NATIONAL RENEWABLE ENERGY LABORATORY

# Refrigeration Playbook: Heat Reclaim

## Optimizing Heat Rejection and Refrigeration Heat Reclaim for Supermarket Energy Conservation


Chuck Reis, Eric Nelson, James Armer, and Tim Johnson
*CTA Architects Engineers*
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**Technical Report**
NREL/TP-5500-63786
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Prepared under Task No. ARCB1102
Prepared under Subcontract No. LEA-0-40383-01

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# Acknowledgments

*Refrigeration Playbook: Heat Reclaim* was developed as a joint effort between the National Renewable Energy Laboratory and CTA Architects Engineers. This project was made possible with American Recovery and Reinvestment Act of 2009 funds.

The authors would like to thank Arah Schuur and Glenn Schatz of the U.S. Department of Energy Building Technologies Office for their support and guidance and to acknowledge reviewers who provided helpful feedback during the development of the playbook, including Ren Anderson, Eigen Energy; Ryon Browning, CLEAResult; Harrison Horning, Delhaize; Ron Judkoff, National Renewable Energy Laboratory; K.C. Kolstad and Mike Saunders, Emerson Climate Technologies; Tom Land, U.S. Environmental Protection Agency; Dustin Lilya, DC Engineering; Dustin Searcy, Parker Hannefin Corporation; and Caleb Nelson, CTA Architects Engineers.

## Abbreviations and Acronyms

| | |
|---|---|
| ASHRAE | American Society of Heating, Refrigerating and Air-Conditioning Engineers |
| ANSI | American National Standards Institute |
| Btu | British thermal unit |
| CBP | Commercial Building Partnerships |
| CFM | cubic feet per minute |
| db | dry bulb |
| DOE | U.S. Department of Energy |
| DSH | desuperheat |
| $\varepsilon$ | effectiveness or efficiency value |
| EER | energy efficiency ratio - Btu/W-h |
| EIR | energy input ratio |
| EWT | entering water temperature |
| GPM | gallons per minute |
| HVAC | heating, ventilation, and air-conditioning |
| in. | inch |
| IP | Imperial units |
| kW | kilowatt |
| LAT | leaving air temperature |
| LWT | leaving water temperature |
| MAT | mixed air temperature |
| NREL | National Energy Renewable Laboratory |
| OAT | outdoor air temperature |
| p-h | pressure-enthalpy |
| PLR | part-load ratio |

| | |
|---|---|
| ppm | part per million |
| SI | International System of Units, also referred to as metric units |
| SHW | service hot water |
| THR | total heat of rejection |
| TAB | test adjust and balance |
| TC | technical committee |
| TD | temperature differential |
| U | thermal transmittance - $Btu/h \cdot ft^2 \cdot °F$ |
| UPS | uninterruptible power supply |
| USGBC | U. S. Green Building Council |
| W | Watts |
| wb | wet bulb |
| WSHP | water-source heat pump |

# Executive Summary

## Background

This refrigeration playbook for optimizing heat rejection and refrigeration heat reclaim for supermarket energy conservation emerged from work done as part of the U.S. Department of Energy's Commercial Building Partnerships (CBP) program. CBP was a public/private, cost-shared initiative that demonstrated cost-effective, replicable ways to achieve dramatic energy savings in commercial buildings. It aimed to reduce energy use by 50% in new construction and 30% in existing buildings compared with minimum code requirements or with pre-retrofit energy use. Building owners teamed with the U.S. Department of Energy, national laboratory staff, and private sector experts to explore and implement energy-saving ideas and strategies. These strategies were then applied to specific building projects that could be replicated across an organization's building portfolio and eventually across the commercial building market.

Much of the analysis presented here emerged from a CBP pilot retrofit of a 213,000-ft$^2$ Walmart supercenter in Colorado in which waste heat reclaim was used to heat ventilation air for the grocery sales area. It saves almost 20,000 therms of natural gas per year. While the results support Walmart's 20% energy savings commitment under the Better Buildings Challenge, they are also applicable to the whole supermarket sector. This includes companies with commercial refrigeration, such as Target, Whole Foods Market, SUPERVALU, and the Defense Commissary Agency, that also participated in CBP.

While heat reclaim systems have been used in supermarkets for many years, their performance is not well understood. At the same time, recent legislation in some parts of the country, including California and Washington, requires refrigeration waste heat to be recovered in supermarkets that meet certain criteria. This makes the calculation of energy savings from heat reclaim strategies critical to many store designs. This guide attempts to demystify the energy savings associated with heat reclaim strategies by providing information and tools to help experienced refrigeration system designers make informed decisions that add value to a building design by reducing operating and life cycle costs.

## Purpose

The purpose of this playbook and accompanying spreadsheets is to generalize the detailed CBP analysis and to put tools in the hands of experienced refrigeration designers to evaluate multiple applications of refrigeration waste heat reclaim across the United States. Supermarkets with large portfolios of similar buildings can use these tools to assess the impact of large-scale implementation of heat reclaim systems. In addition, the playbook provides best practices for implementing heat reclaim systems to achieve the best long-term performance possible. It includes guidance on operations and maintenance as well as measurement and verification.

## Scope

This playbook was written with a traditional supermarket of 40,000–60,000 ft$^2$ in mind, but the concepts apply to smaller and larger facilities with commercial refrigeration systems. The concepts and methods do not provide a complete design or precise calculations for determining the energy savings associated with managing heat rejection. Long-term savings will also depend on weather variability and the degree to which systems are properly maintained. Rather, they provide ideas and tools to assist in the design process through building operations to achieve long-term savings. When considering other applications, such as convenience stores and industrial refrigeration, care should be taken to determine which parameters should be adjusted to match the application. Brief chapter summaries follow.

- Chapter 1 covers the goals and scope of the playbook in more detail and describes the approach to energy analysis.
- Chapter 2 provides guidance for minimizing refrigeration waste heat before discussing how to reclaim it; this is a necessary first step that will save more energy than reclaiming heat from a wasteful system.
- Chapter 3 describes different applications of refrigeration waste heat reclaim, including service hot water preheating, mixed air heating, and outdoor air preheating; it also discusses desuperheating versus full condensing and different methods for capturing and delivering the heat. System diagrams accompany each application.
- Chapter 4 is a step-by-step tour through the thermodynamic calculations needed to evaluate each heat reclaim application, including the waste heat available and the demand for that heat, that are built into the spreadsheets that accompany the playbook.
- Chapter 5 includes best practices for implementing heat reclaim methods, including an introduction to financial analysis techniques, operations and maintenance, and measurement and verification, that are needed to estimate an expected return on investment of a heat reclaim system and then ensure good long-term performance.
- Appendix A tabulates results from the baseline supermarket energy model and energy savings provided by different waste heat reclaim strategies in 17 locations spanning U.S. climate types. The results also include the impacts of air-cooled, evaporatively cooled, and hybrid condensing strategies.
- Appendix B includes best practices for accurately modeling commercial refrigeration systems in EnergyPlus.
- Appendix C provides all the details of the EnergyPlus baseline supermarket used to benchmark whole-building energy use.

Each chapter provides resources for further study and discussion.

## Methods

The playbook and worksheets rely on simple thermodynamic relationships to determine the amount of heat available for reclaim, the demand for that waste heat by different building end uses, and the resulting potential energy savings of different waste heat reclaim applications across U.S. climate zones.

EnergyPlus was used to model the whole-building energy use of a reference supermarket that corresponds as closely as possible to the assumptions of the playbook and spreadsheets; however, it was not used to calculate the heat reclaim savings because of current limitations in modeling superheat and refrigerant mass flow. For now, the EnergyPlus results should be considered as a broad-brush picture of the variability that can be expected in whole-building energy consumption across climate zones. Differences in the way the refrigeration evaporator load was modeled between EnergyPlus and the spreadsheet models prevent a direct apples-to-apples comparison. In the long run, EnergyPlus is expected to be the preferred approach to estimating waste heat reclaim savings because it accounts for the dynamic interaction between the refrigeration system and the rest of the store and allows representation of heat pump-based systems that could not be captured with the spreadsheet calculations.

## Conclusions

From the results and the authors' experience, domestic hot water seems to be the simplest and most cost-effective method of heat reclaim and is the most broadly deployed heat reclaim strategy today. The service hot water load in a supermarket is fairly consistent throughout the year. These systems can work in any climate zone. Outdoor air preheat can be a good solution in colder climates, especially in 100% outdoor air systems. A heat reclaim system for space heating can result in the most energy savings of the methods discussed, but also can be the most difficult to implement because it requires placing a large heating coil in the primary airstream and is difficult to implement in retrofits.

The playbook is intended for use in conjunction with the accompanying spreadsheet models. It includes descriptions of heat reclaim systems, considerations for implementing each method, and the thermodynamic theory behind the spreadsheet models. The spreadsheets are provided so that an experienced designer can quickly generate energy performance results for each heat reclaim method and optimize the recovery of waste heat for use in building heating, ventilation, and air-conditioning and service hot water systems.

Before a heat reclaim strategy is implemented on a large scale, the authors recommend testing the technology and measuring its performance. A good measurement and verification plan can provide valuable information with minimal cost. This is discussed in more depth in Chapter 5.

This playbook addresses the major concepts that should be considered when implementing a heat reclaim strategy, but does not address products available for purchase or specific details for designing the systems. To implement an efficient heat reclaim system, the design must be integrated with experts from other disciplines. The owner; refrigeration designer; heating, ventilation, and air-conditioning or plumbing designer; and contractor must be involved early in the design to achieve a favorable outcome.

## Resources


1. 2013 Nonresidential Compliance Manual, California Energy Commission, 10.5.5, June 2014.
2. 2012 Washington State Energy Code, Commercial Provisions, C403.2.6.3, July 2013.

# Table of Contents

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# List of Tables

# Chapter 1. Introduction

*Refrigeration Playbook: Heat Reclaim* provides guidance for reducing the energy consumption of refrigeration systems by reducing the load on the system and reclaiming waste heat for use in service hot water (SHW) and heating, ventilation, and air-conditioning (HVAC) systems. The intended audience includes building owners, operators, designers, and installers, though most of the guidance and many of the calculation procedures are developed with experienced designers in mind.

A refrigeration system is generally a supermarket's largest energy demand. It often accounts for more than half of a store's energy consumption. These systems generally represent the greatest opportunity for energy savings in this building type.

Refrigeration heat reclaim systems in supermarkets have historically been applied in a "one size fits all" manner, targeting a capacity based on the heat rejection of the system at summer design conditions without considering annual operation. The annual performance of reclaim systems can be difficult to calculate, and ancillary losses are often not identified or considered. Thus, heat reclaim systems are often not applied where appropriate or are applied in such a way that their operation costs exceed their savings. Recent changes to Title 24 requirements in California combined with rising energy costs, have raised awareness of heat reclaim systems and the need for better understanding of design strategies among building owners, operators, and designers.

The authors' intent is to advance the understanding of refrigeration heat reclaim systems by providing system designers and owners with transparent methods for calculating system performance at design and hourly conditions. First, the means for calculating the heat available from the refrigeration system and the heat required in the building are described; second, methods for calculating the energy impacts are shown for some of the most common heat reclaim systems. These common systems include refrigerant desuperheating (DSH) for service water heating and for mixed air heating. Several less common but viable alternatives are also addressed, including full refrigerant condensing for makeup air preheating via a water loop, full refrigerant condensing to mixed air heating via a water loop, refrigerant DSH directly to makeup air preheat, and water-source heat pump (WSHP) loop integration. Methods are also described for integrating these systems with several refrigeration heat rejection methods, including air-cooled, evaporative, and hybrid condensing methods.

Other resources such as *Waste Heat Recapture from Supermarket Refrigeration Systems* by Brian A. Fricke, Ph.D. at Oak Ridge National Laboratory include detailed descriptions of heat reclaim methods.

DSH and condensing are frequently referred to in this playbook. *DSH* refers to lowering to the boiling point the temperature of a refrigerant that was heated beyond its boiling point. *Condensing* refers to the process of changing the refrigerant's phase from gas to liquid.

## 1.1 Goal of This Guide

This playbook was developed to equip those involved with designing supermarket refrigeration systems to make informed decisions about load reduction and heat reclaim systems that maximize value to the building owner. In particular, the playbook is intended to provide experienced refrigeration system designers with tools to estimate the performance of waste heat reclaim systems.

## 1.2 Scope: How To Use This Playbook

This playbook was written with a traditional supermarket of 40,000–60,000 $ft^2$ in mind, but the concepts apply to smaller and larger facilities with commercial refrigeration systems. The concepts and methods do not provide a complete design or precise calculations for determining the energy savings associated with managing heat rejection. Rather, they provide ideas and tools to assist in the design process through building operations to achieve long-term savings. Brief chapter summaries follow.

### Chapter 2: Heat Rejection Management

This chapter includes tips for minimizing waste heat such as various condensing methods and reduction of thermal loads on the refrigeration system. In general, lowering the refrigeration load before attempting heat reclaim strategies is more cost effective than using these strategies alone. This involves high- and low-pressure sides of the refrigeration cycle.

- **Load reduction.** Reduce heat entering the system.
- **Low-pressure side.** Raise suction temperature strategies.
- **High-pressure side.** Lower condensing pressure strategies.

### Chapter 3: Heat Reclaim Methods

This chapter includes descriptions of common heat reclaim strategies. Once the refrigeration load has been reduced, heat reclaim methods can be applied to further offset the energy impact of the refrigeration system.

### Chapter 4: Heat Reclaim Calculations

This chapter includes a step-by-step approach to quickly quantify energy savings from heat reclaim strategies. Calculations are intended to be used alongside the heat reclaim spreadsheets provided with this playbook.

### Chapter 5: Implementing Heat Reclaim Methods

This chapter includes tips and considerations for implementing heat reclaim systems in various configurations and provides practical information about how these methods can influence design decisions.

### Appendix A: Energy Results

This appendix includes sample results from spreadsheet and energy modeling calculations based on a baseline refrigeration system (detailed in Appendix C).

### Appendix B: Notes to Energy Modelers

Heat rejection strategies discussed in the report can be modeled using building energy simulation software. This appendix was written from the authors' experience modeling supermarkets with EnergyPlus™, but many of the same concepts apply to other simulation software packages.

**Appendix C: Baseline Energy Modeling Assumptions**
Evaluating the savings from heat reclaim requires a benchmark that reflects the performance of a typical supermarket. This appendix outlines the assumptions made to model a baseline system for each climate zone in EnergyPlus.

## 1.3 Approach to Energy Analysis

The energy analysis is intended to be simple enough to be easily understood, yet sophisticated enough to accurately estimate energy performance. The overall refrigeration performance and some of these reclaim methods can be modeled with software such as EnergyPlus; however, the authors concluded that, at the time of the study, the calculation methods within EnergyPlus were not adequate to accurately model some refrigeration heat reclaim systems. The calculation of superheat and mass flow has specific limitations, so the authors determined that these were better performed using spreadsheets. In addition, the authors wanted to provide the ability to do heat reclaim calculations to the largest possible audience, including designers who are accustomed to using spreadsheets for their design calculations. Some of these findings are outlined in Appendix C. The calculation methods are carefully outlined in Chapter 4 and can be compared against EnergyPlus methods in the *EnergyPlus Engineering Reference*.

## 1.4 Resources


1. "EnergyPlus Engineering Reference." The Board of Trustees of the University of Illinois and the Regents of the University of California through the Ernest Orlando Lawrence Berkeley National Laboratory, 2013.
2. 2013 Nonresidential Compliance Manual for the Building Energy Efficiency Standards. California Energy Commission, 2014.
3. Fricke, B.A. (2011).Waste Heat Recapture from Supermarket Refrigeration Systems. Oak Ridge, TN: Oak Ridge National Laboratory.

# Chapter 2. Heat Rejection Management

## 2.1 Introduction

Reducing unwanted thermal loads on the refrigeration system is more energy efficient than recovering and using the excess waste heat that results. Measures taken to reduce the source of heat into the refrigeration system earlier in the cycle generate direct energy savings as well as the indirect savings resulting from not having to remove the unwanted heat. For instance, using more efficient evaporator fans reduces the electricity consumed by the fans and the heating load on the case. This reduces the pressure losses in lines and valves, the work done by the compressors, and the work done by heat rejection devices. A few opportunities for system-wide savings are identified in Section 2.2. For more in-depth discussions about these opportunities, see the Resources section.

## 2.2 Lowering the Load

At the most basic level, the sources of heat into the refrigeration system can be defined by two categories: external loads and internal loads. *External loads* are usually generated because of temperature and humidity differences between the refrigerated case or walk-in box and the surrounding area. This load consists of three modes of heat transfer: conduction, convection, and radiation. The most effective methods for reducing these loads follow:

- **Conductive loads**
    - Raise the temperature of the refrigerated control volume.
    - Allow case temperature set points to float up where food safety concerns allow, such as with beer cases, to reduce the temperature difference with the surrounding area.
    - Lower the temperature of the surrounding area.
        - Avoid placing cases or walk-in boxes in unconditioned areas or against exterior walls.
        - Use back-to-back cases in lieu of two separate cases.
        - Group walk-in boxes with shared walls to reduce the overall surface area in contact with nonrefrigerated spaces.
        - Place access doors into walk-in box freezers inside coolers rather than directly in nonrefrigerated spaces.
    - Lower the thermal conductivity of the enclosures.
        - Increase walk-in box wall and ceiling insulation thickness to meet or exceed federal requirements.
        - Reduce uninsulated areas.
- **Convective loads**
    - Wherever possible, use cases with doors.
    - Use night curtains on remaining open cases.

    - Apply door alarms to walk-in doors to encourage efficient operation.
    - Reduce the size of walk-in doors.
    - Avoid refrigeration in areas directly communicating with the sales floor, such as food preparation areas.
    - Install and maintain strip curtains on walk-in doors.
- **Radiative loads.** These are generally considered negligible compared with convective and conductive loads.

When designs reduce the external loads, the internal loads become more significant. These loads are largely composed of electrical components such as fans, lights, defrost coils, and door heaters that are contained within the refrigerated volume. Various methods can be used to reduce both the direct electricity load of these components and the part of this load that causes a heat load inside the refrigerated space. The most common methods for reducing these loads (many of which are required by the 2012 requirements found in DOE 2010) follow:

- Use high-efficiency fans.
- Use high-efficiency lighting (light-emitting diodes).
- Reduce or remove lighting.
- Reduce anti-sweat heater operation based on ambient humidity levels.
- Use low-energy doors (low/no anti-condensate heat, low conductivity).
- Move lighting sources outside the refrigerated volume.
- Add motion sensors to lighting circuits.
- Limit equipment in refrigerated areas.

System inefficiencies also contribute to the heat rejection load. Proper system design can vastly improve energy performance and reduce the overall heat rejection load by minimizing the heat added by the refrigeration system components. System design strategies to increase efficiency follow:

- Subcool the liquid refrigerant.
- Add compressor capacity modulation (either digital or variable speed).
- Add liquid/suction heat exchangers.
- Allow suction and condensing temperatures or pressures to float.
- Increase the efficiency of heat rejection methods.
    - Evaporative condensing
    - Hybrid condensing
    - Water-cooled condensing.

## 2.3 Resources


1. Goetzler, W.; Goffri, S.; Jasinski, S.; Legett, R.; Lisle, H.; Marantan, A.; Millard, M.; Pinault, D.; Westphalen, D.; Zogg, R. (2009). Energy Savings Potential and R&D Opportunities for Commercial Refrigeration. Work performed by Navigant Consulting, Inc., Chicago, Illinois. Washington, DC: U.S. Department of Energy. http://apps1.eere.energy.gov/buildings/publications/pdfs/corporate/commercial_refrig_report_10-09.pdf.

2. Energy Conservation Program: Energy Conservation Standards for Walk-in Coolers and Freezers. 10 CFR Part 431. Office of Energy Efficiency and Renewable Energy, Department of Energy (2010): RIN 1904-AB86.

3. Energy Conservation Program: Test Procedure for Commercial Refrigeration Equipment; Final Rule. 10 CFR Parts 429 and 431. Office of Energy Efficiency and Renewable Energy, Department of Energy (April 21, 2014): RIN 1904-AC99.

# Chapter 3. Heat Reclaim Methods

## 3.1 Introduction

This chapter describes the basic configuration of common heat reclaim methods. The diagrams are schematic for illustration purposes and do not represent actual designs. The system descriptions and associated diagrams are used as a starting point for further system discussions.

Heat reclaim systems should be designed to save the most heat possible with the lowest installed and operating costs. This goal can be achieved in several ways, depending on the quantity and availability of the heat available and the heat needed to satisfy service water or space heating requirements. In supermarkets, reclaim systems generally fall into two categories: water reclaim and air reclaim. Water reclaim is used to heat service water for food preparation, cleanup, and hand washing. On the refrigeration side, the most common heat reclaim systems remove superheat from the refrigerant vapor discharging from the compressors. By avoiding condensing, the temperature of the refrigerant remains higher. The physical layout limitations it introduces are also minimized. Condensing systems may also be used to increase the quantity of heat available for reclaim. These condensing systems require additional design considerations for proper operation. (Refer to Chapter 5 for more information about design and operational considerations.)

Every reclaim design must address the common concerns associated with these systems, including:

- How consistent is the amount of heat available?
- How consistent is the amount of available usable heat?
- What are the temperatures of the refrigerant and the medium to be heated?
- Where are systems located relative to one another?
- Are there negative impacts on the efficiency of the refrigeration, water, and air systems? Are they consistent, or how do they vary?
- How does the reclaim system affect the refrigerant charge in the refrigeration system?

The answers to these questions will direct decisions about the most suitable system for the application of heat reclaim and how the system is laid out and sized.

## 3.2 Refrigeration System

Figure 3-1shows a simplified refrigeration cycle.

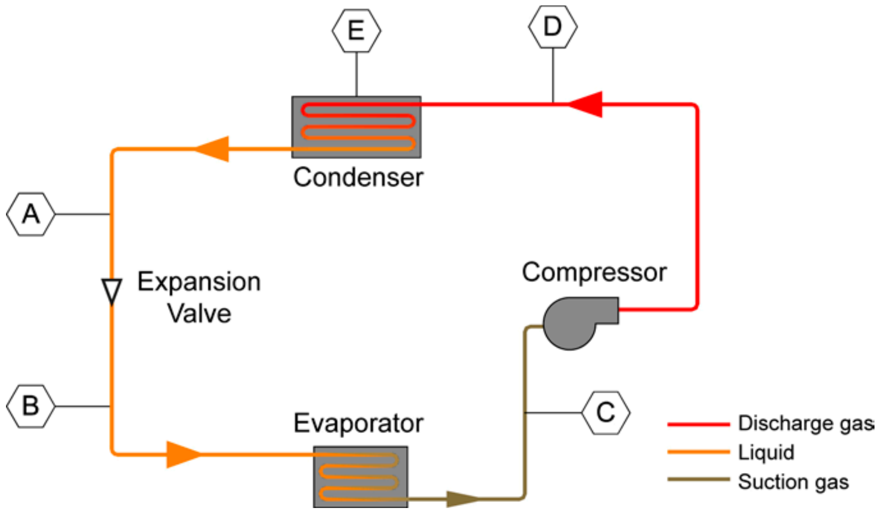


**Figure 3-1. Basic refrigeration cycle**

The refrigerant's properties as it passes through this cycle can be plotted on a pressure-enthalpy (p-h) diagram for R-404a (see Figure 3-2). Such diagrams are readily available from sources such as the *ASHRAE Handbook, Fundamentals* or from the refrigerant manufacturer. The p-h diagram is a resource commonly used by designers to visualize and calculate the performance of refrigeration systems. The refrigerant pressure is plotted on the y-axis and the refrigerant enthalpy is plotted on the x-axis. Enthalpy is the heat in Btu per pound of refrigerant. Differences in enthalpy multiplied by the refrigerant mass flow yield the heat transferred by the refrigerant. The p-h diagram is also used to visualize temperatures at various points in the cycle. The temperatures inside the dome-shaped curve (for example, between points A and E or between B and C) are flat. In this area, the refrigerant changes phase between liquid and vapor (it does not change temperature). This is true in the evaporator and condenser segments of the refrigeration cycle. To the left of the dome, the refrigerant is 100% liquid, or subcooled. To the right of the dome, refrigerant is 100% vapor, or superheated.

