is operating 5,500 hrs per year and if the cost of electricity is \$0.08/kWh, determine the simple payback period when the constant volume fan is retrofitted to:

(a) A temperature measurement indicated that when the outdoor air is at 50°F, the mixed air is at 70°F and the return air is at 75°F. Determine the heating fuel savings by reducing the outdoor air to the minimum ventilation requirements (i.e., 20 cfm/person). Outdoor air is provided only during the occupied period.

(b) Determine the heating energy and cost savings due to a temperate setback to 60°F during the unoccupied period.

(c) If the system is converted into a VAV system, determine the energy and cost savings due to the fan. The average load on the fan is 60 percent of the peak.

7.3 Consider a 200,000 sqft enclosed parking garage that can hold 1,000 cars.

(a) Determine the size of the supply/exhaust fans to ventilate this garage if 1.5 cfm/sq.ft are required. Assume a fan pressure of 1.5 in. of water and a fan efficiency of 0.75.

(b) Using the design method, determine the needed cfm/sqft to ventilate the garage if at peak hour, 30 percent of the cars are moving (assume the worst-case scenario for emissions: winter and cold start). Assume that the acceptable CO level to maintain is 25 ppm. Size the supply/exhaust fans. Discuss the energy saving if the garage is operated 12 hours/day, 7days/week, and 50 weeks/year. Is it worth it to retrofit the ventilation system if the energy price is \$0.08/kWh (the cost of fans and their drives can be estimated as the cost of motors multiplied by 2.5)?

(c) Discuss any other means to reduce the energy cost in ventilating the garage.

7.4 Consider a 80,000-sqft two story building. A 60,000 cfm air-handling unit services the building. The building is occupied 50 hrs per week. The fuel cost is \$5.00 per 10^6 Btus and electricity cost is \$0.09 per kWh. Determine the energy and cost savings for the following ECOs for three locations: Denver, Colorado; Chicago, Illinois; Miami, Florida:

(a) Reset the hot deck temperature: 3°F in summer and 4°F in winter. Assume 50 percent of airflow is in the hot deck.
(b) Reset the cold deck temperature: 2.5 Btu/lb if enthalpy. Assume 50 percent of airflow is in the cold deck.
(c) Reduce minimum outdoor air: from 25 percent to the minimum required. Assume 20 cfm/person and 300 people in the building. Average indoor temperature is 70°F.
(d) Low leakage dampers: reduce damper leakage form 10 percent to 1 percent.
(e) Install a dry-bulb temperature economizer or an enthalpy economizer.

7.5 An 80,000 sq.ft office building located in Chicago has a dual-duct air-conditioning system with a 60,000 cfm air-handling capacity (60 hp). Determine the annual energy and cost savings that could be achieved with a VAV conversion. The following data apply:

Mixed air temp.	65°F	80°F
Diffuser discharge temp.	60°F	67°F
Room temp.	70°F	74°F
Cold deck temp.	57°F	61°F
Hot deck temp.	110°F	95°F
Operating hours	72 hrs/wk	72 hrs/wk
Length of season	30wks	22wks
Cost of electricity	\$0.06/kWh	\$0.09/kWh
Cost of fuel	\$5/MMBtu	\$5/MMBtu

7.6 Consider a building in Denver, Colorado with 70,000 cfm single-duct CV air-handling unit. The building is occupied 60 hours per week with 500 people. The indoor temperature is always maintained at 73°F. Fuel cost is \$4.50 per 10^6 Btu. Electricity cost is \$0.09 per kWh. The fan pressure differential inside the AHU is 4.0 in. The fan has efficiency of 0.78 and is operating 80 hours per week. Determine the cost savings incurred by converting this CV to a VAV system.

7.7 Consider a building in Denver with 90,000 cfm single-duct CV air-handling unit. The building is occupied 60 hours per week with 600 people. The indoor temperature is always maintained at 72°F. Fuel cost is \$4.50 per MMBtu. Electricity cost is \$0.10 per kWh. The fan pressure differential inside the AHU is 5.0 in. The fan is operating 80 hours per week.
(a) A temperature measurement indicated that when the outdoor air is at 50°F, the mixed air is at 70°F, and the return air is at 75°F. Determine the heating fuel savings by reducing the outdoor air to the minimum ventilation requirements (i.e., 20 cfm/person). Outdoor air is provided only during the occupied period.
(b) If the system is converted into a VAV system, determine the energy and cost savings due to the fan. The average load on the fan is 60 percent of the peak.

For Denver, the average outdoor DB temperatures are 35.2°F for winter and 77.9°F for summer. The length of the heating season is 29.4 weeks whereas the length of cooling season is 22.6 weeks.

8 Central Heating Systems

8.1 Introduction

According to a survey reported by the U.S. Energy Information Administration (EIA, 2006), four types of heating systems are used extensively in commercial buildings including:

1. Boilers
2. Packaged heating units
3. Individual space heaters
4. Furnaces

As indicated in Table 8.1, boilers provide heating to almost 33 percent of the total heated floor-space of U.S. commercial buildings. However, boilers are used in only 15 percent of heated commercial buildings, significantly less than furnaces, which are used in more than 42 percent of U.S. commercial buildings. This difference in usage stems from the fact that boilers are more commonly used in larger buildings whereas furnaces are the heating system of choice for smaller buildings. The same EIA survey (EIA, 2006) indicated that small buildings [less than 500 m^2 (5,000 ft^2)] constitute more than half (52 percent) of the total U.S. commercial building stock.

Of the existing boilers used in U.S. commercial buildings, 65 percent are gas-fired, 28 percent are oil-fired, and only 7 percent are electric. The average combustion efficiency of the existing boilers is in the range of 65 to 75 percent. New energy-efficient gas or oil fired boilers can be in the range of 85 to 95 percent.

8.2 Basic Combustion Principles

8.2.1 Fuel Types

Fuels used in boilers consist of hydrocarbons including alkynes (C_nH_{2n-2}) such as acetylene (n = 2), alkenes (C_nH_{2n}) such as the ethylene (n = 2), and alkanes (C_nH_{2n+2}) such as octane (n = 8). A typical combustion reaction involves an atom of carbon with two atoms of oxygen with a generation of heat according to the following generic combustion reaction:

$$C + O_2 \rightarrow CO_2 + Heat \tag{8.1}$$

The heat generated in the combustion reaction is referred to as the heating value HV of a fuel. Typically, the heating value is given when the fuel is dry. The moisture actually reduces the heating value of fuels according to the following simplified equation:

$$HV = HV_{dry}.(1 - M) \tag{8.2}$$

TABLE 8.1 Heating Equipment Used for Main and Other Uses in U.S. Commercial Buildings in 2005

Heating System Type	Percentage of Heated Floor-Space (%)	Percentage of Heated Buildings (%)
Boilers	33	15
Package heating units	31	26
Individual space heaters	30	29
Furnaces	26	42
District heating	13	3
Heat pumps	11	10
Other	12	4

Source: EIA, www.doe.eia.gov, 2006

TABLE 8.2 Results of Proximate Analysis of Coal Extracted from Two Sites in the United States

Coal Type	Moisture	Volatile Matter	Fixed Carbon	Ash	Sulfur
Lackawanna, PA	2.0	6.3	79.7	12	0.6
Weld, CO	24.0	30.2	40.8	5	0.3

TABLE 8.3 Results of Ultimate Analysis of Coal Extracted from Two Sites in the United States

Coal Type	Carbon	Hydrogen	Oxygen	Nitrogen	Heating Value
Lackawanna, PA	93.5	2.6	2.3	0.9	13,000
Weld, CO	75.0	5.1	17.9	1.5	9,200

where M is the moisture content of the fuel.

In addition, the heating value of fuels decreases with altitude. As a rule of thumb, the heating value reduces by 4 percent for every 300 m (1,000 ft) increase in altitude.

Two analyses are typically used to determine the basic components of a fuel. The first analysis is called proximate analysis and determines the fuel content in percentage by weight of moisture, the volatile matter, fixed carbon, ash, and sulfur. The second analysis is referred to as the ultimate analysis and determines the fuel content in percentage by weight of carbon, hydrogen, nitrogen, and oxygen. It should be noted that the heating value of a fuel increases with its carbon content. Tables 8.2 and 8.3 provide the results of proximate and ultimate analyses, respectively, for coal extracted from two sites in the United States.

Table 8.4 illustrates the results of the ultimate analysis of another solid fuel: wood. It is clear that pine has a higher carbon content (by weight) and thus higher heating value.

Liquid or distillate fuels are generally graded in different categories depending on their properties. For fuel oils, there are six different grades depending on the viscosity level. Table 8.5 provides the heating values and common usage of five fuel oils commonly sold in the United States. Fuel oil No. 3 has now been incorporated as part of fuel oil No. 2.

Similar grades are used for diesel fuels with diesel No. 1 used for high-speed engines and diesel No. 2 used for industrial applications and heavy cars.

Liquid petroleum gas (LPG) is a mixture of propane and butane and natural gas is a mixture of methane and ethane.

TABLE 8.4 Results of Ultimate Analysis of Selected Wood Types

Wood Type	Carbon	Hydrogen	Oxygen	Nitrogen	Heating Value
Oak	49.5	6.6	43.7	0.2	7,980
Pine	59.0	7.2	32.7	1.1	10,400
Ash	49.7	6.9	43.0	0.3	8,200

TABLE 8.5 Heating Value and Specific Gravity of Oil Fuels Used in the United States

Oil Grade	Specific Gravity	Heating Value KWh/L (MBtu/gal)	Applications
No. 1	0.805	9.7 (134)	For vaporizing pot-type burners
No. 2	0.850	10.4 (139)	For general purpose domestic heating
No. 3	0.903	10.9 (145)	For burners without preheating
No. 5	0.933	11.1 (148)	Requires preheating to 75–95°C
No. 6	0.965	11.3 (151)	Requires preheating to 95–115°C

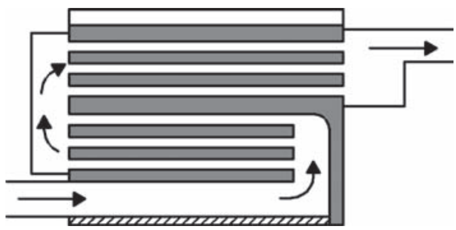


FIGURE 8.1 Typical configuration for fire-tube boilers. (Courtesy of ASHRAE, 2009.)

8.2.2 Boiler Configurations and Components

Typically, boilers have several parts including an insulated jacket; a burner; a mechanical draft system; tubes and chambers for combustion gas, water, or steam circulation; and controls.

There are several factors that influence the design of boilers including fuel characteristics, firing method, steam pressure, and heating capacity. However, commercial and industrial boilers can be divided into two basic groups—fire-tube or water-tube—depending on the relative location of the hot combustion gases and the fluid being heated within the boiler. In the following sections, brief descriptions of common boiler and burner types are provided.

8.2.2.1 Boiler Types

Most commercial boilers are manufactured of steel. Some smaller-size boilers are made up of cast iron. The steel boilers transfer combustion heat to the fluid using an assembly of tubes that can be either water-tubes or fire-tubes.

8.2.2.1.1 Fire-Tube Boilers

In these boilers, the hot combustion products flow through tubes submerged in the boiler water as depicted in Figure 8.1. To increase the contact surface area between the hot gases and the water, two to four passes are used for the tubes. The multipass tubes increase the efficiency of the boiler but require greater fan power. Due to economics, the highest capacity of fire-tube boilers is currently in the 10,000 kg of steam per hour with an operating pressure of 16 atm (250 psi).

The fire-tube boilers are generally simple to install and maintain. Moreover, they have the ability to meet sudden and wide load fluctuations with only small pressure changes.

8.2.2.1.2 Water-Tube Boilers

In these boilers, the water flows inside tubes surrounded by flue combustion gases as shown in Figure 8.2. The water flow is generally maintained by the density variation between cold feed water and the hot water/steam mixture in the riser. The water-tube boilers are classified in several groups depending on the shape and drum location, capacity, and number. The size of water-tube boilers can be as small as 400 kg of steam per hour and as large as 1,000 MW units. The largest industrial boilers are generally about 250,000 kg of steam per hour.

8.2.2.1.3 Cast Iron Boilers:

These boilers are used in small installations (below 1 MW) where long service life is important. These boilers are made up of precast sections and thus are more readily field assembled than steel boilers. At similar capacities, the cast-iron boilers are typically more expensive than fire-tube or water-tube boilers. Figure 8.3 illustrates selected types of cast iron boilers.

8.2.2.2 Firing Systems

The firing system of a boiler depends on the fuel used. The characteristics of the firing system for each fuel type are summarized below.

8.2.2.2.1 Gas-Fired Units:

Natural gas is the simplest fuel to burn because it mixes easily with the combustion air supply. Gas is generally introduced at the burner through several orifices that provide jets of natural gas that mix rapidly with the combustion air supply. There is a wide range of burner designs depending on the orientation, the number, and the location of the orifices.

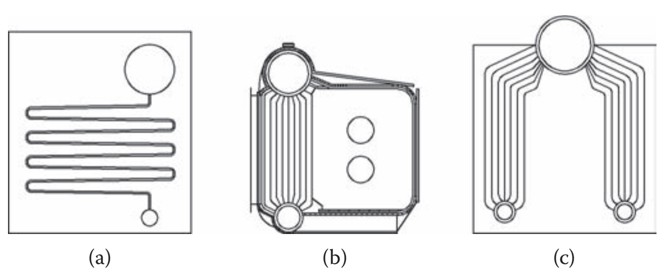


FIGURE 8.2 Common types of water-tube boilers. (Courtesy of ASHRAE, 2009.)

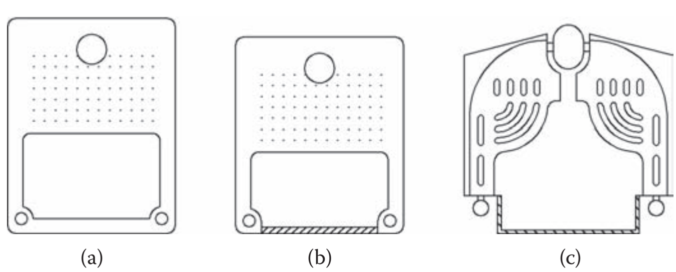


FIGURE 8.3 Selected types of cast iron boilers. (Courtesy of ASHRAE, 2009.)

As part of the routine tune-up and maintenance of gas-fired boilers, it is important to inspect gas injection orifices to check that all the passages are unobstructed. In addition, it is important to identify and replace any burned-off or missing burner parts.

8.2.2.2.2 Oil-Fired Units

Oil fuels need some form of preparation and treatment before final delivery to the burner. The preparation of the oil fuels may include the following:

- Use strainers and filters to clean the oil fuel and remove any deposit or solid foreign material.
- Add flow line preheaters to deliver the fuel oil with a proper viscosity.
- Use atomizers to deliver the fuel oil in small droplets before mixing with the combustion air supply. The atomization of the fuel oil can be carried out by a gun fitted with a tip that has several orifices that can produce a fine spray. Moreover, oil cups that spin the oil into a fine mist are also used on small boiler units.

During tune-up of central heating systems, it is important to check that the burner is adequate for the boiler unit. In particular, it is important to verify that the atomizer has the proper design, size, and location. In addition, the oil-tip orifices should be cleaned and inspected for any damage to ensure a proper oil-spray pattern.

8.2.2.2.3 Coal-Fired Units

In some central heating systems, coal can be used as the primary fuel to fire the burner. There are two main coal-firing systems:

(i) Pulverized coal-fired systems that pulverize, dry, classify, and transport the coal to the burner's incoming air supply. The pulverized coal-fired systems are generally considered to be economical for units with large capacities (more than 100,000 kg of steam per hour).
(ii) Coal stoker units that have a bed on the boiler grate through which the combustion air is supplied. There are currently several stoker-firing methods used in industrial applications such as underfed, overfed, and spreader. Both underfed and overfed firing methods require that the coal be transported directly to the bed combustion and usually respond slowly to sudden load variations. The spreader stokers partially burn the coal in suspension before transporting it to the grate. The spreader stokers can burn a wide range of fuels including waste products.

The efficiency of coal-firing systems depends on the firing system, the type of boiler or furnace, and the ash characteristics of the coal. Some units are equipped with ash reinjection systems that allow collected ash that contains some unburned carbon to be redelivered into the burner.

8.2.3 Boiler Thermal Efficiency

As illustrated by Eq. (8.1), fuel combustion involves a chemical reaction of carbon and oxygen atoms to produce heat. The oxygen comes from the air supplied to the burner that fires the boiler. A specific amount of air is needed to ideally complete the combustion of the fuel. This amount of air is typically referred to as the stoichiometric air. However, in actual combustion reactions, more air than the ideal (or stoichiometric) amount is needed to totally complete the combustion of fuel. The main challenge to ensure optimal operating conditions for boilers is to provide the proper excess air for the fuel combustion. It is generally agreed that 10 percent excess air provides the optimum air-to-fuel ratio for complete combustion. Too much excess air causes higher stack losses and requires more fuel to heat ambient air to stack temperatures. On the other hand, if insufficient air is supplied, incomplete combustion occurs and the flame temperature is reduced.

The general definition of overall boiler thermal efficiency is the ratio of the heat ouput E_{out} over the heat input E_{in}:

$$\eta_b = \frac{E_{out}}{E_{in}} \tag{8.3}$$

The overall efficiency accounts for combustion efficiency, the stack heat loss, and the heat losses from the outside surfaces of the boiler. The combustion efficiency refers to the effectiveness of the burner in providing the optimum fuel-to-air ratio for complete fuel combustion.

To determine the overall boiler thermal efficiency, some measurements are required. The most common test used for boilers is the flue gas analysis using an Orsat apparatus to determine the percentage by volume of the amount of CO_2, CO, O_2, and N_2 in the combustion gas leaving the stack. Based on the flue gas composition and temperature, some adjustments can be made to tune up the boiler and to determine the best air-to-fuel ratio in order to improve the boiler efficiency. The following general rules of thumb can be used to adjust the operation of the boiler:

- *Stack temperature:* The lower the stack temperature, the more efficient is the combustion. High flue gas temperatures indicate that there is no good heat transfer between the hot combustion gas and the water. The tubes and the chambers within the boiler should be cleaned to remove any soot, deposit, and fouling that may reduce the heat transfer. However, the stack temperature should not be too low to avoid water condensation along the stack. The water from the condensation mixes with sulfur and can cause corrosion of the stack. Table 8.6 provides the minimum stack exit temperature for common fuel types to avoid corrosion problems.
- *CO_2 level:* The higher the CO_2 level, the more efficient is the combustion. The low limits acceptable for the CO_2 level are 10 percent for gas-fired boilers and 14 percent for oil-fired boilers. If the CO_2 levels are lower than these limits, the combustion is most likely incomplete. The air-to-fuel ratio should be adjusted to provide more excess air.
- *CO level:* No CO should be present in the flue gas. Indeed, any presence of CO indicates that the combustion reaction is incomplete and thus that there is not enough excess air. The presence of CO in the flue gas can be detected by the presence of smoke which leads to soot deposit in the boiler tubes and chambers.
- *O_2 level*: The lower the O_2 level, the more efficient is the combustion. Indeed, a high level of O_2 is an indication of too much excess air. The high limit acceptable for the O_2 level is 10 percent. When O_2 levels greater than 10 percent are found, the excess air should be reduced.

When the excess air is not adequate, the following boiler adjustment procedure can be used:

1. Operate the boiler for a specific firing rate and put the combustion controls on manual.
2. After stable operation, take a complete set of measurements (decomposition and temperature of the stack flue gas).
3. Increase the excess air by 1 to 2 percent and take a new set of measurements (after reaching stable boiler operating conditions).

TABLE 8.6 Minimum Exit Flue Gas Temperatures to Avoid Stack Corrosion

Fuel Type Used by the Boiler	Temperature Limit (°C)
Fuel oil	200
Bituminous coal	150
Natural gas	105

4. Decrease the excess air by small steps until a minimum excess O_2 condition is reached (i.e., when the combustion becomes incomplete and a noticeable CO level—above 400 ppm—can be detected in the flue gas). Take measurements following each change (allow the boiler to reach stable operating conditions).
5. Plot the measured data to determine the variation of the CO level as a function of the percentage of O_2 in the flue gas. A margin of excess O_2 above the minimum value can be established. Typically, a margin ranging from 0.5 to 2 percent O_2 above the minimum value is used.
6. Reset the burner controls to maintain the excess O_2 within the margin established in Step 5.
7. Repeat Steps 1 through 6 for various firing rates to be considered in the operation of the boiler. It is recommended that the tests be performed from higher to lower firing rates.

The new operating controls should be closely monitored for a sufficient length of time (one to two months) to ensure proper operation of the boiler.

Monographs are available to determine the overall boiler efficiency based on measurement of flue gas composition and temperature. One of these monographs applies to both gas-fired and oil-fired boilers and is reproduced in Figure 8.4 (for IP units) and Figure 8.5 (for SI units) and Example 8.1 illustrates how the monograph can be used to determine the boiler efficiency.

Example 8.1

A flue gas analysis of an oil fuel-fired boiler indicates that the CO_2 content is 11 percent with a gas flue temperature of 343°C (650°F). Determine the overall thermal efficiency of the boiler. Use oil No. 2.

Solution

By reading the monograph of Figure 8.4, the combustion occurs with excess air of 38 percent and excess O_2 of 6 percent. The overall boiler thermal efficiency is about 78 percent.

8.3 Boiler Efficiency Improvements

There are several measures by which the boiler efficiency of an existing heating plant can be improved. Among these measures:

- Tune up the existing boiler.
- Replace the existing boiler with a high-efficiency boiler.
- Use modular boilers.

The net effect of all these measures is some savings in fuel use by the heating plant. To calculate the savings in fuel use ΔFU related to the change in the boiler efficiency, the following equation can be used:

$$\Delta FU = \frac{\eta_{eff} - \eta_{std}}{\eta_{eff}} . FU_{std} \tag{8.4}$$

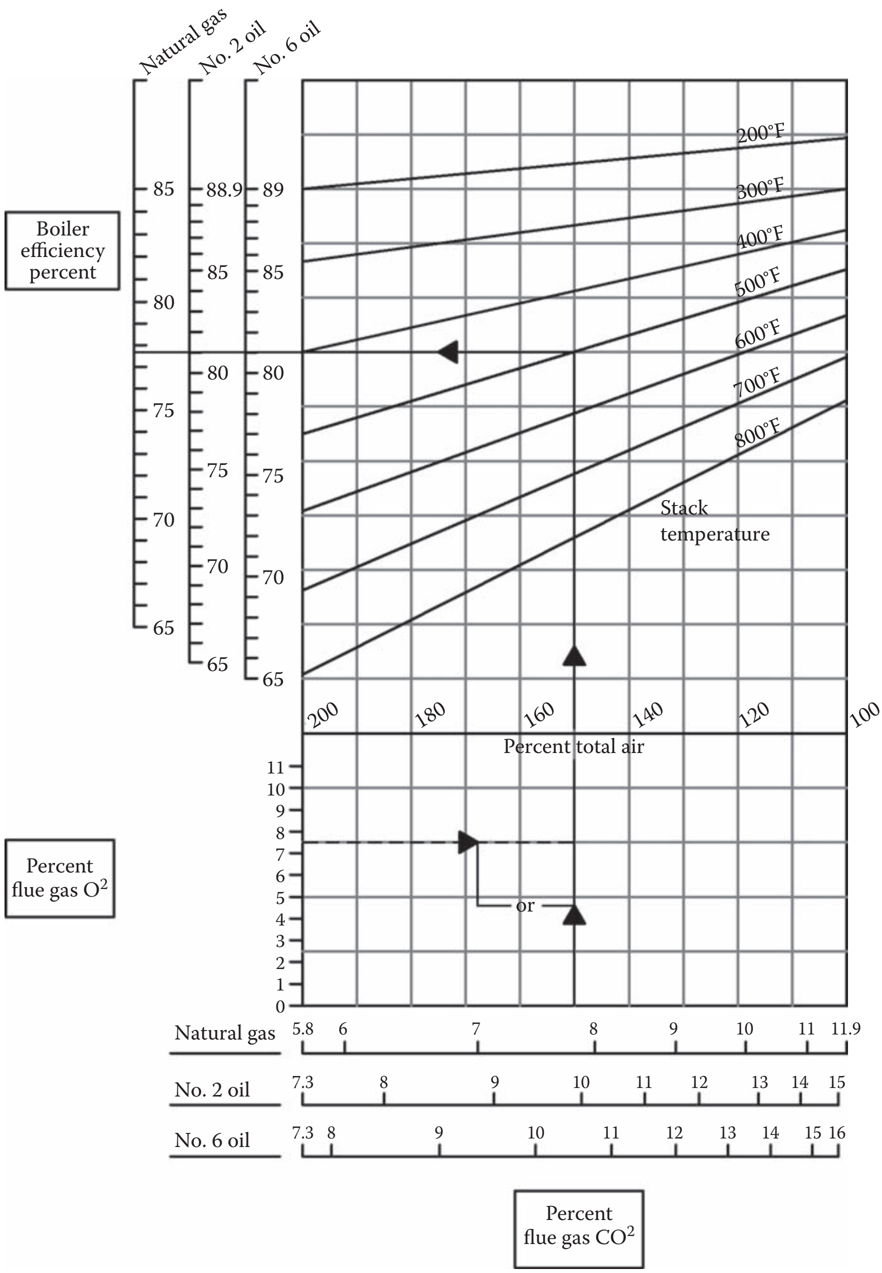


FIGURE 8.4 Monograph for boiler efficiency estimation using IP units.

where

η_{std}, η_{eff} are, respectively, the old and new efficiency of the boiler.

FU_{std} is the fuel consumption before any retrofit of the boiler system.

It is therefore important to obtain both the old and the new overall thermal efficiency of the boiler to estimate the energy savings. The following sections provide a more detailed description of the various boiler improvement measures.

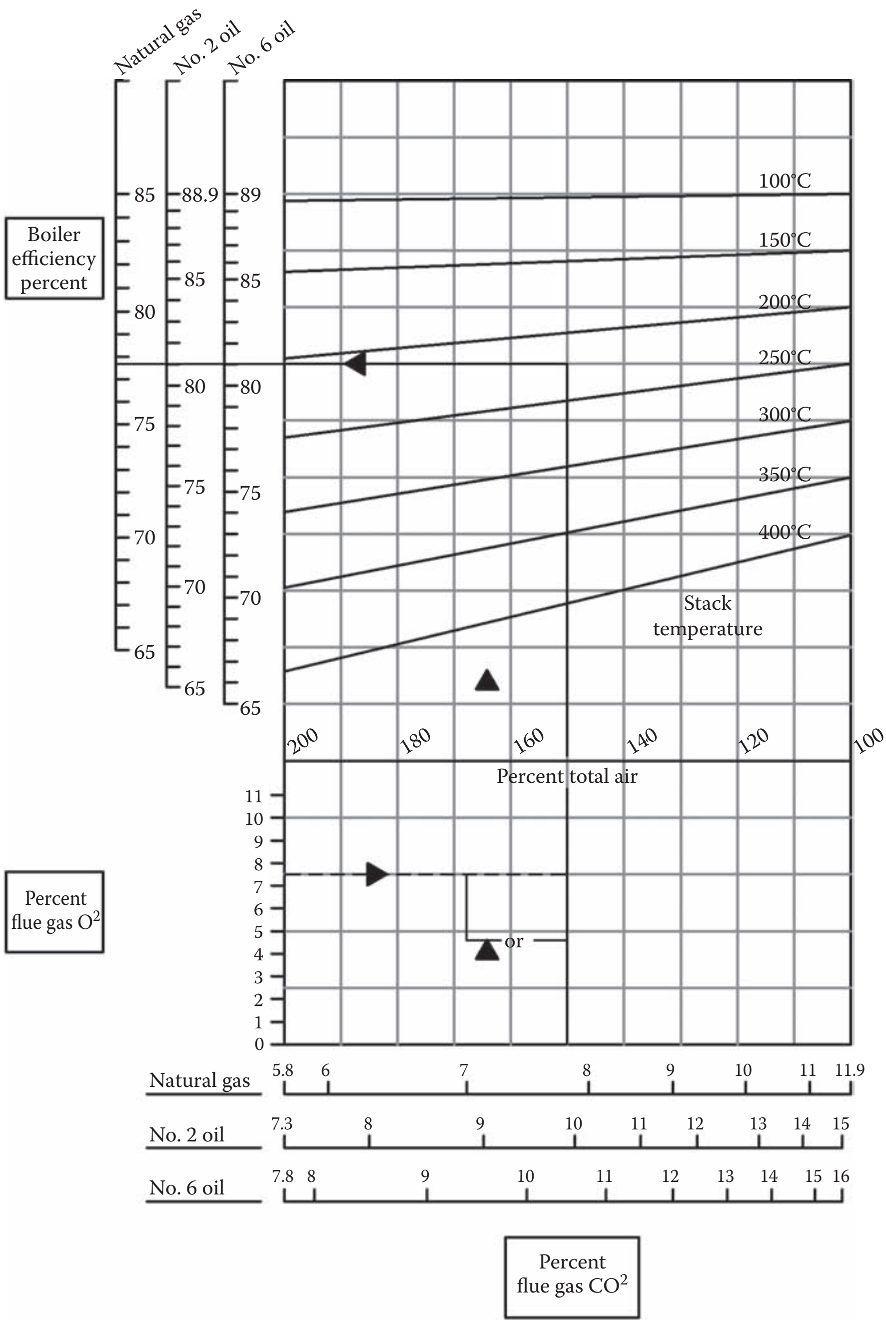


FIGURE 8.5 Monograph for boiler efficiency estimation using SI units.

8.3.1 Existing Boiler Tune-Up

By analyzing the flue gas composition and temperature, the boiler thermal efficiency can be estimated using the monograph provided in Fig. 8.4. If it were found that the efficiency was low due to inappropriate excess air, the boiler could be adjusted and its efficiency improved as described in the step-by-step procedure in the previous section. To perform this adjustment, some instrumentation is needed (gas flue analyzer and a temperature measurement device). Example 8.2 illustrates how the cost-effectiveness of a boiler tune-up can be evaluated.

Example 8.2

The boiler of Example 8.1 uses 1,500,000 L of fuel oil per year. An instrumentation kit is purchased at a cost of $20,000 and is used to adjust the boiler operation so its excess of O_2 is only 3 percent. Determine the payback of the instrumentation if the cost of fuel oil is $0.20/L.

Solution

From Example 8.1, the existing boiler has excess O_2 of 6 percent and an overall boiler thermal efficiency of about 78 percent (i.e., $\eta_{std} = 0.78$).

After the boiler tune-up, the excess O_2 is 3 percent. Using the monograph of Figure 8.1, the new boiler efficiency can be determined from the flue gas temperature (343°C or 650°F) and the excess O_2 (3 percent). It is found to be $\eta_{eff} = 84\% = 0.84$. Using Eq. (8.4), the fuel savings can be calculated:

$$\Delta FU = \frac{0.84 - 0.78}{0.84}.1{,}500{,}000 = 107{,}140\, L/yr$$

Therefore, the simple payback period for the instrumentation is:

$$SBP = \frac{\$20{,}000}{10{,}714 L/yr * \$0.20/L}. \approx 1.0\ year$$

Other measures that can be considered to increase the overall efficiency of the boiler are summarized below:

- Install turbulators in the fire-tubes to create more turbulence and thus increase the heat transfer between the hot combustion gas and the water. The improvement in boiler efficiency can be determined by measuring the stack flue gas temperature. The stack gas temperature should decrease when the turbulators are installed. As a rule of thumb, a 2.5 percent increase in the boiler efficiency is expected for each 50°C decrease in the stack flue gas temperature.
- Insulate the jacket of the boiler to reduce heat losses. The improvement of the boiler efficiency depends on the surface temperature.
- Install soot-blowers to remove boiler tube deposits that reduce heat transfer between the hot combustion gas and the water. The improvement in the boiler efficiency depends on the flue gas temperature.
- Use economizers to transfer energy from stack flue gases to incoming feedwater. The stack temperature should not be lowered below the limits provided in Table 8.6 to avoid corrosion problems. As a rule of thumb, a 1 percent increase in boiler efficiency is expected for each 5°C increase in the feedwater temperature.
- Use of air preheaters to transfer energy from stack flue gases to combustion air. Again, the stack temperature should not be lower than the values provided in Table 8.6.

The stack flue gas heat recovery equipment (i.e., air preheaters and economizers) are typically the most cost-effective auxiliary equipment that can be added to improve the overall thermal efficiency of the boiler system.

8.3.2 High-Efficiency Boilers

Manufacturers continue to improve both the combustion and the overall efficiency of boilers. Currently, commercially sized units can achieve over 95 percent combustion efficiency. For conventional boilers, anything over 85 percent is traditionally considered efficient. One of the most innovative combustion technologies currently available on the market is the gas-fired pulse-combustion boiler. This technology was first introduced in the early 1980s for residential water heaters, and is now available in several commercial-size boilers for both space heating and hot water heating.

Pulse-combustion boilers operate essentially as do automotive internal combustion engines. First, air and gas are introduced in a sealed combustion chamber in carefully measured amounts. This gas/air mixture is then ignited by a spark plug. Almost all the heat from the combustion is used to heat the water in the boiler. Indeed, the exhaust gases have only a relatively low temperature of about 50°C. Once the combustion chamber is fully heated, successive air–fuel mixtures or "pulses" ignite spontaneously (without the need for an electrical spark). Thus, no fuel-consuming burner or standing pilot light is required. When pulse boilers operate, they extract latent heat from the products of combustion by condensing the flue gas. Therefore, the boiler efficiency is increased and the flue gas is left with low water vapor content. The corrosion problems at the stack are then avoided.

The combustion efficiency of the pulse-combustion boilers can reach 95 to 99 percent. When combined with other high-performance elements for heat transfer, the overall thermal efficiency of the combustion-pulse boilers can attain 90 percent. In addition to saving energy, the pulse-combustion boilers can reach operating temperatures in as little as one-half the time of conventional boilers. Moreover, pulse-combustion burners produce lower emissions than conventional gas burners.

8.3.3 Modular Boilers

Almost all the heating systems are most efficient when they operate at full capacity. Improvements in peak-load efficiency result in lower energy use. However, the reduction in fuel use is not necessarily proportional to improvement in the heating system efficiency. Indeed, peak loads occur rarely in most heating installations. Therefore, the boiler is most often operating under part-load conditions. Some boilers may be forced to operate in an on/off cycling mode. This on/off cycling is an inefficient mode of operating the boiler. Indeed, the boiler loses heat through the flue and to the ambient space when it cycles off. Moreover, the water in the distribution pipes cools down. All these losses have to be made up when the boiler restarts. If the boiler capacity is much higher than the load, the cycling can be frequent and the losses can increase, thus significantly reducing the seasonal efficiency of the heating system.

Instead of operating the boiler in an on/off mode when the load is lower than its capacity, controls using step-firing rates (high/low/off) or modulating firing rates (from 100 percent to 15 percent) can be specified. Another effective measure to avoid cycling the boilers is to install a group of smaller boilers or modular boilers. In a modular heating plant, one boiler is first operated to meet small heating loads. Then, as the heating load increases, new boilers are fired and enter online to increase the capacity of the heating system gradually. Similarly, as the heating load decreases, the boilers are taken offline one by one.

Some manufacturers offer preassembled modular boiler packages of various sizes, ranging from approximately 50 kW to 1 MW. However, individual units can be piped and wired together in the field to form an efficient modular heating plant system. In addition to energy savings, modular boilers allow more flexibility in the use of space because they can be transported through doors that cannot accommodate a large boiler. Thus, a modular boiler can be located in confined spaces.

Modular boiler plants are suitable for applications with widely varying heating, steam, and hot water loads, such as hotels, schools, or high-rise buildings. The modular boilers can increase the overall seasonal efficiency of the heating system by 15 to 30 percent. For instance, a 5,000 m^2 shopping mall in Iowa with more than 16 stores and a food service area was retrofitted with 12 modular boilers (each with

40-kW capacity). According to the system manufacturer, the heating cost savings is about 33 percent relative to a conventional gas-fueled boiler (Tuluca, 1997).

8.4 Summary

Basic types of central heating systems are discussed in this chapter. In addition, cost-effective measures to improve the energy efficiency of boilers are presented. Simplified energy analysis methods are illustrated using examples of calculations to estimate energy savings incurred from higher efficiency central heating plants. As a rule of thumb, the auditor should consider simple operating and maintenance measures to increase the energy efficiency of existing heating systems before recommending new, more efficient, boilers.

Problems

8.1 A recent analysis of your gas-fired boiler showed that you have 30 percent excess combustion air. Discussion with the local gas company has revealed that you could use 10 percent excess air if the burner's controls were better adjusted. This represents calculated efficiency improvements of 8 percent.

(a) What would be the change in the flue gas temperature (due to better controls) if the current temperature is measured to be 600°F?

(b) How large an annual gas bill is needed before adding a maintenance person for the boiler alone is justified if this person would cost $35,000/yr?

8.2 A crude but effective approach to estimate steam leak (in lb/hr) from an orifice is to use the following expression:

$$lb/hr = C_d.k.\Delta t.A.P_i^{0.97}$$

where

C_d = the coefficient of discharge (for perfectly round orifice C_d = 1. In most cases, C_d = 0.7).

k = a constant (k = 0.0165).

Δt = the number of seconds in one hour (Δt = 3,600).

A = the area of the orifice (in in^2).

P_i = the pressure inside the steam pipe (in psia).

Estimate the hourly, monthly, and annual costs of steam leaks from 300-psig pipe. The steam is generated from a gas boiler with an efficiency of 80 percent. The cost of natural gas is $0.90/therm (1 therm is equivalent to 100,000 Btu/hr). The heating season spans 7 months per year. Provide the costs for the following orifice diameters: 1/16; 1/8; 1/4; 3/8; 1/2; and 1 in.

8.3 An efficiency test of a boiler fired by fuel No. 2 indicated a flue gas temperature of 700°F and excess air of 40 percent. The annual fuel consumption of the boiler is 85,000 gal/year. The cost of fuel No.2 is $1.15/gal.

(a) Estimate the efficiency of the boiler as well as the percentage in CO_2 and O_2 of the flue gas.

(b) Determine the new boiler efficiency and the annual cost savings in fuel use if the percentage of O_2 in the flue gas is reduced to 3 percent and the stack temperature is set to 500°F.

(c) Determine the payback period of installing an automatic control system at a cost of $8,500) to maintain the same efficiency found in (b) throughout the life of the boiler.

8.4 A boiler has an annual fuel oil No. 2 consumption of 325,000 gallons. The cost of fuel is $1.50 per gallon. A combustion efficiency test done on the boiler indicated that the stack temperature is 650°F and the CO_2 content is 11 percent.

(a) Determine the excess combustion air and the boiler efficiency.

(b) If the fuel–air ratio is adjusted so that the boiler efficiency is increased to 84 percent, determine the payback period, the net present worth, and the rate of return (d = 5 percent over 10 years) of installing a full metering kit with O_2 trim control that cost $45,000. Comment on the cost effectiveness of the kit.

8.5 A combustion efficiency test done on a gas-fired boiler indicated that the stack temperature is 750°F and the CO_2 content is 7 percent.

(a) Determine the excess combustion air and the boiler efficiency.

(b) The boiler efficiency is increased to 83 percent by adjusting the fuel–air ratio so that the excess air is limited to 30 percent. Estimate the new stack temperature and the CO_2 content.

(c) Estimate the annual natural gas consumption (in therms) if a combustion gas analyzer kit, installed at a cost of $9,500, is paid back after four years. The kit helps maintain the boiler efficiency at 83 percent. The annual cost of natural gas is $1.40 per therm.

8.6 A combustion efficiency test done on a boiler indicated that the stack temperature is 350°C and the CO_2 content is 10 percent.

(a) Determine the excess combustion air and the boiler efficiency if the boiler uses fuel No. 2.

(b) Determine the excess combustion air and the boiler efficiency if the boiler uses fuel No. 6.

(c) Determine the excess combustion air and the boiler efficiency if the boiler uses natural gas.

8.7 A combustion efficiency test performed on the boiler using fuel oil has indicated that the stack temperature is 700°F and the CO_2 content is 15 percent.

(a) Determine the excess combustion air and the boiler efficiency.

(b) If the fuel–air ratio is adjusted (so that the O_2 level is only 4 percent and the stack temperature is 400°F), determine the new boiler efficiency and the new CO_2 content.

(c) Determine the payback period and cost-effectiveness (d = 5 percent over 15 years) of installing a full metering kit with O_2 trim control that costs $30,000. The boiler uses 400,000 gallons of fuel oil per year. The cost of fuel is $1.75 per gallon.

(d) In fact, the auditor is not sure if oil No. 2 or No. 6 is used! If you want to be conservative in terms of payback period estimation, what fuel type would you use?

9 Cooling Equipment

9.1 Introduction

According to a survey reported by the U.S. Energy Information Administration (EIA, 2006), several types of cooling systems are used in commercial buildings including:

- Packaged air-conditioner units
- Central chillers
- Individual air conditioners
- Heat pumps
- Residential-type central air conditioners
- District chilled water
- Swamp (or evaporative) coolers

Table 9.1 summarizes the results of an EIA study to assess the percentage, respectively, of floor space and number of commercial buildings that are conditioned by each type of cooling equipment. In particular, packaged air-conditioner (AC) units are the main equipment used to condition buildings in the United States, both in terms of percentage of total cooled buildings (44 percent) and percentage of the total cooled floor-space (53 percent). On the other hand, central chillers are used in only 3 percent of conditioned commercial buildings but cool 20 percent of the floor area. Indeed, central chillers are typically used in larger buildings whereas packaged AC units are installed in smaller buildings. Specifically, the same EIA study indicates that for buildings over 20,000 m^2 (or 200,000 ft^2) of floor area, central chillers are used to cool almost 54 percent of the total cooled floor area associated with U.S. commercial buildings.

9.2 Basic Cooling Principles

A typical cooling system consists of several components including a compressor, a condenser, an expansion device, an evaporator, and other auxiliary equipment. Figure 9.1 illustrates a simple chilling system using a vapor compression cycle.

Note that chilling (or heat extraction) occurs at the evaporator whereas heat rejection is done by the condenser. Both the evaporator and the condenser are heat exchangers. At the evaporator, heat is extracted by the refrigerant from water that is circulated through cooling coils of an air-handling unit. At the condenser, heat is extracted from the refrigerant and rejected to the ambient air (for air-cooled condensers) or water (for water-cooled condensers connected to cooling towers).

Before the 1980s, there was no regulation for the selection of refrigerants to operate air-conditioning systems. Generally, most manufacturers of HVAC systems use chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs). However, international agreements (UNEP, 1999) and federal regulations (such as the 1993 EPA regulations in the United States) have placed some restrictions

TABLE 9.1 Cooling Equipment Used for Main Space Conditioning Other Uses in U.S. Commercial Buildings in 2003

Cooling System Type	Cooled Floor-Space[a] (%)	Cooled Buildings[a] (%)
Packaged AC units	53	44
Central chillers	20	3
Individual air conditioners	22	20
Residential-type AC units	19	28
Heat pumps	16	14
District chiller water	5	1
Swamp coolers	3	3
Other	2	1

Source: EIA, Energy Information Agency, 2003 Commercial Building Energy Consumption Survey, Washington, DC, 2006.

[a] More than one cooling system may apply.

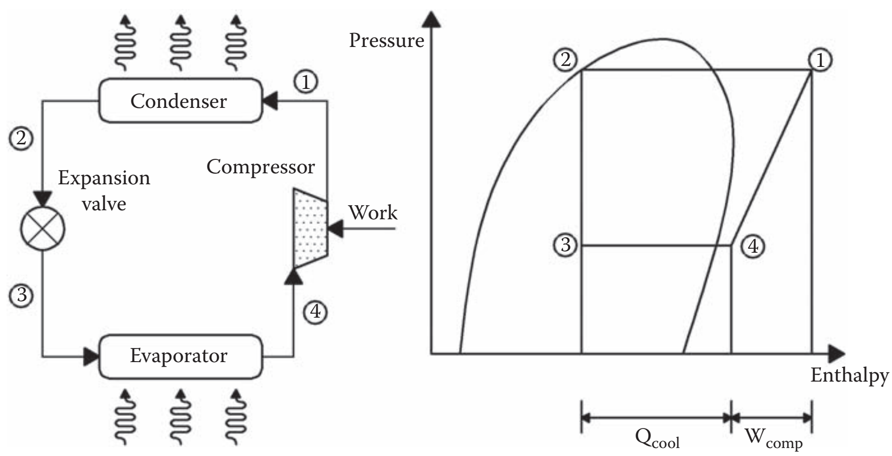


FIGURE 9.1 Typical vapor compression cycle (a) schematic and (b) P-h diagram.

on the use of CFCs and HCFCs due to their high potential in ozone depletion and global warming. Hydrofluorocarbons (HFCs) are now being used. Table 9.2 summarizes some of the characteristics of commonly used refrigerants in air-conditioning applications. Two ratings are typically used to determine the harmful effects of a refrigerant on the environment: the ozone depletion potential (ODP) and the halocarbon global warming potential (HPWP). The first rating, OPD, measures the ability of refrigerant molecules to destroy ozone in the stratosphere and depends mainly on the atmospheric life of the refrigerant. The second rating, HGWP, provides the potential of a refrigerant to contribute to the greenhouse effect. Both ratings, OPD and HGWP, are normalized to the value of R-11. As indicated in Table 9.2, HFCs have no harmful effects on the environment. Because they are nontoxic and nonflammable, R-11, R-22, and R-134a are refrigerants commonly used in building air conditioning applications.

Generally, the energy efficiency of a cooling system is characterized by its coefficient of performance (COP). The COP is defined as the ratio of the heat extracted divided by the energy input

TABLE 9.2 Characteristics of Commonly Used Refrigerants in Air-Conditioning Applications

Refrigerant	Boiling Temperature (°C)	Potential for Flammability	Ozone Depletion Potential (ODP)	Halocarbon Global Warming Potential (HGWP)	Common Applications
CFCs:					
R-11	24	None	1.0	1.0	Cent. Chillers
R-12	−30	None	1.0	3.05	Auto A/C
R-22	−41	None	0.051	0.37	Package A/C
R-123	27	Moderate	0.016	0.019	Low P. Chillers
HCFCs:					
R-32	−52	Moderate	0	0.13	Package A/C
R-125	−49	None	0	0.58	Package A/C
R-134a	−26	None	0	0.285	Refrigeration, Chillers
HFCs:					
R-290	−42	High	0	0	Replace R-22

required. In the case of an electrically driven cooling system as represented in Figure 9.1, the COP can be expressed as:

$$COP = \frac{Q_{coot}}{W_{comp}} \tag{9.1}$$

Both Q_{cool} and W_{comp} should be expressed in the same unit (i.e., W or kW), so that the COP has no dimension.

The maximum theoretical value for COP can be estimated using the ideal Carnot cycle COP. The Carnot cycle consists of isentropic compression and expansion and isothermal evaporation and condensation. In real cooling systems, the energy efficiency of the Carnot cycle cannot be attained because of irreversible losses. Among these are the irreversible losses in the compression and expansion of the refrigerant, the pressure losses in the lines, and the heat losses in the rejection and absorption processes due to the nonuniform temperature through the heat exchangers. However, it is useful to compare the COP of an actual cooling system to that of the Carnot cycle operating between the same temperatures to determine the potential of any added energy efficiency improvements in the design of cooling units. The COP of an ideal Carnot cycle can be expressed in terms of the absolute temperature of the evaporator T_C (the lowest temperature in the cycle), and the condenser T_H (the highest temperature in the cycle) as follows:

$$COP_{Carnot} = \frac{T_C}{T_H - T_C} \tag{9.2}$$

For instance, using the ARI standard 550/590 (1998) rated conditions for water chillers, T_H is 308 K and T_C is 280 K, the COP of Carnot cycle can be estimated by Eq. (9.2) to be 9.88. Currently, the most energy-efficient centrifugal water chiller has a COP of about 7.0 or about 70 percent of the ideal Carnot cycle.

Most manufacturers typically provide the COP of their cooling systems for full load conditions. The capacity of cooling systems is expressed in kW and is defined in terms of the maximum amount of heat that can be extracted. In the United States, manufacturers and HVAC engineers use refrigeration tons to rate the capacity of the cooling systems (1 ton is about 3.516 kW), and kW/ton to express their energy efficiency. In addition, the energy efficiency of the electrically powered cooling systems can be expressed in terms of the energy efficiency ratio (EER) which is defined as the ratio of the heat extracted (expressed in Btu/hr) over the energy input required (expressed in watts). Therefore, the relationship between the EER and the COP is given as follows:

$$EER = 3.413 * COP \tag{9.3}$$

The definition of the EER provided above is specific to the U.S. HVAC industry. In Europe, the EER is defined to be exactly the same as the COP. However, the adopted European standard EN 14511 (PrEN, 2003) specifies that the term COP is to be used only for the heating mode operation of heat pumps. Otherwise, the standard requires the use of the term EER to rate the energy efficiency of air conditioners and heat pumps.

ARI standard 550/590 allows rating the energy efficiency of water chilling systems using the vapor compression cycle by one of the three parameters: COP, EER, or kW/ton. These three parameters are related as indicated below:

$$kW/ton = {}^{3.516}\!/_{COP} = {}^{12}\!/_{EER} \tag{9.4}$$

The rating tests for chillers typically need to be performed under specific conditions for leaving chilled water temperatures and entering condenser water temperatures or air dry-bulb and wet-bulb temperatures.

Generally, the energy efficiency of a cooling system varies under part-load conditions. Cooling systems often operate under part-load conditions throughout the year, therefore other energy efficiency coefficients have been proposed in an attempt to provide a better estimation of the energy performance of the cooling units over a wide range of operating conditions. Currently, two parameters are commonly used in the HVAC industry: the seasonal energy efficiency ratio (SEER) and the integrated part load value (IPLV). ARI 550/590 standard defines the IPLV based on the COP (or EER) values at 100, 75, 50, and 25 percent using the following equation:

$$IPLV = 0.01 * A + 0.42 * B + 0.45 * C + 0.12 * D \tag{9.5}$$

where

A = the COP (or EER) at 100 percent load.
B = the COP (or EER) at 75 percent load.
C = the COP (or EER) at 50 percent load.
D = the COP (or EER) at 25 percent load.

The COP and the EER at part-loads are determined using specific conditions. Table 9.3 summarizes the conditions for both the entering water temperatures (EWT) for water-cooled condensers and the entering air dry-bulb temperatures (EDB) for air-cooled condensers. For all conditions and types, the evaporator leaving chilled water temperature is specified to be 6.7°C.

The following section provides a brief description of commonly used cooling systems and their typical energy efficiencies. In later sections, some common improvement measures to increase the energy efficiency of cooling systems are discussed.

TABLE 9.3 Part-Load Conditions for ARI 550/590 Standard Rating

Cooling Load (% of Capacity)	Water-Cooled Condensers EWT (°C) [°F]	Air-Cooled Condensers EDB (°C) [°F]
100	29.4 [85]	35.0 [95]
75	23.9 [75]	26.7 [80]
50	18.3 [65]	18.3 [65]
25	18.3 [65]	12.8 [55]
0	18.3 [65]	12.8 [55]

9.3 Types of Cooling Systems

As mentioned in the introduction, several types of cooling systems are currently available for space air conditioning. The most common cooling systems used for space air conditioning can be grouped into major categories: unitary AC systems and chillers. The unitary AC systems include packaged AC units, individual air conditioners, residential type AC units, and heat pumps.

9.3.1 Unitary AC Systems

Unitary AC systems are typically factory-assembled units to provide either cooling only or both cooling and heating. Compared to chillers, the unitary AC systems have a shorter life span and lower energy efficiency. They are typically installed in small commercial buildings (with fewer than three floors) including small office buildings, retail spaces, and classrooms.

9.3.2 Packaged AC Units

The packaged AC units are compact cooling systems encased in cabinets. There are various types of packaged AC units including:

- *Rooftop systems* are typically located on the roof (thus, the name rooftop AC units). For commercial buildings, the rooftop AC units are available in the range of 17 to 70 kW (or 5 to 20 tons) even though custom-built units can have larger capacities (up to 350 kW or 100 tons). For residential buildings, capacities between 3 to 7 kW (or 0.75 to 2 tons) are common. Most units are equipped with a heating system (a built-in gas furnace, an electric resistance, or a heat pump) to provide both cooling and heating.
- *Vertical packaged systems* are typically designed for indoor installation. Most systems have water-cooled condensers.
- *Split packaged systems* typically have an air-cooled condenser and compressor installed outdoors and the evaporator installed in an indoor air-handling unit.

9.3.3 Heat Pumps

Heat pumps can be used for both cooling and heating by simply reversing the refrigeration flow through the unit. The heat sink (or source) for the heat pump can be air, water, or ground. For commercial and industrial applications, air-to-air heat pumps can have capacities up to 90 kW (or 25 tons); hydronic heat pumps can have higher cooling capacities. Ground-coupled heat pumps are still small and are mostly suitable for residential applications.

9.3.4 Central Chillers

In large buildings, central chillers are used to cool water for space air conditioning. Central chillers are powered by electric motors, fossil fuel engines, or turbines. Some chillers use hot water or steam to generate chilled water. A description of various types of central chillers is provided below.

9.3.4.1 Electric Chillers

There are currently three major types of electric chillers available on the market using centrifugal, reciprocating, or rotary compressors. All these chillers use a mechanical vapor compression cycle.

1. Centrifugal compressors use rotating impellers to increase refrigerant gas pressure and temperature. Chillers with centrifugal compressors have capacities in the range of 300 kW to 25,000 kW (or 85 to 7,000 tons). For capacities above 4,500 kW (or 1,250 tons), the centrifugal compressors are typically field erected.
2. Reciprocating compressors use pistons to raise the pressure and the temperature of refrigerant gases. Two or more compressors can be used under part-load conditions to achieve higher operating efficiencies. Capacities of 35 kW to 700 kW (or 10 to 200 tons) are typical for chillers with reciprocating compressors.
3. Rotary compressors use revolving motions to increase refrigerant gas pressure. One of the most ingenious rotary compressors is the scroll compressor. The most conventional rotary compressors are the screw compressors that can have several configurations. The capacity of the rotary chillers can range from 3 kW to 1750 kW (1 to 500 tons).

9.3.4.2 Absorption Chillers

Absorption chillers operate using a concentration–dilution cycle to change the energy level of the refrigerant (water) by using lithium bromide to alternately absorb heat at low temperatures and reject heat at high temperatures. The absorption chillers can be direct-fired (using natural gas or oil fuel), or indirect-fired. Indirect-fired units may use steam or hot water (from a boiler, a district heating network, an industrial process, or waste heat) as a heat source. A typical absorption chiller includes an evaporator, a concentrator, a condenser, and an absorber.

- *Direct-fired absorption chillers* can be cost-effective when the price of natural gas is favorable. Some of the direct-fired chillers can be used to produce both chilled and hot water. Thus, they can provide cooling and heating and are sometimes referred to as chillers/heaters. These chillers/heaters can be cost-effective especially when heating needs exist during the cooling season (for instance, buildings with large service hot water requirements). Two types of absorption chillers are available on the market: single-effect and double-effect chillers. Some prototypes of triple-effect absorption chillers have been built in the United States. Capacities ranging from 100 kW to 5,000 kW (or 30 to 1,500 tons) are available for direct-fired chillers.
- *Indirect-fired absorption chillers* operate with steam (with pressures as low as 15 psig) or hot water (with temperature as low as 140°C). Some small absorption chillers using solar energy (to generate hot water) have been proposed and some prototypes have been developed and evaluated. Cooling capacities from 15 kW to 425 kW (4 to 120 tons) are available even though typical sizes range from 200 kW to 5,000 kW (55 to 1,450 tons). Double-effect chillers can be considered only for high-temperature hot water and steam or with hot industrial waste gases.

9.3.4.3 Engine-Driven Chillers

As do electrically driven chillers, the engine-driven chillers can use reciprocating, rotary, or centrifugal compressors to provide mechanical refrigeration. The compressors can be powered by turbines or gas-fired engines. The engine-driven chillers can have large capacities up to 15,000 kW (4,250 tons) but have usually high first costs.

Table 9.4 provides typical energy efficiency indicators for commonly available chiller units.

TABLE 9.4 Typical COP for Different Chiller Types

Type of Chillers	Range for COP
Small electric chillers	
Air cooled	2.2–3.2
Large electric chillers	
Air cooled	3.7–4.1
Water cooled	4.6–5.3
Absorption chillers	
Single-effect	0.4–0.6
Double-effect	0.8–1.1
Engine-driven chillers	1.2–2.0

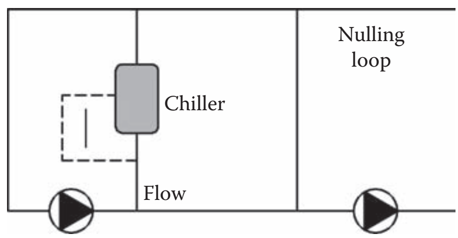


FIGURE 9.2 Typical primary/secondary loop system.

9.4 Water Distribution Systems

Most chillers are operated on a primary/secondary distribution loop system as illustrated in Figure 9.2. This guarantees flow through the evaporator barrel and ensures against freezing.

There are several methods that can be used to distribute chilled water in order to meet cooling loads. Figure 9.3 illustrates some of the distribution systems used in cooling (as well as heating) central plants. The distribution systems include direct return, reverse return, and flow bypass. The source of cooling is generally a chiller. However, when outdoor conditions are suitable, cooling towers can be used to cool chilled water through what is often referred to as a water-side economizer cycle. The load typically consists of cooling coils (part of air-handling units). The volume rate as well as the temperature of the chilled water supplied to the coils can be controlled using three-way mixing valves.

9.4.1 Pumps

In most cooling systems, chilled water is moved by pumps. Pumps are very much like centrifugal fans and follow similar principles except they use an impeller rather than fan blades. A typical centrifugal pump is shown in Figure 9.4.

The performance of a pump depends on the power of the motor and the shape and size of the impeller. A fluid passing through a pump will show an increase in pressure. The pressure produced by the pump is often called the pump head (as in head pressure.) The power added to the fluid is found from the product of the flow rate and the total head as expressed in Eq. (9.6) for SI unit and Eq. (9.7) for IP unit:

$$kW = Flowrate.Head \tag{9.6}$$

$$HP = \frac{GPM \cdot feet\ head \cdot S.G.}{3960} \tag{9.7}$$

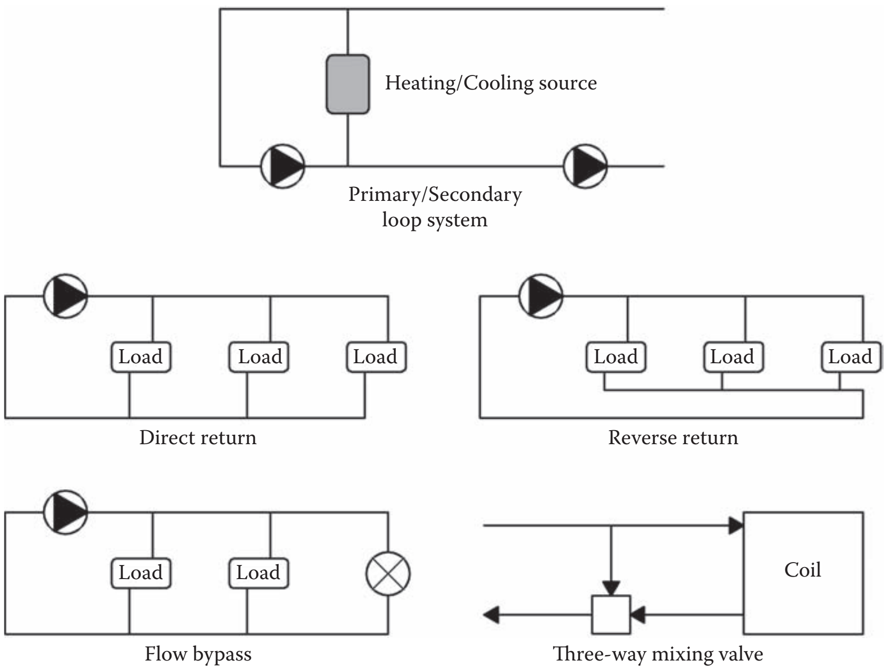


FIGURE 9.3 Types of distribution systems for central cooling plants.

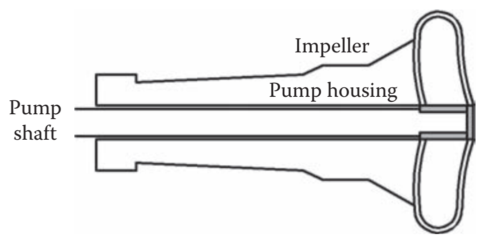


FIGURE 9.4 Typical centrifugal pump.

The basic operation of pumps follows the affinity laws. Pump affinity laws are very similar to the fan laws described in Chapter 7. If the impeller size is a constant but the impeller speed changes, the following affinity laws apply to the operation of pumps:

$$\frac{GPM_2}{GPM_1} = \frac{RPM_2}{RPM_1}; \quad \frac{\Delta P_2}{\Delta P_1} = \left(\frac{RPM_2}{RPM_1}\right)^2; \quad \frac{kW_2}{kW_1} = \left(\frac{RPM_2}{RPM_1}\right)^3 \tag{9.8}$$

If the impeller speed is a constant but the impeller size changes, the following laws can be applied:

$$\frac{GPM_2}{GPM_1} = \frac{D_{i,2}}{D_{i,1}}; \quad \frac{\Delta P_2}{\Delta P_1} = \left(\frac{D_{i,2}}{D_{i,1}}\right)^2; \quad \frac{kW_2}{kW_1} = \left(\frac{D_{i,2}}{D_{i,1}}\right)^3 \tag{9.9}$$

The pressure drop in a piping system will vary as a function of the square of the volumetric flow rate of fluid passing through the pipe:

$$\frac{\Delta P_2}{\Delta P_1} = \left(\frac{GPM_2}{GPM_1}\right)^2 \tag{9.10}$$

Water systems are usually broken down into five categories:

1. Low temperature water (LTW) with maximum temperatures and pressures of 250°F and 150 psi
2. Medium temperature water (MTW) at 325°F and 150 psi
3. High temperature water (HTW) at 450°F and 300 psi
4. Chilled water (CHW) at 40–50°F and 150 psi
5. DTW that satisfies both heating and cooling

9.4.2 Pump and System Curves

To analyze a water distribution system, it is convenient to utilize the pump performance and system curves. The pump curve is a graphical representation of the pressure rise across the pump as a function of the flow rate. The system curve provides the pressure drop variation through a system as a function of the flow rate. It is often convenient to superimpose these curves so that the operating point of the system can be easily identified (it is where the pump and system curves intersect). Figure 9.5 indicates the pump and system curves when two pumps are arranged either in series or parallel.

There several methods to control chilled water flows in central cooling plants. The commonly used control methods include:

- VSD (controls system DP)
- Three-way valves (constant flow)
- Throttling valve
- Flow bypass
- Direct return
- Reverse return

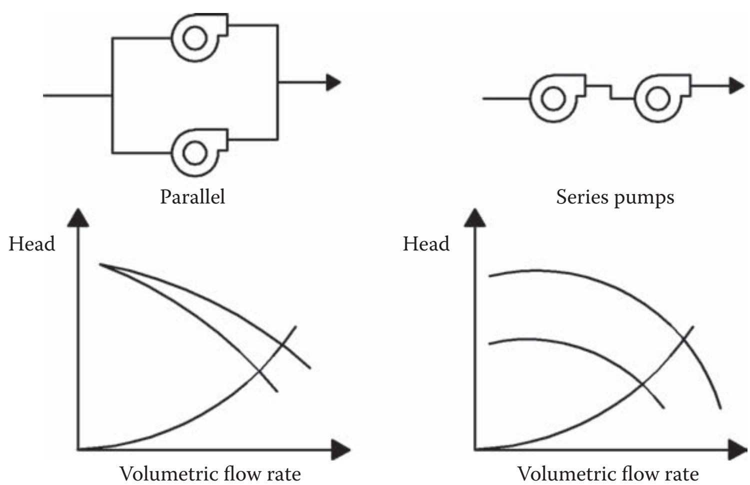


FIGURE 9.5 Pump and system performance curves for both parallel and series arrangements.

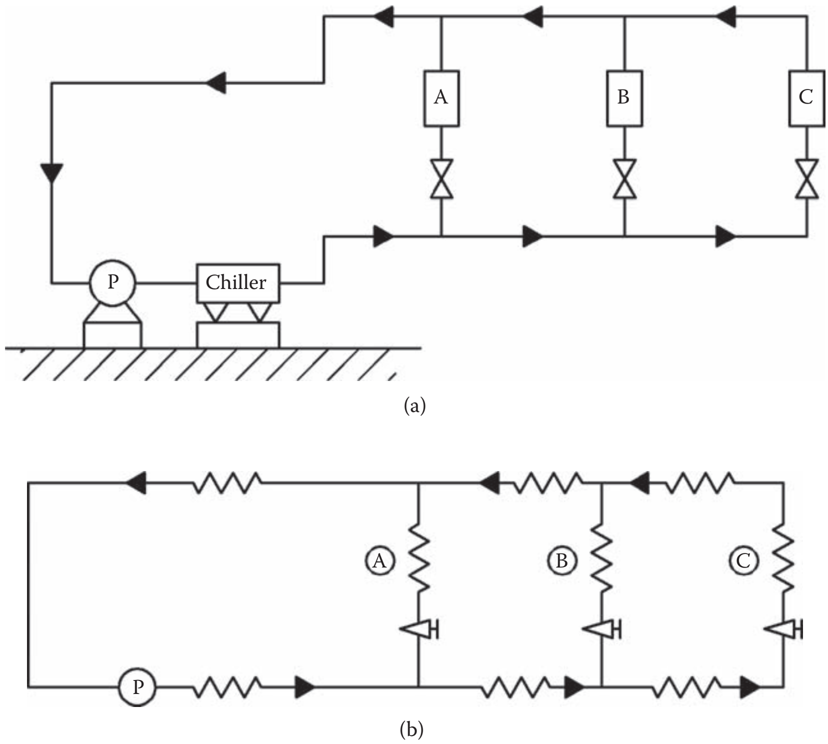


FIGURE 9.6 Chilled water system with one pump in the primary loop, a chiller, and three secondary loops: (a) actual distribution system and (b) equivalent electrical circuit.

9.4.3 Analysis of Water Distribution Systems

In order to evaluate and determine the optimal primary/secondary distribution system configuration that minimizes energy use, a thermal network analysis can be carried out using the analogy between an electrical circuit network and water distribution system. This analogy is illustrated in Figure 9.6 for a chilled water system with one pump in the primary loop, a chiller, and three secondary loops. The water flow rate, pressure drop, and total dynamic heads are equivalent to current, voltage drop, and electrical resistances. The valves in various circuits can be adjusted to balance the secondary loops in order to ensure that the total dynamic head remains the same for all the loops. Example 9.1 shows how to estimate the valve settings to balance a chilled water distribution system and the potential energy use and cost savings from an optimal primary/secondary system.

Example 9.1

Consider the water distribution system with only one primary pump as depicted in Figure Ex 9.1. Determine the hp size of the primary pump and check the settings of the valves for various secondary loops to balance the system. Determine the best primary/secondary pumping arrangement for retrofitting the water distribution system. Estimate the total energy cost savings from the retrofit assuming the system is operated 5,000 hours per year and that the cost of energy is $0.08/kWh. Assume the pump efficiency to be 70 percent and the motor efficiency to be 90 percent.

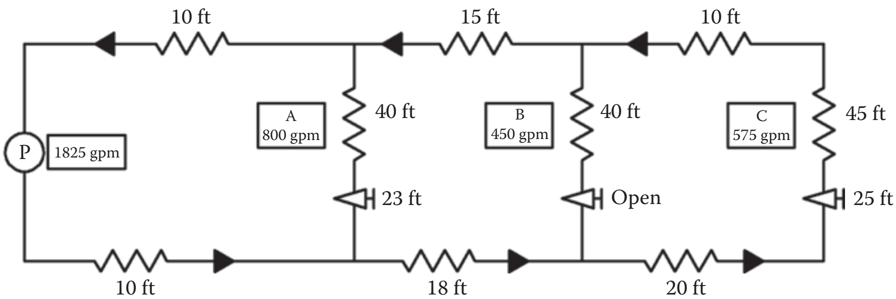


FIGURE Ex-9.1 Water distribution system with one primary pump

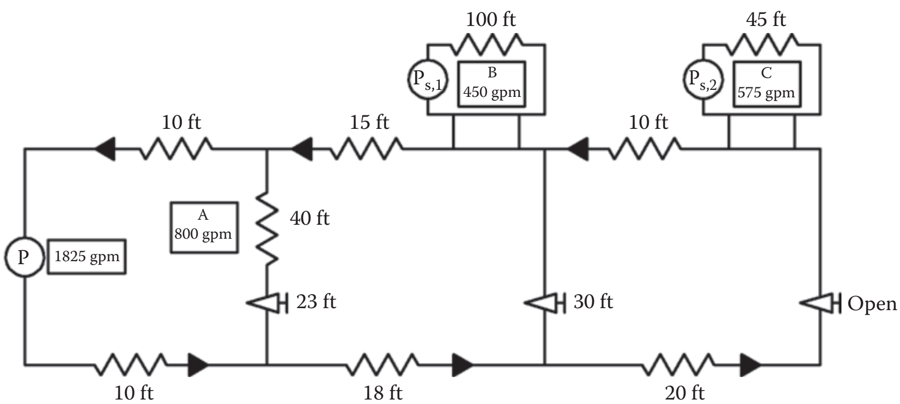


FIGURE Ex-9.2 Water distribution system with primary and secondary pumps.

Solution

For the water distribution system presented by the network above, the heads for the three circuits are first estimated as shown below:

$$\text{Head (circuit A)} = 10 \text{ ft} + 40 \text{ ft} + 10 \text{ ft} = 60 \text{ ft}$$

$$\text{Head (circuit B)} = 10 \text{ ft} + 15 \text{ ft} + 100 \text{ ft} + 18 \text{ ft} + 10 \text{ ft} = 153 \text{ ft}$$

$$\text{Head (circuit C)} = 10 \text{ ft} + 15 \text{ ft} + 10 \text{ ft} + 45 \text{ ft} + 20 \text{ ft} + 18 \text{ ft} + 10 \text{ ft} = 128 \text{ ft}$$

Thus the total head for the system is 153 ft corresponding to the longest run for circuit B. The horsepower for the primary pump is obtained using Eq. (9.15):

$$\text{Pump horsepower} = (1825 \text{ gpm})(153 \text{ ft})/(.70 * 3{,}960) = 101 \text{ hp}$$

We can check that the valve for circuit B should be open and the valves for circuits A and C should be set to 93 ft and 25 ft, respectively.

To minimize energy use for the water distribution system, a primary/secondary pumping system can be considered. The best primary/secondary pumping arrangement is shown in the Figure Ex 9.2 with secondary pumps in circuits B and C. The longest run for the primary pump becomes 83 ft (corresponding to loop C without the secondary loop of 45 ft). The horsepower sizes of the two secondary pumps and the new primary pump are:

$$\text{Secondary pump B horsepower} = (450 \text{ gpm})(100 \text{ ft})/(.70 * 3{,}960) = 16 \text{ hp}$$

$$\text{Secondary pump C horsepower} = (575 \text{ gpm})(45 \text{ ft})/(.70 * 3{,}960) = 9 \text{ hp}$$

Primary pump horsepower = (1,825 gpm)(83 ft)/(.70 * 3,960) = 55 hp

Thus the total hp requirement for the new primary/secondary distribution system is:

Total hp = 55 + 16 + 9 = 80 – hp

Thus, a total savings of 21 hp can be achieved by retrofitting the water system to a primary/secondary pumping arrangement as shown above. Assuming an operation of 5,000 hours/year, a motor efficiency of 0.90, and a cost of $0.0.08/kWh, the energy cost savings achieved by the retrofit of the water distribution system is estimated to be:

Cost Savings = ($0.08/kwh) * (5,000-hrs) * (21– hp * 0.746 – kW/hp/0.90) = $6,250/year

9.5 District Cooling Systems

District cooling systems are centralized cooling plants that distribute chilled water or other media to multiple buildings for air conditioning or other uses. District cooling is now widely used in downtown business districts and institutional settings such as college campuses. Generally, district cooling systems that serve buildings of a single entity are categorized as institutional systems. All others are referred to as commercial district cooling systems.

District cooling has its roots in early nineteenth century schemes to distribute clean cool air to houses through underground pipes. However, the first known built district cooling system was built and operated by the Colorado Automatic Refrigerator Company in Denver in late 1889. Many early systems supplied ammonia and brine for the refrigeration of meat, as well as cooling restaurants, theatres, and other public buildings. Large district cooling systems were built in the 1930s in Rockefeller Center and the United States Capitol complex.

In order for district cooling systems to be cost-effective, the cooling load density, that is, the cooling requirements per unit area, should be relatively high. Indeed, it is uneconomical to distribute energy to sparsely populated areas where distribution piping costs and thermal losses are relatively high compared to conventional systems where cooling is generated within each building. High total cooling loads improve thermal efficiencies of cooling equipment (ASHRAE, 2008).

Typically, apartment complexes, hospitals, universities, groups of office buildings, and industrial facilities are suitable for district cooling and heating. Several cities around the world are served by district heating and cooling systems.

In this section, the components of a typical district cooling system are described. In addition, the benefits and advantages of district cooling are outlined. Specific technologies that can improve the cost-effectiveness of district cooling systems are discussed. Figure 9.7 illustrates a typical district cooling system and its components including (ASHRAE, 2008):

- Central cooling plant with chillers, thermal energy storage systems, and pumps
- Distribution networks (piping system in tunnels or buried)
- Consumer systems (direct or indirect)

The types and performance of various components of district cooling systems are described in the following sections. The distribution networks and consumer systems can be used for both district heating and cooling systems. Thus, the discussion of these two components considers both heating and cooling applications.