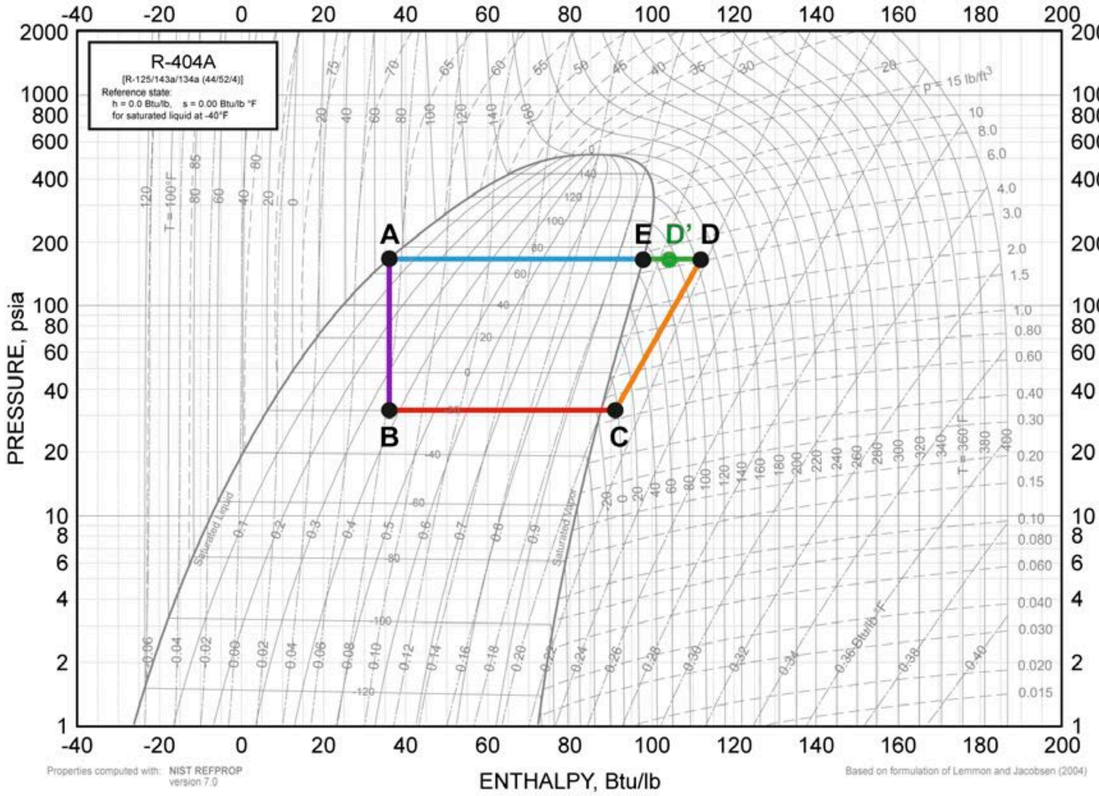


**Figure 3-2. R-404a p-h diagram**

Segment A to B is the expansion process, where pressure is reduced from condensing pressure to evaporator pressure through an expansion valve. This process is assumed to be adiabatic, meaning no heat is transferred. Segment B to C represents the evaporation process, where heat from the refrigerated cases is absorbed by the refrigerant, causing the refrigerant to boil. Segment C to D' represents the ideal isentropic, or constant entropy, compression process. Segment C to D represents an actual compression process that accounts for compression inefficiency. Segment D to E represents the desuperheat process, where heat is rejected and the temperature of the refrigerant is reduced to condensing temperature. Segment E to A represents condensing, where heat is rejected and the refrigerant changes from a gas to a liquid, but the temperature does not change significantly. R-404a is considered a near-azeotropic blend of refrigerants, meaning its components have similar pressure and temperature relationships at phase-change conditions. This means that the lines of constant temperature in the two-phase region are at nearly constant pressure from saturated liquid to saturated vapor. For zeoptropic refrigerants, known as *high glide refrigerants* (such as R-407a), the lines of constant temperature change pressure dramatically from saturated liquid to saturated vapor.

## 3.3 Service Hot Water Preheat

SHW most often captures waste heat by using a refrigerant coil submerged in one or multiple service water tanks, known as *reclaim tanks*. A reclaim tank is installed in the cold water line to serve as the first stage of heating upstream of a gas- or electricity-fired water heater. Depending on the plumbing system design, building hot water may include a recirculation loop to maintain appropriate water temperatures at remote fixtures. In this case, the recirculation loop may be connected to the cold water line upstream of the reclaim tank, or sometimes directly in the tank.

On the refrigeration side, superheated refrigerant vapor leaving the compressor is diverted by a three-way valve to one or multiple reclaim tanks. The vapor travels through the coil from the top of the tank to the bottom of the tank then out to the condenser. The three-way refrigerant diverting valve is controlled based on the water temperature leaving the reclaim tank to a maximum temperature equal to the SHW set point. A typical layout of this system is shown in Figure 3-3.

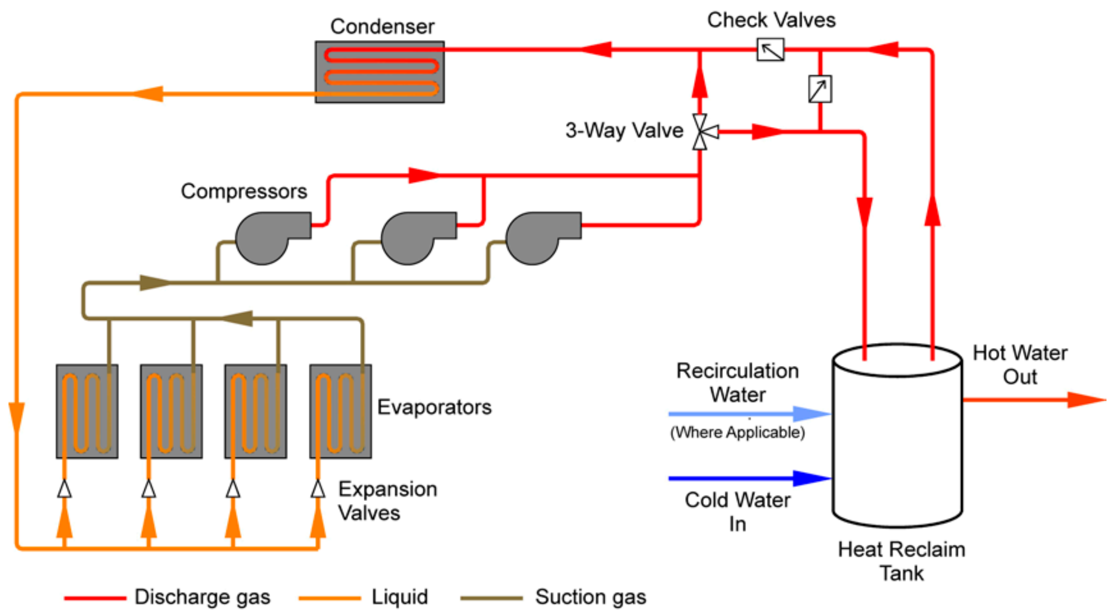


**Figure 3-3. SHW preheat schematic**

SHW temperature set points are 120°–140°F, so at low SHW demand (high tank temperature), the refrigerant cannot condense. During periods of high SHW demand, when cold water flow rates increase in the reclaim tank, the tank temperature drops well below the tank temperature set point, allowing for more heat reclaim. Refrigerant coils are designed to allow the refrigerant to pass through fast enough to prevent condensing and its associated refrigerant and oil management problems.

## 3.4 Mixed Air Heating

Heat reclaim for space heating or dehumidification reheat can be designed similarly to service water heating on the refrigeration side. A refrigerant coil is placed in an air handler instead of in a water tank. The coil is usually sized for DSH only, so the refrigerant leaves the reclaim coil as a gas and travels to the condenser. A schematic of this layout is shown in Figure 3-4.

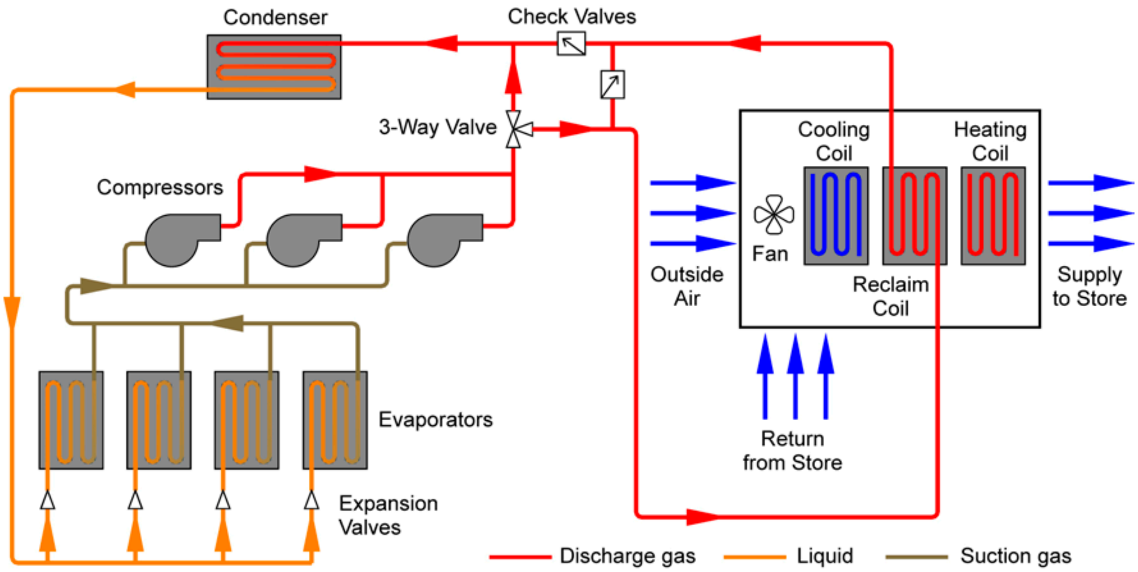


**Figure 3-4. Mixed air heating schematic for DSH**

This system can also be designed to allow for condensing in the reclaim coil. For a condensing reclaim system, a three-way valve in the compressor discharge line diverts the refrigerant to a water-cooled reclaim condenser, which is physically placed above the normal condenser. The refrigerant passes through the reclaim condenser and desuperheats, partially condenses, or fully condenses. It then travels to the condenser and is either condensed or passed through as liquid. A water loop conducts the heat from the reclaim condenser to a coil mounted in the air handler. This configuration allows for heat reclaim when the air handler is located far from the refrigeration system. If the air handler is located above the condenser, the water loop may be omitted from the design. The layout with a water loop is shown in Figure 3-5.

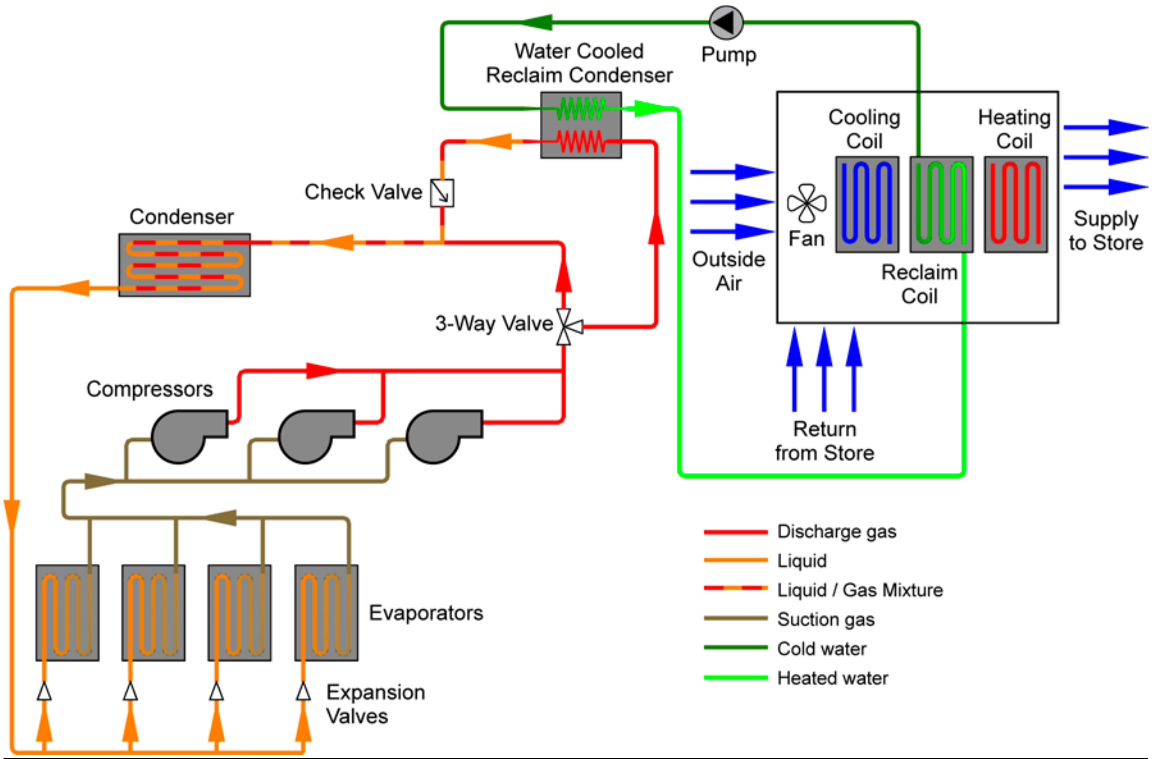


**Figure 3-5. Mixed air heating schematic for condensing**

If the heating load is consistently large enough, this system may be modified to a parallel configuration for condensing. In this layout, any refrigerant passing through the heat reclaim condenser must be fully condensed and is piped directly to the receiver. This allows the reclaim condenser to be located vertically below the primary condenser, but it must still be above the receiver. Capacity control of the reclaim system must be used to ensure full condensing.

## 3.5 Outdoor Air Preheat

Heat reclaim can be used to heat only the outside air entering the air system for ventilation and building pressurization. This system takes advantage of lower outdoor air temperatures (OATs) compared with mixed air temperatures (MATs), allowing for heat transfer at lower refrigerant temperatures. Outdoor airflow rates are significantly lower than mixed airflows. As with mixed air heating, the system can be designed for DSH only, or for partial and full condensing. The schematics for these two layouts are shown in Figure 3-6 and Figure 3-7.

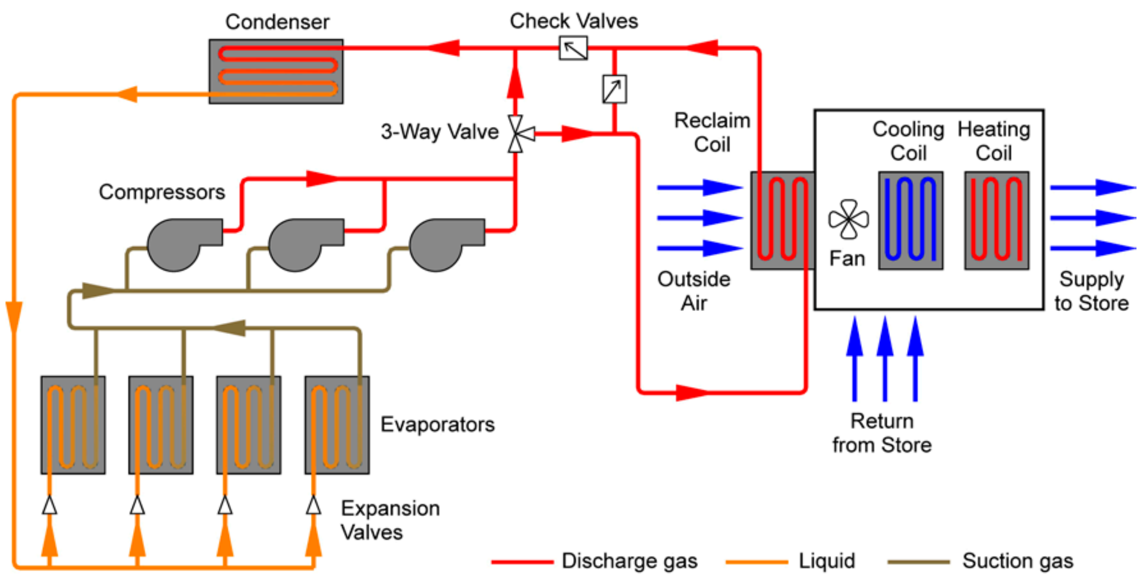


**Figure 3-6. Outdoor air preheat schematic designed for DSH**

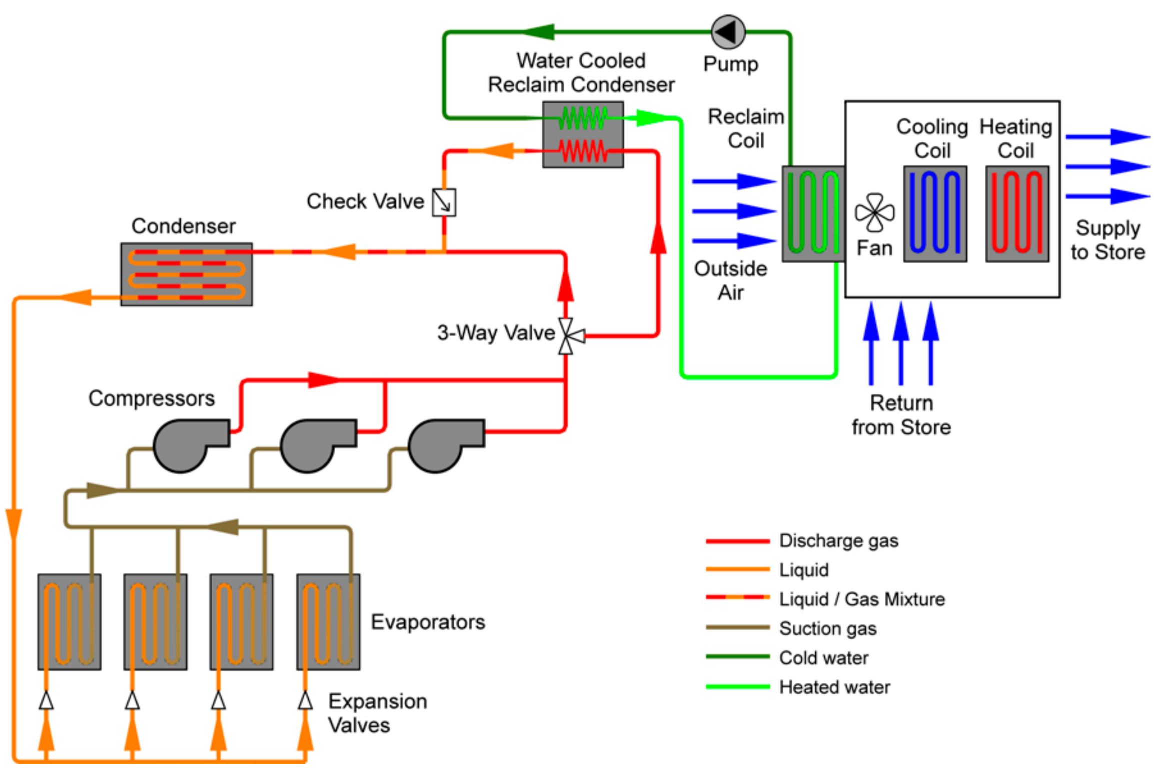


**Figure 3-7. Outdoor air preheat schematic designed for condensing**

## 3.6 Water-Source Heat Pumps

Integrating refrigeration waste heat into a WSHP loop is one of the more complex heat reclaim strategies used today. Refrigeration systems reject heat to a water loop that is shared with WSHPs used for space conditioning (see Figure 3-8). Under the right conditions, the refrigeration system and WSHPs can operate at much higher efficiencies than traditional air-cooled refrigeration systems and packaged rooftop units. When the heat pumps are in heating mode, they remove heat from the heat pump loop. If the refrigeration and heat pump loads are evenly matched, the heat pump loop can maintain temperature if no additional heat is added to, or removed from, the system.

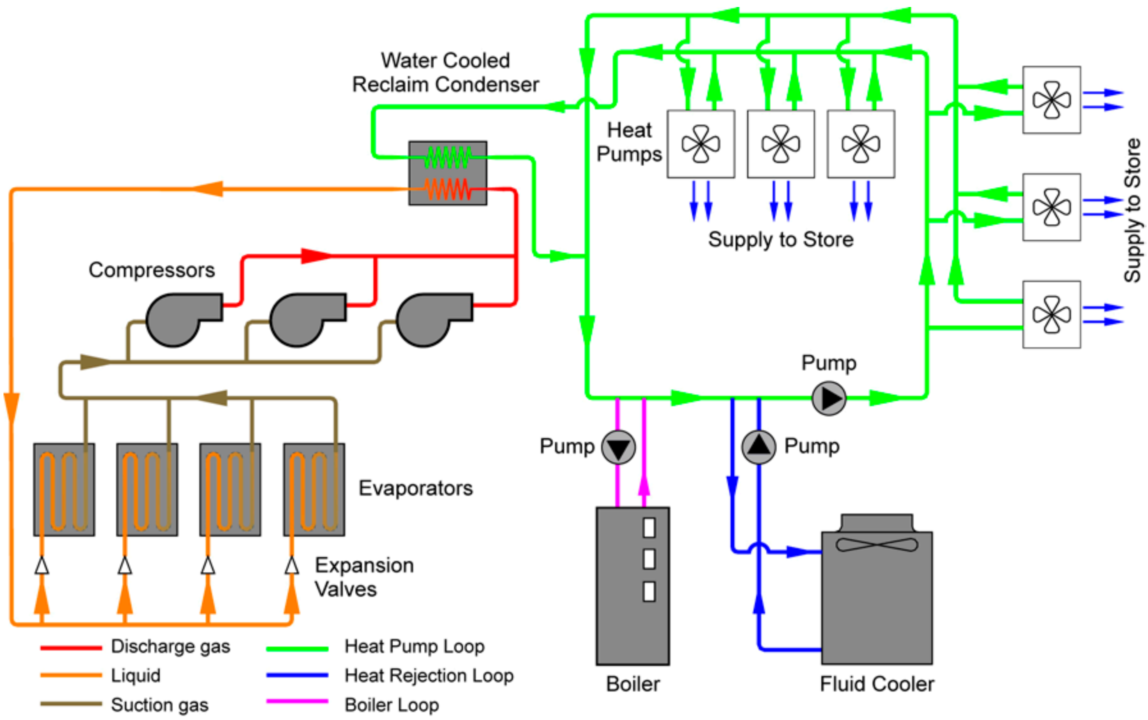


**Figure 3-8. WSHP integration designed for condensing**

The HVAC and refrigeration systems are not perfectly balanced for most of the year, so additional systems are required to maintain the loop temperature. A boiler and cooling tower combination, a ground-loop heat exchanger, or any other technology that allows the heat pump loop to maintain the desired temperature range can be used for this.

A traditional WSHP system allows the loop temperature to be approximately 60°–90°F. When a system is in cooling mode, the loop temperature tends to float to the high end of the range. When a system is in heating mode, the loop temperature drops toward the low end. When a refrigeration system is added to a WSHP system, the optimal loop temperature control strategy is less clear. Refrigeration systems operate most efficiently when the condensing temperature is lower. Although 70°F is a typical minimum condensing temperature, some systems may be able to condense at much lower temperatures. The refrigeration efficiency increases as the condensing temperature decreases; however, the heat pump efficiency decreases in heating mode. Determining the correct control strategy for this system is critical to maximizing energy savings.

## 3.7 Resources


1. Fricke, B.A. (2011). Waste Heat Recapture from Supermarket Refrigeration Systems. Oak Ridge, TN: Oak Ridge National Laboratory.
2. ASHRAE Handbook – Fundamentals. Atlanta, GA: ASHRAE, 2013; pp. 30.30.

# Chapter 4. Heat Reclaim Calculations

## 4.1 Introduction

Energy savings associated with heat reclaim strategies are not well understood and, historically, have been difficult to quantify. This chapter guides the user through the process of quantifying the energy savings associated with heat reclaim strategies and should be considered as a supplement to the heat reclaim calculation spreadsheets provided with this playbook.

To determine the energy savings associated with using waste heat from refrigeration systems, the quantity of heat available must first be determined. The total heat available from the refrigeration system is referred to as the *total heat of rejection* or *THR*. This value includes the heat rejected to cool the superheated discharge gas leaving the compressor to saturated vapor ($\dot{Q}_{DSH}$) and the heat rejected to condense the saturated vapor to saturated liquid ($\dot{Q}_{cond}$). For this analysis, the heat removed to subcool the refrigerant can typically be neglected.

$$\mathrm{THR} = \dot{Q}_{cond} + \dot{Q}_{DSH}$$

Where,

| | | |
|---|---|---|
| THR | = | the total amount of heat that must be rejected from the refrigeration system |
| $\dot{Q}_{cond}$ | = | the component of the THR from refrigerant condensing |
| $\dot{Q}_{DSH}$ | = | the component of the THR from refrigerant desuperheating |

When considering heat reclaim from refrigeration systems, understanding the quality (temperature) and quantity of heat available during times when it is useful for water or space heating is important. *Quality* refers to the refrigerant temperature at which the heat is transferred. A higher quality indicates a higher temperature. Superheated refrigerant discharge vapor temperatures are typically 120°–225°F, depending primarily on the refrigerant, evaporating temperature, return gas temperature entering the compressor, condensing temperature, and compressor type. Condensing temperatures are generally 70°–120°F and track closely to a constant offset above the OAT, down to a minimum point. Although the heat available from superheat is higher quality than that available from condensing, less heat is available. The heat available from superheat as a percentage of THR varies such that lower condensing temperatures, higher suction temperatures, reduced suction gas superheat, and more efficient compressors reduce the available compressor superheat. Accounting for all these factors, the superheat is generally 15%–30% of the THR.

For example, using heat reclaim to preheat outdoor air might be a good option in a cool climate, because the OAT when heat is required is almost always lower than the condensing temperature. Conversely, using an SHW tank that maintains a water temperature of 120°F to condense refrigerant is ineffective, because the water temperature is always higher than the refrigerant condensing temperature. In this application, superheat is more effective.

Figure 4-1 shows that the amount of superheat available is the difference in enthalpy between points D and E times the mass flow of the refrigerant.

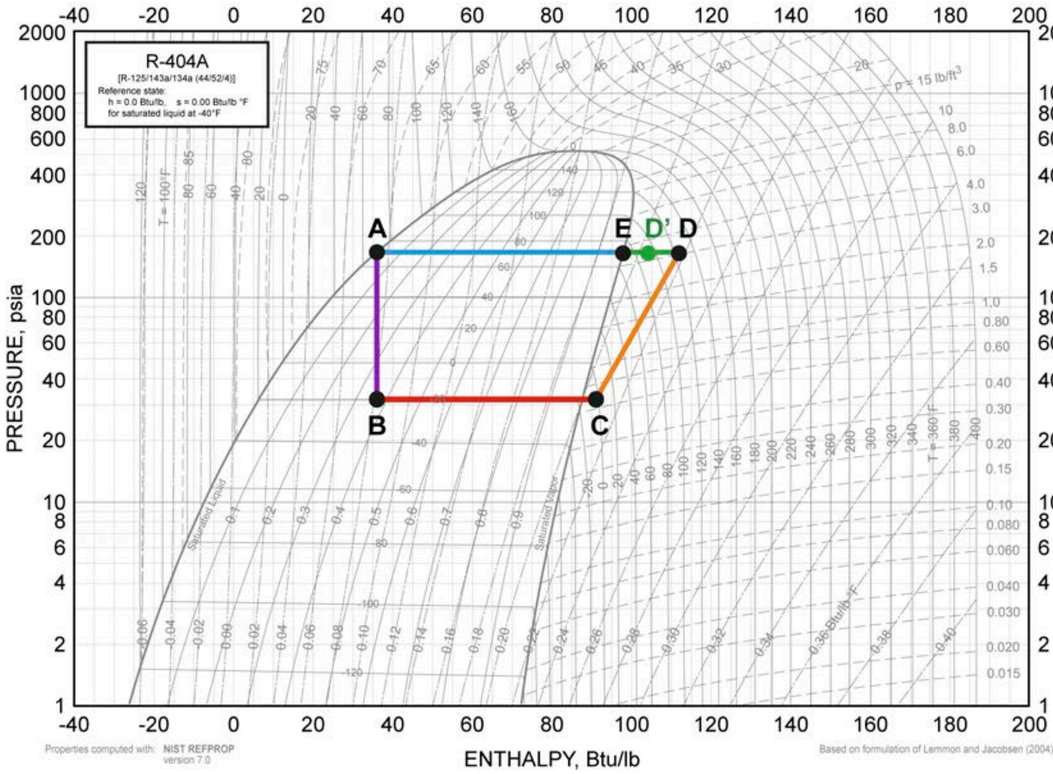


**Figure 4-1. R-404a p-h diagram**

$$\dot{Q}_{DSH} = \dot{m}(h_D - h_E)$$

Where,

$\dot{m}$ = the mass flow of the refrigerant

$h_D$ = the enthalpy of the refrigerant when it leaves the compressor at point D

$h_E$ = the enthalpy of the refrigerant at the saturated liquid point

Similarly, the heat rejected to condense the refrigerant is the difference in enthalpy between points A and E times the mass flow of the refrigerant.

$$\dot{Q}_{cond} = \dot{m}(h_E - h_A)$$

Where,

$h_A$ = the enthalpy of the refrigerant at the saturated vapor point

The enthalpy value at a given point may vary based on the source, because the assumed reference condition varies. Thus, all enthalpy values should be obtained from a single source.

The thermodynamic properties of the refrigerant at these points are affected by properties inherent to the refrigerant, outdoor air conditions, system type, and the specific design parameters of the system. The refrigerant mass flow can be determined based on the evaporator load and the specific enthalpy values for the refrigerant entering and leaving the evaporator, shown as points B and C in Figure 4-1. The mass flow may also be determined from an energy model or from the compressor operating characteristics. Some manufacturers provide compressor coefficients that allow the user to calculate the mass flow at various compressor suction and discharge temperatures. Some factors that can influence the amount of superheat available for heat reclaim follow:

- **Evaporator load.** Typical rated case design load conditions in a grocery sales area are dry-bulb (db) temperature of 75°F and relative humidity of 55%. Design loads typically include extra capacity to reduce the temperature to set point after defrost, among other considerations and safety factors. The combination of these factors can cause the evaporator loads to fall far below the design loads, which directly reduces the total heat available for reclaim.
- **Floating condensing temperature and pressure.** As the condensing temperature and pressure of a system decrease, the temperature and amount of available superheat decrease.
- **Compressor suction superheat.** Compressor manufacturers require a certain amount of superheat to ensure no liquid returns to the compressor. The rated condition for compressors is typically 65°F return gas temperature per American National Standards Institute/American Heating and Refrigeration Institute (ANSI/AHRI) 540. The actual superheat temperature for a typical supermarket system usually exceeds evaporator temperature by 30°–40°F, based on typical piping runs and insulation thicknesses, resulting in return gas temperatures of 20°–50°F. This value may be higher with liquid suction heat exchangers, which are used to subcool liquid refrigerant with suction gas. This condition is represented by point C in Figure 4-1.
- **Compressor efficiency.** The efficiency of the compressors determines the slope of the line between C and D in Figure 4-1. Ideal compression is an isentropic or constant entropy process and follows the line of constant entropy on the p-h diagram resulting in a compressor discharge point at point D'. A real compressor produces more superheat in the compression process and results in a discharge point at D.

The state points of the refrigerant can be determined using refrigerant saturation tables and manufacturer selection software programs. The drawback to this method is that determining the length of time a system operates at a given condition is difficult. Load profiles can be paired with weather bin data to estimate these values, but the best way to determine the superheat and condensing heat rejection is with an energy model or the spreadsheets provided with this playbook. Several energy modeling packages, such as EnergyPlus, are available that can simulate a refrigeration system.

Several variables need to be defined to determine the amount of heat available from superheat and condensing. Rules of thumb for percent superheat are sometimes useful; however, calculating the percentages every hour is more accurate. The following variables are needed to calculate the condensing and DSH heat transfer rates:

- **THR.** This is the total heat rejected by the condenser. It is represented by the difference in enthalpy between points A and D in Figure 4-1 times the mass flow.
- **Condensing temperature.** This value varies with OAT, especially in systems that can float the condensing temperature. It is roughly the same at points A and E for azeotropic refrigerants, but can vary dramatically for zeotropic refrigerants.
- **Mass flow.** This parameter allows the enthalpy differences to be converted to heat transfer values.

Once these values have been determined, the condensing temperature can be used to find the enthalpy at points A and E in the refrigerant saturation tables. The difference in enthalpy is then multiplied by the mass flow to determine the condensing heat transfer ($\dot{Q}_{cond}$). This quantity can be subtracted from the THR to determine the total DSH ($\dot{Q}_{DSH}$) available.

Five spreadsheets were created to support calculation of energy savings of the common heat reclaim systems described in Chapter 3:

- **Refrigeration System Front End.** Used to calculate the amount of heat available from the refrigeration system each hour.
- **Space Heating Load.** Used to calculate space heating requirements each hour.
- **SHW.** Used to calculate energy reclaimed for SHW.
- **Space Heating Reclaim.** Used to calculate energy reclaimed for space heating.
- **Outdoor Air Preheat.** Used to calculate energy reclaimed to preheat outdoor air before it enters the air system.

## 4.2 Refrigeration System Front-End Spreadsheet

The Refrigeration System Front End spreadsheet is a tool that a refrigeration designer can use to rapidly calculate refrigeration system performance if a more detailed calculation or an energy model is not feasible or is otherwise unavailable. The outputs from this spreadsheet are intended to be used as inputs to the subsequent playbook spreadsheets to determine the feasibility of various heat recovery strategies. This tool assumes that the refrigeration system being analyzed has a single suction group. Condenser options include air-cooled, evaporative, and hybrid, with constant- or variable-speed fan control. More complicated systems may be analyzed by combining the results of multiple spreadsheet runs. A screenshot of the inputs page of this tool is shown in Figure 4-2.

| System User Inputs Section | | | | | |
|---|---|---|---|---|---|
| Refrigeration System Inputs | | DESIGN Condensing Conditions - Compressor Characteristics | | MINIMUM Condensing Conditions - Compressor Characteristics | |
| Refrigerant | R404A | Condensing Temperature (°F) | 101.0°F | Condensing Temperature (°F) | 70°F |
| System Operating Temperature | Low | Compressor Capacity (BTU/hr) | 31,300 | Compressor Capacity (BTU/hr) | 40,900 |
| Rated Evaporator Capacity (BTU/hr) | 100,000 | Net Refrigeration Capacity (BTU/hr) | 27,300 | Net Refrigeration Capacity (BTU/hr) | 36,800 |
| Rated Runtime Ratio | 0.85 | System Massflow Rate (lb/hr) | 660 | System Massflow Rate (lb/hr) | 695 |
| Condensing Method | Air-Cooled | Compressor Input Power (W) | 6,250 | Compressor Input Power (W) | 5,300 |
| Condenser ΔT (°F) | 10°F | ***Note**: Refrigeration estimates do not account for scheduled operation of integrated refrigeration system components, i.e. anti-sweats, lighting, defrost, etc.* | | | |
| Condenser Fan Control Strategy | Constant | | | | |
| THR to Fan Power Ratio (BTU/hr/W per 1°F ΔT) | 5.0 | | | | |

| Site Location | | CZ5A - Boston, MA | | *Click Here to Copy Results Data for Export* | Results Section for Export to Refrigeration Playbook Spreadsheets | | | | | | | | Superheat to THR Ratio |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Index | Month | Day | Hour | Wet-Bulb Temp. (°F) | Dry-Bulb Temp. (°F) | Discharge Temp. (°F) | Saturated Vapor Temp. (°F) | Massflow (lb/hr) | Condenser THR (BTU/hr) | Superheat Capacity (BTU/hr) | Compressor Power (W) | Fan Power (W) | |
| | | | | -5.7 MIN | -4.0 MIN | 157.0 MIN | 70.5 MIN | 1,284 MIN | 108,991 MIN | 28,829 MIN | 9,793 MIN | 293 MIN | 0.264 MIN |
| | | | | 45.5 AVG | 51.1 AVG | 161.2 AVG | 74.2 AVG | 1,429 AVG | 120,225 AVG | 32,888 AVG | 11,269 AVG | 1,502 AVG | 0.272 AVG |
| | | | | 77.5 MAX | 99.0 MAX | 204.0 MAX | 109.3 MAX | 2,225 MAX | 174,273 MAX | 63,062 MAX | 22,184 MAX | 3,374 MAX | 0.362 MAX |
| 1 | 1 | 1 | 1:00 | 31.4 | 35.1 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 616 | 0.26 |
| 2 | 1 | 1 | 2:00 | 31.4 | 35.1 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 616 | 0.26 |
| 3 | 1 | 1 | 3:00 | 30.8 | 34.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 598 | 0.26 |
| 4 | 1 | 1 | 4:00 | 30.8 | 34.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 598 | 0.26 |
| 5 | 1 | 1 | 5:00 | 30.5 | 34.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 598 | 0.26 |
| 6 | 1 | 1 | 6:00 | 30.5 | 34.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 598 | 0.26 |
| 7 | 1 | 1 | 7:00 | 30.0 | 33.1 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 583 | 0.26 |
| 8 | 1 | 1 | 8:00 | 29.7 | 33.1 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 583 | 0.26 |
| 9 | 1 | 1 | 9:00 | 29.7 | 33.1 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 583 | 0.26 |
| 10 | 1 | 1 | 10:00 | 28.5 | 32.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 567 | 0.26 |
| 11 | 1 | 1 | 11:00 | 29.6 | 34.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 598 | 0.26 |
| 12 | 1 | 1 | 12:00 | 29.9 | 34.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 598 | 0.26 |
| 13 | 1 | 1 | 13:00 | 30.8 | 36.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 632 | 0.26 |
| 14 | 1 | 1 | 14:00 | 30.9 | 37.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 652 | 0.26 |
| 15 | 1 | 1 | 15:00 | 30.9 | 37.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 652 | 0.26 |
| 16 | 1 | 1 | 16:00 | 30.0 | 36.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 632 | 0.26 |
| 17 | 1 | 1 | 17:00 | 28.6 | 34.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 598 | 0.26 |
| 18 | 1 | 1 | 18:00 | 26.9 | 32.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 567 | 0.26 |
| 19 | 1 | 1 | 19:00 | 24.4 | 30.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 539 | 0.26 |
| 20 | 1 | 1 | 20:00 | 23.0 | 28.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 514 | 0.26 |
| 21 | 1 | 1 | 21:00 | 21.4 | 26.1 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 491 | 0.26 |
| 22 | 1 | 1 | 22:00 | 19.9 | 24.1 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 470 | 0.26 |
| 23 | 1 | 1 | 23:00 | 18.3 | 21.9 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 449 | 0.26 |
| 24 | 1 | 1 | 24:00 | 17.3 | 19.9 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 431 | 0.26 |
| 25 | 1 | 2 | 1:00 | 17.1 | 19.9 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 431 | 0.26 |
| 26 | 1 | 2 | 2:00 | 16.3 | 19.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 424 | 0.26 |
| 27 | 1 | 2 | 3:00 | 15.4 | 18.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 415 | 0.26 |
| 28 | 1 | 2 | 4:00 | 15.4 | 18.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 415 | 0.26 |
| 29 | 1 | 2 | 5:00 | 15.3 | 18.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 415 | 0.26 |
| 30 | 1 | 2 | 6:00 | 14.6 | 17.1 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 408 | 0.26 |
| 31 | 1 | 2 | 7:00 | 14.6 | 17.1 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 408 | 0.26 |
| 32 | 1 | 2 | 8:00 | 14.8 | 17.1 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 408 | 0.26 |
| 33 | 1 | 2 | 9:00 | 16.4 | 19.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 424 | 0.26 |
| 34 | 1 | 2 | 10:00 | 18.5 | 21.9 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 449 | 0.26 |
| 35 | 1 | 2 | 11:00 | 20.6 | 24.1 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 470 | 0.26 |
| 36 | 1 | 2 | 12:00 | 22.8 | 26.1 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 491 | 0.26 |
| 37 | 1 | 2 | 13:00 | 25.2 | 28.9 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 525 | 0.26 |
| 38 | 1 | 2 | 14:00 | 26.7 | 30.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 539 | 0.26 |
| 39 | 1 | 2 | 15:00 | 26.7 | 30.0 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 539 | 0.26 |
| 40 | 1 | 2 | 16:00 | 26.2 | 28.9 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 525 | 0.26 |
| 41 | 1 | 2 | 17:00 | 26.2 | 28.9 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 525 | 0.26 |
| 42 | 1 | 2 | 18:00 | 26.2 | 28.9 | 157.0 | 70.5 | 1,284 | 108,991 | 28,829 | 9,793 | 525 | 0.26 |

**Figure 4-2. Refrigeration System Front End spreadsheet**

The designer needs to obtain design information about the system to be analyzed before using the spreadsheet tool. This information is typically provided in the refrigeration design documents, if they are available. The designer should perform a site visit and gain access to system control logs to verify the performance data and minimize assumptions about system design and operation.

The Refrigeration System Front End spreadsheet tool estimates hourly system performance over the course of an entire year by interpolating minimum and maximum design conditions to hourly weather data and applying this performance as a weighting factor of estimated hourly end-use loads. Performance data may also be refined with a more detailed spreadsheet analysis using ANSI/AHRI Standard 1200 compressor coefficients, which are provided by compressor manufacturers for commercial equipment, or energy modeling software such as OpenStudio, EnergyPlus, and eQuest.

Hourly OAT data for the spreadsheet tool may be acquired from multiple sources. The National Renewable Energy Laboratory (NREL) freely provides Typical Meteorological Year data that may be used to perform common annual calculations. Actual hourly data may be purchased from third-party providers of weather data or downloaded from hourly station observations reported to weather databases such as Weather Underground. For air-cooled condensing systems, db temperatures must be used for proper calculation. For evaporative and hybrid condensing systems, wet-bulb (wb) temperatures must be entered. Hourly wb data can be output from EnergyPlus, eQuest, or other building energy modeling software.

The spreadsheet tool provides a set of enthalpy properties for several refrigerants commonly used in commercial refrigeration: R22, R134a, R404a, R407a, R407c, and R507. The enthalpy table provides the enthalpy differential at a given condensing temperature for the equation: $\dot{Q}_{cond} = \dot{m}(h_E - h_A)$. A final "Spare" field is provided in the table so the designer can enter the enthalpy properties of a refrigerant not included in the spreadsheet. Enthalpy properties may be obtained from the *ASHRAE Handbook of Fundamentals* Chapter 30, NIST Refprop, or the refrigerant manufacturer's documentation. The latent enthalpy differential is calculated by subtracting the saturated liquid enthalpy ($h_A$) from the saturated vapor enthalpy ($h_E$) at the reference pressure point.

The Refrigeration System Front End spreadsheet tool assumes that the refrigeration system performance can be interpolated linearly based on condensing temperature, using the minimum condensing set point and the design condensing temperature as end points. Evaporator capacities are adjusted by the rated runtime fraction input and an algorithm developed by Faramarzi et al. (2004) for the California Energy Commission. The algorithm adjusts low- and medium-temperature display case capacities based on OATs, incorporating assumptions for how the indoor conditions relate to the outdoor conditions. The rated runtime fraction is defined by display case manufacturers and is a factor of oversizing the evaporator coil to allow for quick temperature pulldown after defrost or other periods when heat loads exceed rated conditions.

$$\text{California Energy Commission load factor} = 1 - (1 - \text{min}) \times [(85 - \text{OAT}) \div (85 - 40)]$$

Where,

| | | |
|---|---|---|
| min | = | the minimum fraction of design load (0.66 for medium temperature and 0.8 for low temperature) |
| OAT | = | the ambient db temperature |

Inputs for the spreadsheet tool must be obtained from the refrigeration design documents and manufacturer's rated data for compressors and condensers. Refrigeration performance data must include the proper return gas temperature. The ANSI/AHRI standard rating of 65°F return gas temperature may cause the available superheat to be overestimated. Return gas temperature is the temperature of the suction gas entering the compressor. Compressor manufacturers specify a minimum amount of superheat to prevent the liquid from returning to and damaging a compressor, which should be addressed at the evaporator coil. After the refrigerant vapor leaves the evaporator, the suction gas is again superheated when the refrigerant gas draws heat from a conditioned space as it returns to the compressor. The amount of superheat depends on the distance between the evaporator and the compressor and the effectiveness of the insulation on the suction gas piping. Typical commercial refrigeration system return gas temperatures are 20°–30°F over the system evaporating temperature set point.

Similarly, when gathering compressor performance data, the evaporator temperature input should be reduced by 3°F for low-temperature systems and 2°F for medium-temperature systems. This is a common design practice to account for the pressure drop that is caused by pipe friction. If actual temperature set points can be obtained from the refrigeration system controller, they should be used in lieu of documented design data.

The output results from the Refrigeration System Front End spreadsheet may then be copied or linked to the subsequent spreadsheets included with this playbook.

## 4.3 Space Heating Load Calculations

To calculate the quantity of refrigeration heat that can be reclaimed for building heating, several calculations must be performed to determine the building's heating requirements at every hour of the year. These calculations could be performed using bin hour calculations; however, Typical Meteorological Year data sets are available and the relative accuracy has improved, so hourly calculations are used in this playbook.

A detailed energy model is advantageous because it takes into account nearly every aspect of building performance each hour, accounts for interactions between building systems, and is more accurate than hand calculations. If an energy model of a building is available, the authors highly recommend that the designer use it as a starting point for heat recovery calculations. Important energy model outputs are hourly total heating load, hourly MAT, hourly fan airflow rate, and hourly outdoor airflow rate.

If these values are not available from an energy model, they must be approximated using a simplified method. One such method is the Space Heating Load spreadsheet available with this guide.

A simplified method can be used if load calculations are available. This method assumes a linear relationship between building load and the OAT. To use this method, the designer must assume a balance point for the zone being evaluated and assign an occupied mode schedule to the HVAC unit. A typical balance point for a refrigerated area may be 60°–75°F.

Assuming a linear relationship between OAT and zone load, the hourly heating load can be calculated as:

$$\dot{Q}_{des} = A \times OAT + B$$

Where,

| | | |
|---|---|---|
| $\dot{Q}_{des}$ | = | Peak design heating load |
| OAT | = | Outdoor air temperature |
| A | = | the slope of the linear curve fit |
| B | = | the y-axis intercept of a line equation |

The slope of the line (A) can be determined by a rise over the run calculation. The rise, or maximum Y-value, is the peak heating load. The run, or difference in X-value, is the balance point temperature ($T_{bal}$) minus the temperature at which the peak heating load occurs ($T_{des}$).

$$A = \dot{Q}_{des} / (T_{bal} - T_{des})$$

Where,

| | | |
|---|---|---|
| $T_{bal}$ | = | the balance point temperature |
| $T_{des}$ | = | the temperature at which the peak heating load occurs |

Once the slope of the line is determined, the constant load factor (B) can be determined by choosing a point on the line. For example, the design conditions can be plugged in for Q and OAT.

$$B = \dot{Q}_{des} - (A \times T_{des})$$

Figure 4-3 uses the simplified linear relationship to illustrate building heating and cooling loads by OAT.