Because they are usually connected to a diverse group of customers with varying load requirements, chillers in district cooling systems have to accommodate a relatively large total cooling load with potentially wide variations from season to season. Individual customers often experience their peak loads at different times of the day, therefore the daily load curve of the central chillers tends to be smoothed out, with the peak demand reduced, compared to the sum of all the individual peak loads. Thus, the installed

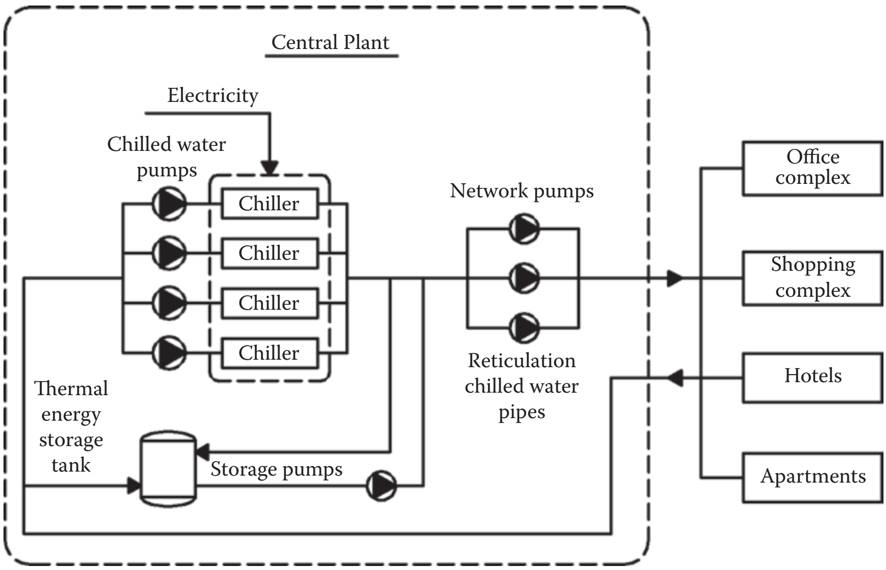


FIGURE 9.7 Typical components of a district cooling system.

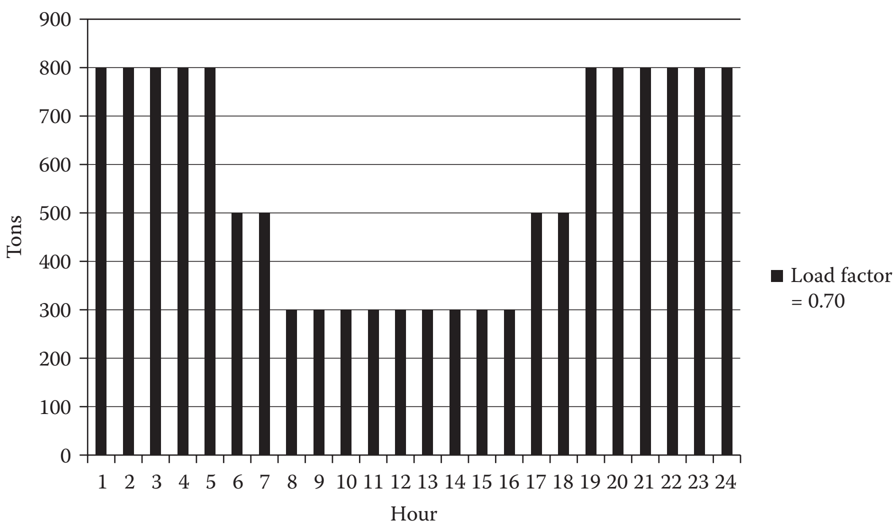


FIGURE 9.8 Cooling load profile for an apartment building.

total cooling capacity of chillers in district cooling systems can be less than that of conventional decentralized systems.

Figures 9.8 and 9.9 show typical hourly cooling demand profiles for two large buildings, one for a commercial office building and the other for an apartment building. Both buildings demonstrate significant cooling demand variation with a similar load factor (i.e., average to peak load ratio). However, the peak occurs during daytime hours for the office building and during night hours for the apartment building. Figure 9.10 shows the demand profile for both buildings and demonstrates the flattening effect on cooling demand for the district cooling system when used by a variety of customer types.

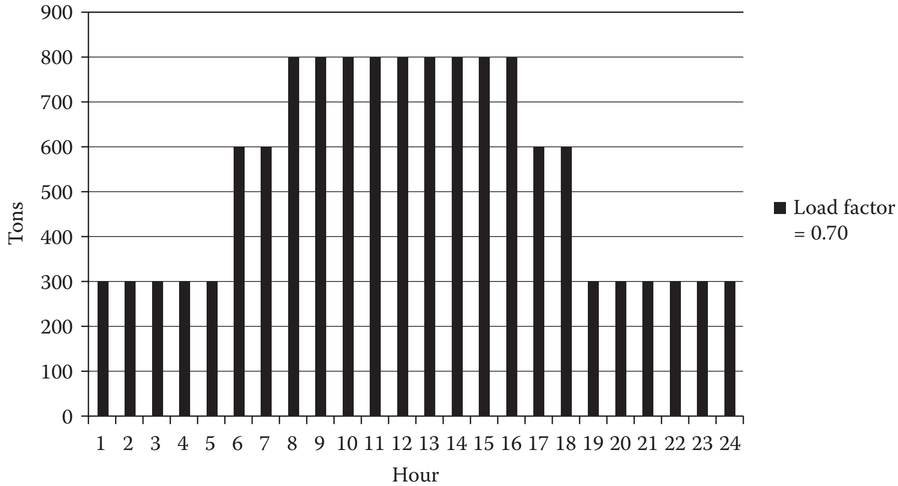


FIGURE 9.9 Cooling load profile for an office building.

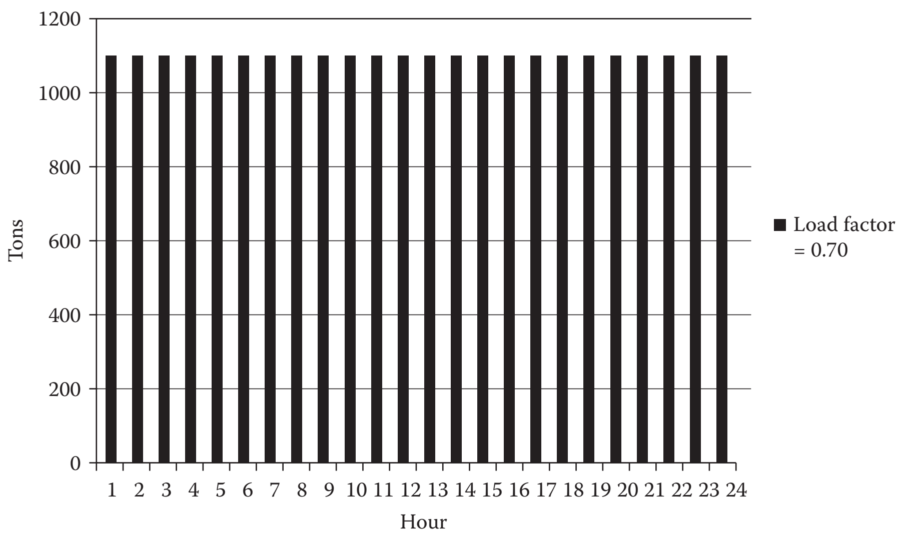


FIGURE 9.10 Combined cooling load profile for both the apartment and the office buildings.

Depending on total system peak, average load requirements, and the load variations from day to day and season to season, chillers for district systems can be designed. A relatively simple district cooling system might utilize a single energy production facility, consisting, for example, of an electrically driven centrifugal chiller. Multiple units may also be selected to meet base, intermediate, and peak loads more efficiently, as well as providing standby capacity and increased system reliability. More complicated district cooling systems might utilize several different energy production facilities such as waste heat from manufacturing plant processes to operate absorption chillers. When several thermal energy production units are considered for the district cooling system, different energy sources can be selected depending on fuel prices, availability of such alternatives, proximity of the load to such sources, environmental concerns, and other factors.

TABLE 9.5 Hours at Percent Part-Load Ratio Using One Chiller

Climate	0–10	10–20	20–30	30–40	40–50	50–60	60–70	70–80	80–90	90–100	100+
Denver	539	601	1,214	1,182	703	140	25	3	2	1	0
Chicago	644	523	980	1,290	657	139	19	1	0	0	0
Phoenix	350	456	790	1,270	1,349	600	55	25	16	8	1
Atlanta	589	595	888	1,167	633	635	45	2	2	0	0

9.6 Multichiller Systems

Replacing a single chiller with two or more smaller chillers to meet varying load requirements may be cost-effective. "Parallel staging" of multiple chillers is a common method of meeting peak load in larger installations. Multiple chillers also provide redundancy for routine maintenance and equipment failure. For many typical facilities, sizing one chiller at one-third and another chiller at two-thirds of the peak load enables the system to meet most cooling conditions at relatively high chiller part-load efficiencies. These staged units can also be sized optimally for different conditions. For example, one chiller could be optimized for peak efficiency at summer conditions (85°F condensing water) and the other chiller could be optimized for winter conditions (75°F condensing water).

Furthermore, proper sequencing helps to maintain the flow rate through each evaporator within the range recommended by the chiller manufacturer. As the system flow nears the maximum limit for the operating chiller(s), another machine must be brought online. Similarly, as the system load and flow decrease, chillers must be shut down to reduce the need for bypass water flow.

The energy savings attributed to switching from a single-chiller plant to a multichiller cooling plant depend on the type of building and climate. Table 9.5 shows the number of hours of operating a one-chiller system at various part-load ratios for an office building located in four U.S. climates. The variation of the total building electrical energy use as a function of the number of chillers in a multichiller cooling plant is provided in Figure 9.11.

Figure 9.12 illustrates the potential savings associated with the annual building electrical energy savings when using two-chiller central cooling plants with various chiller sizing ratios. The chiller size ratio SR is defined as the ratio between the sizes of the small chiller to that of the larger chiller in a two-chiller cooling plant. The analysis was carried out by letting the SR vary from 0 (single-chiller configuration) to 1 (two equal chillers).

9.7 Energy Conservation Measures

To reduce the energy use of cooling systems, the energy efficiency of the equipment has to be improved under both full-load and part-load conditions. In general, improvement of the energy efficiency of cooling systems can be achieved by one of the following measures:

- Replace the existing cooling systems by others that are more energy efficient.
- Improve the existing operating controls of the cooling systems.
- Use alternative cooling systems.

The energy savings calculation from increased energy efficiency of cooling systems can be estimated using the simplified but general expression provided by Eq. (9.11):

$$\Delta E_C = .\left(\frac{\dot{Q}_C .N_{h,C} .LF_C}{SEER}\right)_e - .\left(\frac{\dot{Q}_C .N_{h,C} .LF_C}{SEER}\right)_r \tag{9.11}$$

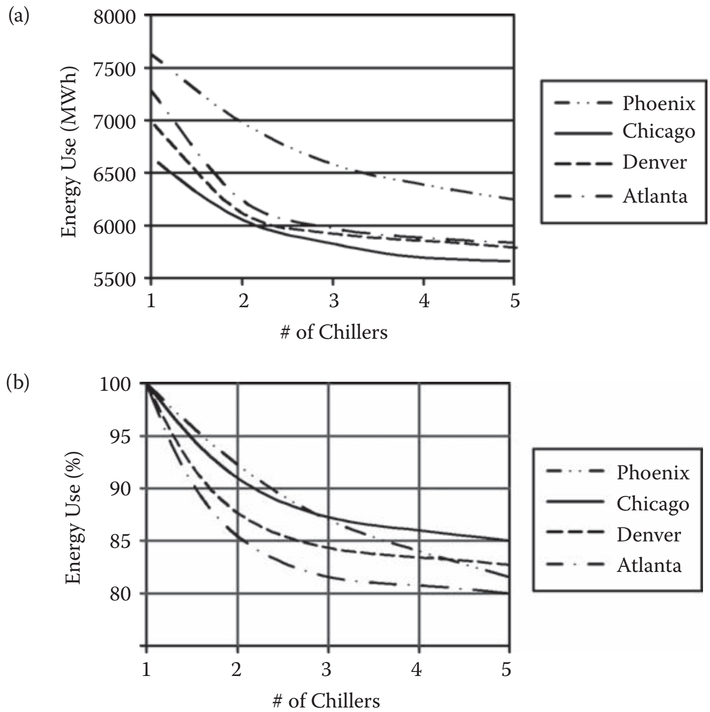


FIGURE 9.11 Energy use for a prototypical office building for four U.S. locations as a function of number of chillers: (a) actual savings; (b) normalized savings.

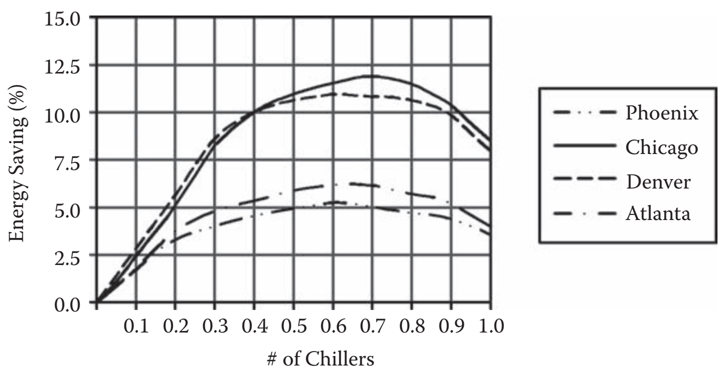


FIGURE 9.12 Normalized annual energy use for a prototypical office building for four U.S. locations for various chiller size ratios in a two-chiller central cooling plant

where

Indices E and R indicate the values of the parameters, respectively, before and after retrofitting the cooling unit.

SEER is the seasonal efficiency ratio of the cooling unit. When available, the average seasonal COP can be used instead of the SEER.

$\dot{Q}_C$ is the rated capacity of the cooling system.

$N_{h,c}$ is the number of equivalent full-load cooling hours.

LF_C is the rated load factor and is defined as the ratio of the peak cooling load experienced by the building over the rated capacity of the cooling equipment. This load factor compensates for oversizing of the cooling unit.

It should be noted that the units for both Q_C and SEER have to be consistent; that is, if SEER has no dimension (using the European definition of EER), Q_C has to be expressed in kW.

When the only effect of the retrofit is improved energy efficiency of the cooling system so that only the SEER is changed due to the retrofit, the calculation of the energy savings can be performed using the following equation:

$$\Delta E_C = \dot{Q}_C.N_{h,C}.LF_C.\left(\frac{1}{SEER_e} - \frac{1}{SEER_r}\right) \tag{9.12}$$

In the following sections, some common energy efficiency measures applicable to cooling systems are described with some calculation examples to estimate energy use and cost savings.

9.7.1 Chiller Replacement

It can be cost-effective to replace an existing chiller by a new and more energy-efficient chiller. In recent years, significant improvements in the overall efficiency of mechanical chillers have been achieved by the introduction of two-compressor reciprocating and centrifugal chillers, variable-speed centrifugal chillers, and scroll compressor chillers. A brief description of each of these chiller configurations is presented below with some estimation of their energy efficiency.

- Multiple compressor chillers can be reciprocating, screw, or centrifugal with capacities in the range of 100 kW to 7,000 kW (i.e., 30 tons to 2,000 tons). They are energy efficient to operate especially under part-load conditions. Some studies indicate that chillers equipped with multiple compressors can save up to 25 percent of the cooling energy use compared to single-compressor chillers (Tuluca, 1997).
- Variable-speed compressor chillers are, in general, centrifugal and operate with variable head pressure using variable speed motors. Therefore, the variable-speed compressor chillers work best when their cooling load is most of the time below the peak. The typical capacity of a variable-speed compressor chiller is in the range of 500 kW to 2,500 kW (i.e., 150 tons to 700 tons). It is reported that chillers with a variable-speed compressor can reduce cooling energy use by almost 50 percent (Tuluca, 1997).
- The scroll compressor is a rotary compression device with two primary components, a fixed scroll and an orbiting scroll, both needed to compress and increase the pressure of the refrigerant. The scroll compressors are more energy efficient than the centrifugal compressor because the heat loss between the discharge and the suction gases is reduced. Manufacturers of scroll compressors report that the COP of the scroll chillers exceeds 3.2.

Example 9.2 illustrates a sample of a calculation to determine the cost-effectiveness of replacing an existing chiller with a high energy efficiency chiller.

Example 9.2

An existing chiller with a capacity of 800 kW and with an average seasonal COP of 3.5 is to be replaced by a new chiller with the same capacity but with an average seasonal COP of 4.5. Determine the simple payback period of the chiller replacement if the cost of electricity is \$0.07/kWh and the cost differential of the new chiller is \$15,000. Assume that the number of equivalent full-load hours for the chiller is 1,000 per year both before and after the replacement.

Solution

In this example, the energy use savings can be calculated using Eq. (9.12) with $SEER_e = 3.5$, $SEER_r = 4.5$, $N_{h,C} = 1{,}000$, $Q_C = 800$ kW, and $LF_C = 1.0$ (it is assumed that the chiller is sized correctly):

$$\Delta E_C = 800kW * 1000hrs/yr * 1.0 * \left(\frac{1}{3.5} - \frac{1}{4.5}\right) = 50{,}800kWh/yr$$

Therefore, the simple payback period for investing in a high-efficiency chiller rather than a standard chiller can be estimated as follows:

$$SPB = \frac{\$15{,}000}{50{,}800kWh/yr * \$0.07/kWh} = 4.2\,years$$

A life-cycle cost analysis may be required to determine if the investment in a high energy-efficient chiller is really warranted.

In some cases, only some parts of the cooling system may need to be replaced. Indeed, regulations have been enacted that phased out the production and use of chlorofluorocarbons including R-11 and R-12 by the end of 1995, after their implication in the depletion of the earth's ozone layer. Inasmuch as CFCs have been extensively used as refrigerants in air-conditioning and refrigeration equipment, the existing stocks of CFCs have been significantly reduced and are becoming expensive. Therefore, replacement and conversion of CFC chillers to operate with non-CFC refrigerants are becoming attractive options. If the existing chiller is relatively new (i.e., less than 10 years old), it may not be cost-effective to replace it with a new non-CFC chiller. Only the conversion of the chiller to operate with non-CFC refrigerants may probably be the most economical option. However, non-CFC refrigerants (such as R-134a and R-717) may reduce the energy efficiency of the chiller by reducing its cooling capacity due to their inherent properties. Fortunately, this loss in energy efficiency can be limited by upgrading some components of the cooling system including the impellers, orifice plates, gaskets, and even compressors. The specifics of a chiller upgrade or conversion are now available from several manufacturers. In some instances, the conversion with equipment upgrade may actually improve the chiller performance (Calm, 2006)

Some of the strategies that can be used to improve the efficiency of existing chillers for an upgrade are listed below:

- Increase the evaporator and the condenser surface area for more effective heat transfer.
- Improve the compressor efficiency and control.
- Enlarge internal refrigerant pipes for lower friction.
- Ozonate the condenser water to avoid scaling and biological contamination.

Oversizing is another problem that may warrant the replacement of cooling systems. Indeed, several existing chillers have a capacity that is significantly higher than their peak cooling load. These chillers operate exclusively under part-load conditions with reduced energy efficiency and thus increased operating and maintenance costs. When the oversized chillers are more than ten years old, it may be cost-effective to replace them with smaller and more energy-efficient chillers operating with non-CFC refrigerants.

9.7.2 Chiller Control Improvement

Before replacing an existing chiller, consideration of alternative cooling systems or simple operating and control strategies are recommended to improve its energy performance. Some common and proven alternative cooling systems such as evaporative cooling and water-side economizers are discussed later in this chapter. In this section, measures involving the use of improved controls are discussed. Among these controls are those based on two fundamental strategies:

1. Supply chilled water at the highest temperature that meets the cooling load.
2. Decrease the condenser water supply temperature (for water-cooled condensers) when the outside air wet-bulb temperature is reduced.

Indeed, chiller performance depends not only on the cooling load but also on the chilled water supply temperature and the condenser water temperature. The Carnot efficiency expressed by Eq. (9.2) can be used to illustrate that the COP increases when the condenser temperature (i.e., T_H) is reduced or when the evaporator temperature (i.e., T_C with $T_C < T_H$) is increased. For typical water-cooled chillers, Figure 9.13 can be used to evaluate the improvement in the COP of a chiller when the leaving water temperature is increased from 4.5°C (40°F). Similarly Figure 9.14 can be used to estimate the effect of

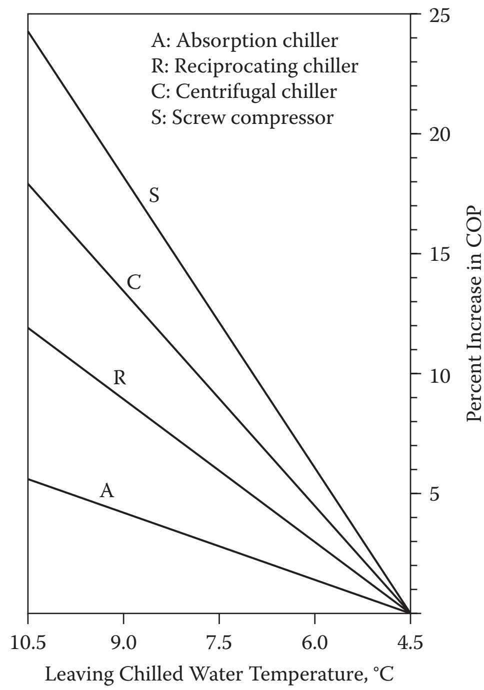


FIGURE 9.13 Effect of leaving chilled water temperature on the chiller COP.

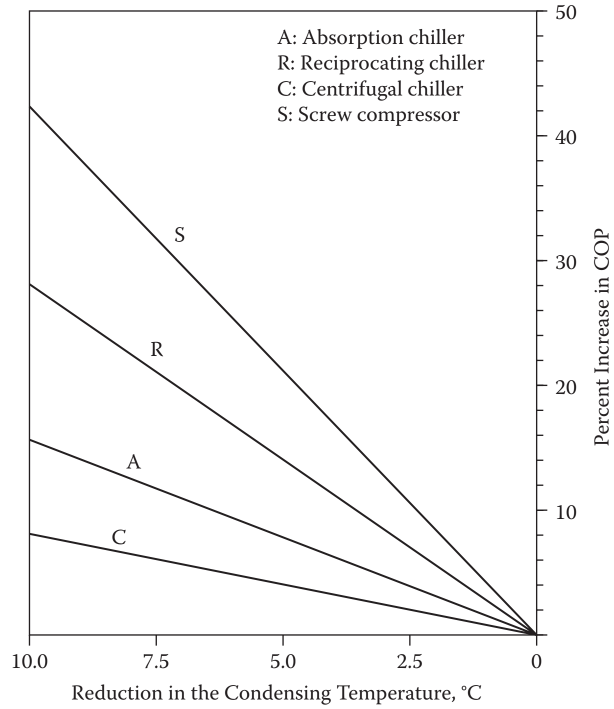


FIGURE 9.14 Effect of a reduction in the condensing temperature on the chiller COP.

reducing the condenser water temperature on the COP of the cooling system. Both Figures 9.13 and 9.14 represent typical chiller performance based on manufacturers' data (LBL, 1980).

Example 9.3

A centrifugal chiller with a capacity of 500 kW and with an average seasonal COP of 4.0 operates with a leaving chilled temperature of 4.5°C. Determine the cost savings incurred by installing an automatic controller that allows the leaving water temperature to be set 2.5°C higher on average. Assume that the number of equivalent full-load hours for the chiller is 1,500 per year and that the electricity cost is $0.07/kWh.

Solution

Using Figure 9.13, the increase in the COP for a centrifugal chiller due to increasing the leaving chilled water temperature from 4.5°C to 7.0°C is about 8 percent. The energy use savings can be calculated using Eq. (9.12) with $SEER_e = 4.0$; $SEER_r = 4.0 * 1.08 = 4.32$; $N_{h,C} = 1{,}500$, $Q_C = 500$ kW, and $LF_C = 1.0$ (assume that the chiller is sized correctly):

$$\Delta E_C = 500kW * 1500hrs/yr * 1.0 * \left(\frac{1}{4.0} - \frac{1}{4.32}\right) = 13{,}890kWh/yr$$

Therefore, the energy cost savings is $970.

9.7.3 Alternative Cooling Systems

There are a number of alternative systems and technologies that can be used to reduce and even eliminate the cooling loads on existing cooling systems. Among the alternative systems and technologies are:

- *Water-side economizers* can be used when the outdoor conditions are favorable. Instead of operating the chillers to provide air conditioning, water can be cooled by using only cooling towers and circulated directly to the cooling coils either through the normal chilled water circuit or through heat exchangers.
- *Evaporative cooling* is a well-established technique that uses water sprays or wetted media to cool supply air either directly or indirectly allowing temperatures to approach the wet-bulb temperature of the ambient air. Direct evaporative cooling humidifies the air supply when its temperature is reduced whereas indirect evaporative cooling is performed through an air-to-air heat exchanger with no humidity addition. Typically, indirect evaporative cooling is less effective and more expensive than direct evaporative cooling. In addition to some energy use (mostly electric energy to power fans), both evaporative cooling methods consume a significant amount of water. Evaporative cooling can be used to reduce the cooling load for a conventional mechanical air-conditioning system in climates characterized by dry conditions either throughout the year or during limited periods. The average COP of evaporative cooling systems can be in the range of 10 to 20 depending on the climate (Huang, 1991).
- *Desiccant cooling* is basically evaporative cooling in reverse because the air temperature is increased but its humidity is reduced. The dried air is then cooled using heat exchangers in contact with ambient air. Finally, the air is further cooled using evaporative cooling. A source of heat is needed to regenerate the desiccant after it has absorbed water from the air. Desiccant cooling has been used mostly in industrial applications and is less commonly used in the commercial sector. However, future developments of gas-fired desiccant dehumidification systems are expected especially to condition outside air required for ventilating office buildings (Tuluca, 1997).
- *Subcooling* of the refrigerant typically increases cooling capacity and can decrease the compressor power and thus increase the overall energy efficiency of the cooling system. Subcooling requires the addition of a device such as a heat exchanger to decrease the enthalpy of the refrigerant entering the evaporator, resulting in an increase in cooling capacity. There are currently three common subcooling technologies. The first technology uses a suction-line heat exchanger of the vapor compression system as a heat sink. The second technology involves a second mechanically driven vapor compression cycle coupled with the main cycle using a subcooling heat exchanger located downstream from the condenser. The third technology requires an external heat sink such as a small cooling tower or ground source water loop. Refrigerant subcooling has long been used in low and medium temperature refrigeration systems (Couvillion et al., 1988). Currently, some manufacturers of packaged and split-systems for air conditioners and heat pumps are integrating subcooling devices with their systems using alternative refrigerants (such as R-134a).

9.8 Summary

This chapter provided a brief analysis of the type and energy efficiency of central cooling equipment and chilled water distribution systems currently available in the United States and other countries. In addition, cost-effective energy conservation measures have been proposed with some specific examples to illustrate the energy savings potential for some of the cooling equipment retrofits. Currently, energy retrofits of cooling systems include improvement in controls, replacement with more energy-efficient systems, and use of alternative cooling systems. In the future, it is expected that higher efficiency cooling equipment will be available as well as other innovative air-conditioning alternatives such as desiccant cooling systems.

Problems

9.1 Consider an 800-ton chiller operating for 1,600 equivalent full-load hours per year. The chiller is a centrifugal chiller that is rated at 0.72 kW/ton. Determine the energy and cost savings when the following operating changes have been made:

(a) The condensing temperature is reduced in its current setting by 5°F, 10°F, or 15°F.

(b) The leaving chilled water temperature is increased from 40°F to 45°F and from 40°F to 50°F.

Assume the electricity cost is \$0.08/kWh.

9.2 Redo Problem 9.1 for an 800-ton absorption chiller using 10 lbs of steam per ton. The cost of steam is \$10/1,000 lbs.

9.3 Consider switching from one large chiller (1,000 ton) to four chillers (each rated at 250 ton). Using the normalized savings shown in Figure 9.11, estimate the annual energy savings for using a multichiller system in a building located in Chicago, Illinois, if the large chiller is operated 5,000 equivalent full-load hours per year and has a rated COP of 4.0.

9.4 A 300-ton chiller with 1.2 kW/ton average seasonal efficiency is operated 3,500 hours per year with an average load factor of 70 percent. This chiller needs to be replaced by either chiller A or chiller B. The manufacturers indicated that the IPLV for chiller A is 4.14 and for chiller B is 4.69.

(a) Estimate the energy cost savings due to replacing the existing chiller with chiller A or with chiller B. The electricity cost is \$0.09/kWh.

(b) The cost differential between chiller B and A is \$25,000. Determine if it is cost-effective to replace the exiting chiller with chiller B rather than chiller A. For this question, a simple payback analysis can be used.

(c) If the discount rate is 6 percent, determine the electricity price at which it is more cost-effective to replace the existing chiller with chiller A rather than chiller B. Assume a life cycle of ten years for both chillers.

9.5 Consider the water distribution system below. Assume the pump efficiency of 0.70:

(a) Determine the primary pump size (in hp) and the valve settings to balance the system.

(b) Determine primary and secondary pump sizes as well as the new valve settings for the optimal primary/secondary distribution system

(c) Estimate the total energy cost savings from the retrofit from existing system to the optimal distribution system. Assume that the system is operated 6,000 hours per year, that the cost of energy is \$0.10/kWh, and that the efficiency of all pump motors efficiency is 90 percent.

10

Energy Management Control Systems

10.1 Introduction

Currently, almost all new buildings have some control systems to manage the operation of various building equipment including HVAC systems. More elaborate control systems can simultaneously operate several pieces of mechanical and electrical equipment dispersed throughout the facility. In particular, these energy management control systems can be used to reduce and limit the energy demand of the entire facility. In the last decade, most of the advances in HVAC equipment are due to modern electronic controls which are now cheap, flexible, and reliable.

The development of energy management and control systems (EMCS) is mostly attributed to the introduction of computerized building automation systems. In fact, energy management represents one of several tasks performed by an integrated building automation system (IBAS). Among other tasks of the IBAS include fire safety, vertical transportation control, and security regulation. Advanced IBAS include logic for interaction among lighting, HVAC, and security systems. For instance, if an automated occupancy sensor detects the presence of people in specific spaces during late hours (during night or weekends), the information can be used to adjust indoor temperature (for comfort) and to reinstate elevator service (to ensure that people can leave the building). Moreover, EMCS can provide facility operators with recommendations on maintenance needs (such as lighting fixture replacement) and alarms for equipment failures (such as motors when they burn out).

The use of energy-efficient equipment does not always guarantee energy savings. Indeed, good management of the operation of this equipment is a significant factor in reducing whole building energy use. Generally, building energy loads are continuously changing with time due to fluctuations in weather and changes in equipment use and occupancy. Thus, effective energy management requires knowledge of the facility loads. Two approaches are typically applied:

1. *Load tracking:* The operation of equipment is modulated to respond to the actual needs in the facility. As an example, the compressor in a centrifugal chiller may change speed to match the cooling demand. The actual needs of a facility can be determined by continuous monitoring. As an example, the load on the chiller can be estimated if the chiller water flow and chilled water supply and return are monitored.
2. *Load anticipation:* In some applications, the needs of a facility have to be predicted to be able to modulate the operation of equipment adequately. For instance, in cooling plants with a thermal energy storage system, it is beneficial to anticipate future cooling loads to be able to decide when and how much to charge and discharge the storage tank. Load prediction can be achieved by analyzing the historical pattern variations of the loads.

Using monitored data and other parameters characterizing the building, energy control systems enable operators and managers to operate HVAC and lighting systems efficiently to maintain the comfort level. In the following sections, building energy control systems and some of their applications are presented and discussed.

10.2 Basic Control Principles

10.2.1 Control Modes

Control systems are used to match equipment operation to load requirements by changing system variables. A typical control system includes four elements as briefly described below:

1. The controlled variable is the characteristic of the system to be controlled (for instance, the indoor temperature is often the controlled variable in HVAC systems).
2. Sensors that measure the controlled variable (for instance, a thermocouple can be used to measure indoor temperatures).
3. Controllers that determine the needed actions to achieve the proper setting for the controlled variable [for instance, the damper position of the VAV (variable-air-volume) box terminal can be modulated to increase the air supply in order to increase the indoor temperature of the zone if it falls below a set-point].
4. Actuators are the controlled devices that need to be activated in order to complete the actions set by the controllers (to vary the air supplied by a VAV box, the position of the damper is changed by an actuator through direct linkages to the damper blades).

Generally, two categories of control systems can be distinguished including closed-loop an.d open-loop systems. In a closed-loop system (also known as a feedback control system), the sensors are directly affected by (and thus sense) the actions of the actuators. A typical control of a heating coil is an example of a closed-loop system. However, in an open-loop system (also called a feedforward control system), the sensors do not directly sense the actions of the controllers. The use of a timer to set the temperature of the heating coils would be an example of an open-loop system inasmuch as the time may not have a direct connection with the thermal load on the heating coils.

Figure 10.1 shows the various components and terms discussed above as well as an equivalent control diagram for a closed-loop control system for a heating coil.

Each control system can use different control modes to achieve the required objectives of the control actions. Four control modes are commonly used in operating HVAC systems. These four control modes are:

1. Two-position: This control mode allows only two values (on–off or open–closed) for the controlled variable and is best suited for slow-reacting systems. Figure 10.2(a) shows the effect of two-position control on the time variation of the controlled variable (such as the air temperature due to the on–off valve position in a heating coil). In order to avoid rapid cycling, a control differential can be used. Due to the inherent time lag in the sensor response and to the thermal mass of the HVAC system, the controlled variable fluctuates with an operating range (called operating differential) with higher amplitude than the control differential. Thus, the operating differential is always higher than the control differential as illustrated in Figure 10.2 (b).

   Examples of two-position controls are domestic hot-water heating, residential space-temperature controls, and HVAC system electric preheat elements.
2. Proportional: This mode has a linear relationship between the incoming sensor signal and the controller's output. The relationship is established within an operating range for the sensor signal.

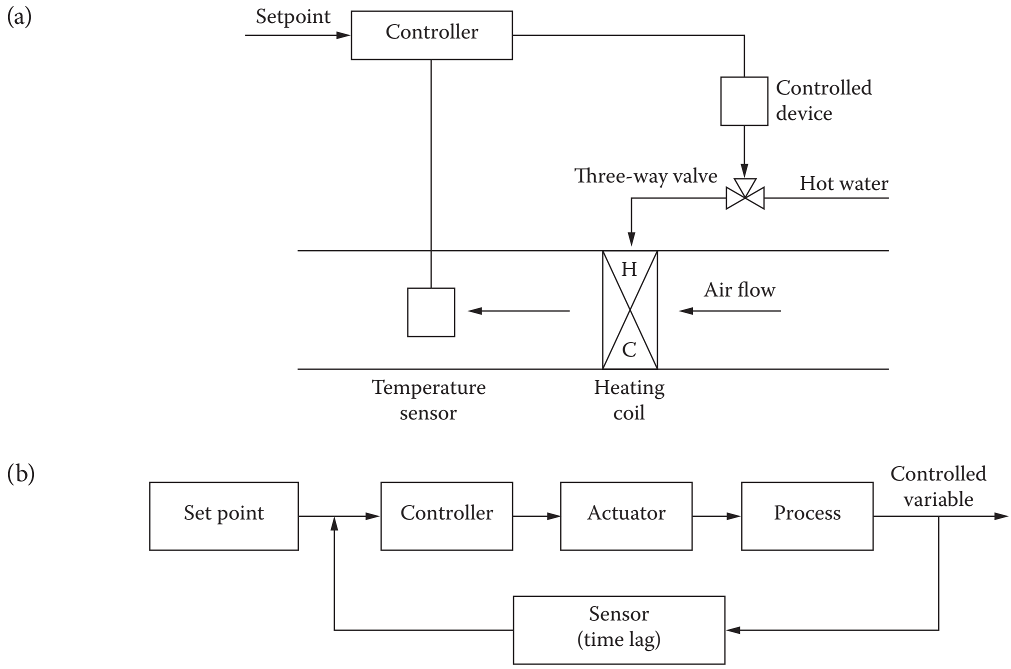


FIGURE 10.1 Typical representations for a heating coil control system: (a) basic closed-loop control for a heating coil; (b) equivalent control diagram for the heating coil.

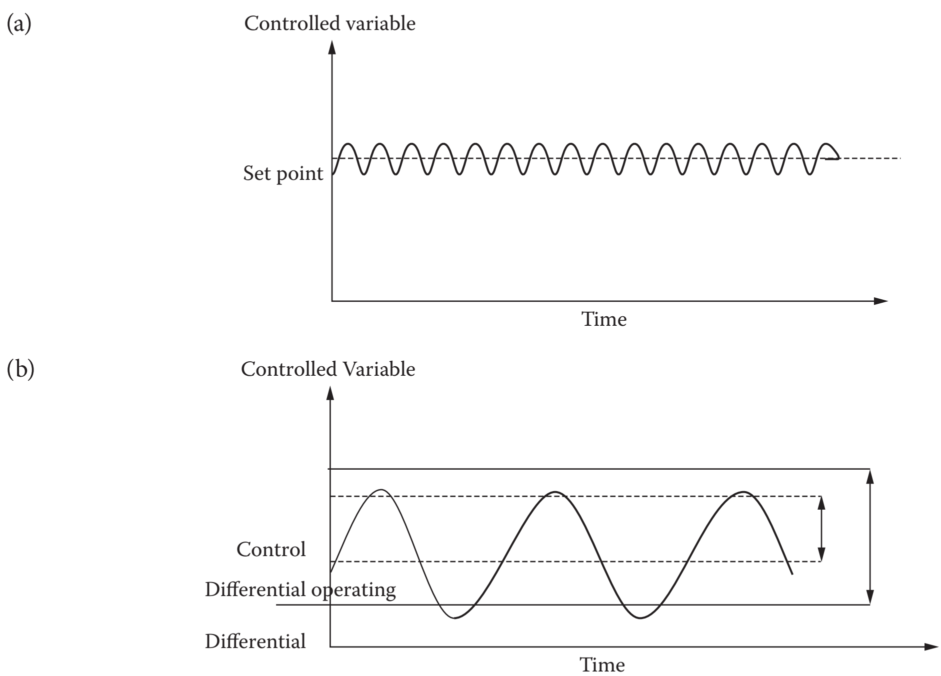


FIGURE 10.2 Effect of two-position control on the time variation of a controlled variable: (a) two-position action when no control differential is used (rapid cycles); (b) two-position action with a control differential.

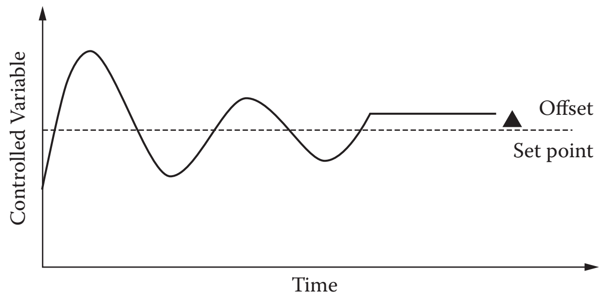


FIGURE 10.3 Proportional control effect on the time variation of a controlled variable.

The set-point of a proportional controller is the sensor input which results in the controller output being at the midpoint of its range. Mathematically, the controller output u is given by the following equation for a proportional control:

$$u = K_p e + u_0 \tag{10.1}$$

The offset or error e is the difference between the set-point and the value of the controlled variable. The proportionality constant K_p is called the proportional gain constant. The controller bias u_0 is the value of the controller output when no error exists.

As depicted by Eq. (10.1), the proportional control is not capable of reducing the error because an error is required to produce any controller action. Therefore, the controlled variable fluctuates within a throttling range as depicted in Figure 10.3.

It should be noted that when the gain constant is very large, an unstable system can be obtained. Example 10.1 shows how the proportional gain constant can be determined.

Example 10.1

A hot water heating coil has a set-point of 35°C with a throttling range of 10°C. The heat output of the coil varies from 0 to 50 kW. Assuming that a proportional controller is used to maintain the air temperature set-point, determine the proportional gain for the controller and the relationship between the output air temperature and the heat rate provided by the coil. Assume steady-state operation.

Solution

Using Eq. (10.1), the relationship between the heat rate Q, and the error in the air temperature at the coil outlet can be put in the form of:

$$Q = K_p\,(T_{\text{set-point}} - T_{\text{air}}) + Q_o$$

(i) When the heat rate $Q = Q_{min} = 0$ kW, the coil outlet air temperature is $T_{air} = T_{min} = 35°C - 5°C = 30°C$.
(ii) When the heat rate $Q = Q_{max} = 50$ kW, the coil outlet air temperature is $T_{air} = T_{max} = 35°C + 5°C = 40°C$.

The proportional gain K_p can be determined as follows:

$$Q_{max} - Q_{min} = K_p(T_{min} - T_{max})$$

or

$$K_p = [Q_{max} - Q_{min}]/[T_{min} - T_{max}] = -50 \text{ kW}/10\ °\text{C} = -5 \text{ kW}/°\text{C}$$

Similarly, the constant Q_o can be determined from

$$Q_{min} = K_p(T_{setpoint} - T_{min}) + Q_o$$

or

$$Q_o = Q_{min} - K_p(T_{set\text{-}point} - T_{min}) = 0 + 5 \text{ kW}/°\text{C} * (35°\text{C} - 30°\text{C}) = +25 \text{ kW}$$

Therefore, the relationship between the heat rate output and the air temperature for the heating coil is:

$$Q = -5\ (T_{set\text{-}point} - T_{air}) + 25$$

Thus, as long as the heat rate is different from $Q_o = 25$ kW, the quantity ($T_{set\text{-}point}$-T_{air}) which is the error in the proportional control equation cannot be equal to zero.

Generally, proportional controllers are used with slow stable systems that have small offset.

3. *Integral:* This control mode is typically incorporated with a proportional control mode to provide an automatic means to reset the set-point in order to eliminate the offset. The combination of proportional and integral actions is called "proportional-plus-integral" or simply PI control. Mathematically, the PI control can be expressed as follows:

$$u = K_i \int e.dt + K_p e + u_0 \tag{10.2}$$

where K_i is the integral gain constant (also known as the reset rate) and has the effect of adding a correction to the controller output whenever an error exists. For HVAC systems, a typical K_p/K_i ratio is less than 60 minutes.

The PI control can be applied to fast-acting systems that require large proportional bands for stability. Typical applications include mixed-air controls, heating or cooling coil controls, and chiller-discharge controls.

4. *Derivative:* This control action is used to speed up the response of the system in case of sudden changes. The derivative control mode is included in a combination of proportional-plus-integral-plus

derivative (PID) control modes for fast-acting systems that tend to be unstable such as duct static-pressure controls. The mathematical model for the PID control is given by Eq. (10.3):

$$u = K_d \frac{de}{dt} + K_i \int e.dt + K_p e + u_0 \tag{10.3}$$

where K_d is the derivative gain constant. The derivative term generates a corrective action proportional to the time rate of change of the error. The ratio K_d/K_p is typically less than 15 minutes for most HVAC applications. If the system has a uniform offset, the derivative term has little effect. The use of PID controls is typically less common than the PI controls for HVAC systems inasmuch as no rapid control responses are needed.

To illustrate the action of P, PI, and PID control modes, Figure 10.4 compares the response of the system to an input step change. As expected the proportional control results in an offset and the controlled variable does not reach the set-point. The correction term due to the PI control slowly forces the controlled variable to reach the set-point value. Finally, the derivative term of the PID control provides a faster action to allow the controlled variable to attain the set-point.

In addition to these conventional control modes, other intelligent controllers have been investigated in various engineering fields including HVAC equipment controls as discussed in the following section.

10.2.2 Intelligent Control Systems

In the future, it is expected that intelligent control systems will be commonly available to operate HVAC systems. A number of research studies are applying intelligent system methodologies so

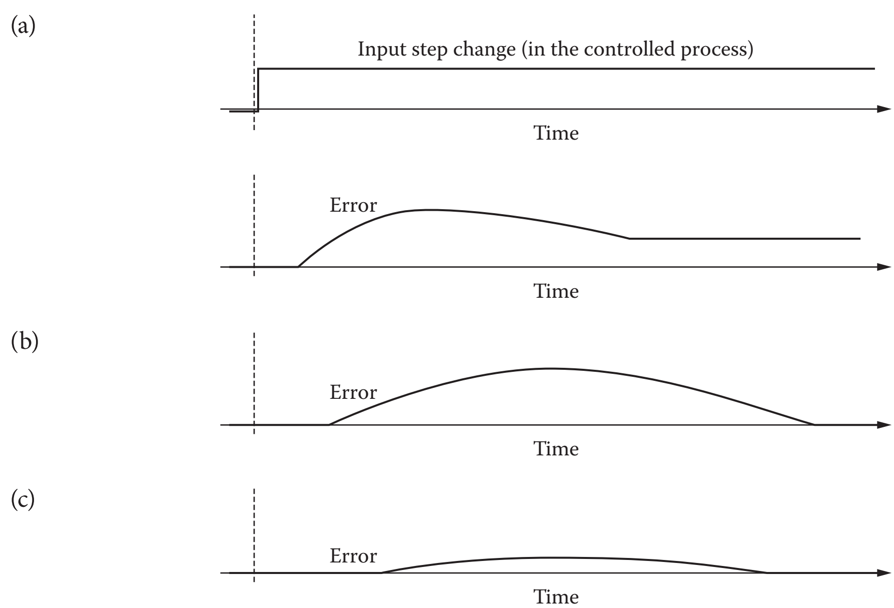


FIGURE 10.4 Comparison of the reaction of three control modes to an input step change: (a) proportional (P) control action; (b) proportional-plus-integral (PI) control action time; (c) proportional-plus-integral-plus-derivative (PID) control action time.

that control systems have humanlike capabilities such as pattern recognition, adaptation, learning, reasoning, and associative memory to operate complex systems that conventional control techniques cannot handle. Various intelligent control environments have been considered for HVAC systems including:

- *Expert controls:* Rule-based expert systems are based on humanlike reasoning and can be powerful in solving practical engineering problems. Indeed, expert control systems have been applied to self-tuning, structured knowledge-based adaptive control, fault diagnosis, and scheduling and planning. However, rule-based expert systems are not yet suitable to process numerical knowledge and thus to provide precise solutions. The development of an expert system environment that can deal with both qualitative and quantitative knowledge, and to automatically explore knowledge, is still in the realm of future expert control research.
- *Fuzzy controls:* Since the development of fuzzy logic in the 1960s, fuzzy control has been one of the most attractive strategies in controlling complex systems with imprecise or uncertain knowledge of system information and behavior. Fuzzy logic can be applied to system modeling, estimation, optimal control, and adaptive control with the requirements of only fuzzy system knowledge and input/output data. Applications of fuzzy logic in expert systems are becoming attractive for establishing intelligent expert control systems with fuzzy knowledge representation and fuzzy reasoning.
- *Artificial Neural Networks:* Similar to biological neural networks, artificial neural networks consist of large number of simple nonlinear processing elements, typically called nodes or neurons, which are interconnected with adjustable weights. Well-trained neural network models can provide both qualitative and quantitative knowledge and have powerful functions in learning and self-organization. These features make neural networks more suitable in dealing with numerical data than expert systems.

10.2.3 Types of Control Systems

To achieve the actions of the control systems discussed above, several types of energy sources are used. In particular, the following types of control systems are used in HVAC applications:

- Pneumatic devices are used with low-pressure compressed air at 0 to 20 psig. Pneumatic systems are common in older installations.
- Electric devices using 24 to 120 volts or even higher voltage sources.
- Electronic devices with low direct current voltages varying from 0 to 10V. These devices are being installed in new commercial buildings especially with direct digital control (DDC) systems.
- Hydraulic systems when large forces are required with pressure larger than 100 psi.
- Self-generated energy derived from the change of state of the controlled variable or from the energy available in the process plant.

For HVAC retrofit applications, it is recommended that direct digital control systems be considered. Indeed, currently developed DDC systems use the latest digital technology including features such as intelligent controllers, high-speed communication networks, and sophisticated control algorithms. All these features allow more energy-efficient control strategies to be implemented. Moreover, digital devices present additional advantages compared to pneumatic or electronic devices as outlined below:

- Little or no maintenance is required for digital devices.
- Calibration of digital devices can be performed through remote instructions issued over a network. Some digital devices have the advantage of being continuously self-calibrating.
- Better accuracy is obtained from digital controls compared to pneumatic or electric devices.

Although DDC devices have been used since the 1980s, it is only recently that manufacturers have developed systems that house interposing devices (such as relays, transducers, and hard-wired logic) in the same package with the electronic devices. These new DDC systems are suitable for retrofitting applications because it is now economical to convert existing pneumatic and electric analog controls to DCC systems. The best candidates for DDC retrofitting are air-handling units (AHUs), heat exchangers, distribution pumps, and cooling towers. In general, the larger the equipment size, the faster the payback period is. Replacing the controls for small HVAC equipment such as package unitary systems (including unit ventilators, heat pumps, and fan coils) may not be cost-effective.

Retrofit of pneumatic controllers can be made using electronic-to-pneumatic (E/P) transducers to convert signals so electronic and pneumatic control components can be combined in the same control loop. For instance, the controller and the sensor in an existing pneumatic HVAC control system can be converted into electronic devices and the actuator can remain pneumatic. The pressure output of the E/P transducer should match the electric signal. The use of E/P transducers allows retrofit of control systems with minor interruption in the operation of the controlled system.

10.3 Energy Management Systems

10.3.1 Basic Components of an EMCS

To control and operate equipment for heating, ventilating, and air conditioning, or for lighting and process equipment, an energy monitoring (or management) and control system can be used. A typical EMCS is configured into a network that includes sensors and actuators at the bottom level, microprocessor controllers in the middle, and a computer at the top with a modem to allow remote monitoring and control of the building energy systems. For a typical commercial building, an EMCS system can be cost-effective in reducing energy use for HVAC and lighting systems.

Energy management is often just one element of an integrated building automation system which regulates security, fire safety, lighting, HVAC systems, and elevators. Advanced IBAS systems include logic for interaction among various systems such as HVAC, lighting, and security systems. Indeed, the automated occupancy count information obtained for different spaces in a facility can be used to adjust indoor temperature settings, reduce or turn off lights, and ensure elevator operation.

The size of an EMCS system is typically classified based on the total number of points connected to a system. Five size categories are generally considered for EMCS systems:

1. Large EMCS systems with more than 2,000 points
2. Medium EMCS systems with 500 to 2,000 connected points
3. Small EMCS systems with 200 to 1,000 points
4. Small centralized EMCS systems with 50 to 500 points
5. Micro-EMCS systems with less than 100 points

A typical EMCS system is depicted in Figure 10.5 and includes a central control unit (CCU), a processing memory, storage devices, input–output devices, a central communications controller (CCC), data transmission medium (DTM), a field interface devices (FID), multiplexers, instruments, and controls. A brief description of the major components of an EMCS is presented in the following sections.

- The central control unit generally consists of a computer with memory for the operating system software, command software, and implementation of application algorithms. In particular, computations and logical decision functions for central supervisory control and monitoring are performed by the CCU. However, data and programs are stored in and retrieved from the memory or storage devices such as magnetic tape and disk systems. Typically, the CCU has input–output (I/O) ports for specific equipment such as printers and terminals.
- The central communications controller is typically a computer with enough memory to execute specific programs required to reformat, transfer, and perform error checks on data coming

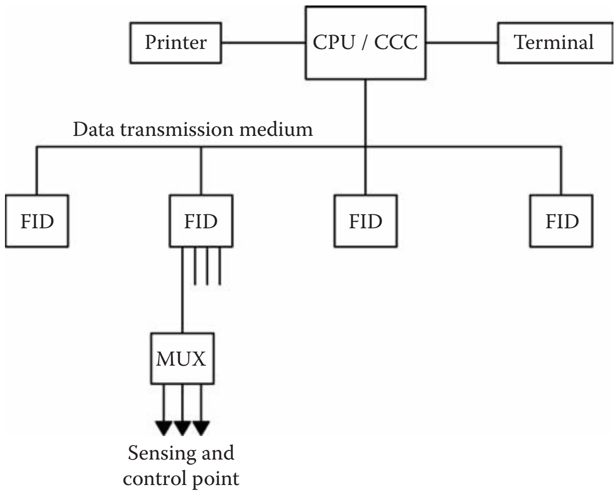


FIGURE 10.5 Typical EMCS components and configuration.

from the CCU or the field interface devices. The CCC may have backup capabilities in case of CCU failure.

- Field interface devices generally consist of computer devices with memory, I/O, communications, and power supply. The FIDs record the monitored and controlled data, perform calculations and logical operations, and accept and process system commands. They should be capable of operating in case of CCU failure.
- A multiplexer (MUX) is a device that communicates between the data environment and its associated FID. The MUX is functionally part of the FID and can thus be in the same enclosure.
- The data transmission medium is the communications link that allows the transfer of data between the CCU and its associated FIDs such as telephone lines, optical fibers, or coaxial cables.

10.3.2 Typical Functions of EMCS

Several control operations and functions can be performed by an EMCS including but not limited to:

1. Equipment operation such start–stop, on–off, and open–close controls.
2. Alarm functions such as abnormal equipment status or high or low parameter values (temperature, pressure, refrigerant level, etc.).
3. Computer programming and table lookup functions for energy management and equipment use optimization including enthalpy economizer controls, chiller plant optimization, load shedding based on demand monitoring, and lighting control by zones.
4. Monitoring of operation conditions such as temperatures, pressures, and energy end-uses.

Other functions and applications of EMCS are being developed and implemented in some facilities. Some of these applications are discussed in the following sections.

EMCS can be used with expert systems to make intelligent operating decisions based on stored and accumulated knowledge. For instance, integrated EMCS/expert systems can be used to diagnose faults and inefficiencies in HVAC and lighting systems. Typically, an EMCS/expert system would be programmed with basic information about the building equipment and design operation characteristics. Over time, actual operating data would be recorded and used to train the expert system and thus improve its diagnostic accuracy. If a fault were detected, the EMCS/expert system would notify the building operator of the probable problems and suggest remedial actions or simply automatically initiate countermeasures in the case of serious problems. As an example, if the indoor temperature for a specific space is noted to be rising significantly beyond the throttling range, the EMCS/expert system would examine all the relevant sensor readings (such as airflow rates, fan energy use, and cooling water flow and temperature). Then the EMCS/expert system would indicate using its knowledge base that the probable causes for the problem are faulty fan operation or leakage in chilled water pipe (70 percent faulty fan and 30 percent chilled water pipe leakage).

Another function of EMCS that has been implemented in several buildings especially in the last decade is the maintenance of acceptable indoor air quality (IAQ) levels. Indeed, the cost of sensors for monitoring air pollution compounds such as carbon monoxide (CO), carbon dioxide (CO_2), and volatile organic compounds (VOCs) has decreased sufficiently to be incorporated in commercial and institutional buildings. For instance, CO_2 demand-controlled ventilation has been implemented in several spaces including classrooms, conference rooms, theaters, and auditoriums.

10.3.3 Design Considerations of an EMCS

When an EMCS is recommended for a facility, it is important to consider some practical issues to ensure successful design and operation. In particular, it is recommended to:

- Allow redundancy into the control systems in case of system failure. In particular, it is important that the local control devices are able to manage and operate the system for a reasonable period of time when the central control unit malfunctions or needs servicing.
- Provide clear information about the system operation. For instance, color graphic displays are preferred to numerical data for the use of facility operators and managers.
- Perform a thorough commissioning of all the components and functions of the EMCS under various operating conditions (peak cooling and peak heating modes as well as part-load conditions).
- Train the operating staff to use the EMCS to their best advantage. In particular, the operators should understand the EMCS capabilities and benefits.

The selection of an EMCS for a given facility typically depends on the required functions and on economic considerations. The desired functions and controls from an EMCS are based on several aspects including the building type, HVAC system zoning, occupancy profiles, accuracy requirements, and the objectives of the owners. For instance, owner–developers may be more interested in having an EMCS with the lowest first cost rather than a system that provides a high-quality environment which may be one of the main objectives of owner–occupants. Moreover, high-rise buildings tend to have centralized systems with builtup fans and associated controls. On the other hand, low-rise buildings typically use package rooftop fan systems with local controls.

The cost–benefits of the EMCS can be evaluated after identification of all the desired controls. Some of the important factors that affect the cost of an EMCS include:

- System size and number of control points
- Degree of automation in the control functions
- Accuracy requirements for sensors and controls

To ensure that the control systems conserve energy and save operating costs, it is important to follow basic principles in their design including:

1. Energy-consuming equipment should be operated only when needed. For instance, heating temperature set-points should be set back during unoccupied periods. The heating equipment should be operated only to maintain these setback temperatures (typically between 50°C and 55°C to prevent freezing damage for various components of the HVAC system).
2. Simultaneous heating and cooling should be avoided. Proper zoning and HVAC system selection can minimize—if not eliminate—the need for providing heating and cooling at the same time.

3. The outdoor air intake should be controlled. In the United States, only minimum requirement ASHRAE Standard 62-1999 for ventilation needs to be supplied to the building when no economizer cycles are used.
4. The heating and cooling should be provided efficiently. In particular, only actual heating and cooling load requirements should be met. In addition, free cooling/heating or low-cost energy sources should be considered first to maintain comfort within the building.

Finally, it should be noted that energy control systems should be designed to be simple and easy to operate and maintain.

10.3.4 Communication Protocols

A communication protocol consists of a set of rules that have to be applied to exchange data between two parts of an EMCS system. Most manufacturers of building automation systems have their proprietary communication protocols. Therefore, building owners are forced to purchase equipment only from the original manufacturer if they want to expand the existing automation system. Due to the lack of interoperability between communication protocols from various manufacturers, it is almost impossible to take advantage of a facilitywide approach to control optimization and energy savings. A single manufacturer does not provide all the best possible control strategies for electrical demand limiting, heating and cooling optimization, and similar energy-saving options. Moreover, several control subsystems in a building including HVAC controls, lighting/daylighting interface, fire alarm and life safety, security, and communication systems are generally manufactured by different companies. Integrating all these subsystems is a difficult task without a common communication protocol.

As a solution to the limitations and difficulties inherent to proprietary protocols, a number of open communication protocols have been developed in recent years. These open protocols have some interoperability capabilities and thus allow building owners and managers to keep the door open to competition on any future expansion projects. One of the most widespread and widely accepted open protocols is the BACnet, a data communication protocol for Building Automation and Control networks (Bushby and Newman, 1991). BACnet is a nonproprietary open protocol standard that supports various communication networks ranging from high-speed Ethernet local area networks (LANs) to low-cost networks. Because ASHRAE developed BACnet, no one specific company or consortium has an advantage or an influence on future development of the standard. Any changes for BACnet are published for public review and comment after discussions on an ASHRAE open committee that includes representatives from industry, academia, and government.

To design a BACnet device, a manufacturer needs to identify the BACnet objects and services required to achieve the intended functionality for the device. A BACnet object is a standard data structure defined with a set of properties and data types. In its current version, the BACnet standard defines 20 objects such as loops, tables, schedules, commands, and programs. BACnet services are the programmed actions that use the data objects to achieve the function of the device. Services, defined by the current version of the BACnet standard, include alarm and event services (to notify of any alarm and event), file access services (to read and write files), object access services (to read or write the properties of objects), remote device management services (to troubleshoot and maintain devices), and virtual terminal services (to allow interaction between a terminal and the device). Moreover, the BACnet device has to conform to a set of specifications using a series of conformance classifications. Each conformance classification adds functional services to the device. Thus, each BACnet device design would have a protocol implementation conformance statement (PICS) prepared by the manufacturer to identify the BACnet options available in the device.

In the last few years, some manufacturers have already developed BACnet control devices. However, the integration and the application of these devices in real installations have not yet been documented.

10.4 Control Applications

Energy management and control systems can be used to perform several functions and tasks. Early building automation systems were limited to simple functions such as simple on–off programming including duty cycling and load shedding. Currently more complex functions and controls can be achieved by EMCS. Some of these controls are now available in standard program packages such as:

- Duty cycling for motor loads to provide sequential shutdown for short periods for equipment such as supply fans of small air-handling units. However for large motors, frequent duty cycling is not generally recommended due to adverse effects on belts, bearings, and motor drives.
- Demand shedding to limit electrical loads. However, it is generally difficult to identify loads that can be shed without affecting building performance especially for HVAC systems that are not generally needed when electrical demand is high. Therefore, equipment not related to HVAC systems such as plug loads or lighting fixtures are typically considered for demand shedding. In particular, programmable lighting controls can be combined with other energy-saving lighting measures including dimming and occupancy sensors.
- Partial space conditioning to allow systems to cool only a small portion of the building by controlling supply air dampers serving various zones. With the use of variable frequency drives, it is now possible to adjust fan speed to match small loads for most central fan systems.

When demand charges are a significant part of the electric utility bills, EMCS can be applied to reduce the electric demands, especially cooling applications. Indeed, accurate and reliable controls are required to ensure cost savings because even one mistake made in one month can increase utility bills not only for the month but also for future billing periods. Therefore, software capable of storing past data and anticipating future effects related to probable weather and occupancy conditions is required to effectively operate building energy systems. Some of the demand-limiting strategies that require effective control software are described below:

- *Precooling of building thermal mass:* The storage capabilities of a building structure can be used to shift a portion of on-peak cooling loads to off-peak periods and thus reduce electrical demand and energy charges. This measure can be achieved by precooling the building thermal mass. Studies (Braun, 1992; Morris, Braun, and Treado 1994) have shown that when an effective control strategy is used, up to 35 percent in energy cost savings can be achieved when an effective control strategy is used to determine when and how much to precool the building. Some additional savings in operating costs can be achieved if free cooling is used for precooling when cool outdoor air temperature is introduced to the building during the night using the air-handling fans. In some cases, the cost of operating air-handling fans may be less than the reduction in operating costs for mechanical cooling during occupied periods.
- Precooling building thermal mass is an example of the application of the EMCS to reduce operating costs. Precooling of the building thermal mass can be effective to lower building operating costs. This strategy can have a large impact when chillers have high loads during periods of high occupancy and high outdoor temperatures (which typically coincide with on-peak periods in rate structures). By reducing the on-peak cooling load it is possible to reduce chiller energy use during these critical periods, thereby reducing energy costs.
- Based on long-term simulation analysis, the annual energy cost savings associated with precooling has been estimated for various time-of-use utility rates (Morgan and Krarti, 2006). For time-of-use rates, the on-peak to off-peak ratio for energy and demand charges is defined as follows:

R_e: Ratio of on-peak to off-peak energy charges:

$$R_e = \frac{PeakEnergyRate(\$/kWh)}{Off - PeakEnergyRate(\$/kWH)} \tag{10.4}$$

R_d: Ratio of on-peak to off-peak demand charges:

$$R_d = \frac{PeakDemandRate(\$/kW)}{Off - PeakDemandRate(\$/kW)} \tag{10.5}$$

- Figures 10.6 and 10.7 show the variation of the annual energy cost savings for typical office buildings in four U.S. locations due to a 4-hr precooling period as a function of R_d and R_e, respectively. The office building has a heavy thermal mass of 105 lbm/ft2 (513.7 kg/m2) and the time-of-use rate has an 8-hr on-peak period (Morgan and Krarti, 2006).

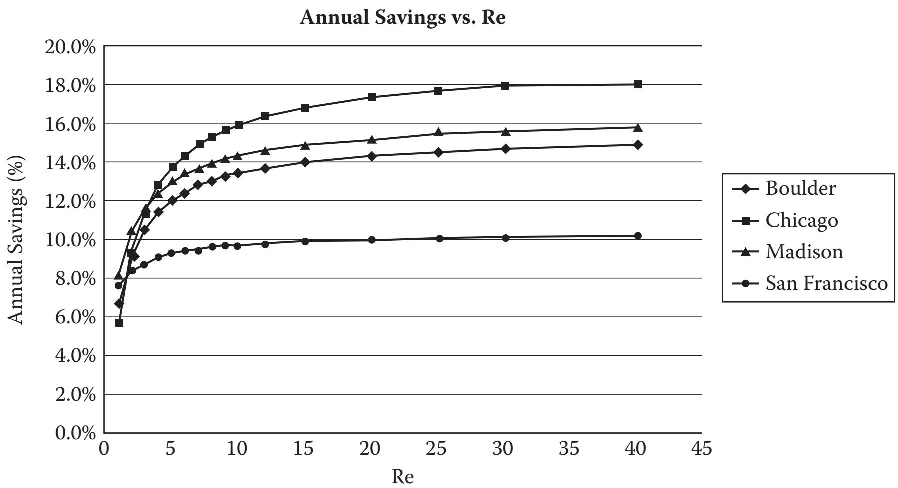


FIGURE 10.6 Annual energy cost savings due to precooling relative to conventional controls as a function of R_e.

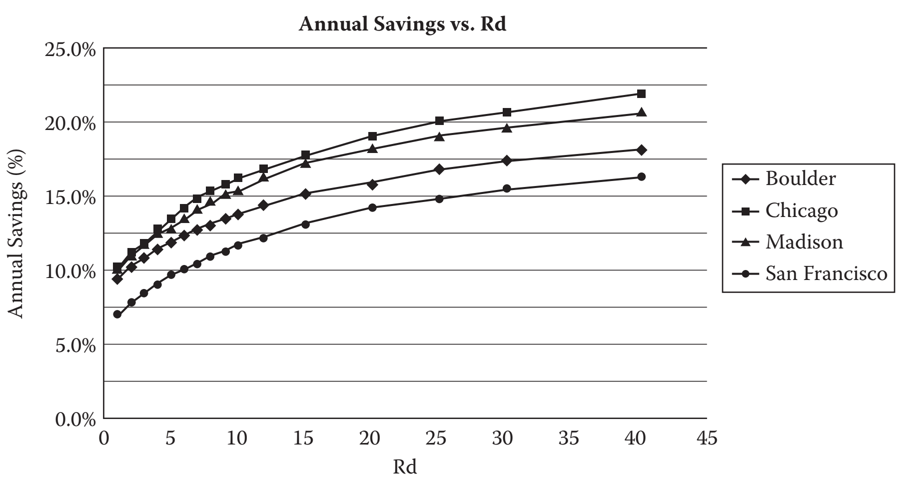


FIGURE 10.7 Annual energy cost savings due to precooling relative to conventional controls as a function of R_d.

- *Thermal energy storage (TES) systems:* Cooling energy can be stored in the form of either chilled water or ice using storage tanks during off-peak periods. The stored chilled water or ice can then be used to meet cooling loads during on-peak periods when electricity cost is generally high. The optimal operation of cooling plants with TES systems requires anticipation of future cooling loads and effective control logic. Some of the control strategies of TES systems are discussed in Chapter 12.
- *Cogeneration systems:* These systems allow the simultaneous production of electricity and heat using engine generators. Combining absorption chillers with on-site engine generators can make cogeneration cost-effective. Sophisticated control strategies may be needed to operate engine generators to avoid excessive costs. Even small engine generators that are required for emergency power can be used—with proper controls—to reduce peak electrical loads and associated demand charges.

In the following sections, a more detailed description of other selected applications of EMCS is given, including duty cycling of motor loads, controls for outdoor air intake, optimum start of heating systems, and central heating and cooling plant operation.

10.4.1 Duty Cycling Controls

Frequent turning on and off of HVAC systems (and in particular fan motors) may actually be detrimental and may not be cost-effective over the life cycle of the equipment due to added maintenance and repair costs. However, on–off cycling of motor loads can be performed safely without long-term damage when minimum on and off times are respected. The National Electrical Manufacturers Association (NEMA) provides a set of recommendations for minimum on–off times of duty cycling of motor loads. Some of these recommendations are summarized in Table 10.1. It is highly recommended however, that the motor manufacturers be directly consulted to determine their suggested minimum off-time and allowable number of starts per hour.

To benefit fully from duty cycling energy cost savings (due mostly to reduction in demand charges), two methods can be used. In the first method known as parallel duty cycling, all the motors are cycled on and off at the same time. This method can provide energy cost savings when the duty period is less than the demand period (typically 15 minutes in most utility rates). However, when the duty period exceeds the demand period, there is no reduction in demand changes if all the motors are cycled on and off at the same time. In this case, it is recommended to use the second method of duty cycling called staggered duty cycling which alternates the on and off times of the motors.

TABLE 10.1 Allowable Number of Starts per Hour and Minimum Off-Time for Motor Loads

Motor Size HP (kW)	2-Pole Motors Max. Min. Starts/hr Off-Time (Seconds)	4-Pole Motors Max. Min. Starts/hr Off-Time (Seconds)	6-Pole Motors Max. Min. Starts/hr Off-Time (Seconds)
2.0 (1.5)	11.5 77	23.0 39	26.1 35
5.0 (3.75)	8.1 83	16.3 42	18.4 37
7.5	7.0 88	13.9 44	15.8 39
15.0	5.4 100	10.7 50	12.1 44
20.0 (15.0)	4.8 110	9.6 55	10.9 48
25.0 (18.75)	4.4 115	8.8 58	10.0 51
30.0 (22.5)	4.1 120	8.2 60	9.3 53
40.0 (30.0)	3.7 130	7.4 65	8.4 57
50.0	3.4 145	6.8 72	7.7 64

Source: NEMA, *Standard Publications MG-10*, National Electrical Manufacturers Association, Rosslyn, VA, 2003.

Most mechanical equipment manufacturers recommend extended duty cycling periods (higher than typical demand periods of 15 minutes). Therefore, a staggered duty cycling approach should be considered in most applications to ensure the safety of HVAC equipment while reducing operating costs. Example 10.2 illustrates the calculation procedure for energy cost savings incurred from a staggered duty cycling measure.

Example 10.2

Determine the reduction in the annual energy costs due to staggered duty cycling of three identical fan motors (each rated at 30 kW [40 hp]). The motor manufacturer specifies 20 minutes on and 10 minutes off as the minimum duty cycle. The utility monthly demand charge is $10/kW. First, determine the recommended NEMA duty period assuming 2-pole motors.

Solution

(a) Using Table 10.2, the allowable number of starts per hour for 40 hp, 2-pole motors is 3.7 starts/hr. Thus each start should last:

$$\text{Start period} = (60 \text{ min/hr})/(3.7 \text{ starts/hr}) = 14.4 \text{ min/start}$$

The duty period is thus about 15 minutes. Because the duty period is the sum of off-time and on-time, and because the minimum off-time is about 2 minutes (130 seconds based on Table 10.2), the maximum on-time allowable by the NEMA standard is 13 minutes. Thus, the manufacturer on-time is longer than that recommended by NEMA.

(b) The reduction in the electrical demand peak due to the staggered duty cycling approach is 1/3 of the total demand of all three motors (or 1/3 * 90 kW = 30 kW). Indeed, at any given time, only two out of the three motors are operating. Thus, the annual savings in electrical demands charges is given as follows:

$$\Delta kW = 12 * 30 kW = 360 kW/yr$$

Therefore, the staggered duty cycling control provides an annual energy cost savings of $3,600.

10.4.2 Outdoor Air Intake Controls

Due to the increase in the number of occupant complaints regarding poor indoor air quality (IAQ) and the increase in buildings diagnosed with sick building syndrome, the control and measurement of outside air intake rates has come to the forefront of the attention of many HVAC engineers and designers. The majority of HVAC system designers today rely on the ASHRAE "Ventilation Rate Procedure" described in ASHRAE Standard 62-2004 (ASHRAE, 2004), *Ventilation for Acceptable Indoor Air Quality*. ASHRAE Standard 62 specifies minimum ventilation rates as a function of building use and occupancy to provide adequate IAQ for conditioned spaces.

Unfortunately, the necessary monitoring equipment and control logic to maintain minimum outdoor intake rates are often nonexistent or are used improperly if they are installed. Consequently, several commercial buildings, and in particular those with variable-air-volume systems, have been found to have

TABLE 10.2 Summary of Comparative Results for the Control and Measurement Techniques Tested by Krarti et al. (2000).

					Outside Air Intake Rate Measurements					
					Averaging Pitot-Tube Array			Electronic Thermal Anemometry		
System Description	Measurement Control[1]	Case	Set-Point (cfm)	Validity	Mean (cfm)	stdev (cfm)	RMS (cfm)	Mean (cfm)	stdev (cfm)	RMS (cfm)
Fixed Damper Position	NA	1A	1,600	14%	656	658	1,150	682	564	1,048
Fixed Damper Position	NA	1B	2,400	23%	1,410	678	1,199	1,407	680	1,199
Fixed Damper Position	NA	1C	3,200	26%	2,178	819	1,309	2,124	870	1,513
Plenum Pressure Control	NA	2A	1,600	100%	1,630	69	75	1,544	70	89
Plenum Pressure Control	NA	2C	3,200	100%	3,288	94	129	3,279	88	118
Direct Control with Economizer Duct	P	3A	1,600	100%	1,635	38	52	1,546	46	71
Direct Control with Economizer Duct	P	3C	3,200	100%	3,192	50	51	3,225	47	53
Direct Control with Economizer Duct	E	4A	1,600	94%	1,695	55	110	1,637	55	67
Direct Control with Economizer Duct	E	4C	3,200	100%	3,228	49	57	3,263	50	80
Volume Tracking	E	5A	1,600	0%	2,427	439	936	2,436	458	953
Direct Control with Dedicated Duct	P	6A	1,600	100%	1,639	39	55	1,634	47	58
Direct Control with Dedicated Duct	P	6B	2,400	100%	2,430	41	51	2,457	51	77
Direct Control with Dedicated Duct	E	7A	1,600	100%	1,643	36	56	1,640	43	58
Direct Control with Dedicated Duct	E	7B	2,400	100%	2,404	39	40	2,428	42	50
Injection Fan	P	8A	1,600	100%	1,621	31	37	1,622	36	42
Injection Fan	P	8B	2,400	100%	2,429	28	40	2,440	36	54
Injection Fan	E	9A	1,600	100%	1,622	38	44	1,617	36	40
Injection Fan	E	9B	2,400	100%	2,418	33	37	2,427	32	42
Direct Control	C	NA[2]	1,600	75%	1,632	137	141	1,605[3]	119[3]	119[3]

[1] P = Averaging Pitot-Tube Array, E = Electronic Thermal Anemometer, C = CO_2 Concentration Balance.
[2] A different system setup was used for testing the concentration balance measurement technique.
[3] Value is for CO_2 concentration balance measurement technique, not electronic thermal anemometry.

inadequate ventilation (Sterling, Collet, and Turner, 1992). The use of appropriate airflow measurement or VAV control techniques is critical to maintain minimum outside air intake rates. In a recent work (Krarti, Brandemuehl, and Schroeder, 1999b), theoretical and experimental analyses have been performed to determine the accuracy of various techniques for outside airflow measurement and control applicable to VAV systems.

Descriptions of control strategies commonly used in the field are presented in the following sections. For a more complete discussion, refer to Krarti, Brandemuehl, and Schroeder (1999b). It is important to note that the following descriptions apply for VAV system operation during minimum outside air intake mode. Control at other times may differ, especially during economizer cycles.

10.4.2.1 VAV Control Techniques for Economizer Systems

In economizer systems, the size of the outside air duct must be large enough to safely provide 100 percent of the design flow. This large size, however, results in very low airflow velocities during minimum outside air intake rate mode which can make measurement difficult with pressure-based airflow measurement devices. Labeling of system diagrams presented here follows the approach adopted by Kettler (1998) for consistency.

10.4.2.1.1 Fixed Minimum Outdoor Air Damper Position

A fixed outside air damper position is a common method used to meet minimum outside airflow intake rates in VAV systems. Under design flow conditions, the outside air damper is positioned to meet the minimum outside air requirements. This predetermined damper position is then used when only minimum outside airflow is required, even as the supply fan speed is reduced.