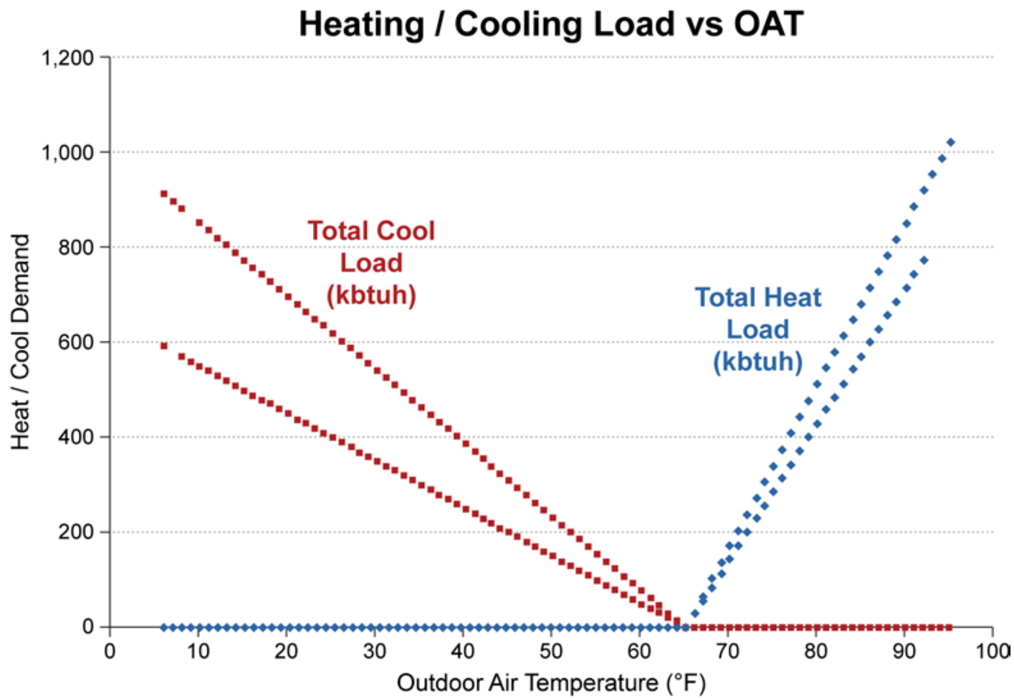


**Figure 4-3. Heating and cooling loads by OAT**

This figure shows two lines associated with heating and cooling. The lower line is the envelope load only and represents the unoccupied hours when no ventilation is supplied to the zone. The higher line represents the envelope load plus the hourly outdoor air thermal load.

The hourly heating loads and airflows are used when calculating the heat recovered for space heating. The outdoor airflows and conditions are used when calculating outdoor air preheating.

## 4.4 Service Water Preheating

SHW is heat reclaimed using a refrigerant coil submerged in a service water tank (assumed here to be isothermal). Superheated refrigerant vapor travels through the coil from the top of the tank to the bottom of the tank. The water temperature in the tank is controlled to a maximum temperature equal to the SHW set point. Typical SHW temperatures are 120°–140°F, so in most cases, the refrigerant cannot condense or fully desuperheat. However, much of the time (during periods of high SHW use), the tank temperature drops well below the tank temperature set point, allowing for more heat reclaim. The p-h diagram in Figure 4-4 illustrates the DSH process, where D is the compressor discharge point, D’ is a variable point that represents the state of the refrigerant leaving the tank, and E represents a point where the refrigerant is fully desuperheated.

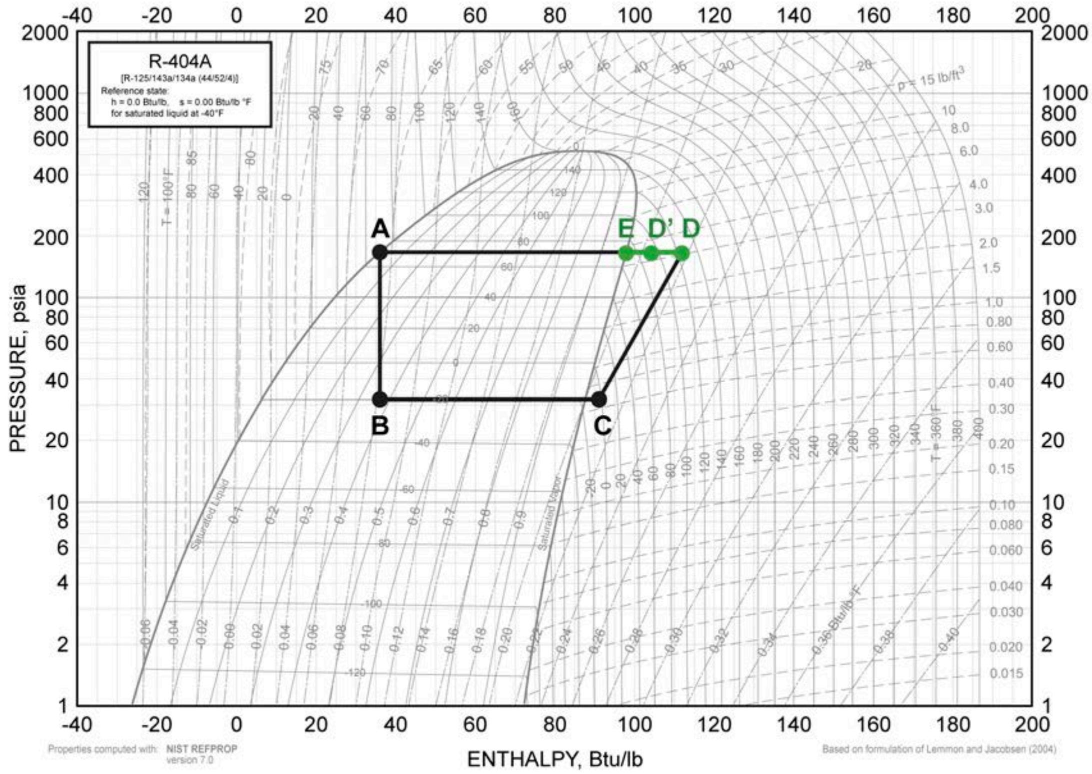


**Figure 4-4. R-404a p-h diagram (ASHRAE 2013)—DSH highlighted**

To accurately model the performance of a heat recovery tank, one must consider the benefit of storage capacity. For example, building SHW use might be very high in the evenings as employees are cleaning and dishwashers are running. During this peak period, the heat reclaim tank temperature likely approaches the domestic cold water temperature. For the first hour after this period of peak use, the heat that the tank can reclaim from the refrigeration system is the heat required to bring the tank temperature up to set point plus the heat required to bring the incoming cold water temperature up to set point.

To factor in the storage effects, the tank temperature is calculated at each hour:

$$T_{tank} = [(V_{Win} \times T_{Win}) + ((V_{Wtank} - V_{win}) \times T_{Wtank})] / V_{Wtank}$$

Where,

| | | |
|---|---|---|
| $T_{tank}$ | = | the hourly temperature of the tank after the hourly hot water flow has left the tank and been replaced by the same amount of domestic cold water |
| $V_{Win}$ | = | the volume of water that enters and leaves the tank |
| $T_{Win}$ | = | the temperature of the domestic cold water entering the tank |
| $V_{Wtank}$ | = | the heat reclaim tank volume |
| $T_{Wtank}$ | = | the temperature of the water in the tank from the previous hour |

Once the hourly tank temperature has been calculated, the amount of heat that can be transferred to the water can be calculated. To fully understand the heat transfer between the water and the refrigerant, the refrigeration side of the heat balance must be evaluated.

In the p-h diagram in Figure 4-4, point D represents the compressor discharge state point. This is the maximum temperature in the cycle. The superheat region available for water heating is the range from point D to E. Under many conditions, the water temperature is higher than the condensing temperature. During these periods, the tank cannot remove all the superheat from the refrigerant. This state point is represented by point D' on the p-h diagram.

An energy balance calculation must be performed to determine the conditions at this point. To calculate the impact of the storage tank, a steady-state heat transfer equation is used in place of a heat transfer rate equation on the water side of the energy balance. Because these calculations take place every hour, the heat transfer rate is assumed to be steady state for 1 hour. When writing the equations, both heat transfer quantities are written as a heat transfer rate.

$$\dot{Q}_R = \dot{Q}_w$$

Where,

$\dot{Q}_W$ = the heat added to the domestic water

$\dot{Q}_R$ = the heat removed from the refrigerant

Using these equations and the first law of thermodynamics, an equation can be derived to solve for the temperature of the water leaving the tank and temperature of the refrigerant, leaving the coil starting with the first law of thermodynamics for the refrigerant side:

$$\dot{Q}_R = \dot{m}_R \times c_{pR} \times \Delta T_R$$

Where,

$\Delta T_R$ = the temperature difference between the entering and leaving refrigerant

$c_{pR}$ = the specific heat of the refrigerant

A similar equation can be written for the water side:

$$\dot{Q}_W = \dot{m}_W \times c_{pW} \times \Delta T_W$$

Where,

$\Delta T_w$ = the temperature difference between the entering and leaving water

$c_{pW}$ = the specific heat of the water

An assumption must be made about the relative water and refrigerant outlet temperatures. For these calculations—and under ideal conditions—the temperature of the refrigerant leaving the coil ($T_{Ro}$) is assumed to approach the tank temperature, which is equivalent to the temperature of the water leaving the tank (LWT).

$$LWT_{ideal} = T_{Ro}$$

Where,

| | | |
|---|---|---|
| $T_{Ro}$ | = | the temperature of the refrigerant leaving the coil |
| LWT | = | the temperature of the water leaving the tank |

The temperature difference on the refrigerant side is the difference between the temperature of the refrigerant into the coil and its temperature leaving the coil. For these calculations—and in an ideal system—the $LWT_{ideal}$ is assumed to approach the temperature of the refrigerant out of the coil ($T_{ro}$):

$$\Delta T_R = T_{ri} - T_{ro} = T_{Ri} - LWT_{ideal}$$

Where,

| | | |
|---|---|---|
| $LWT_{ideal}$ | = | the ideal leaving water temperature at the end of the hour, or the temperature of the water leaving the tank |
| $T_{ro}$ | = | the temperature of the refrigerant out of the coil |
| $T_{Ri}$ | = | the discharge temperature |

The amount of heat transferred from the refrigerant must be equal to that added to the water, so the water-side and refrigerant-side heat transfer equations can be equated:

$$\dot{m}_R \times c_{pR} \text{ x } (T_{Ri} - LWT_{ideal}) = \dot{m}_W \times c_{pW} \times (LWT_{ideal} - EWT)$$

Where,

| | | |
|---|---|---|
| EWT | = | temperature of the water entering the tank at the beginning of the hour |
| $\dot{m}_R$ | = | the refrigerant mass flow |
| $\dot{m}_W$ | = | the water mass flow |

To solve for the LWT, we can divide both sides by ($\dot{m}_W \times c_{pW} \times \Delta T_W$):

$$[(\dot{m}_R \text{ x } c_{pR}) / (\dot{m}_W \times c_{pW})] \times (T_{Ri} - LWT_{ideal}) = (LWT_{ideal} - EWT)$$

To simplify this equation we create a constant:

$$C = (\dot{m}_R \times c_{pR}) / ((\dot{m}_W \times c_{pW})$$

The equation is then rewritten and solved for the $LWT_{ideal}$.

$$LWT_{ideal} = (C \times T_{Ri} + EWT) / (1 + C)$$

Once the ideal LWT is determined, the ideal heat transfer can be calculated using the first law of thermodynamics for water systems.

$$\dot{Q}_{ideal} = \dot{m}_W \times c_{pW} \times (LWT_{ideal} - EWT)$$

When using imperial units (IP), this equation can be written in terms of water flow in GPM, temperatures in °F, and heat transfer rate in Btu/h.

$$\dot{Q}_{ideal} = 500.5 \times GPM \times (LWT_{ideal} - EWT)$$

The result of this equation is the ideal heat transfer for an infinitely large heat exchanger. An effectiveness or efficiency value ($\varepsilon$) must be assigned to the coil to correct this. This value can be determined from coil selection software using the actual inlet and outlet temperatures.

$$\dot{Q}_{actual} = \dot{Q}_{ideal} \times \varepsilon$$

Where,

$\dot{Q}_{actual}$ = the actual heat transfer considering effectiveness

$\dot{Q}_{ideal}$ = the ideal heat transfer

$\varepsilon$ = the effectiveness of the heat exchange

To provide the hourly domestic water use input for these calculations, several assumptions must be made. Measured data are usually not available for building hot water use. Very few public data are available to influence or advise assumptions. The *ASHRAE 90.1-2004 User Manual* (Table G-L) provides a good resource for determining schedules of hot water consumption for retail buildings. The "Refrigeration Playbook - Hot Water Recovery Calculator" applies the hourly schedules from Table G-L to a daily demand volume to simulate an hourly hot water demand. The daily demand volume of a supermarket can range substantially based on the scale of food preparation departments. The authors recommend discussing the operations of these areas with the building manager and using water billing data that are available to assist with assumptions for building hot water consumption.

The other major influence in determining hot water recovery is the building supply water temperature. The "Refrigeration Playbook - Hot Water Recovery Calculator" uses an algorithm described in the technical report *Towards Development of an Algorithm for Mains Water Temperature*. This algorithm provides a method for estimating temperatures of water mains throughout a year based on monthly average temperatures and the annual average temperature. Constants provided in the paper were incorporated into the spreadsheet to generate a sinusoidal hourly building supply water temperature profile using the input weather data. If a simpler input is desired, the groundwater temperature may be assumed to be consistent throughout the year per *ASHRAE Handbook of HVAC Applications 2011*, Chapter 34, Figure 17.

## 4.5 Mixed Air Heating

Heat reclaim for space heating is similar to heat reclaim for outdoor air preheat; the primary difference is that the heat from the refrigeration system is transferred to the HVAC unit's entire mixed airstream instead of only outdoor air. The term *mixed air* refers to the mix of return air from the space being served by the unit and the outdoor air being introduced into the space through the unit.

Placing the heat reclaim coil in the mixed airstream has advantages and disadvantages. The major advantage over outdoor air preheat is that more heat can be transferred, because usually more airflow is in the mixed airstream than in the outdoor air alone. As a rule, the total mixed airflow is about 5 times greater than the outdoor airflow. Also, the heat load is larger because it includes the outdoor air preheat load as well as the building envelope and refrigeration loads.

The biggest drawback to using heat reclaim for space heating is that the temperature differential (TD) between the refrigeration waste heat and the mixed airstream is smaller than that of the outdoor air. For example, if the OAT is 0°F, the space return air temperature is 65°F, and outdoor airflow represents 20% of the total airflow, the MAT is 52°F. The MAT can be calculated using the following equation:

$$\text{MAT} = \text{OAT}(\text{OA ratio}) + \text{RAT}(\text{RA ratio})$$

| | | |
|---|---|---|
| MAT | = | the mixed air temperature |
| OAT | = | the outdoor air temperature |
| RAT | = | the return air temperature |
| OA ratio | = | the outdoor airflow divided by the total airflow |
| RA ratio | = | the return air divided by the total airflow |

Because the TD between the condensing temperature and the mixed airstream is generally fairly low, other strategies for reclaiming this heat need to be considered. In climates where significant dehumidification is required, the waste heat may be used to reheat the air coming off the cooling coil. In a typical dehumidification system, air is cooled to 50°–52°F, then reheated to a more comfortable temperature before entering the space. This results in a larger TD between the airstream and the condensing temperature.

Alternatively, a coil in the HVAC unit airstream can be used to desuperheat—rather than completely condense—the refrigerant. Because superheated refrigerant temperatures are much higher than condensing temperatures, this is a much more usable heat source; however, as discussed earlier, superheat represents only about 15%–30% of a refrigeration system's THR.

This study addresses four options for heat reclaim for space heating:

- DSH with a water coil
- DSH with a refrigerant coil
- Full condensing with a water coil
- Full condensing with a refrigerant coil.

## 4.6 Desuperheat for Space Heating

The first part of refrigeration heat reclaim for space heating that will be addressed is DSH. Figure 4-5 illustrates the DSH process.

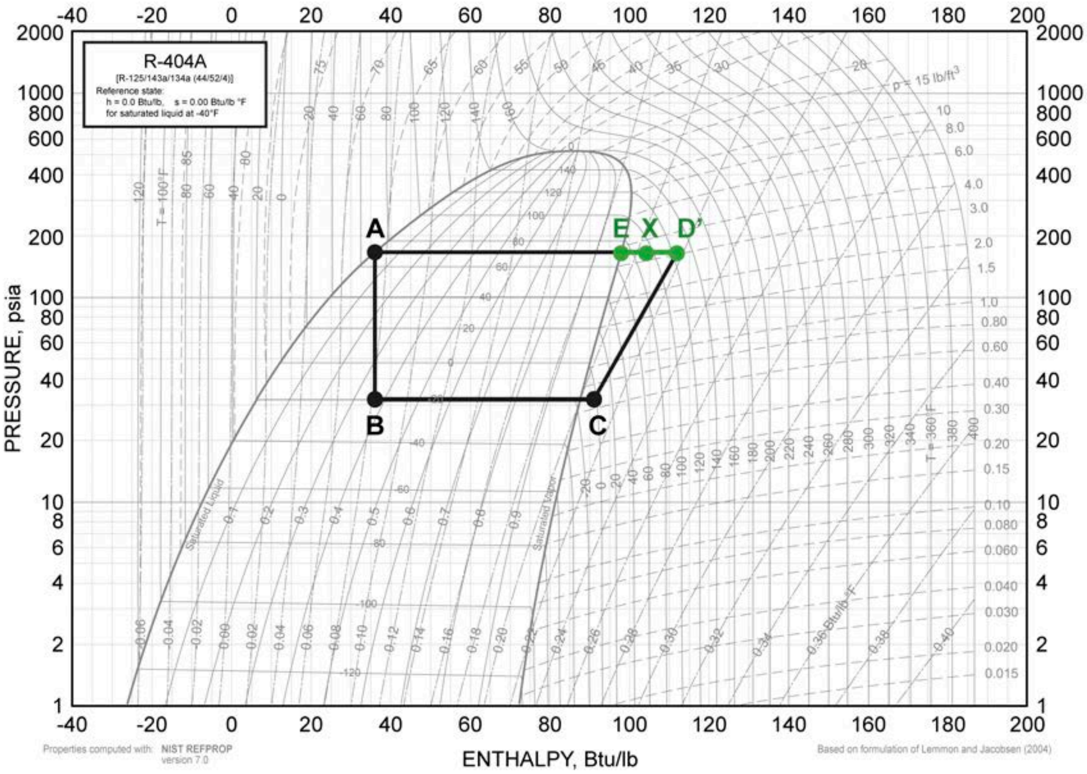


**Figure 4-5. R-404a p-h diagram (ASHRAE 2013)—DSH highlighted**

In Figure 4-5, point D represents the compressor discharge state point. This is the maximum temperature in the cycle. The superheat region available for space heating is the range from point D to point E. Under some conditions, the MAT may approach or surpass the condensing temperature. During these periods, the mixed air cannot remove all the superheat from the refrigerant. This state point is represented by point X. An energy balance calculation must be performed to determine the conditions at this point. The heat transferred to the airstream ($\dot{Q}_A$) must be equivalent to the heat removed from the refrigerant ($\dot{Q}_R$).

$$\dot{Q}_R = \dot{Q}_A$$

Where,

| | | |
|---|---|---|
| $\dot{Q}_A$ | = | the heat transferred to the airstream |
| $\dot{Q}_R$ | = | the heat removed from the refrigerant |

These equations and the first law of thermodynamics are used to derive an equation to solve for the temperature of the air leaving the coil and temperature of the refrigerant leaving the coil, starting with the refrigerant side:

$$\dot{Q}_R = \dot{m}_R \times c_{pR} \times \Delta T_R$$

Where,

| | | |
|---|---|---|
| $\dot{m}_R$ | = | the refrigerant mass flow |

| | | |
|---|---|---|
| $c_{pR}$ | = | the specific heat of the refrigerant |
| $\Delta T_R$ | = | the temperature difference between the entering and leaving refrigerant |

A similar equation can be written for the airside:

$$\dot{Q}_A = \dot{m}_A \times c_{pA} \times \Delta T_A$$

Where,

| | | |
|---|---|---|
| $\dot{m}_A$ | = | the water mass flow |
| $c_{pA}$ | = | the specific heat of the water |
| $\Delta T_A$ | = | the temperature difference between the entering and leaving air |

In addition to the energy balance, these calculations assume that under ideal conditions the temperature of the refrigerant leaving the air coil ($T_{Ro}$) approaches the temperature of the air leaving the coil (LAT). This is not entirely accurate, because a coil with multiple rows could theoretically approach the discharge temperature. However, this is a close approximation of typical conditions.

$$LAT_{ideal} = T_{Ro}$$

Where,

| | | |
|---|---|---|
| $LAT_{ideal}$ | = | temperature of the air leaving the coil |
| $T_{Ro}$ | = | temperature of the refrigerant leaving the air coil |

The temperature difference on the refrigerant side is the difference between the temperature of the refrigerant into the coil and the temperature of the refrigerant leaving the coil. For these calculations—and assuming an ideal system—the $LAT_{ideal}$ approaches the temperature of the refrigerant leaving the coil ($T_{ro}$):

$$\Delta T_R = T_{ri} - T_{ro} = T_{Ri} - LAT_{ideal}$$

Where,

| | | |
|---|---|---|
| $T_{Ri}$ | = | the discharge temperature |
| $T_{Ro}$ | = | the refrigerant outlet temperature |
| $LAT_{ideal}$ | = | the MAT leaving the coil |

The heat transfer of the refrigerant must be equal to the heat transfer of the air, so the airside and refrigerant-side heat transfer equations can be equated:

$$\dot{m}_R \times c_{pR} \text{ x } (T_{Ri} - LAT_{ideal}) = \dot{m}_A \times c_{pA} \times (LAT_{ideal} - MAT)$$

Where,

| | | |
|---|---|---|
| MAT | = | the mixed air temperature entering the coil |

To solve for the LAT, divide both sides by $(\rho_{act}/\rho_{std}) \times 1.08 \times CFM)$:

$$[(\dot{m}_R \times c_{pR}) / (\dot{m}_A \text{ x } c_{pA})] \times (T_{Ri} - LAT_{ideal}) = (LAT_{ideal} - MAT)$$

To simplify this equation, a constant can be created:

$$C = (\dot{m}_R \times c_{pR}) / (\dot{m}_A \times c_{pA})$$

The equation is then rewritten and solved for the $LAT_{ideal}$.

$$LAT_{ideal} = (C \times T_{Ri} + MAT) / (1 + C)$$

Once the $LAT_{ideal}$ is determined, the ideal heat transfer can be calculated using the first law of thermodynamics for air systems.

$$\dot{Q}_{ideal} = \dot{m}_A \times c_{pA} \times (LAT_{ideal} - MAT)$$

This equation can be rewritten in IP units in terms of airflow rate in CFM, temperatures in °F, and heat transfer rate in Btu/h.

$$\dot{Q}_{ideal} = (\rho_{act}/\rho_{std}) \times 1.08 \times CFM \times (LAT_{ideal} - MAT)$$

Where,

CFM = the flow rate of the air through the coil in cubic feet per minute

The result of this equation is the ideal heat transfer for an infinitely large heat exchanger. To correct this, Ɛ must be assigned to the coil. To calculate the actual value, the ideal value is multiplied by Ɛ. This value can be determined from coil selection software using the actual inlet and outlet temperatures.

$$\dot{Q}_{actual} = \dot{Q}_{ideal} \times \varepsilon$$

Where,

Ɛ = the effectiveness of the heat exchange

This value should be compared to the heat required by the HVAC system and the available superheat, as determined from the HVAC and refrigeration calculations. These can be determined by methods described earlier in this playbook or with an energy model. The heat recovery calculation should never exceed the HVAC heat required or the available superheat.

## 4.7 Full Condensing for Space Heating

The refrigerant can be fully or partially condensed with an HVAC coil under the right conditions. The p-h diagram in Figure 4-6 shows that the heat available when the coil is allowed to fully condense refrigerant far exceeds that of superheat alone. A fully condensing coil would allow the refrigerant to travel from state point D to state point A. However, at times the result of the heat balance is somewhere between state points D and A. This is represented by state point X.

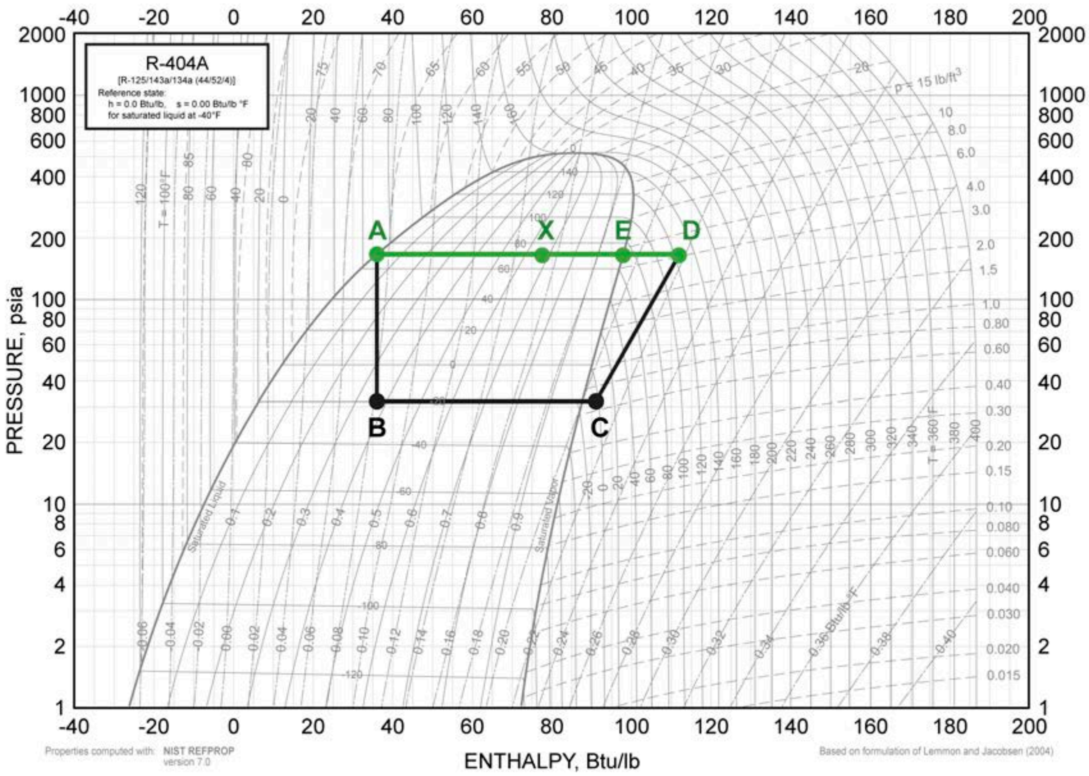


**Figure 4-6. R-404a p-h diagram (ASHRAE 2013)—condensing highlighted**

To calculate the heat reclaimed by a fully condensing coil, an energy balance must be performed to consider the properties of the refrigerant in each part of the heat rejection process. In the condensing part of the p-h diagram, also known as the two-phase region, the refrigerant rejects heat and the temperature does not change. The refrigerant is only changing phase. In this area, a temperature-based first law of thermodynamics equation is not useful; rather, an enthalpy-based equation must be used.

$$\dot{Q}_{Rcond} = \dot{m} \times \Delta h$$

| | | |
|---|---|---|
| $\dot{Q}_{Rcond}$ | = | the heat transfer required to condense the refrigerant, not including DSH |
| $\dot{m}$ | = | the mass flow of refrigerant |
| $\Delta h$ | = | the change in enthalpy between state points in the refrigeration circuit |

Using this equation in combination with the energy balance performed for the DSH process, an energy balance for the THR can be developed.

$$\dot{Q}_{air} = \dot{Q}_{RDSH} + \dot{Q}_{Rcond}$$

Where,

| | | |
|---|---|---|
| $\dot{Q}_{air}$ | = | the heat transfer of the airstream |
| $\dot{Q}_{RDSH}$ | = | the heat transfer required to desuperheat the refrigerant |

This equation is not easily solved without some assumptions about ideal temperature relationships. For estimation purposes, the LAT is assumed to approach the condensing temperature. As with the DSH assumptions, this is not necessarily true. With a large enough coil with many rows, the LAT could theoretically approach the refrigerant discharge temperature. However, this is a close approximation of typical conditions.

$$LAT_{ideal} = T_{cond}$$

Where,

| | | |
|---|---|---|
| $LAT_{ideal}$ | = | the ideal leaving air temperature |
| $T_{cond}$ | = | the condensing temperature |

Once a relationship between refrigerant temperature and LAT has been established, the energy balance equation can be rewritten.

$$\dot{m}_A \times c_{pA} \times (T_{cond} - MAT) = \dot{Q}_{Rcond(ideal)} + [\dot{m}_R \times c_{pR} \times (T_{ri} - T_{cond})]$$

Where,

| | | |
|---|---|---|
| $T_{Ri}$ | = | the refrigerant discharge temperature |
| $T_{Ro}$ | = | the refrigerant outlet temperature |
| MAT | = | the mixed air temperature entering the coil |
| $\dot{m}_R$ | = | the refrigerant mass flow |
| $\dot{m}_W$ | = | the water mass flow |
| $c_{pR}$ | = | the specific heat of the refrigerant |
| $c_{pW}$ | = | the specific heat of the water |

An energy balance has already been established for the DSH part of the refrigerant heat rejection, so the condensing heat transfer ($\dot{Q}_{Rcond}$) is the variable of interest. The designer can use simple algebra to quickly solve for the condensing part of the THR.

$$\dot{Q}_{Rcond(ideal)} = [\dot{m}_A \times c_{pA} \times (T_{cond} - MAT)] - [\dot{m}_R \times c_{pR} \times (T_{ri} - LAT_{ideal})]$$

Where,

| | | |
|---|---|---|
| $\dot{Q}_{Rcond(ideal)}$ | = | the ideal condenser heat rejection based on the ideal leaving air temperature |

This equation can be rewritten in IP units in terms of airflow rate in CFM, temperatures in °F, and heat transfer rate in Btu/h.

$$\dot{Q}_{Rcond(ideal)} = [(\rho_{act}/\rho_{std}) \times 1.08 \times CFM \times (T_{cond} - MAT)] - [\dot{m}_R \text{ x } c_{pR} \times (T_{ri} - LAT_{ideal})]$$

To correct this for nonideal conditions, Ɛ must be assigned to the coil. To calculate the actual value, the ideal value is multiplied by Ɛ. This value can be determined from coil selection software using the actual inlet and outlet temperatures.

$$\dot{Q}_{Rcond(actual)} = \dot{Q}_{Rcond(ideal)} \times \varepsilon$$

Where,

$\dot{Q}_{Rcond(actual)}$ = the actual condenser heat rejection based on the coil effectiveness

This equation assumes that the HVAC heat required is the limiting factor of the equation and solves for the condensing heat transfer that balances the equation. This equation is useful only for periods of the year when the HVAC heat required is the limiting factor. If the result of this equation exceeds the available THR, the system can fully condense the refrigerant and the available THR is the maximum amount of heat reclaimed by the HVAC system.

$$\dot{Q}_{reclaim} = \min\left(\dot{Q}_{Rcond(actual)}, THR_{available}\right)$$

Where,

$\dot{Q}_{reclaim}$ = the quantity of heat that is transferred as part of the reclaim process

$THR_{available}$ = the total amount of heat available for rejection from the refrigeration system

See Section 4.12 for guidance on calculating additional pump energy consumption, HVAC fan energy consumption, and refrigeration condenser fan energy savings.

## 4.8 Net Energy Savings From Heat Reclaim for Mixed Air Heating

Once all the factors described above have been calculated, the net energy savings can be calculated, taking account of all associated sources of energy savings and energy penalties:

$$\text{Net energy savings} = E_{reclaim} - E_{pump} - E_{fan} + E_{cond}$$

Where,

$E_{reclaim}$ = the quantity of heat energy that is transferred as part of the reclaim process

$E_{pump}$ = the amount of heat required to operate a heat reclaim pump. This quantity is zero if the transfer is direct from refrigerant to air.

$E_{fan}$ = the extra energy consumed by the system fan to overcome the extra pressure associated with the reclaim coil in the airstream

$E_{cond}$ = the energy saved by reducing the condenser fan power from the heat reclaim

When calculating the energy savings of a refrigerant-to-air heat exchanger, the reclaim coil is assumed to be configured such that no significant load is added to the compressor and no pump energy is required. If the design uses a refrigerant-to-water-to-air configuration, pump energy should be included in the net energy savings calculation.

## 4.9 Outdoor Air Preheat

Heat reclaim from a refrigeration system may be used to preheat outdoor air used for ventilation. This involves adding either a water coil or a refrigerant coil in the outdoor airstream. The advantage of this method is the relatively high temperature difference between the outdoor air and the condensing refrigerant. Figure 4-7 illustrates the refrigeration cycle where D is the compressor discharge point, E represents a point where the refrigerant is fully desuperheated, and A represents the point where all refrigerant is fully condensed, leaving the coil as a saturated liquid.

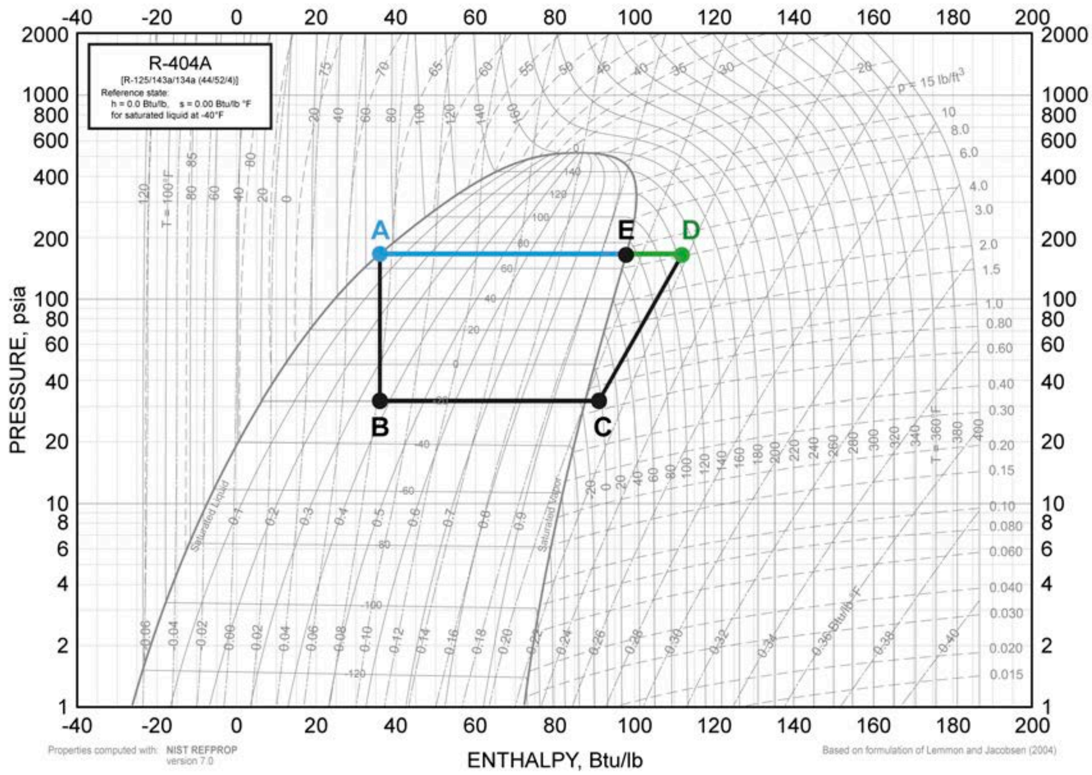


**Figure 4-7. R-404a p-h diagram (ASHRAE 2013)—DSH in green, condensing in blue**

Four options for heat reclaim for outdoor air preheat are addressed:

- DSH with a water coil
- DSH with a refrigerant coil
- Full condensing with a water coil
- Full condensing with a refrigerant coil.

Outdoor air preheat benefits a store only when heating is required. A typical supermarket building does not require heat at outdoor temperatures higher than 60°–75°F based on indoor temperature set points and internal loads. Buildings with large-capacity refrigeration systems may require heat in some localized areas at warmer temperatures, because the refrigerated cases may cool those areas. Warmer climates where the OAT is rarely below 65°F should not be considered for outdoor air preheat.

For calculation purposes, the maximum amount of heat recovered from the heat reclaim process can be determined by taking the lesser of the hourly airside equipment heat demand and the hourly heat available to be reclaimed from the refrigeration system. This calculation method assumes that if the refrigerant does not fully condense in the HVAC unit, it proceeds to the condenser to complete the condensing process.

An outdoor air preheat strategy may involve heating the outdoor air to the balance point temperature or implementing an outdoor air reset schedule that increases the supply OAT set point as the OAT drops. For this example, the designer presumably chooses a constant preheat OAT that is close to the store balance point and below the minimum condensing temperature to allow for heat transfer, without modifying refrigeration system set points. Heat required by the outdoor airstream using this strategy can easily be calculated using a derivation of the first law of thermodynamics, as follows:

$$\dot{Q} = \dot{m}_A \times c_{pA} \times \Delta T$$

Where,

| | | |
|---|---|---|
| $\dot{m}_A$ | = | the water mass flow |
| $c_{pA}$ | = | the specific heat of the water |
| $\Delta T_A$ | = | the temperature difference between the entering and leaving air |

This equation can be rewritten in IP units in terms of airflow rate in CFM, temperatures in °F, and heat transfer rate in Btu/h.

$$\dot{Q} = (\rho_{act}/\rho_{std}) \times 1.08 \times CFM_{OA} \times \Delta T$$

Where,

| | | |
|---|---|---|
| $\dot{Q}$ | = | the heat transfer in Btu/h |
| $CFM_{OA}$ | = | the outdoor airflow rate in cubic feet per minute |
| $\Delta T$ | = | the temperature difference in °F between the current OAT entering and leaving the heat reclaim coil |

The 1.08 constant contains necessary unit conversions and constants based on air density at sea level, so it is multiplied by an air density correction factor (ρact/ρstd) at elevations above sea level. These correction factors are shown in Table 4-1.

**Table 4-1. Density Correction Factors by Altitude Above Sea Level**

| Air Density by Altitude | |
|---|---|
| **Altitude (ft)** | **Density Correction** |
| 0 | 1.00 |
| 1,000 | 0.97 |
| 2,000 | 0.93 |
| 3,000 | 0.90 |
| 4,000 | 0.86 |
| 5,000 | 0.83 |
| 6,000 | 0.80 |
| 7,000 | 0.77 |

Once the heat required by the outdoor airstream is determined, it must be compared to the amount of heat available from the refrigeration system for DSH or condensing to determine the amount of heat reclaimed. (Refer to the refrigeration calculations earlier in this report for a description of how to determine these values.) The minimum of the heat required by the outdoor airstream and the heat available from the refrigeration system is the maximum heat that can be reclaimed.

$$\dot{Q}_{ideal} = \min(\dot{Q}_{air}, \dot{Q}_{ref})$$

Where,

$\dot{Q}_{ideal}$ = the ideal heat transfer between the airstream and refrigeration systems

$\dot{Q}_{air}$ = the heat transfer of the airstream

$\dot{Q}_{ref}$ = the heat transfer of the refrigerant

An actual air coil cannot achieve this ideal heat transfer, so the authors recommend that $\varepsilon$ be applied to the heat transfer. An $\varepsilon$ of 0.70 can be used as a starting point if the actual $\varepsilon$ is not known.

$$\dot{Q}_{actual} = \dot{Q}_{ideal} \times \varepsilon$$

$\dot{Q}_{actual}$ = the actual heat transfer between the airstream and refrigeration systems

$\varepsilon$ = the effectiveness of the heat exchange

This method does not accurately calculate the heat reclaimed if the design preheat temperature exceeds the minimum condensing temperature. If a designer is considering an outdoor air preheat temperature higher than the condensing temperature, he or she should use the same methodology as for space heating calculations.

See Section 4.12 for guidance on calculating additional pump energy consumption, HVAC fan energy consumption, and refrigeration condenser fan energy savings.

## 4.10 Net Energy Savings From Outdoor Air Preheat

Once all the factors have been calculated, the net energy savings can be calculated. This is the sum of all the factors calculated above as follows:

$$\text{Net Energy Savings} = E_{reclaim} - E_{pump} - E_{fan} + E_{cond}$$

Where,

$E_{reclaim}$ = the quantity of heat energy that is transferred as part of the reclaim process

$E_{pump}$ = the amount of heat required to operate a heat reclaim pump; this quantity is zero if the transfer is direct from refrigerant to air

$E_{fan}$ = the extra energy consumed by the system fan to overcome the extra pressure associated with the reclaim coil in the airstream

$E_{cond}$ = the energy saved by reducing the condenser fan power from the heat reclaim

The heat recovered by the outdoor airstream will be limited by the amount of heat required by the HVAC unit as the OAT approaches the discharge set point. As the difference between the OAT and the discharge air set point increases, the heat reclaimed is limited by the refrigeration system heat rejection. Figure 4-8 displays actual data from a 5,000-cfm air preheat coil installed near Denver, Colorado.

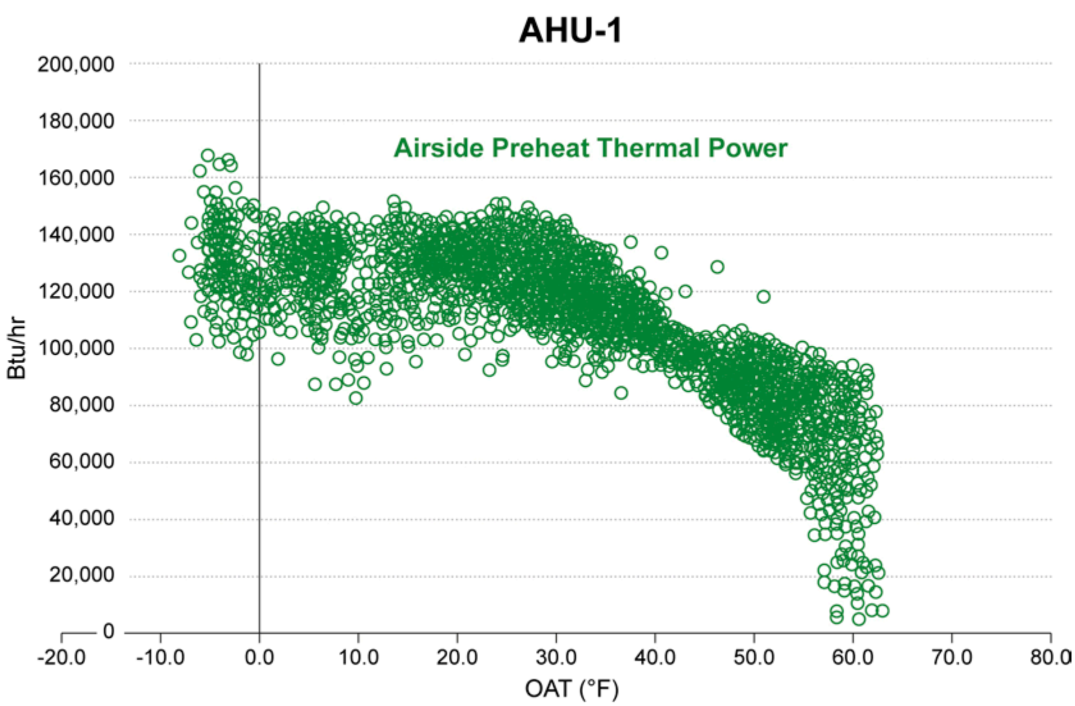


**Figure 4-8. Performance of an actual outdoor air preheat coil in the Denver metropolitan area**

The Outdoor Air Preheat spreadsheet was used to create a similar chart (see Figure 4-9).

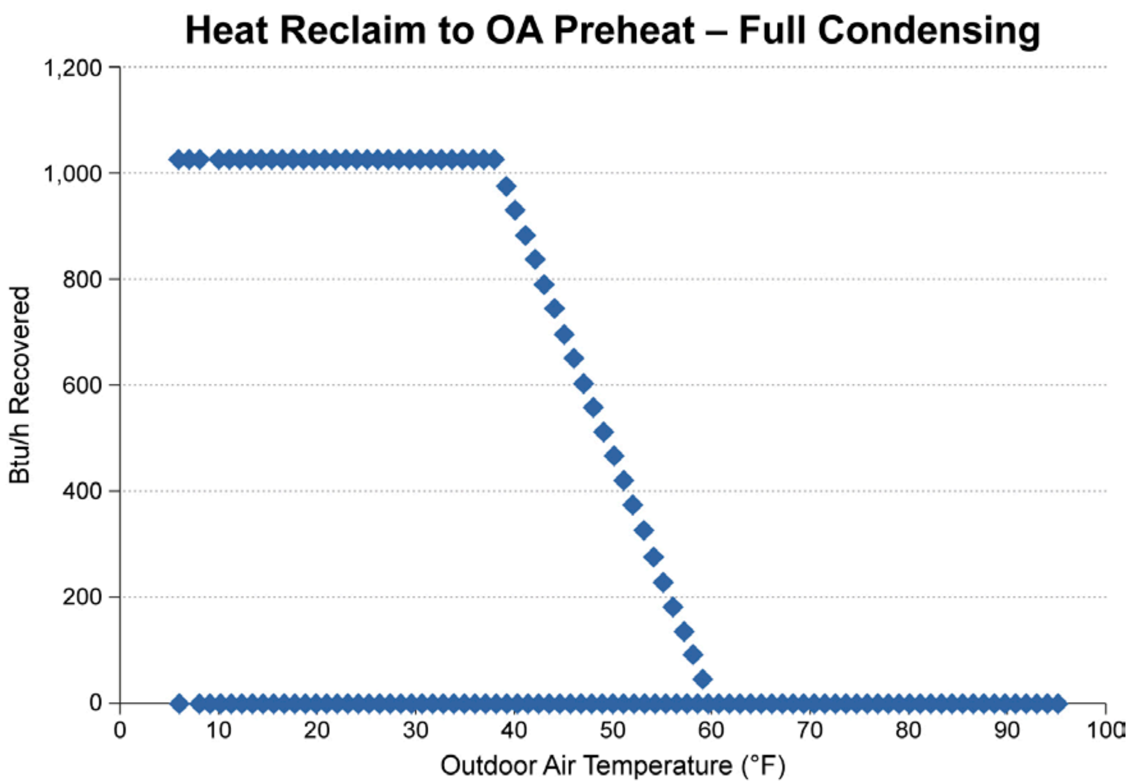


**Figure 4-9. Modeled performance of outdoor air preheat in the Denver area**

Although the trend is much cleaner in the simulated data, the inflection points are roughly the same in both charts. Both charts show the heat reclaimed starting to drop off around 30°F and reaching zero at 60°F, the preheat set point. The inflection point is where the limiting factor changes from the amount of heat available from the refrigeration system to the ability of the air preheat coil to deliver that heat.

Because of the large heating loads from refrigerated cases, the HVAC unit serving the grocery sales floor is usually the best choice for heat reclaim; however, in some situations several HVAC units that serve the sales floor or other areas would benefit from heat reclaim. These should be carefully evaluated to determine the best application for each project.