In VAV systems, this control method does not deliver the minimum outside air intake due to variation in the static pressure of the mixing plenum (Drees, Wenger, and Janu, 1992; Mumma and Wong, 1990). Outside air intake rates are much closer to a constant percentage of supply air than a constant volume flow rate, a fundamental flaw of this control method (Janu, Wenger, and Nesler, 1995). Another problem with this method is stack and wind effects on the outside air intake rate (Solberg, Dougan, and Damiano, 1990). In addition to the limitations inherent to the technique of using a fixed minimum outside damper position, Ke and Mumma (1997) found through simulation that this method was the least effective control strategy to maintain a minimum outside airflow rate among those commonly used.

10.4.2.1.2 Volumetric Fan Tracking

Figure 10.8 shows a schematic of the volumetric fan tracking system. The flow measurement stations, AFS-1 and AFS-2, measure the supply and return airflow rates, respectively. The return fan speed is controlled (AFC-1) to maintain a fixed differential in the return airflow rate compared to the supply airflow rate. The preset fixed differential in the return and supply airflow rates must then be made up by outside air. Damper positions for the return, exhaust, and outside air are generally set to fixed positions during

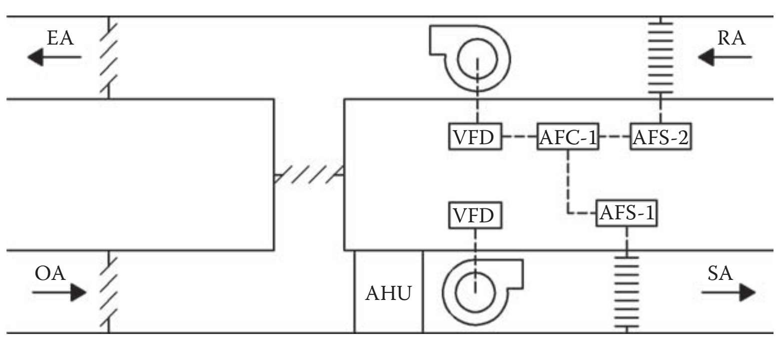


FIGURE 10.8 Outside airflow rate control schematic for system using volumetric fan tracking control strategy.

minimum outside air intake mode. The fixed flow differential is typically based upon the initial system air balancing. The outside air provided to the space maintains a slight positive static pressure within the building to reduce unwanted infiltration.

Volumetric tracking is one of the more common control methods used in VAV systems today (Kettler, 1995; Avery, 1992). The benefit of this method is that airflow rates in the supply and return ducts are generally large enough that standard flow measuring techniques can be sufficiently accurate. However, several authors have alluded to weaknesses in this control method. Elovitz (1995) states that even small measurement errors in large flow rates can translate to large errors in the calculated outside air intake rates and that a fixed differential flow is not versatile enough to account for exhaust and leakage flow rate changes.

Using a fixed position for the outside air damper also limits the flow rates of outside air for space pressurization. If the damper is not sufficiently open, it is possible that there will not be enough outside air available (Janu, Wenger, and Nesler, 1995). Janu, Wenger, and Nesler (1995) also recommend that online measurement of outside air intake rates be provided and that the differential flow vary to compensate for operation of variable exhaust flows and the opening and closing of windows and doors. Finally, Kettler (1995) makes the argument that when typical measurement errors are accounted for, the outside air intake rate can vary by as much as 35 percent.

10.4.2.1.3 Measurement and Control of Outside Airflow Rate with Economizer

A typical arrangement for this type of system is shown in Figure 10.9. The outside air duct is sized to allow for economizer control of the system. During minimum outside airflow intake mode, a flow measurement station (AFS-1) records the flow of outside air and controls the return and outside dampers (M-1 and M-2, respectively) to maintain the required minimum outside airflow intake rate.

Due to the relatively large size of the outside air duct in this system, the measurement of the outside air intake rate at the flow measurement station (AFS-1) can be difficult with pressure-based airflow measurement devices. The accuracy of this control technique depends directly upon the accuracy with which the outside air intake rate can be measured.

10.4.2.1.4 Plenum-Pressure Control

This method relies upon additional instrumentation such as a manometer or differential pressure transmitter to measure the pressure drop across a fixed orifice. By maintaining a constant pressure drop, the minimum outside airflow requirements can be met (Janu, Wenger, and Nesler, 1995; Haines, 1994; Elovitz, 1995). It can be implemented either in a dedicated ductwork or in an existing economizer duct. The fixed orifice in this case is the combination of the outside air louver (L-1) and the damper installed in the outside air duct as suggested by Mumma and Wong (1990). This system is shown schematically in Figure 10.10. The pressure drop must be large enough so it can be accurately measured but not so large as to create an excessive energy penalty (Ower and Pankhurst, 1977; Kettler, 1998). The differential pressure transmitter (DP-1) measures the pressure drop and the return air damper

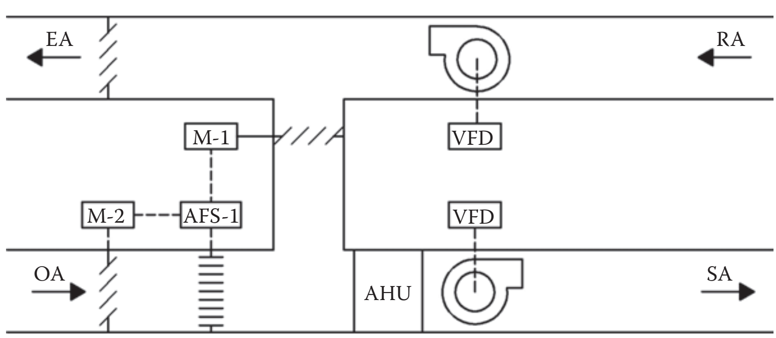


FIGURE 10.9 Outside airflow rate control schematic for system with economizer damper.

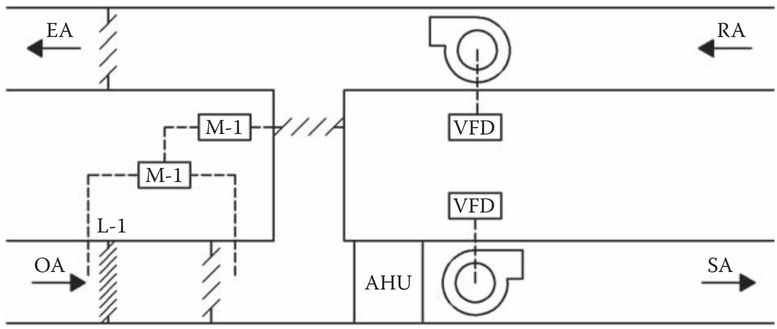


FIGURE 10.10 Plenum-pressure control schematic.

is controlled to maintain a constant value. Obviously, if an actuator is not located on the return air damper, one must be added.

For a fixed damper position, the value of the loss coefficient C for the damper is constant. The outside airflow intake rate is related to the pressure drop across the damper by Eq. (10.6):

$$V = D\sqrt{\frac{\Delta p_j}{\rho \cdot C}} \tag{10.6}$$

where

V = Velocity [fpm] (m/s).
D = Constant [1096.7] (1.4123).
Δp_j = Total pressure loss [inW.G.] (Pa).
ρ = Density [lb_m/ft^3] (kg/m^3).
C = Local loss coefficient [–].

10.4.2.2 VAV Control Techniques for Systems with a Dedicated Outside Air Duct

The next two control strategies attempt to remedy the main disadvantage of an HVAC system equipped with only one outside air duct. By adding another duct through which only the minimum outside air must flow, the size can be made much smaller, thereby increasing the airflow velocities and thus making them easier to measure. Typically, the larger duct is used only during economizer control mode and is closed when minimum outside air intake rates are required.

10.4.2.2.1 Measurement and Control of Dedicated Minimum Outside Duct Airflow Rate

This system is shown schematically in Figure 10.11. In economizer mode, the damper on the larger outside air duct is controlled to regulate the outside air intake rate. During minimum outside air intake mode, the dedicated outside airflow intake duct is opened and the damper in the larger outside air duct is closed. A flow measurement station (AFS-1) records the outside airflow rate and controls the return (M-1) and the dedicated outside air dampers (M-2) to maintain the minimum outside airflow intake rate. The exhaust air damper can be left in a fixed position during minimum outside airflow intake mode, or alternatively, can also be controlled from the flow measurement station (exhaust damper control not shown in Figure 10.11).

10.4.2.2.2 Outside Air Injection Fan

In this control technique, a dedicated minimum outside airflow intake duct contains a fan used to control outside airflow during minimum intake rate mode. This system is the same as that illustrated

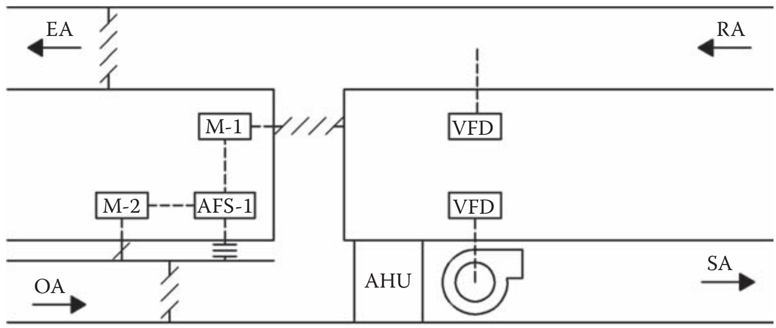


FIGURE 10.11 Outside airflow rate control schematic for system with dedicated minimum outside airflow duct.

in Figure 10.11 except a fan is installed in the dedicated minimum outside air duct. This injection fan is controlled by a supply fan: when the supply fan is on, the injection fan motor is turned on. This control of the fan motor can be modified to allow the injection fan to remain off during unoccupied periods, times of building warmup, or when the system is running in economizer mode. The fan is chosen such that it has a very flat fan curve and operates almost as a constant volume fan over the expected range of pressures (Elovitz, 1995; Avery, 1989). Although this method can be adequate to provide the minimum required outside air intake, it is usually expensive and difficult to implement in existing buildings.

10.4.2.3 Other VAV Control Techniques

In addition to the techniques described above, other control strategies are used to control outdoor air intake. Some of these control strategies are briefly described in this section. For more information on these techniques, the reader is directed to the cited references.

- *Minimum outside air damper position reset:* This control strategy attempts to compensate for the main limitation of the fixed outside air damper control strategy by allowing the damper position to be reset based upon the supply fan speed. The position of the outside air damper can be found from either a linear relationship with the supply fan speed, or a higher-order polynomial equation. As with the fixed minimum outside damper position control method, online measurements of the outside air intake rates are not required. However, the minimum ventilation rate may not be met if the supply airflow rate falls too low (Ke and Mumma, 1997). In addition, because the damper and duct are often the same size, small changes in damper position translate to large changes in flow rates (a highly nonlinear relationship) and normal hysteresis can significantly affect the outside air intake rates (Drees, Wenger, and Janu, 1992). Finally, this control strategy cannot account for wind and stack effects on the system. See Solberg, Dougan, and Damiano (1990) for additional details regarding these errors.
- *Supply/return fan speed or vane position matching:* The supply and return fan speeds are controlled, often from the same control signal, to match each other with a fixed differential to maintain a slight positive pressurization. The outside airflow rate is then equal to the difference between the supply and return airflow rates. However, similar to the volumetric tracking control strategy, this is only true when there is no exhaust airflow. Whenever the exhaust flow is greater than zero, the outside air intake rate will be increased and an energy penalty may result. Although this control method is inexpensive and easy to implement on existing systems, it has generally been unacceptable due to mismatched fan flow characteristics over the typical range of operation (Janu, Wenger, and Nesler, 1995). Elovitz (1995) has also stated that this method is not versatile enough to account for all possible circumstances encountered in building operation, such as fume hoods and the opening and closing of windows and doors.

- *Direct building static pressure control:* By measuring the building pressure relative to the outdoors, a closed-loop control method can be implemented to vary outside air intake and relief airflow rates. This is achieved by varying the return airflow rate or the positions of the return and relief dampers (Janu, Wenger, and Nesler, 1995; Kettler, 1988). The outside air intake rate is controlled by one of two methods based upon the differential pressure between the building space and the outside static pressure sensor. The first method varies the return fan speed to maintain positive space pressurization. The second method controls the return and exhaust dampers to maintain positive space pressurization. This control method is easy to implement in existing systems and has the advantage that the outside air intake rate is not a function of the supply airflow rate (Janu, Wenger, and Nesler, 1995). However, Elovitz (1995) points out several drawbacks to this method. First, the differential pressure between the space and outdoors is very difficult to measure accurately due in part to wind loads and intermittent pressure changes due to the opening of windows and doors (Levenhagen, 1992; Avery, 1992). Secondly, the normal range of pressure differences throughout a large building can be larger than the system is trying to control. Finally, Elovitz (1995) states that the outside air intake rates are a function of the pressurization and leakage area of a building and may not remain constant over time.
- *Fan capacity matching through balancing:* In this method, return fans are controlled by static pressure in the ductwork rather than by tracking the supply fan. The supply and return fans are adjusted during building commissioning so they always lead/lag each other to maintain a difference in airflow. Outside air is the difference in the supply and return airflow rates. A positive building static pressure is usually maintained. Again, when there is an exhaust airflow an energy penalty may be incurred by bringing in too much outside air. Unlike volumetric tracking, there are no flow measuring stations for fan matching through balancing. This method is cheaper and easier to implement than volumetric tracking and is more accurate than direct building pressure control. However, Levenhagen (1992) points out that the balancing contractor must ensure that the two fans are properly matched which is very difficult to achieve. Janu, Wenger, and Nesler (1995) have concluded that generally this is not possible.
- *Characterization of flow through a modulated outside air damper:* By characterizing the outside air intake rate as a function of both the position and pressure drop across the damper, accurate control of ventilation air can be obtained over a wide range of operating conditions. However, this process requires significant amounts of time to properly characterize the airflow rates. In addition, this method is subject to calibration drifts in transmitters, as well as looseness and hysteresis, any of which can cause substantial errors (Janu, Wenger, and Nesler, 1995).

10.4.2.4 Comparative Analysis

Table 10.2 summarizes the results of an experimental comparative analysis performed by Krarti et al. (2000) to evaluate some of the control techniques for outdoor air intake under repeatable laboratory conditions. Specifically, Table 10.2 provides the average value, the standard deviation, the root mean square of the outdoor air intake flow rate, and the validity of each measurement and control method tested in a laboratory setup. In particular, three measurement techniques are used to determine the airflow rates: averaging Pitot-tube array station (P), electronic thermal anemometer (E), and CO_2 concentration balance technique (C). For more details on these measurement techniques, the testing setup, and the experimental results, the reader is referred to Krarti et al. (2000).

The percentages listed in Table 10.2 in the column labeled "validity" were calculated from Eq. (10.7):

$$validity = \frac{n_v}{n} \tag{10.7}$$

where n_v = the number of valid data points.

Each test presented in Table 10.2 is subject to errors from the airflow measurement and the control technique used. Each 10-second data point x_i recorded during testing was considered valid if it met the following two conditions:

$$|x_i - \text{set point}| \leq (\text{set point} \cdot 10\%)$$

and

$$\frac{e_i}{x_i} < 15\%$$

where e_i = the predicted error for the airflow measurement in the laboratory.

The first condition attempts to account for the accuracy of the control technique by requiring the data point to be within 10 percent of the set-point. The second condition attempts to account for the accuracy of the airflow measurement technique by requiring the predicted error of the data point to be less than 15 percent.

In summary, accurate measurement and control of outside air intake rates in VAV systems is possible when careful attention is paid to proper installation and operation of system equipment. In systems where uniform airflow profiles exist, the use of an averaging Pitot-tube array or an electronic thermal anemometry, depending upon the expected velocities, for the direct measurement of outside airflow rates allows for direct control of minimum outside air intake rates. When these conditions are not met, the installation of a separate, dedicated minimum outside air duct, or the use of the concentration balance airflow measurement technique provide adequate alternatives. However, calculating the outside airflow rate using a temperature balance will not provide accurate results for all building operating conditions. Plenum pressure control in systems where measurement of the outside airflow rate is not possible should provide adequate control of minimum outside air intake rates. The traditional CAV control strategy of a fixed minimum outside air damper position, and the more robust volumetric fan tracking technique are not capable of accurately controlling outside airflow rates in VAV systems.

10.4.3 Optimum Start Controls

After the energy crisis of the 1970s, engineers found that building utility bills can be reduced by 12 to 34 percent merely by implementing an occupied thermostat setback (Bloomfield and Fisk, 1977). During the cooling season, the unoccupied zone temperature set-point is raised whereas during the heating season, the set-point is lowered. In some mild climates, the indoor temperatures are allowed to float during night periods rather than be defined by a night set-point. However, the winter night setback is typically set between 13°C (55°F) and 15.5°C (60°F). Thus, building indoor temperature is colder than its occupied set-point in the early mornings. Due to the building thermal mass, the heating system has to be turned on earlier than the scheduled occupied time to achieve thermal comfort when people first enter the building.

The amount of time a building takes to recover from its night setback to its occupied set-point is usually referred to as the building recovery time. The length of the recovery period depends on several factors including outdoor ambient temperatures, indoor temperatures, and building thermal characteristics. Therefore, the recovery period can vary daily throughout the heating season especially in climates with sudden changes in the outdoor temperatures. However, the recovery time is typically set to be the same throughout the entire heating season by building operators to simplify the start controls of the heating system. This recovery time is defined as the earliest time the heating system needs to be started for the coldest day of the year. Although this approach may achieve thermal comfort at the start of the occupancy periods throughout the heating season, it does not ensure optimal start times for the heating system especially during mild winter mornings.

With an energy management and control systems, algorithms can be developed to determine the optimum start times and thus the best recovery periods. Several algorithms have been suggested in the literature. Typically, the recover y times are adjusted daily based on outdoor ambient temperatures and initial building zone temperatures (which may not necessarily be close to the set-point temperatures during unoccupied periods). In the following sections, some of the simplified algorithms for estimating building recovery times are presented.

Method 1: A linear relationship between recovery times τ and outdoor ambient temperatures T_{amb}:

$$\tau = a_0 + a_1 T_{amb} \tag{10.8}$$

where

$$a_0 = \tau_{\max} + \frac{\tau_{\max} T_{amb,\max}}{T_{amb,zero} - T_{amb,\max}}$$

and

$$a_1 = -\frac{\tau_{\max}}{T_{amb,zero} - T_{amb,\max}}$$

where

τ_{max} is the maximum recovery period.

$T_{\mathrm{amb,max}}$ is the outdoor ambient temperature during the time when the maximum recovery period is obtained.

$T_{\mathrm{amb,zero}}$ is the outdoor ambient temperature during the time when the recovery period is zero.

It should be noted that the approach presented by Eq. (10.8) is relatively easy to implement inasmuch as it does not require any regression analysis to determine the relation coefficients a_0 and a_1.

Method 2: A linear relationship between the recovery time τ and both outdoor ambient temperature T_{amb}, and initial zone temperature $T_{\mathrm{zone,initial}}$.

$$\tau = a_0 + a_1 T_{amb} + a_2 T_{zone,initial} \tag{10.9}$$

where the coefficients $a0$, a_1, and a_2 are determined based on a regression analysis. This approach was first developed and implemented by Jobe and Krarti (1997). To determine the regression coefficients, it is recommended that data for at least five days be used.

Method 3: A quadratic relationship among the recovery tim τ and both outdoor ambient temperature T_{amb} and initial zone temperature $T_{\mathrm{zone,initial}}$, using a weighting function in the form of

$$\tau = a_0 + w a_1 T_{amb} + (1-w).[a_2 T_{zone,initial} + a_3 T_{zone,initial}^2] \tag{10.10}$$

TABLE 10.3 Daily Average Reduction in Recovery Periods for Two Buildings Located in Colorado

Building/ Method No.	Maximum Recovery Period (min)	Predicted Recovery Period (min)	Reduced Startup Time (min)	Reduction from Maximum Recovery Period (%)
Building 1				
Method 1	90	64	26	29
Method 2	90	53	37	41
Method 3	90	77	13	14
Building 2				
Method 1	75	19	56	74
Method 2	75	19	56	74
Method 3	75	28	47	62

Source: Jobe and Krarti, Field implementation of optimum start heating controls, in *Proceedings for ASME Solar Engineering*, 305, 1997.

where the weighting parameter w is defined as follows:

$$w = 1000^{-(T_{zone,initial} - T_{setnight}) / (T_{zone,final} - T_{setnight})}$$

with

$T_{setnight}$ is the night (or unoccupied) setback temperature
$T_{zone,final}$ is the occupied set-point temperature

The approach presented by Eq. (10.10) was proposed by Seem, Armstrong, and Hancock (1989) based on results from computer simulations.

A comparative analysis performed by Jobe and Krarti (1997) among the three approaches indicated that all three methods can reduce the startup time for the heating system and thus save energy compared to the common approach that relies on setting the maximum recovery time throughout the entire heating season. Table 10.3 summarizes the daily average reduction time in recovery period for two educational buildings located in Colorado using the three approaches discussed above.

It clear from the results presented in Table 10.3 that the approach described as Method 2 provides the highest recovery time reduction while providing adequate thermal comfort within the two buildings. For the two buildings used in the comparative analysis, the recovery time was found to vary rather linearly with outdoor ambient temperatures. Therefore, because Method 3 places more emphasis on indoor air temperatures [refer to Eq. (10.10)], it may not be adequate for the considered buildings.

10.4.4 Cooling/Heating Central Plant Optimization

Cooling and heating central plants offer several opportunities to reduce energy operating costs through optimal or near-optimal controls for individual equipment (local optimization) and for the entire HVAC system (global optimization). Although optimal controls have been developed and implemented for various components of cooling and heating central plants, global optimization remains a considerably complex endeavor and only a few strategies have been suggested and tested.

In this section, some of the local optimal control strategies are discussed. Moreover, operating strategies for entire cooling/heating plants are briefly discussed.

10.4.4.1 Single Chiller Control Improvement

Before replacing an existing chiller, it may be more cost-effective to consider other cooling alternatives or simple operating strategies to improve cooling plant energy performance. In particular, a significant improvement in the overall efficiency of a chiller can be obtained through the use of automatic controls to:

- Supply chilled water at the highest temperature that meets the cooling load.
- Decrease the condenser water supply temperature (for water-cooled condensers) when the outside air wet-bulb temperature is reduced.

Example 10.3 illustrates typical energy cost savings due to improved controls for a single chiller cooling plant.

Example 10.3

A centrifugal chiller (having a capacity of 500 kW and an average seasonal COP of 4.0) operates with a leaving chilled water temperature of 4.5°C. Determine the cost savings incurred by installing an automatic controller that allows the leaving chilled water temperature to be set 2.5°C higher on average. Assume that the number of equivalent full-load hours for the chiller is 1,500 per year and that the electricity cost is \$0.07/kWh.

Solution

Using Figure 9.13 (refer to Chapter 9), the increase in the COP for a centrifugal chiller due to increasing the leaving chilled water temperature from 4.5°C to 7.0°C is about 8 percent. The energy use savings can be calculated using Eq. (9.12) with $SEER_e = 4.0$; $SEER_r = 4.0 * 1.08 = 4.32$; $N_{h,C} = 1{,}500$; and $Q_C = 500$ kW; $LF_C = 1.0$ (assume that the chiller is sized correctly):

$$\Delta E_C = 500 kW * 1500 hrs/yr * 1.0 * \left(\frac{1}{4.0} - \frac{1}{4.32} \right) = 13{,}890 kWh/yr$$

Therefore, the annual energy cost savings are \$970/yr.

10.4.4.2 Controls for Multiple Chillers

When a central cooling plant consists of several chillers, a number of control alternatives exist to meet a building cooling load. Effective controls would select the best alternative for operating and sequencing the chillers to minimize the cooling plant operating costs.

Simple guidelines can be followed to operate multiple chillers at near-optimal performance. Typically, chiller operating variables such as chilled water temperature and condenser waterflow rate are adjusted to ensure optimal controls. Some of the near-optimal control guidelines to operate electrically driven central chilled water systems are summarized below (ASHRAE, 2007):

- Multiple chillers should be controlled to supply identical chilled water temperatures.
- For identical chillers, the condenser waterflow rates should be controlled to provide identical leaving condenser water temperatures.
- For chillers with different capacities but similar part-load performance, each chiller should be loaded at the same load fraction. The load fraction for a given chiller can be set as the ratio of its capacity to the sum total capacity of all operating chillers.

To determine the optimal chiller sequencing, a detailed analysis is generally needed to account for several factors including the capacity and the part-load performance of each chiller and the energy use associated with all power-consuming devices such as distribution pumps. Chapter 9 provides some guidelines on the potential energy savings associated with multichiller cooling plants.

10.4.4.3 Controls for Multiple Boilers

As discussed in Chapter 8, the use of an array of small modular boilers provides a more energy-efficient heating system than a single large boiler especially under part-load operation conditions. Indeed, each of the modular boilers can be operated close to its peak capacity and thus its highest energy efficiency. To optimally operate multiple boilers, it is important to know when to change the number of boilers online or offline. The mere addition of a second boiler online when one boiler cannot handle the load may not provide the minimum operating cost. Indeed, the increase of firing rate (due to additional heating load) on any given boiler can cause a decrease in thermal efficiency due to higher flue-gas temperatures and thus higher thermal losses. However, the addition of a second boiler online increases the standing losses due to auxiliaries and the thermal losses through the added casing and piping of the second boiler. Therefore, a detailed analysis is needed to determine the changeover points for the multiple boilers. These changeover points depend on the characteristics of each boiler (ASHRAE, 2007).

10.5 Summary

In this chapter, an overview of basic components and applications of HVAC control systems has been presented. In particular, the energy cost savings incurred by various functions of energy management and control systems have been illustrated through selected examples and applications. In addition to being knowledgeable regarding the currently available control systems and applications, the energy auditor should be aware of the development of intelligent control systems, especially those applicable to HVAC systems.

11

Compressed Air Systems

11.1 Introduction

Compressed air is a commonly used utility in industrial processes and represents an important fraction of the operating cost of manufacturing facilities. It is estimated that the energy used by compressed air systems represents about 30 percent of the total energy consumed by electrical motors in France. Typical compressors use electricity to produce compressed air that may be needed for various industrial applications. Unfortunately, most existing compressed air systems have low efficiencies due to several factors including air leaks, inadequate selection of compressors, inappropriate uses of compressed air, and poor controls.

In this chapter, cost-effective energy conservation measures are described to reduce the operating costs of compressed air systems. First, a review of the basic principles of gas compression is provided. Then, the basic components required for the production, distribution, and utilization of compressed air are discussed. Finally, the calculation procedures for the energy savings of selected energy conservation measures are presented with illustrative examples.

11.2 Review of Basic Concepts

Figure 11.1 illustrates a simplified compressed air system operated by an electric motor. The compressed air is generally produced in a centralized location and then distributed to various locations within the facility to be used by equipment involved in either the production process or in pneumatic control.

Generally, a compressed air system consists of several components including:

- One or several compressor(s) connected to a driver. The driver is typically an electric motor.
- A distribution system with piping, valves, fittings, and controls. The distribution system feeds the compressed air to operate several pieces of equipment dispersed throughout the facility.
- Other equipment such as receivers, dryers, and filters.

The overall efficiency of a compressed air system depends on three stages: production, distribution, and utilization. During an audit, it is important to evaluate each of these stages in order to assess the performance and thus the potential for improving the energy efficiency of an air compressed system. A brief review of the basic principles and factors that affect energy use for each stage for the production–distribution–utilization chain of compressed air is given below.

11.2.1 Production of Compressed Air

The basic concept of producing compressed air is relatively simple. Generally, mechanical power is provided to a compressor that increases the pressure of intake air. This intake air is typically drawn at

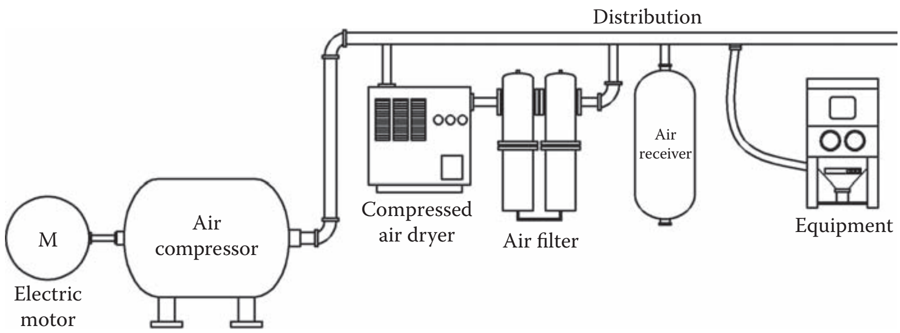


FIGURE 11.1 A schematic diagram for a compressed air system driven by an electric motor.

ambient atmospheric conditions (i.e., pressure of 100 kPa or 1 bar). The compressor can be selected from several types such as centrifugal, reciprocating, or rotary screw with one or multiple stages. For small and medium-sized units, screw compressors are currently the most commonly used in industrial applications. Table 11.1 provides typical pressure, airflow rate, and mechanical power requirement ranges for different types of compressors (Herron, 1999).

A simplified energy analysis of compressed air systems can be carried out using the first law of thermodynamics applied to ideal gases. A basic review of energy analysis applied to the compression of an ideal gas is provided to help identify the important parameters that need to be modified to reduce the energy used for the production of compressed air.

Figure 11.2 represents a piston cylinder during a compression of an ideal gas. In addition, the figure shows the variation of the pressure as a function of the volume occupied by the gas, shown in a P–V diagram, for isothermal, adiabatic, and polytropic compressions. The relationship between temperatures and pressures at the inlet (or suction) and outlet (or discharge) of a compressor following an isothermal, an adiabatic, or a polytropic compression process can be summarized by Eq. (11.1):

$$\frac{T_o}{T_i} = \left(\frac{P_o}{P_i}\right)^{\frac{\gamma-1}{\gamma}} \tag{11.1}$$

where

$\gamma = 1$ when the compression is isothermal.

$\gamma = k$ when the compression is adiabatic (k = 1.4 for dry air).

$\gamma < k$ when the compression is polytropic (typically k.=.1.3 for dry air).

It should be recalled that for dry air, a simple relationship exists among the pressure, temperature, and density. This relationship is typically referred to as the equation of state and is provided by Eq. (11.2) for the compressor inlet air:

$$P_i = \rho_i . Z_{a,i} . R_a . T_i \tag{11.2}$$

For dry air, the constant R_a can be calculated by dividing the ideal gas constant R [R = 8314.4 J(kg. mole.K)] by the molar mass of air M_a (M_a = 28.9). Thus, the value for the constant R_a is 287 J/(kg.K). Moreover, the compressibility factor $Z_{a,i}$ provides an indication of the difference in the behavior of air relative to that of an ideal gas. The value of $Z_{a,i}$ ranges from zero and one and depends on the pressure and temperature of the air. For pressures above 20,000 kPa (200 atm), it can be assumed that dry air behaves as an ideal gas (i.e., $Z_{a,i} = 1$).

It should be noted that an expression similar to Eq. (11.2) could be established for the compressor outlet air.

TABLE 11.1 Typical Ranges of Application for Various Types of Air Compressors

Compressor Type	Airflow Rate (m^3/s)	Absolute Pressure (MPa)	Mechanical Power Requirement (kW/L/s)
Reciprocating	0.0–5.0	0.340–275.9	0.35–0.39
Centrifugal	0.5–70.5	3.5–1034.3	0.46
Rotary screw	0.5–16.5	0.1–1.8	0.33–0.41

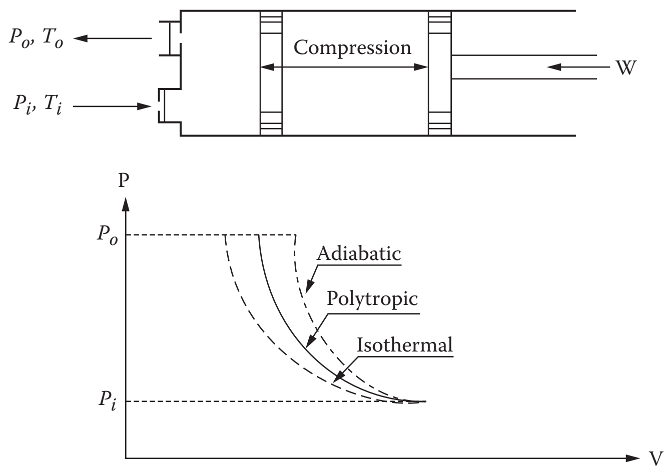


FIGURE 11.2 Ideal compression using isothermal, adiabatic, or polytropic processes.

The work needed to compress a mass flow rate $\dot{m}_a$ of dry air can be estimated by applying the first law of thermodynamics to a compression process of an ideal gas. In particular, the mechanical power required for an isothermal compression can be calculated by Eq. (11.3):

$$\dot{W}_m = \dot{m}_a.R_a.T_i.Ln\left(\frac{P_o}{P_i}\right) \tag{11.3}$$

The mechanical power required for an adiabatic or a polytropic compression cycle can be estimated from Eq. (11.4):

$$\dot{W}_m = \frac{\dot{m}_a.R_a.T_i.\gamma}{\gamma-1}.\left[\left(\frac{P_o}{P_i}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \tag{11.4}$$

Two important observations can be made from both Eqs. (11.3) and (11.4) related to the energy efficiency of compressors:

1. The mechanical power for a compressor increases linearly with inlet air temperature. To increase the energy efficiency of a compressed system, inlet air should therefore be as cool as possible.
2. The mechanical power for a compressor increases with the pressure ratio. Therefore, it is important to produce compressed air with a discharge pressure limited to the maximum pressure needed by the facility. In other terms, overpressurization should be avoided as much as possible.

Moreover, the variation of the inlet air pressure can have a significant impact on the mechanical power requirement by a compressor. The influence of the altitude on the ambient pressure should be accounted for, in particular.

Example 11.1 provides a comparative analysis of the mechanical power requirements for three compression types (i.e., isothermal, adiabatic, and polytropic). It is clear from the results of Example 11.1 that adiabatic compression requires more energy input than all the other compression types. The case of isothermal compression provides the optimal operating conditions of the compressor. In particular, the air temperature should be maintained constant throughout the compression. Therefore, to reduce the compressor mechanical power requirements, cooling of air should be performed throughout the compression process. Unfortunately, ideal conditions of constant air temperature cannot be achieved in real compressions. However, if adequate cooling of the air is provided, a real compression can be represented by a polytropic process. As discussed later, intercoolers are used to dissipate heat during compression.

Example 11.1

Compare the mechanical power requirement to compress 1 kg/s of dry air from 100 kPa and 20°C (ambient conditions) to 800 kPa (absolute pressure) using isothermal, adiabatic, or polytropic ($\gamma = 1.3$) compression .

Solution

Using Eq. (11.3) with P_i = 100 kPa, P_o = 800 kPa, T_i = 293 K, R_a = 287 J/(kg.K); the mechanical power requirement for the isothermal compression of 1.0 kg/s can be estimated:

$$\dot{W}_m = (1.0kg/s).(287J/kg.K).(293K).Ln\left(\frac{800kPa}{100kPa}\right) = 174.86kW$$

The mechanical power for the adiabatic compression is calculated using Eq. (11.4) with $\gamma = k = 1.4$:

$$\dot{W}_m = \frac{(1.0kg/s).(287J/kg.K).(293K).(1.4)}{1.4-1}.\left[\left(\frac{800kPa}{100kPa}\right)^{1.4-1/1.4} - 1\right] = 238.82kW$$

For polytropic compression, the mechanical power requirement can be determined using Eq. (11.4) again but with $\gamma = k = 1.3$:

$$\dot{W}_m = \frac{(1.0kg/s).(287J/kg.K).(293K).(1.3)}{1.3-1}.\left[\left(\frac{800kPa}{100kPa}\right)^{1.3-1/1.3} - 1\right] = 224.42kW$$

As expected, the isothermal compression requires less mechanical power and the adiabatic compression requires more power input than any of the three compression types. Note that the polytropic compression which is more representative of an actual compression process has mechanical power requirements between those of the isothermal and adiabatic compressions.

It is interesting to note that the mechanical power input per unit of airflow rate (i.e., kW/L/s) can be calculated for the three types of compression by estimating the density of air at the inlet conditions. The ideal gas equation Eq. (11.2) can be used to determine the density of the air (with $Z_{a,I} = 1$). For $T_i = 293$ K and $P_i = 100$ kPa, the density of air 1.189 kg/m3. Thus, the mechanical input for the three compression types is 0.21 kW/L/s for an isothermal compression, 0.27 kW/L/s for a polytropic compression, and 0.29 kW/L/s for an adiabatic compression.

For real air compressors, it is recommended that the mechanical power or energy requirements be estimated using the assumption of adiabatic compression for several reasons including:

1. Even though polytropic compression is a better representation of actual compression processes, it is difficult to obtain the value of the exponent γ readily without expensive measurements. In some applications such as turbocompressors (which increase the pressure of the air by increasing its velocity), the exponent γ can in fact be greater than $k = 1.4$.
2. For typical polytropic compressions, the power requirements are close to those estimated from adiabatic compression (see the results of Example 11.1).
3. The power requirement for an adiabatic compression represents an upper limit of the power input used by real compressions and thus provides the worst-case scenario.

When compressors are driven by electric motors, the electrical energy kWh_{comp} used by the compressed air systems can be determined from the following expression:

$$kWh_{comp} = \frac{\dot{W}_m.LF_{comp}.N_{h,comp}}{\eta_M} \tag{11.5}$$

where

$\dot{W}_m$ = the mechanical power required by the compressor estimated using Eq. (11.3) or Eq. (11.4).

$N_{h,comp}$ = the number of hours per year when the compressor is operated.

LF_{comp} = the average load factor of operating the compressor.

η_M = the efficiency of the motor that drives the compressor.

In addition to the compressor, other accessory units are needed for the production of compressed air. These units may increase the mechanical and thus the electrical power requirement for operating compressed air systems. Some of the accessory equipment and their roles are briefly discussed below.

11.2.1.1 Filters

The air at the inlet of the compressor should be as clean as possible to protect the compressor and its associated equipment from any damages that suspended particles in an unfiltered air can cause. Indeed, suspended particles can penetrate various parts of the air compressor system where they can either (i) obstruct small orifices (within the compressors, distribution lines, or machines using compressed air), or (ii) wear by friction some interior surfaces of the compressor (including cylinders, pistons, or rotors). The wear is even more damaging when the particles are mixed with lubricating oil.

Various air filters and air-cleaning techniques can be used at the inlet of compressors depending on the size of the airborne particles. The *ASHRAE Handbook* (2008) provides a description of a number of air-cleaning techniques and their applications. A selection of an air filter for a compressor depends typically on two factors: the efficiency of the filter and the pressure drop caused by the filter. The air filter efficiency is typically defined as the percentage of retained particles. The pressure drop through an air filter—due to either a fine screen mesh or an accumulation of dirt—decreases the capacity of the compressor. As a rule of thumb, it is estimated that for every 100 mm (or 4 in.) of water pressure drop (i.e., 1 kPa), the mass flow rate of compressed air is reduced by approximately 1 percent (assuming the compressor uses the same power).

It is therefore important to select proper air filters and to clean or replace these filters periodically in order to reduce the energy used by compressed air systems. Example 11.2 illustrates the increase in the energy cost for operating an air compressed system with a contaminated (i.e., dirty) air filter.

Example 11.2

Determine the increase in energy cost of a 100-kW air compressor operating 8,000 hours per year due to a contaminated air filter. Assume that:

- The electrical motor efficiency is 90 percent.
- The pressure drop is 1.0 kPa for a clean air filter but becomes 3 kPa for a contaminated filter.
- The cost of electricity is $0.05/kWh.

Solution

Using a rule of thumb, it can be stated that for every 1 kPa increase in the pressure drop across the air filter, the mechanical power is increased by 1 percent (because the mechanical power is proportional to the mass flow rate). Thus, the energy cost penalty EC_{filter}, of using a contaminated air filter, can be estimated as follows:

$$EC_{filter} = \frac{100kW * 8000hrs/yr * \$0.05/kWh}{0.90} * \left(\frac{3kPa}{1kPa}\right) * 0.01 = \$1{,}330/yr$$

11.2.1.2 Receiving Tanks

In some compressed air installations, receiving tanks are used and are placed either close to the compressors or near the end-use machines that have high and variable demands. Three reasons can justify the utilization of central or local air receiving tanks:

1. Storage of energy in the form of compressed air to reduce unwanted cycling of the compressors
2. Regulation of the flow rate of the compressed air based on the load requirements of the end users
3. Control of the discharge pressure for the compressed air especially when piston compressors are used

The design specifications of the receiving tanks should follow the requirements of the local authorities for pressure vessel regulations.

11.2.1.3 Dryers

Ambient air can be moist with high water vapor content. This water vapor can condense either in the compressor, the distribution lines, or the equipment using the compressed air. To reduce or eliminate this water condensation, the compressed air should be dried using dryers. The selection of the dryer type depends on two important factors:

1. The quality of dry air required expressed in terms of the dew point pressure
2. The total cost of the dryer including initial costs and operating and maintenance costs

There are a number of drying methods used with compressed air systems. In most cases, refrigerated dryers are used to bring the pressure dewpoint of the air to levels below 5°C. In other applications such as food processes, desiccant dryers are used to achieve very dry air with dewpoint pressures below 0°C.

In general, heat exchangers commonly known as after-coolers are used to reduce the water content and possibly oil vapor in the compressed air before it gets into the distribution lines or the receiving tanks. In particular, the compressed air is cooled using ambient air or water so condensation occurs in a separator located just after the compressor. As a rule of thumb, the discharge temperature of the compressed air is typically 10°C above the water temperature (for after-coolers using water) and 15°C above the air temperature (for after-coolers using air).

11.2.1.4 Intercoolers

To dissipate the compression heat, it is customary to cool the compressor cylinders and cylinder covers using heat exchange devices commonly referred to as intercoolers. These intercoolers can use air or water to control the temperature of the compressor. Typically, cooling of the compressed air is preferred because of convenience inasmuch as it eliminates the need for a water supply and thus avoids the problems associated with water distribution such as the danger of freezing. However, adequately reducing the temperature of the compressor may be difficult with air cooling in some applications.

11.2.2 Distribution of Compressed Air

The distribution of compressed air to the end-use equipment is achieved by a set of piping networks connected by various fittings. There are several types of pipes that can be used to channel compressed air ranging from rigid metallic pipes to flexible plastic tubing. The selection of the pipe type depends on a number of specifications including the diameter and the pressure required for the distribution piping. Through the piping network, the flow of compressed air is typically regulated by a set of valves and pressure regulators. Moisture traps are often fitted along the distribution lines to separate any condensate from compressed air.

The pressure of compressed air at the end-use equipment is always lower than the discharge pressure from either the compressor or the receiving tanks mostly for two reasons: flow pressure drop in the distribution piping and leaks. A brief description of each source of compressed air pressure loss is provided below.

11.2.2.1 Flow Pressure Drop

Due to the flow resistance of the compressed air through the pipes, valves, and other parts of the distribution system such as fitting and connectors, there is a pressure drop that can be estimated using the Darcy–Weisbach equation:

$$\Delta P = \rho \left(f . \frac{L}{D} + \sum K \right) . \frac{V^2}{2} \tag{11.6}$$

where

ΔP is the total flow pressure drop of the compressed air through the distribution system (Pa).
ρ is the average density of compressed air through the distribution system (m^3/kg).
f is the friction factor.
Σ^K is the loss coefficient due to valves and fittings in the distribution system.
L is the total length of pipe in the distribution system (m).
D is the internal diameter of the pipe (m).
V is the average velocity of the compresses air flow (m/s).

For more details on the calculation procedure using Eq. (11.6) to determine the flow pressure drop through pipes, the reader can refer to the *ASHRAE Handbook* (2009). In general, the calculations of the pressure drop for compressed air flow using equations such as Eq. (11.6) are not recommended because they are complex and are not accurate due to the inherent assumptions. For instance, several assumptions are generally made to determine the friction factor and the loss coefficients for valves and fittings. Rules of thumb and charts can be used instead to determine at least the order of magnitude of compressed air pressure drop through the distribution system. However, Eq. (11.6) can be used to pinpoint some measures that can be considered to reduce the flow pressure drop through the distribution piping for a compressed air system:

- The pressure drop increases with the length of the distribution system. Therefore, an optimized piping layout can reduce the pressure drop and thus energy use of the air compressed system. In particular, it is important to reduce the number of valves and bends in the distribution network.
- The velocity of the compressed airflow should be as low as possible to reduce the pressure drop through the distribution system. This velocity reduction is recommended especially for installations where the distribution system is long. A reduction in the flow velocity can be achieved either by decreasing the mass flow rate required (i.e., end-use load), lowering the temperature of the compressed air, or increasing the pipe diameter.

It should be noted that the pressure drop through the distribution system constitutes a waste of energy. Indeed, the discharge pressure has to be increased to compensate for the flow pressure drop. Equations (11.3) and (11.4) indicate that the increase of the compression ratio (P_o/P_i) leads to an increase in the mechanical power provided to the compressor by the driver (i.e., electrical motor). For instance, a pressure drop of 50 kPa in a compressor designed to increase the pressure from 100 kPa (1 atm) to 700 kPa (7 atm) corresponds to 3 percent waste in the mechanical and thus electrical power of an air compressed system.

The pressure drop problems in the distribution due to flow resistance are generally inherent to the design of the piping layout and therefore should not be the focus for the energy audit of a compressed air system unless the system has to be replaced. Instead, the auditor should focus on the identification of measures to reduce energy waste for the existing air distribution system.

11.2.2.2 Air Leaks

The energy losses due to leaks are generally significant in industrial compressed air installations. It is estimated that leaks can represent as much as 25 percent of the output of an industrial compressed air system (Terrell, 1999). Typically leaks can occur due to poor connectors and fittings, bad valves, or small holes in a rubber or plastic tubing or hoses. Nonproducing machines with compressed air left on can also be common sources of leaks.

To identify air leaks several methods can be used during an audit walk-through. These methods can be grouped into two categories:

1. *Simple Procedures.* Two inexpensive methods can be considered to identify the location of air leaks:
2. *Measurement Methods.* The detection of air leaks can be performed using ultrasound equipment. Ultrasound leak detectors are available and can be used to scan an entire compressed air distribution system to locate any air leaks easily.

Maintenance personnel in an industrial facility can be trained to detect leaks in the compressed air systems during periods when there are limited or no production activities (i.e., weekends, or scheduled plant shutdown periods). It is estimated that with little investment in a regular maintenance program for detecting leaks, significant savings in the operating cost of compressed air systems can be achieved.

When leaks are identified, the waste in the amount of compressed air can be estimated by simple calculations. Indeed, the waste in mass flow rate $\Delta\dot{m}_a$ of compressed air through a hole can be estimated using Fliegner's expression:

$$\Delta\dot{m}_a = \sqrt{\frac{2}{R_a}}.C_L.A_L.P_o.T_o^{-\frac{1}{2}} \tag{11.7}$$

where

$\Delta \dot{m}_a$ is the amount of mass flow rate in compressed air wasted through the leak (kg/s).
P_o is the pressure of the compressed air leaving the leak (Pa).
T_o is the temperature of the compressed air leaving the leak (K).
C_L is the flow coefficient through the hole. It depends on the shape and size of the hole. For round holes, C_L can be set to be 0.65.
A_L is the area of the hole (m^2).

11.2.3 Utilization of Compressed Air

There are several applications for compressed air. In general, end-use equipment can be grouped into classes:

- *Static End-Users* including pneumatic control equipment such as actuators or pressure regulators. The action of these end-users depends only on the presence or absence of compressed air. These end-users can be found in both commercial and industrial facilities.
- *Dynamic End-Users* including assembly tools (such as screwdrivers and nut-runners) and air motors (such as tools for drilling and grinding). The action of these end-users depends on the pressure and flow of compressed air. Most industrial applications of compressed air involve dynamic end-use equipment.

To reduce energy waste, the pressure delivered by the compressed air system should correspond to the highest pressure required by the end-users (with some safety factor to account, for instance, for the flow pressure drop as discussed in Section 11.2.2). If the available pressure is higher than the pressure utilized by the end-use equipment, energy is wasted because the unused high pressure increases the mechanical power required by the compressor [see Eqs. (11.3) and (11.4)].

In general, the use of compressed air should be avoided for applications that can be performed by other resources. For instance, electrically driven tools can be used instead of air-driven tools. Also, in applications involving drying, evaporation from a surface, or biological production within a large tank, blowers can be used instead of compressed air. It is estimated that if the pressure requirement is less than 200 kPa (2 atm), a blower can save 50 percent or more of the energy that would be used by compressed air (Terrell, 1999).

11.3 Common Energy Conservation Measures for Compressed Air Systems

One of the tasks involved in energy auditing is to collect data and information relevant to the design, operation, and maintenance of various systems. For compressed air systems, some of the information may need to be obtained by interviewing plant operators, processing engineers, or maintenance personnel. In particular, the following data can be gathered through one or several interviews with operating personnel:

1. Current operation controls of the compressed air system for the facility
2. Existing problems with the quality, quantity, and pressure of the compressed air
3. Design data and drawings and any maintenance records including compressor operating log and leak detection maintenance schedule
4. Nameplate data for all equipment associated with the compressed air system
5. Future plans for upgrading the compressed air system including compressor replacement, changes in the compressed air usage, and installation of instrumentation

The data gathering can be time consuming depending on the complexity of the facility and the compressed air system. However, it is important that the energy auditor obtain enough data to understand the existing design details, current operation and maintenance procedures, and future plans to be able to propose energy conservation measures that not only save energy use and cost but also provide a more reliable compressed air system.

A description of selected energy conservation measures commonly suitable for compressed air systems are provided below with some specific calculation procedures and illustrative examples.

11.3.1 Reduction of Inlet Air Temperature

In some compressed air installations, the compressor draws inlet air from indoors and in particular from the mechanical room or the space where it is located. This inlet air temperature can be high (30°C or higher) and is most likely warmer than outdoor ambient air. As indicated in both Eqs. (11.3) and (11.4), higher inlet air temperatures imply higher mechanical and thus electrical energy requirements. A simple measure to reduce the inlet air temperature is to install pipes to connect the compressor to outdoors so the compressor draws ambient air. To ensure that the ambient outdoor air remains cool when it gets to the compressor, the pipes should be insulated.

The electrical energy savings ΔkWh_{comp} associated with a reduction of the inlet air compressor can be calculated using the following expression based on Eq. (11.5):

$$\Delta kWh_{comp} = \frac{\dot{W}_m .LF_{comp} .N_{h,comp} .(T_{i,e} - T_{i,r})}{\eta_M .T_{i,e}} \tag{11.8}$$

where

$T_{i,e}$ = the annual average inlet air temperature before the retrofit (K).
$T_{i,r}$ = the annual average inlet air temperature after the retrofit (K).

In general, $T_{i,r}$ can be assumed to be the same as the annual average outdoor temperature. An illustration of the calculation procedure using Eq. (11.8) is presented by Example 11.3.

Example 11.3

A compressed air system has a mechanical power requirement of 100 kW with a motor efficiency of 87 percent. The intake air for the compressor is from the interior of the mechanical room (where the compressor is located and where the average annual temperature is 35°C). Determine the payback period of installing an insulated piping section to connect the intake of the compressor to the ambient outdoor air. Use the following information:

- The total cost of the piping system is $850.
- The annual average outdoor air is 12°C.
- The compressor is operating 5,000 hours per year with an average load factor of 80 percent.
- The cost of electricity is $0.05/kWh.

Solution

Using Eq. (11.8), the electrical energy savings due the reduction in the intake air temperature can be estimated:

$$\Delta kWh_{comp} = \frac{100kW * 0.8 * 5000hrs/yr * [(35 \ -12)K]}{0.87 * [(273+35)K]} = 34330kWh/yr$$

Therefore, the simple payback period for the installation of the piping system to connect the intake of the compressor to the outside air is:

$$SBP = \frac{\$850}{34330kWh/yr * \$0.05/kWh} = 0.5yr = 6months$$

11.3.2 Reduction of Discharge Pressure

When the maximum pressure required by all the end-use equipment in a facility is noticeably less than the air pressure delivered by the compressed air system, it is recommended to reduce the discharge pressure to reduce the compressor energy use. It is estimated that for every 15 kPa higher discharge pressure, 1 percent more energy input is required by the compressor. Higher than needed discharge pressures have other detrimental effects on the compressed air systems. In particular, overpressurization increases the waste of compressed air through leaks.

The electrical energy savings ΔkWh_{comp} due to the reduction in the discharge air pressure of the compressor can be calculated as follows:

$$\Delta kWh_{comp} = \frac{\%\dot{W}_m . W_m . LF_{comp} . N_{h,comp}}{\eta_M} \tag{11.9}$$

where $\%W_m$ is the percent reduction in mechanical power required by the compressor. Using either Eq. (11.3) or Eq. (11.4), the percent reduction in compressor power input $\%W_m$ can be estimated.

Example 11.4

A compressed air system has a mechanical power requirement of 50 kW with a motor efficiency of 90 percent. Determine the cost savings of reducing the discharge absolute pressure from 800 kPa to 700 kPa. Assume that:

- The compressor is operating 4,000 hours per year with an average load factor of 70 percent.
- The cost of electricity is \$0.05/kWh.

Solution

Assuming that the intake air pressure of the compressor is equal to 100 kPa (i.e., 1 atm), the reduction in the discharge pressure corresponds to a reduction in the pressure ratio P_o/P_i from 8 to 7. The percent reduction in the mechanical power requirement $\%W_m$ can be calculated using either Eq.(11.3) or Eq.(11.4):

For an isothermal compression:

$$\%W_m = \frac{Ln(8) - Ln(7)}{Ln(8)} = 6.4\%$$

Using Eq. (11.8), the electrical energy savings can be calculated:

$$\Delta kWh_{comp} = \frac{0.064 * 50kW * 4000hrs/yr * 0.70}{0.90} = 9950kWh/yr$$

Thus, the cost savings due to a reduction in the discharge air pressure are about \$500/yr.

For an adiabatic compression:

$$\%W_m = \frac{(8)^{1.4-1/1.4} - (7)^{1.4-1/1.4}}{(8)^{1.4-1/1.4}} = 5.7\%$$

Using Eq. (11.8), the electrical energy savings can be calculated:

$$\Delta kWh_{comp} = \frac{0.057 * 50kW * 4000hrs/yr * 0.70}{0.90} = 8870kWh/yr$$

Thus, the cost savings for reducing the discharge air pressure are about \$450/yr.

11.3.3 Repair of Air Leaks

As discussed in Section 11.2.2, leaks in the distribution system result in unnecessary waste of compressed air and thus loss in the energy required to operate the compressed air system. Once the air leaks are identified and their size estimated, it is recommended to repair them as soon as possible. The energy savings associated with repair of compressed air leaks can be estimated using the following expressions:

For isothermal compressions:

$$\Delta kWh_{comp} = \frac{\Delta \dot{m}_a . N_{h,comp} . LF_{comp} . R_a . T_i . Ln\left(\frac{P_o}{P_i}\right)}{\eta_M} \tag{11.10}$$

For adiabatic or polytropic compressions:

$$\Delta kWh_{compm} = \frac{\Delta \dot{m}_a . N_{h,,comp} . LF_{comp} . R_a . T_i . \gamma}{(\gamma - 1). \eta_M} . \left[\left(\frac{P_o}{P_i}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \tag{11.11}$$

where $\Delta \dot{m}_a$ represents the mass flow rate of compressed air wasted through the leaks identified in the distribution system and can be estimated from Eq.(11.7). Example 11.5 provides an indication of the magnitude of the energy waste through a typical leak in a compressed air system. As shown in Example 11.5, the payback periods for the leak repair are generally very short.

It should be mentioned that compressed air can be wasted through the leaks even during periods when the end-use tools are not in operation because air is still in the distribution system. Moreover, overpressurization increases the amount of compressed air wasted through the leaks in addition to loss in the mechanical and electrical energy as discussed in Section 11.2.1.

Example 11.5

A leak of 5 mm has been detected in the compressed air distribution lines. Determine the payback period for repairing the leak. Assume an isothermal compression. Use the following information:

- The total cost of the leak repair is $150.
- The annual average compressed air temperature and absolute pressure are 20°C and 900 kPa, respectively.
- The compressor is operating 3,000 hours per year with an average load factor of 70 percent.
- The annual average ambient air temperature and pressure are 15°C and 100 kPa, respectively.
- The electrical motor efficiency is 90 percent.
- The cost of electricity is $0.05/kWh.

Solution

Using Eq. (11.7), the waste in the compressed mass flow rate through the leak can be estimated:

$$\Delta \dot{m}_a = \sqrt{\frac{2}{287}} * 0.65 \ * 0.0000196 * 800000 \ .(273+20)^{-\frac{1}{2}} = 0.050 kg/s$$

For an isothermal compression, the annual electrical energy waste by the leak is calculated using Eq. (11.10):

$$\Delta kWh_{comp} = \frac{(0.050kg/s)*3000hrs/yr*0.70*287J/kg.K.(273+15)*Ln\left(\frac{800}{100}\right)}{0.90} = 20050kWh/yr$$

Therefore, the simple payback period for the piping system that connects the intake of the compressor to the outside air is:

$$SBP = \frac{\$150}{20050kWh/yr*\$0.05/kWh} = 0.15yr = 2months$$

11.3.4 Other Energy Conservation Measures

Other energy conservation measures that can be considered for compressed air systems are listed below:

- Replacement of inefficient compressors with new and high-efficiency compressors.
- Reduction of the compressed air usage and air pressure requirements by making some modifications to the processes.
- Installation of heat recovery systems to use the compression heat within the facility for either water heating or building space heating.
- Installation of automatic controls to optimize the operation of several compressors by reducing part-load operations.
- Use of booster compressors to provide higher discharge pressures. Booster compressors can be more economical if the air with the highest pressure represents a small fraction of the total compressed air used in the facility. Without booster compressors, the primary compressor will have to compress the entire amount of air to the maximum desired pressure.

It should be mentioned that detailed technical and economical analyses should be carried out to ensure that the above-listed measures are both cost-effective and nondisruptive to the proper operation of the entire air compressed system.

11.4 Summary

This chapter provides basic information on the operation procedures for compressed air systems typically used in industrial facilities but also in some commercial buildings. Several energy conservation measures have been presented. In addition, simplified analysis methods based on fundamental thermodynamic principles have been provided to estimate energy and cost savings from easily implemented operation and maintenance measures specific to compressed air systems. For industrial applications, significant energy savings can be obtained by improving the energy performance of compressed air systems in all phases of production, distribution, and utilization.

Problems

11.1 In an industrial facility located in Denver Colorado, a compressed air system has a mechanical power requirement of 250 kW with a motor efficiency of 85 percent. The average compressed air temperature and absolute pressure are 25°C and 900 kPa, respectively. The compressor is operated 5,000 hours/yr with an average load factor of 80 percent. If the cost of electricity is \$0.08/kWh, determine the energy and cost savings by reducing the discharge absolute pressure to (i) 800 kPa, and (ii) 750 KPa. (Use both isothermal and adiabatic compression models.)

11.2 For the same conditions defined in Problem 11.1, estimate the energy and cost savings from repairing a leak. Assume the size varies from 1/8 in. to 2 in. (in an increment of 1/4 of an inch). If the cost of repairing the leak is $1,250, determine the smallest leak worth repairing if the longest payback period that is acceptable is one year.

11.3 A facility requires three levels of compressed air streams: 2 kg/s of 500 kPa (5 atm) air, 4 kg/s of 800 kPa (8 atm) air, and 5 kg/s of 900 kPa (9 atm) air. The existing installation generates a compressed air stream only at 900 kPa. Estimate the energy and cost savings if three separate compressors are used (each to compress air at one of the three levels). The facility is operated 60 hrs/week and 300 days/year. The cost of electricity is $ 0.08/kwh.

11.4 A compressor is operated 5,000 hrs per year to generate 800 kPa air. Heat recovery can be used to increase the inlet compressor air temperature to 35°C instead of compressing air from 100 kPa and 15°C.

(a) Estimate the energy and cost savings for using higher inlet temperature. Assume that the electricity cost is $0.07/kWh.

(b) Determine the required cost for the heat recovery system if the payback period does not exceed five years.

(c) In an economy of 6 percent inflation and with the installed cost found in (b) for the heat recovery system, determine the threshold value of the electricity price for which it is no longer cost-effective to invest in the heat recovery system.

12

Thermal Energy Storage Systems

12.1 Introduction

Thermal energy storage (TES) is generally defined as the temporary storage of energy for later use when heating or cooling is needed. For heating applications, heat storage systems are used with energy stored at high temperatures [above 20°C (68°F)]. For cooling applications, energy is stored at low temperatures [below 20°C (68°F)]. The concept of TES is not new and was used a few centuries ago to cool churches using blocks of ice that were stored in the cellar.

Recently, TES technology has been shown to be effective especially in reducing the operating costs of cooling plant equipment. By operating the refrigeration equipment during off-peak hours to recharge the storage system and discharge the storage during on-peak hours, a significant fraction of the on-peak electrical demand and energy consumption is shifted to off-peak periods. Cost savings are realized because utility rates favor leveled energy consumption patterns. The variable energy rates reflect the high cost of providing energy during relatively short on-peak periods. Hence, these rates constitute an incentive to reduce or avoid operation of the cooling plant during on-peak periods by the cool storage system. A large differential between on- and off-peak energy and peak consumption rates generally makes cool storage systems economically feasible.

Some electric utility companies actually encourage the use of TES systems to reduce the cost required to generate on-peak electric power. Indeed, the need to build new generation plants in order to meet the demand during on-peak hours can be eliminated by promoting the use of off-peak power. Thus, utilities have initiated different rate structures to penalize the use of electric power during on-peak periods. In addition to the differential charges for on-peak versus off-peak energy rates, the utilities have imposed demand charges, based on the monthly peak demand. These demand charges are intended to recover fixed costs such as investing in new generation plants, and transmission or distribution lines. In the United States, it is estimated that 35 percent of the electric peak demand is due to cooling. Therefore, the use of TES systems can be a good alternative to delay the use of chillers to meet space cooling especially for commercial buildings.

TES systems can also be used to reduce the size and thus the initial cost of the cooling equipment especially in applications where peak loads occur only for a limited period during a year such as churches. For instance, a church can have a peak cooling load of 300 kW over a three-hour period that occurs once a week. Instead of using a chiller of 300 kW to operate for three hours in order to provide the required 900 kWh of cooling load, a 90-kW cooling system can be installed. The same cooling load (900 kWh) can then be produced by operating the 90-kW cooling system for a longer period (at least a 10-hour period to account for any storage losses).

Due to compressor efficiency and storage losses, cooling plants with TES systems may actually consume more energy that cooling plants without TES systems. However, TES systems if designed and controlled properly can reduce the overall operating costs of cooling plants. To determine the potential operating cost reduction, the auditor should carefully consider the various factors that affect the design and operation of TES systems. In the following sections, an overview of the types of TES systems as well as factors that affect both the design and operation of cooling plants with TES systems is provided. Finally, some simplified calculation examples as well as results of parametric analyses, reported in the literature, are presented to illustrate some typical cost savings incurred due to adequately installing and properly controlling TES systems for space cooling applications.

12.2 Types of TES Systems

Thermal energy storage can be achieved by two mechanisms:

1. Sensible energy storage by increasing (for heating applications) or decreasing (for cooling applications) the temperature of the storage medium (water for instance)
2. Latent energy storage by changing the phase of the storage medium (phase change materials, PCM, eutectic salt solutions, or ice–water mixtures)

For cooling applications, there are several types of TES systems that have been installed in various commercial buildings and industrial applications. Among these TES systems are:

1. *Chilled water storage systems:* These systems typically consist of tanks where chilled water (temperature above freezing point) is stored before it is used during off-peak periods. There is no change of phase for the water in these systems and thus they can store a limited density of energy. Water is selected because it has the highest specific heat of all common materials (4.18 kJ/kg×°C). Typically, a tank volume varying from 0.09 to 0.17 m^3 is required to store 1 kWh of energy using chilled water.
2. *Eutectic salts:* In these systems, a solution of salts is used to store energy at low temperatures. The advantage of these systems is that temperatures below 0°C can be achieved before the solution is frozen. In addition, some salts have heat of fusion comparable to that of ice. It should be noted that the solution of salts needs to be mixed in a controlled ratio to ensure that the mixture melts completely and has the same composition in both liquid and solid phases. For eutectic salts, the volume requirement for the storage tank is estimated to be 0.05 m^3/kWh.
3. *Ice storage systems:* In these systems, the water is transformed into ice which is stored in tanks. Therefore, the water can be present in the form of two phases (liquid and solid) inside the tank. Typically, the ice is made during the off-peak periods (charging) and is melted during on-peak periods (discharging). Ice storage systems have a higher energy density compared to chilled water systems. Thus, the volume of the storage tank required for ice systems is significantly less than that for chilled water systems (almost one-fourth). In addition, ice storage systems allow for innovative HVAC system design such as cold air distribution systems which have lower initial costs compared to conventional distribution systems. Common ice storage systems include:
   (a) Ice harvesters [Figure 12.1(a)]. In these systems, thin ice layers are formed around vertical plates (evaporator) that are sprayed with water pumped from the tank. The ice layers are harvested to the storage tank by circulating hot gases through the evaporator. The ice mixed with water is stored in the tank to obtain what is often referred to as ice slurry. The volume requirement for the storage tank used in ice harvesters is about 0.025 m^3/kWh.

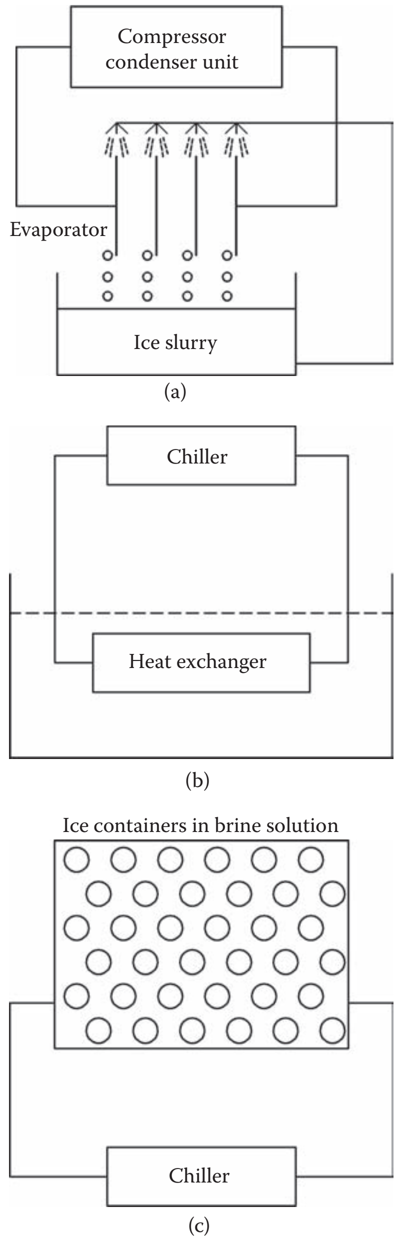


FIGURE 12.1 Commonly installed ice storage systems: (a) ice harvester storage system; (b) ice-on-coil with internal melt storage system; (c) containerized ice storage system.

(b) Internal melt ice-on-soil storage systems [see Figure 12.1(b)]. In these systems, direct expansion coils are fitted inside the storage tank which is filled with water. A brine solution (mixture of water and ethylene glycol) is typically circulated through the coils with a temperature in the range of –6°C to –3°C. In the charging mode, ice layers are formed around the coils. In the discharging mode, the ice is melted by circulating a warm brine solution in the coils to be cooled in order to provide space cooling. The volume of the storage tank required for internal melt ice-on-coil systems varies from 0.019 to 0.023 m^3/kWh.

(c) External melt ice-on-coil storage systems. They are similar to the internal melt ice-on-coil system in that the ice is made around coils filled with brine solution. However, the water that results from melting ice in the storage tank is used directly to provide space cooling. Typically, a volume of 0.023 m^3/kWh is used to size storage tanks for external melt ice-on-coil systems.

(d) Containerized ice storage systems [see Figure 12.1(c)]. In these systems, small containers of various shapes (typically spherical) filled with water are used inside a tank to store energy. The water inside the containers is frozen by directly cooling the solution inside the tank (which acts as the evaporator). The typical volume requirement for containerized ice storage systems is 0.048 m^3/kWh.

12.3 Principles of TES Systems

Figure 12.2 illustrates a typical configuration for a cooling plant with a TES system. Instead of one chiller, some cooling plants may have a base-load chiller that provides cooling up to a threshold load (determined by the capacity of the base chiller). Any additional cooling loads are either met directly by a second chiller (TES chiller) or the storage system. This second chiller is used to charge the TES system during unoccupied periods (or off-peak hours).

As discussed in Chapter 9, the energy efficiency of a cooling system is characterized by its coefficient of performance (COP) which is defined as the ratio of the heat extracted divided by the energy input required. The maximum theoretical value for COP can be estimated using the ideal Carnot cycle COP

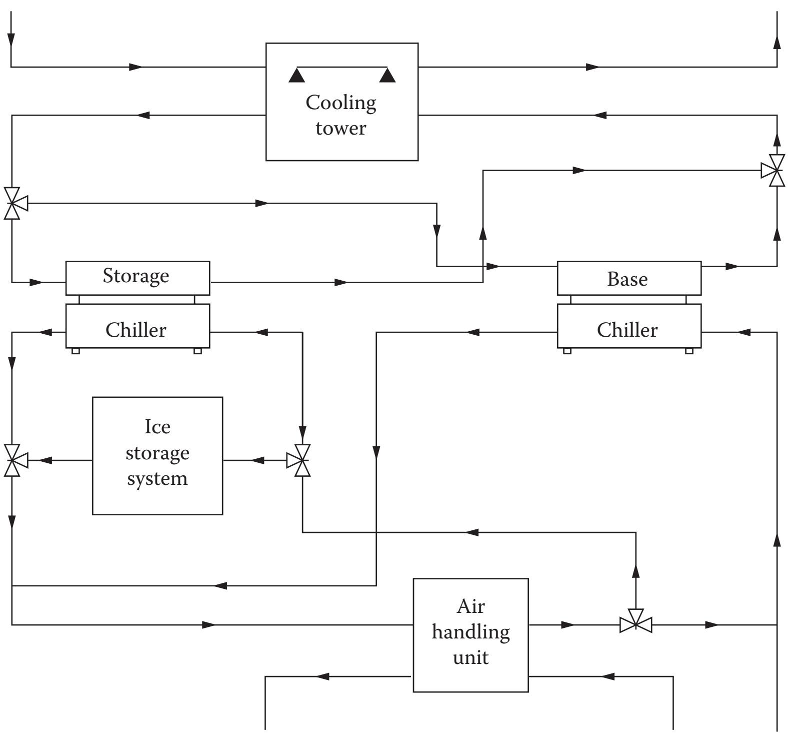


FIGURE 12.2 Typical configuration for a cooling plant with a TES system.

which can be expressed in terms of the absolute temperature of the evaporator T_C (the lowest temperature in the cycle) and the condenser T_H (the highest temperature in the cycle) as follows:

$$COP_{Carnot} = \frac{T_C}{T_H - T_C} \tag{12.1}$$

It can be seen from Eq. (12.1) that decreasing the evaporator temperature reduces the COP. Therefore, the chiller energy efficiency is reduced when it is operated at low temperatures. In particular, the chiller operates at low efficiencies when it is used to charge an ice storage tank rather than to meet the space cooling load directly. In addition, there are energy losses from the storage tanks when ice is kept for long periods of time without being utilized. Therefore, the energy used to meet a cooling load through a TES system may actually be higher than that consumed by a chiller used to provide cooling directly. It is therefore important to pinpoint that the main advantage in the use of cooling plants with TES systems is not primarily to save energy but rather to reduce the electrical demand during on-peak periods or to provide additional cooling beyond the chiller capacity. In most cases, TES systems are used primarily to reduce the cost of operating a cooling plant while maintaining indoor thermal comfort. The factors that affect the operation of cooling plants with TES systems include the following:

- TES and cooling plant performance during charging/discharging
- Control strategies used to operate the TES system
- Utility rate structures (real-time-pricing and time-of-use rates including ratchet clauses)
- Cooling load profile and noncooling electrical load profile

Some of the effects of these factors on operating cost savings of cooling plants are discussed in the following sections.

12.4 Charging/Discharging of TES systems

The operation of TES systems may lead to partial—rather than full—charging and discharging of the storage tank. The performance of TES systems under partial charging or discharging can affect the overall performance of the cooling plant especially for ice storage systems characterized by ice breakage effects for low-charging levels. Typically, charging is performed during nighttime (off-peak period) and discharging occurs during daytime (on-peak period). In some instances, the ice storage tank is neither fully charged nor fully discharged. Partial charging and discharging cycles may occur when the peak load is not high or when the charging/discharging time is limited as is the case for some real-time pricing rates (Krarti et al., 1999c, and Henze and Krarti, 1999). In other cases, the chiller may be too small to fully charge the entire tank during one charging period (i.e., one day). In these cases, the storage tank operates almost exclusively under partial charging and discharging cycles.

During partial charging and discharging sequences, thin ice layers can be trapped between water layers within an internal melt ice-on-coil storage tank. In particular, the coils are surrounded by ice that can be trapped by a water layer during a charging cycle. Figure 12.3 shows a cross-section view of one coil during a charging cycle when the ice layers for adjacent coils do not overlap. When the tank is discharged again, the heat transfer between the brine circulating within the coils and the surrounding ice is relatively high. Similarly, Figure 12.4 gives a cross-section view of the coil during discharging cycle.

Partial charging/discharging cycles affect the TES performance which is generally difficult to model. Some of the effects that characterize ice storage systems especially under partial charging/discharging operation include:

- Effects of water flow within the tank during charging or discharging cycles
- Gravitational effects that deform the ice formations around the coils
- Effects of ice breakage at the end of discharging cycles

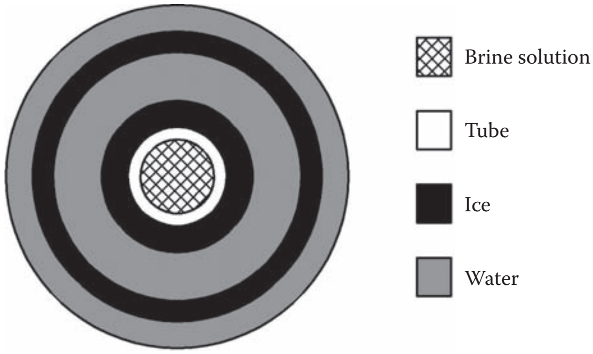


FIGURE 12.3 Cross-section of the coil during charging period.

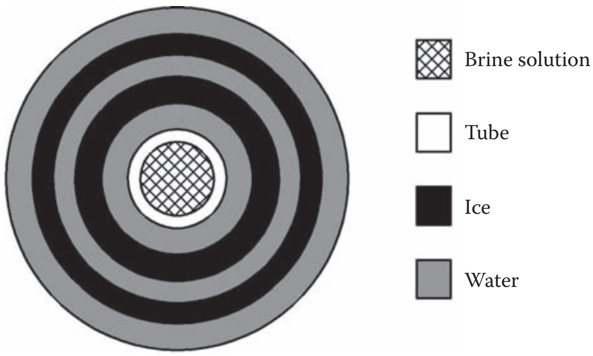


FIGURE 12.4 Cross-section of the coil during discharging period.

Only a few detailed models have been developed to predict the behavior and determine the thermal performance of ice storage systems. Most of these models can typically simulate the ice tank thermal performance during full charging and discharging cycles, but not during partial charging and discharging cycles. In particular, the model developed by Jekel, Mitchell, and Klein (1993) is based on fundamental principles of mass and energy conservation applied to the coil simulated as one node segment. The accuracy of the model is relatively marginal with an average error of 11 percent when compared to the manufacturer's data (Drees, 1994).

Strand, Pederson, and Coleman (1994) have proposed a simplified model for an internal melt ice-on-coil storage system using a nonlinear correlation to predict the heat transfer rate between the brine solution and the ice–water interface. The correlation coefficients were obtained from data specific to one manufacturer's model. Therefore, the model of Strand, Pederson, and Coleman cannot handle different ice tank designs. The model developed by Drees and Braun (1995) is based on the model proposed by Jekel, Mitchell, and Klein (1993) and uses a thermal network technique to find the radius of the ice.

The model developed by Carey, Mitchell, and Beckman (1995) for the indirect ice storage system was integrated in the TRNSYS computer simulation program. However, the model has some restrictions. For instance, the ice–water interface is not allowed to be larger than one-half of the distance between the coils. The numerical model developed by Neto and Krarti (1997a) is more complete and has been thoroughly validated against measured data (Neto and Krarti, 1997b). Neto and Krarti's model uses a thermal network and an iterative approach technique to find the radius of the ice during the charging cycle and the radius of the water during the discharging cycle. In addition, Neto and Krarti's model takes into account the overlapping phenomenon of the ice and water due to layer superposition.

However, all the models listed above simulate only full charging or discharging cycles, and cannot be applied to model accurately the thermal performance of an ice tank under partial charging and discharging cycles. Only the models developed by Vick, Nelson, and Yu (1996), West and Braun (1999), and Kiatreungwattan (1998) can simulate partial charging and discharging cycles. All three models are based on simplifying assumptions to simulate the performance TES systems. For instance, some models neglect the overlapping of the ice and water layers which reduces the heat transfer between the brine solution and the ice/water. However, all three models can be used to evaluate the effects of partial charging/discharging operation on the overall performance of the cooling plant.

Based on the results of experiments performed under a controlled laboratory setup, Kiatreungwattan (1998) compared the chiller energy use (expressed in kWh) operating with a single cycle of charging with that obtained for a sequence of partial charging and discharging cycles. The volume of the ice at the final state is set to be the same for both sequences (i.e., full charge and partial charge). Figures 12.5 and 12.6 show the chiller thermal output and electrical energy use for, respectively, full charging cycle, and a sequence of partial charging and discharging cycles (corresponding to a sequence of charging–discharging–charging cycles). Table 12.1 provides a summary of the measured performance results for both charging and discharging sequences. The results indicate that one full charging cycle consumes significantly less electrical energy than a sequence of partial charging and discharging cycles with a total energy use of 238.5 kWh for a full charging cycle and 531.1 kWh for the sequence of partial charging and discharging cycles. The increase in the chiller electric energy use for the partial charging or discharging operation is attributed to two main factors:

1. Longer chiller operation time (the charging period is 8.7 hours for one single full charging cycle but is 12.7 hours for the partial charging and discharging sequence)
2. More ice made (measured by the total charging energy which is 130.0 ton-hrs for full charging versus. 247.5 for the sequence of partial charging and discharging)

In addition, the results of Table 12.1 indicate that the average chiller efficiency (estimated by dividing the total chiller electrical power use by the total chiller load) is better for a full charging cycle (average

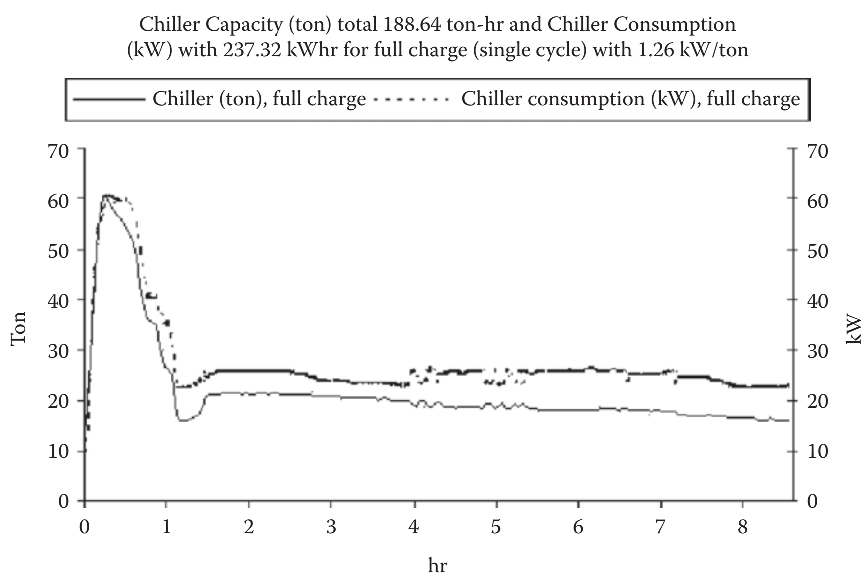


FIGURE 12.5 Chiller capacity and chiller power consumption during a single cycle of full charging (obtained from an experimental study by Kiatreungwattan, MS Thesis, University of Colorado, Boulder, 1998).

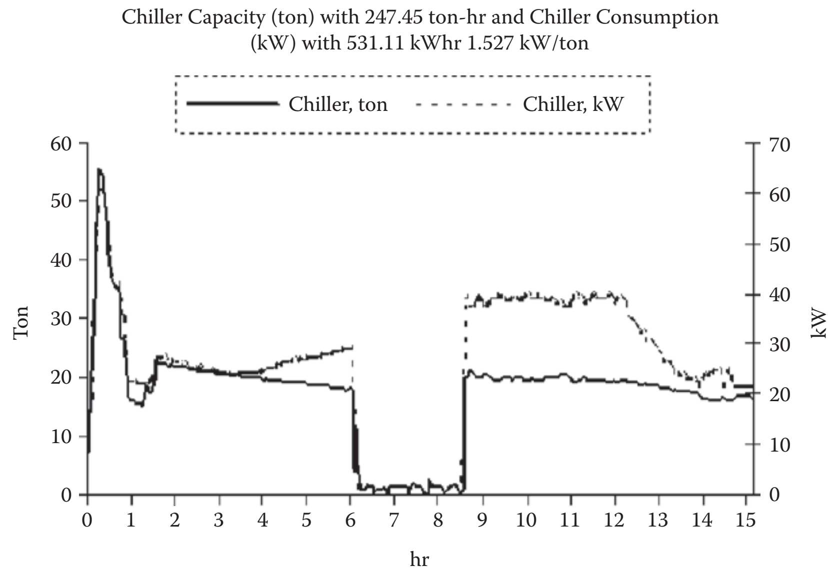


FIGURE 12.6 Chiller capacity and chiller power consumption during partial charging and discharging cycles (obtained from an experimental study by Kiatreungwattan, MS Thesis, University of Colorado, Boulder, 1998).

TABLE 12.1 Average Charging Rate and kW/ton for the Experiments Performed by Kiatreungwattan (1998)

	One Single Full Charging Cycle	Sequence of Partial Charging/Discharging Cycles
Total charging time (hr)	8.65	12.70
Total charging load (ton-hr)	130.04	247.45
Average Charging rate (ton)	15.03	19.48
Total chiller consumption (kWh)	238.47	531.11
Total chiller capacity (ton- hr)	189.67	347.62
Average chiller eåci ency (kW/ton)	1.257	1.527

1.26 kW/ton) than for a sequence of partial charging and discharging cycles (1.53 kW/ton). This result is attributed to the part-load performance of the chiller. On average, the load on the chiller operating under a partial charging and discharging sequence is lower than that experienced by the chiller under a full charging cycle.

An average charging rate (expressed in tons) is estimated as the ratio of total charging energy (ton-hrs) over the cumulative charging time (hrs). The average charging rate is an indicator of the average heat transfer effectiveness of the ice storage tank. Table 12.1 indicates that the average charging rate is higher for a partial charging and discharging cycle (19.5 tons) compared to that for a full charging cycle (15 tons). This result is due to the fact that after discharging, the ice layer is reduced and better heat transfer is achieved between the brine and the water.

Thus, the experimental results indicate that the average heat transfer effectiveness of the ice tank is higher when operating with partial charging and discharging cycles than when operating with a full single charging cycle. However, the same results indicate that the chiller operates less efficiently (higher average kW/ton) under partial charging and discharging operating conditions.

In addition to the experimental analysis, simulation results obtained using numerical models (Vick, Nelson, and Yu, 1996; West and Braun, 1999; Kiatreungwattan, 1998) indicate that several variables affect the heat transfer effectiveness of the ice storage tank and the efficiency of the chiller under partial charging and discharging cycles. In particular, Kiatreungwattan (1998) found that:

- The average heat transfer effectiveness of the ice storage tank improves (by up to 10 percent) when a sequence of partial charging and discharging cycles is considered instead of one single full charging or discharging cycle.
- The average heat transfer effectiveness of the ice storage tank decreases (by up to 15 percent) when a shorter time length for the discharging cycle is used, in a sequence with only one partial discharging cycle.
- The average heat transfer effectiveness of the ice storage tank increases with higher brine flow rate. For instance, when the brine flow rate is increased from 60 gpm to 100 gpm, the charging effectiveness is increased by 28 percent. However, the increased flow rate may increase the pumping energy which is not considered in this analysis.

In summary, the reported results indicate that a sequence of partial charging and discharging cycles provides higher average heat transfer effectiveness than a single charging cycle. Meanwhile, the chiller efficiency for the same sequence of partial charging and discharging cycles is slightly lower than that for a single full charging cycle.

12.5 TES Control Strategies

Once installed, the TES system needs to be properly operated and controlled in order to achieve the desired cost savings. Several control strategies have been proposed and actually used to operate existing TES installations. The control of a TES system depends on the operating mode. Two TES operating modes are commonly considered full versus partial storage as discussed below.

12.5.1 Full Storage

This operating strategy is also called load shifting and consists of generating the entire on-peak cooling load during off-peak periods when no significant cooling load exists. Therefore, the TES system operates at full capacity and the chillers do not operate at all during the on-peak hours. Thus, in order to implement a full storage operating mode, the TES system has to be sized properly so it can hold enough energy to meet the cooling load for the entire design day on-peak hours. The full storage strategy is best suited for applications where the length of the on-peak cooling period is short compared to the off-peak period when the TES system can be charged. It can be an effective operating strategy when the on-peak demand charges are high. Moreover, the control under a full storage operating strategy is simple because all that is needed is a timer clock to define charging and discharging periods. However, the full storage strategy requires large chiller and storage capacities and thus high initial costs.

12.5.2 Partial Storage

It can be defined as an operating strategy when the TES system meets only part of the on-peak cooling load. The remainder of the load is provided directly by the chiller. Partial storage requires lower initial costs than full storage inasmuch as both the chiller and the storage for partial storage are of smaller size. However, partial storage operating strategies require more complex controls than full storage systems. Some of these controls are discussed below.

12.5.2.1 Chiller-Priority Control

The simplest of TES control strategies used is the chiller-priority control. For this strategy, the chiller runs continuously under conventional chiller control (direct cooling) possibly subject to a demand-limit and the storage provides the remaining cooling capacity if required.

The simplicity lies in the fact that the conventional chiller control is not altered, yielding high average part loads and a smooth demand curve, however, the meltdown of the ice is not well controlled to allow for maximal demand reduction. In addition, there is no accounting for the time-of-day dependent energy rate structure.

12.5.2.2 Constant-Proportion Control

This control strategy considers that the storage meets a constant fraction of the cooling load under all conditions. Thus, neither the chiller nor the storage has priority in providing cooling. This simple control strategy provides a greater demand reduction than chiller-priority control because the chiller capacity fraction used for a particular month will track the fraction of the annual cooling design load the building experiences during that month. For example, in a month in which 50 percent of the annual design load occurs, only 50 percent of the chiller capacity will be requested.

Constant-proportion control is rather easy to implement in practice by assigning a fixed fraction of the total temperature difference between brine supply and return flow to be realized by the storage and the remainder by the chiller. Finding the best load fraction for each application is a matter of trial and error. Caution should be exercised to be sure that the chiller can always meet the remaining load fraction.

12.5.2.3 Storage-Priority Control

As the name reflects, storage-priority control requires melting as much ice as possible during the on-peak period. It is generally defined as that control strategy which aims at fully discharging the available storage capacity over the next available on-peak period. Thus both the simultaneous operation of the chiller plus the storage and the terminal state-of-charge are specified; yet how this is accomplished in detail is not known (Tamblyn, 1985).

The following implementation of storage priority has been suggested by Henze, Dodier, and Krarti (1997b): the chiller is base-loaded during off-peak hours to recharge the ice storage for the next on-peak period. The chiller operates in one of two modes during the on-peak period. In the first mode, the chiller operates at a reduced capacity in parallel with the storage during on-peak hours so that at the end of the on-peak period the storage inventory is just depleted. If this is not possible without prematurely depleting storage, the control switches to the second mode in which the storage provides a constant proportion of the load in each on-peak hour similar to constant-proportion control.

12.5.2.4 Optimal Controls

Several optimal control strategies have been proposed in order to minimize energy use and operating cost of active TES systems while maintaining occupant comfort. The main requirement for an optimal control strategy is typically the forecasting of cooling load and weather. The key references for optimal operating controls of active TES systems are Braun (1992); Drees and Braun (1996); Henze, Krarti, and Brandemuehl (1997a); and Henze, Dodier, and Krarti (1997b). Other studies have evaluated the combined use of both building thermal mass (passive TES) and ice or chilled water storage tanks (active TES), a load management strategy to shift on-peak building HVAC cooling load to off-peak time (Kintner-Meyer and Emery, 1995; Henze Clemens, and Knabe, 2004; Henze et al., 2005).

For instance, the optimal control in Henze, Krarti, and Brandemuehl (1997a) is defined as that sequence of control actions that minimizes the total operating cost (i.e., including demand and energy charges) of the cooling plant over the simulation period. The optimization technique used is the predictive optimal controller that is proposed by Henze, Dodier, and Krarti (1997b) and is based on a "closed-loop" optimization approach. In particular, prediction of the weather, price, and loads is updated at the beginning of each time step (typically one hour) over the optimization period. The planning horizon for the proposed controller consists of a fixed-length moving window over an entire simulation period. At each time step, only the action of the first hour is executed. In this approach, a model for the cooling plant and the TES system (the "planning" model) is needed to perform the closed-loop optimization calculation. The operating cost savings incurred by the optimal control is determined using actual plant behavior. A more detailed description and discussion of the predictive optimal controller is provided in Henze, Dodier, and Krarti (1997b).

12.5.3 Utility Rates

There are typically two categories of utility rates that can provide some incentives for considering the installation of TES systems in cooling plants. The first rate structure is common and is known as the time-of-use (TOU) rate whereas the second is relatively new and is currently available from only a limited number of utilities: the real-time-pricing (RTP) rate.

12.5.3.1 TOU Rates

These rates are currently common for most electric utility companies. They typically penalize energy use during predefined on-peak periods. Indeed, the day is divided into two or more periods during which the charges for power demand or energy use are set. Typically, the hours when cooling is needed are part of the on-peak period because the demand for electrical power is the highest. The charges for both energy and demand are greater during the on-peak period in an attempt by the utilities to level off the electrical power demand curve so as to avoid the need to operate power generation plants for short periods of time.

12.5.3.2 RTP Rates

RTP rates are now offered by several utilities, especially in the commercial building sector. Generally the utilities determine their RTP rates based on the actual marginal costs of generating, transmitting, and distributing electricity. Although the details of RTP rate implementations may vary widely, they have several common features. Typically, the energy prices are set for a single day and are provided to the customer the preceding day. In addition, the RTP rates do not generally change in real-time, but are rather constant for periods ranging from one-half hour to five hours (Norford, Englander, and Wiseley, 1996).

Figure 12.7 shows the RTP rates used in the simulation analysis. The selected rates represent two possible scenarios of RTP rates. The sinusoidal cyclic rate is a fictitious rate and could represent an RTP rate for a week with low cost differential between on- and off-peak hours. The actual rate is extracted from real RTP rates that a U.S. utility applied during 1994. The rate is for a week with high cost differential between on- and off-peak hours.

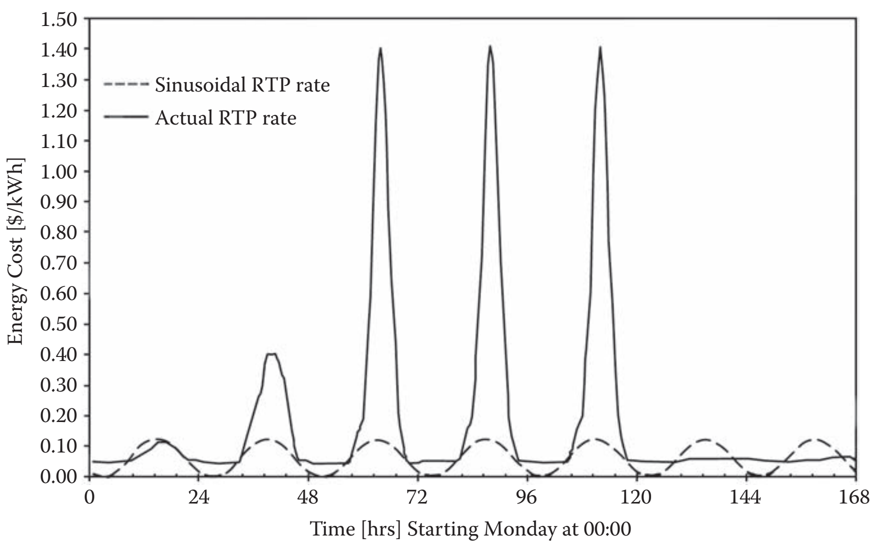


FIGURE 12.7 RTP rate profiles for the sinusoidal cyclic rate and the noncyclic actual rate.

Chapter 2 provides a more detailed discussion of electric utility rates typically available in the United States

12.6 Measures for Reducing Operating Costs

As discussed in the introduction, TES systems can be utilized to reduce the operating costs of cooling systems. Typically, three measures involving TES systems are available. These measures are:

1. Install a TES system in the existing cooling plant.
2. Install a TES system and replace the existing chiller (with a smaller capacity).
3. Improve the existing operating controls of the TES systems.

To assess the cost effectiveness of any of the above-listed measures, a detailed analysis is generally needed. Indeed, the utility rates that provide the major incentives for TES systems are set on an hourly or seasonal basis. Therefore, the use of a dynamic energy analysis tool is typically warranted to determine the overall operating cost of cooling plants with TES systems. However, simplified calculation methods can sometimes be used especially to assess whether a TES system is economically viable.

In this section, a simplified analysis method is presented to evaluate the cost-effectiveness of installing a TES system. In addition, a summary of typical results reported in the literature for selected operating cost savings strategies suitable for TES systems is provided.

12.6.1 Simplified Feasibility Analysis of TES Systems

The electrical power demand reduction due to the use of TES systems depends on the control strategy selected and can be estimated using the following simplified expression:

$$\Delta kW_{TES} = .\left(\frac{\dot{Q}_C}{SEER_{CHW}}\right)_e - .\left(\frac{(1-X).\dot{Q}_C}{SEER_{CHW}}\right)_r \tag{12.2}$$

With the electric demand reduction calculated above, savings in the demand charges can be estimated.

Similarly, the energy cost savings calculation incurred from the use of TES system can be calculated as follows:

$$\Delta EC_{TES} = \dot{Q}_C.N_{h,c}^{TES}.\left(\frac{C_{on-pk}}{SEER_{CHW}} - \frac{C_{off-pk}}{SEER_{ICE}}\right) \tag{12.3}$$

where

indices e and r indicate the values of the parameters, respectively, before and after retrofitting the cooling unit (i.e., adding the TES system).

$SEER$ is the seasonal efficiency ratio of the cooling unit. When available, the average seasonal COP can be used instead of the $SEER$. Typically the $SEER_{\mathrm{CHW}}$ for producing chilled water (to directly cool the space) is higher than $SEER_{\mathrm{ICE}}$ for making ice (to charge the TES system).

$\dot{Q}_C$ is the rated capacity of the cooling system.

X is the fraction of the on-peak cooling load (occurring during the hour when maximum electrical power demand is obtained) shifted to off-peak period.

$N_{h,C}^{TES}$ is the number of equivalent on-peak cooling full-load cooling hours that have been shifted during off-peak periods by using the TES system.

Example 12.1 illustrates a sample of a calculation to determine the cost-effectiveness of installing a TES system for a commercial building.

Example 12.1

Consider the cooling load profile for an office building shown in Figure 12.8. The cooling is provided by a chiller having a capacity of 1,000 kW and with an average seasonal COP of 3.5. The noncooling profile experienced by the same office building is illustrated in Figure 12.9. It is proposed to install an ice storage system. When making ice the chiller has an average COP of 3.0. Determine the simple payback period of installing an ice storage system if the cost of electricity is as follows:

- Energy Charges: \$0.07/kWh for on-peak hours (between 10:00 and 15:00 during weekdays) and only \$ 0.02/Kwh during other hours.
- Demand charges: \$15/kW during on-peak hours and \$ 0/kW during off-peak hours. The demand charges are assessed on a monthly basis.

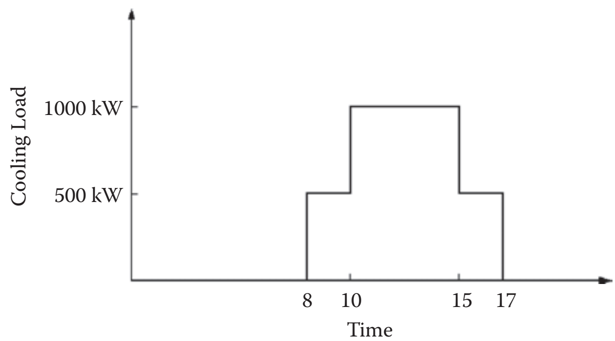


FIGURE 12.8 Cooling load profile for an office building (see Example 12.1).

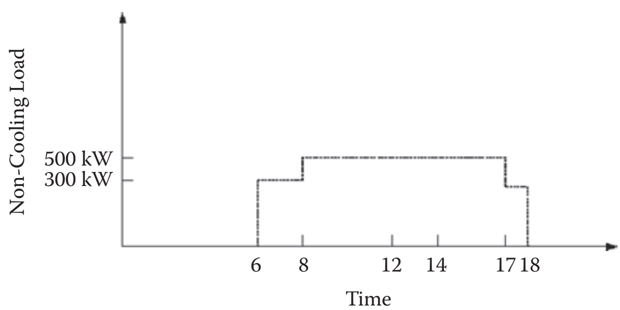


FIGURE 12.9 Electrical noncooling load profile for the office building used in Example 12.1.

The installed cost of the TES system is \$100/kW. Assume that the number of typical cooling days during the entire year both before and after the installation of the TES system is 250 days. The TES system is operated with demand-leveling control so that the power demand during on-peak hours never exceeds 500 kW (which is the maximum noncooling load).

Solution

In this example, the entire cooling load during on-peak has to be shifted and thus in Eq. (12.2) the fraction $X = 1$. Therefore, the savings in the electric power demand can be calculated using $SEER_{CHW} = 3.5$; and $Q_C = 1{,}000$ kW:

$$\Delta kW_{TES} = 1000kW * \left(\frac{1}{3.5}\right) = 286kW$$

Thus, the cost savings due to demand charges are: 286 kW*$15/kW*12 months/yr = $51, 480/yr.

The energy cost savings of using the TES can be calculated using Eq. (12.3) with the assumption that there are 250 typical cooling days per year:

$$\Delta EC_{TES} = 1000kW * 5hrs/day * 250days/yr\left(\frac{\$0.07/kWh}{3.5} - \frac{\$0.02/kWh}{3.0}\right) = \$16{,}667\,/\,yr$$

The size of the ice storage tank is such that it can hold the cooling energy required during on-peak period; that is, 1,000 kW * 5 hr = 5,000 kWh. Therefore, the simple payback period for installing TES is estimated as follows:

$$SPB = \frac{\$100/kWh * 5000kWh}{(\$51480 + \$16667)} = 7.3years$$

A life-cycle cost analysis may be required to determine if the investment in a TES system is really warranted.

It should be noted that more savings can be obtained if the chiller has to be replaced. Indeed, when the TES is installed the chiller capacity can be reduced. For the case of this example, a chiller with a 500-kW cooling capacity is sufficient (instead of a 1,000-kW chiller) to charge the TES system and cool the office building during off-peak periods.

12.6.2 TES Control Improvement

In this section, typical results are reported on operating cost savings due to improvement of TES controls based on a detailed analyses (Krarti et al. 1999c). As discussed earlier, the performance of cooling plants with TES systems is affected by several factors. The effects of some of these factors on the cost savings of various TES control strategies are discussed below.

12.6.2.1 Effect of Plant Size

The size of the cooling plant equipment including the capacity of both the chiller and the storage tank can significantly affect the operating cost savings incurred by using one of the control strategies discussed in Section 12.3. Some of these effects were investigated systematically using different sizes of chillers and storage tanks for both an office building and a hotel (Krarti et al. 1999c). Similar findings were reported for both buildings during both peak and swing periods. In this section, only the results for the swing season cooling load profile for the hotel are presented.

The size of the chillers and the storage tank for various design cases (for the hotel) considered in the analysis are provided in Table 12.2. The performance of all the control strategies is also summarized in Table 12.2, using chiller-priority control as a base-case (i.e., all the cost savings are relative to the savings incurred for the same cooling plant design operated with chiller priority control.

In particular, the results from Table 12.2 indicate that:

TABLE 12.2 Effect of Cooling Plant Design on the Performance of TES Control Strategies for a Hotel. Savings Are Compared to a Plant with Chiller-Controlled TES

System		Base Case	Reduced Storage Size	Increased Chiller Size	Reduced Base-Load Chiller
Base-load chiller size	(tons)	550	550	620	350
TES chiller (chw) size	(tons)	160	160	270	160
TES chiller (ice) size	(tons)	120	120	190	120
Storage tank size	(ton-hr)	2,500	1,600	2,500	2,500
Savings					
Constant proportion	(%)	+0.4	+0.4	0.0	−1.5
Storage priority	(%)	−0.2	−0.3	−0.1	−6.7
Optimal control	(%)	−3.5	−3.4	−1.5	−11.8

TABLE 12.3 Effect of the Cooling Load Profile on Performance of the TES Control Strategies

Cooling Load	Office Peak		Office Swing		Hotel Peak		Hotel Swing	
Noncooling Loads	Without (%)	With (%)	Without (%)	With (%)	Without (%)	With (%)	Without (%)	With (%)
Constant Priority	−5.5	−1.4	−4.4	−0.6	+0.2	0.0	+0.1	0.0
Storage Priority	−6.7	−1.7	−10.6	−1.5	−1.0	−0.2	−1.0	−0.2
Optimal Control	−17.3	−4.4	−20.0	−2.8	−2.8	−0.5	−3.7	−0.7

Source: Krarti, Henze, and Bell, *ASHRAE Transactions*, 105(1), 1999c.

- The optimal controller out-performs all the conventional controls independently of the cooling plant design.
- All the control strategies provide a significant cost saving relative to the chiller priority, for the case of a reduced base-load chiller. However, the optimal controller produces a smaller saving when the sizes of both chillers are increased (i.e., in the case of increased chiller size).
- The optimal controller provides a significant cost saving (11.8 percent) when two favorable conditions are present:
    (i) The base-load chiller is small (or the cooling load to be shifted is significant relative to the storage size).
    (ii) The TES chiller size is small compared to the storage capacity. Table 12.1 shows that meeting condition (ii) by itself does not translate into increased cost savings (i.e., base case versus reduced storage).

12.6.2.2 Effect of the Cooling Load Profile

Table 12.3 summarizes the operating cost savings of the optimal controller and the conventional control strategies relative to the performance of the chiller-priority control during the peak and swing seasons for both an office and a hotel (Krarti, Henze, and Bell, 1999c). The results presented in Table 12.3 are based on the actual RTP rate presented in Figure 12.3.

Table 12.3 indicates that the optimal controller and the two conventional controls (i.e., constant-proportion and storage-priority) provide more operating cost savings for the office building rather than for the hotel. This result is due to the fact that the cooling load profiles for the office building present distinguishable periods of high and low cooling demands unlike those for the hotel.

Other interesting features can be noted from the results shown in Table 12.2. First, the savings percentages for all the control strategies are significantly reduced when the noncooling electrical loads are included in the total operating cost of the cooling plant. This reduction is due to the small contribution of the cooling load to the total building electrical load especially for the swing seasons. Second, the optimal controller outperforms the chiller-priority more significantly during the swing season rather than during the peak season when the noncooling electrical loads are not considered. This result is attributed to two factors: (i) the storage-priority control generally performs poorly during swing seasons as pointed out by Braun (1992) and Henze, Dodier, and Krarti (1997b) because the storage is used only slightly when the cooling loads are low, and (ii) during the swing season, the cooling load to be shifted is smaller, thus the economic benefit for using the storage is greater inasmuch as only a fraction of the cooling load during the peak season can be shifted from high to low rates. However, Table 12.3 shows that when the noncooling electrical loads are considered, the optimal controller seems to perform better during the peak season for the office building. The fact that the cooling load contribution for the office building is much lower during the swing season than that during the peak season explains the lower percentage for the cost savings during the swing season. Meanwhile for the hotel, the cooling contribution to the total electrical load remains almost the same for the two seasons.

12.7 Summary

Some of the benefits of TES systems are discussed in this chapter. In particular, it has been indicated that TES systems, if properly designed and controlled, can save significant operating costs for the cooling plants of commercial and institutional buildings. Operation of TES systems can be affected by several factors including primarily the electrical utility rates. Currently, TES systems in buildings are being considered as possible options for storing electrical energy for a distributed generation environment.

Problems

12.1 Consider a building with 250-tons peak cooling load. On the peak-cooling day, the cooling load occurs for 8 hours with a load factor of 75 percent. During the entire year, there are 1,000 equivalent full-load hours. Cooling is required only during the on-peak period of the utility rate structure.

The utility rate structure includes (i) demand charge: \$10/Kw with 12 months ratchet period, (ii) energy charge: the energy charge differential between on-peak and off-peak is \$0.05/kWh. The installed cost of a chiller is \$60/ton. The cost of a storage system is \$60/ton-hr. Consider that the efficiency of the chiller is on average 1 kW/ton.

(i) Determine the cost-effectiveness (using a simple payback analysis) of installing (a) a full storage system and (b) a partial storage system with 50 percent undersized chiller. For the partial storage system, assume that only 400 of equivalent full-load hours (relative to the new chiller size) have to be supplied by the chiller during the on-peak period.

(ii) Determine the effect of the number of hours that the chiller has to be used during the on-peak period on the cost-effectiveness of the partial storage system.

12.2 Consider a school building with 400-tons peak cooling load. On the peak-cooling day, the cooling load occurs for 8 hours with a load factor of 60 percent. During the entire year, there are 1,500 equivalent full-load hours. Cooling is required only during the on-peak period of the utility rate structure.

The utility rate structure includes (i) demand charge: \$12/kW with 12 months ratchet period, (ii) energy charge: the energy charge differential between on-peak and off-peak is \$0.06/kWh.

The installed cost of a chiller is \$500/ton. The cost of a storage system is \$50/ton-hr. Consider that the efficiency of the chiller to be 0.8 kW/ton.

(i) Determine the cost-effectiveness (using a simple payback analysis) of installing (a) a full storage system and (b) a partial storage system with 50 percent undersized chiller. For the partial storage system, assume that only 400 of equivalent full-load hours (relative to the new chiller size) have to be supplied by the chiller during on-peak period.

(ii) Determine the effect of the number of hours that the chiller has to be used during the on-peak period on the cost-effectiveness of the partial storage system.

12.3 List and comment on the advantages and disadvantages of both full and partial storage systems.

13
Cogeneration Systems

13.1 Introduction

Cogeneration and combined heat and power (CHP) are terms used interchangeably to denote the simultaneous generation of power (electricity) and usable thermal energy (heat) in a single integrated system. A CHP plant derives its efficiency, and hence lower costs, by recovering and utilizing the heat produced as a by-product of the electricity generation process that would normally be wasted in the environment. The overall fuel efficiency of typical CHP installations can be in the range of 70–90 percent, compared with 35–50 percent for conventional electricity generation. Overall, CHP achieves a 35 percent reduction in primary energy usage compared with remote power stations and heat-only boilers. Moreover, CHP also avoids transmission and distribution losses because it supplies electricity generally close to the generation site.

Although the CHP concept is not new, it has only recently been applied to a wide range of commercial buildings. Indeed, until the 1980s, cogeneration systems were used only in large industrial or institutional facilities with high electricity demand (typically over 1,000 kW). After the energy crisis of 1973 during which fuel and electricity prices increased significantly (by a factor of five), in 1978 the U.S. government passed the National Energy Act (NEA) which includes the Public Regulatory Policies Act (PURPA). The PURPA regulations have forced utilities to purchase electricity and to provide supplementary or back-up power to any qualified cogeneration facilities. The Energy Policy Act of 1992 has increased the appeal of cogeneration systems even more by opening up transmission line access and retail wheeling.

The term *wheeling* refers to the process by which utilities can buy or sell electricity to or from other utilities in order to meet peak demands or shed excess generation. The wheeling process enables utilities to spend less on peaking power plants, thereby lowering capital expenditures required to meet high periods of electric power demands.

In addition to the favorable regulations, the development of energy-efficient cogeneration systems and small pre-engineered packaged cogeneration units has provided the needed incentives to encourage the implementation of systems capable of generating electricity and heat for commercial, institutional, and even residential applications. Currently, cogeneration systems are available over a wide range of sizes from less than 50 kW (microsystems) to over 100 MW. Moreover, advances in controls have provided better procedures to operate and integrate the various components of a cogeneration system (including prime movers, electrical generators, and heat recovery systems). More recently, energy service companies (ESCOs) have increased awareness and interest of the public in on-site generation technologies.

In addition to their better overall energy efficiency compared to conventional utility plants, cogeneration systems offer the following benefits:

- Cleaner power generation sources with reduced No_x and carbon emissions
- Increased reliability and quality of electrical power because customers are less vulnerable to blackouts from utility power lines
- Added national energy security with diversified sources and locations of power generation
- Avoided transmission and distribution costs because no new lines would be needed if distributed cogeneration plants were available

To evaluate the feasibility of a cogeneration system, several technical and economical aspects as well as regulatory issues should be considered. This chapter provides some discussion of the main regulatory considerations and financial options for cogeneration in the United States, however, the main focus of the chapter is to provide technical information and engineering principles that are necessary to understand the design and operation of cogeneration systems. First, an overview of the past, current, and future status of the electricity generation industry in the United States is presented.

13.2 History of Cogeneration

Systems for combined heat and power generation have been in existence since the 1880s in the United States and Europe. Indeed, several industrial facilities generated their own electricity and steam using coal-fired boilers and steam-turbine generators. It is estimated that CHP systems produced up to 58 percent of the total electricity generated in the United States.

However, in the mid-1900s, large central electrical power plants were built with reliable utility grids. The electricity from these plants was produced at relatively low cost. As a consequence, industrial facilities started to purchase electricity from the power plants and thus gradually reduce their reliance on on-site generation. The contribution of CHP systems to total U.S. electric power generation represented 15 percent in 1950 and only 4 percent in 1974 (EIA, 2000). Large electric utility holding companies have been formed and expanded since the early 1900s. In 1920, the majority of the U.S. power industry was controlled by a few privately owned electric power holding companies. These holding companies abused their power and charged consumers higher prices for electricity.

To reduce the monopoly of the privately owned electric utility holding companies, the Federal government intervened by passing the Public Utility Holding Company Act (PUHCA) in 1935. Under the provisions of the PUHCA, the electric utility holding companies became regulated by the Securities and Exchange Commission. In a further effort to reduce electricity prices, government-owned hydroelectric power facilities were built including the Hoover Dam in 1936. The Bonneville Project Act of 1937 provided the federal government with means to oversee transmission and marketing of power produced from hydroelectric facilities. By 1941, power produced from publicly owned facilities represented 12 percent of the total utility generation.

Until the early 1970s, utilities were able to meet the ever-increasing electrical power demands at decreasing prices due to the economies of scale, technological advances, and declining fuel costs. However, a series of events that occurred during the 1970s have had a significant impact on the U.S. electric power industry. These events include the oil embargo in 1973–1974, and the passage of the Clean Air Act in 1970, as well as the Energy Supply and Environmental Coordination Act in 1974. As a result of these events, prices of electrical power increased dramatically in the 1980s. Between 1973 and 1985, the prices of fuels and electricity increased by a factor of 5.

As a reaction to the oil embargo, in 1978 the federal government also passed the National Energy Act (NEA) in order to reduce U.S. dependence on foreign oil, develop alternative energy sources, and encourage energy conservation. The NEA comprises five different statutes:

1. Public Utility Policy Act (PURPA)
2. Energy Tax Act

3. National Energy Conservation Policy Act
4. Powerplant and Industrial Fuel Use Act
5. Natural Gas Policy Act

PURPA has had the most significant impact on the electric power industry and the development of cogeneration. Indeed, PURPA allowed nonutility facilities that meet certain ownership and efficiency criteria to sell electric power to utility companies. Specifically, PURPA sets the following legal obligations for the electric utility companies toward cogenerators defined as qualified facilities (QFs):

- Utility companies have to purchase cogenerated energy and capacity from QFs.
- Utility companies have to sell energy and capacity to QFs.
- Utility companies have to provide to QFs supplementary power, backup power, maintenance power, and interruptible power.
- Utility companies have to provide access to transmission grid to wheel to other electric utility companies.

Additional legislation passed in the 1990s, including the Clean Air Act Amendments (CAAA) of 1990 and the Energy Policy Act (EPACT) of 1992, have provided new opportunities for cogeneration. In particular, EPACT provides incentives for nonutility generators to enter the wholesale market for electrical power by exempting them from the PUHCA constraints. The law creates a new category of power producers known as exempt wholesale generators (EWGs). It should be noted that EWGs are different from PURPA QFs because they are not required to meet PURPA's cogeneration criteria. In addition, utilities are not required to purchase electric power from EWGs.

Table 13.1 summarizes the major federal legislation that has marked the U.S. power industry since the early 1930s. Because of the passage of PURPA, multiple megawatt CHP projects have been developed and built especially at large industrial facilities including pulp and paper, steel, chemical, and refining plants. Recent advances in reciprocating engines and microcombustion turbines make CHP more cost-effective for small applications, such as fast-food restaurants, as well as commercial buildings.

It is estimated that in 2005, the CHP generating capacity is about 82 GW accounting for about 6.1 percent of the total electricity generation capacity in the United States (EIA, 2009). In several European countries, the share CHP in national generating capacity exceeds 25 percent such as the case of Denmark (53 percent) and Netherlands (37 percent) as summarized in Table 13.2.

13.3 Types of Cogeneration Systems

There are several types of cogeneration systems that are commercially available. In general, three categories of cogeneration systems can be considered:

1. *Conventional cogeneration systems:* These systems consist of large cogeneration units (more than 1,000 kW) and require a thorough design process to select the size of all equipment and components (i.e., prime movers, electrical generators, and heat recovery systems).
2. *Packaged cogeneration systems:* These systems are small (below 1,000 kW) and are easy to design and install inasmuch as they are pre-engineered and preassembled units.
3. *Distributed generation technologies:* Some cogeneration systems can use a number of technologies that have been recently developed to produce both electricity and heat including fuel cells.

13.3.1 Conventional Cogeneration Systems

A typical cogeneration plant consists of several pieces of equipment to produce electricity and heat (in the form of either steam or hot water). The number and type of equipment in a cogeneration plant

TABLE 13.1 Major Federal Legislation Affecting the U.S. Electric Power Industry

Legislation Name	Year	Main Scope
Public Utility Holding Company Act (PUHCA)	1935	To reduce utility industry abuses by giving the Securities and Exchange Commission the authority to oversee holding companies.
Bonneville Project Act	1937	To create the Bonneville Power Administration (BPA) to oversee the transmission and marketing of power produced in northwest dams.
Clean Air Act	1970	To reduce sulfur dioxide and nitrogen dioxide emissions.
Energy Supply and Environmental Coordination Act (ESECA)	1974	To allow the federal government to prohibit electric utilities from burning natural gas or petroleum products.
Public Utility Regulatory Policies Act (PURPA)	1978	To promote conservation of electric energy by allowing nonutility generators and qualified cogenerators to sell power to utilities.
Energy Tax Act (ETA)	1978	To allow tax credits for investment in cogeneration equipment and renewable technologies. These incentives were curtailed in the mid-1980s.
National Energy Conservation Policy Act	1978	To require utilities to develop residential energy conservation plans in order to reduce growth in electricity demand.
Powerplant and Industrial Fuel Use Act	1978	To replace ESECA of 1974 and extend federal government prohibition on the use of natural gas and petroleum in new power plants.
Pacific Northwest Electric Power Planning and Conservation (PNEPPC) Act	1980	To create the PNEPPC Council in order to coordinate the conservation and resource acquisition of the BPA.
Electric Consumers Protection Act (ECPA)	1986	To set new environmental criteria in licensing hydroelectric power plants; to reduce significantly PURPA benefits for new hydroelectric projects; and to increase the enforcing powers of FERC
Clean Air Act Amendments (CAAA)	1990	To establish a new emissions-reduction program; generators of electricity are made responsible for a large portion of the sulfur dioxide and nitrogen oxide reductions.
Energy Policy Act (EPACT)	1992	To create a new category of electricity producers and to authorize the FERC to open up the national electricity transmission system to wholesale suppliers.

Source: EIA, *The Changing Structure of the Electric Power Industry*, 2000.

TABLE 13.2 Share of CHP in Total National Power Generation for Selected Countries

Country	Denmark	Netherlands	Germany	Italy	United States	United Kingdom	France
Contribution of CHP (%)	53	37	17	8	6	8	4

Source: EIA, *Annual Energy Review,* 2009.

depend on the size of the system and the procedure used to generate electricity and heat. Generally, a conventional cogeneration system includes the following components:

1. *A prime mover:* This is the most important equipment in a cogeneration system. It is typically a turbine that generates mechanical power using a primary source of fuel. There are three turbine types that are commonly used in cogeneration plants: turbines operated by steam generated from boilers, gas turbines fueled by natural gas or light petroleum products, and internal combustion engines fueled by natural gas or distillate fuel oils.
2. *A generator:* This is a device that converts the mechanical power to electrical energy.

3. *A heat recovery system:* This comprises a set of heat exchangers that can recover heat from exhaust or engine cooling and convert it into a useful form, typically hot water.

To operate a cogeneration plant, a robust control system is needed to ensure that all the individual pieces of equipment provide the expected performance. Two basic operation cycles are used to generate electricity and heat: either a bottoming cycle or a topping cycle.

13.3.1.1 Bottoming Cycle

In this cycle, the generation of heat is given the priority to supply process heat to the facility. Thermal energy is produced directly from fuel combustion (in the prime mover). Heat is then recovered and fed to the generator to produce electricity as illustrated in Figure 13.1(a). Industrial plants characterized by high-temperature heat requirements (such as steel, aluminum, glass, and paper industries) typically use bottoming cycle cogeneration systems.

13.3.1.2 Topping Cycle

Unlike the bottoming cycle, the generation of electricity takes precedence over the production of heat as indicated in Figure 13.1(b). The waste heat is then recovered and converted to either steam or hot water. Most existing cogeneration systems are based on topping cycles. A hybrid of a topping cycle commonly

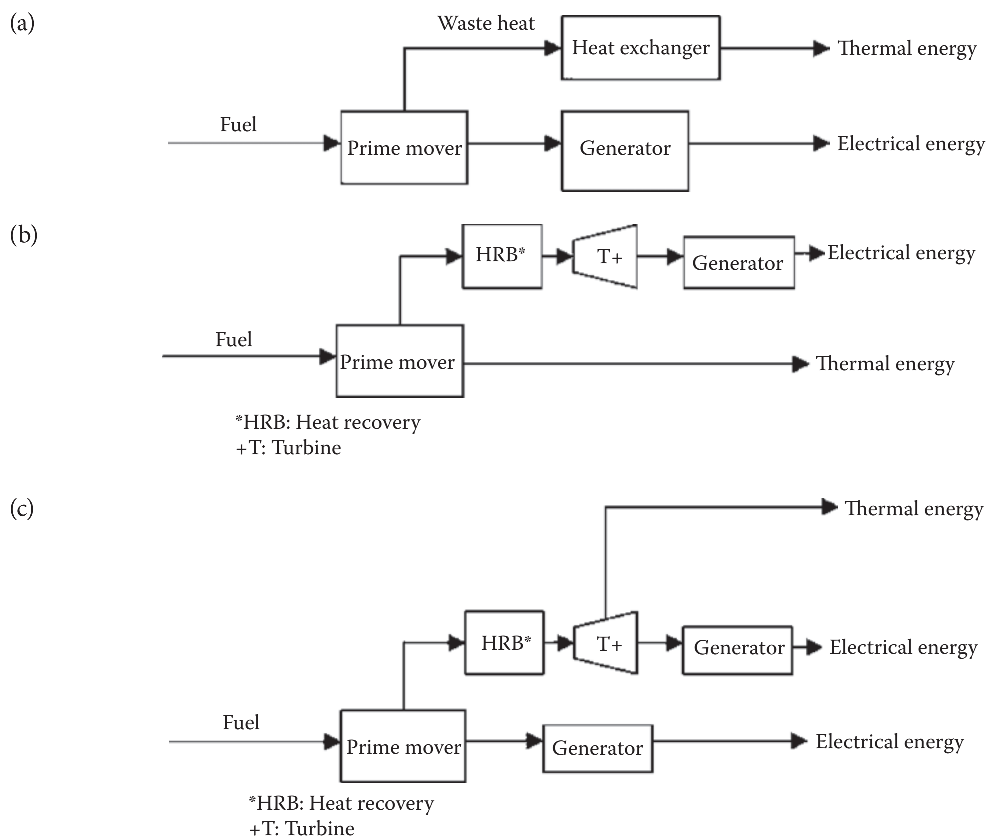


FIGURE 13.1 (a) Topping cycle cogeneration system; (b) bottoming cycle cogeneration system; (c) combined cycle cogeneration system.

used by several industrial facilities and even by electrical utilities is the combined cycle as depicted in Figure 13.1(c). In this cycle, a gas turbine is typically used to produce electricity. The exhaust gas is then fed to a heat recovery steam generator to generate more electricity using a steam turbine. For a cogeneration plant, a small portion of steam can be converted into a useful form of thermal energy.

Three types of prime movers are generally considered for large and medium conventional cogeneration systems: steam turbines, reciprocating engines, and gas turbines. A brief overview of each prime mover type is provided below.

13.3.1.2.1 Steam Turbines

Steam turbines are the oldest and most versatile prime movers used in power generation. In the United States, most of the electricity is generated from power plants using steam turbines. However, steam turbines are also utilized in combined heat and power systems, combined cycle power plants, and district heating systems. The capacity of steam turbines ranges from 50 kW to several hundred MWs. There are several types of steam turbines used today in power generation applications including:

- Condensing turbines are power-only utility turbines. They exhaust directly to condensers that maintain vacuum conditions at the discharge.
- Noncondensing turbines are also referred to as back-pressure turbines. They exhaust steam to the facility mains at conditions close to the process heat requirements.
- Extraction turbines have openings in their casing for extraction of a portion of the steam at some intermediate pressure before condensing the remaining of the steam.

The electrical generating efficiency of steam turbines varies from 37 percent for large electric utility plants to 10 percent for small plants that produce electricity as a by-product of steam generation. The common applications of steam turbines for combined heat and power systems involve industrial processes where solid or waste fuels are readily available. In the United States, it is estimated that over 580 industrial and institutional facilities use steam engines to produce about 19,000 MW of electric capacity. Table 13.3 lists typical cost and performance parameters for cogeneration systems using steam turbines.

13.3.1.2.2 Reciprocating Engines

Reciprocating internal combustion engines are a widely used technology to generate power for several applications including institutional and industrial, and combined heat and power systems. The capacity of reciprocating engines varies from a few kilowatts to over 5 MW. Two basic types of reciprocating engines are utilized for power generation applications:

1. Spark ignition (SI) engines generally use natural gas. SI engines are now commonly used for duty-cycle stationary power applications.
2. Compression ignition (CI) or diesel engines have been historically popular for both small and large power generation applications. However, diesel engines are now restricted to emergency and standby applications due to emission concerns.

TABLE 13.3 Cost and Performance Characteristics for Selected Cogeneration Systems Using Steam Turbines

Cost and Performance Parameters	System Capacity in kW		
	500	3,000	15,000
Total installed cost ($/kW)	$918	$385	$349
Fuel input (MMBtu/hr)	26.7	147.4	549.0
Electric efficiency (%)	6.4	6.9%	9.3%
Steam to process (MMBtu/hr)	19.6	107.0	386.6
Overall efficiency (%)	79.6	79.5	79.7

Source: Energy Nexus Group, *Technology Characterization: Steam Turbines*, 2002a.

Electric efficiency of reciprocating engines ranges from 28 percent for small engines (less than 100 kW) to 40 percent for large engines (above 3 MW). Waste heat can be recovered from four sources in reciprocating engines: exhaust gas, engine jacket cooling water, lube oil cooling water, and turbocharger cooling. High and low pressure steam can be produced from reciprocating engines for combined heat and power applications. Overall efficiency for a CHP system using a reciprocating engine fueled by natural gas can exceed 70 percent.

Reciprocating engine technology has improved significantly over the last decades with increased fuel efficiency reduced emissions, improved reliability, and low first cost. The use of reciprocating engines for combined heat and power generation applications is expected to continue to grow in the next decade. Currently, it is estimated that 1,055 CHP systems operating in the United States use reciprocating engines with an overall power capacity of 800 MW. Table 13.4 provides typical cost and performance parameters for commercially available reciprocating engines suitable for cogeneration applications.

13.3.1.2.3 Gas Turbines

Gas turbines provide one of the cleanest means for electric power generation with very low emissions of carbon dioxide (CO_2) and oxides of nitrogen (NO_x). The available capacity of gas turbines ranges from 500 kW to 250 MW. Gas turbines are well suited for CHP applications because high-pressure steam (as high as 1,200 psig) can be generated from their high-temperature exhaust using heat recovery steam generators (HRSGs). It is estimated that over 575 industrial and institutional facilities in the United States use gas turbines to generate power and heat with a total capacity of 40,000 MW (Energy Nexus Group, 2002c).

Table 13.5 lists performance characteristics of selected commercially available gas turbine CHP systems. As indicated in Table 13.5, both electrical efficiency and overall CHP efficiency increase with the size of the gas turbine.

TABLE 13.4 Cost and Performance Characteristics for Selected Cogeneration Systems Using Reciprocating Engines

Cost and Performance Parameters	System Capacity in kW				
	100	300	800	3,000	5,000
Total installed cost (2001 $/kW)	$1,515	$1,200	$1,000	$920	$920
Fuel input (MMBtu/hr)	1.11	3.29	8.20	28.48	43.79
Electrical efficiency (%)	30.6	31.1	33.3	36.0	39.0
Total heat recovered (MMBtu/hr)	0.57	1.51	3.50	11.12	15.28
Overall efficiency (%)	81.0	77.0	76.0	75.0	74.0

Source: Energy Nexus Group, *Technology Characterization: Reciprocating Engines*, 2002b.

TABLE 13.5 Cost and Performance Characteristics for Selected Cogeneration Systems Using Gas Turbines

Cost and Performance Parameters	System Capacity in kW				
	1,000	5,000	10,000	25,000	40,000
Total installed cost (2000 $/kW)	$1,780	$1,010	$970	$860	$785
Fuel input (MMBtu/hr)	15.6	62.9	117.7	248.6	368.8
Electrical efficiency (%)	21.9	27.1	29.0	34.3	37.0
Steam output (MMBtu/hr)	7.1	26.6	49.6	89.8	128.5
Overall efficiency (%)	68.0	69.0	71.0	73.0	74.0

Source: Energy Nexus Group, *Technology Characterization: Gas Turbines*, 2002c.

13.3.2 Packaged Cogeneration Systems

For cogeneration facilities requiring small systems ranging from less than 50 kW to about 1 MW, pre-engineered and factory-assembled cogeneration units are currently available with reduced construction, installation, and operation costs. In addition, small packaged systems with capacities ranging from 4 to 25 kW have been developed and can be installed in a short period of time with little interruption of service. Almost all packaged cogeneration systems are equipped with advanced controls to improve the reliability and energy efficiency of the units.

Packaged cogeneration units are often sold as turn-key installations. In particular, manufacturers are generally responsible for the testing and installation of the complete cogeneration system. The development of packaged systems has enlarged the appeal of cogeneration to a wide range of facility types including office buildings, restaurants, homes, and multifamily complexes. In addition, packaged cogeneration systems are now typically more cost-effective than conventional systems for small and medium hospitals, schools, and hotels. However, each facility has to be thoroughly evaluated to determine the economical feasibility of any packaged cogeneration system.

Packaged cogeneration systems typically use reciprocating engines. Recently, microturbines have been developed and have been commercially available since 2000. Microturbines are small electricity generators that operate at very high speeds (over 60,000 rpm). They are available in sizes ranging from 30 kW to 350 kW. They have been promoted as ideal generators for distributed generation applications including cogeneration systems due to their connection flexibility (they can be stacked in parallel), their reliability, and their low emissions. Table 13.6 summarizes the performance characteristics of selected commercially available microturbines suitable for CHP applications.

13.3.3 Distributed Generation Technologies

Distributed generation is a relatively recent approach proposed to produce electricity using small modular generators. The small generators with capacities in the range of 1 kW to 10 MW can be assembled and relocated in strategic locations (typically near customer sites) to improve power quality, reliability, and flexibility in order to meet a wide range of customer and distribution system needs. Some technologies have emerged in the last decade that allow the generation of electricity with reduced waste, cost, and environmental impact which may make the future of distribution generation promising especially in a competitive deregulated market. Among these technologies are renewable energy sources (wind and solar), fuel cells, as well as microturbines, combustion turbines, gas engines, and diesel engines. Table 13.7 summarizes some distribution generation technologies with their average efficiency, capacity, and applications. Perhaps the recent developments in fuel cells represent the best opportunity for distributed generation and CHP applications. It is expected that fuel cells will play a significant part in the twenty-first-century electricity market.

TABLE 13.6 Cost and Performance Characteristics for Selected Cogeneration Systems Using Microturbines

Cost and Performance Parameters	System Capacity in kW			
	30	70	100	350
Total installed cost (2000 $/kW)	$2,516	$2,031	$1,561	$1,339
Fuel input (MMBtu/hr)	0.437	0.948	1.264	4.118
Electrical efficiency (%)	23.4	25.2	27.0	29.0
Heat output (MMBtu/hr)	0.218	0.369	0.555	1.987
Overall efficiency (%)	73.0	64.0	71.0	77.0

Source: Energy Nexus Group, *Technology Characterization: Micro-Turbines*, 2002d.

The principle of the fuel cell was first demonstrated over 150 years ago. In its simplest form, the fuel cell is constructed similar to a battery with two electrodes in an electrolyte medium, which serves to carry electrons released at one electrode (anode) to the other electrode (cathode). Typical fuel cells use hydrogen (derived from hydrocarbons) and oxygen (from air) to produce electrical power with other by-products (such as water, carbon dioxide, and heat). High efficiencies (up to 73 percent) can be achieved using fuel cells.

Table 13.8 summarizes various types of fuel cells that are commercially available or under development. Each fuel cell type is characterized by its electrolyte, fuel (source of hydrogen), oxidant (source of oxygen), and operating temperature range. CHP systems using fuel cells are available in sizes ranging from few kW to thousands of kW. Table 13.9 lists some performance characteristics of selected commercially available CHP systems using fuel cell technology.

TABLE 13.7 Characteristics of Current Distributed Generation Technologies

Type	Efficiency (%)	Size	Applications
Combustion turbine	24–40	500 kW–30 MW	Cogeneration (commercial/industrial), transmission and distribution support
Diesel engine	36–42	50 kW–6 MW	Standby, remote, and peak shaving power
Gas engine	28–38	5 kW–2 MW	Cogeneration (commercial/industrial), peak shaving and primary power
Microturbine	21–40	25 kW–300 kW	Cogeneration (commercial/light industrial), primary power
Fuel cell	40–65	1 kW–3 MW	Cogeneration (residential/commercial), primary power

TABLE 13.8 Basic Characteristics of Selected Types of Fuel Cells

Fuel Cell Name	Electrolyte	Fuel	Oxidant	Operating Temperatures (°C)
PAFC	Phosphoric acid	Pure hydrogen	Clear air (without CO_2)	200
AFC	Alkaline	Pure hydrogen	Pure oxygen and water	60–120
SPFC	Solid polymer	Pure hydrogen	Pure oxygen	60–100
MCFC	Molten carbonate	Hydrocarbons	Air and oxygen	650
SOFC	Solid oxide	Any fuel	Air	900–1,000

TABLE 13.9 Cost and Performance Characteristics of Selected CHP Using Fuel Cells

Cost and Performance Parameters	System Capacity in kW			
	10	100	200	2,000
Fuel cell type	PEM	SOFC	PAFC	MCFC
Total installed cost (2002 $/kW)	$5,500	$3,500	$4,500	$2,800
Fuel input (MMBtu/hr)	0.10	0.80	1.90	14.80
Electrical efficiency (%)	30.0	45.0	36.0	46.0
Heat output (MMBtu/hr)	0.04	0.19	0.74	3.56
Overall efficiency (%)	68.0	70.0	75.0	70.0

Source: Energy Nexus Group, *Technology Characterization: Fuel Cells,* 2002e.

13.4 Evaluation of Cogeneration Systems

To evaluate the technical and economical feasibility of a cogeneration system, it is important to collect accurate data about the facility and its energy consumption. In particular, current and projected future energy consumption and costs need to be available. For a detailed evaluation analysis, hourly electrical and thermal energy data are required. However, monthly and even yearly energy data can be sufficient for a preliminary feasibility analysis of cogeneration systems. In this section, basic considerations are discussed to evaluate the feasibility of cogeneration systems.

13.4.1 Efficiency of Cogeneration Systems

To account for the fact that a typical cogeneration system produces both electrical power E_e and thermal energy E_t from fuel energy FU, as illustrated in Figure 13.2, the overall thermodynamic efficiency $\eta_{overall}$ of the cogeneration system is defined as follows:

$$\eta_{overall} = \frac{E_e + E_t}{FU} \tag{13.1}$$

It should be noted that for a cogeneration facility to meet the criteria specified by the PURPA, it has to comply with certain efficiency standards. These standards use a "PURPA efficiency" or η_{PURPA} factor, which is defined by Eq. (13.2):

$$\eta_{PURPA} = \frac{E_e + E_t/2}{FU} \tag{13.2}$$

PURPA states that to be a qualified facility for cogeneration, the efficiency η_{PURPA} has to be at least:

(a) 45 percent for the cogeneration facilities with a useful thermal energy fraction that is larger than 5 percent
(b) 42.5 percent for the cogeneration facilities with a useful thermal fraction that is larger than 15 percent

The main purpose of the PURPA efficiency is to ensure that a sufficient amount of thermal energy is produced so that the cogeneration facility is more efficient than the electric utility. A number of regulations within PURPA provide incentives to develop cogeneration facilities. Among these incentives are the legal obligations of the electric utilities toward the cogeneration facilities set by Section 210 of PURPA (FERC, 1978). For instance, the electric utility has to:

- Purchase cogenerated electrical energy from the qualified facilities.
- Sell electrical energy to the QFs.
- Provide access for the QFs to transmission grid.

Example 13.1 illustrates how to estimate overall thermodynamic and PURPA efficiencies for conventional cogeneration systems.

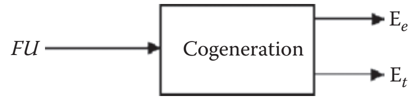


FIGURE 13.2 Energy input–output for a typical cogeneration system.

Example 13.1

Consider a 20-MW cogeneration power plant in a campus complex. An energy balance analysis indicates the following energy fluxes for the power plant:

- Electricity generation 33 percent
- Condenser losses 30 percent
- Stack losses 30 percent
- Radiation losses 7 percent

It is estimated that all the condenser losses but only 12 percent of the stack losses can be recovered. Determine both the overall thermodynamic efficiency as well as the PURPA efficiency of the power plant.

Solution

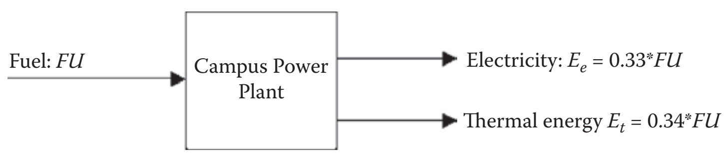
First, the recovered thermal energy is determined:

Et = Condenser Losses + Part of the Stack Losses

This thermal energy output can be expressed in terms of the fuel use (*FU*) of the power plant:

$$Et = 30\% * FU + 12\% * [30\% * FU] = 0.34 * FU$$

The energy flow for the campus power plant is summarized in the diagram below:



Thus, the overall thermodynamic efficiency of the power plant can be easily determined using Eq (13.1):

$$\eta_{overall} = \frac{E_e + E_t}{FU} = \frac{0.33 * FU + 0.34 * FU}{FU} = 0.67$$

The PURPA efficiency can be calculated using Eq. (13.2),

$$\eta_{PURPA} = \frac{E_e + E_t/2}{FU} = \frac{0.33 * FU + 0.34 * FU/2}{FU} = 0.50$$

Therefore, the campus power plant meets the PURPA criteria (η_{PURPA} > 45 percent) and is thus a qualified cogeneration facility.

13.4.2 Simplified Feasibility Analysis of Cogeneration Systems

To determine if a cogeneration system is cost-effective, simplified analysis procedures can be used first. A further evaluation with more detailed energy analysis tools may be warranted to determine the optimal design specifications of the cogeneration system.

Example 13.2 illustrates one simplified calculation procedure that can be used to determine the cost-effectiveness of installing a cogeneration system for a hospital building.

Example 13.2

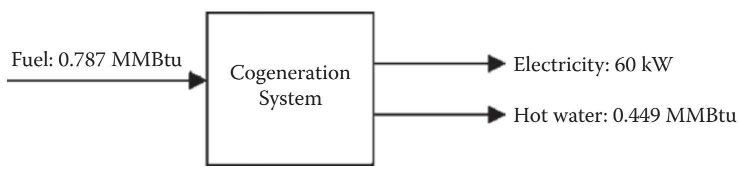
Consider a 60-kW cogeneration system that produces electricity and hot water with the following efficiencies: (a) 26 percent for the electricity generation, and (b) 83 percent for the combined heat and electricity generation. Determine the annual savings of operating the cogeneration system compared to a conventional system that consists of purchasing electricity at a rate of $0.08/kWh and producing heat from a boiler with 70 percent efficiency. The cost of fuel is $5/MMBtu. The maintenance cost of the cogeneration system is estimated at $1.00 per hour of operation. Assume that all the generated thermal energy and electricity are utilized during 6,500 hrs/year.

Determine the payback period of the cogeneration system if the installation cost is $2,500/kW.

Solution

First, the cost of operating the cogeneration system is compared to that of the conventional system on an hourly basis:

(a) *Cogeneration System:* For each hour, 60-kW of electricity is generated (at an efficiency of 26 percent) with fuel requirements of 0.787 MMBtu [(= 60 kW * 0.003413 MMBtu/kW/0.26). At the same time, a thermal energy of 0.449 MMBtu (= 0.787 MMBtu * 0.83 – 0.26) is obtained. The hourly flow of energy for the cogeneration system is summarized in the diagram below:



Thus, the cost of operating the cogeneration on an hourly basis can be estimated as follows:

Fuel Cost:	0.787 MMBtu/hr * $ 5/MMBtu =	$3.93/hr
Maintenance Cost:		$1.00/hr
Total Cost:		$4.93/hr

(b) *Conventional System:* For this system, the 60 kW electricity is directly purchased from the utility, and the 0.449 MMBtu of hot water is generated using a boiler with an efficiency of 0.65. Thus the costs associated with utilizing a conventional system are as follows:

Electricity Cost:	60 kWh/hr * 0.08 =	$4.80/hr
Fuel Cost (Boiler):	(0.449 MMBtu/hr)/0.65 * $ 5/MMBt	$3.45/hr
Total Cost:		$8.25/hr

Therefore, the annual savings associated with using the cogeneration system are:

$$\Delta Cost = (\$8.25/hr - \$4.93/hr) * 6500 hr/yr = \$21{,}580/yr$$

Thus, the simple payback period for the cogeneration system:

$$SPB = \frac{\$2500/kW * 60kW}{\$21{,}580} = 7.0\, years$$

A life-cycle cost analysis may be required to determine if the investment on the cogeneration system is really warranted.

For a more detailed evaluation, it is important to determine the main goal of the cogeneration system. Ideally, the cogeneration system can be sized to match exactly both the electrical and thermal loads. Unfortunately, there is almost never an exact match. Therefore, the cogeneration system has to be designed to meet specific load requirements such as the base-load thermal demand, base-load electrical demand, peak thermal demand, or peak electrical demand. The main features of each design scenario are briefly described below.

Base-load cogeneration systems: produce only a portion of the facility's electrical and thermal requirements. Thus, production of supplemental thermal energy (using a boiler, for instance) and the purchase of additional electrical energy are generally required. Base-load cogeneration systems are suitable for facilities characterized by variable thermal and electrical loads but not willing or able to sell electrical power.

Thermal-tracking cogeneration systems: are those systems that produce all the thermal energy required by a facility. In the case where the generated electrical energy exceeds the electrical demand, the facility has to sell power to the utility. In the case where the generated electrical energy is lower than the electrical demand, additional power has to be purchased from the utility. Thermal-tracking cogeneration systems are becoming increasingly attractive to small buildings that have to pay higher utility rates than large industrial and commercial facilities.

Electricity-tracking cogeneration systems: are designed to match electrical loads. Any supplemental energy requirements are produced through boilers. These systems are typically suitable for large industrial facilities with fairly high and constant electrical loads and lower but variable thermal loads.

Peak-shaving cogeneration systems: In the case where the cost associated with peak electrical demand is high, it may be cost-effective to design cogeneration systems specifically for peak shaving even though these systems may operate only a few hours (less than 1,000 hours per year).

Example 13.3 illustrates a monthly analysis to determine the optimum size for a base-load cogeneration system that is designed without selling any generated electrical power sales. The following equations have been used to carry out the analysis summarized in Table 13.4.

The monthly electrical energy kWh_{cogen} and the monthly thermal energy TE_{cogen}, produced by a cogeneration system of capacity kW_{cogen}, are estimated using Eqs. (13.3) and (13.4), respectively:

$$kWh_{cogen} = Min\{24.N_d.kW_{cogen};kWh_{actual}\} \tag{13.3}$$

$$TE_{cogen} = Min\{24.N_d.kW_{cogen};TE_{actual}\} \tag{13.4}$$

where
N_d = the number of days in the month.
kW_{cogen}= the electrical power capacity of the cogeneration system.
kWh_{actual} = the actual electrical energy used by the facility during the month.
TE_{actual} = the actual thermal energy used by the facility during the month.

Example 13.3

Provide a simple payback period analysis for implementing a cogeneration system to be installed in a hospital. Use the following characteristics for the cogeneration system:

- Fuel input rate: 10,000 Btu/kWh
- Heat recovery rate: 5,500 Btu/kWh
- Maintenance cost: $0.02/ kWh
- Maximum electrical output: 200 kW or 300 kW
- Installed equipment cost: $1000/kW

Table 13.10 summarizes the energy usage and cost of the hospital. Assume that the boiler(s) efficiency is 70 percent. For this analysis, assume also that the cogeneration system requires diesel fuel only (the other option is dual fuel). Assume the heating value of diesel fuel is 140,000 Btu/ gal.

TABLE 13.10 Monthly Utility Data for the Hospital Used in Example 13.3

	Utility Summary				
	Electricity			Fuel Oil	
Month	(kWh)	(kW)	($)	(Gallon)	($)
January	226,400	546	2,7020	20,659	14,911
February	273,600	572	28,949	20,555	12,639
March	280,800	564	31,048	16,713	9,670
April	228,000	526	25,251	10,235	4,742
May	246,000	692	28,755	12,193	5,347
June	301,200	884	36,604	12,352	9,001
July	346,800	1,040	45,031	20,604	3,122
August	403,200	944	46,374	17,276	5,711
September	303,600	860	36,541	10,457	3,762
October	276,000	872	33,559	10,890	3,726
November	272,400	662	28,042	13,478	5,255
December	276,000	524	25,041	17,661	7,808
Total	3,434,000		393,215	183,073	85,694

Solution

For each month, the energy cost incurred with a cogeneration system is calculated using a step-by-step procedure based on Eqs. (13.3) and (13.4). Table 13.11 summarizes the results of the step-by-step analysis performed for the month of January. Table 13.12 provides the results for all the months with the payback period for each cogeneration size. In this example, the smaller cogeneration system (200 kW) is more cost-effective because there is no option to sell excess generated power to the utility. However, a detailed economic analysis should be carried out to optimize the size of the cogeneration system.

TABLE 13.11 Details of Step-by-Step Analysis Performed for the Month of January in Example 13.3

Energy/Cost Requirements	Cogeneration 200 kW	System 300 kW
Electrical energy requirements (kWh)	226,400	226,400
Thermal energy requirements (MMBtu)	2,024	2,024
Cogenerated electrical energy, kWh_{cogen} (kWh)	148,808	223,400
Cogenerated thermal energy, TE_{cogen} (MMBtu)	808	1,228
Electrical energy to be purchased from utility (kWh)	77,600	3,000
Thermal energy to be directly generated (MMBtu)	1,206	796
Fuel use for cogeneration (gal)	10,628	15,957
Fuel energy for direct generation of thermal energy (gal)	8,614	5,686
Total fuel use requirements (gal)	19,242	21,643
Cost of utility electrical energy ($)	9,258	557
Cost of fuel ($)	13,893	15,627
Cost of maintenance ($)	2,976	4,468
Total cost ($)	26,127	20,450

TABLE 13.12 Cogeneration Cost Savings and Payback Periods

	Output(kW) Conventional E cost ($/month)	200 Cogen E Cost ($/month)	300 Cogen Cost, Cogen ($/month)
	41,931	26,127	20,450
	41,588	28,920	27,687
	40,718	27,108	21,753
	29,993	14,527	14,813
	34,102	20,178	14,121
	45,605	31,658	26,034
	49,153	32,167	24,142
	52,085	37,815	31,536
	40,303	26,388	20,414
	37,285	22,687	16,339
	33,297	21,581	16,512
	32,849	22,097	17,892
Total ($)	478,909	311,251	251,692
Equipment cost ($)	0	200,000	300,000
Payback period (yr)	—	1.19	1.32

13.4.3 Financial Options

To finance a cogeneration system, several financial options are generally available. Selecting the most favorable financial arrangement is critical to the success of a cogeneration project. A number of factors affect the selection of the best financial arrangement for a given cogeneration project. These factors include ownership arrangements, risk tolerance, tax laws, credit markets, and cogeneration regulations. In the United States, the most common financial approaches for cogeneration facilities are the following:

(i) *Conventional ownership and operation:* In this financing structure, the owner of the cogeneration facility funds the project either totally or partially from internal sources. In the case of partial funding, the owner can borrow the remaining funds from a conventional lending institution. Operation and maintenance of the cogeneration system can be performed by an external contractor.

(ii) *Joint-venture partnership:* This structure is an alternative to the conventional ownership and operation with shared financing and ownership with a second partner such as an electric utility. Indeed, PURPA regulations provide the option for an electric utility to own up to 50 percent of a cogeneration facility. The joint-venture financing structure reduces the risks for both partners but may increase the complexity of the various contracts among all the involved parties: including the owner and its partner, gas provider, electric utility, lending institution, and possible operation and maintenance contractor.

(iii) *Leasing:* In this financing option, a company builds the cogeneration facility with a leasing agreement from the owner to use part or all the thermal and electrical energy output of the cogeneration plant. The construction of the cogeneration system by the lessor (i.e., the builder of the facility) can be financed through funds from lenders or investors. The owner is generally heavily involved in the construction phase of the cogeneration facility.

(iv) *Third-party ownership:* This financing structure is similar to that described for the leasing case. However, in third-party ownership, the owner is not involved in both the financing and construction of the cogeneration facility. Instead, a third party or a lessee develops the project, and arranges for gas/fuel supply, electrical power and heat sales, and operation and maintenance agreements. The finances can be arranged by a lessor through funds from investors or lenders.

(v) *Guaranteed savings contracts:* In this financing option, a developer first builds and maintains the cogeneration facility. Then, the developer enters in a guaranteed savings contract with the energy consumer (the owner). This contract is typically made for a period ranging from five to ten years with a guaranteed fixed savings per year. This type of financial structure is common for small cogeneration systems (i.e., packaged units) inasmuch as it shifts all the financing and operation risks from the owner to the facility developer (i.e., the guaranteed savings contractor).

13.5 Case Study

A power plant, serving all the buildings of a university campus, utilizes natural gas with fuel oil No. 6 as a backup fuel. Due to increasing demands in electricity and steam use, the university retrofitted the power plant into a cogeneration plant. The specifications of the components for the cogeneration systems are summarized in Table 13.13.