Determining the best refrigeration rack to use for heat reclaim is generally more complex than picking the right HVAC unit. To evaluate which rack should be used, each rack's potential for heat reclaim should be compared using the heat reclaim spreadsheets, considering DSH as well as full condensing based on design load, temperature settings, load profile, and any other aspects of the system that may significantly influence the available heat. Table 4-2 outlines the results of an example comparison for the baseline energy model, as outlined in Appendix C, for Denver, Colorado.

**Table 4-2. Modeled Annual Results for Outdoor Air Preheat in Denver, Colorado**

| | Heat Recovered (Condensing) | | | | |
|---|---|---|---|---|---|
| | **Rack A** | **Rack B** | **Rack C** | **Rack D** | **All 4 Racks** |
| Heat Recovered Condensing (kBtu) | 141,200 | 132,029 | 143,547 | 143,547 | 143,547 |
| New Rooftop Unit Heat (kBtu) | 2,437 | 11,518 | 0 | 0 | 0 |
| Savings (kBtu) | 141,200 | 132,029 | 143,547 | 143,547 | 143,547 |
| Savings ($) | $1,765 | $1,650 | $1,794 | $1,794 | $1,794 |
| | **Heat Recovered (DSH)** | | | | |
| DSH (kBtu) | 37,899 | 28,498 | 46,924 | 44,207 | 114,905 |
| New Rooftop Unit Heat (kBtu) | 105,648 | 115,049 | 96,623 | 99,340 | 28,642 |
| Savings (kBtu) | 37,899 | 28,498 | 46,924 | 44,207 | 114,905 |
| Savings ($) | $474 | $356 | $587 | $553 | $1,436 |

As expected, the analysis shows that full condensing results in significantly more energy savings than DSH alone. The energy model shows that rack C or D has enough capacity alone to meet the entire outdoor air preheat load for the year and result in the highest energy cost savings whereas DSH from all 4 racks together are not sufficient to meet the load, requiring supplemental gas consumption. The cost savings values shown in Table 4-2 were tabulated based on 80% efficient gas heat at $1/therm and ancillary electricity uses and savings at $0.10/kWh.

Once the annual cost savings have been determined, the cost of installing the heat reclaim system can be reviewed to determine the economic impact. (See Appendix C for guidelines on this type of analysis.)

## 4.11 Water-Source Heat Pumps

WSHPs are an attractive strategy for recovering waste heat, because they do not require a high grade of heat from the refrigeration system to heat the space effectively. They also generally have much higher cooling efficiencies than air-source direct expansion cooling systems.

The energy impacts of a heat pump are significantly more difficult to quantify than the other heat reclaim strategies outlined in this playbook because many system configurations and control sequences are possible. To define the energy savings, the designer must also decide which system to use to benchmark the performance of the WSHP HVAC system integrated with the refrigeration system: a packaged rooftop HVAC system with no refrigeration integration or a WSHP system with no refrigeration integration.

To understand the energy impacts of combining refrigeration and HVAC loads into a single heat pump loop, the designer must first understand the impact of loop temperature on refrigeration and heat pump efficiency. As the temperature of the heat pump loop drops, the refrigeration efficiency rises and the efficiency of heat pumps in heating mode drops. Rising heat pump loop temperatures has the opposite effect on the refrigeration and heat pump efficiencies. Heat pumps in cooling mode follow roughly the same trend as the refrigeration systems. The rate at which these efficiencies change relative to heat pump loop temperature and the loads of each system on the loop determines the optimum control strategy.

The performance of a refrigeration system that is integrated with an HVAC heat pump loop is best quantified with a robust energy simulation software. However, this performance can also be estimated using spreadsheet calculations.

To quantify the energy impact of this system type, determining the loop temperature is critical; however, the loop temperature depends on system loads. This creates a circular argument, so a beginning loop temperature must be assumed. This temperature is somewhat arbitrary, but should be within the controlled range. The loop temperature change is a function of the heat rejected by the refrigeration system and the heat added or rejected by the heat pumps.

$$\Delta T_{loop} = f(Q_{HP}, THR)$$

Where,

| | | |
|---|---|---|
| $\Delta T_{loop}$ | = | the temperature change in the heat pump loop |
| $Q_{HP}$ | = | the heat added or removed from the loop by the HVAC system heat pumps |
| THR | = | the total heat of rejection, the heat added to the loop by the refrigeration system |

This equation can be further defined in terms of the first law of thermodynamics as:

$$\Delta T_{loop} = (Q_{HP} + THR) / \dot{m}_{HP} \times c_p$$

When using IP units, this can be simplified in terms of GPM instead of mass flow:

$$\Delta T_{loop} = (Q_{HP} + Q_{ref}) / 500 \times GPM \times (\text{heat capacity correction})$$

Where,

| | | |
|---|---|---|
| $\dot{m}_{HP}$ | = | the mass flow of the heat pump loop |
| $c_p$ | = | the specific heat of the heat pump loop fluid |
| gpm | = | the flow rate of the heat pump loop |

Heat capacity correction. If glycol is used in the heat pump loop, a correction factor should be added. (See Figure 4-13 for correction factors.)

The heat transfer to the loop from the refrigeration systems is the sum of the compressor power input and the evaporator load with suction superheat. The evaporator capacity does not depend on the condensing method and can be obtained from the Refrigeration Front End spreadsheet.

$$THR = Q_{comp} + Q_{evap}$$

Where,

| | | |
|---|---|---|
| $Q_{comp}$ | = | the compressor power input to the system |
| $Q_{evap}$ | = | the refrigeration evaporator load, including suction superheat |

The compressor power input is a function of the part-load ratio (PLR) and the condensing temperature. The PLR is obtained from the Refrigeration Front End spreadsheet. The condensing temperature can be calculated based on the previous hour's loop temperature. Typically, the condensing temperature is the loop temperature plus a TD. If this differential is unknown, 10°F can be used as a starting point.

$$Q_{comp} = f(PLR, T_{cond})$$

Where,

| | | |
|---|---|---|
| PLR | = | the compressor part-load ratio |
| $T_{cond}$ | = | the refrigeration condensing temperature |

The effects of varying PLR and condensing temperatures on the compressor power depend on the model of compressor being used. If this is unknown, typical values can be obtained from energy simulation software programs such as eQUEST or EnergyPlus. For this example, eQUEST curves are used to illustrate the procedure.

$$Q_{comp(Tcond)} = 1.364 - 0.0264 \times T_{cond} + 0.00023 \times T_{cond}^2 \text{ (low temperature)}$$

$$Q_{comp(Tcond)} = 0.938 - 0.0125 \times T_{cond} + 0.00013 \times T_{cond}^2 \text{ (medium temperature)}$$

$$Q_{comp(PLR)} = 0.0383 + 1.0778 \times PLR - 0.1161 \times PLR^2$$

Where,

| | | |
|---|---|---|
| $Q_{comp(Tcond)}$ | = | the compressor power multiplier dependent on condensing temperature |
| $Q_{comp(PLR)}$ | = | the compressor power multiplier dependent on PLR |

These two modifier values are multiplied by the compressor power as defined in the refrigeration front end. If a compressor has not been selected, the design compressor power may need to be calculated based on evaporator load and rated compressor energy efficiency ratio (EER).

$$Q_{comp,design} = Q_{evap} / EER$$

Where,

| | | |
|---|---|---|
| EER | = | the energy efficiency ratio defined as evaporator capacity divided by compressor wattage |
| $Q_{comp,design}$ | = | the design compressor power |

The heat transferred between heat pumps and the heat pump loop is a function of space load and heat pump efficiency. The calculation of this value is different in heating mode versus cooling mode. In heating mode, the heat from the heat pump compressor and condenser are added to the space and the evaporator also adds heat from the water loop. In cooling mode, the reverse is true: the evaporator removes heat from the space while the compressor heat and condenser heat from the heat pump are added to the loop.

$$Q_{HPh} = Q_{evap}$$

$$Q_{HPc} = Q_{cond} + Q_{comp}$$

Where,

| | | |
|---|---|---|
| $Q_{HPh}$ | = | the heat removed from heat pump loop by heat pumps in heating mode |
| $Q_{HPc}$ | = | the heat added to heat pump loop by heat pumps in cooling mode |
| $Q_{evap}$ | = | the heat absorbed by the heat pump evaporator |
| $Q_{cond}$ | = | the heat rejected by the heat pump condenser |
| $Q_{comp}$ | = | the heat added to the system by the heat pump compressor |

The heat added or removed by the heat pumps from the heat pump loop depends on the wb temperature into the heat pump, the EWT, and the PLR. The wb temperature is obtained from the weather file. The water temperature depends on the refrigeration and heat pump loads on the loop. This creates another circular argument, so a beginning water temperature should be assumed within the control range.

$$Q_{HP} = f(T_{wb}, EWT, PLR)$$

Where,

| | | |
|---|---|---|
| $Q_{HP}$ | = | the heat added or removed from the heat pump loop by the heat pumps |
| $T_{wb}$ | = | the wb temperature at the air coil, assumed to be the outdoor air wb from the weather data |
| EWT | = | the entering water temperature, the source water temperature to the heat pump |
| PLR | = | Part-load ratio, the fraction of hourly load to maximum load |

Heat pump performance varies significantly between manufacturers based on the conditions described above. Manufacturers publish data that can help a designer understand the performance characteristics. However, these data are typically broken down into two equations: (1) a multiplier for performance based on PLR only; and (2) wb temperature and EWT. Developing a curve fit for a two-variable equation is difficult, so the authors recommend that typical performance curves from EnergyPlus or eQUEST be used to describe heat pump performance. Sample equations from eQUEST follow.

$$EIR_{cool} = 0.499 - 0.008 \times T_{wb} + 0.013 \times EWT$$

$$EIR_{heat} = 0.646 + 0.008 \times T_{wb} - 0.006 \times EWT$$

$$EIR = 0.010 + 1.080 \times PLR - 0.105 \times PLR^2 + 0.015 \times PLR^3$$

Where,

EIR = The energy input ratio, the power consumed divided by the usable capacity for heating and cooling, respectively. These values are actually multipliers for the EIR.

Once the EIR fractions are determined, they can be multiplied by the EIR at rated conditions and the space load at each hour to determine the amount of heat added or rejected to the heat pump loop. Space loads can be determined by the Space Heating spreadsheet.

Once the refrigeration and heat pump effects on the loop have been calculated, the loop temperature difference can be determined. The loop temperature difference is then added to the previous hour's temperature. To simplify the calculations, the authors suggest determining the loop flow rate based on a constant temperature difference at design conditions. This determines the peak flow rate of the system. If variable-speed pumps are used, this flow rate can then be scaled back based on the PLRs of the refrigeration and heat pump systems. Although each system is sized for a constant TD, the loop TD is not constant, because heat pumps fluctuate between heating and cooling. In heating mode, the loop TD is somewhat lower than the individual system design TD.

Because the heat pump and refrigeration systems are not perfectly balanced for much of the year, heat needs to be added or removed from the heat pump loop. This can be done in many ways, but is typically done with a boiler and cooling tower. To fully understand the energy impact of this system, the energy consumption of the heat addition and rejection devices must be quantified.

The heat addition and rejection devices are possibly the most difficult aspects of this system to quantify, because their energy consumption depends on the control strategy used. A traditional heat pump loop control strategy maintains a loop temperature between upper and lower temperatures (typically 60°–90°F) and lets the temperature float inside that range. With refrigeration systems tied to the loop, driving the loop temperature down when possible may be beneficial. This sequence needs case-by-case evaluation.

An uncontrolled heat pump loop temperature that is calculated from the refrigeration and heat pump heat balance rises or falls outside reasonable values. Thus, a second loop temperature must be calculated. To maintain the loop temperature within a range, the new loop temperature must be the same as the original temperature (if it is within the range). If it is not within the range, the temperature is either the maximum or the minimum limit. The TD between the original loop temperature and the second loop temperature can be used in combination with the hourly loop flow to determine the amount of heat that must be added or removed from the loop.

Once the amount of heat required to be added or rejected from the loop is determined, the efficiencies of the heat addition and rejection devices can be used to quantify the energy consumed by this process. Typical curves may be used to approximate these values.

The preceding calculation methods provide the energy performance of the system, but the total energy is of little use without a baseline for comparison. If the baseline of interest is a WSHP system with a separate refrigeration system, the refrigeration information can be removed from the heat pump calculations.

If the baseline of interest is a packaged rooftop design, the comparison is more difficult. The rooftop unit heating efficiency calculation is fairly straightforward, because furnace efficiency varies only slightly. The cooling energy consumption depends on outdoor air conditions, indoor conditions, and PLR. These could be defined with typical performance curves, as previously discussed. In addition to these factors, the pumping power associated with the WSHP system would need to be quantified and added to the heat pump system energy calculations. Pump calculations are addressed in Section 4.12.

## 4.12 Other Considerations

Once the amount of heat recovered by a given heat reclaim method has been determined, several other contributions to building energy consumption need to be considered, including:

- **Circulating pump power.** A pump is required when a water loop is used to recover heat. This power is used only when the reclaim system is in operation.
- **Heat reclaim coil airside pressure drop.** Adding a coil to the airstream causes a pressure drop and consumes additional fan power. This pressure drop occurs even when the reclaim system is not in operation.
- **Heat reclaim coil refrigerant side pressure drop.** Depending on the system, adding a coil to the refrigerant flow path may require higher head pressures, leaving the compressor to maintain the required pressure for proper condensing. This is not usually an issue, because refrigeration systems typically operate at minimum condensing pressures during times when outside air preheat would be beneficial. However, it may require consideration.
- **Condenser fan power savings.** When the refrigerant condenser system does not need to be used to reject heat to the outdoor airstream, condenser fans may be shut off or run at a lower speed.

### *4.12.1 Circulating Pump Power*

For energy estimation purposes, the only difference between a refrigerant-to-water-to-air heat exchange and a refrigerant-to-air heat exchange is assumed to be the pumping power. Circulating pump power is considered in a water loop system only. The circulating pump operates whenever heat is required to warm the outdoor airstream. A control valve at the air coil is used to modulate the flow through the coil.

The only way flow can be adjusted with a constant-speed pump is to adjust the pressure on the pump. With a constant-speed pump, the flow is reduced when system pressure increases; however, the power is not reduced significantly. This is commonly referred to as *riding the pump curve*. Figure 4-10 shows the pump curve and the system curve. The system curve reflects the effects of the piping systems that the pump serves as the flow requirements change. Pump selection involves choosing a pump that delivers the desired flow rate at a given system pressure as determined by all the resistance to flow through the system.

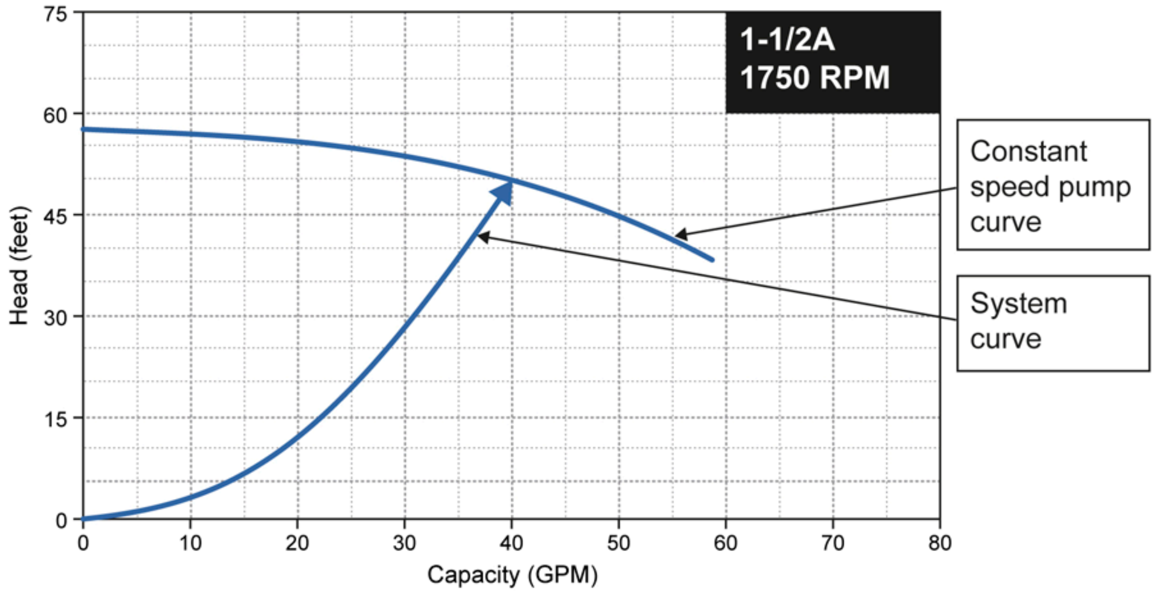


**Figure 4-10. Sample constant-volume pump curve**

A variable-speed pump is generally designed to maintain system pressure by adjusting pump speed. Such a pump lowers energy consumption more than does a constant-speed pump. Figure 4-11 shows an example of a variable-speed pump selection. In this case, the pump speed can be adjusted to follow the system curve while it delivers the right amount of heat to match a varying load.

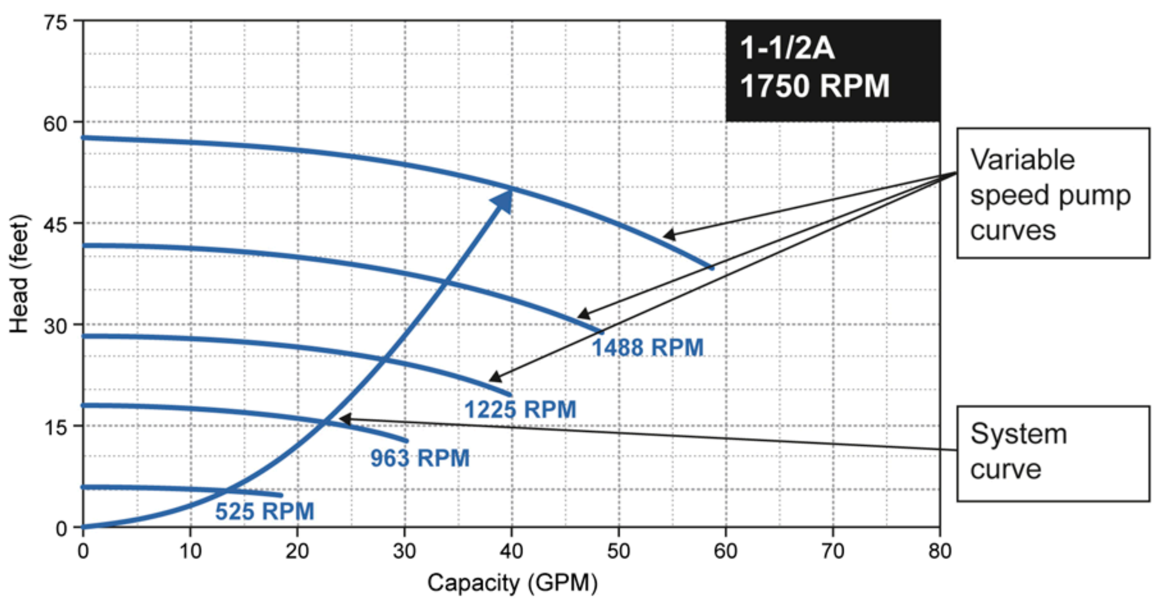


**Figure 4-11. Sample variable-volume pump curve**

Each pump speed has its own pump curve. Although a variable-speed pump does not exactly follow the system curve (because adjusting the pump speed to maintain system pressure takes time), it comes much closer than a constant-speed pump. From the constant-speed curve, the designer should note the horsepower reduction associated with pump speed reduction. For example, when the system flow drops to 30 gpm, the variable-speed pump needs less than ½ horsepower to operate; the constant speed pump requires almost 1 horsepower.

If the system pump, coil, flow, and fluid properties have been designed, these values should be used to calculate energy consumption by the circulating pump. If system components have not been designed, the pumping power can be estimated based on rules of thumb, assuming 10°F temperature difference, 10 ft of head loss through the condenser, 10 ft of head loss through the reclaim coil, and 6 ft through the piping, figuring 2 ft loss/100 ft of pipe and 300 ft of pipe. These values can be used as inputs for the pump selection software that outputs pump motor horsepower. The GPM value is determined based on a derivation of the first law of thermodynamics as follows:

$$\dot{Q} = 500 \times GPM \times \Delta T \times (\text{heat capacity correction})$$

or

$$GPM = \dot{Q} / [(500 \times c_p \text{ x } \Delta T) \times (\text{heat capacity correction})]$$

In this formula, $\dot{Q}$ is the heat transfer calculated from the airside heat recovery calculations and $\Delta T$ is the temperature difference between the EWT and LWT. The heat capacity correction is for systems with glycol. The more glycol is in a fluid, the lower the heat capacity. These correction factors can be determined from the manufacturer at design temperature. Table 4-3 represents typical correction factors for propylene glycol. It is based on the Engineering Toolbox.

**Table 4-3. Heat Capacity Correction Factor by Percent Propylene Glycol**

| Propylene Glycol/Water Derate | |
|---|---|
| **% Propylene Glycol** | **Heat Capacity Correction** |
| 0% | 1.00 |
| 10% | 0.98 |
| 20% | 0.96 |
| 30% | 0.94 |
| 40% | 0.90 |
| 50% | 0.85 |
| 60% | 0.81 |

Once the flow rate has been determined, one of two methods can be used to determine the peak pumping power:

- Pump selection software, which is readily available from the major manufacturers, can be used to specify an inline recirculating pump. The software provides a motor horsepower selection corresponding to the required flow rate. The pumping power estimate can then be converted to Watts and used to calculate the hourly energy consumption.

- Quick, back-of-the-envelope calculations. ASHRAE 90.1 Appendix G is a guideline for developing a baseline energy model. In section G3.1.3.5, the guideline specifies a baseline hot water pump to consume 19 W/gpm.

With a constant-volume pump, the pumping power is equivalent to the pump motor power for each hour that heating is required. If a variable frequency drive is used with the pump, the power consumed is proportional to the cube of the speed as shown below.

$$P_2 = P_1 \, (PLR)^3$$

In this equation, PLR is the ratio of the hourly capacity to the peak capacity of the heating coil. A variable frequency drive in a pumping system costs more than a constant-speed pump. The variable frequency drive adds some cost; however, with the widespread use of electronically commutated motors, the cost premium is relatively low. In addition to the initial cost increase of the pump, the pump controller must be more sophisticated and needs an input from a differential pressure sensor downstream of the pump. Consult a pump manufacturer about the cost of a variable-speed pump compared with a constant-volume pump.

The inefficiencies associated with a pump result in heat output. In the instance of heat reclaim, heat generated by the motor is rejected to the surrounding space. Heat generated from impeller inefficiencies is rejected to the fluid. In both cases, these values are considered to be negligible and are not included in the heat recovery calculations.

### 4.12.2 Additional Fan Power

The heat reclaim coil in the airstream increases energy consumption. If equipment has been designed, the actual equipment ratings such as pressure drop and fan power should be used to calculate energy consumption. If the system has not been designed, assumptions must be made to estimate the additional fan power. As a starting point, assume 60% fan efficiency and 0.25 in. of static pressure drop across the preheat coil. The design outdoor airflow is also required to calculate the energy consumption.

$$AHP = CFM \times \Delta P / 6356$$

Where,

| | | |
|---|---|---|
| AHP | = | air horsepower, the power delivered to the airstream in horsepower |
| $\Delta P$ | = | the pressure drop discussed above |
| CFM | = | the total system airflow of the unit being evaluated |

The pressure drop of the coil may directly affect the outdoor airstream only, depending on the heat reclaim method; however, an added pressure drop in the outdoor airstream reduces the flow of outdoor air. Fan speed needs to bring the outdoor airflow back up to design levels, which also increases the return airflow. To balance these flows, the return air damper needs to be partially closed, resulting in pressure drops in the return airstream and in the outdoor airstream. Thus, the total system airflow is used rather than the outdoor airflow in each case. Once the air horsepower has been calculated, the brake horsepower and total horsepower can be determined using the following equations:

$$BHP = AHP / \text{fan efficiency}$$

$$P_{HP} = BHP / \text{motor efficiency}$$

In these equations, BHP is the brake horsepower, which is the power required to drive the fan. Sixty percent is a good starting point for fan efficiency, but actual values should be used if available. Motor efficiency can be determined from ASHRAE 90.1 motor efficiency tables. The location of the building determines which version of ASHRAE 90.1 is appropriate, because these requirements vary by jurisdiction. This table is located in Table 10.8A in ASHRAE 90.1-2010. Totally enclosed fan-cooled motors at 1800 rpm is a good starting point. The result of these calculations is the total fan power input ($P_{HP}$) in horsepower. This can be converted to Watts for each hour and summed to calculate the additional annual power consumed from fan power through the preheat coil.