The primary systems of the cogeneration plant include two combustion gas turbines, two heat recovery steam generators, high-pressure steam boilers, and low-pressure absorption chillers. The two gas turbines are each capable of producing 16 megawatts of power while providing sufficient hot exhaust gas to generate 80,000 pounds per hour of 300 psig steam in each heat recovery steam generator (HRSG). The steam turbine generator consists of dual topping turbines driving a common generator. The turbine reduces the incoming steam pressure by expanding it through the turbines which in turn, drive the

TABLE 13.13 Equipment Used for the Cogeneration Plant

Utility	Qty.	Equipment	Description
Electric power	2	Mitsubishi Industrial gas turbine sets	16 MW each each includes a dual fuel Mitsubishi Heavy Industries MF-111AB gas turbine driving a RENK single reduction gearbox coupled to a Brush two pole synchronous generator. Operates with a shaft speed of 9,645 rpm
Electric power	1	Dresser Rand steam turbine set	1 MW induction generator double-ended low and high pressure steam
Steam	2	Zurn (HRSG) heat recovery steam generators	Supplemental fire capability; Davis duct burners 80,000 lb/hr maximum steam output 300 psig
Steam	1	Erie City 1966 boiler	Front fired 150,000 lb/hr maximum steam output 130 psig
Steam	1	Combustion engineering 1957 boiler	Tangentially fired 115,000 lb/hr maximum steam output 130 psig
Chilled water	3	Steam absorption chillers	One – Trane – 1470 tons cooling; two – York – 900 tons cooling; each utilizes 10 psig steam lithium bromide

TABLE 13.14 Monthly Fuel Used and Electricity/Steam Produced by University Cogeneration

	Cogeneration System Performance				
		Electricity			
	Nat. Gas	Used	Sold	Total	130 psig Steam
Month	MMBtu	kWh	kWh	kWh	1,000 lbs
Jan	195,300	9,803,800	5,952,000	15,755,800	84,932
Feb	161,000	9,332,820	5,376,000	14,708,820	79,452
Mar	141,600	10,011,670	5,952,000	15,963,670	84,932
Apr	141,600	10,055,890	5,760,000	15,815,890	82,192
May	150,350	9,776,410	5,952,000	15,728,410	84,932
Jun	141,000	9,899,130	5,760,000	15,659,130	82,192
Jul	181,757	10,752,740	5,95,2000	16,704,740	84,932
Aug	201,802	10,604,980	5,952,000	16,556,980	84,932
Sep	180,480	10,492,980	5,760,000	16,252,980	82,192
Oct	150,474	10,389,270	5,952,000	16,341,270	84,932
Nov	166,230	10,553,990	5,760,000	16,313,990	82,192
Dec	154,039	9,441,120	5,952,000	15,393,120	84,932
Year	1,965,632	121,114,800	7,080,000	191,194,800	1,000,000

generator. The exhaust steam is then exported to either the 130 psig or the 10 psig steam header. The 130 psig steam is delivered to various buildings on the campus for heating purposes. The 10 psig steam is utilized to operate three absorption chillers that deliver chiller water to the campus. The high-pressure steam boilers deliver 300 psig steam to the gas turbines for NO_x control and steam injection for power augmentation.

The cogeneration plant was designed to deliver over 30 MW of electrical power. Absorption chillers with over 3000-ton capacity were added to increase the thermal load on the plant especially during the summer period. A large portion of the cogenerated electricity is sold to the local utility. As stipulated by PURPA, the plant can purchase electricity from the same utility in the case of maintenance or emergency periods. A typical monthly performance of the cogeneration plant is outlined in Table 13.14.

As indicated in Table 13.14 the university campus used about 63 percent of the electricity produced by the cogeneration plant and the remaining 37 percent was sold to a local utility. Due to the ever-increasing

TABLE 13.15 Comparison of Annual Energy Use and Carbon Emissions in 1990 and 2000

	Year 1990		Year 2000	
	Energy Purchased or Sold	Carbon Emissions (Tons)	Energy Purchased or Sold	Carbon Emissions (Tons)
Natural gas purchases				
Central plant/cogeneration	634,159 MMBtu	10,115 tons	1,936,341 MMBtu	30,885 tons
Individual buildings	116,500 MMBtu	1,849 tons	632,419 MMBtu	10,080 tons
Electricity purchases				
Central plant/cogeneration	66,024,000 kWh	18,476 tons	39,937,364 kWh	10, 637 tons
Individual buildings	20,782,510 kWh	5,816 tons	9,970,008 kWh	2,656 tons
Electricity sales				
Cogeneration plant	0 kWh	—	74,893,631 kWk	(19,947 tons)
Total natural gas purchased	750,659 MMBtu	11,994 tons	2,568,760 MMBtu	40,965 tons
Net electricity purchased/sold	86,806,510	24,292 tons	(24,986,259 kWh)	(6,655tons)
Net carbon emissions	—	36,296 tons	—	34,310 tons

electricity demands at the university (due to new buildings and additional research laboratories), the cogeneration plant may not be able to meet the university load in the near future. The university is currently investigating plans to add a new cogeneration plant.

From the data provided in Table 13.15, the electrical and overall efficiency of the cogeneration plant during 2002 was estimated at about 33 and 80 percent, respectively. The PURPA efficiency [refer to Eq. (13.2)] of the plant is over 56 percent, well above the requirements set by PURPA.

Over the last two decades, electricity and steam consumption have increased substantially at the university with a growth rate of about 5 percent. Table 13.15 illustrates the progression of energy use and carbon emissions at the university between 1990 (before the construction of the cogeneration plant) and 2000 (after the construction of the cogeneration plant). As indicated in Table 13.15, the cogeneration plant has substantially decreased the carbon output per unit of energy consumed. Even though natural gas use has tripled between 1990 and 2000, the net carbon emissions were actually reduced by 5 percent.

13.6 Summary

In this chapter, existing types and designs for cogeneration systems are briefly discussed. Moreover, simplified feasibility analyses of cogeneration systems are described with some illustrative examples. In the future, cogeneration is expected to become more attractive for small buildings especially with new developments in fuel cell technologies and microprocessor-based control systems. These developments will make small cogeneration systems cost-effective, reliable, and efficient even for nontraditional cogeneration applications such as residential complexes and office buildings.

Problems

13.1 Provide a simple payback analysis for implementing a cogeneration system in a hospital. Assume the following characteristics for the cogeneration system:

- Fuel input rate: 8,000 Btu/kWh
- Heat recovery rate: 4,800 Btu/kWh
- Maintenance cost: $0.02/ kWh

- Maximum electrical output: 200 kW, 300kW, 400 kW, 500 kW, 600 kW, 700 kW, or 800 kW
- Installed equipment cost: $700/kW, $1,000/kW, or $1,500/kW

Table 13.10 summarizes the energy usage and cost of the hospital. Assume that the boiler(s) efficiency is 75 percent. The heating value of diesel fuel is 140, 000 Btu/gal.

Present the results in one graph: the payback period versus the equipment cost for various equipment sizes.

13.2 Same as Problem 13.1. Assuming that the electric cost varies from $0.05/kWh to $0.15/kWh (including demand charge), determine the variation of the cogeneration system payback period versus electricity cost. For this question, assume that the capacity and the cost of the cogeneration system are 300 kW and $2,000/kW, respectively.

13.3 Same as Problem 13.1. If the life of the cogeneration system is 40 years, provide the optimal size of the cogeneration system for the hospital when the average interest rate is 8 percent and the inflation rate is 4 percent.

13.4 A 400-kW cogeneration system has an overall efficiency of 86 percent. The cogeneration system produces hot water at a thermal efficiency of 24 percent. Determine the cost-effectiveness of installing the cogeneration system in a facility at a cost of $60,000. The cost of electricity, if purchased directly from the utility, is $0.09/kWh and the cost of natural gas, used to operate the cogeneration system, is $1.50/therm. Without the cogeneration system, a gas-fired boiler with an efficiency of 85 percent is utilized to produce hot water needed by the facility. Assume that:

(a) The cogeneration system, as well as the facility, is operated 6,500 hours/year.

(b) The maintenance cost of the cogeneration system is $0.015/kWh.

Perform a life-cycle analysis using a life cycle of 20 years and a discount rate of percent.

14

Heat Recovery Systems

14.1 Introduction

Several processes inherent to the operation of the heating, ventilating, and air-conditioning (HVAC) systems result in heat rejection to the outside. All or part of this heat can be recovered and used to perform other useful functions. Improvements in air-to-air heat exchangers have made the recovery of waste heat cost-effective for some building systems. In industrial applications, the concept of heat recovery is relatively old and common because heat is a by-product of several manufacturing processes. For instance, waste incinerators are commonly used in industrial facilities to generate steam or hot water.

Both sensible and latent heat can be recovered from various HVAC systems such as exhaust air ducts, chillers, heat pumps, and cogeneration systems. The recovery of sensible heat generally results in temperature increase of a fluid (such as outdoor intake air). Meanwhile, the latent heat mainly affects the humidity level of air streams. In some cases, the addition of latent heat can also change the air temperature when a phase change occurs. Specifically, when the humid air condenses due to contact with a cold surface, air temperature increases. However, when the humid air is evaporated, the air temperature is decreased. Several heat recovery devices allow sensible heat recovery including air-to-air plate heat exchangers, heat pipes, and glycol heat reclaiming systems. Latent heat is recovered using desiccant systems.

In this chapter, a brief description of common heat recovery systems used in buildings is provided. In particular, the thermal efficiencies and the applications of various heat reclaiming devices are presented. Moreover, simplified calculation procedures are outlined to estimate the potential energy use and cost savings due to addition or improvement of heat recovery devices.

14.2 Types of Heat Recovery Systems

Waste heat recovery can be achieved by heat exchangers and can take several forms and shapes depending on the systems involved in the exchange of thermal energy. In particular, heat exchangers can be grouped into three categories depending on the temperature involved:

1. Low-temperature heat exchangers with fluid temperatures less than 230°C (450°F). Applications of low-temperature heat exchangers are common in buildings such as preheating of ventilation air with exhaust air.
2. Medium-temperature heat exchangers with fluid temperatures ranging from 230°C (450°F) and 650°C (1,200°F). Examples of medium-temperature heat recovery systems include incinerators.
3. High-temperature heat exchangers with fluid temperatures above 650°C (1,200°F). Generally, the use of high-temperature heat exchangers is specific to industrial processes such as in steel/aluminum furnaces.

Moreover, the fluids involved in the exchange of heat can determine the type of waste heat recovery system that is the most suitable for a given application. Three common types of heat exchangers are considered:

1. Gas-to-gas waste heat recovery systems that include heat pipes, rotary thermal wheels, liquid coupled heat exchangers, and plate-fin heat exchangers.
2. Gas-to-liquid waste heat recovery systems such as fire-tube or water-tube boilers, heat pipes, and economizers.
3. Liquid-to-liquid waste heat recovery systems including shell-and-tube heat exchangers and plate heat exchangers.

There are several types of heat recovery systems that can be considered to reclaim waste heat in buildings. Some of the commonly used air-to-air heat recovery systems and their applications are described in the following sections.

These systems typically reclaim heat between intake and exhaust air streams and consist of plates, fins, or coils that are placed and extended in both intake and exhaust ducts. The air-to-air heat exchangers can be used to heat intake air during winter and cool it during summer when conditions are favorable. The energy efficiency of the air-to-air heat exchangers depends on the configuration and the temperature difference. Typically, the air-to-air heat exchangers have an energy efficiency that ranges from 45 to 65 percent.

1. *Plate-air-to-air heat exchangers:* These heat recovery systems have the advantage that the exhaust air does not mix with intake air and thus they provide an effective method to retrieve heat virtually free of cross-contamination. The plate air-to-air heat exchangers are attractive for buildings that require large amounts of outside air. Therefore, the commercial applications for these heat recovery systems include hospitals and restaurants. Figure 14.1 illustrates one type of plate air-to-air heat exchanger.
2. *Heat-pipes:* Based on a concept developed for nuclear energy applications during the 1940s, heat pipes provide simple and effective devices to reclaim heat. A heat pipe consists of a copper tube lined with a wick medium and filled with refrigerant. When one end of the heat pipe is heated (by placing it in the exhaust air stream, for instance), the refrigerant is vaporized and flows to the other end to provide heat to the intake air by condensation of the refrigerant. Typically, the heat pipe has a recovery rate that ranges from 50 to 70 percent. Even though heat pipes are more expensive than the plate air-to-air heat exchangers, their maintenance requirements are small because they have no moving parts. The useful life expectancy of heat pipes can be over 25 years. Figure 14.2 presents a basic configuration of a heat exchanger equipped with heat pipes.

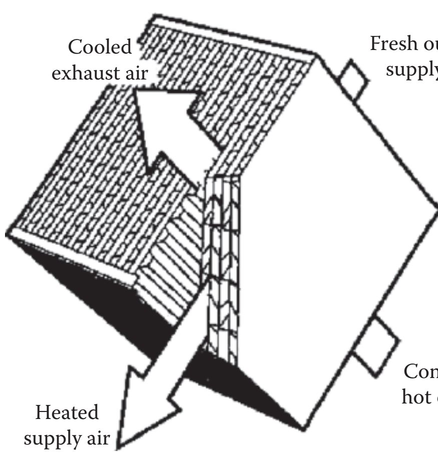


FIGURE 14.1 One configuration of plate air-to-air heat exchanger. (*Source:* ASHRAE, *Handbook of HVAC Systems and Equipment,* 2008.)

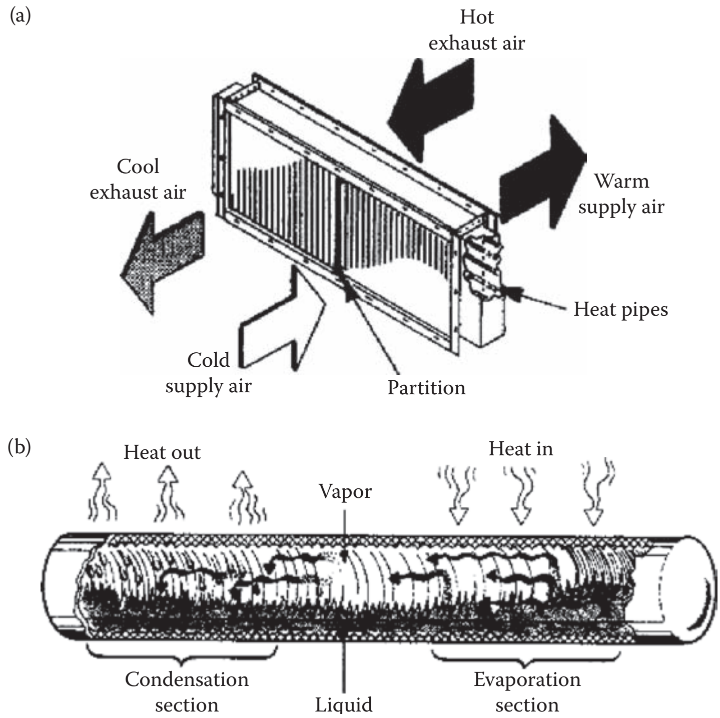


FIGURE 14.2 (a) Air-to-air heat exchanger equipped with heat pipes; (b) construction details of one heat pipe. (*Source*: ASHRAE, *Handbook of HVAC Systems and Equipment,* 2008.)

3. *Rotary thermal wheels:* include a rotating cylinder filled with an air-permeable medium with a large internal surface area. The medium can be appropriately selected either to recover sensible heat only or to reclaim total heat (i.e., sensible and latent heat). Typically, the air streams flow in a counterflow configuration as depicted in Figure 14.3 to increase the heat transfer effectiveness. Generally, rotary thermal wheels include a purge section to reduce cross-contamination between air streams. This cross-contamination occurs by carryover air becoming entrained within the rotating heat exchanger medium or by leakage due to differential pressure across the two air streams.
4. *Glycol loop heat exchangers:* consist generally of finned-tube water coils placed in the supply and exhaust air streams. These coils are part of a closed-loop system that transfers heat from one air stream to the other using a glycol (antifreeze) solution. These systems, often referred to as run-around coils, are suitable for sensible heat recovery applications. Figure 14.4 illustrates one configuration of runaround coil systems that is commonly used to preheat or precool fresh outdoor air with exhaust air.

Table 14.1 summarizes some of the characteristics of the four heat recovery systems discussed above.

14.3 Performance of Heat Recovery Systems

The performance of heat recovery systems can be determined using laboratory testing. For instance, ASHRAE Standard 84 (ASHRAE-84, 2008) presents a systematic procedure for testing and evaluating the performance of air-to-air heat exchangers under controllable laboratory conditions. Moreover, ARI Standard 1060 (ARI, 2001) provides an industry-established standard to rate and verify the performance of air-to-air

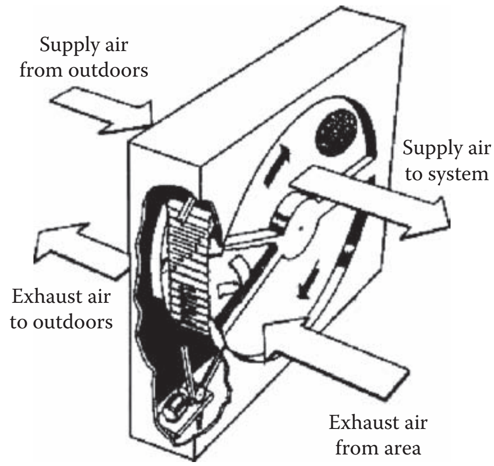


FIGURE 14.3 Basic components of a rotary thermal wheel. (*Source*: ASHRAE, *Handbook of HVAC Systems and Equipment*, 2008.)

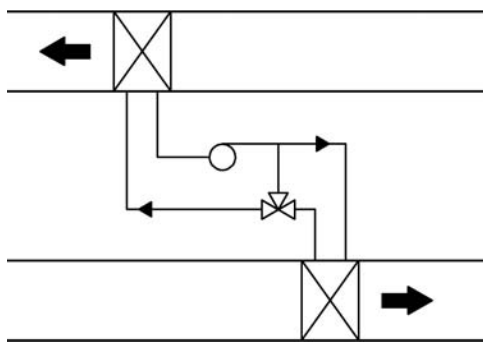


FIGURE 14.4 Basic set-up of a glycol loop heat exchanger.

heat exchangers for use in energy recovery ventilation equipment. However, the in situ performance of heat recovery systems can be significantly different from that obtained through laboratory testing. Indeed, balances in mass airflow as required by both ASHRAE and ARI standards are generally not achieved in field operation of heat recovery systems. Therefore, the actual—rather than the rated—performance of the heat recovery systems should be used to determine their cost-effectiveness in retrofit applications.

The performance of air-to-air heat exchangers depends on the type of heat and mass transfer involved and is typically measured in terms of:

- Sensible energy transfer using dry-bulb temperature
- Latent energy transfer using humidity ratio
- Total energy transfer using enthalpy

Figure 14.5 illustrates a basic model for a heat exchanger between two streams of fluids (typically air). The figure indicates the following parameters:

- X_i (i =1,2,3, and 4) is one of three possible characteristics of the fluid: temperature (for sensible heat), humidity ratio (for latent heat), and enthalpy (for total heat).

TABLE 14.1 Characteristics of Air-to-Air Waste Heat Recovery Systems Commonly Used in Building Applications

	Plate HX	Heat Pipe	Rotary Wheel	Runaround Coil
Temperature range	−56°C to 815°C	−40°C to 35°C	−56°C to 93°C	−45°C to 480°C
Type of heat transfer	−70°F to 1,500°F	−40°F to 95°F	−70°F to 200°F	−50°F to 900°F
Effectiveness range:	Sensible/Total	Sensible	Sensible/Total	Sensible
Sensible	50 to 80%	45 to 65%	50 to 80%	55 to 65%
Total	55 to 85%		55 to 85%	
Heat rate control	By-pass dampers/ ducts	Change tilt angle	Change wheel speed	By-pass valve or change fluid flow
Advantages	(i) Has no moving parts	(i) Has no moving parts	(i) Has a compact size	(i) Allows a separation between two air streams
	(ii) Can be easily cleaned	(ii) Provides flexibility for fan location	(ii) Allows latent heat transfer	(ii) Provides flexibility for fan location
	(iii) Provides low-pressure drop		(iii) Provides low-pressure drop	
Main disadvantage	Not suitable for latent heat transfer	Low effectiveness values	Potential cross-air contamination	Hard to optimize its performance

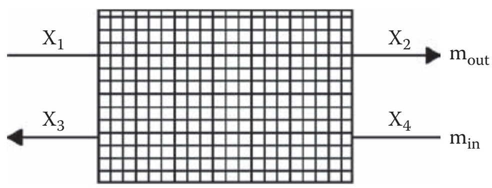


FIGURE 14.5 Typical air-handling unit for an air HVAC system.

- m_{in} is the mass flow rate of the fluid stream from which heat is recovered.
- m_{out} is the mass flow rate of the fluid stream that recovers heat.

To characterize the heat recovery capability of a heat exchanger, an index referred to as the effectiveness is usually provided and is defined as the ratio of the actual heat transfer and the maximum possible heat transfer that can occur between the two streams. Using the heat balance applied to the fluid streams as indicated in Figure 14.1, it can be shown that the effectiveness can be estimated using the following expression:

$$\varepsilon = \frac{m_{in}.(X_2 - X_1)}{Min[m_{in}, m_{out}].(X_4 - X_1)} \tag{14.1}$$

It should be noted, when only sensible heat is recovered, the expression of Eq. (14.1) involves temperature values as indicated in Eq. (14.2):

$$\varepsilon = \frac{m_{in}.c_{p,in}.(T_2 - T_1)}{Min[m_{in}.c_{p,in}; m_{out}.c_{p,out}].(T_4 - T_1)} \tag{14.2}$$

For air-to-air heat exchangers, when the two streams have the same mass flow (such as in the case of make-up air systems), the expression for the effectiveness (sometimes referred to as the efficiency) can be further simplified to

$$\varepsilon = \frac{(T_2 - T_1)}{(T_4 - T_1)} \tag{14.3}$$

In general, the effectiveness of a heat recovery system depends on two main factors including:

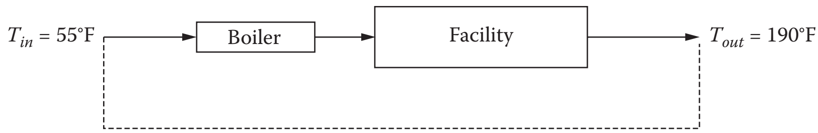
1. The contact surface between the heat exchanger and the fluid streams. The higher this contact surface, the more heat is recovered. However, the increase in the surface area—using fins, plates, and coils—required to improve the heat recovery effectiveness results in an increase in the pressure drop across the heat exchanger which has to be overcome by additional fan/pump power. Another factor that significantly affects the performance of a heat recovery system is fouling which is basically the undesired accumulation of particulates such as dust on the contact surfaces. Fouling results in additional resistance to heat transfer and thus reduces the heat exchanger performance. Moreover, fouling increases surface roughness and increases the pressure drop, and thus may lead to higher fan or pump energy requirements. Therefore, it is important to clean the heat exchanger surfaces regularly in order to maintain a good performance of the heat exchanger.
2. Temperature difference between the two fluid streams. The higher the temperature difference, the more efficient is the heat recovery system. For instance, in cold climates where the temperature difference between air intake and air exhaust is higher than 30°F (20°C), the effectiveness of heat recovery systems is higher than in mild climates where the temperature difference may not exceed 15°F (10°C).

14.4 Simplified Analysis Methods

To assess the feasibility of heat recovery systems, simplified analysis methods can be used. These simplified methods are based on fundamental thermodynamics and heat transfer principles. Example 14.1 presents a simplified energy and cost analysis of reusing waste hot water for a thermal process, and Example 14.2 provides a method to assess the cost-effectiveness of an air-to-air heat exchanger for a laboratory make-up air system. Finally, Example 14.3 outlines a bin-bases analysis method to estimate energy and cost savings for a rotary thermal wheel installed in an air-handling unit serving a hospital building. For a more accurate analysis of the performance of heat recovery systems, detailed simulation tools may be used.

Example 14.1

A thermal process within a facility rejects 10,000 lbm/hr of hot water (T_{out} = 190°F). An energy audit of the facility revealed that instead of using an open water system which feeds fresh cold water (T_{in} = 55°F) to the boiler, a closed water loop system can be set up (with a total cost of $90,000) to reuse the hot water wasted by the facility at the end of the thermal process as depicted in the sketch below. Determine the simple payback period for the closed-loop installation if the cost of fuel is $6/MMBtu and the boiler efficiency is 80 percent. Assume that the facility is operated 4,000 hours per year.



Solution

First the annual fuel savings is estimated based on the energy saved by using water at T_{out} = 190°F rather than T_{in} = 55°F:

$$\Delta Q = \frac{\dot{m}c_p.(T_{out} - T_{in})}{\eta_b} = \frac{(10{,}000 lbm/hr) * (1.0 Btu/hr.^{o}F) * [(190 - 55)^{o}F]}{0.80} = 1.687 * 10^6 Btu/hr$$

Then, the total fuel cost savings during one year can be estimated:

$$\Delta E = \Delta Q * N_h * Cost = 1.687 MMBtu/hr * 4000 hrs/yr * \$6/MMBtu = \$40{,}500/yr$$

Thus, the simple payback for installing the closed-loop water system is:

$$SP = \frac{Initial\_Cost}{Annual\_Savings} = \frac{\$70{,}000}{\$40{,}500} = 2.22 years$$

Therefore, the project of installing a closed water loop system is cost-effective and should be considered to reduce the facility operating cost.

Example 14.2

A heat exchanger is considered to recover heat from the exhaust air of a 5,000-cfm laboratory make-up air system located in Denver, Colorado. The indoor temperature within the laboratory is kept at T_{in} = 70°F. The exhaust air temperature is T_{ea} = 120°F.

(a) Estimate the amount of outdoor air that needs to by-pass the heat recovery system under winter design conditions (T_{oa} =-10°F) to ensure that the supply air temperature is equal to the indoor temperature (i.e., $T_{sa} = T_{in}$).

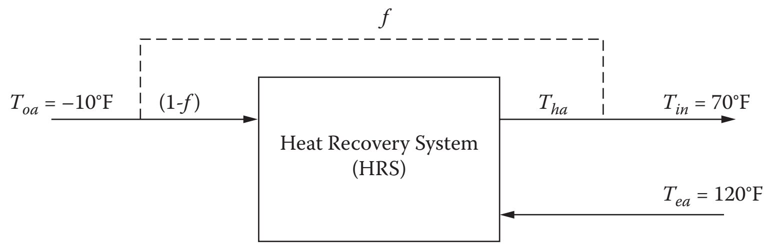
(b) Determine the simple payback period if the installation cost of the heat recovery system is $1/cfm. For Denver, the average winter season outdoor temperature is T_{oa} =42.4°F and the cost of gas is $0.10/CCF. The heat recovery system is operated 24 hours/day, 271 days/year (winter season). The heat content of gas is 840 Btu/ft^3 (Denver is located at an altitude of about 5,280 ft).

Solution

The by-pass factor *f* can be determined using the fact that the supply temperature provided to the laboratory is the result of mixing two air streams as depicted in the diagram below:

- Outdoor air (representing a fraction *f* of the total air supply and kept at the temperature T_{oa} = –10°F)

- Heated air [coming from the heat recovery system at the temperature T_{ha} (to be determined), and representing a fraction (1 – *f*) of the total supply air]



First, the heated air temperature is calculated using the definition of the heat recovery system effectiveness provided by Eq. (14.3):

$$T_{ha} = T_{oa} + \varepsilon.(T_{ea} - T_{oa}) = -10^o F + 0.70 * (120^o F + 10^o F) = 81^o F$$

Then the fraction *f* can be determined by setting $T_{sa} = T_{in}$:

$$T_{in} = T_{oa}.f + T_{ha}.(1-f)$$

Thus the fraction *f* is:

$$f = \frac{(T_{in} - T_{ha})}{(T_{oa} - T_{ha})} = \frac{70-81}{-10-81} = 0.12$$

Using the average winter conditions, the energy rate saved by the heat recovery system can be estimated as following:

$$\Delta E = \dot{m}_a.c_p.(T_{sa} - \overline{T}_{oa})$$

where $T_{sa} = T_{in} = 70^o F$ (assuming that the waste energy is recovered only to temper the make-up air; more thermal energy can actually be recovered if it can be used for other purposes such as space heating) and $\overline{T}_{oa} = 42.4^o F$. Thus:

$$\Delta E = (5{,}000 cfm) * (0.91 Btu/hr.^o F.cfm) * (70^o F - 42.4^o F) = 124{,}200 Btu/hr$$

The savings in fuel use ΔFU can then be calculated using a gas boiler efficiency of 80 percent and a number of operating hours of $N_h = 6{,}504 \quad hrs/yr$ (=24 hrs/day * 271 days/yr):

$$\Delta FU = \frac{\Delta E.N_h}{\eta_b} = \frac{(124{,}200 Btu/hr) * (6{,}504 hrs/yr)]}{0.80} = 1.007 * 10^6 Btu/yr = 21{,}021 CCF/yr$$

Thus, the simple payback period for installing the waste heat recovery system is:

$$SP = \frac{Initial\_Cost}{Annual\_Savings} = \frac{\$1/cfm * 5{,}000 cfm}{\$0.1/CCF * 21{,}021 CCF/yr} = 2.1 \quad years$$

Therefore, the installation of the heat recovery system in the laboratory is cost-effective.

Example 14.3

A thermal wheel is considered to recover heat from the exhaust air of 20,000 air-handling unit operating supplying a hospital in Denver, Colorado. The indoor temperature within the hospital is kept at T_{in} = 68°F during the winter and T_{in} = 75°F during the summer. The effectiveness of the thermal wheel is 75 percent and it is operated during both the heating and cooling seasons 24 hours/day (365 days/year).

Determine the annual cost savings of using the thermal wheel if the gas-fired boiler efficiency is 80 percent and the chiller COP is 3.5. Use a bin analysis to estimate the annual savings in heating, cooling, and fan energy use. Estimate the payback period for installing the thermal wheel if its initial cost is $30,000. The gas cost is $1/MMBtu and electricity cost is $0.05/kWh. The static pressure added by the thermal wheel on both supply and exhaust fans is 0.55 in. of water.

Solution

The annual energy savings can be estimated using a bin analysis based on the following 5°F-bin data for Denver, Colorado (ASHRAE, 1997):

Temperature Range (°F)	Average Temperature (°F)	Hours of Occurence (# of Hrs)
−10 to −6	−8	4
−5 to 0	−3	32
0 to 4	2	31
5 to 9	7	74
10 to 14	12	110
15 to 19	17	207
20 to 24	22	500
25 to 29	27	706
30 to 34	32	700
35 to 39	37	660
40 to 44	42	665
45 to 49	47	671
50 to 54	52	741
55 to 59	57	663
60 to 64	62	655
65 to 69	67	708
70 to 74	72	514
75 to 79	77	353
80 to 84	82	360
85 to 89	87	258
90 to 94	92	131
95 to 99	97	17

The energy analysis of the thermal wheel can be divided into three parts:

- *Heating energy savings*: The annual heating energy savings can be estimated using a bin analysis. Each bin is characterized by an average outdoor temperature $\bar{T}_{oa,b}$ and a number of hours of occurrence N_b:

$$\Delta E_H = \dot{m}_a . c_p . \sum_b N_b . (T_{in} - \bar{T}_{oa,b})$$

Thus:

$$\Delta E_H = (5{,}000 \quad cfm) * (0.91 \quad Btu/hr.^o F.cfm) * \sum_b N_b * (68^o F - \bar{T}_{oa,b})$$

The heating will be needed as long as the outdoor temperature is below the building heating balance temperature, assumed to be 60°F in this case (due to internal gains; see Chapter 6 for more details); that is $\bar{T}_{oa,b} \le 57^o F$.

The fuel use savings can be obtained from the heating energy savings ΔE_H, and the boiler efficiency η_b ,as indicated below:

$$\Delta FU_H = \frac{\Delta E_H}{\eta_b} = \frac{\Delta E_H}{0.80}$$

- *Cooling energy savings:* The sensible cooling energy savings can be estimated by following the same bin analysis considered for the heating energy savings. Thus, cooling energy use savings for each bin can be estimated as indicated below:

$$\Delta E_C = \dot{m}_a . c_p . N_b . \eta_{th} . (\bar{T}_{oa,b} - T_{in,c})$$

Therefore:

$$\Delta E_C = (20{,}000 \quad cfm) * (0.91 \quad Btu/hr.^o F.cfm) * N_b * 0.75 * (\bar{T}_{oa,b} - 75^o F)$$

The electricity savings can be obtained from the cooling energy savings ΔE_C , and the chiller COP as indicated below:

$$\Delta kWh_C = \frac{\Delta E_C}{COP} * \frac{1.0 kWh}{3{,}412 Btu}$$

- *Fan energy penalty*: Because the thermal wheel adds static pressure on both the supply and exhaust fans, additional fan energy is needed to move the air through the duct system. This additional electrical power for one fan can be estimated in terms of horsepower (see Chapter 7):

$$\Delta Hp_{fan} = \frac{cfm.\Delta P_s}{6{,}356 * \eta_s} = \frac{20{,}000 cfm * (0.45 in)}{6{,}356 * 0.65} = 2.18\, Hp$$

Thus, the total fan energy penalty for each bin is determined as follows:

$$\Delta kWh_{fan} = (1.0Hp + 2 * 2.18Hp) * (0.746kW/Hp) * N_b$$

The calculation of the energy savings for both heating and cooling as well as the fan energy penalty has to be performed for each bin as indicated above. The cost savings can then be easily obtained based on the fuel and electricity rates which are, respectively \$3/MMBtu and \$0.05/kWh.

The results of the bin analysis are summarized in the table below:

Temperature Range (°F)	Fuel Use Savings (MMBtu)	Cooling Electricity Savings (kWh)	Fan Energy Penalty (kWh)	Total Cost Savings (\$)
–5 to–1	27	0	88	76
0 to 4	90	0	320	254
5 to 9	68	0	260	190
10 to 14	101	0	424	283
15 to 19	143	0	656	395
20 to 24	313	0	1,595	860
25 to 29	395	0	2,259	1,073
30 to 34	438	0	2,851	1,171
35 to 39	401	0	3,031	1,051
40 to 44	295	0	2,659	752
45 to 49	257	0	2,871	628
50 to 54	214	0	3,131	485
55 to 59	143	0	3,047	277
60 to 64	0	0	0	0
65 to 69	0	0	0	0
70 to 74	0	0	0	0
75 to 79	0	892	1,559	–33
80 to 84	0	2784	1,391	70
85 to 89	0	3223	940	114
90 to 94	0	2293	472	91
95 to 99	0	75	12	3
Total Annual Cost Savings:				**7,739**

It should be noted that the thermal wheel is not operated during the period when the outdoor temperature is in the range of 60 to 74°F (when free cooling can be obtained). The results of the bin analysis indicate clearly that the thermal wheel should also not be operated when the temperature is in the range of 75 to 79°F.

From the annual cost savings of \$7,739, the simple payback period can be estimated to be:

$$SP = \frac{Initial\_Cost}{Annual\_Savings} = \frac{\$30,000}{\$7,739} = 3.9 \quad years$$

Therefore, the installation of the thermal wheel is cost-effective.

14.5 Summary

In this chapter, selected types of heat recovery systems are discussed. Simplified analysis methods are presented to evaluate the cost-effectiveness of air-to-air heat recovery systems in some HVAC applications. In the future, it is expected that heat recovery systems will be more commonly integrated with other HVAC equipment such as chillers, cogeneration engines, and heat pumps. Typically, heat recovery systems are cost-effective even though they are fairly expensive to install.

Problems

14.1 Provide a simple payback period of installing a heat wheel system in an 80,000-cfm constant volume AHU. During the winter, air is exhausted at 72°F, and during the summer at 76°F. The heat recovery system operates during both heating and cooling seasons with 78 percent efficiency.

The boiler is 75 percent efficient and uses natural gas above 50°F ambient and diesel below 50°F ambient. The chiller uses 0.8 kW/ton. Assume that the energy costs are $0.10/kWh, $0.90/gal (diesel), and $1.50/CCF (gas).

The heat recovery system costs $1.50 per cfm and increases the static pressure in both supply and exhaust ducts by 0.75 in WG. Assume that the fan static efficiency is 75 percent. To move the wheel, a 1 .5 hp motor is required.

To solve this problem, use the bin calculation for the Denver climate.

14.2 Use the data provided in Problem 14.1. Determine the critical cost of fuel, diesel and natural gas (assuming the cost of electricity remains constant), when using the heat wheel is not economical if the interest rate is 7 percent, the inflation rate is 3 percent, and the life cycle is 20 years.

14.3 Determine the simple payback period of installing a heat recovery system at a cost of $1.25/cfm for a 10,000-cfm make-up air system in a laboratory located in Chicago, Illinois. The indoor temperature within the laboratory is kept at $T_{in} = 68^o F$. The exhaust air temperature is $T_{ea} = 110^o F$. Assume that the cost of gas is $1.25/CCF and that the heat recovery system is operated 24 hours/day, 175 days/year.

14.4 Determine the simple payback period of installing a heat recovery system at a cost of $0.15/L/s for a 7,000/L/s make-up air system in a laboratory located in New York, New York. The indoor temperature within the laboratory is kept at $T_{in} = 20^o C$. The exhaust air temperature is $T_{ea} = 50^o C$. Assume that the cost of gas is $1.50/therm and that the heat recovery system is operated 24 hours/day during 200 days/year.

15

Water Management

15.1 Introduction

In recent years, the cost of water usage has increased significantly and represents an important portion of the total utility bills especially for residential buildings. In some western U.S. cities where the population growth has been high, the cost of water has increased by more than 400 percent during the past ten years. In the future, it is expected that the cost of water will increase at higher rates than the cost of energy and the consumer price index (CPI). In the United States, energy used for domestic hot water accounts for 15 percent of residential energy consumption, the third largest energy end-use in homes after space heating (47 percent) and lighting and appliances (24 percent) as indicated by a survey conducted for residential energy consumption (EIA, 2009). Therefore, it is worthwhile to explore potential savings in water use and water heating expenditures during a building energy audit.

Under the Energy Policy Act of 1992, the U.S. government recognized the need for water management and required federal agencies to implement any water conservation measure with a payback period of ten years or less. The Federal Energy Management Program (FEMP) has been established to help federal agencies identify and implement cost-effective conservation measures to improve both energy and water efficiency in federal facilities. The technical assistance offered by FEMP includes development of water conservation plans, training information resources, and software tools. In particular, FEMP has developed WATERGY, computer software that can be used to estimate potential water and associated energy savings for buildings.

There are several water conservation strategies that can be considered for buildings. These strategies can be grouped into three main categories:

1. Indoor water management with the use of water-efficient plumbing systems (such as low-flow showerheads and water-efficient dishwashers and washing machines)
2. Outdoor water management associated with irrigation and landscaping (including the use of low-flow sprinkler heads, irrigation control systems, and xeriscape)
3. Recycling of water usage by installing processing systems that reuse water

Some of the proven water conservation technologies and techniques are described in the following sections with a special emphasis on indoor water management.

15.2 Indoor Water Management

The use of water-conserving fixtures and appliances constitutes one of the most common methods of water conservation, particularly in residential buildings. In general, the retrofit of toilets, showerheads, and faucets with water-efficient fixtures can be performed with little or no change in lifestyle for the building occupants. Similarly, water-saving appliances such as dishwashers and clothes washers can provide an effective method to reduce indoor water usage in buildings. Another common and generally cost-effective method to conserve water is repairing leaks. It is estimated that up to 10 percent of water is wasted due to leaks (deMonsabert, 1996). In addition to water savings, energy use reduction can be achieved when the water has to be heated as in the case of domestic hot water applications (i.e., showering and hand washing). In the following sections, selected water-conserving technologies are described with illustrative examples to showcase the potential of water and associated energy savings due to implementation of these technologies.

15.2.1 Water-Efficient Plumbing Fixtures

Water, distributed through plumbing systems within buildings, is used for a variety of purposes such as hand washing, showering, and toilet flushing. In recent years, water-efficient plumbing fixtures and equipment have been developed to promote water conservation. Table 15.1 summarizes the typical U.S. household end-use of water with and without conservation. The average U.S. home can reduce inside water usage by about 32 percent by installing water-efficient fixtures and appliances and by reducing leaks.

In this section, some of the proven water-efficient products are briefly presented with some calculation examples that illustrate how to estimate the cost-effectiveness of installing water-efficient fixtures. As a general rule, it is recommended to test the performance of water-efficient products to ensure user satisfaction before any retrofit or replacement projects.

15.2.1.1 Water-Saving Showerheads

The water flow rate from showerheads depends on the actual inlet water pressure. In accordance with the Energy Policy Act of 1992, the showerhead flow rates are reported at an inlet water pressure of 80 psi. The water flow rate is about 4.0 gpm (gallons per minute) for older showerheads, and is 2.2 gpm for newer showerheads. The best available water-efficient showerheads have flow rates as low as 1.5 gpm. In addition to savings in water usage, water-efficient showerheads provide savings in heating energy cost. The calculation procedure for the energy use savings due to reduction in the water volume to be heated is presented in Section 15.2.3 and is illustrated in Example 15.1.

15.2.1.2 Water-Saving Toilets

Typical existing toilets have a flush rate of 3.5 gpf (gallons per flush). After 1996, toilets manufactured in the United States were required to have flush rates of at least 1.6 gpf. Since 2003, high-efficiency toilets with water usage of less than 1.3 gpf have been available. Therefore, significant water savings can be achieved by retrofitting existing toilets especially when they become leaky. Leaks in both flush-valve and gravity tank toilets are common and are often invisible. The use of dye tablet testing helps the detection of toilet water leaks.

TABLE 15.1 Average U.S. Household Indoor Water End Use with and without Conservation

End Use	Without Conservation (gallons/capita/day)	With Conservation (gallons/capita/day)	Savings (gallons/capita/day)
Toilets	20.1 (27.7%)	9.6 (19.3%)	10.5 (52%)
Clothes washers	15.1 (20.9%)	10.6 (21.4 %)	4.5 (30%)
Showers	12.6 (17.3%)	10.0 (20.1%)	2.6 (21%)
Faucets	11.1 (15.3%)	10.8 (21.9%)	0.3 (2%)
Leaks	10.0 (13.8%)	5.0 (10.1%)	5.0 (50%)
Other domestic	1.5 (2.1%)	1.5 (3.1%)	0 (0%)
Baths	1.2 (1.6%)	1.2 (2.4%)	0 (0%)
Dishwashers	1.0 (1.3%)	1.0 (2.0%)	0 (0%)
Total	72.5 (100%)	49.6 (100%)	22.9 (32%)

Source: AWWA, *Water Use Inside the Home*, Report of American Water Works Research Foundation, 1999.

Example 15.1

Determine the annual energy, water, and cost savings associated with replacing an existing showerhead (having a water flow rate of 2.5 gpm) with a low-flow showerhead (1.6 gpm). An electric water heater is used for domestic hot water heating with an efficiency of 95 percent. The temperature of the showerhead water is 110°F. The inlet water temperature for the heater is 55°F. The showerhead use is 10 minutes per shower, 2 showers per day, and 300 days per year. Assume that the electricity price is \$0.07/kWh and that the combined water and waste water cost is \$4/1,000 gallons.

Solution

The annual savings in water usage due to replacing a showerhead using 2.5 gpm by another using only 1.6 gpm can be estimated as follows:

$$\Delta m = 2 * [(2.5 - 1.6) gpm] * 10 \min/day * 300 days/yr = 5{,}400 gal/yr$$

The energy savings incurred from the reduction in the hot water usage can be estimated as indicated below:

$$\Delta E = 5{,}400 gal/yr * 8.33 Btu/gal.{}^\circ F[(110 - 55)^\circ F]/0.95 = 2.604 * 10^6 Btu/yr$$

Therefore, the annual cost savings in energy use and in water use are, respectively:

$$\Delta Cost = [(2.604x106 Btu/yr)/(3.413 Btu/Wh)] * 1000 W/kW * \$0.07/kWh = \$53/yr$$

and

$$\Delta Cost = 5{,}400 gal/yr * \$4.0/1000 gal = \$22/yr$$

Thus, the total annual savings incurred from the water-efficient showerhead are \$75. These savings make the investment in water-saving showerheads virtually certain to be cost-effective.

15.2.1.3 Water-Saving Faucets

To reduce water usage for hand washing, low-flow and self-closing faucets can be used. Low-flow faucets have aerators that add air to the water spray to lower the flow rate. High-efficiency aerators can reduce the water flow rates from 4 gpm to 0.5 gpm. Flow rates as low as 0.5 gpm are adequate for hand washing in bathrooms. Self-closing faucets are metered and are shut off automatically after a specified time (typically 10 seconds) or when the user moves away from the bathroom sink (as detected by a sensor placed on the faucet). The water flow rates of self-closing faucets can be as low as 0.25 gpc (gallons per cycle).

15.2.1.4 Repair Water Leaks

It is important to repair leaks in water fixtures even if these leaks consist of few water drips per minute. Over a long period of time, the amount of water wasted from these drips can be significant as indicated in Table 15.2. The daily, monthly, and annual water wasted due to leaks can be estimated using Table 15.2 by simply counting the number of drips in one minute from the leaky fixture. It should be noted that a leak of 300 drips per minute (i.e., 5 drips per second) corresponds to steady water flow.

15.2.1.5 Water/Energy Efficient Appliances

In addition to water-efficient fixtures, water can be saved in residential buildings by using water-efficient appliances such as clothes washers and dishwashers. The reduction of water needed to clean dishes or clothes can actually increase the energy efficiency of the household appliances. Indeed, a large fraction of the electrical energy used by both clothes washers and dishwashers is attributed to heating the water (85 percent for clothes washers and 80 percent for dishwashers). Typical water and energy performances of conventional and efficient models available for residential clothes washers and dishwashers are summarized in Table 15.3. Example 15.2 provides an estimation of the potential water and energy savings due to the use of efficient clothes washers.

Example 15.2

Estimate the annual cost savings incurred by replacing an existing clothes washer (having 2.65 ft^3 tub volume) with a water/energy-efficient appliance that has an energy factor of 2.50 ft^3/kWh and uses 42 gallons per load. The washer is operated based on 400 cycles (loads) per year. The water is heated using an electric heater. Assume that the electricity cost is \$0.07/kWh and that water/sewer cost is \$5/1000 gallons.

Solution

Using the information provided in Table 15.3, the water savings due to using efficient clothes washer is 13.0 gallons per load. Based on 400 loads per year, the annual water savings are 5,200 gallons which amounts to cost savings of \$26.

The annual electrical energy savings per load can be calculated using the energy factors as indicated in Table 15.3:

$$\Delta E = 2.65 f^3t/load * [(1/1.18 - 1/2.50) kWh/ft^3] * 400.loads/yr = 474.3 kWh/yr$$

Thus the annual electrical energy cost savings are \$33.

Therefore, the total annual cost savings achieved by using an energy/water-efficient clothes washer are \$59. The cost-effectiveness of the efficient clothes washer depends on the cost differential in purchasing price (between the efficient and the conventional clothes washer models). For instance, if the cost differential is \$200, the payback period for using an efficient clothes washer can be estimated to be:

$$Payback = \frac{\$200}{\$59} = 3.4\, years$$

In this case, the use of water/energy-saving clothes washer is cost-effective.

15.2.2 Domestic Hot Water Usage

In most buildings, hot water is used for hand washing and showering. To heat the water, electric or gas boilers or heaters are generally used. The energy input E_w required to heat the water can be estimated using a basic heat balance equation as expressed by Eq. (15.1):

TABLE 15.2 Volumes of Water Wasted from Small Leaks

Number of Drips per Minute	Water Wasted per Day (gal/day)	Water Wasted per Month (gal/month)	Water Wasted per Year (gal/year)
1	0.14	4.3	52.6
5	0.72	21.6	262.8
10	1.44	43.2	525.6
20	2.88	86.4	1,051.2
50	7.20	216.0	2,628.0
100	14.40	432.0	5,256.0
200	28.80	864.0	10,512.0
300	43.20	1,296.0	15,768.0

TABLE 15.3 Water and Energy Efficiencies of Residential Dishwashers and Clothes Washers

Performance	Dishwashers	Clothes Washers
Older models:		
Water use (gal/cycle)	14.0	55.0
Energy factor[a]	0.46	1.18
Energy Star models:		
Water use (gal/cycle)	6.5	7.5
Energy factor[a]	0.46	1.8

[a] The energy factor is a measure of the energy efficiency of the appliance. For dishwashers, the energy factor is the number of full wash cycles per kWh. For clothes washers it is the volume of clothes washed (in ft^3) per kWh per cycle. The water use for Energy Star clothes washers is expressed in gallons used per ft^3 of volume of clothes washed per cycle.

$$E_w = m_w c_{w,p}(T_{w,i} - T_{w,o}) / \eta_{WH} \tag{15.1}$$

where

m_w = the mass of the water to be heated. The hot water requirements depend on the building type. The *ASHRAE Applications Handbook* (2007) provides typical hot water use for various building types.

$C_{w,p}$ = the specific heat of water.

$T_{w,i}$ = the water temperature entering the heater. Typically, this temperature is close to the deep ground temperature (i.e., well temperature).

$T_{w,o}$ = the water temperature delivered by the heater and depending on the end-use of the hot water.

η_{WH} = the efficiency of the water heater.

Based on Eq. (15.1) there are four approaches to reduce the energy required to heat the water. These approaches are:

TABLE 15.4 Typical Hot Water Temperature for Common Residential Applications

Applications	Hot Water Temperatures
Dishwashers	140–160°F (60–71°C)
Showers	105–120°F (41–49°C)
Faucet flows	80–120°F (27–49°C)
Clothes washers	78–93°F (26–34°C)

Source: Adapted from ASHRAE, *Handbook of HVAC Systems and Equipment,* 2008.

TABLE 15.5 Minimum NAECA Efficiency Standards for Water Heaters

Water Heater Fuel Type	Minimum Acceptable Energy Factor (EF)
Electric storage	0.97 – (0.00132 × rated storage volume in gallons)
Electric: instantaneous	0.93 – (0.0013 × rated storage volume in gallons)
Gas: storage	0.67 – (0.0019 × rated storage volume in gallons)
Gas: instantaneous	0.62 – (0.0019 × rated storage volume in gallons)
Oil	0.59 – (0.0019 × rated storage volume in gallons)

1. Reduction of the amount of water to be heated (i.e., m_w). This approach can be achieved using water-saving fixtures and appliances as discussed in Section 15.2.1. Further reduction in domestic hot water use can be realized through changes in water use habits of building occupants.
2. Reduction in the delivery temperature. The desired delivery temperature depends mostly on the end-use of the hot water. It should be noted that a decrease of the water temperature occurs through the distribution systems (i.e., hot water pipes). The magnitude of this temperature decrease depends on several factors such as the length and the insulation level of the piping system. For hand washing, the delivery temperature is typically about 120°F (49°C). For dish washing, the delivery temperature can be as high as 160°F (71°C). However, booster heaters are recommended for use in areas where high delivery temperatures are needed such as dishwashers. A detailed analysis of water/energy efficient appliances can be found in Koomey, Dunham, and Lutz (1994).
3. Increase the temperature of the water entering the water heater using heat recovery devices including drain-water heat recovery systems. When hot water goes down the drain, it carries away with it energy, typically 80–90 percent of the energy used to heat water in a home. To capture this energy, drain-water heat recovery systems can be used to preheat cold water entering the water heater. These drain-water heat recovery systems can be used effectively with all types of water heaters, especially with tankless and solar water heaters. In particular, they can be used to recover heat from hot water used in showers, bathtubs, sinks, dishwashers, and clothes washers. The payback period, for installing a drain-water heat recovery system, ranges from 2.5 to 7.0 years depending on how often the system is used.
4. Increase in the overall efficiency of the hot water heater. This efficiency depends on the fuel used to operate the heater and on factors such as the insulation level of the hot water storage tank. The National Appliance Energy Conservation Act (NAECA, 2004) defined the minimum acceptable efficiencies for various types and sizes of water heaters as indicated in Table 15.5. The energy factor (EF) is a measure of the water heater efficiency and is defined as the ratio of the energy content of the heated water to the total daily energy used by the water heater.

Example 15.3

(a) Estimate the annual energy cost of domestic hot water heating for an office building occupied by 1,000 people. Each person uses on average 3 gallons per day of hot water. The water temperature at the bathroom sink is 120°F. Due to the length of the distribution system, an average of 20°F decrease in water temperature occurs before the hot water reaches the bathroom sink. The heater is an oil-fired boiler with an efficiency of 75 percent. The cost of fuel oil is $1.20/gal. Assume that the entering water temperature is 55°F and that the building is occupied 300 days per year.

(b) Determine the cost savings if the losses in hot water temperature in the distribution system are virtually eliminated by insulating the pipes.

Solution

(a) Using Eq. (15.1), the annual fuel use by the water heater can be estimated. First, the water mass can be determined

$$m_w = 3.0\,gal/pers/day * 1000\,pers * 300\,days/yr * 8.33\,lbm/gal = 7.497 * 10^6\,lbm/yr$$

Therefore, the fuel use required to heat the water from 55°F to 140°F (=120°F + 20°F) is:

$$FU = 7.497 * 10^6\,lbm/yr * 1.0\,Btu/lbm \cdot {}^\circ F * [(140 - 55)^o F)]/(0.75 * 138 * 10^3\,Btu/gal)z$$

or

$$FU = 6{,}157\,gals/yr$$

Thus, the annual fuel energy cost for water heating is:

$$Cost = [6{,}157\,gals/yr] * \$1.20/gal = \$7{,}388/yr$$

When the temperature decrease across the piping system is reduced from 20°F to 10°F (so that the actual hot water temperature from the heater is reduced from 140°F to 130°F), a reduction in the annual cost of heating the water results and is estimated to be:

$$\Delta Cost = [1 - (130 - 55)/(140 - 55)] * \$7{,}388/yr = \$869/yr$$

It should be noted that the losses through the distribution systems are generally referred to as parasite losses and can represent a significant fraction of the fuel use requirements for water heating.

An economic evaluation analysis should be conducted to determine the cost-effectiveness of insulating the piping system.

15.3 Outdoor Water Management

Outdoor water conservation includes mostly innovative strategies for landscaping and irrigation of lawns and trees. Specifically, water savings can be achieved by reducing the overwatering of lawns using adequate irrigation control systems, or by replacing all or part of a landscape with less water-dependent components such as rocks and indigenous vegetation, a method known as xeriscaping. Other areas to conserve outdoor water use include swimming pools and HVAC equipment such as evaporative cooling systems.

15.3.1 Irrigation and Landscaping

In addition to its esthetics, vegetation consisting of trees, shrubs, or turfgrass can have a positive impact on the energy use in buildings by reducing cooling loads, especially in hot and arid climates. However, with water costs rising, it is important to reduce the irrigation cost for a vegetated landscape. The amount of water necessary for irrigation depends on several factors including the type of plant and the climate conditions.

In a study conducted in a residential neighborhood consisting of 228 single-family homes near Boulder, Colorado, Mayer (1995) found that 78 percent of the total water used in the test neighborhood during the summer is attributed to lawn irrigation. Therefore, outdoor water management provides a significant potential to reduce water use for buildings. Some of the practical recommendations to reduce irrigation water use include:

- Water lawns and plants only when needed. The installation of tensiometers to sense the soil moisture content helps to determine when to water.
- Install automatic irrigation systems that provide water during early mornings or late evenings to reduce evaporation.
- Use a drip system to water plants.
- Add mulch and water-retaining organic matter to conserve soil moisture.
- Install windbreaks and fences to protect the plants against winds and reduce evapotranspiration.
- Install rain gutters and collect water from downspouts to irrigate lawns and garden plants.
- Select trees, shrubs, and groundcovers based on their adaptability to the local soil and climate.

The amount of irrigation water use is generally difficult to determine exactly and depends on the type of vegetation and the local precipitation. Typically, the water needed by a plant is directly related to its potential evapotranspiration (ET) rate. The ET rate of a plant measures the amount of water released through evaporation and transpiration of moisture from the leaves. Please see Table 15.6. Figure 15.1 illustrates various factors that may affect the ET and the local climate around a typical residential building (Conchilla and Krarti, 2002a). The maximum possible ET rate can be estimated using one of several

TABLE 15.6 Annual Normal Precipitation and ET Rates for Selected U.S. Locations

Location	ET for Turfgrass[a] (in.)	ET for Common Trees[a] (in.)	Precipitation[a] (in.)
Phoenix, AZ	48.10	30.05	3.77
Austin, TX	38.86	24.29	21.88
San Francisco, CA	26.90	16.82	3.17
Boulder/Denver, CO	26.96	16.85	13.92
Boston, MA	24.13	15.08	22.38

[a] The values are provided for a growing season assumed to extend from April to October for all locations.

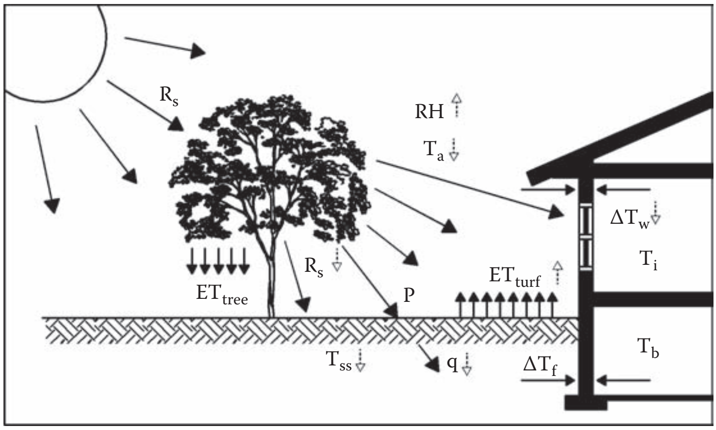


FIGURE 15.1 Effects of ET and other factors on the local climate around a typical house.

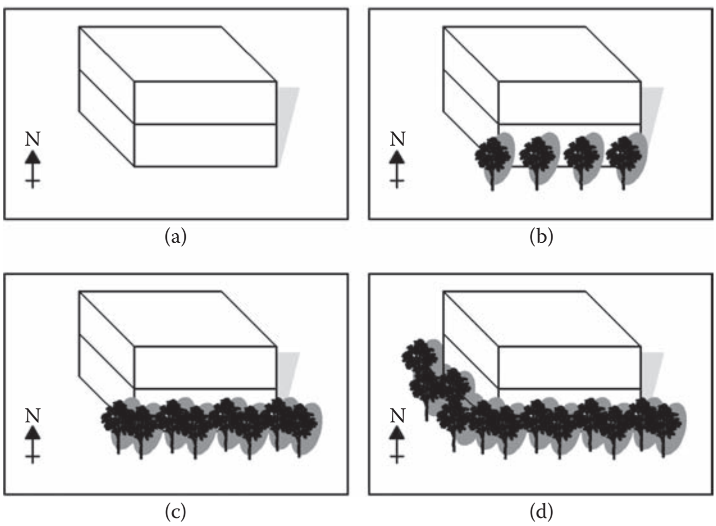


FIGURE 15.2 Number and location of trees around a house considered in the analysis performed by Conchilla and Krarti *ASHRAE Transactions,* Vol. 108, Part 1, 345–356 (2002a).

calculation procedures including the Penman method (Periera, 1996) based on climate driving forces (i.e., solar radiation, wind speed, ambient temperature).

A well-designed landscaping, although it consumes water, can actually save energy to heat and cool a building. Indeed, some computer simulation studies indicated that well-positioned trees around a house can save up to 50 percent in energy use for cooling. Table 15.7 provides the magnitude of energy savings in the cooling loads for a typical residential dwelling surrounded by a variable number of trees in selected U.S. locations based on a simulation analysis (Conchilla and Krarti, 2002b). The placement configurations of the trees are illustrated in Figure 15.2. The savings in building cooling energy use attributed to vegetation are the effects of the cooler microclimate that trees and soil cover create due to shading and evapotranspiration effects. A number of studies (Huang, 1987; Taha, 1997; Akbari, 1993; Akbari, Huang, and Davis, 1997; Conchilla, 1997) showed that summertime air temperatures can be 2°F to 6°F (1°C to 3°C) cooler in tree-shaded areas than in treeless locations. During the winter, the trees provide windbreaks to shield the buildings from wind effects such as air infiltration and thus reduce heating energy use.

TABLE 15.7 Percent Savings of Annual Cooling Loads for Single-Family Homes for Various Vegetation Types[a]

Location	Turfgrass (%)	4 Trees (%)	8 Trees (%)	12 Trees (%)
Phoenix, AZ	1.2	4.5	8.0	13.1
Austin, TX	2.9	4.8	9.2	14.7
San Francisco, CA	18.7	27.2	40.3	57.0
Denver, CO	3.4	10.4	18.7	30.7
Boston, MA	8.9	8.6	19.7	34.8

[a] These savings are estimated relative to a bare soil with no groundcover and no trees.

15.3.2 Waste Water Reuse

The reuse options for building applications are generally limited to graywater use and rainwater harvesting. However, the available options depend on the type and location of the building, and the legal regulations applicable to water reuse. Sewage water treatment facilities are other options that are available but require large investments that are too costly for individual buildings to consider.

Graywater is a form of waste water with a lesser quality than potable water but higher quality than black water (which is water that contains significant concentrations of organic waste). Sources of black water include water that is used for flushing toilets, washing in the kitchen sink, and dish washing. Graywater comes from other sources such washing machines, baths, and showers and is suitable for reuse in toilet flushing. In addition, graywater can be used instead of potable water to supply some of the irrigation needs of a typical domestic dwelling landscaped with vegetation. It is believed that graywater can actually be beneficial for plants because it often includes nitrogen and phosphorus which are plant nutrients. However, graywater may also contain sodium and chloride which can be harmful to some plants. Therefore, it is important to chemically analyze the content of graywater before it is used to irrigate the vegetation around the building.

Several graywater recycling systems are available ranging from simple low-cost systems to sophisticated high-cost systems. For instance, a small water storage tank can be easily connected to a washing machine to recycle the rinse water from one load to be used in the wash cycle for the next load. The most effective systems include settling tanks and sand filters for the treatment of graywater.

Another method to conserve water used for irrigation is rainwater harvesting especially in areas where rainfall is scarce. The harvesting of rainwater is suitable for both large and small landscapes and can be easily planned in the design of a landscape. There is a wide range of harvesting systems to collect and distribute water. Simple systems consist of catchment areas and distribution systems. The catchment areas are places from which water can be harvested such as sloped roofs. The distribution systems such as gutters and downspouts help direct water to landscape holding areas which can consist of planted areas with edges to retain water. More sophisticated water harvesting systems include tanks that can store water between rainfall events or periods. These systems result in larger water savings but require higher construction costs and are generally more suitable for large landscapes such parks, schools, and commercial buildings.

15.4 Swimming Pools

The energy performance for recreational centers has not been thoroughly analyzed in the United States. However, in the United Kingdom several studies have been conducted to address the energy consumption of these buildings. *The Good Practice Guide* (*Guide 52*) is a series of papers developed to provide the average energy consumption of different buildings in the United Kingdom and measures to reduce it, breaks down recreational centers in two broad categories: good or poor. Tables 15.8 and 15.9 show the typical energy use associated with good and poor recreation centers, respectively.

TABLE 15.8 United Kingdom Good Practice Guide Results of "Good" Recreational Centers Average Energy Use

Good	Gas, kWh/m^2 (kWh/ft^2)	Electricity, kWh/m^2 (kWh/ft^2)
Dry sports	<215 (20)	<75 (7)
Sports and pool	<360 (33)	<150 (14)
Pool only	<775 (72)	<165 (15)

Source: Haberl J.S., and Bou-Saada T.E. *ASME Solar Energy Engineering Journal,* 120(3), 193, 1998. With permission.

TABLE 15.9 United Kingdom Good Practice Guide Results of "Poor" Recreational Centers Average Energy Use

Poor	Gas, kWh/m^2 (kWh/ft^2)	Electricity, kWh/m^2 (kWh/ft^2)
Dry sports	>325 (30)	>85 (8)
Sports and pool	>540 (50)	>205 (19)
Pool only	>1120 (104)	>235 (22)

For poor type facilities with a dry sports center and swimming pools, the total consumption (electric and gas) is nearly 70 kWh/ft^2. The average electric consumption is 30 percent of the total consumed, with the remaining 70 percent fuel used for domestic hot water, space heating, and swimming pool water heating.

In this section, calculation procedures are outlined to estimate water and energy use associated with operating indoor and outdoor swimming pools. The impact of using covers on water and energy use is also discussed.

15.4.1 Evaporative Losses

Several factors affect water and energy losses from indoor swimming pools including pool surface area, pool water temperature, ambient air temperature and relative humidity, air velocity at the water surface, and the number of swimmers. The main losses from the pool to the ambient air occur through radiation, convection, and evaporation. Typically, evaporative losses contribute a significant portion of water losses and more than 60 percent of the total energy losses, whereas convection and radiation are responsible for most of the remaining energy losses. Conductive energy losses can occur along the pool's surfaces in contact with the ground but are typically very small.

Several methods are available to estimate evaporative losses. Most of these models are based on rather simplified assumptions and ignore the effect of swimmers on evaporative losses including methods developed by Carrier (1918) and presented in the *ASHRAE Applications Handbook* (ASHRAE, 2007). In particular, the Carrier method is limited to inactive pools and simply lists different activity factors dependent on the type of pool (i.e., lap pool, residential pool, and diving pool) to adjust for the variations in the evaporative losses. The Shah method, a recently developed method to estimate evaporative losses, allows the user to input the actual number of people/ft^2 (Shah, 2004). The Shah method can also be used to estimate evaporative losses for both active and inactive pools. Figure 15.3 illustrates the results of a comparative analysis among experimental data and predictions from three calculation methods for experiments conducted for six pools (Mozes and Krarti, 2009). It is found that the Shah method provides the closest estimation results, with generally lower evaporative losses than the other two calculation methods. Although the Shah method underestimates the evaporative losses for the first two tests, the results are still closer to the measured data than the predictions from the other two methods.

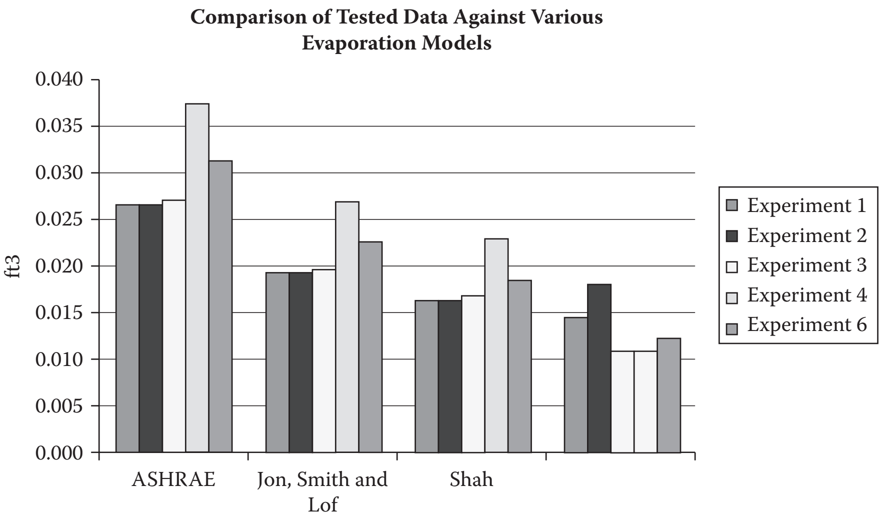


FIGURE 15.3 Shah method results with tested data.

This method was also derived from empirical testing but accounts for several factors (Shah, 2003). The method provides two correlations, one suitable for active pools and the other applicable to inactive pools. For inactive pools, the evaporative losses are calculated as follows:

$$E = A_p K D_w (D_r - D_w)^{1/3} (W_w - W_r) \tag{15.2}$$

where

E = evaporative loss, lbm_w/hr (kg/s).
A_p = area of pool, ft³ (m³).
D_w = density of saturated air at pool temperature, lb/ft³ (kg/m³).
D_r = density of air at room temperature and humidity, lb/ft³ (kg/m³).
W_w = humidity ratio of saturated air at pool temperature, lb_w/lbd_a (kg_w/kg_a).
W_r = humidity ratio of air at room temperature and humidity, lb_w/lbd_a (kg_w/kg_a).
K = 290 if $D_r - D_w > 0.00125$, or 333 if $D_r - D_w < 0.00125$.

For active pools, the evaporative losses are obtained from Equations (15.3) and (15.4):

$$E_o = A_p(0.023 - (3.347 \times 10^{-7})(A_p/N) + 0.041(p_w - p_r)) \tag{15.3}$$

where

N = number of occupants per unit pool area.
p_w = partial water pressure in saturated air, at pool temperature, inHg (Pa).
p_r = partial water pressure in air, at room temperature, inHg (Pa).

$$E = E_o(160(N/A_p) + 1) \text{ if } A_p/N > 500,$$

$$E = E_o(62.9(N/A_p) + 1.2) \text{ if } 50 < (A_p/N) < 500 \tag{15.4}$$

$$E = 2.5E_o \text{ if } A_p/N < 50$$

15.4.2 Impact of Pool Covers

Pool blankets are the most effective measure for significantly reducing evaporative and convective losses from the pool to the environment while the pool is unoccupied. These blankets can be either manually operated or automatic, and can be found in at least three different types of materials:

1. *Bubble PVC:* normally used with manually operated systems. It is the cheapest material and needs replacement more often (some manufacturers claim a ten-year life, but the warranty never covers more than two years).
2. *Foamed rubber sheet:* more expensive and durable than bubble PVC, with expected life between four and five years.
3. *Heavy duty covers:* manufactured from either bubble PVC or foamed rubber, but these are the most durable of all pool covers.

The cost effectiveness of pool blankets depends on several factors including initial costs and operating costs. To reduce operating costs and facilitate manipulation, it is recommended that pool blankets be kept at a convenient location. A nearby wall may be used and a pool "rack" installed so that the cover can be rolled in and out easily. If a wall is not available, racks may have to be left near the pool but at some distance to allow access to the pool by swimmers. Enclosed roller housing may be used to hide the pool blanket during hours when the pool is used. An integrated design with pool blankets built into the side walls of the swimming pools can be an alternative for new swimming pools, but usually are not cost effective for existing pools.

Table 15.10 indicates the impact of using covers on the energy use of four indoor swimming pools (Mozes and Krarti, 2009). A significant energy savings is achieved by covering swimming pools ranging from 17 to 30 percent. On average, a saving of 0.10 MBtu/ft^2-year (of pool area) can be attained by using pool covers.

Table 15.11 illustrates the impact of using covers on energy use of outdoor pools. For outdoor pools, energy savings are greater because evaporative losses and convective losses are higher than indoor swimming pools. An average savings of 0.60 MBtu/ft^2 of pool area can be achieved for outdoor pools that use covers.

TABLE 15.10 Indoor Swimming Pool Loads with and without Cover and Percent Reduction

Facility	Pool	Without Cover	With Cover	Reduction (%)
Facility # 1	Children's pool	2,067	1,669	19
	Lap pool	2,551	1,867	27
Facility # 2	Lap pool	2,323	1,709	26
Facility # 3	Children's pool	1,408	1,165	17
	Lap pool	1,504	1,135	25
Facility # 5	Indoor lap	1248	877	30

TABLE 15.11 Outdoor Swimming Pool Loads with and without Cover and Percent Reduction

Facility	Pool	Without Cover	With Cover	Reduction (%)	Mbtu Difference	Pool Area, ft^2	Mbtu/ft^2
Facility # 5	Outdoor lap	4,890	2,713	45	2,177	3,608	0.60
Facility # 6	Outdoor lap	6,420	3,896	39	2,524	4,200	0.60

15.5 Summary

This chapter outlined some measures to reduce water usage in buildings. In particular, landscaping and waste water reuse, water-saving plumbing fixtures, heat recovery systems, and swimming pool covers can provide substantial water and energy reduction opportunities. The auditor should perform water use analysis and evaluate any potential water management measures for the building.

Problems

15.1 An office building has 200 occupants, each of whom uses 3.5 gallons of hot water per day for 250 days each year. The temperature of the water as it enters the heater is 55°F (an annual average). The water must be heated to 150°F in order to compensate for a 20°F temperature drop during storage and distribution, and still be delivered at the tap at 130°F. The hot water is generated by an oil-fired boiler (using fuel No. 2) with an annual efficiency of 0.60. The cost of fuel is $ 0.80 per gallon. Determine:

(a) The fuel cost savings if the delivery temperature is reduced to 90°F

(b) The fuel cost savings if hot water average usage is reduced by two-thirds

15.2 Determine the best insulation thickness, using simple payback period analysis, to insulate a hot water storage tank with a capacity of 500 gallons. The cost of fiberglass insulation is as follows: $2.60/sqft for 1-in., $2.95/sqft for 2.-in., and $3.60/sqft for 3-in. fiberglass insulation. The average ambient temperature surrounding the tank is 65°F. Assume appropriate dimensions for the 500-gallon tank.

15.3 Determine the annual evaporative losses from a 3,500-sqft indoor swimming pool with indoor conditions of 78°F and 60 percent RH. The water temperature is kept at 85°F throughout the year. Determine the cost-effectiveness of a cover if the pool is operated on average 12 hours per day and 300 days per year. The cost of the cover is $1.25/sqft. Assume the energy cost to heat the water is $20/MMBtu.

15.4 Using monthly calculations based on weather conditions of Denver, Colorado, determine the annual evaporative losses from a 4,000-sqft outdoor swimming pool with water temperature kept at 80°F throughout the year. Determine the cost-effectiveness of a cover if the pool is operated on average 10 hours per day and 250 days per year. The cost of a cover is $1.50/sqft. Assume a gas-fired boiler is used to heat the water at a cost of $2/gal and a boiler efficiency of 85 percent.

16

Methods for Estimating Energy Savings

16.1 Introduction

After an energy audit of a facility, a set of energy conservation measures (ECMs) are typically recommended. Unfortunately, several of the ECMs that are cost-effective are often not implemented due to a number of factors. The most common reason for not implementing ECMs is the lack of internal funding sources (available to owners or managers of the buildings). Indeed, energy projects have to compete for limited funds against other projects that are perceived to have more visible impact such as improvements in productivity within the facility.

Over the last decade, a new mechanism for funding energy projects has been proposed to improve the energy efficiency of existing buildings. This mechanism, often called performance contracting, can be structured using various approaches. The most common approach for performance contracting consists of the following steps:

- A vendor or contractor proposes an energy project to a facility owner or manager after conducting an energy audit. This energy project would save energy use but most likely energy cost and thus would reduce the facility operating costs.
- The vendor or contractor funds the energy project typically using borrowed money from a lending institution.
- The vendor or contractor and facility owner or manager agree on a procedure to repay the borrowed funds from energy cost savings that may result from the implementation of the energy project.

An important feature of performance contracting is the need for a proven protocol for measuring and verifying energy cost savings. This measurement and verification protocol has to be accepted by all the parties involved in the performance contracting project: the vendor or contractor, the facility owner or manager, and the lending institution. For different reasons, all the parties have to ensure that cost savings have indeed incurred from the implementation of the energy project and are properly estimated.

The predicted energy savings for energy projects based on an energy audit analysis are generally different from the actual savings measured after implementation of the energy conservation retrofits. For instance, Greeley et al. (1990) found in a study of over 1,700 commercial building energy retrofits, that a small fraction (about 16 percent) of the energy projects produced predicted savings within 20 percent of the measured results. Therefore, accepted and flexible methods to measure and verify savings are needed to encourage investments in building energy efficiency.

Direct "measurements" of energy savings from energy efficiency retrofits or operational changes are almost impossible to perform because several factors can affect energy use such as weather conditions, levels of occupancy, and HVAC operating procedures. For instance, Eto (1988) found that during abnormally cold and warm weather years, energy consumption for a commercial building can be, respectively, 28 percent higher and 26 percent lower than the average weather year energy use. Thus, energy savings cannot be easily obtained by merely comparing the building energy consumption before (pre) and after (post) retrofit periods.

Over the last few years, several measurement and verification (M&V) protocols have been developed and applied with various degrees of success. Among the methods proposed for the measurement of energy savings are those proposed by the National Association of Energy Service Companies (NAESCO, 1993), the Federal Energy Management Program (FEMP, 1992, 2000, and 2008), the American Society of Heating Refrigeration and Air Conditioning Engineers (ASHRAE, 1997; 2002), the Texas LoanSTAR program (Reddy et al., 1994), and the North American Energy Measurement and Verification Protocol (NEMVP) sponsored by DOE and later updated and renamed the International Performance Measurement and Verification Protocol (IPMVP, 1997, 2002, 2007).

In this chapter, general procedures and methods for measuring and verifying energy savings are presented. Some of these methods are illustrated with calculation examples or with applications reported in the literature.

16.2 General Procedure

To estimate the energy savings incurred by an energy project, it is important to first identify the implementation period of the project, that is, the construction phase where the facility is subject to operational or physical changes due to the retrofit. Figure 16.1 illustrates an example of the variation of the electrical energy use in a facility that has been retrofitted from constant volume to a variable-air-volume HVAC system. The time-series plot of the facility energy use clearly indicates the duration of the construction period, the end of the preretrofit period, and the start of the postretrofit period. The duration of the construction period depends on the nature of the retrofit project and can range from few hours to several months.

The general procedure for estimating the actual energy savings ΔE_{actual} from a retrofit energy project is based on the calculation of the difference between the preretrofit energy consumption predicted from a model and the postretrofit energy consumption obtained directly from measurement (Kissock, Reddy, and Claridge, 1998):

$$\Delta E_{actual} = \sum_{j=1}^{N} \Delta E_j = \sum_{j=1}^{N} (\tilde{E}_{pre,j} - E_{post,j}) \tag{16.1}$$

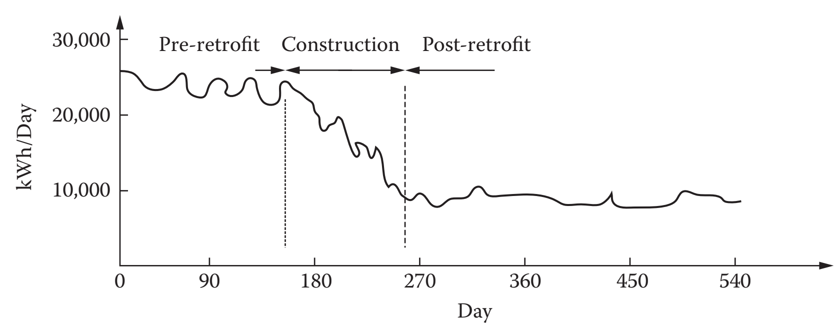


FIGURE 16.1 Daily variation of building energy consumption showing preretrofit, construction, and postretrofit periods after CV to VAV conversion.

where

N = the number of postretrofit measurements (for instance, during one year, N = 365 daily data can be used).

$\tilde{E}_{pre,j}$ = the energy use predicted from a preretrofit model of the facility using the weather and operating conditions observed during the postretrofit period.

$E_{post,j}$ = the energy used by the facility measured during the postretrofit period.

Therefore, it is important to develop a preretrofit energy use model for the facility before estimating the retrofit energy savings. This preretrofit model helps determine the baseline energy use of the facility knowing the weather and operating conditions during the postretrofit period.

In some instances, the energy savings estimated using Eq. (16.1) may not be representative of average or typical energy savings from the retrofit project. For instance, the measured energy use during the postretrofit period may coincide with abnormal weather conditions and thus may lead to retrofit energy savings that are not representative of average weather conditions. In this case, a postretrofit energy use model for the building can be used instead of measured data to estimate normalized energy savings that would occur under typical weather and operating conditions. The normalized energy savings ΔE_{norm} can be calculated as follows:

$$\Delta E_{norm} = \sum_{j=1}^{N} \Delta \tilde{E}_j = \sum_{j=1}^{N} (\tilde{E}_{pre,j} - \tilde{E}_{post,j}) \tag{16.2}$$

where

$\tilde{E}_{pre,j}$ = the energy use predicted from a preretrofit model of the facility using normalized weather and operating conditions.

$E_{post,j}$ = the energy use predicted from a postretrofit model of the facility using normalized weather and operating conditions.

It should be noted that the energy savings calculation procedures expressed by Eqs. (16.1) and (16.2) can be applied to subsystems of a building (such as lighting systems, motors, chillers, etc.) as well as to the entire facility. In recent years, several techniques have been proposed to estimate various energy end-uses of a facility. Some of these approaches are discussed in Section 16.3.

There are several approaches to estimate the energy savings from energy retrofits. These methods are used as part of the M&V protocol in performance contracting projects and range from simplified engineering approaches to detailed simulation and measurement techniques. For specific projects, the method to be used in M&V of savings depends on the desired depth of the verification, the accuracy level of the estimation, and on the accepted cost of the total M&V project. In general, the cost of the M&V procedure depends on the metering equipment needed to obtain detailed data on the energy consumption of the facility and its end-uses. The installation cost of the metering equipment can represent up to 5 percent of the total energy project costs, especially if no energy management system is available in the facility. A number of suggestions have been proposed to reduce the cost of metering equipment including:

- Use of statistical sampling techniques to reduce metering requirements such as the case for lighting retrofit projects where only a selected number of lighting circuits are metered. A complete metering of all the lighting circuits within a facility, although feasible, can be very costly.
- End-use metering for specific systems directly affected by the retrofit project. For a CV to VAV (variable-air-volume) conversion project, for example, metering may be needed only for the retrofitted air-handling unit.

- Limited metering requirements with stipulated calculation procedures to verify savings. For instance, the energy savings incurred from a replacement of motors can be estimated using simplified engineering methods that may require only the metering of operation hours.

Some of the most commonly used methods for verifying savings are briefly presented in the following section.

16.3 Energy Savings Estimation Models

16.3.1 Simplified Engineering Methods

These methods are commonly used in performance contracting projects that include retrofits of lighting systems or motors. The calculation methods have to be accepted by all the parties involved in the retrofit project to be representative of actual savings. The step-by-step calculation procedure is generally part of the written agreement between the parties involved in the energy project. Examples 16.1 and 16.2 provide illustrations of some of the simplified engineering methods that one can use to verify savings in lighting and motor retrofit projects. The reader is referred to Chapter 5 for further details about the calculation procedures to estimate the energy and cost savings for lighting retrofit projects.

Example 16.1

A medium office building has 800 luminaires equipped with four 40-W lamps and standard magnetic ballasts. The manager of the building agreed with an energy service company to replace these luminaires with four 32-W lamps and electronic ballasts. In the agreement, a specific calculation procedure was defined to verify the energy savings due to the lighting retrofit. In particular, it was agreed that the lighting system is operated 8 hours per day, 5 days per week,and 50 weeks per year. In addition, the wattage rating is set to be 192 W for the standard luminaire and 140 W for the energy-efficient luminaire.

Estimate the annual energy savings stipulated by the agreement. Determine the cost savings due to the lighting retrofit if the electricity cost is \$0.07/kWh.

Solution

The reduction in the electrical energy input for all the luminaires is first determined:

$$\Delta kW_{light} = [(0.192 - 0.140)kW] * 800 = 41.6 \quad kW$$

The total number of hours for operating the lighting system during one year can be estimated as follows

$$\Delta t_{light} = 8hrs/day * 5days/week * 50weeks/year = 2{,}000hrs/yr$$

Thus, the annual energy savings due to the lighting retrofit are:

$$\Delta kWh_{light} = 41.6kW * 2{,}000hrs/yr = 83{,}200kWh/yr$$

Therefore, the annual cost savings that resulted from the stipulated savings are:

$$\Delta C_{light} = \$0.07/kWh * 83{,}200kWh/yr = \$5{,}824/yr$$

Example 16.2

An energy service company agreed to convert all 12 constant volume air-handling units (AHUs) within a commercial building into variable-air-volume systems by installing variable frequency drives (VFDs) and VAV terminal boxes. To verify fan energy savings, it was agreed to meter only one AHU. Metering all 12 AHUs was determined to be economically infeasible. The metering of one AHU revealed the following: (a) the supply fan is rated at $kW_{e,\,fan}$ = 40 kW and is operated for 6,000 hours/year, and (b) the VFD operates according to the following load profile:

- At 100 percent of its speed, 500 hours per year (i.e., 8.33 percent of total operating hours)
- At 80 percent of its capacity, 3,500 hours per year (i.e., 58.33 percent of total operating hours)
- At 60 percent of its capacity, 1,500 hours per year (i.e., 25.0 percent of total operating hours)
- At 40 percent of its capacity, 500 hours per year (i.e., 8.33 percent of total operating hours)

It was agreed that the energy use of one motor is proportional to its speed squared (The reader is referred to Chapter 7 for more details on fan performance). Determine the fan energy and cost savings for the CV to VAV conversion for all the AHUs as stipulated in the agreement which states that the annual fan energy savings for all the AHUs is simply 12 times the fan energy savings estimated from the metered AHU. Assume that electricity costs $0.07/kWh.

Solution

The electrical energy input of the supply fan motor is first determined for the metered AHU by calculating a weighted average fan rating (see Example 7.3 for more details):

$$k\bar{W}^{VSD}_{e,fan} = 40kW * (1.0 * 0.083 + 0.64 * 0.583 + 0.36 * 0.25 + 0.16 * 0.083) = 22.4kW$$

The total number of hours for operating the fan during one year is $N_{f,h}$ = 6,000 hours. The electrical energy use for one supply fan can be estimated as indicated below:

$$kWh^{CV}_{e,fan} = 40kW * 6{,}000hrs = 240{,}000kWh/yr$$

and

$$kWh^{VSD}_{e,fan} = 22.4kW * 6{,}000hrs = 134{,}400kWh/yr$$

Thus, the annual fan energy savings due to the CV to VAV conversion for all 12 AHUs are:

$$\Delta kWh_{AHUs} = 12 * [(240{,}000 - 134{,}400)kWh/yr] = 1{,}267{,}200kWh/yr$$

The annual fan energy cost savings that resulted from all 12 retrofitted AHUs are:

$$\Delta C_{light} = \$0.07/kWh * 1{,}267{,}200kWh/yr = \$88{,}704/yr$$

16.3.2 Regression Analysis Models

Using metered data, models for building energy use for pre- or postretrofit periods can be established using regression analysis. The first developed regression model for building energy use estimation has been an application of the variable-base degree-days (VBDD) method (the reader is referred to Chapters 4 and 6 for a more detailed description of the VBDD method). Indeed, Fels (1986) proposed the Princeton scorekeeping method (PRISM) to correlate the monthly utility bills and the outdoor temperatures to estimate energy use for heating and cooling and estimate any energy savings for residential retrofit measures. A similar approach has been used to develop a regression analysis method, FASER, for large groups of commercial buildings (OmniComp, 1984).

In recent years, general regression approaches have been proposed to establish baseline models for commercial building energy use which can be used to estimate retrofit savings. Two main regression models have been developed and applied successfully to predict energy use in commercial buildings:

1. Single-variable regression analysis models that assume that the building energy use is driven by one variable (typically, the ambient temperature)
2. Multivariable regression models that account for other independent variables (such as solar radiation, humidity ratio, and internal gain) in addition to temperature to predict the building energy use

A brief description of these two regression analysis models is provided in the following sections.

16.3.2.1 Single-Variable Regression Analysis Models

These models constitute the main procedures adopted by the International Performance Measurement and Verification Protocol (IPMVP, 1997, 2002, 2007). A simple linear correlation is assumed to exist between building energy use and one independent variable. The ambient temperature is typically selected as the independent variable especially to predict commercial/residential building heating and cooling energy use. Degree-days with properly selected balance temperature can be used as another option for the independent variable.

Ambient-temperature-based regression models have been shown to predict building energy use with an acceptable level of accuracy even for daily datasets (Kissock et al., 1992; Katipamula, Reddy, and Claridge, 1994; Kissock and Fels, 1995) and can be used to estimate energy savings (Claridge et al., 1991; Fels and Keating, 1993). Four basic functional forms of the single-variable regression models have been proposed for measuring energy savings in commercial and residential buildings. The selection of the function form depends on the application and the building characteristics. Figure 16.2 illustrates the four basic functional forms commonly used for ambient-temperature linear regression models. The regression models, also called change-point or segmented-linear models, combine both search methods and least-squares regression techniques to obtain the best-fit correlation coefficients. Each change-point regression model is characterized by the number of correlation coefficients. Therefore, the two-parameter model has two correlation coefficients (β_o and β_1) and consists of a simple linear regression model between building energy use and ambient temperature. Table 16.1 summarizes the mathematical expressions of four change-point models and their applications. In general, the change-point regression models are more suitable for predicting heating rather than cooling energy use. Indeed, these regression models assume steady-state conditions and are insensitive to the building dynamic effects, solar effects, and nonlinear HVAC system controls such as on–off schedules.

Figure 16.3 provides an example of a three-parameter model for gas usage of a home in Colorado using an ambient-temperature linear regression method. Figure 16.4 illustrates, for the same home, a two-parameter model using a heating degree-day linear regression model.

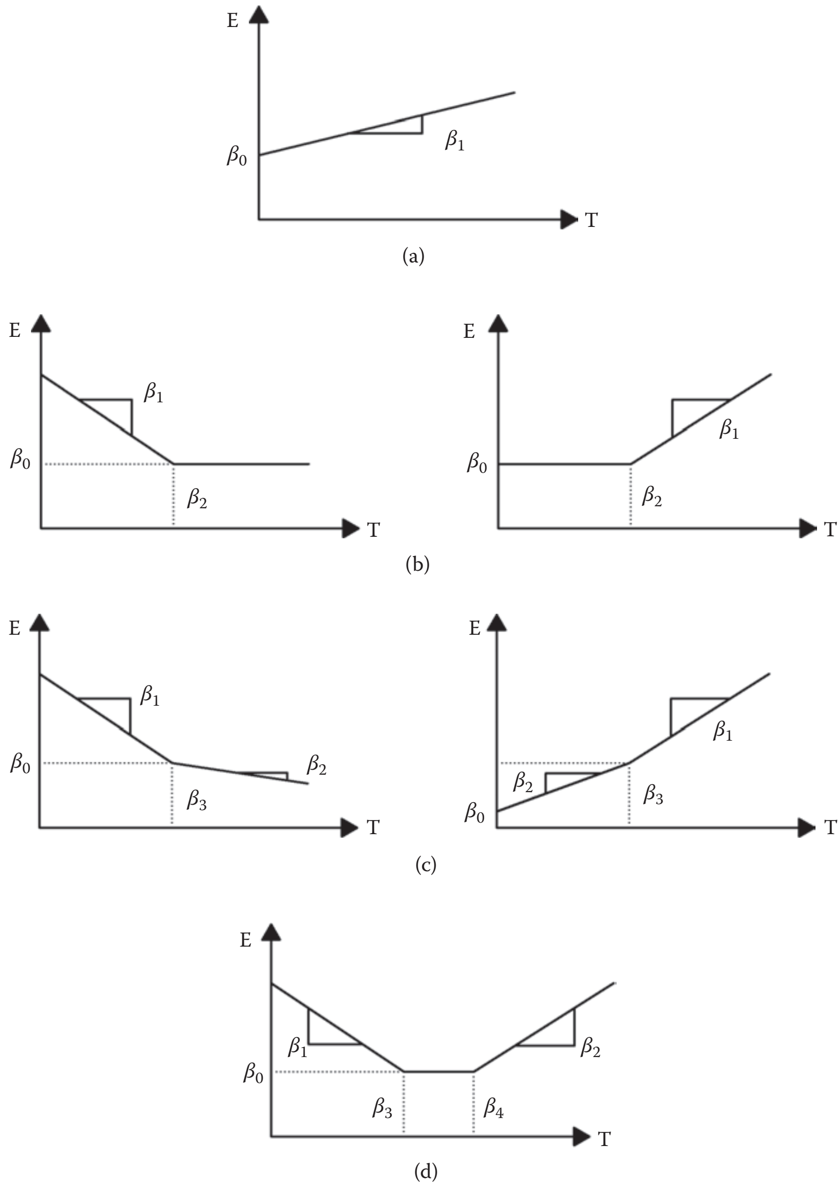


FIGURE 16.2 Basic forms of single-variable regression models: (a) two-parameter model; (b) three-parameter models; (c) four-parameter models; (d) five-parameter model.

Other types of single-variable regression models have been applied to predict the energy use of HVAC equipment such as pumps, fans, and chillers. For instance, Phelan, Brandemuehl, and Krarti (1996) used linear and quadratic regression models to obtain a correlation between the electrical energy used by fans and pumps and the fluid mass flow rate.

16.3.2.2 Multivariable Regression Analysis Models

These regression models use several independent variables to predict building energy use or energy savings due to retrofit projects. Several studies indicated that the multivariable regression models provide

TABLE 16.1 Mathematical Expressions and Applications of Change-Point Regression Models

Model Type	Mathematical Expression	Applications
Two-parameter (2-P)	$E = \beta_0 + \beta_1 \cdot T$	Buildings with constant air volume systems and simple controls
Three-parameter (3-P)	Heating: $E = \beta_0 + \beta_1 \cdot (\beta_2 - T)^+$ Cooling: $E = \beta_0 + \beta_1 \cdot (T - \beta_2)^+$	Buildings with envelope-driven heating or cooling loads (i.e., residential and small commercial buildings)
Four-parameter (4-P)	Heating: $E = \beta_0 + \beta_1 \cdot (\beta_3 - T)^+ - \beta_2 \cdot (T - \beta_3)^+$ Cooling: $E = \beta_0 + \beta_1 \cdot (\beta_3 - T)^+ + \beta_2 \cdot (T - \beta_3)^+$	Buildings with variable-air-volume systems or with high latent loads. Also, buildings with nonlinear control features (such as economizer cycles and hot deck reset schedules)
Five-parameter (5-P)	$E = \beta_0 + \beta_1 \cdot (\beta_3 - T)^+ + \beta_2 \cdot (T - \beta_4)^+$	Buildings with systems that use the same energy source for both heating and cooling (i.e., heat pumps, electric heating and cooling systems)

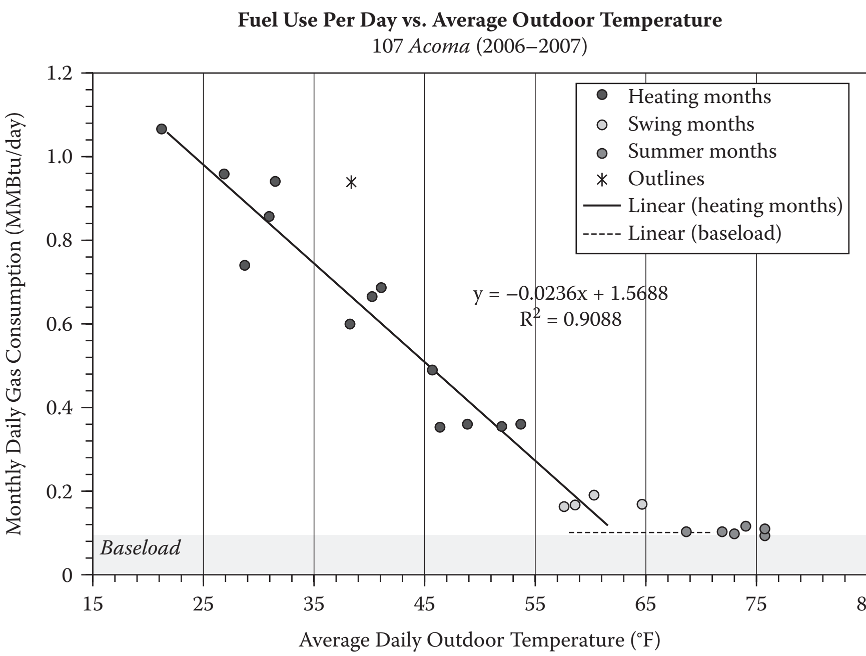


FIGURE 16.3 Three-parameter regression model based on outdoor ambient temperature for a home in Colorado.

better predictions of monthly, daily, and even hourly energy use of large commercial buildings than the single-variable models (Haberl and Claridge, 1987; Katipamula, Reddy, and Claridge, 1994, 1995, 1998). In addition to outdoor temperature, multivariable regression models use internal gain, solar radiation, and humidity ratio as independent variables. For instance, the cooling energy use for commercial buildings conditioned with VAV systems can be obtained using the following functional form of a multivariate regression model (Katipamula, Reddy, and Claridge, 1998):

$$E = \beta_0 + \beta_1 T_a + \beta_2 I + \beta_3 I T_a + \beta_4 T_{dp}^{+} + \beta_5 q_i + \beta_6 q_s \tag{16.3}$$

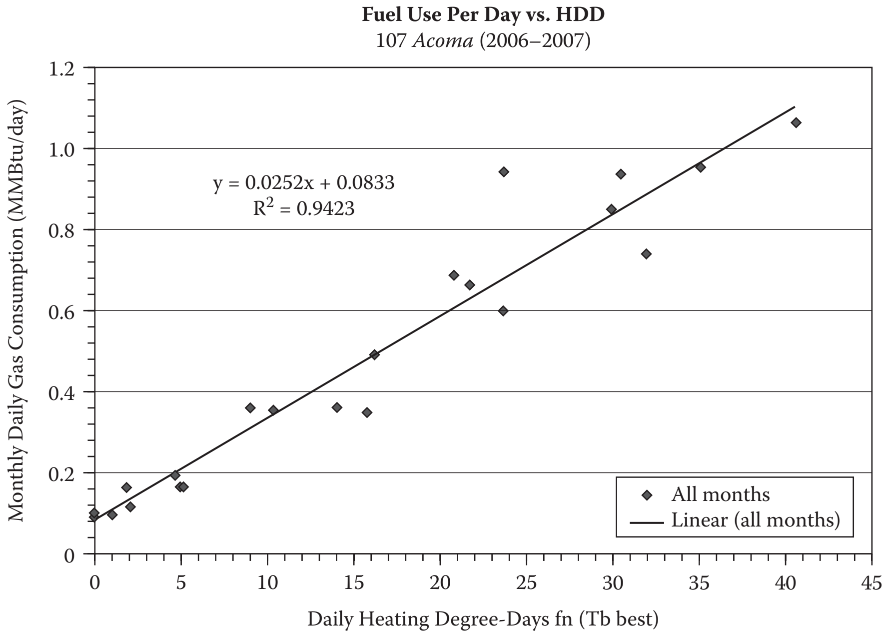


FIGURE 16.4 Two-parameter regression model based on heating degree-day for the same home considered in Figure 16.3.

where

β_0 , through β_6 are regression coefficients.
I is an indicator variable that is used to model the change in slope of the energy use variation as a function of the outdoor temperature.
T_a is the ambient outdoor dry-bulb temperature.
T_{dp}^+ is the outdoor dew-point temperature.
q_i is the internal sensible heat gain.
q_s is the total global horizontal solar radiation.

For commercial buildings conditioned with a constant-volume HVAC system, the cooling energy use can be predicted using a simplified model of Eq. (16.3) as follows:

$$E = \beta_0 + \beta_1 T_a + \beta_2 T_{dp}^+ + \beta_3 q_i + \beta_4 q_s \tag{16.4}$$

The models represented by Eqs. (16.3) and (16.4) have been applied to predict cooling energy consumption in several commercial buildings using various time-scale resolutions: monthly, daily, hourly, and hour-of-day (HOD). The HOD predictions, which require a significant modeling effort because the energy use data has to be regrouped in hourly bins corresponding to each hour of the day and 24 individual hourly models have to be obtained, are found to have better accuracy (Katipamula, Reddy, and Claridge, 1994). Table 16.2 indicates some of the advantages and disadvantages of the different multivariable regression modeling approaches. Generally, monthly utility bills can be used to develop monthly regression models but metering is required to establish daily, hourly, and HOD models.

TABLE 16.2 Typical Advantages and Disadvantages of Multivariable Regression Models with Different Time Resolution

Advantages/Disadvantages	Monthly	Daily	Hourly	HOD
Modeling Effort	Minimum	Minimum	Moderate	Difficult
Metering needs	None	Required	Required	Required
Data requirements	At least 12 months	At least 3 months	At least 3 months	At least 3 months
Application to savings estimation	In some cases	In most cases	All cases	All cases
Prediction accuracy	Low	High	Moderate	High

Source: Katipamula, Reddy, and Claridge, Multivariate regression modeling, *ASME Solar Energy Engineering Journal*, 120(3), 177, 1998.

It should be noted that the multivariable regression models discussed above are developed without retaining the time-series nature of the data. Other regression models can be considered to preserve the time variation of the building energy use. For instance, Fourier series models can be used to capture the daily and seasonal variations of commercial buildings energy use (Dhar, Reddy, and Claridge, 1998).

Multivariable regression models have been applied not only to estimate total building energy use but also to predict the behavior of individual pieces of HVAC equipment such as chillers, fans, and pumps. In particular, polynomial models have been widely used to model chiller energy use as a function of part-load ratio, evaporator leaving temperature, and condenser entering temperature (LBL, 1982). Other regression models for chillers have been obtained based on fundamental engineering principles (Gordon and Ng, 1994).

16.3.3 Dynamic Models

The regression models discussed in Section 16.3.2 generally cannot account for transient effects such as thermal mass that can cause short-term temperature fluctuations during warmup or cooldown periods of a building. To capture building energy use transient effects, dynamic models are typically recommended. Most of the dynamic models that are based on a physical representation of the building energy systems are complex in nature and require detailed calibration procedures. These models can be grouped into four major types: (i) thermal network models, (ii) time-series models, (iii) differential equation models, and (iv) modal analysis models. Some of these models are briefly discussed in Chapter 4. For more detailed evaluation of the four types of dynamic models, the reader is referred to studies by Rabl (1988) and Reddy (1989).

In recent years, other types of dynamic models based on connectionist approaches have been applied to predict building energy use and estimate retrofit energy savings. An international competition (Kreider and Haberl, 1994; Haberl and Thamilseran, 1996) has indicated that the connectionist approach provides superior accuracy for predicting both long- and short-term energy use in buildings compared to traditional methods. Techniques based on neural networks (NNs) have been developed and applied to forecast building energy use for both short- and long-term periods (Kreider and Wang, 1992; Anstett and Kreider, 1993; Gibson and Kraft, 1993; Kreider et al., 1995, 1997). In particular, NNs have been used to predict hourly building thermal and electrical energy use for a period of one day, one week, and even one month. Typically, weather data, occupancy profiles, and day types have been considered as input parameters for the NNs to predict building energy use.

A typical neural network consists of several layers of neurons that are connected to each other. A connection is a unique information transport link from one sending to one receiving neuron. Figure 16.5 shows a schematic diagram of the structure of a typical neural network. The first and last

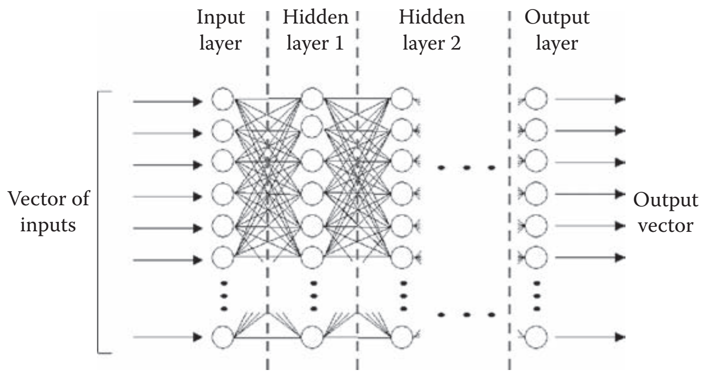


FIGURE 16.5 A typical structure of a neural network with hidden layers interposed between input and output layers.

layers of neurons are called input and output layers; between them are one or more hidden layers. The connections between the layers are determined using a training dataset to "learn" the weights between various neurons.

It should be noted that if not used properly neural networks may tend to "memorize" the noise in training data. Various techniques that exist reduce this overtraining problem. These techniques are discussed by Kreider et al. (1995). In general, however, NNs can be very flexible models that can approximate many kinds of input–output mappings.

A number of applications of NNs to estimate savings from energy conservation measures have been proposed and are discussed by Krarti et al. (1998). For instance, a NN-based approach to determine energy savings from building retrofits has been proposed by Cohen and Krarti (1995, 1997). The retrofit savings were estimated using the difference between predicted preretrofit energy consumption extrapolated into the post period (using NNs) and the actual postretrofit measured data. The following steps summarize the procedure used to determine retrofit savings with NNs:

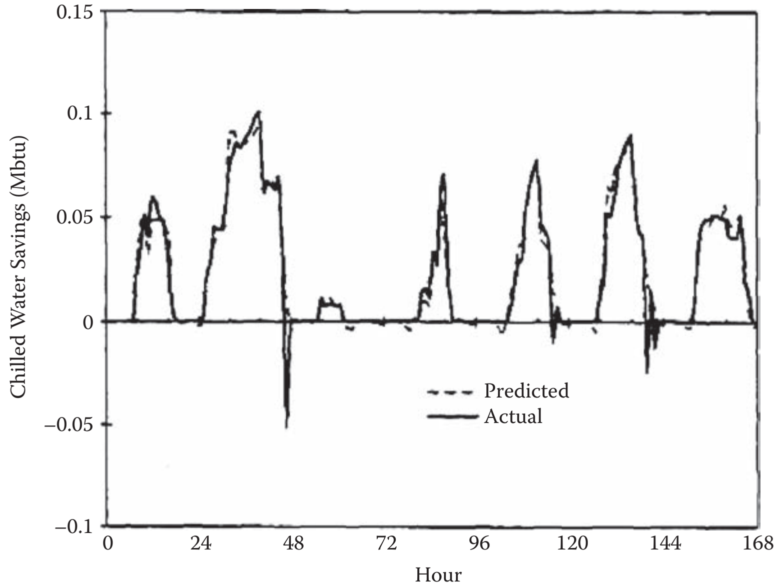
- From the preretrofit building energy use data, a preretrofit NN model is developed to predict building energy consumption.
- Using the preretrofit NN model, prediction is made of what future (i.e., in the postretrofit period) building energy use would be if the retrofit were not implemented.
- Retrofit energy savings are estimated from the difference between the actual postretrofit measured data and the energy use obtained from the previous step.
- A second NN model is developed to correlate selected input variables to the retrofit energy savings.

To illustrate the performance of the NN-based approach to estimate energy savings, Cohen and Krarti (1995) considered three energy conservation measures applied to an educational building:

1. High-efficiency lighting retrofit (changing fluorescent lamps from F40-T12 to high-efficiency T-8 lamps and electronic ballasts).
2. Variable-air-volume system retrofit to replace a dual-duct constant-volume system (in particular, the supply fans are equipped with a variable-speed drive to allow the supply air flow to vary with load).
3. Combination variable-speed chiller drive retrofit and thermal storage system addition (the chiller is equipped with an adjustable-frequency drive on the compressor motor. In addition, a chilled water tank is installed. The chiller provides off-peak cooling and supplements the tank during the on-peak period if necessary).

To evaluate the performance of the trained NNs, a one-week testing set was selected. The week used for testing was not part of the training set used to determine the weights of the NNs. Figure 16.6 illustrates the performance of the NNs in predicting the energy savings for the three retrofits when the educational building is assumed to be located in Chicago, Illinois. In particular, Figure 16.6(a) compares the predicted and actual savings for the whole building chilled water consumption when lighting retrofit is implemented. The NN accurately estimates the hourly chilled water savings for the entire

(a)



(b)

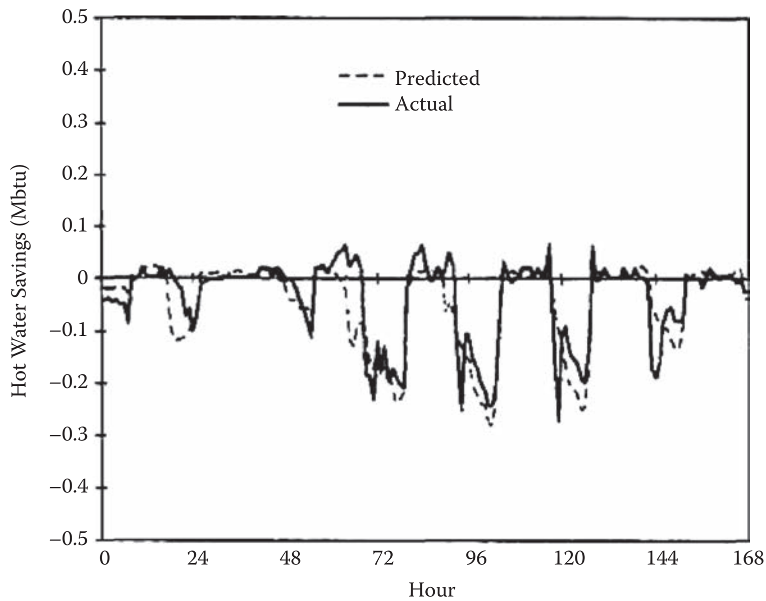


FIGURE 16.6 Energy savings estimation from a NN-based approach: (a) chilled water savings due to lighting retrofit; (b) hot water savings due to CV to VAV retrofit: (c) whole building electrical energy savings due to cooling plant retrofit. (Courtesy of Krarti M., Kreider J.F., Cohen D., and Curtiss P. Estimation of energy savings for building retrofits using neural networks, *ASME Journal of Solar Energy Engineering*, 120(3), 211, 1998.)

(c)

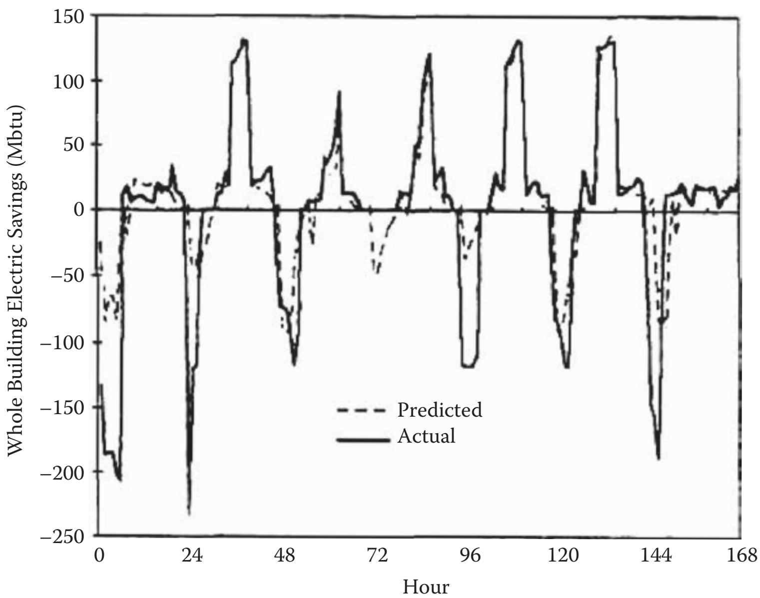


FIGURE 16.6 (Continued) Energy savings estimation from a NN-based approach: (a) chilled water savings due to lighting retrofit; (b) hot water savings due to CV to VAV retrofit: (c) whole building electrical energy savings due to cooling plant retrofit. (Courtesy of Krarti M., Kreider J.F., Cohen D., and Curtiss P. Estimation of energy savings for building retrofits using neural networks, *ASME Journal of Solar Energy Engineering*, 120(3), 211, 1998.)

week. For the VAV retrofit, Figure 16.6(b) indicates that the NN model determines the hot water savings fairly well. Finally, for the ASD chiller drive/thermal energy storage (TES) retrofit, Figure 16.6(c) shows that the daytime peaks are well predicted by the NN model, but some of the evening charging peaks are not.

Currently, the NN-based approach to estimate savings has been applied to only a few building types. Generalization from one building to other similar buildings in other climates has not been totally successful (Krarti et al., 1998). Only NNs trained on the specific building to be retrofitted are suggested at this time.

16.3.4 Computer Simulation Models

Detailed computer simulation programs can be used to develop baseline models for building energy use. A brief discussion of the existing computer programs that can be applied to building energy simulation is provided in Chapter 4. The main feature specific to the application of computer simulation models to estimate retrofit energy savings is the calibration procedure to match the baseline model results with measured data (Yoon, Lee, and Claridge, 2003; Zhu, 2006). Typically, the calibration procedure of a computer simulation model can be time-consuming and require significant efforts especially when daily or hourly measured data is used. Unfortunately, there is no consensus on a general calibration procedure that can be considered for any building type. To date, calibration of building simulation programs is rather an art form that relies on user knowledge and expertise. However, several authors have proposed graphical and statistical methods to aid in the calibration process and specifically to compare the simulated results with measured data. These comparative methods can then be used to adjust certain input values either manually or automatically until the simulated results match the measured data according to predefined accuracy level. Figure 16.7 illustrates a typical calibration procedure for building energy simulation models.

Calibration of computer simulation models is generally time-consuming especially for hourly predictions. Two methods have been used to compare the simulated results to the measured data and help reduce the efforts involved in the calibration procedure:

1. Graphical techniques to efficiently view and compare the simulated results and the measured data. Recently, several graphical packages have been utilized or developed for energy computer

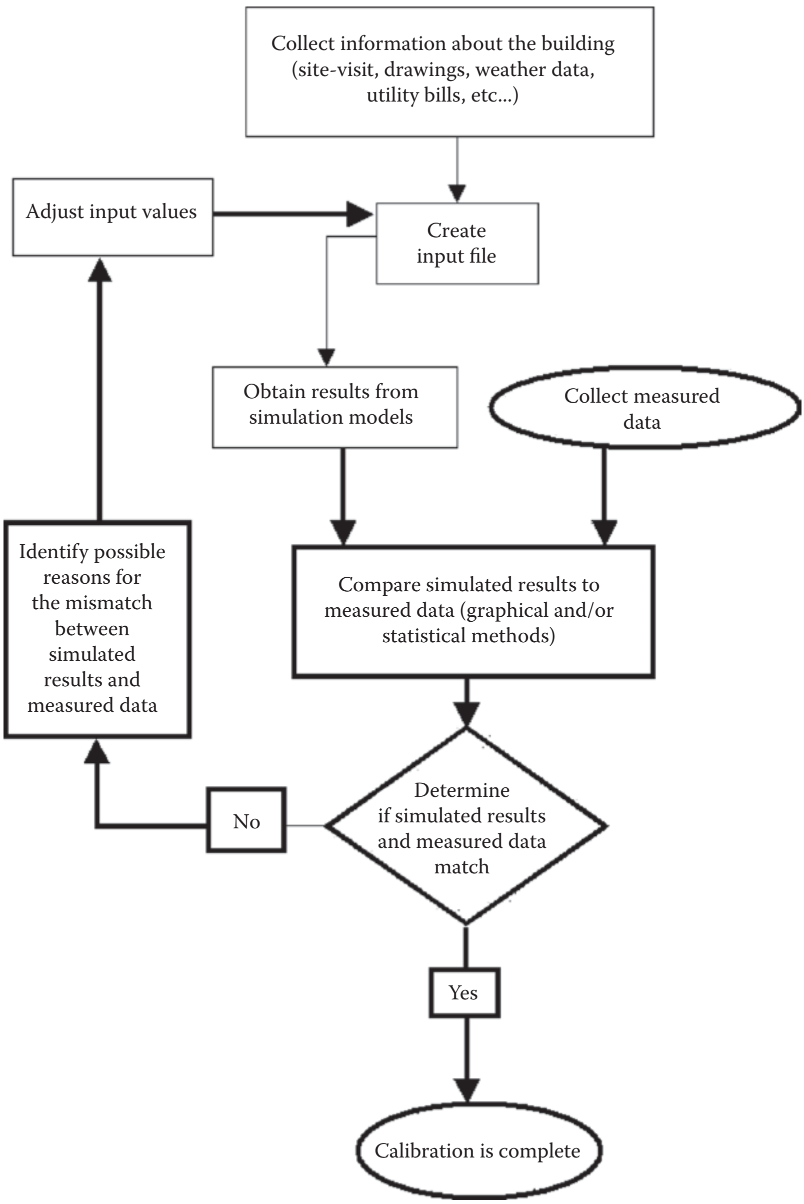


FIGURE 16.7 Typical calibration procedure for building energy simulation models. (Note that the steps that are specific to the calibration process are highlighted in bold.)

simulation calibration including: 2-D time-series plots and x–y plots (for monthly and daily data), 3-D surface plots (for hourly data), BWM or box-whisker-mean plots (for weekly data). Haberl and Abbas (1998a,b) and Haberl and Bou-Saada (1998) provide a detailed discussion of the most commonly used graphical tools to calibrate energy computer simulation models. Figure 16.8 illustrates 3-D surface plots for hourly energy use before, during, and after an energy retrofit project.
2. Statistical indicators to evaluate the goodness-of-fit of the simulated results compared to the measured data. Several indicators have been considered including mean difference, mean bias error

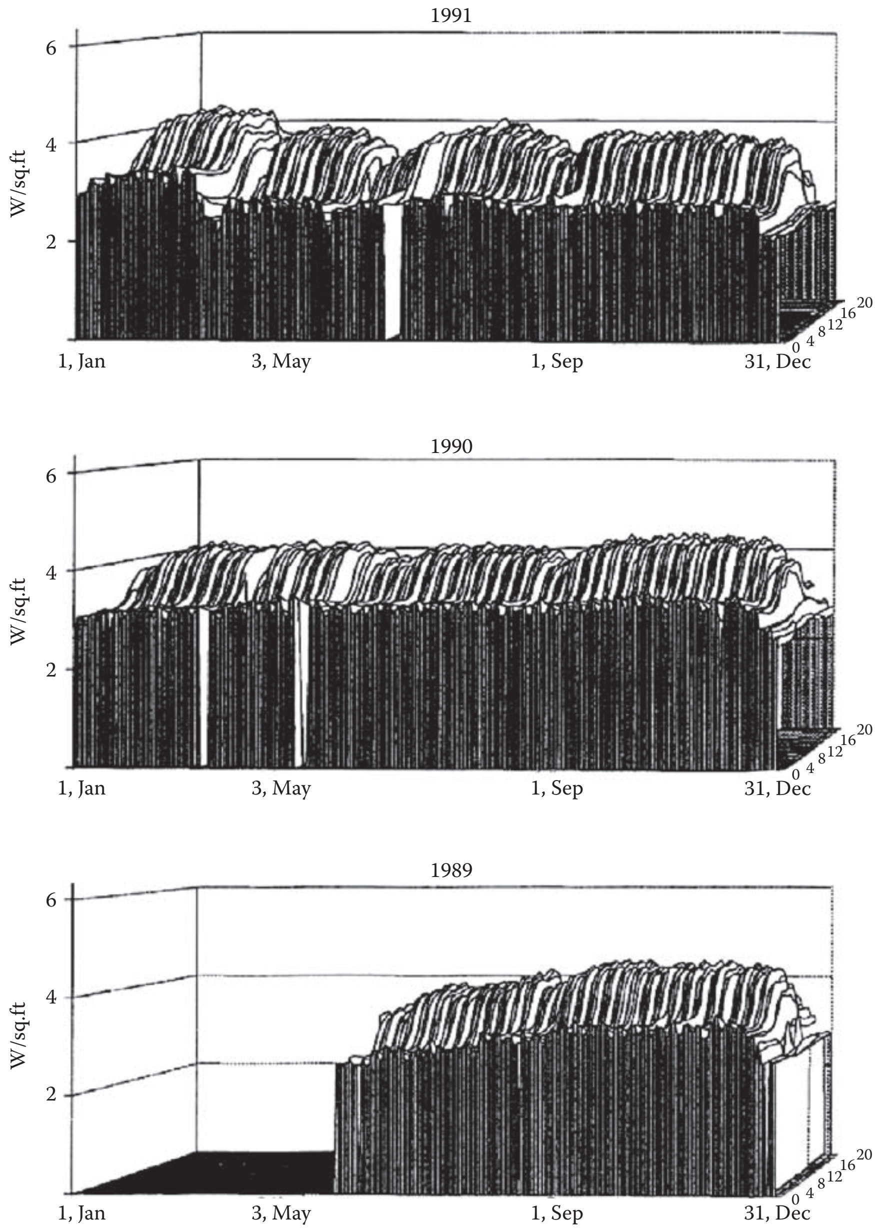


FIGURE 16.8 Typical 3-D graphs to show the hourly simulation results before and after a retrofit measure. (Courtesy of Haberl, J.S. and Abbas, M. Development of graphical indices for viewing building energy data: Part II, *ASME Solar Energy Engineering Journal*, 120(3), 162, 1998b.)

(MBE), root mean square error (RMSE), and coefficient of variance (CV). A discussion of these indicators can be found in Haberl and Bou-Saada (1998). To obtain these indicators, classical statistical tools can be used such as SAS (1989).

ASHRAE Guideline 14 (ASHRAE, 2002) provides procedures for the reliable measurement and verification of energy savings from any implemented ECMs. The procedures outlined in the guideline are applicable to commercial, residential, and industrial applications to determine savings with reasonable accuracy. Savings are determined through comparison of pre- and postretrofit data with modeled or predicted data. Guideline 14 provides an outline calculation of energy savings procedures for three different methods. The first method defined by Guideline 14 is the whole building approach which uses whole-building metered energy consumption data normalized for external factors such as weather for the pre- and postretrofit period to determine savings. The second method is the retrofit isolation approach which uses measured end-use consumption data for pre- and postretrofit periods to determine savings. The third method defined by Guideline 14 is the whole-building calibrated simulation approach which defines procedures for creating calibrated building energy simulation models for the determination of savings.

ASHRAE Guideline 14 (2002) recommends a set of statistical indicators as calibration tolerances including monthly errors, annual error, and coefficient of variation of the root mean square of monthly errors. The error between the simulated energy consumption and the measured energy consumption (utility data) is analyzed on a monthly basis:

$$ERR_{month}(\%) = \left[\frac{(M-S)_{month}}{M_{month}} \right] \times 100 \tag{16.5}$$

where the variable M is the measured energy consumption for the month in kWh or Btu, and S is the predicted energy consumption obtained from the simulation model. The annual error is defined as:

$$ERR_{year}(\%) = \sum_{month} \left[\frac{ERR_{month}}{N_{month}} \right] \tag{16.6}$$

As indicated by Eq. (16.6), the annual error is an average of the monthly error over the analysis year and may not accurately represent the true error due to sign value differences; therefore it is typically useful to calculate and provide the coefficient of variation. The coefficient of variation is found by first calculating the root mean square monthly error using the following equation:

$$RSME = \sqrt{\frac{\sum_{month} (M-S)^2}{N_{month}}} \tag{16.7}$$

The coefficient of variation can be found using Eq. (16.8) where A_{month} is the average measured monthly consumption.

$$CV = \frac{RSME}{A_{month}} \tag{16.8}$$

ASHRAE Guideline 14 and FEMP measurement and verification guidelines can be used to determine the acceptance criteria for calibrating the simulation models as outlined in Table 16.4 (ASHRAE, 2002; FEMP, 2000).

16.4 Applications

The analysis methods presented in this chapter are used to measure and verify savings from energy retrofit projects. These methods are recommended as part of the International Performance for Monitoring and Verification Protocol (IPMVP, 2002, 2007) which identifies four different options for energy savings estimation procedures depending on the type of measure, accuracy required, and the cost involved:

- *Option A* uses simplified calibration methods to estimate stipulated savings from specific energy end-uses (such as lighting and constant speed motors). This option may use spot or short-term measurements to verify specific parameters such as the electricity demand when lighting is in use. Common applications of this option involve lighting retrofits (such as replacement with more energy-efficient lighting systems and use of lighting controls).
- *Option B* typically involves long-term monitoring in an attempt to estimate the energy savings from measured data (rather than stipulated energy consumption). Simplified estimation methods can be used to determine the savings from the monitored data. The energy savings obtained from motor replacements (i.e., use of more energy-efficient motor or installation of variable speed drives) are commonly estimated using option B.
- *Option C* is generally applied to estimate savings by monitoring whole-building energy use and by using regression analysis to establish baseline models. The regression models can be developed from monthly or daily measured data. Examples of energy retrofit projects for which the Option C procedure can be used include conversion of a constant-volume system to a variable-air-volume system and chiller or boiler replacement.
- *Option D* is typically used when hourly savings need to be estimated. In addition to hourly monitoring, this option requires complex data analysis to establish baseline models. Dynamic models and calibrated simulation models are typically used to estimate hourly energy savings. The impact of energy management control systems (EMCS) is one of several applications for which option D can be considered.

TABLE 16.3 Basic Characteristics of the Various Options for Procedures to Estimate Energy Savings

Option	Models for Data Analysis	Accuracy (%)	Cost (%)	Metering Requirements	Applications		
					End-Use	Load	Operation
A	Simplified methods	20–100	1–5	Spot/short-term	Subsystem	Constant	Constant
B	Simplified methods	10–20	3–10	Long-term	Subsystem	Constant/ variable	Constant/ variable
C	Regression analysis	5–10	1–10	Long-term (daily or monthly)	Building	Variable	Variable
D	Dynamic analysis/ simulation	10–20	3–10	Long-term (hourly)	Subsystem/ building	Variable	Variable

TABLE 16.4 Acceptance Criteria

Criteria	FEMP	ASHRAE 14
ERR_{Month}	(±)15%	(±)5%
ERR_{Year}	(±)10%	-
$CV(RMSE_{Month})$	(±)10%	(±)5–15%

Table 16.3 summarizes the requirements of each option including typical accuracy levels of savings estimation and the average cost (expressed as a percentage of the total cost required to implement the energy retrofit project) needed for metering equipment and for data analysis.

16.5 Uncertainty Analysis

ASHRAE Guideline 14 (2002) provides a method to estimate the uncertainty in determining the energy savings from retrofit projects. The uncertainty in the determined savings is a function of the CVRSME of the baseline model, corresponding *t*-statistic, the fractional savings, and number of pre- and postretrofit months. The equation for the fractional uncertainty in savings measurements in either utility analysis or a computer simulation model is as follows:

$$\frac{\Delta E_{save,m}}{E_{save,m}} = t \cdot \frac{1.26 \cdot CVRSME \left[\left(1 + \frac{2}{n}\right) \frac{1}{m} \right]^{1/2}}{F} \tag{16.9}$$

where

$$RMSE = \left[\sum_{i=1}^{n} \left(E_i - \hat{E}_i\right)^2_{pre} / (n - p) \right]^{1/2} \tag{16.10}$$

t = Student's t-statistic at selected confidence interval.
n = Number of preretrofit months used in model.
m = Number of postretrofit months used in savings analysis.
F = Percent savings.
p = Number of model parameters, (p = 3 for utility analysis, p = 1 for calibrated simulation).

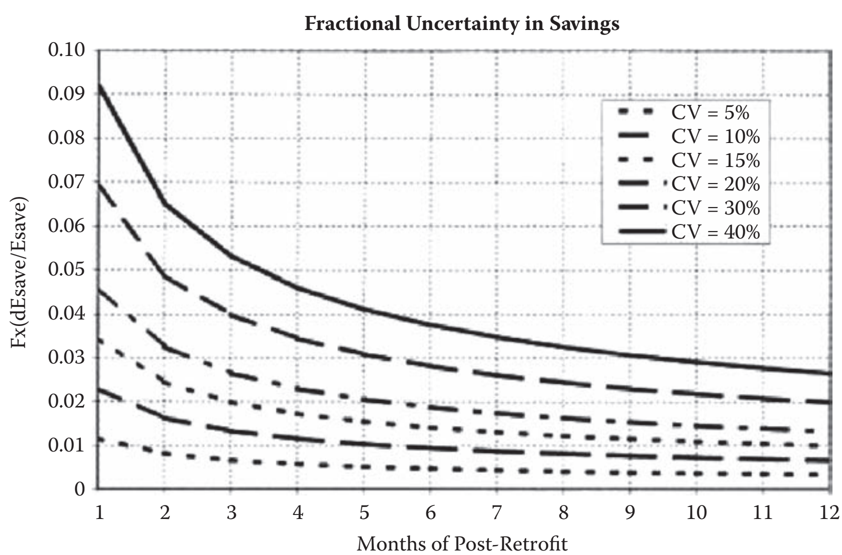


FIGURE 16.9 Fractional uncertainty in savings for varying CVs.

This method of uncertainty analysis assumes zero measurement error, which holds true when using monthly utility data. The fractional uncertainty in the savings formula taken from Guideline 14 is shown graphically in Figure 16.9 for varying coefficients of variation, 12 months of preretrofit data, and a 68 percent confidence interval.

Figure 16.9 indicates that fractional uncertainty in predicted energy or cost savings decreases with the availability of more postretrofit data and better baseline models [i.e., lower CV values as estimated using Eq. (16.8)]. The fractional uncertainty in predicted savings is also dependent on the estimated fraction of savings. For example, a model with a CV of 15 percent, an expected savings fraction of 15 percent, and 3 months of postretrofit data has a y-ordinate value of .02 from the chart and yields a fractional uncertainty in savings of (.02/.15) * 100 = 13 percent, whereas the same model with an expected savings fraction of 40 percent would yield a fractional uncertainty in predicted savings of (.02/.4) * 100 = 5 percent.

16.6 Summary

In this chapter, methods used to estimate energy savings from retrofit projects are briefly presented. The common applications of the presented methods are discussed with some indication of their typical expected accuracy and additional cost. As mentioned throughout the chapter, the presented measurement and verification protocols are applicable to estimate energy savings measured from energy conservation measures applied to existing commercial buildings. However, similar procedures have been developed for other applications such as energy efficiency in new buildings, water efficiency, renewable technology, emissions trading, and indoor air quality.

17

Case Studies

17.1 Reporting Guidelines

In order to summarize the findings of various tasks carried out in an energy audit, it is important to submit a well-documented report. The report should describe in detail the tasks completed and the analysis considered for the energy audit. The level of detail in the report depends on the type of energy audit completed. Chapter 1 defines and discusses the various energy audit types. In the following sections, a set of guidelines is provided to help the auditor write the final report after completing any energy audit type for an existing building.

17.1.1 Reporting a Walk-Through Audit

The walk-through audit can be a standalone task or one part of a standard energy audit. Typically, this type of audit is sufficient for small buildings with simple energy systems including residential buildings and low-rise commercial buildings. The basic tasks to be carried out for a walk-through audit include:

Task 1: Describe the basic energy systems of the building including building envelope, mechanical systems, and electrical systems. Observations from the walk-through as well as specifications from architectural, mechanical, and electrical drawings can be utilized to describe the building features.

Task 2: Perform basic testing and measurements to assess the performance of various energy systems. These measurements may depend on the type of building and its systems as well as on time available for the auditor. For residential buildings, it is highly recommended to perform a pressurization or a depressurization testing using the blower door test kit as outlined in Chapter 6. In all building types, spot measurement, and, if possible, monitoring indoor air temperature and relative humidity within the space for at least one day, is helpful to estimate the indoor temperature settings and identify or check any comfort issues.

Task 3: Meet and discuss with building occupants or building operators to identify any potential discomfort problems and sources of energy waste within the building. This task is often helpful to define potential operation and maintenance measures as well as energy conservation measures,

Task 4: Identify some potential operation and maintenance (O&M) measures and energy conservation measures (ECMs) as well as any measures required to improve comfort problems. Provide the implementation details and the cost of implementation (try to seek direct quotations from local contractors/stores).

Task 5: Evaluate the energy savings (or requirements if measures are needed to improve comfort) using simplified analysis methods presented in this book. Compare the results between the two approaches and comment on the accuracy of both approaches.

Task 6: Perform cost analyses based on the simple payback period method to determine the cost-effectiveness of the identified O&Ms and ECMs. You should make appropriate assumptions and, if necessary, estimate the energy cost savings. Provide recommendations based on the economic analyses. Cost data should be taken from actual estimations from contractors. This cost data will be provided.

The report of a walk-through energy audit can be brief and should include at least the basic recommendations for cost-effective O&Ms and ECMs, that is, the results of Task 6 described above. However, it is strongly recommended that a more detailed report be drafted and delivered to the client to document the findings and observations obtained from the completed tasks. In particular, the report should describe the basic features of the audited building as well as any potential problem areas identified during the walk-through. Moreover, the calculations to estimate energy use and cost savings should be presented for the recommended energy conservation measures. In addition, references and specifications for implementing the recommended O&Ms and ECMs should be provided. A final report for a walk-through energy audit can include the following sections:

(i) Legible and complete drawings showing the floor plan and at least two elevation views.
(ii) Brief description of the architectural features of the building (construction type, orientation, solar systems, etc.).
(iii) An analysis of the utility bills to estimate energy use intensity, the building BLC, the balance temperature, and base-loads. It is useful to perform this task before visiting the building.
(iv) Description of any testing or measurements carried out during the walk-through audit including temperature and air leakage. For air leakage testing, provide all the relevant details of your testing and calculation analysis including any assumptions. Ensure that you provide the air leakage area as well as the infiltration rates (in ACH) under reference conditions (i.e., $\Delta P = 4$ Pa) and for average weather conditions (annual average and heating season average).
(v) Discussion of the walk-through audit tasks and the outcome. In particular, highlight any occupant concerns and complaints and any identified potential O&Ms and ECMs.
(vi) Describe the calculation details to estimate energy use and cost savings for the considered O&Ms and ECMs. References for these calculations should be provided in the report including any assumptions that were made to carry out the estimations.
(vii) Discuss the results of the economic analysis. In particular, provide the general procedure and the cost of implementing each ECM.
(viii) Provide specific recommendations to the client to reduce the utility bills or to improve the indoor environment within the building.
(ix) Take some photos to highlight some of the features and the problem areas of the house.

Case Study 1 provides an example of a report for a walk-through energy audit performed for a single-family house.

17.1.2 Reporting a Standard Audit

The report of a standard audit is more comprehensive than a report for the walk-through audit outlined above. Indeed, a standard audit, as defined in Chapter 1, includes additional tasks and requires more effort and time to complete. This type of audit is typically suitable for large buildings such as those with complex energy systems. Moreover, the utility bills for large buildings such as commercial and institutional are significantly high and can justify the level of detail required by a standard audit. In addition to the tasks described for the walk-through audit, the following tasks can be carried out as part of a standard audit:

Task 1: Carry out a detailed survey of lighting and electrical equipment. The main goal of this task is to estimate the lighting and equipment power densities within various spaces of the building.

Task 2: Identify heating, ventilating, and air-conditioning (HVAC) systems and their operation schedules. This task is often crucial because energy used by HVAC systems is a significant portion of the total energy consumed in large buildings.

Task 3: Determine the main discomfort and complaints of occupants through a well-designed questionnaire. Surveying occupants very often provides valuable information about the performance of the building and its energy systems throughout the year.

Task 4: Collect and analyze utility data for at least three years. Utility data for only one year is often insuà cient to estimate the historical energy performance of a building. In some cases, certain conditions such as special events or extreme weather may create biases in the energy use of the building.

Task 5: Perform any relevant measurements such as lighting levels, IR photos, indoor temperatures, airflow rates supplied by air-handling units, and electrical energy end-uses as well as indicators of electrical power quality.

Task 6: Model the existing building using a detailed energy simulation tool. Ensure that the simulation model is well calibrated using utility data. Typically, monthly calibration within 10 percent is required to increase the confidence level in the predictions of the building energy simulation model

Task 7: Carry out calculations to estimate energy savings from potential energy conservation measures using both the calibrated energy simulation model and simplified calculation procedures outlined in this book

Task 8: Perform an economic analysis using simple payback, net present worth, or life-cycle cost (LCC) analysis methods for all the energy conservation measures. The implementation details and costs should be provided for each measure.

Task 9: Select the energy conservation measures to be recommended for implementation. In addition, specify the additional benefits of each measure (such as improving thermal or visual comfort), the implementation costs, and any information to help the client implement these measures.

The report of a standard energy audit should summarize the results of all the completed tasks. A recommended outline for the standard energy audit report is provided below. It should be noted that the same outline can be used to report the findings for a detailed energy audit.

(i) Legible and complete drawings showing the floor plan and elevation views of the building.
(ii) Brief description of the features of the building and its systems including building envelope, lighting systems, office equipment, appliances, HVAC systems, and heating and cooling plant.
(iii) Summary of the walk-through audit findings and results from any testing and measurements. Typically for a standard audit, monitoring of indoor temperature and relative humidity is carried out for at least one week.
(iv) Summary of the survey results for lighting, electrical, and HVAC systems as well as the results of the questionnaire of occupants.
(v) Discussion of the results of the utility data analysis. The energy use intensity of the building can be compared to some available and established benchmarks.
(vi) Basic assumptions made to model the building using a detailed simulation tool (such as eQUEST, VisualDOE, and EnergyPlus).
(vii) Description of the calibration process including discussion of the parameters that were utilized to match simulation tool predictions with utility data.

(viii) Discussion of the implementation details for the various ECMs that are considered for the building. The assumptions made to estimate the energy use and cost savings should be provided.
(ix) Summary of the economic analysis using simple payback and LCC methods. Specific quotes from local contractors should be used in the economic analysis. However, established references such as Means (2008) can also be considered for cost estimates.
(x) List the implementation priority for the recommended ECMs.

Case Studies 2 and 3 illustrate the analysis procedures and reporting guidelines for standard audits performed for, respectively, a single-family house and a museum building.

17.2 Case Study 1: Walk-Through Audit of a Residence

In this section, an example of a residential building walk-through is presented. As outlined in Chapter 1, a walk-through audit allows the collection of basic information about the building envelope (windows, walls, and doors) as well as the lighting fixtures, appliances, and HVAC equipment. During a walk-through audit, the auditor should question the building owners and occupants to determine any problematic areas of the building related to thermal comfort and energy performance. The main purpose of a walk-through audit is to provide recommendations for improving the energy efficiency of the residence by investigating selected operating and maintenance measures and energy efficiency measures (EEMs) with short payback periods. A report of a walk-through audit is provided in the following section.

17.2.1 Building Description

Table 17.1 provides basic features of an audited house located in Colorado. The main concern communicated by the occupants was the draftiness of the house, especially the downstairs area. Another concern was energy required to heat the hot tub, which is used from October through May.

The following sections describe the basic findings of the walk-through audit:

17.2.1.1 Building Envelope

By inspecting the building envelope, the walls of the upper level of the house were found to have some insulation including R-11 batt insulation. However, with the downstairs exterior walls, no insulation was found (the original wall consists of logs plus drywall). The roof is built from a standard woodframe construction with an aluminum exterior finish. The roof appears to be in good condition. The insulation level is adequate. The homeowner has indicated that R-38 insulation was added to the ceiling below the roof during a recent renovation of the house. Most of the windows are double paned and in fairly good condition. However, some window frames were found to be leaky and could be better sealed.

17.2.1.2 Building Infiltration

A blower door test was performed using the pressurization technique. It was found that the rate of infiltration was 0.89 air changes per hour (i.e., 0.89 ACH). A typical house has a rate of infiltration in ACH

TABLE 17.1 Basic Features of the Audited House

Age of building	70 years (upgrades made over the past 30 years)
Square footage	2,200 ft^2
Number of floors	2 floors mostly above grade
Basement or crawlspace	Some cellar/storage space
Method of construction	Original log construction downstairs; wood siding upstairs
Utilities	Propane and electricity
Condition of house	Fair–Good
Additional notes	Mountain property

of 0.4–0.75. An infiltration rate higher than 0.75 ACH indicates a “leaky” house. In the audited house, air infiltration is occurring through visible gaps in the window frames, gaps between the walls and the ceiling/floor, through piping/plumbing fixtures, and through the fireplace doors.

17.2.1.3 HVAC System

A heating and cooling system consumes about 55 percent of a typical U.S. residence’s energy use. The audited house has a unique HVAC system: a combination of passive solar heating, electric baseboard heaters, heat fans, a unit propane heater in the living area, and a furnace in the cellar area. There is no central AC unit in the residence.

17.2.1.4 Water Management

The domestic hot water tank was observed to be uninsulated, and the plumbing fixtures to be conventional without low-flow devices.

17.2.1.5 Appliances

In the audited house, it was found that the appliances consuming the most energy were two refrigerators. Both refrigerators were fairly old models and could be replaced with new, energy-efficient models. Similarly, the clothes washer is rather old and could be replaced with a more efficient model.

17.2.1.6 Thermal Comfort

The greatest concern outlined by the homeowner during the walk-through audit regarding thermal comfort is that the downstairs area is drafty, especially the north television/guest room and bathroom. Due to the more recent renovations, the upstairs is less drafty and more comfortable.

17.2.2 Energy Efficiency Measures

This section describes the potential operation and maintenance and the energy efficiency measures applicable to this house to reduce its energy use and cost. Eight measures have been identified relating to the building envelope, appliances, water management, thermal comfort, and solar water heating for the hot tub. The following sections briefly describe the rationale for each EEM, the general plan for implementation, the estimated cost of installation (including labor and materials), and the payback period for its implementation.

17.2.2.1 Building Envelope

A typical house has a rate of infiltration in ACH of 0.4–0.75. An infiltration rate higher than 0.75 ACH indicates a rather leaky house. This is the case for the audited house with a 0.89 ACH air infiltration rate.

17.2.2.1.1 EEM-1: Weatherization

For the audited house, basic weatherization measures such as applying weatherstripping to unsealed windows, and caulking gaps between the ceiling and floor can save energy and improve thermal comfort. Several studies for similar houses have indicated that a 20–50 percent reduction in air infiltration can be achieved after weatherization of residential buildings. For this house, a 33 percent (i.e., 1/3) reduction in air infiltration through the building envelope is assumed due to caulking and weatherstripping of window frames and doors. The expected cost of basic weatherization for the house (including labor) is about $450. Using the degree-day method outlined in Chapter 6, the projected annual energy savings is $215 attributed to reduced heating energy cost resulting in a payback period of 2.2 years. As a note, one easy recommendation is to apply magnetic tape around the fireplace to seal the fireplace doors. Simple black magnetic tape (available at most hardware stores) is an easy do-it-yourself fix to reduce infiltration.

17.2.2.2 Water Management

Domestic hot water (DHW) accounts for 15–25 percent of the energy consumed in a typical U.S. residence. However, the actual DHW load can vary significantly depending on usage. Whether the water heater is gas or electric, significant energy savings can often be found through implementation of water heating efficiency improvements. In general, the four measures to reduce water heating costs are:

1. Use less hot water.
2. Turn down thermostat on hot water tank.
3. Insulate hot water tank.
4. Replace existing water heater with a more efficient model.
5. Use low-flow plumbing fixtures.

17.2.2.2.1 EEM-2: Hot Water Tank Insulation

For the audited house, the propane-fired domestic hot water tank is uninsulated. Even though the tank sits in a conditioned area of the house (the kitchen), it is still losing heat to the surroundings. The tank surface temperature is measured to be 74°F for a surrounding air temperature of 65°F. This nine-degree difference indicates that heat is being lost from the tank. Wrapping the tank with R-12 insulation could save $111 annually. Tank insulation kits can be purchased from any hardware store and cost about $30–$60. Assuming the cost of implementing this measure is $50, the resulting payback period is 0.4 years.

17.2.2.2.2 EEM-3: Low-Flow Showerheads

Because propane is expensive relative to natural gas, water conservation measures such as low-flow fixtures also save energy. Assuming that the occupants in the audited house take two showers per day and that a typical shower lasts eight minutes, there would be a saving of about $175 in energy costs per year by replacing the conventional showerheads with low-flow fixtures. Moreover, the homeowner would also conserve about 12,000 gallons of water per year.

Low-flow showerheads cost between $25 and $50 and can be purchased at most hardware stores. For an installation cost of $100 for this measure (two showerheads), the payback period is less than one year.

17.2.2.2.3 EEM-4: Low-Flow Faucets

In the audited house, two faucets (4 gpm) can be replaced with low-flow fixtures (1.5 gpm) at a cost of $25 per faucet. Assuming each faucet is used for three minutes per day at an average temperature of 80°F, 5,370 gallons of water and $53 of energy cost per year can be saved. The initial investment of about $50 results in a payback period of one year.

17.2.2.3 Appliances

Appliances and electronics can be responsible for as much as 20 percent of a U.S. residence's energy bills. Refrigerators, clothes washers, and clothes dryers are often the greatest energy consumers. Substantial energy use savings can be achieved by altering the usage patterns and by investing in more efficient, ENERGY STAR® appliances.

17.2.2.3.1 EEM-5: Cold Water Wash

Washing clothes in cold water is an easy way to save energy. Because hot water is not used, no energy is needed to heat the water, thus the energy savings. Based on the occupancy level of the audited house, the estimated laundry load is three washes per week. By washing all three loads in cold water a saving in energy cost of $95 per year can be obtained. There is no cost required to implement this ECM, resulting in an immediate payback period.

17.2.2.3.2 EEM-6: Phantom Loading

Several plugged appliances are utilized in the audited house. Appliances such as computers, televisions and coffeemakers draw current even when turned off. These "phantom loads" can be eliminated by plugging these appliances into power strips. Using only five power strips around the house to reduce phantom loading can save about $70 per year. With an initial investment of $50 ($10/power strip), the payback period is about 0.7 years.

For this EEM to be effective, at least three appliances should be plugged into each strip, and the strip should be turned off for at least 12 hours per day.

17.2.2.3.3 EEM7: ENERGY STAR Washing Machine

ENERGY STAR washing machines use less water for each wash cycle, which saves both water and energy. The homeowner can save about $70 in annual energy use by replacing a conventional washing machine with an ENERGY STAR model. With an initial investment of $600, the payback period is about 8.7 years.

17.2.2.3.4 EEM8: ENERGY STAR Refrigerator

In most homes, the refrigerator is the most energy-consuming appliance. ENERGY STAR qualified refrigerators use about half as much energy as models manufactured before 1993! Replacing the two refrigerators used in the audited house with new ENERGY STAR models could reduce the refrigerator's energy consumption by 50 percent, saving about $131 annually. With an initial investment of $1400 (for two new refrigerators), the payback period is about 10.7 years.

17.2.3 Economic Analysis

Table 17.2 recaps the proposed EEMs, outlines the projected cost of installation and annual energy savings and then ranks each measure in terms of its simple payback period (initial investment divided by annual savings).

17.2.4 Recommendations

The total cost of implementing the eight recommended EEMs outlined in Table 2 is $2,920 and would yield an energy savings of $900. This number may vary depending on the actual behavioral patterns in the house (e.g., the occupants may take fewer than two eight-minute showers per day, in which case the energy savings realized by installing low-flow shower fixtures would be less).

EEMs with a payback of less than four years in addition to the weatherization EEM (to improve thermal comfort) would cost about $720 to implement. With an annual energy savings of $700, the payback period of these EEMs is a little more than one year. The recommended EEMs would improve thermal comfort while saving energy in the home.

17.3 Case Study 2: Standard Audit of a Residence

In this section, a standard energy audit of a residential building is presented. As indicated in Chapter 1, the standard energy audit includes a walk-through energy audit, utility data analysis, energy modeling, and economic analysis to recommend cost-effective energy efficiency measures. The report outlining the findings of a standard audit for a house is provided in the following sections.

TABLE 17.2 Economic Analysis of Recommended EEMs

EEM Descriptions	Cost of Measure ($/total cost)	Annual Savings ($/year)	Simple Payback (years)
EEM5: Cold Water Wash: Wash clothes in cold water instead of hot	0	$95	Immediate
EEM2: Hot Water Tank Insulation: Wrap DHW tank with R-12 insulation	50	111	0.4
EEM3: Low-Flow Showerheads: Replace the existing showerheads with low-flow fixtures	100	175	0.6
EEM6: Phantom Loading: Eliminate phantom loads by plugging appliances into power strips	70	50	0.7
EEM4: Low-Flow Faucets: Replace the existing faucets with low-flow fixtures	50	53	1.0
EEM1: Weatherization: Reduce infiltration by caulking and sealing drafty windows and doors	450	215	2.2
EEM7: ENERGY STAR® Washing Machine: Save water and energy by replacing conventional machine with energy-efficient model	600	70	8.7
EEM8: ENERGY STAR Refrigerator: Save electricity by replacing conventional machine with energy-efficient model	1,400	131	10.7
TOTAL	**2,920**	**900**	**3.2 years**

FIGURE 17.1 Audited house for Case Study No. 2.

17.3.1 Architectural Characteristics

The house audited in this report is a 1950s split-level design with 8-in. brick walls and is located in Denver, Colorado, as shown in Figure 17.1. The interior is finished with plaster, the ceiling consists of gypsum with 8 in. of blown-in insulation in the attic, and windows and doors are the only types of fenestrations. The windows were replaced two years ago with new double-paned models. The house has two roofs that have been modeled as one, while still accounting for the volume of the two halves of the house. The floor is concrete slab covered with carpet and hardwood floors. The floor plan and elevation views of the house can be found in the appendix.

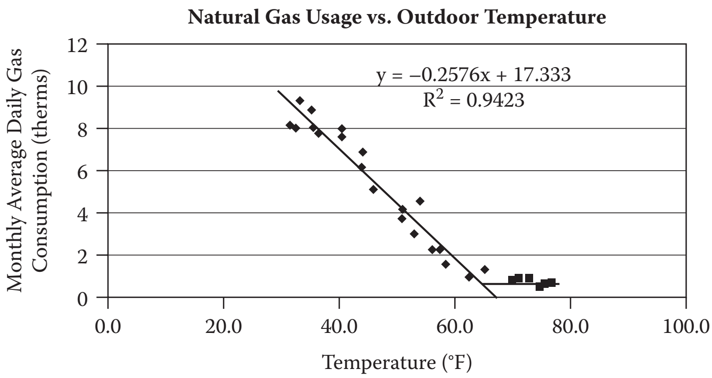


FIGURE 17.2 Natural gas consumption as a function of average monthly outdoor temperature.

17.3.2 Utility Analysis

One of the first steps in evaluating a building is to analyze its utility bills as indicated in Chapter 6. In particular, the average daily fuel use (over a month) is plotted against the average ambient temperature in that month to assess the energy efficiency of the building envelope, by calculating the building load coefficient (BLC) as well as the balance temperature and the base-load. One method used to determine the BLC is to examine the latter plot mentioned above, which is shown in Figure 17.2.

The operating BLC_{UTILITY} that was determined by Eq. (17.1) is:

$$BLC_{UTILITY} = \frac{-S \cdot \eta}{24} \tag{17.1}$$

For this calculation, S is the slope (therms/average monthly temperature) of the linear regression shown in Figure 17.2, η is the efficiency of the heating system (0.80), and 24 is used because the house is heated 24 hours per day during the winter. The slope was determined by using a linear curve fit on the plot of natural gas consumption versus outdoor temperature; see Figure 17.2. The BLC was calculated using Eq. (17.1) to be 773 (Btu/hr-F).

The base-load, or fuel used to heat the domestic hot water, is determined by fitting a line to the bottom series of points and finding where it crosses the y-axis. The base-load for this house is approximately 0.96 therms per day and remains relatively constant throughout the year.

The point where these two linear regressions cross is known as the balance temperature. This represents the outdoor temperature at which no additional heating is required to maintain the indoor temperature. For this house the balance temperature is approximately 65°F, which as a rule of thumb means the thermal performance of the envelope should be improved. This performance is affected by both infiltration and exterior insulation.