The fans are located in the airstream of the HVAC unit, so the additional fan energy consumed as a result of the preheat coil is added to the airstream as heat. A typical coil does not add significant fan heat to the airstream in this manner and can be neglected in the heat recovery calculations. The additional fan power is expected to add less than 0.5°F to the airstream, downstream of the outdoor air preheat part of the unit.

### 4.12.3 Condenser Fan Power

The next component that needs to be calculated is the condenser fan power savings. For calculation purposes, the designer should assume that the condenser fan speed is proportional to the fraction of condenser capacity used each hour. To determine these values, the hourly condenser heat rejection and hourly heat reclaim are required. The hourly condenser heat rejection can be determined from the energy modeling outputs. The hourly heat reclaim can be calculated using the procedures described above. Once these values have been determined, the hourly fan speed fraction can be determined:

$$\text{Fan speed ratio} = \text{Heat rejection ratio} = (\text{condenser heat rejection} - \text{heat reclaim}) / \text{condenser heat rejection}$$

If the condenser fans have single-speed control, the new hourly fan energy is linearly related to the fan part load, because the fan is cycling on and off to meet the load. The equation for this follows:

$$\text{Part load fan power} = \text{Total condenser fan power} \times (\text{fan speed ratio})$$

If variable-speed fans are used and the condenser fans have minimal downstream pressure drop, the fan similarity laws can be used to calculate part load fan power as follows:

$$\text{Part load fan power} = \text{Total condenser fan power} \times (\text{fan speed ratio})^3$$

## 4.13 Resources


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# Chapter 5. Implementing Heat Reclaim Methods

## 5.1 Introduction

Once a heat reclaim strategy has been identified for potential implementation, several practical issues must be addressed. The first and most basic is cost. Energy savings must be weighed against initial costs, operating costs, and future costs. A quality design also includes plans for operations and maintenance. It may also include measurement and verification strategies. This chapter does not include a comprehensive list of practical concerns or a set of instructions to design a system. Rather, it outlines some high-level considerations for implementing and operating heat reclaim systems.

## 5.2 Financial Analysis

Several methods are available to evaluate the economic impacts of equipment investments. A method is often selected based on company standards or familiarity. The type of system, complexity of costs and benefits, and investor requirements all dictate which method is selected. Of the numerous methods available, simple payback, net present value, and internal rate of return are discussed in this playbook. See the entries in the Resources section for more in-depth discussions of these and other methods.

Any financial analysis requires an understanding of the costs and savings of implementing a heat reclaim system. This includes the cost of the equipment, the incremental costs of installation, energy savings, and incremental maintenance or operational savings or costs. Calculation methods for these costs and savings are described in Chapter 4. Actual values for costs or savings should be used when available to improve the accuracy of the results.

Several tools are available to assist in life cycle cost analysis. These include BLCC5 (a program maintained by the U.S. Department of Energy) and the "User Friendly" Building Life-Cycle Costing spreadsheet, an adaptation of the BLCC5 program in spreadsheet format. More information about these tools can be found in the Resources section at the end of this chapter.

### *5.2.1 Simple Payback*

The most basic financial analysis method typically used is commonly referred to as *simple payback*. As the name indicates, the method calculates the payback of a measure in a simple manner. The equipment and installation costs of the measure are divided by the yearly savings to produce a payback, usually reported in years. For instance, if implementing a heat reclaim system costs $10,000 and the system is expected to save $2,000 per year, the system will "pay back" in 5 years. This method is popular because measures can be quickly and easily compared with simple calculations and minimal information. It is also effective for communicating financial comparisons in a manner that is easy for wide audiences to understand. Its primary drawback is that it does not account for the time value of money, which is particularly important for measures that are expected to be in operation for a long time.

$$\text{Simple Payback(years)} = \frac{\text{First Cost}}{\text{Yearly Savings}}$$

### 5.2.2 Net Present Value

As the name indicates, this method takes into account the time value of money, using compound interest rates to determine the present value of each periodic cost or savings. These individual present values are then added together to determine the net present value.

$$\text{Net Present Value} = \frac{\text{Year 1 Saving}}{(1 + \text{Rate})^1} + \frac{\text{Year 2 Saving}}{(1 + \text{Rate})^2} + \cdots + \frac{\text{Year X Saving}}{(1 + \text{Rate})^\text{X}} - \text{Initial Cost}$$

The discount rate is typically set at a value consistent with a company's cost of capital or returns available from alternative investment of resources. At this rate, a positive net present value can be considered a good investment; a negative net present value is a poor investment. Savings that are made many years after the initial investment become less and less significant, especially with a high return rate.

### 5.2.3 Internal Rate of Return

Internal rate of return refers to the return rate for an investment at which the net present value is zero. This value can be used when no required investment return rate is known. Unfortunately, the structure of the formula does not allow a direct calculation of rate. The easiest way to calculate the internal rate of return is to use an iterative approach, starting with an estimate and calculating the net present value based on that estimate. If the net present value is positive, the rate estimate is too low; if the net present value is negative, the rate estimate is too high. Based on the outcome of net present value, the rate may be adjusted and recalculated until the error is small enough to be deemed insignificant. A project with an internal rate of return exceeding the company's cost of capital is generally considered worthy of investment.

## 5.3 Refrigeration Systems

Refrigeration systems require several design considerations to account for variations introduced by heat reclaim systems. One critical design consideration is refrigerant charge variations. Any refrigeration system requires the correct mass of refrigerant (commonly referred to as *charge*) to operate properly, ensuring the correct balance of liquid and gas in each component. The charge never leaves the overall system unless a leak occurs, but the balance of liquid and gas shifts in each component as operating conditions change. For example, many systems control condenser capacity by allowing a variable part of the condenser volume to be flooded with liquid refrigerant. This condenser capacity control may be required because the system load decreases or ambient conditions increase the condenser effectiveness. Operation of a heat reclaim system further reduces the required condenser capacity in the winter, which increases the liquid in the condenser.

This increase, in addition to the charge required to appropriately fill the reclaim components, must be accounted for when the size of the receiver and the associated refrigerant charge are determined. If the receiver is too small and the system is undercharged, low load conditions may cause enough refrigerant to collect in the condenser and reclaim components to allow refrigerant gas instead of liquid to be fed to evaporators. If the same system is overcharged, liquid may fill the receiver and back up into the condenser, which reduces capacity under high load conditions. A system with a properly sized receiver accounts for the variations in charge location and contains an appropriate amount of liquid and gas with any system operating condition, but must be appropriately charged with refrigerant to do so.

To achieve optimal thermodynamic efficiency of a refrigeration system, the condensing pressure and associated temperature must be driven as low as possible. This goal conflicts with optimal operation of a heat reclaim system, because the heat transfer in this system is directly proportional to the approach temperature. At optimal condensing temperature, the refrigerant temperature may not reach that of the fluid intended to absorb the reclaimed heat, reducing or eliminating heat transfer. To account for this, the refrigeration system may be controlled to raise the condensing temperature set point targets when the reclaim system is operating. This strategy must account for the incremental energy required to raise the condensing temperature to evaluate if this strategy results in a net energy cost savings.

A reclaim system expected to cycle off should be equipped with a means for removing most of the refrigerant from the system, typically referred to as a *pumpout*. If the refrigerant is not pumped out of the reclaim coil, its temperature drops to equalize with the surrounding fluid. When the reclaim system is restarted, hot discharge gas causes rapid boiling in the reclaim coil, leading to rapid pressure buildup. This condition may lead to liquid hammer and associated failure of system components. To prevent this, a connection is commonly made between the reclaim part of the system volume and the suction side of the compressor. This pumpout line is closed when the reclaim system is operating. It is then opened when the reclaim system turns off, preventing liquid refrigerant from remaining in the reclaim system. The pumpout line size should be kept small, or a restriction should be included to prevent liquid refrigerant from entering the compressor.

If a reclaim system is designed for partial or full condensing, it must be laid out to allow liquid to flow freely out of the reclaim condenser. For series configurations, the reclaim condenser outlet must be located vertically above the condenser inlet. For parallel configurations, the reclaim condenser outlet must be located above the receiver. If this height restriction is not followed, liquid does not flow out of the reclaim condenser at an acceptable rate. As a result, the condenser may flood, reducing capacity, causing refrigerant charge issues, and liquid may flood back to the compressors. This restriction does not apply to DSH reclaim systems; however, line sizing and layout must account for entrained oil to be carried through the system under any load condition.

## 5.4 Service Hot Water Systems

Heat is typically reclaimed for SHW systems with two tanks, one for reclaim and one that includes the primary heat source. This system allows the heat reclaim tank temperature to rise until it reaches the SHW temperature set point. As the temperature of the tank rises, the heat that can be transferred from the refrigerant is reduced. In some instances a heat exchanger between the entering cold water and the refrigerant might be a better alternative. The temperature difference is always maximized in this configuration. Its drawbacks are its high initial cost and its inability to transfer heat without hot water demand. A tank configuration allows heat to be transferred from the refrigeration system to the tank at all times, even without domestic hot water demand.

## 5.5 Airside Heating, Ventilation, and Air-Conditioning Systems

Calculating energy savings from heat reclaim for space heating is a good place to start when evaluating the practicality of implementing this type of system, but how the heat recovery equipment functions in practice must be considered. This type of heat reclaim strategy is easiest to implement in a new construction, but it can also be implemented in an existing building.

The heat reclaim coil should be placed downstream of the cooling coil to take advantage of the ability to provide reheat during dehumidification. In this scenario, a warm and humid return and outdoor air mixture is cooled to a temperature lower than comfortable supply temperatures to remove moisture. This cold, dry air is then reheated to the design supply temperature. Much of the time, HVAC units performing dehumidification include a direct expansion coil with a DSH coil downstream for reheat. In this scenario, the waste heat from the HVAC system cooling process (as opposed to the refrigeration system) consumes little additional energy. If this is the case, the available savings from using the refrigeration system waste heat for dehumidification reheat may be minimal. Depending on the sequence of operations, the HVAC unit may at times simultaneously heat and cool, which is generally not permitted by energy code. The exception is with systems serving zones that have specific humidity requirements. It generally applies to supermarket sales areas where excessive humidity significantly increases refrigeration energy.

Ideally, the heat reclaim coil should be placed upstream of a system's primary heating coil and used as the first stage of heat. If the coil is installed in this manner, the two heating coils could be controlled to turn on in stages based on system supply temperature or space temperature. A sequence could be set up to provide a certain supply air temperature during cooling, ventilation, and heating modes. If the temperature downstream of the heat reclaim coil is lower than the desired supply air temperature, the primary heating coil is activated to provide the remainder of the heat required.

## 5.6 Water-Source Heat Pumps

Integrating refrigeration systems with WSHPs can be an effective strategy for reclaiming waste heat under the right conditions, but several design aspects must be considered. This type of system has a much higher first cost than most typical supermarket systems and may require more maintenance. The associated energy savings must be weighed against the cost impacts.

Integrating refrigeration systems with WSHPs is most efficient when the heat pumps are in heating mode and the refrigeration system is rejecting heat to the loop. This allows the two systems to "share" heat, eliminating the need for condenser fans in either system. When heat pumps operate in cooling mode, no benefit accrues to including both systems on a single loop. Thus, integrating refrigeration and heat pump systems should only be considered in heating-dominated climates or in heating-dominated microclimates within a building.

Proper control depends on the size of each load on the system. The designer needs to understand the balance of heat pump capacity with the refrigeration THR to optimize the performance of the project. The load balance, combined with the efficiency curves associated with the refrigeration and heat pump systems, helps determine the control strategy. For instance, the refrigeration system operates most efficiently at low condensing temperatures and heat pumps in heating mode operate most efficiently with a higher heat pump loop temperature. If lowering the condenser water temperature drastically improves the refrigeration efficiency but has little effect on the heat pump efficiency, controlling the loop to a lower temperature may be beneficial. In some cases, installing a fluid cooler for the refrigeration systems and a boiler to maintain heat pump loop temperature may be beneficial.

The heat pump loop heat addition and rejection systems influence the efficiency of the total system. A boiler and fluid cooler configuration is controlled much differently than a ground source heat exchanger.

In general, reliable refrigeration system performance is more critical than building HVAC system operation. When designing a supermarket, providing some redundancy in the refrigeration circuit is often critical. This might involve two fluid coolers on the heat pump loop and a sequence that shuts off heat pump cooling if a cooling tower is out of service.

Heat pump integration with refrigeration systems may be the most difficult of the methods described to successfully implement, but under the right conditions can be an effective solution.

## 5.7 Operations and Maintenance

Although recovering heat from a system has several advantages, refrigerant management and additional system complexity issues must be considered.

Heat reclaim systems typically involve at least 100 ft of additional refrigerant piping. Whether the system is intended for DSH or full condensing, every joint, valve, and Schrader port is a potential location for a leak. With the increasingly stringent environmental regulations and the higher costs of refrigerant and hydrochlorofluorocarbon phase-out, accounting for every pound of refrigerant that goes into a system is more important than ever.

Table 5-1 shows operations and maintenance checks for each heat reclaim system noted in the report.

**Table 5-1. Maintenance Considerations**

| | 3-Way Valves Seating Properly | Remove Scale From Heat Exchanger | Verify Pumpout Operation | Check Refrigerant Level |
|---|---|---|---|---|
| **DSH** | | | | |
| Domestic Water Preheat | * | * | * | * |
| Mixed Air Heating | * | | * | * |
| Outdoor Air Preheat | * | * | * | * |
| **Condensing** | | | | |
| Mixed Air Heating | * | | * | * |
| Outdoor Air Preheat | * | | * | * |

A series of valves is required to switch the system between operation under normal conditions and heat reclaim mode as shown in system schematics. The additional valves found in a heat reclaim system include a three-way valve to divert the hot gas to the heat exchanger and a pumpout solenoid to remove excess refrigerant from the heat reclaim circuit when it is not in use. Other heat reclaim applications have different valve types. Additional wiring and programming are required for the heat reclaim system to work as intended. The programmer and technicians must understand the added complexity of the system to detect and diagnose faults before alarms are triggered; this decreases product loss, equipment damage, and costly service calls. Adding a few extra steps to the refrigeration maintenance program keeps the system operating as intended.

## 5.8 Measurement and Verification

Understanding how the system operates compared with how it is intended to operate is crucial, because the high side of the refrigeration system is the largest contributor to energy consumption, longevity, and proper operation. Reviewing historical system operation data provides the information necessary to tune and adjust the system parameters to optimize system efficiency. This review requires measurement and logging of key data points.

Depending on the type of heat reclaim system and control strategy, sensor types and locations differ. Table 5-2 shows common set points and sensors needed to make the most of the heat reclaim system. Combinations of measurements can be used to diagnose whether control sequences are working properly; for example if the outdoor air preheat set point is 60°F, the discharge air temperature off of the outdoor air preheat coil is 40°F and the three-way reclaim valve is off, there is likely a problem with the control sequence.

**Table 5-2. Measurements Required**

| Maintenance Considerations—Review Historical Data Points and Trend Logs | |
|---|---|
| **DSH** | |
| Domestic Water Preheat | 3-way valve command, tank temperature, receiver level |
| Mixed Air Heating | 3-way valve command, fan speed, MAT, discharge air temp, receiver level |
| Outdoor Air Preheat | 3-way valve command, fan speed, OAT, damper position, discharge air temp, receiver level |
| | |
| **Condensing** | |
| Mixed Air Heating | 3-way valve command, fan speed, MAT, discharge air temperature, drop-leg temp, receiver level |
| Outdoor Air Preheat | 3-way valve command, fan speed, OAT, damper position, discharge air temperature, drop-leg temperature, receiver level |

Knowing the refrigerant level in the receiver at all times is important for refrigeration system operation. If the receiver does not have sufficient refrigerant to provide 100% liquid refrigerant to the expansion valves, the refrigerant flashes in the liquid line, causing improper valve operation, high superheat, and minimal refrigeration, which result in alarms and product loss. A good time to monitor the receiver level is when a system is coming out of heat reclaim. The transition from one heat exchanger to the other could drain the receiver faster than the pumpout system can add refrigerant to the receiver.

For the system to work correctly, the major sensors must be checked annually for accuracy. Compare the following sensor controller readings to gauge measurements, add offsets, and adjust as necessary.

- Condenser drop-leg pressure sensor
- Heating reclaim tank temperature
- MAT sensor
- Discharge air temperature sensor
- Airflow monitoring station or airflow switch
- Fan/pump speed command and speed reference
- OAT.

## 5.9 Resources


1. Methodology and Procedure for Life Cycle Cost Analyses. U.S. National Archives and Records Administration. Code of Federal Regulations (2006): Title 10 Sec 435-436.
2. Fuller, S.K.; Petersen, S.R. *Life-Cycle Costing Manual for the Federal Energy Management Program. NIST Handbook 135*. Gaithersburg, MD: Building and Fire Research Laboratory, 1996.
3. "Capital Budgeting." Accounting Explained, 2013. http://accountingexplained.com/managerial/capital-budgeting/.
4. "Building Life Cycle Cost Programs." U.S. Department of Energy, 2013. http://www1.eere.energy.gov/femp/information/download_blcc.html.
5. "User Friendly Building Life-Cycle Costing: a spreadsheet implementation of BLCC." DOE2, 2012. http://www.doe2.com/.

# Appendix A: Energy Results

## Introduction

Energy results from the baseline energy model and spreadsheets have been tabulated in Figure A-1. Tables A-2 through A-7 illustrate the energy savings associated with each climate zone, reclaim method, and condensing method. Appendix C includes descriptions of the baseline energy model. The spreadsheet instructions tab includes spreadsheet descriptions. These results are for illustration purposes only and are valid for the assumptions outlined in this playbook. They are not typical for every store in a given climate zone. These results do, however, illustrate general trends that can be used to help designers select heat reclaim methods for a project. They should not be compared directly with the spreadsheet results, which use different assumptions for how the evaporator loads are modeled.

The EnergyPlus results and the modeling guidance in Appendix B are included because the authors expect that in the long run, EnergyPlus and OpenStudio (an easy-to-use user interface for developing and running EnergyPlus models) will accurately simulate all types of heat reclaim strategies once superheat and mass flow modeling limitations are addressed. Researchers and refrigeration system designers already use EnergyPlus and OpenStudio to estimate refrigeration system performance and potential savings from a range of energy conservation strategies. They are well-supported platforms that have the advantage of capturing the dynamic interaction between the refrigeration system and the rest of the store.

## Baseline Energy Model: Results

Table A-1 indicates the modeled annual energy whole-building consumption for the baseline energy model in 17 U.S. locations. Each baseline model includes air-cooled refrigeration condensers. The values are included for reference when calculating heat reclaim energy savings.

**Table A-1. Baseline Energy Model Results by Climate Zone**

| No. | ASHRAE Climate Zone | Representative City | Modeled Baseline Electrical Consumption (kWh) | Modeled Baseline Gas Consumption (therms) | Modeled Baseline Annual Energy Consumption (kBtu) | Modeled Baseline Annual Energy Use Index(kBtu/ft²) |
|---|---|---|---|---|---|---|
| 1 | 1A | Miami, FL | 2,071,838 | 22,649 | 9,334,007 | 198.6 |
| 2 | 2A | Houston, TX | 1,874,150 | 30,303 | 9,424,925 | 200.5 |
| 3 | 2B | Phoenix, AZ | 1,738,818 | 22,769 | 8,209,769 | 174.7 |
| 4 | 3A | Atlanta, GA | 1,664,190 | 35,469 | 9,225,118 | 196.3 |
| 6 | 3B | Los Angeles, CA | 1,566,439 | 31,002 | 8,444,890 | 179.7 |
| 5 | 3B | Las Vegas, NV | 1,581,969 | 27,807 | 8,178,369 | 174.0 |
| 7 | 3C | San Francisco, CA | 1,451,676 | 35,310 | 8,484,148 | 180.5 |
| 8 | 4A | Baltimore, MD | 1,581,993 | 43,865 | 9,784,225 | 208.2 |
| 9 | 4B | Albuquerque, NM | 1,464,846 | 35,412 | 8,539,267 | 181.7 |
| 10 | 4C | Seattle, WA | 1,427,381 | 43,558 | 9,225,998 | 196.3 |
| 12 | 5A | Chicago, IL | 1,521,790 | 51,334 | 10,325,761 | 219.7 |
| 11 | 5A | Boston, MA | 1,500,659 | 49,447 | 10,064,965 | 214.1 |
| 13 | 5B | Denver, CO | 1,437,353 | 41,737 | 9,077,936 | 193.1 |
| 14 | 6A | Minneapolis, MN | 1,481,217 | 56,297 | 10,683,587 | 227.3 |
| 15 | 6B | Helena, MT | 1,393,835 | 51,413 | 9,897,100 | 210.6 |
| 16 | 7 | Duluth, MN | 1,413,584 | 63,228 | 11,145,977 | 237.1 |
| 17 | 8 | Fairbanks, AK | 1,351,434 | 75,302 | 12,141,298 | 258.3 |

U.S. climate zones begin at tropical climates with climate zone 1 and transition gradually from warm to cold through climate zone 8. Subcategories A, B, and C indicate moisture conditions related to the climate zones: A indicates humid, B indicates dry, and C indicates a marine climate. The results demonstrate anticipated energy use per climate zone; warmer climates have higher electricity (refrigeration) consumption and colder climates have higher natural gas (heating) consumption.

A comparison of the resulting energy use intensities of the analysis to benchmarks such as the U.S. Environmental Protection Agency's Target Finder may demonstrate that the baseline energy model performs considerably well to available metrics (ENERGY STAR® scores: ~80–90). When making comparisons to benchmarks (Target Finder or existing utility billings), several factors of the energy model must be considered. Benchmarks represent actual consumption of existing buildings, which may contain aging equipment affected by human behavior, under real weather conditions. An energy model represents a scenario of a building performing ideally with all components (equipment, constructions, people, etc.) behaving predictably and to specification, which is seldom the case in any building that is not continuously commissioned.

Inputs to the energy model of note leading to high baseline energy performance are that many real refrigeration systems may not typically be controlled to a 70°F minimum condensing temperature, nor may most connected display cases be DOE 2012 compliant (installed pre-2012). Furthermore, the assumptions used from the ASHRAE 90.1-2004 User Manual for miscellaneous equipment loads may not be sufficient to satisfy the requirements of all supermarkets. The combination of these items may cause the model to appear too efficient compared to existing supermarkets, but is sufficient when considered that the baseline is being used only for the purposes of "economy of scale" comparisons between refrigeration system components. (Refer to Appendix C for further information about the refrigeration baseline inputs.)

## Heat Reclaim Spreadsheets: Results

Tables A-2 through A-7 illustrate the energy savings associated with each climate zone, reclaim method, and condensing method. These results are based on the baseline energy model as described in Appendix C and use the spreadsheet tools provided with this playbook. The results are discussed at the bottom of this section.

## Low-Temperature Refrigeration Spreadsheet Calculation Results

Table A-2 through Table A-4 illustrate the energy savings for low-temperature refrigeration systems, which include ice cream and frozen food. These systems have a lower evaporator temperature and pressure and a higher compressor discharge temperature than medium-temperature systems. Different tables show results with different condensing strategies.