There is also a method of calculating the BLC, known as the BLC_{CALC} based on the envelope thermal resistance levels and air infiltration rate. The BLC_{CALC} can be calculated using Eq. (17.2):

$$BLC_{CALC} = \sum (U \cdot A) + (\dot{m}c_P)_{INF} \tag{17.2}$$

where U is the thermal conductivity (Btu/hr/°F/ft^2) of each external surface, A is the corresponding surface area (ft^2), $\dot{m}$ is the mass flow rate due to air infiltration (lbm/hr), and c_P is the specific heat of air (Btu/lbm°F). Calculations based on the brick walls, double-paned glazing, R-19 insulation in the attic, and no floor insulation, determined a (UA) value of 675. When combined with the infiltrations $(\dot{m}c_P)_{INF}$

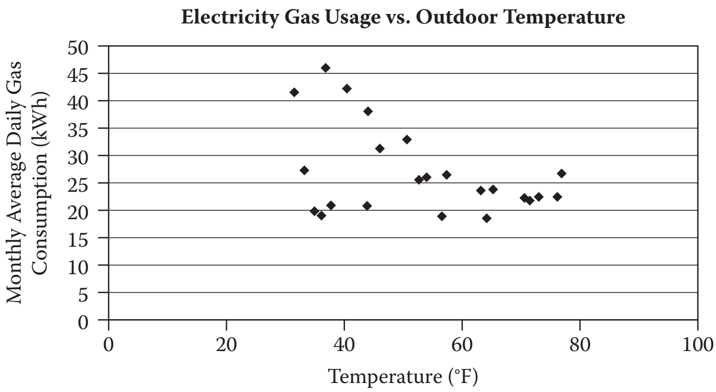


FIGURE 17.3 Electricity consumption as a function of outdoor temperature.

of 115.8 Btu/hr°F this residence has a calculated BLC of 791 Btu/hr°F. The modeled BLC_{CALC} was compared against the BLC_{UTILITY} of 773 Btu/hr°F to show good congruence. The BLC value obtained using the utility data is usually lower than the calculated BLC because of thermal coupling in the walls (i.e., infiltrating air recovers heat escaping the building envelope). This thermal coupling is described as the efficiency of thermal coupling between heat transmission and air infiltration/exfiltration within the wall, and is calculated using Eq. (17.3):

$$\eta_{wall} = \frac{BLC_{CALC} - BLC_{UTILITY}}{BLC_{CALC}} \tag{17.3}$$

The audited house had a thermal coupling wall efficiency of 2.2 percent.

The electricity use in the audited house was also analyzed by using utility data. The average daily electricity use (over a month) was plotted against the average ambient temperature in that month and is shown in Figure 17.3.

There are two features of this house that make the electrical energy consumption atypical of modern homes currently built in Colorado: no mechanical cooling, and an outdoor hot tub that uses electric resistance heat. There is no mechanical cooling provided in the house, therefore the electrical consumption does not increase in the summer with the increase in outdoor air temperature. In the winter, when several other residences are consuming at their base-load level, this house is using a large amount of electricity. Due to the colder outdoor air temperatures, greater electric resistance heat is needed to keep the hot tub warm. From the data shown in Figure 17.3, the base electric load for this home is approximately 20 kWh per day.

17.3.3 Air Leakage Testing

Air can flow in or out of the building envelope through leaks in a process called air infiltration. Air infiltration can affect the energy use, thermal comfort, and structural integrity (due to humidity transport) of a residential home. For this project, the infiltration rate is measured using a blower door test as outlined in Chapter 6.

The blower door test determines how the volumetric flow rate varies with the pressure difference between the interior of the house and outside. Before pressurizing the house, all window and exterior doors are closed, the furnace turned off, and the water heater set to the "Pilot" setting. Moreover, all interior doors are left open and any exhaust fans and the vented dryer turned off.

First, the blower fan was positioned in the front door so as to introduce air into the house and to perform the pressurization test. For the depressurization test, the fan was simply turned around to face the outdoors. Table 17.3 summarizes the measurements for the pressurization and depressurization tests of the house.

Using the data from Table 17.2, a correlation is found for $V_{inf} = C * \Delta P^n$, where C and n are correlation coefficients determined by fitting the measured data of pressure differentials and flow rates as described in Chapter 6. The correlation regressions are presented in Figures 17.4 and 17.5, respectively, for the pressurization test and for the depressurization test.

TABLE 17.3 Data from Blower Door Tests

Pressurization Test		Depressurization Test	
House Pressure (Pa)	Fan Pressure (cfm)	House Pressure (Pa)	Fan Pressure (cfm)
9	1,080	7	1,250
12	1,475	9	1,375
15	1,775	11	1,900
18	2,150	12	2,100
20	2,300	13.5	2,250
22	2,550	15	2,550

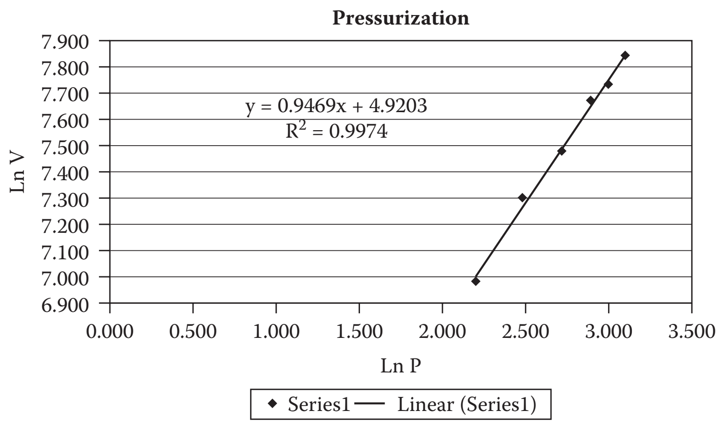


FIGURE 17.4 Correlation for pressurization test.

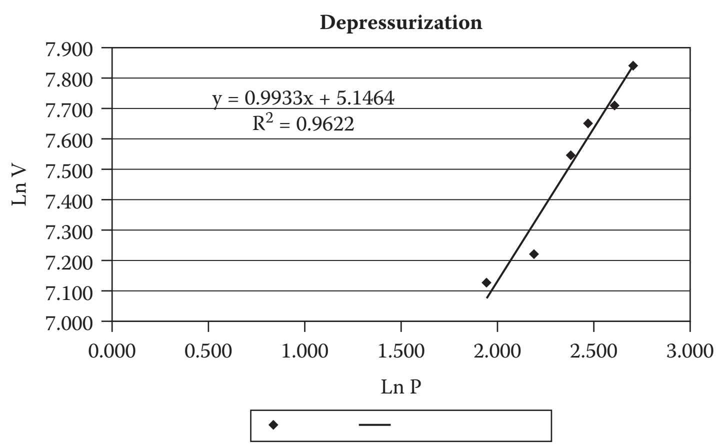


FIGURE 17.5 Correlation for depressurization test.

TABLE 17.4 Data from Blower Door Tests

	n	C	V_{ref} (cfm)	A_{leak} (in^2)
Pressurization	0.9469	137.04	509.27	132.12
Depressurization	0.9933	171.81	680.89	176.64

TABLE 17.5 Air Infiltration Rate Using LBL Model

A_{leak} (in.^2)	fs	fw	ΔT (°F)	V (mph)	V_{inf} (cfm)	V_{house} (ft^3)	ACH	Qinf (Btu/hr)
154.38	0.0156	0.0065	11.2	8.5	123.9	16,771	0.443	111.5

Note: IP units for fs: ((ft ^ 3/min) ^ 2 * in. ^ 4 * °F); IP units for fw. ((ft ^ 3/min)^2*in. ^ 4 * mph).

Using the correlation coefficients from Figures 17.4 and 17.5, the effective leakage area of the house can be estimated using Eq. (17.3) as outlined in Chapter 6:

$$A_{leak} = V_{inf} * \sqrt{\frac{\rho}{2\Delta P}} \ (in^2) \tag{17.3}$$

where $\Delta P = 4$ Pa. The correlation coefficients n and C as well as the leakage area and the reference air infiltration rate V_{inf} at $\Delta P = 4$ Pa are summarized in Table 17.4 for both the pressurization and depressurization tests.

For the subsequent energy analysis, the average of the two leakage areas is considered as the effective leakage area of the house, where

$$A_{leak,avg} = 1.07 \text{ ft}^2$$

Potential leakage areas were identified during a walk-through inspection. The two most prominent were in the utility room, a partially attached vent tube that allows air into the home, and this infiltrating air may pass easily under the utility room door. Pictures in the appendix clearly depict the situation.

As outlined in Chapter 6, the LBL model can be utilized to estimate the seasonal variation of the air infiltrating the house. The following assumptions are made for the LBL model to estimate the average air infiltration during the winter season:

- $V_w = 8.5$ mph (average winter wind speed for Denver).
- $T_{indoor} = 63$°F (to account for temperature setback in the winter).
- $T_{out} = 51.8$°F (6-month winter average for Denver).
- $P_{air} = 0.063$ lb/ft^3.
- One story house with moderate local shielding.
- $V_{inf} = A_{leak} * \sqrt{f_s \Delta T + f_w * v_w{}^2}$.

Table 17.5 summarizes the calculations for the average air infiltration for the audited house during winter using the LBL model.

17.3.4 Energy Modeling

The DOE-2 simulation tool, based on the VisualDOE interface, was used to evaluate various energy conservation measures to reduce the energy use and cost for the audited house. Various assumptions were used in modeling the house and its thermal loads.

- No shading except the mountains, which have been modeled as a 1,000-ft high and 3,000-ft wide wall, 2,500 ft west of the house.
- There is no attic and one flat roof of equivalent volume is modeled to represent the two slanting roofs.

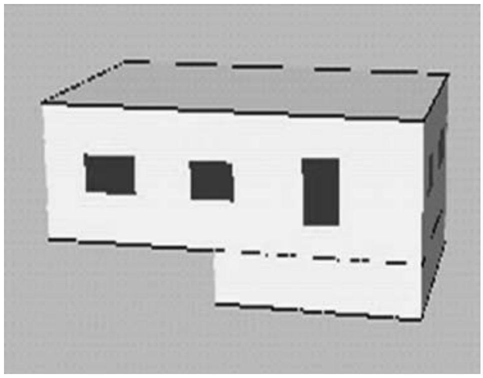


FIGURE 17.6 Model residence from VisualDOE.

- The radiant heat system was modeled as a floor panel heating system.
- Construction types and R-values based on data available from the *ASHRAE Handbook of Fundamentals*.
- The brick walls have no internal insulation.
- The floors are not insulated.
- Occupancy schedules based on owner's estimates.
- Lighting and equipment schedules based on owner's estimates.
- Temperature schedule based on the information provided by the owner as well as data collected during one day of monitoring.
- No cooling load for this house.
- Thermostat type: proportional.
- New windows modeled as double clear 3/12/3 mm glazing.
- Equipment schedule adjusted to reflect winter usage of hot tub.

Figure 17.6 shows the 3-D rendering of the model residence generated by the VisualDOE tool.

17.3.5 Model Calibration

The calibration process of the energy model involved a sensitivity analysis by evaluating the impact of input parameters that seemed most likely to affect usage patterns. As noted earlier, the appropriate material and construction information was entered into VisualDOE. Approximate schedule and occupancy data based on the owner's estimates were input. The temperature settings were based on the owner's information and one day's worth of temperature recording. However, the day that the temperature was recorded was unusually hot for the month of October and the temperatures were most likely a result of the ambient outdoor air temperature and not a heating set-point. The modeled house has hot water radiant heating, so it was treated as having a floor panel heating system in the VisualDOE simulation tool. After entering all known pertinent and available information, a few adjustments were made to match the simulation model predictions to the historical billing data for monthly consumption of electricity and natural gas. The first adjustments focused on the base-load due to domestic hot water heating. Because the house has no cooling system, the only gas used during the summer months is attributed to domestic hot water. Some changes to the occupancy/usage schedule for hot water were made. With the base-load gas usage roughly calibrated, adjustments to lighting and equipment densities and schedules were considered to calibrate the electrical energy usage. With these changes, a calibrated model was achieved within 3.79 percent and 1.34 percent for annual electrical and natural gas consumption, respectively, as summarized in Table 17.6.

TABLE 17.6 Annual Energy Use: Utility Data Compared to Predictions from the Calibrated Energy Model

Energy Type	Actual Use (kWh)	Calibrated Use (therms)	Percent Change
Electricity	9,141	8,794	3.79
Natural gas	1,607	1,585	1.34

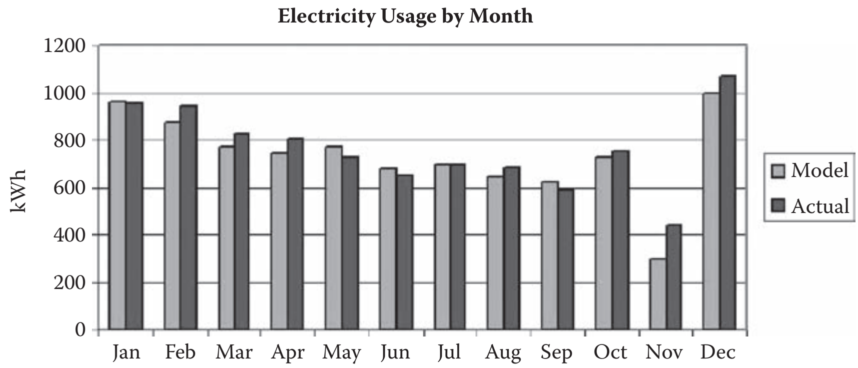


FIGURE 17.7 Comparison of model predictions and utility data for the monthly electricity usage.

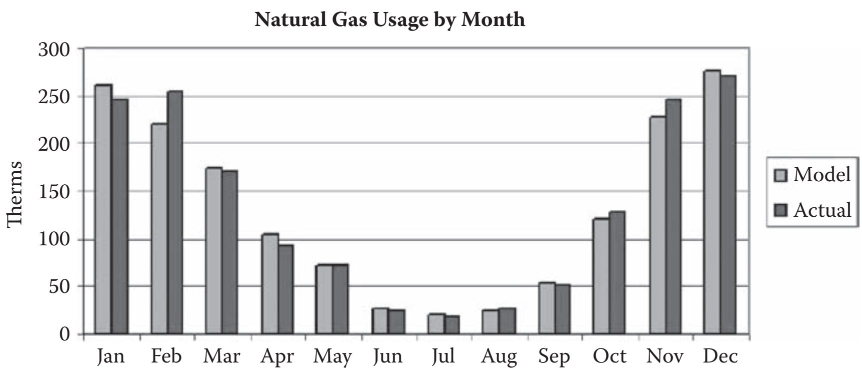


FIGURE 17.8 Comparison of model predictions and utility data for the monthly natural gas usage.

Monthly energy use was also calibrated to ensure an accurate energy model by some adjustments to occupancy schedules. Monthly comparisons for electrical and natural gas consumption are located in Figures 17.7 and 17.8, respectively. There are a few months that show some discrepancies between the model's predictions and actual energy consumption data. Most of these discrepancies are attributed to random occupancy pattern changes by the owners. For example, the occupants took occasional vacations and purchased an electrically heated hot tub during the period of the analysis. Because of these discrepancies, the model was compared to the average of the two years of available utility data. Other than a few anomalies due to vacations and the winter use of the hot tub, monthly energy consumption for our model remained within 10 percent of the actual use.

17.3.6 Energy Conservation Measures

Using the calibrated simulation model for the audited house, a number of energy conservation measures have been analyzed. The results of this analysis are summarized in this section.

Due to the type of building construction with little insulation, additions of insulation for both the floor and the ceiling are considered and simulated as ECMs. Moreover, decreasing air infiltration, switching from incandescent to compact fluorescent lighting fixtures (CFLs), incorporating nighttime temperature setbacks, and replacing the boiler were also analyzed. A summary of the ECMs is provided in Table 17.7.

Installing CFLs was the only measure, among the ECMs listed in Table 17.7, to reduce electricity energy usage attributed to the fact that there is no cooling system in the house and thus there are no electrical savings due to any building envelope improvements. Findings of the energy analysis are summarized in Table 17.8.

For the economic analysis, a discount rate $d = 4.7$ percent was used based on an annual interest rate of 7 percent and an annual inflation rate of 2.4 percent. The cost for implementing each measure was estimated using Means' cost data (Means, 2008). Based on the utility rate, the natural gas price was found to be \$0.60/therm and the electricity costs were \$0.075/kWh. As noted in Chapter 3, the following equation can be used to calculate the *USPW*(*d*,*N*):

$$USPW(d,N) = \frac{1-(1+d)^{-N}}{d} \tag{17.4}$$

Based on the energy savings and the economic analysis, the best ECMs for energy use and cost savings appear to be the nighttime temperature setback and the lighting upgrade as indicated by the results of Table 17.8. These measures have the shortest payback period and the highest net present worth (NPW) values. Other economically feasible measures are reducing the air infiltration rate and replacing the boiler with a more efficient model. Due to very high installation costs, increasing the insulation in the ceiling and floor did not prove to be cost-effective. However, this cost assessment is not sufficient for comparison as the alternatives have different lifetimes.

For an accurate comparison, it is necessary to calculate the life-cycle costs of all the alternatives. For this analysis, a project life of 30 years is considered. During this period, some of the ECMs have to be replaced over the course of the life cycle (refer to Table 17.8 for each ECM lifetime). The replacement costs are accounted for in the annual maintenance costs. An annual baseline maintenance cost of \$75

TABLE 17.7 Summary of the Evaluated ECMs

Energy Conservation Measure	Current Status	Proposed Change
1 – Reduce infiltration	0.44 ACH	0.3 ACH
2 – Increase floor insulation	4-in. concrete with carpet	Add 2-in. expanded polystyrene
3 – Nighttime temperature setback	No setback	Setback to from 69°F to 60°F for 5 hrs/day
4 – Replace incandescent with CFL	Incandescent lights (0.6 W/ft^2)	CFLs (0.3 W/ft^2) for A =1344 ft^2
5 – Increase boiler efficiency	$\eta = 0.8$	$\eta = 0.9$
6 – Increase ceiling insulation	R-19	R-30

TABLE 17.8 Evaluation of the Cost-Effectiveness of the ECMs Using Detailed Simulation Modeling

	Therm Change	kWh Change	Annual Savings (\$)	Initial Cost (\$)	Simple Payback Period (yrs)	N (life of ECM)	USPW	NPW (\$)
Base	0	0	—	—	—	—	—	—
Alt1	–54	0	32.4	150	4.63	10	7.835	103
Alt2	–65	0	39	1,170	30	30	14.527	–603
Alt3	–112	0	67.2	0	0	30	14.527	976
Alt4	69	–1,292	55.5	151	2.72	10	7.835	283
Alt5	–128	0	76.8	775	10.1	15	10.593	38
Alt6	–14	0	8.4	655	78	20	12.785	–548

TABLE 17.9 Summary of the LCC Analysis Results for all ECMs Using Detailed Simulation Modeling

	Do Nothing	Alt. 1	Alt. 2	Alt. 3	Alt. 4	Alt. 5	Alt. 6
Initial cost ($)	0	150	1,170	0	151	775	655
Annual energy costs ($)	1,610.55	1,578.15	1,571.55	1,543.35	1,555.05	1,533.75	1,602.15
Annual maintenance costs ($)	75	75 + 38	75 + 50	75	75 + 38.50	75 + 73.16	75 + 50
Total annual costs ($)	1,685.55	1,691.15	1,696.55	1,618.35	1,668.55	1,681.91	1,727.15
USPW	14.527	14.527	14.527	14.527	14.527	14.527	14.527
LCC ($)	24,486	24,717	25,816	23,509	24,390	25,208	25,745

TABLE 17.10 Evaluation of the Cost-Effectiveness of the ECMs Using Simplified Calculation Methods

	Therm Change	kWh Change	Annual Savings ($)	Initial Cost ($)	Simple Payback Period (yrs)	N (Life of ECM)	USPW	NPW ($)
Base	0	0	—	—	—	—	—	—
Alt1	−66	0	39.6	150	3.79	10	7.835	160
Alt2	−68	0	40.8	1,170	28.7	25	14.527	−577
Alt3	−108	0	64.8	0	0	25	14.527	941
Alt4	−43	−1236	66.9	151	2.3	10	7.835	373
Alt5	−158	0	94.8	775	8.2	15	10.593	229
Alt6	−42	0	25.2	655	26	20	12.785	−328

TABLE 17.11 Summary of the LCC Analysis Results for All ECMs Using Simplified Calculation Methods

	Do Nothing	Alt. 1	Alt. 2	Alt. 3	Alt. 4	Alt. 5	Alt. 6
Initial cost ($)	0	150	1,170	0	151	775	655
Annual energy costs ($)	1,649.78	1,610.18	1,608.98	1,584.98	1,582.88	1,554.98	1,624.58
Annual maintenance costs ($)	75	75 + 38	75 + 50	75	75 + 38.50	75 + 73.16	75 + 50
Total annual costs ($)	1,724.78	1,723.18	1,733.98	1,659.98	1,696.38	1,693.14	1,749.58
USPW	14.527	14.527	14.527	14.527	14.527	14.527	14.527
LCC ($)	25,056	25,183	26,340	24,115	24,794	25,371	26,071

is assumed for all cases even if no ECMs were implemented. Table 17.9 shows the results of the LCC analysis of all the measures.

From the LCC analysis, it appears the option of a nighttime temperature setback (*LCC* = $23,509) is the most economically feasible, followed closely by replacing the lighting fixtures with CFLs (*LCC* = $24,390). The worst alternative in terms of life-cycle costs is replacing the floor insulation (*LCC* = $25,816).

The same analysis was performed based on the results obtained from the simplified calculation methods outlined in various chapters of this book, using the actual utility data as the baseline for both natural gas and electricity energy use (9141 kWh/yr and 1607 therms/yr). The results of the energy use savings and the economic analyses including the LCC analysis are summarized in Tables 17.10 and 17.11.

The results obtained from both the detailed simulation analysis and simplified calculation methods were quite similar as illustrated in Figure 17.9. For both approaches, the nighttime temperature setback yielded the lowest LCC and the increased floor insulation proved to be the most costly alternative.

17.3.7 Conclusions and Recommendations

A detailed energy audit was carried out for a family residence in Denver, Colorado. Based on the utility data, the house has a building load coefficient of 773 (Btu/hr-F), a base-load of 0.96 therms per day, a base-load electrical load of 20 kWh per day, and a balance temperature of 65°F. Utilizing the blower door

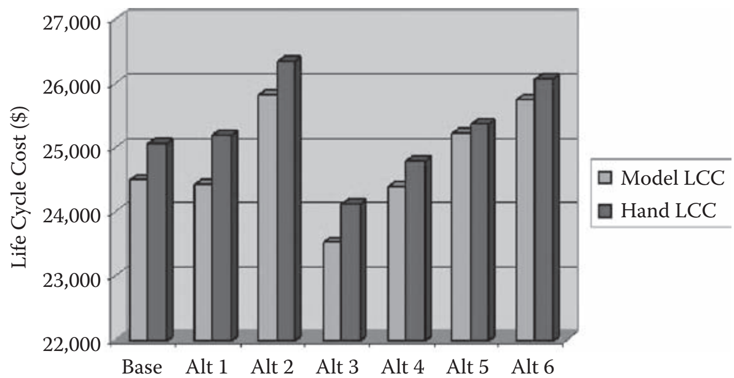


FIGURE 17.9 Comparison of LCC results of all ECMs using detailed simulation analysis and simplified calculation methods.

method, it was determined that this residence has an effective leakage area of 154 in.^{2}, resulting in an infiltration rate of 0.44 air changes per hour (ACH). The heat transmission and air infiltration account for, respectively 85.4 and 14.6 percent of the total BLC of the house.

In order to evaluate the potential savings, a detailed energy simulation model using VisualDOE was developed and calibrated based on monthly utility data. The energy savings for selected energy conservation measures were estimated based on the calibrated energy simulation model as well as simplified calculation methods outlined throughout this book. The results from both the simulation model and the simplified calculation methods have indicated that two measures were cost-effective and are highly recommended to be implemented. These two measures include nighttime temperature setback and replacement of incandescent lamps with CFLs.

17.4 Case Study 3: Audit of a Museum

In this section, findings from a detailed audit of a museum are presented. First, a description of the building's general features is provided. Then, the results of a walk-through audit are outlined including findings from a survey of building occupants. Next, records of the building's utility data are analyzed. Then, selected measurements are described to assess the building performance. Finally, a building model is developed and calibrated using the eQuest simulation tool and the economic performance of several energy conservation measures is considered. The report concludes with recommendations to reduce energy use and cost of operating the building.

17.4.1 Building Description

The Museum Collections building is a 44,820 ft^2 structure originally constructed in the early 1900s. It is located at the main campus of a university in Colorado. The main purpose of the building is to store valuable specimens and artifacts while also housing office, laboratory, and classroom spaces.

Since the original construction of the building, numerous upgrades have been completed including the addition of a major west wing and a lecture hall adjoining the west wing. The last major renovation was completed in 2003 and included major changes to the space layout and mechanical systems. The south façade of the building is shown in Figure 17.10. Historically the building was oriented along the north–south axis and the majority of the building's area continues to be oriented in this way; however, with the additions more surface area has been added on the east–west axis.

Table 17.12 shows the distribution by floor area of the primary space types in the building. It should be noted that specimen storage (collections) accounts for almost half of the overall building floor area but aside from conditioned air and intermittent lighting, it does not impose a significant

FIGURE 17.10 South façade of the Museum Collections building.

TABLE 17.12 Space Allocation

Space Type	Area (SF)	Total Area (%)
Collections	18,160	41
Classrooms	6,413	14
Offices	6,313	14
Labs	4,988	11
Auxiliary	8,945	20
Total	44,820	100

load on the building. It should also be noted that the labs are atypical in that they are not particularly equipment intensive.

17.4.1.1 HVAC Systems

The building is controlled by a single variable-speed air-handling unit (AHU) with heating and cooling coils as well as direct evaporative cooling. Air is distributed through a single-duct system to variable-air-volume boxes with hot water reheat in each zone. A schematic of the AHU is seen in Figure 17.11. The AHU is located on the fourth floor. There are 43 variable-air-volume boxes with a variety of air flows and heating capacities.

Cooling is provided primarily through evaporative cooling and there is a backup direct exchange cooling coil located in the AHU for supplemental cooling. Chilled water is supplied by a central chiller. The central chiller supplies 42°F chilled water to the Museum Collections building as well as two additional buildings. A cooling tower for the central chilled water plant is located inside the building in the fourth floor mechanical room. There are eight fume hoods located in labs in this building.

The building converts high pressure steam (HPS) from the campus central steam plant into low pressure steam (LPS) which is used by the hot water coils in the AHU, cabinet heaters, and domestic hot water. HPS is supplied continuously throughout the year to the building and it is converted to LPS with a heat exchanger system located in the basement.

The HVAC system is controlled by a pneumatic system with a supervisory digital controller. The university facilities management recommends 76°F for the occupied cooling set-point, 70°F for the occupied heating set-point, and 82°F for the unoccupied cooling set-point and 68°F for the unoccupied heating

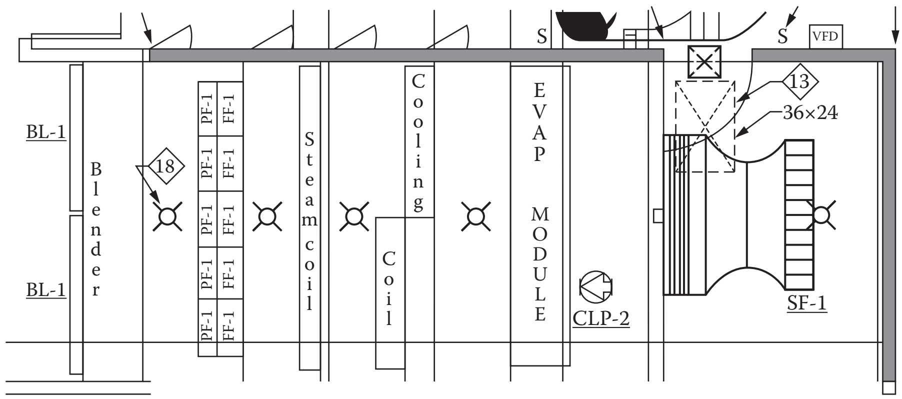


FIGURE 17.11 Building air-handling unit schematic illustrating sequence and components.

TABLE 17.13 Lighting Power Density by Space Type

Audited Space	Area (SF)	Number of Bulbs	Power (W)	LPD (W/ft2)
Hallways	900	14	23	0.36
Collections 1	1,000	40	35	1.40
Collections 2	375	24	35	2.24
Average collections				1.82
Labs	375	24	35	2.24
Classrooms		Same as labs		
Offices	70	6	35	3.00

TABLE 17.14 Overall Building LPD

Audited Space	Weight	LPD	Weighted LPD
Collections	0.41	1.82	0.74
Classrooms	0.14	2.24	0.32
Halls	0.20	0.36	0.07
Labs	0.11	2.24	0.25
Office	0.14	3.00	0.42
Total	**NA**		**1.80**

set-point. A brief survey of thermostat settings throughout the building during occupied times on a November day revealed thermostats with set-points varying from 70°F to 76°F with no apparent setbacks in their settings. The building is typically occupied between 8 a.m. and 6 p.m. during the months of August through May when school is in session. The same schedule with reduced occupancy is considered for the summer months of June and July.

17.4.1.2 Electrical Systems

Surveys were taken in sample spaces to estimate the overall lighting power density (LPD) and equipment power density (EPD) for the building. A somewhat more complex challenge with this building is identifying the equipment power density (plug loads) and specific equipment utilization. Lighting and equipment scheduling and assumptions are further discussed in the modeling section.

A survey was conducted to determine the average lighting power density of the building in W/ft^2. Table 17.13 summarizes the LPD for each space type. Table 17.14 uses a weighted average (based on floor area allocated for each space) to determine an average building LPD of 1.80 W/ft^2. ASHRAE 90.1

TABLE 17.15 Equipment Power per Room Type

		Number of Pieces per Room					
	No. of Rooms	Server	Chest Freezer	Walk-In Freezer	Computer	Projector	Printer
Classrooms	6	0	0	0	1	1	0
Labs	6	0	0	0	0	0	0
Offices	23	0	0	0	2	0	1
Grad Offices	4	0	0	0	5	0	1
Misc	1	2	2	1	0	0	0
Total	**NA**	**2**	**2**	**1**	**72**	**6**	**27**

TABLE 17.16 Total Building Equipment Power

Equipment	Watts	No. of units	Total Watts
Server	750	2	1,500
Chest freezer	440	2	880
Walk-in freezer	4,400	1	4,400
Computer	100	72	7,200
Projector	500	6	3,000
Printer	100	27	2,700
Total	**NA**	**NA**	**19,680**

recommends that classroom/lecture/training and laboratory areas have an LPD of 1.4 W/SF which is fairly aggressive and targeted at new construction. Given that this is an older building, a LPD of 1.80 W/ft^2 provides a reasonable estimation for overall building lighting load.

A similar survey was conducted to determine the equipment power density for the building. Table 17.15 shows the equipment in each typical room and the number of rooms of each space type. Table 17.16 shows the total estimated equipment power in the building. With 44,820 ft^{2}of space, the building has an overall equipment power density of 0.45 W/ft^2. The low EPD is a result of the fact that this building is mostly populated by storage spaces. The lab spaces are generally used for specimen examination and do not have significant equipment loads.

17.4.2 Walk-Through Audit

On a November day, a walk-through audit was performed on the building with the building proctor. The goal of this audit was to gather basic information about the building's operating schedule and to identify any problem areas for the occupants. This section focuses on the potential problem spots in the building and identifies initial energy conservation measures for the building including operations and maintenance measures.

17.4.2.1 Lighting Systems

As most of the building's space is used for collections storage, one of the highest energy use priorities is to turn off lights while not in use in the collections areas. Besides saving energy, turning off these lights helps to reduce UV damage to the sensitive specimens. While touring this building it was observed that lights were off in the majority of collections spaces that were not in use. The building proctor emphasized that the building occupants were meticulous in keeping unused lights off. Posted reminders were found throughout the building requesting occupants to turn off the lights when they leave the rooms.

It was noted that some delamping had already been considered for the hallways. However, the labs, offices, and teaching spaces appeared to be well lit, to the point of overlit. This observation was later confirmed by the lighting measurements.

No incandescent lamps were used in the building, with all lighting coming from fluorescent fixtures. The majority of the lamps were General Electric F32T8/SP35/ECO/CVG lamps. These 32-watt lamps are efficient, producing 2,850 mean lumens, or close to 90 lumens per watt.

In summary, the initial walk-through audit indicated that there was some potential for electricity savings by reducing lighting use in some of the work spaces and to add automatic occupancy sensors to some of the work and collections spaces.

17.4.2.2 Mechanical Systems

As part of the initial walk-through audit, a small survey was conducted to evaluate the thermostat settings. It was found that the thermostat settings varied between 70°F and 76°F with no setbacks. Anecdotally, many of the occupants found it difficult to control the temperature in their spaces due to controls based on thermostats in other rooms or apparent calibration problems with thermostats. It was found that occupants respond to poor control by opening windows to relieve overheating or by using small electric resistance heaters to compensate for cold spaces.

These observations point toward an ECM based on improved HVAC control. There is a strong need to recommission the space so that the control systems operate as intended. Additionally, enforcing seasonal and nighttime setbacks should be considered.

17.4.2.3 Building Shell

During the walk-through audit, it was found that there were two major opportunities for improvements to the building envelope. All of the windows throughout the building were single-pane clear glass windows. Many of the windows had visibly leaky frames. The building proctor noted on the tour that several of the windows allow water to enter the building and, during the winter months, that infiltrating water freezes, expands, and pops opens the windows. The windows then get frozen in the open position causing the heated air to flow directly outside. At one point in the past few years, some funding was secured to upgrade the windows but it was not carried out because the available funds were not sufficient to do a full window replacement. In addition, there was concern that new windows could violate the historic look of the campus building.

Air infiltration could be greatly reduced by improving the seals of the existing windows and repairing those windows that pop open on cold days. The thermal efficiency of the building shell could be improved by replacing the single-pane windows with higher-efficiency windows.

Although the roof of the building has some insulation (R-30) and the north wall is moderately insulated (R-11), the other walls of the building are uninsulated. An obvious potential ECM is to improve the insulation in the roof and add insulation to the walls, although practically, this could be somewhat challenging. Given the structural design of the building, adding insulation using blown cellulose is the best option for the roof.

17.4.2.4 Other Issues

While conducting the walk-through it was noted that the occupants were dissatisfied with several elements of the building. An important concern was security for the collections. There was special concern that the basement storage area, with many rare items, was easy to access through an elevator pass code. The building manager would like to see a key card system to track authorized access to the space.

There have been problems with flooding in the basement, damaging collections. This problem currently seems to be under control with two sump pumps.

Many collections are sensitive to damage from UV light. Past efforts to shade UV radiation from bulbs have created a fire risk. Efforts are currently underway to replace lamps in sensitive areas with low UV lights.

The fume hood exhaust fan can cause severe vibrations in several of the working spaces. It was not possible to examine the fan, but it would be possible to consider improving the fan mounting system.

17.4.3 Utility Data Analysis

The following section presents the available utility data and the analysis approach to determine basic energy performance indicators of the building including the average building energy use, base-loads, building load coefficient, and the balance point temperature. In addition, the building energy use intensity is compared to other similar buildings.

Electricity and steam use data for this building was collected for three years (2007 through 2009). However, only three months of data are available for 2009. Graphical representations of the utility data are provided in Figure 17.12 and Figure 17.13 for, respectively, steam and electricity use.

17.4.3.1 Base-Load Determination

Base-loads for steam and electricity account for general building usage without seasonal variations. Steam base-load is used for DHW and electric base-load for building equipment and plug loads.

The steam base-load was found to be 220 MBtu/month by averaging the steam load for the summer months. This high load could be attributed, in addition to domestic hot water use, to the process loads for the lab equipment, the stringent cleanliness required in the collections storage, or leaky faucets in the restrooms. The electric base-load was found to be 51,000 kWh/month by averaging electric loads for the months of October through March.

Assuming all non-base-load electricity is attributed to space cooling and all non-base-load steam use is due to space heating, a breakdown of building energy end-use can be developed. As seen in Table 17.17, space heating accounts for 36 percent of the building energy use whereas space cooling accounts for only 13 percent. This indicates that this is a shell-dominated structure, not an internal

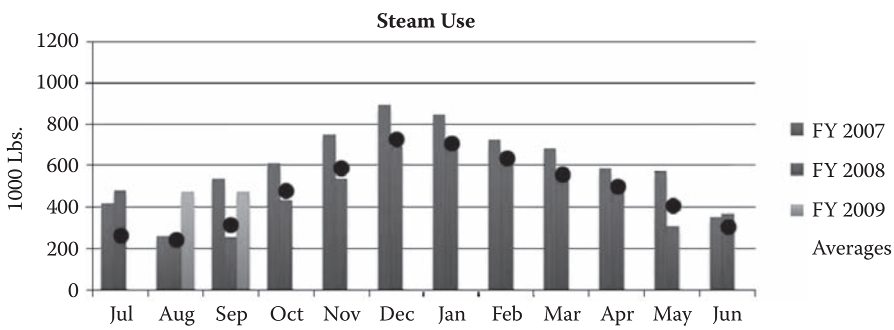


FIGURE 17.12 Monthly steam use for Museum Collections.

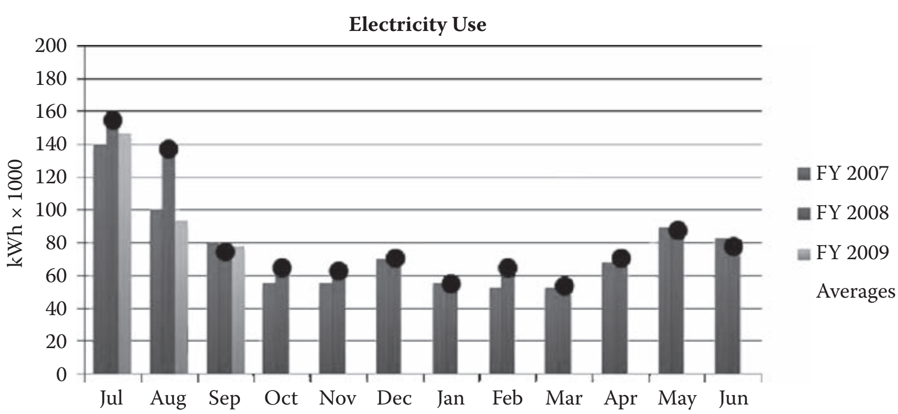


FIGURE 17.13 Monthly electricity use for Museum Collections.

TABLE 17.17 Annual Building Energy End-Uses

End-Use	MMBtu/Year	Load (%)
Heating	3,235	36
Cooling	1,143	13
DHW	2,640	29
Lighting/plug loads	2,047	23

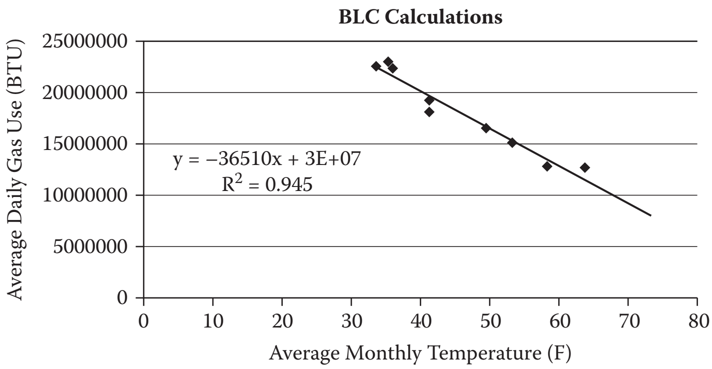


FIGURE 17.14 BLC estimation using utility data.

load-dominated structure, as one might expect. This is a reasonable conclusion given that the majority of the building floor area is unoccupied (essentially storage area) and most occupied spaces are located along the building perimeter near exterior walls. These observations are verified by the energy model.

17.4.3.2 Building Load Characteristics

Using the monthly steam data, it was useful to determine a building load coefficient and balance point temperature (*Tb*) to characterize the performance of the building shell. As outlined in Chapter 6, the building's BLC and *Tb* can be determined by plotting average daily steam use for the heating months against average monthly outdoor ambient temperature as illustrated in Figure 17.14.

Using Eq. (17.5), the building's BLC was determined to be 13,520 Btu/hr-F assuming a heat exchanger conversion efficiency of 90 percent from steam to hot water. With the building's approximate 8,000 square feet of surface area and poor insulation discussed above, along with the high infiltration from leaky windows, this number is very reasonable.

$$
\begin{aligned}
a &= \frac{-24 * BLC}{\eta},\\
-365102 &= \frac{-24 * BLC}{0.90},\\
BLC &= 13{,}520[BTU/hr - F]
\end{aligned}
\tag{17.5}
$$

17.4.4 Occupant Survey

A survey was developed and distributed to building occupants to determine their satisfaction with the building. The following sections describe the survey and its results.

The survey consisted of three major sections. The first part of the survey asked about the amount of time spent in the building and type of work done. The second part requested information about the type of work space and its proximity to windows. The third part consisted of seven sections characterizing the occupant's impression of the space. The seven sections characterized general impressions, lighting, acoustics, air quality, ventilation, overall temperature satisfaction, and seasonal temperature profile.

The most significant conclusions that can be drawn from the survey include:

- The user space distribution confirms the shell-dominated nature of the structure.
- Over half of the occupants are generally satisfied with the space and 38 percent are dissatisfied.
- Occupants have inadequate control of the building systems in their space including lighting and HVAC systems.
- Lighting is adequate, but could be better controlled.
- The building is inadequately heated during the winter months.

17.4.5 Field Testing and Measurements

In addition to the walk-through audit, field testing and measurements were conducted to gain a better understanding of the building's performance. These measurements characterized lighting quality, thermal and humidity performance, and thermal performance of the building shell.

17.4.5.1 Lighting Quality

A field test was conducted to determine the average illuminance in a collections space, lab space, and office space. The following sections show results and analysis of this field testing. Measurements were taken at a work surface or three feet off the ground. None of the tested spaces had windows so measurements were not affected by daylight.

Figure 17.15 shows the results of the illuminance measurements for a collections storage space. Each of the luminaires in the space has three 35-W lamps. The collections spaces typically need illuminance values between 30 and 50 footcandles (fc). On average, the measurements indicate that the collections spaces meet this need with some areas near the walls having inadequate lighting

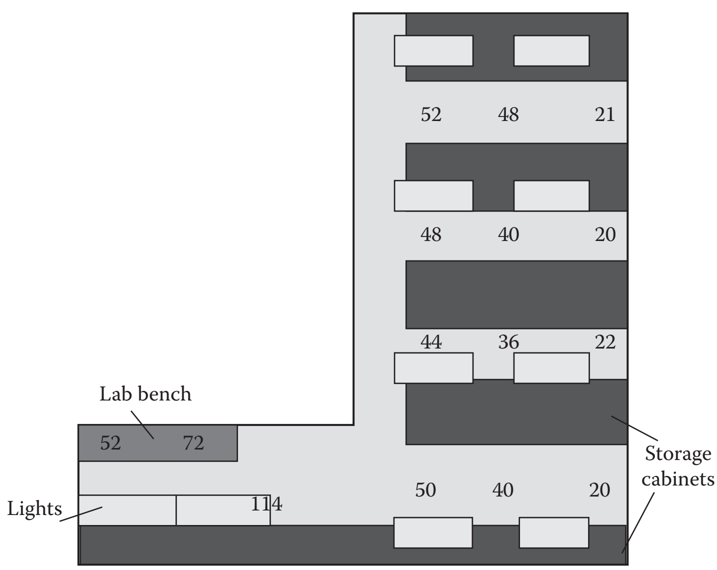


FIGURE 17.15 Lumen (fc) measurements in collections storage room 125.

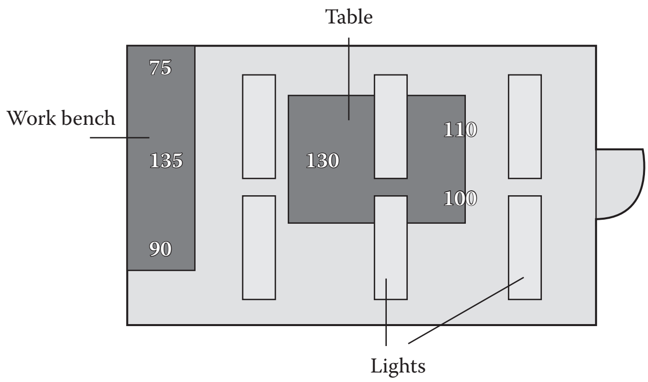


FIGURE 17.16 Lumen (fc) measurements in lab room W130.

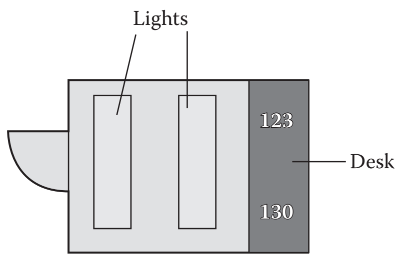


FIGURE 17.17 Lumen (fc) measurements in office room W126a.

levels. However from field observations of the space *s*, the low light at the walls should not affect task effectiveness.

Similar measurements were conducted in a typical lab space with the results shown in Figure 17.16. Each of the luminaires has four 35-W lamps. The Illuminating Engineering Society of North America (IESNA) recommends 50 fc on the surface for lab-type visual tasks. This requirement assumes that workers are primarily under 40 years of age, the countertop surfaces are light colored, and activities include medium contrast or small size, medium bench and machine work, and difficult inspection tasks. The lab should not need illuminance values in excess of 50 fc at the work surfaces yet it exceeds this need by almost double in several locations. Thus, there is an opportunity to consider delamping for the lab spaces.

Figure 17.17 illustrates lighting levels on a work surface of a typical office space. Each of the luminaires has four 35-W lamps. Similar to lab spaces, 50 fc is an adequate amount of illuminance for the office spaces. The measured lighting levels far exceed the IESNA requirements presenting an opportunity to delamp the office spaces.

17.4.5.2 Space Temperature and Humidity Profiles

Small battery-powered data loggers were used to measure the temperature and humidity levels for two building spaces: an interior common space and an exterior lab. Measurements were taken continuously for five days from Monday, December 1st to Friday, December 5th as shown in Figures 17.18 for the common room and 17.19 for the lab space. The relative humidity variation is represented by the lower curves for both spaces.

The measurement results shown in Figures 17.18 and 17.19 indicate that there is no clear temperature setback during unoccupied nighttime hours with large fluctuations from day to day. The common room, located in the core of the building, maintains a reasonable temperature during all hours of the

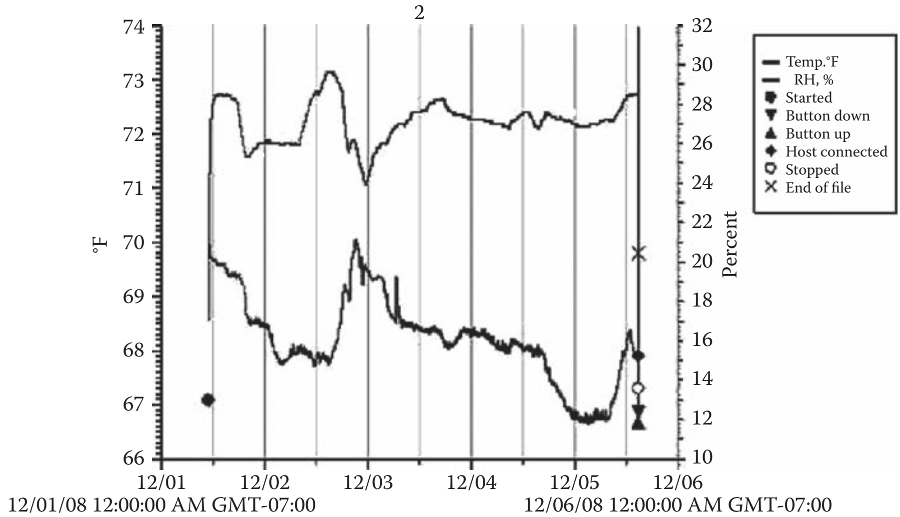


FIGURE 17.18 Temperature and humidity levels measured in a core common room.

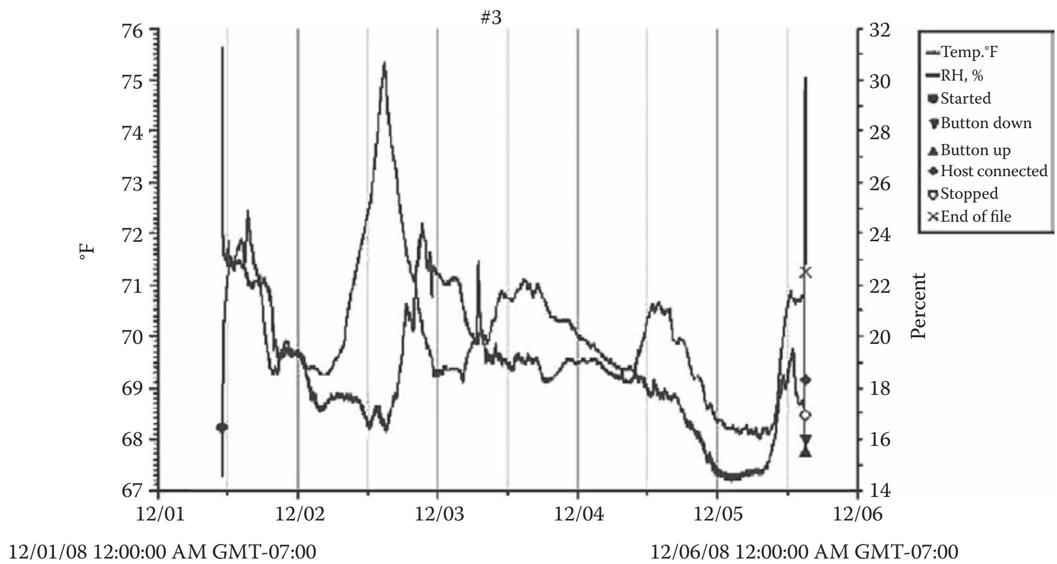


FIGURE 17.19 Temperature and humidity levels measured in a perimeter lab.

day. The lab, located in the perimeter of the building, maintains reasonable temperatures through most days. However, on the colder night of December 5th, the room's temperature dropped down close to 67°F and did not recover until later in the day. This is not a severe thermal comfort violation, but the long recovery time may indicate that there is inadequate heating available to heat the space on the coldest days of the year.

17.4.5.3 Thermal Imaging

On the evening of November 20th, a thermal imager was used to examine the thermal performance of the building. The goal of using thermal imaging was to identify locations where the building shell was leaking heat to the outside. The analysis was done on a mild warm evening so cool temperatures indicate

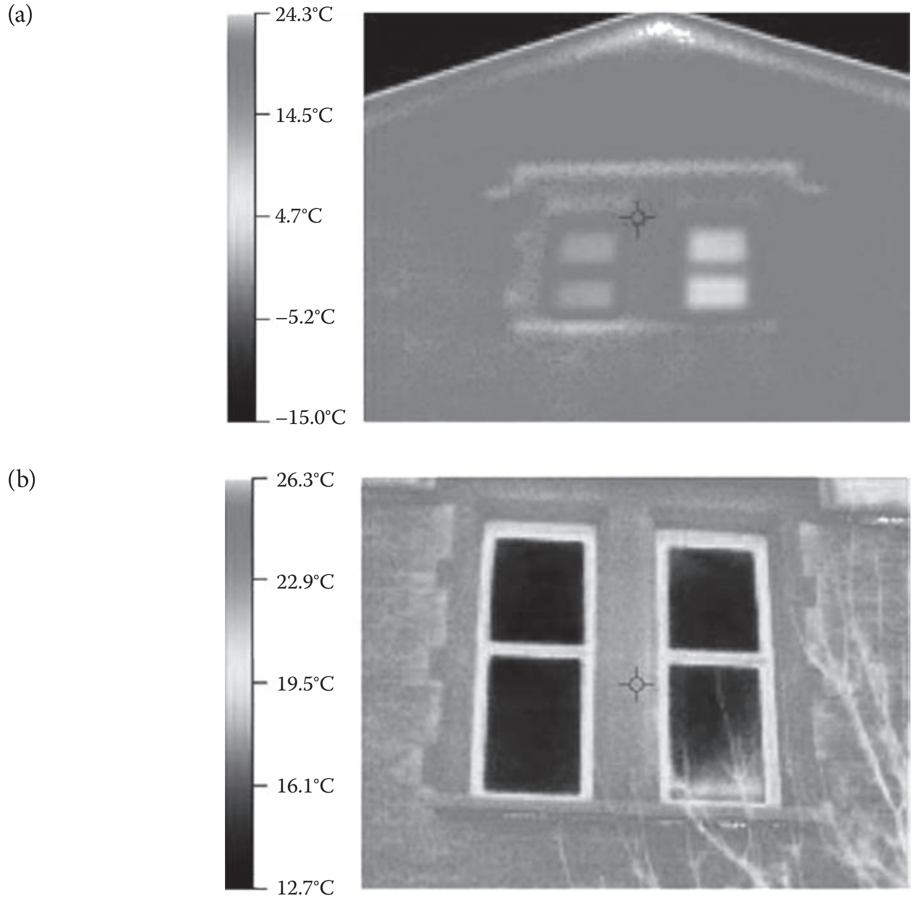


FIGURE 17.20 IR images for windows on south face of west wing: (a) upper windows; (b) lower windows.

heat losses. Because there was only a small differential between the building temperature and ambient temperature, even small temperature differences may be indicative of problem spots in the building shell. It should be noted that the west wing is a newer addition whereas the east wing is the original, much older, building.

When examining the south face of the west wing, it was found that the only major area of heat loss was the windows and surrounding frames. The higher window frames seem to have better energy performance than the lower frames as depicted by Figures 17.20.

Figure 17.21 shows a thermal image of the uninsulated east façade part of the original structure. Significant heat losses can be noted as indicated by the rather low temperature characterizing the entire wall including the windows. Similar results were found for the uninsulated south façade.

17.4.6 Energy Modeling

The building was modeled primarily based on the architectural drawings provided by the university facilities management. The DOE2.2-based energy simulation tool eQUEST was used to model the building and assess the energy-saving potential for various energy conservation measures. Within eQUEST, the DD wizard was used to enter most of the assumptions and create the building envelope and geometry features. Some additional modifications as well as adjustments needed for the calibration effort were

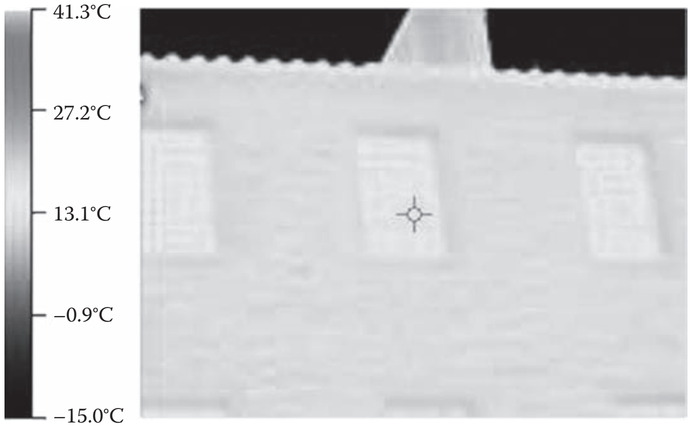


FIGURE 17.21 IR images for the east façade of east wing.

TABLE 17.18 Summary of Input Data for the Building Energy Model

Component	Notes
	Envelope
Roof	R-30 batt insulation between joists
Walls	2 × 4″, steel studs, no insulation (R-11 on North), prefinished composite panel on exterior, assumed 12″ concrete for below grade walls
Windows	Single pane, aluminum frame, no break
Window to wall ratios	West 13.1%, north 16.8%, east 10.8%,
	south (elevations missing, assumed 16.8%)
Doors	Solid wood core 1 3/4″ (both interior and exterior)
Slab	6″ concrete
	Schedules
Occupied hours	8:00 a.m.–6:00 p.m., August–May
Reduced occupancy	8:00 a.m.–6:00 p.m., June–July
	Systems & Equipment
Setpoints	Occupied: heating = 70°F, cooling = 76°F
	Unoccupied: heating = 82°F, cooling = 68°F
Lighting power density	Refer to Table 17.13
Equipment power density	Specific per zone, refer to Table 17.15
Heating equipment	Distributed campus steam
Cooling equipment	Chilled water from shared chiller, 236 tons, water cooled condenser
Air distribution	Custom AHU: 4 coils and a total system capacity of 46,390 cfm and 1555 MBH. VAV boxes in each zone; 8 fume hoods

carried out using the Detailed Edit mode available within eQUEST. The following section describes the energy modeling process and the calibration efforts.

17.4.6.1 Building Envelope, Geometry, and Thermal Zones

The building model was started in the DD Wizard using the input data summarized in Table 17.18. To enter the proper geometry for the building, four shells were modeled in detailed mode; the basement, the first floor, second and third floors (combined in one shell), and the fourth floor. The window-to-wall

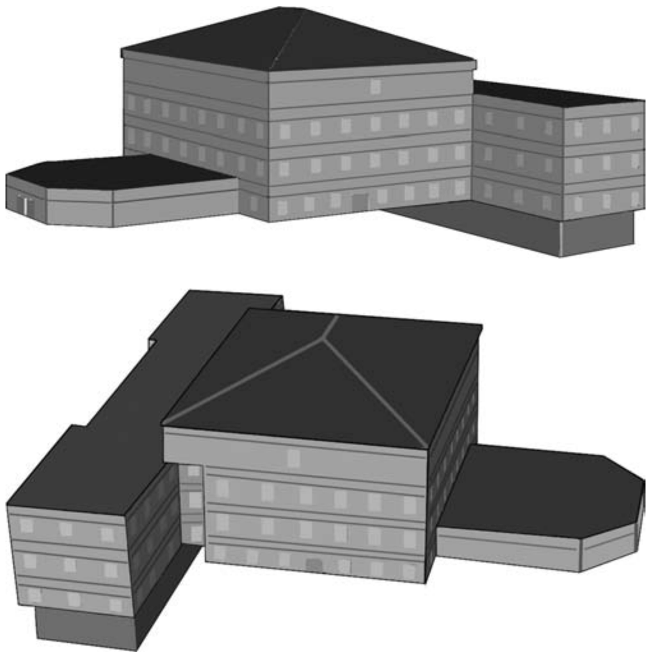


FIGURE 17.22 3-D Renderings of the building energy model.

ratios were adjusted for each façade. Figure 17.22 illustrates the 3-D renderings of the energy model developed using the eQUEST simulation tool. Individual zones were created and the space allocations were specified for each shell.

The building energy model has seven thermal zones on the first floor: auditorium, west group of bathrooms, lab storage, central corridor, classrooms on the south, interpretive research lab, and the offices along the north. The zoning was selected based on the air-conditioning system types and the typical occupancy schedules. Figure 17.23 presents the basic outlines of the individual thermal zones.

Floors 2 and 3 were created in the same shell inasmuch as they have the same footprint and similar zoning. The space functions and areas were matched so that the same zoning layout could be considered for both floors. There were nine zones for these two floors: offices along the south, museum storage in the central west end, lab in the northeast wing of the west end, office on the very north tip of the east wing, classroom/teaching lab for E280, offices south of the teaching lab, restrooms in the central east wing, and storage on the south tip of the east wing and the central corridors.

Two zones were created for the fourth floor, one for the mechanical room and the other for the offices. Similarly, two zones were created for the basement, one for the mechanical room on the north end of the basement and the other for the remainder of the storage.

17.4.6.2 HVAC Components

In general, the cooling equipment is located on the fourth floor and the heating and DHW equipment are located in the basement. The air-handling unit for this building is a custom unit with 4 coils and a total system capacity of 46,390 cfm and 1555 MBH. It is approximately 11 ft 6 in. wide and 11 ft tall. It contains a supply fan, an evaporative coil (direct/indirect, 46,325 cfm, 1200 MBH), a cooling coil (11,600 cfm, 389 MBH), a steam coil (46,325 cfm, 1980 MBH), filters, and a blender. The cooling tower is also located on the fourth floor.

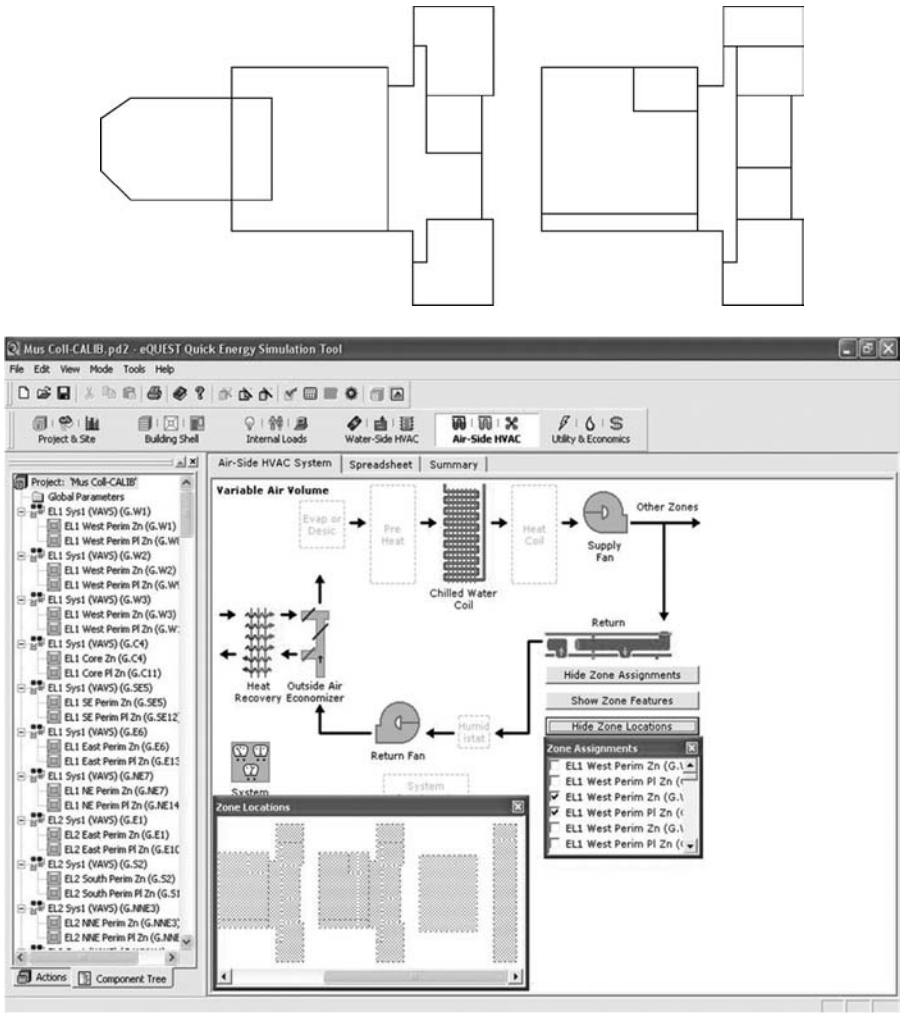


FIGURE 17.23 Thermal zoning for each floor used for the building energy model.

Heating is provided through the air system for the classrooms, labs, offices, and storage spaces. Cabinet unit heaters are located in the corridors, vestibules, and the restrooms.

The eight fume hoods were modeled using the flow rate specified in the building design drawings.

In the DD wizard, the initial plant equipment was modeled as noted by the manufacturers' specifications. The chilled water system consists of a single chilled water pump. It is a base-mounted Taco model FE 3010 pump. The FE line of pumps from Taco has been discontinued so data was sparse. It was assumed to be a constant-volume pump using 473 gpm and 73 ft of head as indicated on the pump schedule. It was modeled as a standard efficiency motor (shown to be 80 percent in the schedule) because standard efficiencies are typically found in existing central plants.

The chiller was modeled as a water-cooled screw chiller using default eQUEST performance curves. The actual chiller is a Dunham Bush WCFx24 which is a custom unit with no available data. As indicated on the chilled waterflow diagram, the chiller is shared with two other neighboring buildings. The efficiency of this chiller was calculated to be 0.60 kW/ton based on a kW input of 143 and a 236 ton capacity.

The water-cooled condenser was modeled with a single pump (Taco model FE4010) with a head of 81 ft and 667 cfm. The cooling tower was modeled as an open cross-flow heat exchanger.

The operating schedule for the chilled water system was assumed to be 7 a.m. to 6 p.m., 5 days a week. The set-point was based on the outside air reset and had a CHW minimum temperature of 42° and a CHW max temperature of 54°, as indicated on the chiller schedule. It had scheduled operation based on an OA temperature of 55°.

The hot water system was modeled with a single pump system with two pumps in parallel. The loop head of each is 54 ft and the flow is 48 gpm. A standard motor efficiency was assumed inasmuch as it is an older system.

The hot water system was controlled based on an outside air reset. The minimum HW temperature is set at 110°F and because it is operating continuously, the operation was modeled as continuous in eQUEST. It is controlled by an on–off controller with a schedule of 24 hours, 7 days per week.

The air side system uses chilled water coils as its cooling source and hot water coils for its heating source. The conditioned air is distributed to the zones with VAV boxes with HW reheat coils. The return air is ducted and the exhaust air from the fume hoods and from the bathrooms is vented separately as direct exhaust with no recirculation.

The occupied set-points were modeled at 76°F for the occupied cooling set-point, 70°F for the occupied heating set-point, 82°F for the unoccupied cooling set-point, and 68°F for the unoccupied heating set-point. These values were based on general recommendations that the facilities management department provides to building proctors. The actual set-points in the spaces varied from 70°F to 76°F as depicted by the field measurements.

The supply fans were modeled at 75 hp and the exhaust fans were modeled at 20 hp per the information found in the mechanical drawings. Standard efficiency fans were used for both. The flow of the supply fan was modeled at 46,325 cfm and the return fan was modeled at 28,755 cfm. The fan system was assumed to operate one hour before and after the building's regular hours of operation. The economizer was set to operate based on the dry-bulb temperature of 65°F, a measurement that is common for the university buildings.

17.4.6.3 Calibration of the Energy Model

After completing the initial model it was necessary to calibrate it to the actual building utility data. Both the steam and electricity predictions from the initial energy simulation model were significantly lower than the monthly average utility data. The following adjustments were made in order to calibrate the energy model:

- The LPD was increased in key areas, including some areas, such as the storage areas, that may have lighting densities exceeding the averages found in the survey carried out during the walk-through audit. Task lighting was also added to offices.
- The EPD was increased to account for the fact that some of the storage areas have individual computers, microscopes, and other pieces of equipment.
- The air infiltration was increased from 1.5 ACH to 2.5 ACH in the perimeter zones and from 0.125 to 0.5 in the core zones. Because the windows freeze open in the winter and several open windows were observed during the survey this is a reasonable assumption.
- The DHW load was increased in order to match the steam loads to actual base load steam usage. As discussed in the utility analysis section the DHW load is rather large and is found to be roughly 220 MMBtu/month.
- The schedules for internal loads have been adjusted. Specifically, the lighting, occupancy, and miscellaneous equipment loads in January and March were reduced to take into account holiday breaks when the school is closed down. Loads were also reduced in October to account for fall break. Heating and DHW loads were also reduced during these periods.

After these adjustments, the modeled monthly electricity and steam use are within 10 percent of the actual use indicated by the utility data as illustrated in Figure 17.24 for electricity use and Figure 17.25

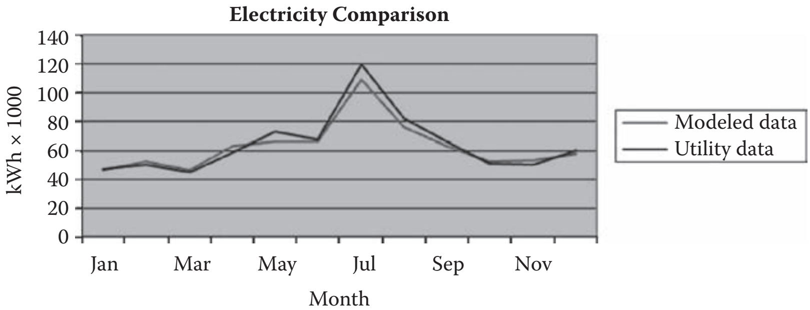


FIGURE 17.24 Monthly electricity use predicted by the calibrated energy model compared to utility data.

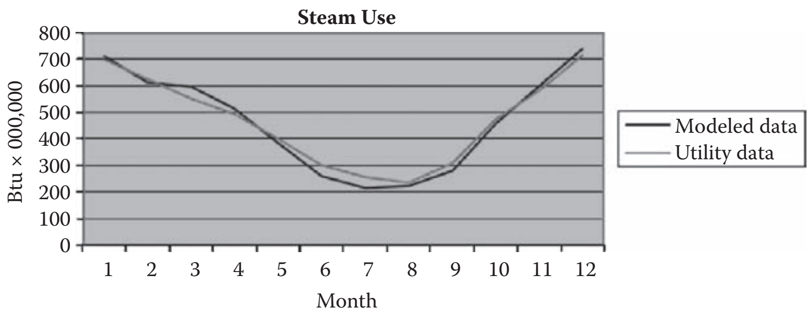


FIGURE 17.25 Monthly steam use predicted by the calibrated energy model compared to utility data.

for steam use. Tables 17.19 and 17.20 provide a comparative analysis of the calibrated energy end-uses against utility data for electricity and steam, respectively.

17.4.7 Analysis of Energy Conservation Measures

17.4.7.1 Overview

Based on the results of the calibrated building energy model, the primary end-uses of energy in this building are space heating (41 percent), hot water (27 percent), and lighting (16 percent) as summarized in Table 17.21. It should be noted that space cooling energy use predicted by the model is lower than expected, most probably due to the use of the air economizer cycle. Therefore, space heating, hot water, and lighting systems are the key target areas for improving the energy performance of the building.

The energy conservation measures that were analyzed include operational and maintenance improvements with little or no capital costs. Other measures that would require higher capital costs were also considered.

In the following section, energy and cost analyses of selected ECMs are described. Implementation costs including labor costs for each measure are considered. Energy use and cost savings are calculated using the calibrated building energy model developed using eQUEST as well as approximate estimations based on the simplified calculation methods outlined in various chapters of this book. A discussion of the differences for the two calculation approaches is included with each ECM analysis.

For the simplified calculations, a few assumptions were consistently used including: 5,487 heating degree-days and 691 cooling degree-days (for Denver, Colorado). The heating process is assumed to be 80 percent efficient; the cooling process operated at an efficiency of 0.6 kW/ton.

TABLE 17.19 Monthly Electricity End-Uses Predicted by the Calibrated Energy Model Compared to Utility Data

Item		Base Case	ECM1: Delamping	ECM2: Increase Ceiling Insulation	ECM3: Window Replacement	ECM4: Occupancy Sensors	ECM5: Premium Efficiency Pumps	ECM6: Fume Hood Sensors	ECM7: Reduce DHW Demand	ECM8: Optimized Package of ECMs
Initial cost	IC	$0.00	$0.00	$14,142.00	$124,740.00	$9,620.00	$8,037.00	$40,000.00	$900.00	$182,397.00
Annual energy costs	Electricity (kWh)	77,1000	67,0770	771,000	759,435	702,381	75,1725	726,205	771000	553809
	Electricity cost	92,520	80,492	92,520	91,132	84,286	90,207	87,145	92520	66457
	Gas (MMBtu)	5,635	5,860	5,619	4,683	5,782	5,635	4,880	5226	4821
	Gas cost	13,6311	14,1763	135,926	113,274	139,855	13,6311	118,045	126428	116614
	Total cost	22,8831	22,2255	228,446	204,406	224,140	22,6518	208,690	218948	183071
	Annual savings	NA	6,575	385	24,424	4,690	2,313	20,141	9883	45760
	% Annual savings	NA	0	0	0	0	0	0	0	0
Economic indices	SPP (years)	NA	0.00	36.77	5.11	2.05	3.47	1.99	0.09	3.99
	USPW	14	14	14	14	14	14	14	14	14
	LCC	32,25126	31,32456	323,3848	3,005,632	316,8643	32,00564	2,981,260	3086743	2762588
	NPW	NA	92,670	-8,721	219,495	56,483	24,562	243,867	138,384	462,539

TABLE 17.20 Monthly Steam End-Uses Predicted by the Calibrated Energy Model Compared to Utility Data

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Utility data (Btu × 000,000)	700	627	550	493	398	298	256	234	307	472	583	717	5635
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total
Space cool	0	0	0	0	0	0	0	0	0	0	0	0	0
Heat reject.	0	0	0	0	0	0	0	0	0	0	0	0	0
Refrigeration	0	0	0	0	0	0	0	0	0	0	0	0	0
Space heat	551	468.3	438.4	298.5	174.5	50.4	2.8	17.9	83.4	264.9	413.3	584.6	3,348.00
Hp supp.	0	0	0	0	0	0	0	0	0	0	0	0	0
Hot water	157.1	140.9	154	213.5	203.9	205.9	206.5	202.7	192	188.8	184.6	152.1	2,202.00
Vent. fans	0	0	0	0	0	0	0	0	0	0	0	0	0
Pumps & aux.	0	0	0	0	0	0	0	0	0	0	0	0	0
Ext. usage	0	0	0	0	0	0	0	0	0	0	0	0	0
Misc. equip.	2.3	2.1	2.3	2.4	2.4	2.2	4.6	2.4	2.2	2.4	2.1	2.3	29.8
Task lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Area lights	0	0	0	0	0	0	0	0	0	0	0	0	0
Total	710.5	611.3	594.7	514.3	380.8	258.5	213.9	223	277.7	456	600	739	5,579.80
Difference	102%	97%	108%	104%	96%	87%	84%	95%	90%	97%	103%	103%	99%

TABLE 17.21 Building Energy End-Uses

Building Energy End-Use	Percent of Total
Space heating	41
Domestic hot water	27
Electrical lighting	16
Electric plug loads	8
Circulation fans	4
Space cooling	2
Water pumps	2

17.4.8 Energy Savings Estimation

17.4.8.1 ECM 1: Delamping 30 Percent of Lamps

It was observed from both the walk-through energy audit and lighting measurement that some spaces within the building were overlit compared to the IESNA requirements based on the tasks performed in those spaces. For these spaces, it is worth considering delamping by removing unnecessary lamps from selected lighting fixtures.

17.4.8.1.1 Implementation and Costs

This ECM is rather simple to implement. The building maintenance staff can remove unnecessary lamps while ensuring that the desired lighting levels are maintained within the overlit spaces. The capital cost of this measure consists of the labor for the maintenance staff. This cost can be neglected inasmuch as it will be offset by the following benefits:

- The removed bulbs can be stored and used when the currently installed bulbs reach the end of their lives, thus saving money.
- Delamping would reduce bulb replacement and labor costs because there are fewer overall bulbs to replace.

17.4.8.1.2 Energy Model Predictions

In the energy simulation model, the LPD was reduced by 25 percent compared to the existing LPD. The lighting in the corridors will likely not be able to be reduced because these spaces are equipped with two-lamp round fixtures. Thus delamping is not feasible in the corridors which account for roughly 16 percent of the total floor area. However, deplamping can be carried in all other spaces. This measure resulted in reducing electricity demand by 100,230 kWh/year (13.0 percent reduction) and a heating increase of 225.4 MMBtu/year (4.0 percent increase).

17.4.8.1.3 Simplified Calculations

In order to determine the potential savings from delamping, it is first necessary to determine the amount of light energy that is currently being used in the space. The lighting load is a function of the LPD and number of hours that lighting fixtures are operated. As discussed in the utility analysis section, the building electrical energy base-load is 51,000 kWh/month. This load is distributed to lighting, plug loads, and equipment loads. Based on the LPD value and the number of hours of operation, it can be estimated that the energy model indicates 50 percent of the electricity base-load can be attributed to lighting (this estimate is confirmed by the energy model predictions).

The energy savings due to delamping can be estimated by multiplying the annual lighting energy use by the 25 percent reduction. Using this method the initial annual lighting load was estimated at 3.06×10^5 kWh/year and the reduced lighting load was found to be 2.29×10^5 kWh/year; this is a reduction of 76,500 kWh/year (10 percent).

The effect of delamping on heating has been neglected in this simplified calculation even though it can be estimated by using the variable-base degree-day method with higher balance temperature due to lower internal gains (refer to Chapter 6).

17.4.8.1.4 Discussion

Both methods produce similar results with the energy simulation model indicating a 13.0 percent energy reduction and the simplified calculations indicating a 10 percent reduction. This is not surprising: both models are based on similar assumptions for the lighting operating hours and the impact of delamping. The energy model predictions account for daily and seasonal scheduling variations as well as the effect of delamping on heating energy use.

17.4.8.2 ECM 2: Increased Roof Insulation

In an attempt to reduce heating load, the largest energy end-use for the building, increasing roof insulation from R-30 to R-50 is considered. This measure would be accomplished with blown cellulosic insulation.

17.4.8.2.1 Implementation and Costs

To install additional insulation to the roof, insulation would be blown below the roof deck of all roof areas including the auditorium and the east and west wings. The cost of this ECM is estimated to be \$1.29/ft^2, or \$14,142 for the entire 10,963 ft^2 of roof space (Means, 2008).

17.4.8.2.2 Energy Model Predictions

The ceiling insulation was added to the model by changing a global parameter in detailed mode and modifying the material constructions dialogue box to include a global parameter for R-50 roof insulation. Verification was performed to ensure that this parameter was applied to all three roof areas including the auditorium roof, the fourth floor roof, and the roof above the third floor east wing.

The energy savings of this measure were only a 15.9 MMBtu reduction is steam use (0.3 percent). There was no change in cooling load.

17.4.8.2.3 Simplified Calculations

The effect of increasing roof insulation on cooling and heating load was estimated using the degree-day method outlined in Chapter 6. Because the insulation was increased from R-30 to R-50 and the roof has an area of 10,963 ft^2, the change in BLC was found to be 146 Btu/hr°F. This value is almost inconsequential when compared to the building BLC of 12,170 Btu/hr°F. The savings in space heating energy use are estimated to be ΔFU of 24 MMBtu/year or 0.16 percent in annual steam use reduction (using the heating degree-day of 5487°F-days). The savings in space cooling energy are found be ΔEU = 1210 kWh/year (using a cooling degree-day of 691°F-days), a reduction of 0.43 percent in annual electricity use.

17.4.8.2.4 Discussion

Both methods indicated an inconsequential energy savings from this ECM. This result is primarily due to the already adequate roof insulation levels

17.4.8.3 ECM 3: Window Replacement

Replacing the windows could not only provide an opportunity to improve glazing performance but also reduce air infiltration and provide better thermal comfort for the occupants. The current windows consist of older, operable, single-pane types with metal frames and no seals. While deciding on the window replacement option it is important to consider the requirement of maintaining the campus' architectural look.

17.4.8.3.1 Implementation and Costs

Windows with double-pane insulating glass with spacer bars would maintain the historic look of the facades while providing superior performance compared to the existing windows with single panes. In order to determine the new window thermal properties, the WINDOW 5 program was used to estimate the thermal and solar optical properties of glazing and window systems. Refer to Figures 17.26 through 17.29 for sample outputs from the WINDOW 5 program.

The capital cost for the proposed windows was estimated to be $309 per unit. With 162 windows in the space this translates to a cost of $50,058, installed (Means, 2008).

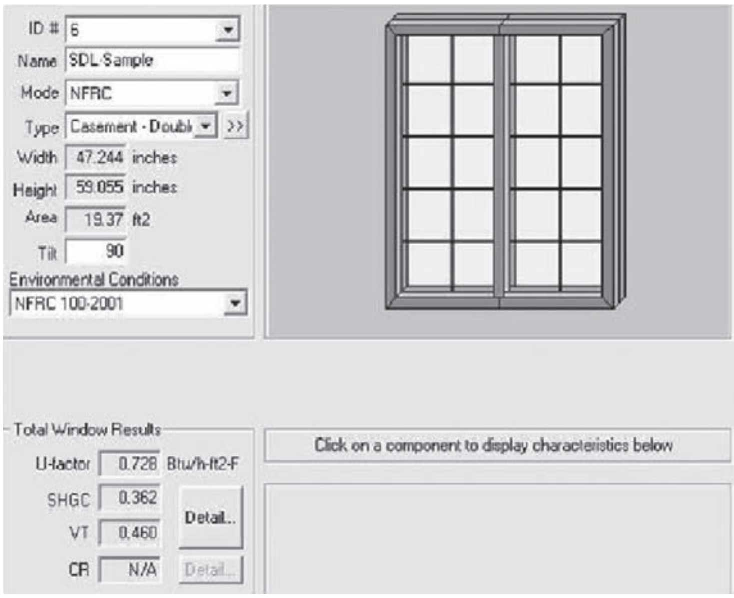


FIGURE 17.26 Overall window performance data and image of window from WINDOW 5.

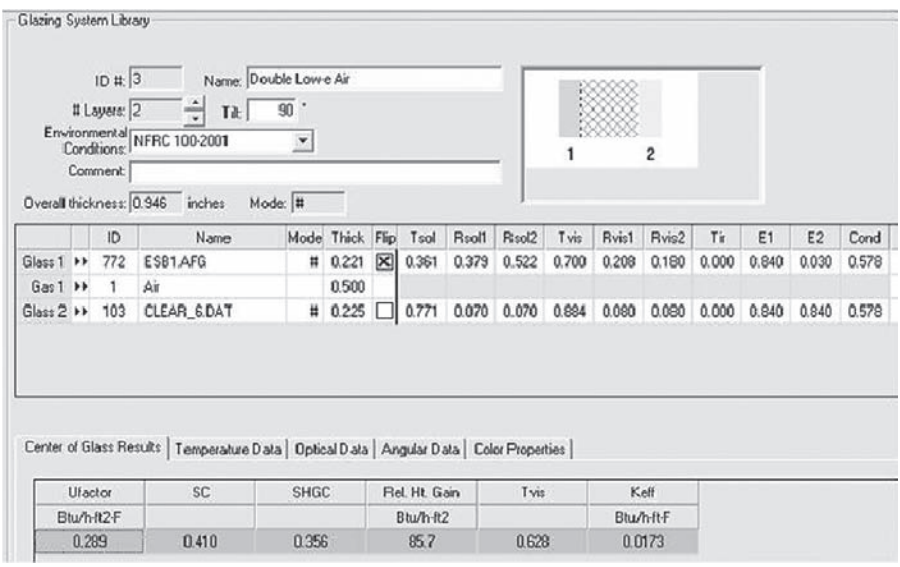


FIGURE 17.27 Glazing system performance from WINDOW 5.

Frame Library (C:\Program Files\LBNL\WINDOW5\w5.mdb)

ID	Name	Source	Type	Frame Uvalue	Edge Uvalue	Edge Correlation	Glazing Thickness	Pfd	Abs	Color
				Btu/h-ft2-F	Btu/h-ft2-F		inches	inches		
1	Al no break	ASHRAE	N/A	1.900	N/A	Class1	N/A	2.250	0.90	
2	Al w/break	ASHRAE	N/A	1.000	N/A	Class1	N/A	2.250	0.90	
3	Al flush	ASHRAE	N/A	0.699	N/A	Class1	N/A	2.250	0.90	
4	Wood	ASHRAE	N/A	0.400	N/A	Class1	N/A	2.750	0.90	
5	Vinyl	ASHRAE	N/A	0.299	N/A	Class1	N/A	2.750	0.90	
6	sample-head.THM	Therm	Head	0.353	0.414	N/A	1.043	1.689	0.30	
7	sample-jamb.THM	Therm	Jamb	0.359	0.414	N/A	1.043	1.689	0.30	
8	sample-sill.THM	Therm	Sill	0.352	0.413	N/A	1.043	1.689	0.30	

FIGURE 17.28 Frame performance from WINDOW 5.

Select Cancel Find ID 1931 records found.

ID	Name	ProductName	Manufacturer	Source	Mode	Color	Thickness	Tsol	Rsol1	Rsol2	Tvis	Rvis1	Rvis2	Tir	emis1	emis2
							inches									
769	ES752.AFG	Silver Med%T LowE on Azurlite	AFG Industries	IGDB v11.4	#		0.2230	0.215	0.304	0.058	0.514	0.048	0.065	0.000	0.090	0.840
770	ES840.AFG	Silver Low%T LowE on Evergr	AFG Industries	IGDB v11.4	#		0.2200	0.163	0.350	0.071	0.390	0.076	0.099	0.000	0.103	0.840
771	ES852.AFG	Silver Med%T LowE on Evergr	AFG Industries	IGDB v11.4	#		0.2210	0.201	0.315	0.063	0.471	0.054	0.066	0.000	0.087	0.840
772	ES81.AFG	Sunbelt LowE on Clear	AFG Industries	IGDB v11.4	#		0.2210	0.361	0.522	0.379	0.700	0.180	0.208	0.000	0.030	0.840
773	ES82.AFG	Sunbelt LowE on Green	AFG Industries	IGDB v11.4	#		0.2330	0.246	0.523	0.131	0.591	0.181	0.164	0.000	0.029	0.840
774	ES83.AFG	Sunbelt LowE on Gray	AFG Industries	IGDB v11.4	#		0.2140	0.197	0.520	0.149	0.367	0.165	0.089	0.000	0.030	0.840
775	ES84.AFG	Sunbelt LowE on Bronze	AFG Industries	IGDB v11.4	#		0.2210	0.215	0.524	0.170	0.414	0.177	0.106	0.000	0.025	0.840
776	ES85.AFG	Sunbelt LowE on Blue	AFG Industries	IGDB v11.4	#		0.2290	0.215	0.521	0.140	0.433	0.168	0.109	0.000	0.032	0.840
777	ES86.AFG	Sunbelt LowE on Blue-Green	AFG Industries	IGDB v11.4	#		0.2220	0.262	0.516	0.151	0.602	0.172	0.172	0.000	0.034	0.840
778	ES87.AFG	Sunbelt LowE on Azurlite®	AFG Industries	IGDB v11.4	#		0.2230	0.216	0.515	0.094	0.566	0.169	0.149	0.000	0.034	0.840

FIGURE 17.29 Glass-type selection from WINDOW 5.

17.4.8.3.2 Energy Model Predictions

The window replacement was modeled by directly changing the window characteristics in the energy simulation model to match the specifications provided by WINDOW 5. The frame conductance was modified from U = 3.079 Btu/hr.ft^2.°F (eQUEST default value for unbroken aluminum frames) to U = 1.00 Btu/hr.ft^2.°F based on aluminum with a break as obtained from WINDOW 5 output. To model the reduced air infiltration, the infiltration flow area for all perimeter spaces was set to be 0.0937 cfm/sqft (a 75 percent reduction in infiltration from the original value of 0.375 cfm/sqft). The core and plenum infiltration rates were unchanged as they would not be affected by a window upgrade.

The energy use savings predicted by the energy simulation model due to replacing the windows are 952 MMBtu/year (16.9 percent) in steam use and an 11,565 kWh/year (1.5 percent) in electricity use.

17.4.8.3.3 Simplified Calculations

Window improvements to the space result in better building envelope performance through a combination of improved thermal resistance and reduction in air infiltration. The potential energy savings from improved thermal resistance was estimated using the degree-day method as was the case for the roof insulation addition (ECM-2). In this case the original U value for the window was 2.38 and the new window U value was 0.728. The total window area was estimated to be 2,480 ft^2. The improvement saves 674 MMBtu/year and 3,400 kWh/year.

The same degree-day method was utilized to determine the savings from reduction in the air infiltration rate. The air infiltration reduction was estimated to be from 0.375 cfm/ft^2 to 0.0937 cfm/ft^2 in the perimeter zones. The area of perimeter zones was estimated to be 17,251 ft^2 resulting in a total reduction of air infiltration of 4,852 cfm. The energy savings associated with reduction in the air infiltration rate are estimated 707 MMBtu/year for annual steam use and 3,564 kWh/year for annual electricity use.

The total annual energy savings due to replacing the windows are 1,381 MMBtu/year (24.5 percent) for steam use and 6,964 kWh/year (0.9 percent) for electricity use.

17.4.8.3.4 Discussion

The resulting energy savings from both methods are fairly comparable. The discrepancy is likely a result of the energy model's ability to capture the dynamic effects of heat transmission through the building envelope.

17.4.8.4 ECM 4: Occupancy Sensors

Due to the intermittent occupancy of several spaces within the building such as classrooms and storage and office rooms, it would be beneficial for the building to install occupancy sensors in all multioccupant spaces to better control lighting. The impact of occupancy sensors is assumed to be a 10 percent reduction in lighting power density as specified by Appendix G of ASHRAE 90.1 (ASHRAE, 2004).

17.4.8.4.1 Implementation and Cost

This measure can be implemented by installing occupancy sensors on the light switches for all offices, classrooms, and storage areas. This would require installing new switches. The installed cost of infrared ceiling mounted occupancy sensors is estimated to be $185.50/unit. With 52 units, the total implementation cost would be $9,620 (Means, 2008).

17.4.8.4.2 Energy Model Predictions

To model this measure, the lighting power density was reduced by 10 percent from the base case using the eQUEST detailed mode spreadsheet for each applicable space. As a consequence of this measure, the annual electricity use is reduced by 68,619 kWh/year (8.9 percent) and the steam use is decreased by 146.5 MMBtu/year (2.6 percent).

17.4.8.4.3 Simplified Calculations

The same calculation approach outlined for energy savings attributed to the delamping measure (ECM-1) was used with a 10 percent reduction in lighting load. The energy savings are estimated to be 30,600 kWh, a 6.0 percent reduction. As in the delamping measure, the effect of light reduction on heating load was not considered.

17.4.8.4.4 Discussion

The model predicts savings of twice that of the hand calculations. This is likely due to the energy simulation model's more accurate accounting of schedules and overall lighting usage. In addition, the energy model's ability to account for heating energy increase due to lighting reduction is valuable and may offset savings in electricity use.

17.4.8.5 ECM 5: Premium Efficiency Pumps

Inasmuch as the pumps are used to circulate hot and cold water, it could be beneficial to upgrade the motors of these systems from standard efficiency to premium efficiency. As per the mechanical schedules the current pumps have low efficiencies.

17.4.8.5.1 Implementation and Costs

Using Means' data, the cost data, and a 120 percent premium for efficient pumps, the heating coil pumps were assumed to cost $4,917, installed (Means, 2008). The cooling coil pump cost was estimated by examining several available retail pumps and applying a cost premium and labor allowance of \$3,120.

17.4.8.5.2 Energy Model Predictions

A parametric component was added in the calibrated building energy model that increased the motor class of the HVAC pumps from standard to premium. This measure resulted in annual electricity energy savings of 19,275 kWh/year (2.5 percent). There was no change to steam use.

17.4.8.5.3 Simplified Calculations

Per the mechanical drawings the cooling coil pump is a 3-hp pump that is 65 percent efficient and the two heating coil pumps are 2 hp and 57 percent efficient. The new premium efficiency pumps will have 91 percent efficiency. Based on the length of both winter and summer seasons for Denver, Colorado, it was assumed that the heating coil pumps are operating 35.2 weeks per year and the cooling coil pump is operating 29.4 weeks per year. It was assumed that the pumps run 24 hours a day and have a load factor of unity. Using Eqs. (17.6) and (17.7) as outlined in Chapter 5, the heating coil pumps can save 11,566 kWh/year and the cooling coil pump can save 4,859 kWh/year resulting in a total savings of 16,524 kWh/year (2.1 percent).

$$\Delta P_r = P_m\left(\frac{1}{\eta_e} - \frac{1}{\eta_r}\right) \tag{17.6}$$

$$\Delta kWh = 0.746 * \Delta P_r[hp] * N_h * LF_m \tag{17.7}$$

17.4.8.5.4 Discussion

The results from both calculation methods are quite similar. However, the energy simulation model takes better accounting of energy demands and actual pump operating hours.

17.4.8.6 ECM 6: Improved Fume Hood Controls—Demand-Controlled Ventilation

Fume hoods are typically a significant part of the overall energy use in lab facilities. Recent developments in improved technology can provide increased control over the duration of use for the fume hoods. For instance, an Aircuity system is able to sense contaminants and moderate hoods rather than run them continuously at high exhaust rates. From Aircuity promotional documents: "...with minimum air change rates, rather than hood makeup air or thermal load requirements, this concept for dynamically varying air change rates should significantly increase the energy efficiency of both new and existing facilities. Using this strategy, we can maintain occupant safety while furthering the goals of sustainable laboratory design" (Sharp, 2010). This system operates by continuously sampling and analyzing laboratory air for the presence of key parameters (i.e., chemicals or particulates). If contaminants are present, the system ramps up the exhaust fume hood and if the air is "clean", the system is ramped down to a minimal rate of 4–6 ACH. A representative for Aircuity recommended using particulate, VOC, and CO sensors for this building's laboratory spaces. Several buildings on the university campus are considering such technology.

17.4.8.6.1 Implementation and Costs

The proposed hoods can sense air quality and control fume hood dampers to regulate air flow in the space. The exhaust fan can be upgraded to a variable-speed fan. Based on discussions with the manufacturer's representative, it is estimated that the air flow to the space can be reduced by 75 percent. The cost of this measure was provided by Aircuity and estimated to be \$40,000 with annual fees of \$3,500 to keep the system properly maintained.

17.4.8.6.2 Energy Model Predictions

The ventilation air volume was reduced by 75 percent for all of the fume hoods in the energy simulation model (a reduction of 16 ACH to 4 ACH). This measure resulted in annual electricity use savings of 44,795 kWh/year (5.8 percent) and annual steam use reduction of 755 MMBtu/year (13.4 percent).

17.4.8.6.3 Simplified Calculations

With a 75 percent reduction the flow was reduced from 11,080 cfm to 8,310 cfm. The original system had a 10-hp motor. The heating savings were calculated using Eq. (17.8) as outlined in Chapter 7. The parameters $T_{out}$and $N_{wk,winter}$ were set to be 35.32°F and 29.4°F, respectively. The indoor temperature was set at 70°F and the system ran 168 hours/week. This resulted in savings of 527 MMBtu/year (9.4 percent).

$$\Delta FU = \frac{\Delta cfm * 0.886 * (T_{in} - T_{out}) * N_{hrs,wk} * N_{wk,winter}}{\eta} \tag{17.8}$$

The fan power saved was found using Eqs. (17.9) and (17.10). The Δhp was found to be 5.78 hp. Assuming that the fans run for 168 hours per week, the resulting energy savings are 37,772 kWh/year (4.9 percent).

$$\frac{HP_{new}}{HP_{old}} = \left(\frac{CFM_{new}}{CFM_{old}}\right)^3, \Delta HP = HP_{old} - HP_{new} \tag{17.9}$$

$$\Delta kWh = 0.746 * \Delta HP[hp] * N_h \tag{17.10}$$

17.4.8.6.4 Discussion

The building energy simulation model predicted a larger reduction in heating energy and electrical energy use savings. The heating energy use difference is a result of the energy model taking into account actual hourly weather variations as well as fluctuations in indoor temperature. The lower electrical reduction predicted by the simplified analysis is attributed to the fact that savings in cooling outdoor air intake from the fume hoods are not considered.

17.4.8.7 ECM 7: Improved Water Fixture Efficiency

The base-load steam use attributed to domestic hot water is rather high for this building; therefore improving fixture flow rates could have some significant impact in reducing the overall steam use.

17.4.8.7.1 Implementation and Costs

Several options exist including adding aerators to existing fixtures and low-flow valves to others. It was assumed that by implementing this measure savings of 30 percent in water usage could be achieved at a cost of roughly $30/fixture. Assuming roughly 30 fixtures are needed for this upgrade, the capital cost is estimated to be $900.

17.4.8.7.2 Energy Model Predictions

Using the detailed building energy simulation model, the DHW demand was reduced by 30 percent resulting in heating energy savings of 408.5 MMBtu (7.3 percent). There was no impact on electricity use.

17.4.8.7.3 Simplified Calculations

For the simplified analysis, 50 percent of the base-load steam use is attributed to DHW load because steam and hot water are also used for processing some of the lab equipment. The DHW load was reduced by 30 percent. In this case the base-load steam use due to DHW is reduced by 396 MMBtu or 7 percent.

17.4.8.7.4 Discussion

The simplified calculations predict savings similar to those predicted by the calibrated building energy simulation model.

17.4.8.8 ECM 8: Optimized Package of ECMs

This package includes all of the ECMs described and evaluated above except the roof insulation which proved to be not cost-effective and is not recommended as outlined in the economic analysis section.

17.4.8.8.1 Implementation and Costs

Implementation of this package of measures is described on an individual measure basis above. The total capital costs are summed to be $107,715.

17.4.8.8.2 Energy Model Predictions

Using the calibrated building energy simulation model, all of the recommended ECMs were modeled simultaneously. The resulting set of ECMs had an electricity savings of 217,190 kWh/year (28.2 percent) and steam energy savings of 814.3 MMBtu/year (14.5 percent).

17.4.8.8.3 Simplified Calculations

In the simplified analysis, no interactive effect was assumed between the various ECMs so the savings attributed to the combined set of ECMs (except ECM-2) were simply added. As a result, the simplified calculations indicated the total annual electricity savings of 168,360 kWh/year (21.9 percent) and annual steam use savings of 2,304 MMBtu/year (40.8 percent).

17.4.8.8.4 Discussion

It is clear that the two methodologies produce different results especially for the steam energy use savings. These differences are most likely attributed to the significant interactive effect that can be ignored as was assumed in the simplified analysis. The predictions of the building energy simulation model are likely more accurate because the simulation tool takes into account the interactive effects between various systems within the building. In the following economic analysis, only the results of the building energy model are considered.

17.4.9 Economic Analysis

An economic analysis was performed to determine the cost effectiveness of the various ECMs discussed above. In order to complete this analysis, the following basic parameters were defined: electricity was assumed to cost $0.12/kWh and steam to cost $24.29/MMBtu. These rates were provided by the university. Additionally, a life cycle of 25 years and a discount rate of 5 percent were assumed for the life-cycle cost and net present worth (NPW) analyses.

The results of the economic analysis are summarized in Table 17.22 using the simple payback period (SPP), life-cycle cost, and net present worth methods as defined in Chapter 2. Aside from increasing ceiling insulation (ECM-2), which, with a simple payback period of 37 years is not cost-effective, all of the other ECMs considered are economically viable. As indicated by Table 17.22, delamping (ECM-1) has an

TABLE 17.22 Economic Analysis of All Considered ECMs for the Museum Building

Item		Base Case	ECM1: Delamping	ECM2: Increase Ceiling Insulation	ECM3: Window Replacement	ECM4: Occupancy Sensors	ECM5: Premium Efficiency Pumps	ECM6: Fume Hood Sensors	ECM7: Reduce DHW Demand	ECM8: Optimized Package of ECMs
Initial cost	IC	$0.00	$0.00	$14,142.00	$124,740.00	$9,620.00	$8,037.00	$40,000.00	$900.00	$182,397.00
Annual energy costs	Electricity (kWh)	771,000	670,770	771,000	759,435	702,381	751,725	726,205	771,000	553809
	Electricity cost	92,520	80,492	92,520	91,132	84,286	90,207	87,145	92,520	66457
	Gas (MMBtu)	5,635	5,860	5,619	4,683	5,782	5,635	4,880	5,226	4821
	Gas cost	136,311	141,763	135,926	113,274	13,9855	136,311	11,8045	126,428	116614
	Total cost	228,831	222,255	228,446	204,406	224,140	226,518	208,690	218,948	183071
	Annual savings	NA	6,575	385	24,424	4,690	2,313	20,141	9,883	45760
	% Annual savings	NA	0	0	0	0	0	0	0	0
Economic indices	SPP (years)	NA	0.00	36.77	5.11	2.05	3.47	1.99	0.09	3.99
	USPW	14	14	14	14	14	14	14	14	14
	LCC	3,225,126	3,132,456	3,233,848	3,005,632	3,168,643	3,200,564	2,981,260	3,086,743	2762588
	NPW	NA	92,670	-8,721	219,495	56,483	24,562	243,867	138,384	462,539

TABLE 17.23 Recommended ECM Implementation Ranking Based on NPW and LCC Analysis

Implementation ECM Rank	ECM Description
1	ECM-6: Fume hood sensors
2	ECM-3: Window replacement
3	ECM-7: Low DHW flow devices
4	ECM-1: Delamping
5	ECM-4: Occupancy sensors
6	ECM-5: Premium efficiency pumps

immediate payback because it does not require any capital costs. Using LCC or NPW as the economic indicator, Table 17.23 shows a prioritized ranking of implementing the different measures. Generally for a measure to be cost-effective, the SPP should be less than 5 years, LCC should be lower than the base-case LCC value, and the NPW should be positive. As indicated in Table 17.22, implementing all ECMs (except ECM-2) would provide the best savings over the life cycle considered in the analysis (i.e., 25 years) and is highly recommended.

17.5 Summary and Recommendations

After the completion of a standard energy audit of the Museum Collections building, it was found that several opportunities exist to reduce energy use and cost. Specifically, six energy conservation measures were found to be financially attractive. In order of priority of recommended implementation, the measures are:

1. Replace windows.
2. Install fume hood sensors and controls.
3. Reduce DHW demand.
4. Remove unnecessary lamps.
5. Install occupancy sensors.
6. Install premium efficiency pumps.

As a package, all the above six measures can provide 217,190 kWh/year (28.2 percent) of electricity savings and 814.3 MMBtu/year (14.5 percent) of steam use savings at a cost of implementation estimated to be $107,715 and result in a payback period of 4 years and a total savings of $462,539 over a 25-year period.

Appendix A: Conversion Factors

Conversion Factors (Metric to English)

Area	1 m^2	= 1550.0 in^2
		= 10.764 ft^2
Energy	1 J	= 9.4787×10^{-4} Btu
Heat transfer rate	1 W	= 3.4123 Btu/h
Heat flux	1 W/m^2	= 0.3171 Btu/hr×ft^2
Heat generation rate	1 W/m^3	= 0.09665 Btu/hr×ft^3
Heat transfer coefficient	1 W/m^2×K	= 0.17612 Btu/hr×ft^2×°F
Latent heat	1 J/kg	= 4.2995×10^{-4} Btu/lb_m
Length	1 m	= 3.2808 ft
	1 km	= 0.62137 mile
Mass	1 kg	= 2.2046 lb_m
Mass density	1 kg/m3	= 0.062428 lb_m/ft^3
Mass flow rate	1 kg/s	= 7936.6 lb_m/h
Mass transfer coefficient	1 m/s	= 1.1811×10^4 ft/h
Pressure	1 N/m^2	= 0.020886 lb_f/ft^2
		= 1.4504×10^{-4} lb_f/in.^{2}
		= 4.015×10^{-3} in. water
Power	1 kW	= 1.340 hp
		= 3,412 Btu/hr
	1×10^5 N/m^2	= 1 bar
Refrigeration capacity	1 kJ/hr	= 94,782 Btu/hr
		= 7.898×10^{-5} ton
	1 kW	= 0.2844 ton
Specific heat	1 J/kg×K	= 2.3886×10^{-4} Btu/lb_m×°F
Temperature	K	= (5/9)°R
		= (5/9)(°F + 459.67)
		= °C + 273.15
Temperature difference	1 K	= 1°C
		= (9/5) °R = (9/5) °F
Thermal conductivity	1 W/m×K	= 0.57782 Btu/hr×ft×°F
Thermal resistance	1 K/W	= 0.52750°F/hr×Btu
Volume	1 m^3	= 6.1023×10^4 in^3
		= 35.314 ft^3
		= 264.17 gal
Volume flow rate	1 m^3/s	= 1.2713×10^5 ft^3/hr
	1 L/s	= 127.13 ft^3/hr
		= 2.119 ft^3/min

Appendix B: Weather Data

ANNUAL HEATING DEGREE DAYS (HDD), COOLING DEGREE DAYS (CDD), AND COOLING DEGREE HOURS (CDH) INTERNATIONAL SITES (SI UNITS)

Country	City	HDD Tb = 10°C	HDD Tb = 18.3°C	CDD Tb = 10°C	CDD Tb = 18.3°C	CDH Tb = 10°C	CDH Tb = 18.3°C
Albania	TIRANA	292	1644	2321	633	5927	2318
Algeria	ANNABA	25	924	2999	856	7302	2459
American Samoa	PAGO PAGO WSO AP	0	0	6442	3400	37333	11474
Antarctica	DAVIS	7436	10478	0	0	0	0
Antigua and Barbuda	VC BIRD INTL AIRPOR	0	0	6249	3208	32044	8623
Argentina	EZEIZA AERO	129	1211	2597	637	5565	1923
Armenia	YEREVAN/ YEREVAN-ARA	1211	2804	2191	744	8569	3923
Aruba	QUEEN BEATRIX AIRPO	0	0	6790	3748	41777	14922
Australia	CANBERRA AIRPORT	390	2113	1564	246	2823	1076
Austria	SALZBURG-FLUGHAFEN	1343	3372	1166	153	1382	371
Azerbaijan	LANKARAN	598	2107	2183	651	5013	1405
Bahamas	NASSAU AIRPORT NEW	0	9	5643	2609	23534	7304
Bahrain	BAHRAIN (INT. AIRPORT)	0	103	6153	3214	43452	26351
Barbados	GRANTLEY ADAMS	0	0	6308	3267	32611	8441
Belarus	MINSK	2154	4405	876	84	742	147
Belgium	BRUXELLES NATIONAL	864	2933	1069	96	823	200
Belize	BELIZE/PHILLIP GOLD	0	0	6145	3103	30986	10421
Benin	BOHICON	0	0	6416	3374	32351	12746
Bermuda	BERMUDA INTL	0	88	4596	1643	11676	2542
Bolivia	LA PAZ/ALTO	925	3941	27	0	0	0
Bosnia and Herzegovina	SARAJEVO-BJELAVE	1288	3186	1359	217	1938	596

(*Continued*)

Country	City	HDD Tb = 10°C	HDD Tb = 18.3°C	CDD Tb = 10°C	CDD Tb = 18.3°C	CDH Tb = 10°C	CDH Tb = 18.3°C
Botswana	SERETSE KHAMA INTER	5	435	3976	1364	14449	6227
Brazil	BRASILIA (AEROPORTO)	0	23	4364	1346	9734	2571
Bulgaria	SOFIA (OBSERV.)	1318	3169	1431	239	2431	784
Burkina Faso	OUAGADOUGOU	0	0	6844	3802	49567	27676
Cape Verde	SAL	0	0	5125	2083	11379	1826
Chad	NDJAMENA	0	1	6870	3829	50915	30309
Chile	PUERTO MONTT	462	2916	592	2	47	6
China	BEIJING	1310	2830	2370	848	7954	3031
Christmas Island	CHRISTMAS ISLAND AE	0	0	5414	2373	12402	643
Cocos (Keeling) Islands	COCOS ISLAND AERO	0	0	6203	3161	30969	5908
Colombia	BOGOTA/ ELDORADO	1	1752	1292	0	3	0
Congo	BRAZZAVILLE/ MAYA-M	0	0	5841	2799	21976	7507
Cook Islands	AMURI/AITUTAKI ISL	0	0	5858	2817	23341	3743
Costa Rica	JUAN SANTAMARIA INT	0	0	4867	1826	8564	1674
Cote d'Ivoire	ABIDJAN	0	0	6239	3197	30476	8942
Croatia	ZAGREB/MAKSIMIR	1107	2873	1576	301	2636	791
Cuba	AEROPUERTO JOSE MAR	0	21	5411	2391	19541	6592
Cyprus	PAPHOS AIRPORT	16	765	3198	906	7760	1927
Czech Rep.	PRAHA/RUZYNE	1565	3754	941	89	918	231
Dem. People's Rep. of Korea	PYONGYANG	1629	3298	1947	574	3808	917
Denmark	KOEBENHAVN/ KASTRUP	1357	3653	790	45	199	17
Diego Garcia	DIEGO GARCIA NAF	0	0	6427	3386	35509	9748
Dominican Rep.	SANTO DOMINGO	0	0	6023	2982	25939	8221
Ecuador	QUITO AEROPUERTO	0	1402	1640	1	4	1
Egypt	CAIRO AIRPORT	1	393	4416	1767	19113	9392
Estonia	TALLINN	2193	4649	617	31	244	32
Falkland Islands (Malvinas)	MOUNT PLEASANT AIRP	1477	4364	153	0	2	0
Faroe Islands	TORSHAVN	1291	4239	94	0	0	0
Fiji	NADI AIRPORT	0	0	5675	2633	22351	6795
Finland	HELSINKI-VANTAA	2411	4856	637	39	337	39
France	PARIS-AEROPORT CHAR	743	2649	1301	164	1404	413
French Polynesia	TAHITI-FAAA	0	0	6026	2984	27652	7815
Gabon	LIBREVILLE	0	0	6016	2974	27177	6108
Gambia	BANJUL/YUNDUM	0	1	6154	3113	28227	12066
Georgia	TBILISI	806	2371	2134	659	5657	2244

Country	City	HDD Tb = 10°C	HDD Tb = 18.3°C	CDD Tb = 10°C	CDD Tb = 18.3°C	CDH Tb = 10°C	CDH Tb = 18.3°C
Germany	BERLIN/DAHLEM	1270	3390	1039	118	979	234
Gibraltar	GIBRALTAR	3	620	3122	697	2989	654
Greece	ATHINAI (AIRPORT)	119	1165	3076	1079	10117	3957
Greenland	NUUK (GODTHAAB)	4173	7203	12	0	0	0
Grenada	POINT SALINES AIRPO	0	0	6378	3336	34507	9339
Guam	ANDERSEN AFB	0	0	6229	3187	30847	6544
Guatemala	GUATEMALA (AEROPUERTO)	0	71	3616	645	1874	125
Guernsey	GUERNSEY AIRPORT	426	2567	924	23	77	11
Guiana	ROCHAMBEAU	0	0	6052	3010	25655	7857
Guyana	TIMEHRI\CHEDDI JAG	0	0	6136	3094	25396	8681
Honduras	TEGUCIGALPA	0	12	4525	1495	9513	2407
Hungary	BUDAPEST/ FERIHEGY I	1322	3188	1433	258	2803	928
Iceland	REYKJAVIK	2049	4990	103	0	0	0
India	NEW DELHI/PALAM	1	286	5767	3011	42516	25343
Indonesia	JAKARTA/ SOEKARNO-HA	0	0	6439	3398	34051	12496
Iran (Islamic Rep. of)	TEHRAN-MEHRABAD	428	1588	3421	1540	20017	10807
Ireland	DUBLIN AIRPORT	717	3135	630	6	12	0
Isle of Man	ISLE OF MAN/ RONALDS	620	3079	586	4	11	0
Israel	BEN-GURION INT. AIR	10	619	3721	1289	12346	4565
Italy	ROMA FIUMICINO	201	1525	2273	555	4304	1022
Jamaica	KINGSTON/ NORMAN MAN	0	0	6608	3567	39379	14343
Japan	TOKYO	341	1611	2671	902	7421	2427
Jersey	JERSEY AIRPORT	484	2581	982	37	170	30
Jordan	AMMAN AIRPORT	184	1291	2974	1037	10217	4263
Kazakhstan	ASTANA	3571	5717	1101	206	2291	736
Kenya	NAIROBI/ KENYATTA AI	0	104	3459	523	3023	333
Kerguelen	PORT-AUX-FRANCAIS	1918	4925	36	0	0	0
Kiribati	TARAWA	0	0	6698	3656	42259	14305
Korea (Rep. of)	SEOUL	1183	2721	2202	699	4928	1318
Kuwait	KUWAIT INTERNATIONA	11	426	5987	3360	54703	38978
Kyrgyzstan	BISHKEK	1531	3218	1950	596	6961	3046
Latvia	RIGA	1910	4193	827	69	451	81
Lebanon	BEYROUTH (AEROPORT)	4	464	3966	1383	11850	3063

(*Continued*)

Country	City	HDD Tb = 10°C	HDD Tb = 18.3°C	CDD Tb = 10°C	CDD Tb = 18.3°C	CDH Tb = 10°C	CDH Tb = 18.3°C
Libya	TRIPOLI INTERNATION	8	668	4016	1633	18871	10227
Liechtenstein	VADUZ (LIECHTENSTEIN)	1147	3119	1239	169	1087	232
Lithuania	VILNIUS	2077	4361	830	72	624	124
Luxembourg	LUXEMBOURG/ LUXEMBOU	1217	3383	994	119	893	217
Macao	TAIPA GRANDE	8	309	4728	1987	18408	5667
Macedonia	SKOPJE- AIRPORT	1031	2653	1920	500	6006	2633
Madagascar	ANTANANARIVO/ IVATO	0	330	3356	643	3006	406
Malaysia	KUALA LUMPUR SUBANG	0	0	6601	3559	35356	13689
Maldives	MALE	0	0	6779	3737	45547	16941
Mali	BAMAKO/ SENOU	0	0	6572	3530	42710	23263
Malta	LUQA	11	782	3293	1024	7332	2244
Marshall Islands	MAJURO WSO AP	0	0	6547	3505	39609	11397
Martinique	LE LAMENTIN	0	0	6087	3045	27854	7906
Mauritania	NOUAKCHOTT	0	2	5998	2959	29233	13642
Mauritius	VACOAS (MAURITIUS)	0	25	4245	1229	4048	302
Mayotte	DZAOUDZI/ PAMANZI	0	0	6102	3060	29903	7871
Mexico	AEROP. INTERNACIONA	2	563	2670	190	1905	241
Micronesia	YAP ISLAND WSO AP	0	0	6458	3416	35953	10617
Moldova	KISINEV	1511	3337	1541	325	2332	631
Mongolia	ULAANBAATAR	4546	6964	701	79	1016	303
Morocco	RABAT-SALE	6	800	2780	533	2858	722
Mozambique	MAPUTO/ MAVALANE	0	22	4958	1938	14446	4288
Namibia	WINDHOEK	8	379	3719	1049	9511	3222
Netherlands	AMSTERDAM AP SCHIPH	883	3038	951	65	486	103
New Caledonia	NOUMEA (NLLE-CALEDONIE)	0	2	4849	1810	10103	1880
New Zealand	WELLINGTON AIRPORT	135	1877	1346	47	42	1
Nicaragua	MANAGUA A.C.SANDINO	0	0	6457	3416	35517	15527
Niger	NIAMEY-AERO	0	0	7192	4151	57896	34902
Niue Is.	ALOFI	0	0	5438	2397	16208	3099
Norfolk Island	NORFOLK ISLAND AERO	0	308	3232	499	408	2
Northern Mariana Islands	SAIPAN	0	0	6541	3499	36653	10852
Norway	OSLO/ GARDERMOEN	2447	4982	528	21	232	26

Country	City	HDD Tb = 10°C	HDD Tb = 18.3°C	CDD Tb = 10°C	CDD Tb = 18.3°C	CDH Tb = 10°C	CDH Tb = 18.3°C
Oman	SEEB INTL AIRPORT	0	1	6692	3652	48896	28848
Pakistan	ISLAMABAD AIRPORT	16	652	4388	1982	25718	13954
Palau	KOROR WSO	0	0	6523	3481	36552	10433
Panama	MARCOS A GELABERT I	0	0	6621	3579	36052	12472
Paraguay	ASUNCION/ AEROPUERTO	6	254	4841	2049	20329	8939
Peru	LIMA-CALLAO/ AEROP	0	165	3698	822	2619	344
Philippines	MANILA	0	0	6737	3695	43723	16608
Poland	WARSZAWA-OKECIE	1637	3771	1021	112	1079	266
Portugal	LISBOA/PORTELA	35	1012	2664	599	4084	1513
Puerto Rico	SAN JUAN INTL ARPT	0	0	6159	3118	29006	8330
Qatar	DOHA INTERNATIONAL	0	73	6496	3527	49126	31403
Reunion Is.	SAINT-DENIS/ GILLOT	0	0	5154	2113	13305	2522
Romania	BUCURESTI AFUMATI	1332	3069	1686	382	4134	1507
Russia	MOSKVA	2502	4747	903	107	759	145
Saint Helena Is.	ST. HELENA IS.	0	309	3057	326	48	1
Saint Lucia	HEWANORRA INTL AIRP	0	0	6429	3388	36468	10501
Samoa	APIA	0	0	6160	3118	30271	8176
Saudi Arabia	RIYADH OBS. (O.A.P.)	6	301	6009	3264	51690	35063
Senegal	DAKAR/YOFF	0	1	5363	2322	17989	4546
Serbia	BEOGRAD	961	2558	1944	498	4316	1583
Seychelles	SEYCHELLES INTERNAT	0	0	6329	3288	33339	8399
Singapore	SINGAPORE/ CHANGI AI	0	0	6579	3537	37142	12225
Slovakia	BRATISLAVA-LETISKO	1251	3099	1458	265	2533	813
Slovenia	LJUBLJANA/ BEZIGRAD	1123	2953	1482	269	2178	636
Solomon Islands	HONIARA/ HENDERSON	0	0	6048	3007	26449	8396
South Africa	PRETORIA (IRENE)	23	811	2728	473	3217	666
Spain	MADRID/BARAJAS RS	451	2023	2084	612	8047	3864
Sri Lanka	KATUNAYAKE	0	0	6454	3412	36566	12590
Suriname	ZANDERIJ	0	0	6264	3222	26783	10101
Svalbard and Jan Mayen	HOPEN	5460	8502	0	0	0	0
Sweden	STOCKHOLM/ ARLANDA	2015	4409	692	46	414	69

(*Continued*)

Country	City	HDD Tb = 10°C	HDD Tb = 18.3°C	CDD Tb = 10°C	CDD Tb = 18.3°C	CDH Tb = 10°C	CDH Tb = 18.3°C
Switzerland	ZURICH-KLOTEN	1240	3303	1112	132	1334	358
Syria	DAMASCUS INT. AIRPO	298	1527	2874	1060	14780	8331
Taiwan	TAIBEI	1	237	4850	2044	19554	7584
Tajikistan	DUSHANBE	603	1941	2655	952	12143	6284
Tanzania	DAR ES SALAAM AIRPO	0	0	5851	2809	24708	8192
Thailand	BANGKOK METROPOLIS	0	0	6915	3873	46537	20777
Togo	LOME	0	0	6356	3314	33283	10981
Tonga	FUAAMOTU	0	3	5001	1963	11098	1666
Tunisia	TUNIS-CARTHAGE	19	814	3432	1186	11128	4799
Turkey	ESENBOGA	1373	3299	1344	227	3548	1271
Turkmenistan	ASHGABAT KESHI	633	1909	3221	1454	19659	11118
Tuvalu	FUNAFUTI NF	0	0	6694	3653	42887	14536
Ukraine	KYIV	1878	3907	1194	180	1249	272
United Arab Emirates	ABU DHABI INTER. AI	0	30	6577	3565	49047	31114
United Arab Emirates	DUBAI INTERNATIONAL	0	24	6461	3442	47226	29061
United Kingdom	LONDON WEATHER CENT	464	2344	1291	129	694	155
United States Minor Outlying Islands	MIDWAY ISLAND NAS	0	28	4867	1854	12637	2543
Uruguay	CARRASCO	98	1221	2379	461	2736	719
Uzbekistan	TASHKENT	780	2162	2679	1019	13323	7046
Vanuatu	ANEITYUM	0	0	5002	1961	12066	1685
Venezuela	CARACAS/ MAIQUETIA A	0	0	6284	3242	30706	9906
Vietnam	HA NOI	0	168	5223	2348	23638	9309
Wallis and Futuna	HIHIFO (ILE WALLIS)	0	0	6291	3249	33102	8432
Zimbabwe	HARARE (KUTSAGA)	0	348	3424	731	4586	889

ANNUAL HEATING DEGREE DAYS (HDD), COOLING DEGREE DAYS (CDD), AND COOLING DEGREE HOURS (CDH) US SITES (SI UNITS)

State	City	HDD Tb = 10°C	HDD Tb = 8.3°C	CDD Tb = 10°C	CDD Tb = 18.3°C	CDH Tb = 10°C	CDH Tb = 18.3°C
AK	JUNEAU INT'L ARPT	1937	4629	352	2	49	3
AL	MONTGOMERY DANNELLY FIELD	252	1191	3372	1268	12048	5233
AR	LITTLE ROCK ADAMS FIELD	489	1653	3058	1180	12054	5413
AZ	PHOENIX SKY HARBOR INTL AP	17	523	5067	2532	38386	24726
CA	SACRAMENTO METROPOLITAN AP	179	1327	2676	782	9969	5181

State	City	HDD Tb = 10°C	HDD Tb = 8.3°C	CDD Tb = 10°C	CDD Tb = 18.3°C	CDH Tb = 10°C	CDH Tb = 18.3°C
CO	DENVER INTL AP	1434	3301	1606	432	5288	2342
CT	HARTFORD BRADLEY INTL AP	1499	3329	1621	411	3772	1299
DE	DOVER AFB	930	2509	2116	653	5314	1804
FL	TALLAHASSEE REGIONAL AP	128	852	3741	1424	13428	5569
GA	ATLANTA HARTSFIELD INTL AP	381	1497	2949	1023	8753	3266
HI	HONOLULU INTL ARPT	0	0	5624	2583	19731	5259
IA	DES MOINES INTL AP	1734	3467	1887	578	5294	1929
ID	BOISE AIR TERMINAL	1295	3143	1687	494	6908	3416
IL	SPRINGFIELD CAPITAL AP	1382	3016	2038	631	6112	2255
IN	INDIANAPOLIS INTL AP	1315	2957	1986	586	5003	1647
KS	TOPEKA FORBES FIELD	1198	2773	2202	736	7287	3069
KY	LOUISVILLE STANDIFORD FIELD	888	2316	2445	831	7694	2883
LA	BATON ROUGE RYAN ARPT	152	894	3772	1474	13556	5429
MA	BOSTON LOGAN INT’L ARPT	1297	3123	1632	417	3116	1045
MD	BALTIMORE BLT-WASHNGTN INT’L	959	2537	2145	682	6287	2397
ME	AUGUSTA AIRPORT	2059	4097	1232	228	1705	449
MI	LANSING CAPITAL CITY ARPT	1884	3827	1416	317	3017	895
MN	ST PAUL DOWNTOWN AP	2271	4183	1500	372	3103	971
MO	JEFFERSON CITY MEM	1008	2508	2309	767	7604	3049
MS	JACKSON INTERNATIONAL AP	309	1284	3326	1258	11998	5184
MT	HELENA REGIONAL AIRPORT	2137	4266	1119	208	3138	1301
NC	RALEIGH DURHAM INTERNATIONAL	551	1846	2627	877	7764	3036
ND	BISMARCK MUNICIPAL ARPT	2658	4706	1293	299	3857	1598
NE	LINCOLN MUNICIPAL ARPT	1616	3329	1987	658	7293	3206
NH	CONCORD MUNICIPAL ARPT	1973	3989	1281	256	2848	931
NJ	TRENTON MERCER COUNTY AP	1151	2858	1883	548	4887	1763
NM	ALBUQUERQUE INTL ARPT	762	2261	2293	749	7964	3439
NV	LAS VEGAS MCCARRAN INTL AP	176	1169	3908	1860	27352	16813
NY	ALBANY COUNTY AP	1773	3671	1472	329	2798	814
NY	NEW YORK J F KENNEDY INT’L AR	1007	2682	1911	543	3369	909
OH	COLUMBUS PORT COLUMBUS INTL A	1290	2957	1914	539	4783	1606

(*Continued*)

State	City	HDD Tb = 10°C	HDD Tb = 8.3°C	CDD Tb = 10°C	CDD Tb = 18.3°C	CDH Tb = 10°C	CDH Tb = 18.3°C
OK	OKLAHOMA CITY WILL ROGERS WOR	663	1953	2822	1070	11496	5467
OR	SALEM MCNARY FIELD	626	2542	1289	162	2603	1021
PA	HARRISBURG CAPITAL CITY ARPT	1211	2904	1899	550	4863	1724
RI	PROVIDENCE T F GREEN STATE AR	1293	3106	1633	405	3004	919
SC	COLUMBIA METRO ARPT	332	1405	3139	1171	11588	5059
SD	PIERRE MUNICIPAL AP	2071	3948	1684	520	6404	3117
TN	NASHVILLE INTERNATIONAL AP	666	1968	2676	935	8907	3522
TX	AUSTIN/BERGSTROM	167	919	3949	1661	17897	8648
UT	SALT LAKE CITY INT'L ARPT	1298	3067	1935	663	7963	3837
VA	VIRGINIA TECH ARPT	1036	2689	1798	409	3250	881
VA	WASHINGTON DC REAGAN AP	772	2223	2439	847	7497	2825
VT	MONTPELIER AP	2413	4565	1036	146	1355	294
WA	OLYMPIA AIRPORT	822	2984	937	56	1345	422
WI	MADISON DANE CO REGIONAL ARPT	2064	3998	1447	338	3160	954
WV	CHARLESTON YEAGER ARPT	947	2468	2113	592	5132	1726
WY	CHEYENNE MUNICIPAL ARPT	1866	3971	1117	180	2523	818

ANNUAL HEATING DEGREE DAYS (HDD), COOLING DEGREE DAYS (CDD), AND COOLING DEGREE HOURS (CDH) CANADIAN SITES (SI UNITS)

State	City	HDD Tb=10°C	HDD Tb=18.3°C	CDD Tb=10°C	CDD Tb=18.3°C	CDH Tb=10°C	CDH Tb=18.3°C
AB	CALGARY INT'L A	2655	5086	648	37	727	166
AB	EDMONTON CITY CENTRE A	2947	5275	778	63	639	125
BC	VANCOUVER INT'L A	801	2932	951	41	146	11
BC	VICTORIA INT'L A	779	3022	821	22	316	55
MB	WINNIPEG RICHARDSON INT'L A	3552	5750	1011	168	1723	490
NB	FREDERICTON A	2497	4692	979	132	1219	304
NF	ST JOHN'S A	2381	4907	543	28	109	6
NS	HALIFAX STANFIELD INT'L A	2142	4356	927	98	555	92
NT	YELLOWKNIFE A	5714	8306	482	33	170	18
NU	IQALUIT A	7056	10076	23	0	1	0
ON	OTTAWA MACDONALD-CARTIER INT'	2517	4563	1233	236	1819	476
ON	TORONTO LESTER B. PEARSON INT	1954	3956	1316	276	2190	612
PE	CHARLOTTETOWN A	2438	4703	873	94	406	43

State	City	HDD Tb=10°C	HDD Tb=18.3°C	CDD Tb=10°C	CDD Tb=18.3°C	CDH Tb=10°C	CDH Tb=18.3°C
QC	MONTREAL/MIRABEL INT'L A	2710	4849	1064	162	1291	273
QC	QUEBEC/JEAN LESAGE INTL A	2891	5094	970	132	988	201
QC	SHERBROOKE A	2803	5058	880	93	956	173
SK	REGINA A	3436	5701	902	126	1774	581
SK	SASKATOON DIEFENBAKER INT'L A	3558	5861	844	105	1488	462
YT	WHITEHORSE A	4070	6803	313	6	197	36

ANNUAL HEATING DEGREE DAYS (HDD), COOLING DEGREE DAYS (CDD), AND COOLING DEGREE HOURS (CDH) INTERNATIONAL SITES (IP UNITS)

Country	City	HDD Tb=50°F	HDD Tb=65°F	CDD Tb=50°F	CDD Tb=65°F	CDH Tb=74°F	CDH Tb=80°F
Albania	TIRANA	525	2959	4178	1140	10669	4173
Algeria	ANNABA	45	1663	5399	1541	13143	4427
American Samoa	PAGO PAGO WSO AP	0	0	11595	6120	67200	20654
Antarctica	DAVIS	13385	18860	0	0	0	0
Antigua and Barbuda	VC BIRD INTL AIRPOR	0	0	11249	5774	57679	15522
Argentina	EZEIZA AERO	232	2180	4674	1147	10017	3462
Armenia	YEREVAN/YEREVAN-ARA	2179	5047	3944	1340	15424	7062
Aruba	QUEEN BEATRIX AIRPO	0	0	12222	6747	75199	26859
Australia	CANBERRA AIRPORT	702	3804	2816	443	5082	1936
Austria	SALZBURG-FLUGHAFEN	2417	6069	2098	275	2488	667
Azerbaijan	LANKARAN	1076	3792	3930	1172	9024	2529
Bahamas	NASSAU AIRPORT NEW	0	16	10157	4697	42361	13148
Bahrain	BAHRAIN (INT. AIRPORT)	0	186	11075	5785	78213	47431
Barbados	GRANTLEY ADAMS	0	0	11355	5880	58700	15194
Belarus	MINSK	3877	7929	1577	151	1335	265
Belgium	BRUXELLES NATIONAL	1555	5279	1924	173	1482	360
Belize	BELIZE/PHILLIP GOLD	0	0	11061	5586	55774	18757
Benin	BOHICON	0	0	11549	6074	58232	22942
Bermuda	BERMUDA INTL	0	159	8273	2957	21016	4576
Bolivia	LA PAZ/ALTO	1665	7093	48	0	0	0
Bosnia and Herzegovina	SARAJEVO-BJELAVE	2319	5735	2447	390	3488	1072
Botswana	SERETSE KHAMA INTER	9	783	7156	2455	26009	11209
Brazil	BRASILIA (AEROPORTO)	0	42	7855	2422	17521	4627
Bulgaria	SOFIA (OBSERV.)	2372	5704	2575	430	4376	1412

(*Continued*)

Country	City	HDD Tb=50°F	HDD Tb=65°F	CDD Tb=50°F	CDD Tb=65°F	CDH Tb=74°F	CDH Tb=80°F
Burkina Faso	OUAGADOUGOU	0	0	12319	6844	89220	49816
Cape Verde	SAL	0	0	9225	3750	20482	3287
Chad	NDJAMENA	0	2	12366	6893	91647	54556
Chile	PUERTO MONTT	832	5249	1066	4	84	11
China	BEIJING	2358	5094	4266	1527	14318	5456
Christmas Island	CHRISTMAS ISLAND AE	0	0	9746	4271	22323	1158
Cocos (Keeling) Islands	COCOS ISLAND AERO	0	0	11165	5690	55745	10635
Colombia	BOGOTA/ELDORADO	2	3153	2326	0	5	0
Congo	BRAZZAVILLE/ MAYA-M	0	0	10514	5039	39556	13512
Cook Islands	AMURI/AITUTAKI ISL	0	0	10545	5070	42013	6738
Costa Rica	JUAN SANTAMARIA INT	0	0	8761	3286	15415	3014
Cote d'Ivoire	ABIDJAN	0	0	11230	5755	54857	16096
Croatia	ZAGREB/MAKSIMIR	1993	5171	2837	542	4745	1423
Cuba	AEROPUERTO JOSE MAR	0	37	9740	4304	35173	11865
Cyprus	PAPHOS AIRPORT	28	1377	5756	1630	13968	3469
Czech Rep.	PRAHA/RUZYNE	2817	6757	1694	160	1652	415
Dem. People's Rep. of Korea	PYONGYANG	2933	5937	3504	1034	6855	1650
Denmark	KOEBENHAVN/ KASTRUP	2442	6575	1422	81	359	31
Diego Garcia	DIEGO GARCIA NAF	0	0	11569	6094	63917	17547
Dominican Rep.	SANTO DOMINGO	0	0	10842	5367	46690	14798
Ecuador	QUITO AEROPUERTO	0	2523	2952	1	8	1
Egypt	CAIRO AIRPORT	2	708	7949	3180	34403	16906
Estonia	TALLINN	3948	8369	1110	55	439	58
Falkland Islands (Malvinas)	MOUNT PLEASANT AIRP	2659	7855	276	0	3	0
Faroe Islands	TORSHAVN	2324	7630	169	0	0	0
Fiji	NADI AIRPORT	0	0	10215	4740	40232	12231
Finland	HELSINKI-VANTAA	4340	8740	1146	71	607	71
France	PARIS-AEROPORT CHAR	1338	4768	2342	295	2528	744
French Polynesia	TAHITI-FAAA	0	0	10846	5371	49774	14067
Gabon	LIBREVILLE	0	0	10829	5354	48918	10995
Gambia	BANJUL/YUNDUM	0	1	11078	5604	50808	21719
Georgia	TBILISI	1450	4267	3841	1186	10182	4039
Germany	BERLIN/DAHLEM	2286	6102	1871	213	1763	422
Gibraltar	GIBRALTAR	5	1116	5619	1255	5380	1177
Greece	ATHINAI (AIRPORT)	215	2097	5536	1942	18210	7123
Greenland	NUUK (GODTHAAB)	7512	12965	21	0	0	0
Grenada	POINT SALINES AIRPO	0	0	11480	6005	62113	16810
Guam	ANDERSEN AFB	0	0	11212	5737	55524	11780
Guatemala	GUATEMALA (AEROPUERTO)	0	128	6509	1161	3374	225

Country	City	HDD Tb=50°F	HDD Tb=65°F	CDD Tb=50°F	CDD Tb=65°F	CDH Tb=74°F	CDH Tb=80°F
Guernsey	GUERNSEY AIRPORT	766	4621	1663	42	139	19
Guiana	ROCHAMBEAU	0	0	10893	5418	46179	14143
Guyana	TIMEHRI\CHEDDI JAG	0	0	11044	5569	45713	15625
Honduras	TEGUCIGALPA	0	21	8145	2691	17123	4333
Hungary	BUDAPEST/FERIHEGY I	2379	5739	2579	464	5045	1670
Iceland	REYKJAVIK	3689	8982	185	0	0	0
India	NEW DELHI/PALAM	2	515	10381	5419	76529	45618
Indonesia	JAKARTA/ SOEKARNO-HA	0	0	11591	6116	61291	22492
Iran (Islamic Rep. of)	TEHRAN-MEHRABAD	770	2858	6158	2772	36030	19452
Ireland	DUBLIN AIRPORT	1290	5643	1134	11	22	0
Isle of Man	ISLE OF MAN/ RONALDS	1116	5543	1055	8	19	0
Israel	BEN-GURION INT. AIR	18	1115	6697	2320	22223	8217
Italy	ROMA FIUMICINO	362	2745	4092	999	7748	1840
Jamaica	KINGSTON/NORMAN MAN	0	0	11895	6420	70882	25818
Japan	TOKYO	614	2900	4808	1623	13358	4368
Jersey	JERSEY AIRPORT	872	4645	1768	66	306	54
Jordan	AMMAN AIRPORT	332	2324	5353	1866	18390	7673
Kazakhstan	ASTANA	6427	10291	1981	371	4123	1324
Kenya	NAIROBI/ KENYATTA AI	0	187	6227	942	5441	600
Kerguelen	PORT-AUX-FRANCAIS	3453	8865	64	0	0	0
Kiribati	TARAWA	0	0	12056	6581	76066	25749
Korea (Rep. of)	SEOUL	2130	4897	3964	1258	8870	2372
Kuwait	KUWAIT INTERNATIONA	19	766	10777	6048	98465	70161
Kyrgyzstan	BISHKEK	2756	5793	3510	1072	12529	5483
Latvia	RIGA	3438	7548	1489	124	812	146
Lebanon	BEYROUTH (AEROPORT)	8	835	7139	2490	21330	5513
Libya	TRIPOLI INTERNATION	15	1202	7228	2940	33967	18408
Liechtenstein	VADUZ (LIECHTENSTEIN)	2065	5615	2230	305	1956	417
Lithuania	VILNIUS	3739	7849	1494	130	1124	223
Luxembourg	LUXEMBOURG/ LUXEMBOU	2190	6089	1790	215	1608	390
Macao	TAIPA GRANDE	14	556	8511	3577	33134	10201
Macedonia	SKOPJE- AIRPORT	1856	4776	3456	900	10811	4740
Madagascar	ANTANANARIVO/ IVATO	0	594	6041	1158	5411	730
Malaysia	KUALA LUMPUR SUBANG	0	0	11882	6407	63640	24641
Maldives	MALE	0	0	12202	6727	81985	30494
Mali	BAMAKO/SENOU	0	0	11829	6354	76878	41874

(*Continued*)

Country	City	HDD Tb=50°F	HDD Tb=65°F	CDD Tb=50°F	CDD Tb=65°F	CDH Tb=74°F	CDH Tb=80°F
Malta	LUQA	19	1407	5928	1843	13197	4040
Marshall Islands	MAJURO WSO AP	0	0	11784	6309	71297	20514
Martinique	LE LAMENTIN	0	0	10956	5481	50138	14230
Mauritania	NOUAKCHOTT	0	4	10797	5327	52619	24555
Mauritius	VACOAS (MAURITIUS)	0	45	7641	2212	7286	543
Mayotte	DZAOUDZI/PAMANZI	0	0	10983	5508	53825	14168
Mexico	AEROP. INTERNACIONA	4	1014	4806	342	3429	434
Micronesia	YAP ISLAND WSO AP	0	0	11624	6149	64716	19110
Moldova	KISINEV	2719	6006	2773	585	4198	1136
Mongolia	ULAANBAATAR	8182	12536	1261	142	1828	545
Morocco	RABAT-SALE	11	1440	5004	960	5144	1299
Mozambique	MAPUTO/MAVALANE	0	39	8924	3488	26003	7719
Namibia	WINDHOEK	14	683	6694	1889	17120	5799
Netherlands	AMSTERDAM AP SCHIPH	1589	5468	1712	117	874	186
New Caledonia	NOUMEA (NLLE-CALEDONIE)	0	3	8729	3258	18185	3384
New Zealand	WELLINGTON AIRPORT	243	3378	2423	84	76	1
Nicaragua	MANAGUA A.C.SANDINO	0	0	11623	6148	63931	27948
Niger	NIAMEY-AERO	0	0	12946	7471	104212	62823
Niue Is.	ALOFI	0	0	9789	4315	29174	5579
Norfolk Island	NORFOLK ISLAND AERO	0	554	5817	898	735	3
Northern Mariana Islands	SAIPAN	0	0	11774	6299	65975	19534
Norway	OSLO/GARDERMOEN	4405	8967	951	37	418	47
Oman	SEEB INTL AIRPORT	0	2	12045	6573	88013	51927
Pakistan	ISLAMABAD AIRPORT	29	1174	7899	3568	46293	25118
Palau	KOROR WSO	0	0	11741	6266	65793	18779
Panama	MARCOS A GELABERT I	0	0	11918	6443	64893	22450
Paraguay	ASUNCION/ AEROPUERTO	11	457	8714	3688	36593	16091
Peru	LIMA-CALLAO/AEROP.	0	297	6657	1480	4715	620
Philippines	MANILA	0	0	12126	6651	78701	29895
Poland	WARSZAWA-OKECIE	2947	6787	1838	201	1943	478
Portugal	LISBOA/PORTELA	63	1822	4796	1079	7352	2724
Puerto Rico	SAN JUAN INTL ARPT	0	0	11087	5612	52210	14994
Qatar	DOHA INTERNATIONAL	0	132	11693	6349	88426	56526
Reunion Is.	SAINT-DENIS/GILLOT	0	0	9278	3803	23949	4539
Romania	BUCURESTI AFUMATI	2398	5525	3035	687	7442	2713
Russia	MOSKVA	4504	8545	1625	193	1366	261
Saint Helena Is.	ST. HELENA IS.	0	556	5503	586	86	2
Saint Lucia	HEWANORRA INTL AIRP	0	0	11573	6098	65643	18902

Country	City	HDD Tb=50°F	HDD Tb=65°F	CDD Tb=50°F	CDD Tb=65°F	CDH Tb=74°F	CDH Tb=80°F
Samoa	APIA	0	0	11088	5613	54488	14717
Saudi Arabia	RIYADH OBS. (O.A.P.)	10	541	10817	5875	93042	63113
Senegal	DAKAR/YOFF	0	1	9653	4180	32381	8182
Serbia	BEOGRAD	1730	4605	3500	897	7768	2849
Seychelles	SEYCHELLES INTERNAT	0	0	11393	5918	60010	15119
Singapore	SINGAPORE/ CHANGI AI	0	0	11842	6367	66856	22005
Slovakia	BRATISLAVA-LETISKO	2252	5579	2625	477	4559	1463
Slovenia	LJUBLJANA/BEZIGRAD	2022	5315	2668	485	3920	1145
Solomon Islands	HONIARA/ HENDERSON	0	0	10887	5412	47608	15112
South Africa	PRETORIA (IRENE)	42	1459	4910	852	5791	1199
Spain	MADRID/BARAJAS RS	812	3641	3752	1102	14484	6955
Sri Lanka	KATUNAYAKE	0	0	11617	6142	65819	22662
Suriname	ZANDERIJ	0	0	11275	5800	48210	18181
Svalbard and Jan Mayen	HOPEN	9828	15303	0	0	0	0
Sweden	STOCKHOLM/ ARLANDA	3627	7937	1246	82	745	124
Switzerland	ZURICH-KLOTEN	2232	5945	2002	238	2402	644
Syria	DAMASCUS INT. AIRPO	537	2748	5173	1908	26604	14995
Taiwan	TAIBEI	1	426	8730	3680	35198	13651
Tajikistan	DUSHANBE	1085	3493	4779	1713	21857	11312
Tanzania	DAR ES SALAAM AIRPO	0	0	10531	5056	44474	14746
Thailand	BANGKOK METROPOLIS	0	0	12447	6972	83767	37399
Togo	LOME	0	0	11441	5966	59910	19766
Tonga	FUAAMOTU	0	5	9002	3533	19977	2999
Tunisia	TUNIS-CARTHAGE	34	1466	6177	2135	20030	8638
Turkey	ESENBOGA	2471	5939	2420	409	6387	2287
Turkmenistan	ASHGABAT KESHI	1139	3436	5798	2617	35387	20013
Tuvalu	FUNAFUTI NF	0	0	12050	6575	77197	26165
Ukraine	KYIV	3380	7033	2149	324	2248	489
United Arab Emirates	ABU DHABI INTER. AI	0	54	11838	6417	88284	56005
United Arab Emirates	DUBAI INTERNATIONAL	0	43	11629	6196	85007	52310
United Kingdom	LONDON WEATHER CENT	835	4219	2324	232	1250	279
United States Minor Outlying Islands	MIDWAY ISLAND NAS	0	51	8760	3337	22747	4577
Uruguay	CARRASCO	176	2198	4282	829	4925	1294
Uzbekistan	TASHKENT	1404	3892	4822	1835	23981	12682
Vanuatu	ANEITYUM	0	0	9003	3529	21719	3033

(*Continued*)

Country	City	HDD Tb=50°F	HDD Tb=65°F	CDD Tb=50°F	CDD Tb=65°F	CDH Tb=74°F	CDH Tb=80°F
Venezuela	CARACAS/ MAIQUETIA A	0	0	11311	5836	55271	17831
Vietnam	HA NOI	0	302	9401	4227	42548	16756
Wallis and Futuna	HIHIFO (ILE WALLIS)	0	0	11324	5849	59584	15177
Zimbabwe	HARARE (KUTSAGA)	0	627	6164	1316	8254	1601

ANNUAL HEATING DEGREE DAYS (HDD), COOLING DEGREE DAYS (CDD), AND COOLING DEGREE HOURS (CDH) US SITES (IP UNITS)

State	City	HDD Tb=50°F	HDD Tb=65°F	CDD Tb=50°F	CDD Tb=65°F	CDH Tb=74°F	CDH Tb=80°F
AK	JUNEAU INT'L ARPT	3487	8333	634	3	89	6
AL	MONTGOMERY DANNELLY FIELD	454	2143	6069	2282	21686	9420
AR	LITTLE ROCK ADAMS FIELD	881	2976	5505	2124	21697	9744
AZ	PHOENIX SKY HARBOR INTL AP	30	941	9120	4557	69095	44506
CA	SACRAMENTO METROPOLITAN AP	323	2389	4817	1408	17945	9325
CO	DENVER INTL AP	2581	5942	2890	777	9519	4216
CT	HARTFORD BRADLEY INTL AP	2699	5992	2917	739	6789	2338
DE	DOVER AFB	1674	4517	3808	1176	9566	3247
FL	TALLAHASSEE REGIONAL AP	230	1534	6733	2563	24170	10024
GA	ATLANTA HARTSFIELD INTL AP	686	2694	5309	1841	15755	5878
HI	HONOLULU INTL ARPT	0	0	10124	4649	35515	9466
IA	DES MOINES INTL AP	3121	6240	3397	1041	9530	3472
ID	BOISE AIR TERMINAL	2331	5658	3036	890	12435	6149
IL	SPRINGFIELD CAPITAL AP	2487	5429	3668	1135	11001	4059
IN	INDIANAPOLIS INTL AP	2367	5322	3575	1055	9005	2965
KS	TOPEKA FORBES FIELD	2157	4992	3964	1325	13116	5524
KY	LOUISVILLE STANDIFORD FIELD	1599	4168	4401	1496	13849	5190
LA	BATON ROUGE RYAN ARPT	274	1610	6790	2653	24400	9772
MA	BOSTON LOGAN INT'L ARPT	2334	5621	2938	750	5609	1881
MD	BALTIMORE BLT-WASHNGTN INT'L	1726	4567	3861	1228	11317	4315
ME	AUGUSTA AIRPORT	3707	7375	2217	410	3069	809
MI	LANSING CAPITAL CITY ARPT	3392	6889	2548	570	5431	1611
MN	ST PAUL DOWNTOWN AP	4087	7529	2700	669	5586	1747
MO	JEFFERSON CITY MEM	1815	4514	4156	1380	13687	5489
MS	JACKSON INTERNATIONAL AP	557	2311	5986	2265	21596	9332
MT	HELENA REGIONAL AIRPORT	3847	7679	2014	374	5649	2342
NC	RALEIGH DURHAM INTERNATIONAL	991	3322	4728	1579	13975	5465
ND	BISMARCK MUNICIPAL ARPT	4785	8471	2328	539	6942	2877
NE	LINCOLN MUNICIPAL ARPT	2908	5993	3576	1184	13127	5770
NH	CONCORD MUNICIPAL ARPT	3551	7180	2305	461	5126	1676
NJ	TRENTON MERCER COUNTY AP	2072	5144	3389	987	8796	3173
NM	ALBUQUERQUE INTL ARPT	1372	4069	4128	1348	14336	6190

State	City	HDD Tb=50°F	HDD Tb=65°F	CDD Tb=50°F	CDD Tb=65°F	CDH Tb=74°F	CDH Tb=80°F
NV	LAS VEGAS MCCARRAN INTL AP	316	2105	7034	3348	49234	30264
NY	ALBANY COUNTY AP	3192	6608	2649	592	5036	1465
NY	NEW YORK J F KENNEDY INT'L AR	1813	4828	3439	978	6064	1636
OH	COLUMBUS PORT COLUMBUS INTL A	2322	5322	3446	971	8610	2890
OK	OKLAHOMA CITY WILL ROGERS WOR	1193	3516	5079	1926	20693	9840
OR	SALEM MCNARY FIELD	1127	4576	2320	292	4685	1837
PA	HARRISBURG CAPITAL CITY ARPT	2180	5228	3419	990	8754	3104
RI	PROVIDENCE T F GREEN STATE AR	2327	5591	2940	729	5408	1654
SC	COLUMBIA METRO ARPT	598	2529	5651	2108	20858	9106
SD	PIERRE MUNICIPAL AP	3727	7107	3031	936	11527	5610
TN	NASHVILLE INTERNATIONAL AP	1199	3542	4817	1683	16032	6340
TX	AUSTIN/BERGSTROM	300	1654	7108	2989	32215	15566
UT	SALT LAKE CITY INT'L ARPT	2337	5521	3483	1193	14334	6907
VA	VIRGINIA TECH ARPT	1865	4841	3236	737	5850	1586
VA	WASHINGTON DC REAGAN AP	1389	4001	4390	1524	13494	5085
VT	MONTPELIER AP	4343	8217	1865	262	2439	530
WA	OLYMPIA AIRPORT	1479	5372	1686	101	2421	759
WI	MADISON DANE CO REGIONAL ARPT	3715	7197	2604	608	5688	1718
WV	CHARLESTON YEAGER ARPT	1704	4443	3803	1066	9238	3107
WY	CHEYENNE MUNICIPAL ARPT	3358	7148	2011	324	4542	1472

ANNUAL HEATING DEGREE DAYS (HDD), COOLING DEGREE DAYS (CDD), AND COOLING DEGREE HOURS (CDH) CANADIAN SITES (IP UNITS)

State	City	HDD Tb=50°F	HDD Tb=65°F	CDD Tb=50°F	CDD Tb=65°F	CDH Tb=74°F	CDH Tb=80°F
AB	CALGARY INT'L A	4779	9154	1167	67	1308	298
AB	EDMONTON CITY CENTRE A	5305	9495	1401	114	1150	225
BC	VANCOUVER INT'L A	1442	5278	1712	73	262	19
BC	VICTORIA INT'L A	1402	5439	1478	40	568	99
MB	WINNIPEG RICHARDSON INT'L A	6393	10350	1819	303	3102	882
NB	FREDERICTON A	4494	8445	1763	238	2194	547
NF	ST JOHN'S A	4285	8832	977	51	196	11
NS	HALIFAX STANFIELD INT'L A	3856	7840	1669	177	999	166
NT	YELLOWKNIFE A	10286	14951	867	59	306	33
NU	IQALUIT A	12701	18136	41	0	1	0
ON	OTTAWA MACDONALD-CARTIER INT'	4531	8213	2219	425	3274	857
ON	TORONTO LESTER B. PEARSON INT	3518	7120	2368	496	3942	1102
PE	CHARLOTTETOWN A	4389	8465	1571	170	730	77
QC	MONTREAL/MIRABEL INT'L A	4878	8729	1915	292	2324	492

(*Continued*)

State	City	HDD Tb=50°F	HDD Tb=65°F	CDD Tb=50°F	CDD Tb=65°F	CDH Tb=74°F	CDH Tb=80°F
QC	QUEBEC/JEAN LESAGE INTL A	5203	9169	1746	238	1779	361
QC	SHERBROOKE A	5045	9105	1584	167	1721	311
SK	REGINA A	6185	10262	1623	227	3193	1045
SK	SASKATOON DIEFENBAKER INT'L A	6405	10550	1520	189	2679	832
YT	WHITEHORSE A	7326	12246	564	10	354	64

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