**Table A-2. Tabulated Results for Heat Reclaim From Baseline Low-Temperature Refrigeration Systems With Air-Cooled Condensers**

| No. | ASHRAE Climate Zone | Representative City | DHW Desuperheating Savings (kbtu) | Space Heating Full Condensing Savings (kbtu) | Space Heating Desuperheating Savings (kbtu) | Ventilation Full Condensing Savings (kbtu) | Ventilation Desuperheating Savings (kbtu) |
|---|---|---|---|---|---|---|---|
| 1 | 1A | Miami, FL | 265,761 | 512,782 | 189,195 | -8,879 | -14,038 |
| 2 | 2A | Houston, TX | 264,778 | 782,508 | 231,934 | 110,603 | 25,414 |
| 3 | 2B | Phoenix, AZ | 284,859 | 676,161 | 204,035 | 72,113 | 16,861 |
| 4 | 3A | Atlanta, GA | 256,082 | 942,543 | 262,520 | 210,782 | 54,640 |
| 6 | 3B | Los Angeles, CA | 228,388 | 1,176,558 | 334,633 | 55,856 | 25,410 |
| 5 | 3B | Las Vegas, NV | 275,570 | 898,983 | 246,113 | 178,066 | 47,721 |
| 7 | 3C | San Francisco, CA | 223,711 | 1,317,392 | 330,268 | 241,068 | 100,152 |
| 8 | 4A | Baltimore, MD | 251,461 | 1,013,205 | 278,475 | 330,594 | 85,328 |
| 9 | 4B | Albuquerque, NM | 254,925 | 974,207 | 270,468 | 302,007 | 80,190 |
| 10 | 4C | Seattle, WA | 230,693 | 1,271,816 | 315,505 | 426,737 | 124,657 |
| 12 | 5A | Chicago, IL | 252,309 | 1,005,135 | 288,159 | 399,669 | 100,007 |
| 11 | 5A | Boston, MA | 244,659 | 1,102,669 | 299,485 | 419,316 | 107,921 |
| 13 | 5B | Denver, CO | 255,031 | 976,418 | 281,565 | 374,205 | 96,568 |
| 14 | 6A | Minneapolis, MN | 254,703 | 1,001,394 | 292,792 | 444,013 | 108,594 |
| 15 | 6B | Helena, MT | 247,772 | 1,063,681 | 301,019 | 473,759 | 121,963 |
| 16 | 7 | Duluth, MN | 246,398 | 1,103,733 | 311,989 | 516,599 | 131,694 |
| 17 | 8 | Fairbanks, AK | 249,030 | 1,133,295 | 317,776 | 566,786 | 144,271 |

**Table A-3. Tabulated Results for Heat Reclaim From Baseline Low-Temperature Refrigeration Systems With Evaporative-Cooled Condensers**

| No. | ASHRAE Climate Zone | Representative City | DHW Desuperheating Savings (kbtu) | Space Heating Full Condensing Savings (kbtu) | Space Heating Desuperheating Savings (kbtu) | Ventilation Full Condensing Savings (kbtu) | Ventilation Desuperheating Savings (kbtu) |
|---|---|---|---|---|---|---|---|
| 1 | 1A | Miami, FL | 301,320 | 507,319 | 221,815 | -9,135 | -13,943 |
| 2 | 2A | Houston, TX | 290,883 | 805,051 | 269,563 | 108,593 | 26,143 |
| 3 | 2B | Phoenix, AZ | 239,173 | 669,785 | 205,470 | 70,492 | 16,661 |
| 4 | 3A | Atlanta, GA | 272,887 | 986,226 | 290,610 | 207,749 | 55,137 |
| 6 | 3B | Los Angeles, CA | 266,206 | 1,204,270 | 397,184 | 54,246 | 26,622 |
| 5 | 3B | Las Vegas, NV | 232,410 | 879,630 | 238,007 | 174,899 | 46,981 |
| 7 | 3C | San Francisco, CA | 253,631 | 1,354,297 | 386,191 | 236,592 | 105,034 |
| 8 | 4A | Baltimore, MD | 266,747 | 1,070,377 | 301,993 | 326,171 | 85,603 |
| 9 | 4B | Albuquerque, NM | 241,554 | 999,005 | 275,112 | 297,902 | 79,170 |
| 10 | 4C | Seattle, WA | 246,967 | 1,298,171 | 347,375 | 421,353 | 127,784 |
| 12 | 5A | Chicago, IL | 265,444 | 1,089,346 | 310,932 | 395,358 | 100,348 |
| 11 | 5A | Boston, MA | 258,332 | 1,163,408 | 322,984 | 414,417 | 108,691 |
| 13 | 5B | Denver, CO | 244,384 | 1,023,899 | 288,123 | 369,976 | 95,835 |
| 14 | 6A | Minneapolis, MN | 263,267 | 1,068,262 | 309,058 | 439,658 | 108,557 |
| 15 | 6B | Helena, MT | 244,111 | 1,095,216 | 305,983 | 469,134 | 121,358 |
| 16 | 7 | Duluth, MN | 256,009 | 1,171,270 | 332,516 | 511,877 | 132,270 |
| 17 | 8 | Fairbanks, AK | 251,910 | 1,162,237 | 328,579 | 562,863 | 144,543 |

**Table A-4. Tabulated Results for Heat Reclaim From Baseline Low-Temperature Refrigeration Systems With Hybrid Condensers**

| No. | ASHRAE Climate Zone | Representative City | DHW Desuperheating Savings (kbtu) | Space Heating Full Condensing Savings (kbtu) | Space Heating Desuperheating Savings (kbtu) | Ventilation Full Condensing Savings (kbtu) | Ventilation Desuperheating Savings (kbtu) |
|---|---|---|---|---|---|---|---|
| 1 | 1A | Miami, FL | 263,189 | 499,484 | 197,616 | -9,120 | -14,166 |
| 2 | 2A | Houston, TX | 254,906 | 783,160 | 241,246 | 108,594 | 24,757 |
| 3 | 2B | Phoenix, AZ | 223,499 | 640,993 | 194,694 | 70,435 | 16,271 |
| 4 | 3A | Atlanta, GA | 256,141 | 965,049 | 275,822 | 207,550 | 54,074 |
| 6 | 3B | Los Angeles, CA | 281,114 | 1,218,500 | 417,024 | 54,193 | 27,712 |
| 5 | 3B | Las Vegas, NV | 228,949 | 873,176 | 235,594 | 174,824 | 46,874 |
| 7 | 3C | San Francisco, CA | 285,264 | 1,397,358 | 431,246 | 237,525 | 113,996 |
| 8 | 4A | Baltimore, MD | 246,578 | 1,031,493 | 282,507 | 326,172 | 84,151 |
| 9 | 4B | Albuquerque, NM | 258,306 | 1,044,925 | 290,182 | 297,926 | 80,224 |
| 10 | 4C | Seattle, WA | 266,148 | 1,324,322 | 376,332 | 424,335 | 134,752 |
| 12 | 5A | Chicago, IL | 249,997 | 1,040,860 | 293,966 | 395,211 | 98,966 |
| 11 | 5A | Boston, MA | 249,235 | 1,143,039 | 311,781 | 414,126 | 107,361 |
| 13 | 5B | Denver, CO | 261,004 | 1,077,177 | 305,280 | 370,270 | 97,468 |
| 14 | 6A | Minneapolis, MN | 250,654 | 1,018,472 | 294,834 | 439,465 | 107,437 |
| 15 | 6B | Helena, MT | 265,211 | 1,178,203 | 335,454 | 470,801 | 126,089 |
| 16 | 7 | Duluth, MN | 252,652 | 1,154,187 | 327,347 | 511,708 | 131,584 |
| 17 | 8 | Fairbanks, AK | 269,010 | 1,226,570 | 355,054 | 564,886 | 149,724 |

## Medium-Temperature Refrigeration Spreadsheet Calculation Results

Table A-5 through Table A-7 illustrate the example energy savings for medium-temperature refrigeration systems such as produce, beverage, dairy, and meat cases. These systems have a higher evaporator temperature and pressure and a lower compressor discharge temperature than low-temperature systems.

### Table A-5. Tabulated Results for Heat Reclaim From Baseline Medium-Temperature Refrigeration Systems With Air-Cooled Condensers

| No. | ASHRAE Climate Zone | Representative City | DHW Desuperheating Savings (kbtu) | Space Heating Full Condensing Savings (kbtu) | Space Heating Desuperheating Savings (kbtu) | Ventilation Full Condensing Savings (kbtu) | Ventilation Desuperheating Savings (kbtu) |
|---|---|---|---|---|---|---|---|
| 1 | 1A | Miami, FL | 356,905 | 595,202 | 360,339 | -8,564 | -11,167 |
| 2 | 2A | Houston, TX | 356,831 | 1,047,545 | 451,413 | 156,141 | 54,799 |
| 3 | 2B | Phoenix, AZ | 373,551 | 890,376 | 381,158 | 85,090 | 38,603 |
| 4 | 3A | Atlanta, GA | 347,809 | 1,323,460 | 499,955 | 333,769 | 104,099 |
| 6 | 3B | Los Angeles, CA | 309,797 | 1,592,459 | 659,309 | 57,055 | 42,578 |
| 5 | 3B | Las Vegas, NV | 367,257 | 1,217,414 | 475,195 | 228,220 | 91,551 |
| 7 | 3C | San Francisco, CA | 301,730 | 1,886,813 | 646,912 | 256,126 | 165,677 |
| 8 | 4A | Baltimore, MD | 343,325 | 1,570,234 | 522,044 | 584,549 | 157,508 |
| 9 | 4B | Albuquerque, NM | 349,493 | 1,419,049 | 507,312 | 459,303 | 147,687 |
| 10 | 4C | Seattle, WA | 310,649 | 1,935,072 | 597,371 | 603,911 | 221,849 |
| 12 | 5A | Chicago, IL | 349,765 | 1,670,015 | 530,989 | 787,166 | 182,733 |
| 11 | 5A | Boston, MA | 333,134 | 1,790,361 | 556,136 | 769,613 | 196,138 |
| 13 | 5B | Denver, CO | 352,584 | 1,526,120 | 518,970 | 640,940 | 175,821 |
| 14 | 6A | Minneapolis, MN | 357,824 | 1,720,725 | 532,131 | 902,820 | 198,932 |
| 15 | 6B | Helena, MT | 348,183 | 1,774,639 | 547,968 | 883,509 | 219,551 |
| 16 | 7 | Duluth, MN | 355,084 | 1,920,180 | 566,204 | 1,048,248 | 236,100 |
| 17 | 8 | Fairbanks, AK | 373,176 | 2,024,307 | 570,196 | 1,198,192 | 257,540 |

### Table A-6. Tabulated Results for Heat Reclaim From Baseline Medium-Temperature Refrigeration Systems With Hybrid Condensers

| No. | ASHRAE Climate Zone | Representative City | DHW Desuperheating Savings (kbtu) | Space Heating Full Condensing Savings (kbtu) | Space Heating Desuperheating Savings (kbtu) | Ventilation Full Condensing Savings (kbtu) | Ventilation Desuperheating Savings (kbtu) |
|---|---|---|---|---|---|---|---|
| 1 | 1A | Miami, FL | 374,032 | 599,272 | 378,039 | -8,821 | -11,183 |
| 2 | 2A | Houston, TX | 366,853 | 1,168,477 | 483,991 | 153,325 | 55,022 |
| 3 | 2B | Phoenix, AZ | 290,387 | 904,126 | 369,894 | 83,174 | 38,014 |
| 4 | 3A | Atlanta, GA | 350,604 | 1,427,758 | 525,752 | 328,860 | 103,617 |
| 6 | 3B | Los Angeles, CA | 340,368 | 1,639,598 | 711,948 | 55,407 | 42,534 |
| 5 | 3B | Las Vegas, NV | 285,498 | 1,170,219 | 444,155 | 224,170 | 89,952 |
| 7 | 3C | San Francisco, CA | 327,588 | 2,175,386 | 711,231 | 250,886 | 168,593 |
| 8 | 4A | Baltimore, MD | 347,237 | 1,667,417 | 542,669 | 576,891 | 156,516 |
| 9 | 4B | Albuquerque, NM | 307,776 | 1,397,168 | 495,573 | 453,114 | 145,429 |
| 10 | 4C | Seattle, WA | 321,444 | 2,208,427 | 633,687 | 595,074 | 223,413 |
| 12 | 5A | Chicago, IL | 355,157 | 1,776,814 | 553,998 | 778,372 | 181,849 |
| 11 | 5A | Boston, MA | 340,881 | 1,903,480 | 578,465 | 760,183 | 195,654 |
| 13 | 5B | Denver, CO | 319,810 | 1,542,731 | 516,170 | 633,495 | 173,784 |
| 14 | 6A | Minneapolis, MN | 358,415 | 1,797,237 | 547,223 | 893,587 | 197,777 |
| 15 | 6B | Helena, MT | 329,390 | 1,799,764 | 545,563 | 874,350 | 217,283 |
| 16 | 7 | Duluth, MN | 362,259 | 2,014,874 | 589,240 | 1,038,213 | 235,268 |
| 17 | 8 | Fairbanks, AK | 371,301 | 2,067,599 | 579,336 | 1,188,971 | 256,528 |

**Table A-7. Tabulated Results for Heat Reclaim From Baseline Medium-Temperature Refrigeration Systems With Evaporative-Cooled Condensers**

| No. | ASHRAE Climate Zone | Representative City | DHW Desuperheating Savings (kbtu) | Space Heating Full Condensing Savings (kbtu) | Space Heating Desuperheating Savings (kbtu) | Ventilation Full Condensing Savings (kbtu) | Ventilation Desuperheating Savings (kbtu) |
|---|---|---|---|---|---|---|---|
| 1 | 1A | Miami, FL | 320,401 | 572,849 | 351,301 | -8,806 | -11,358 |
| 2 | 2A | Houston, TX | 315,813 | 1,024,564 | 443,330 | 153,434 | 53,564 |
| 3 | 2B | Phoenix, AZ | 265,701 | 805,723 | 351,481 | 83,177 | 37,594 |
| 4 | 3A | Atlanta, GA | 324,087 | 1,341,329 | 499,682 | 328,908 | 102,551 |
| 6 | 3B | Los Angeles, CA | 357,837 | 1,649,018 | 732,394 | 55,297 | 43,147 |
| 5 | 3B | Las Vegas, NV | 280,903 | 1,152,203 | 439,867 | 224,101 | 89,836 |
| 7 | 3C | San Francisco, CA | 370,245 | 2,227,210 | 766,798 | 250,271 | 176,275 |
| 8 | 4A | Baltimore, MD | 316,899 | 1,547,331 | 510,995 | 577,060 | 155,079 |
| 9 | 4B | Albuquerque, NM | 333,583 | 1,529,369 | 521,282 | 452,813 | 146,333 |
| 10 | 4C | Seattle, WA | 350,142 | 2,360,951 | 672,106 | 594,308 | 231,724 |
| 12 | 5A | Chicago, IL | 328,955 | 1,664,735 | 524,776 | 778,562 | 180,431 |
| 11 | 5A | Boston, MA | 325,648 | 1,832,677 | 559,927 | 760,218 | 194,125 |
| 13 | 5B | Denver, CO | 343,852 | 1,677,365 | 542,215 | 633,119 | 175,443 |
| 14 | 6A | Minneapolis, MN | 336,594 | 1,698,984 | 521,878 | 893,733 | 196,515 |
| 15 | 6B | Helena, MT | 363,935 | 2,009,368 | 591,212 | 873,644 | 222,541 |
| 16 | 7 | Duluth, MN | 355,257 | 1,981,266 | 579,504 | 1,038,171 | 234,438 |
| 17 | 8 | Fairbanks, AK | 396,679 | 2,165,730 | 617,212 | 1,188,236 | 262,272 |

## Discussion

Several trends emerge from the example results presented in Tables A-1 through A-7. The condensing method has some effect on the total energy recovered through heat reclaim, but the effect is modest when compared to the total heat recovered. The condensing method affects the energy saved in two ways:

- It affects the fan power required. Lower fan power results in less opportunity for condenser fan savings.
- It affects the condensing temperature. When using evaporative and hybrid condensing methods, the condensing temperature approaches the outdoor wb temperature. An air-cooled condenser approaches the outdoor db temperature. A lower condensing temperature results in higher quantities of heat but lower quality of heat. In general, the energy savings from reducing condensing temperature surpass the savings from increasing condensing temperature to boost heat reclaim.

The energy savings from DSH refrigerant for use in SHW are relatively constant throughout each climate zone and condensing method. This heat reclaim method relies somewhat on OAT and groundwater temperature, but the effect is relatively small compared with space heating and ventilation preheat savings.

The heat reclaim methods that reject heat to an airstream are heavily dependent on outdoor air conditions. In general, these technologies provide more benefit in cold climates than in warmer climates. This example shows an energy penalty, caused by ancillary losses, for installing ventilation preheat in Miami, Florida, while Fairbanks, Alaska, shows considerable energy savings from heat reclaim. Los Angeles, California, is somewhat of an anomaly in the data; it shows very good energy savings from heat reclaim for space heating. This particular weather file is located close to the ocean and has little temperature fluctuation throughout the year. For the bulk of the year, this temperature is slightly lower than the store's balance point temperature, meaning the store requires heat. This OAT generally corresponds to a condensing temperature somewhat higher than the minimum condensing temperature, resulting in more heat available for reclaim. This combination of nearly constant heating demand and heat available makes this particular scenario favorable for space-heating heat reclaim strategies.

The trends for heat reclaim in medium-temperature systems, as shown in Tables A-5 through A-7, generally follow the same patterns as the low-temperature systems; however, medium-temperature systems in this example have significantly more capacity than low-temperature systems, resulting in more opportunities for heat reclaim energy savings. Although the THR is much greater in medium-temperature systems, the refrigerant discharge temperature is considerably lower than in a low-temperature system.

The results of the example calculations illustrate that heat reclaim is not a "one size fits all" type of energy conservation strategy. The refrigeration systems affected, climate, and condensing method, among other factors, are critical to heat reclaim energy savings and should be evaluated separately for each application.

# Appendix B: Notes to Energy Modelers

## Introduction

Refrigeration systems are the largest single energy end use in supermarkets, but predicting system performance with an energy model can be difficult and time consuming. This section is intended to equip energy modelers with strategies for creating more accurate refrigeration energy models in general and for heat reclaim, in particular.

## General Notes

The primary difficulty in modeling refrigeration systems is quantifying the heat transfer between the refrigerated cases and walk-ins with the rest of the store. This quantity depends on space temperature, case temperature, humidity, type of case, infiltration, frequency of use, and loads internal to the case such as lights, fans, defrost, and anti-sweat heaters. Energy models simplify all these factors to reduce complexity and simulation time.

Energy models typically assume that the entire space that includes refrigerated systems is the same temperature and humidity. This is not the case in a real supermarket. The refrigerated aisles spill a certain amount of cold air into the space. Because this air is colder than the rest of the space, it generally stays low to the ground. To complicate the matter, supermarket designs often include thermostats mounted higher above the floor than is typical of other locations to ensure that the HVAC systems are not always in heating mode. These factors create a microclimate on a sales floor and complicate the energy modeling process.

The issue of microclimates can be addressed in one of several ways in an energy model. The simplest may be to reduce the case load transferred to the space by a factor and lower the space heating set point. The quantity of heat transferred to the space relative to the evaporator capacity is commonly referred to as *case credits*. Many good resources for determining case credits, such as *Case Credits & Return Air Paths for Supermarkets*, are available. Its disadvantage is that it depends on rules of thumb and may thus be inaccurate in some instances.

A more accurate method of modeling refrigerated systems in a larger sales area is to create separate zones for refrigerated systems. These zones should communicate thermally with the larger sales zone. The refrigerated zone could be defined with air walls around it that allow heat to transfer between the refrigerated areas and the greater sales floor. This method is an attempt to define the microclimate in the refrigerated aisles. The airflow from the HVAC unit to the refrigerated area can then be entered into the zone inputs even if the zone temperature is not explicitly controlled. This model is not entirely accurate as it does not account for convection effects between the zones. Some energy modeling packages can take these effects into account, but the quantities of heat and air transferred are difficult to define.

When this playbook was written, the primary programs used to simulate refrigeration systems were EnergyPlus and DOE2-R. eQUEST, a DOE2 user interface, has somewhat limited refrigeration modeling capacity; however, refrigeration-specific versions are available. These refrigeration versions are less user friendly than standard versions of eQUEST and do not include the same functionality for modeling other building systems. OpenStudio, an EnergyPlus user interface, has nearly the same capabilities as EnergyPlus. Many other commercially available energy modeling packages have focused on HVAC systems and do not contain refrigeration system functionality.

Schedules can have major impacts on refrigeration system performance. For example, refrigeration design best practices dictate that case defrost cycles be offset; that way, case temperatures are not raised and lowered in unison, which increase the load on the compressor and potentially raise the store demand. These schedules should be offset in the simulation program as they would in reality.

## Notes for EnergyPlus Users

EnergyPlus currently has one of the most robust refrigeration simulation algorithms available. Although other software packages have been developed with similar capabilities, EnergyPlus is the tool of choice for this guide. The following notes are specific to EnergyPlus users.

The issue of microclimates can be addressed in EnergyPlus using the methods described. EnergyPlus has added several room air models such as RoomAir:TemperaturePattern: UserDefined, ConstantGradient, Two Gradient, NondimensionalHeight, and SurfaceMapping to account for this issue.

Compressor coefficients are critical to modeling the performance of a refrigeration system. When entering compressor coefficients into EnergyPlus, the order of these coefficients does not match the Air-Conditioning and Research Institute 540 for rating compressor performance. The EnergyPlus IDF Editor and Input Output Reference include guidance for entering the manufacturer compressor coefficients into EnergyPlus.

Heat reclaim calculations in EnergyPlus are currently limited for superheat applications. EnergyPlus does not calculate the amount of superheat available each hour; rather, it relies on a user input for superheat as a ratio of THR. EnergyPlus uses this value as a constant for each hour of operation. Experience and hand calculations show significant fluctuation in the superheat/THR ratio. Thus, a spreadsheet approach was determined to be a more accurate method of calculating the energy performance of DSH reclaim strategies. EnergyPlus also does not have an option for an outdoor air preheat coil. This would have to be modeled using a dedicated outdoor air system.

In the case of the integration of refrigeration and water source heat pump systems, an energy model is preferable to spreadsheets due to the complexity of the water loop controls.

## Resources


1. ANSI/AHRI Standard 540 (formerly ARI Standard 540) – Standard For Performance Rating of Positive Displacement Refrigerant Compressors and Compressor Units. Arlington, VA: Air-Conditioning, Heating, and Refrigeration Institute, 2004.
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5. “EnergyPlus Engineering Reference.” The Board of Trustees of the University of Illinois and the Regents of the University of California through the Ernest Orlando Lawrence Berkeley National Laboratory, 2013.

# Appendix C: Baseline Energy Modeling Assumptions

## Introduction

The purpose of the reference supermarket was to realistically benchmark refrigeration system loads and system performance. The model is intended to represent a building constructed to 10-year-old energy codes to represent the bulk of existing building stock and maximize the energy savings impact.

## Baseline Energy Model: Overview

EnergyPlus was the energy modeling software selected for baseline energy modeling, because it can represent complex supermarket refrigeration systems. The baseline energy model was initially generated using OpenStudio and was then transferred for completion to EnergyPlus, because features required to complete the model were unavailable at that time in OpenStudio.

Climatic design data were selected at the ASHRAE 0.4% db cooling and 99.6% db heating conditions. TMY3-based EPW weather data for EnergyPlus were referenced and the model was applied across 17 U.S. locations. Table C-1 shows the cities that were selected for evaluation to provide geographic depth of results for the baseline energy models.

**Table C-1. Cities Selected for Geographic Depth of Results**

| No. | ASHRAE Climate Zone | Representative City | EPW Weather File Source |
|---|---|---|---|
| 1 | 1A | Miami, FL | Miami International Airport |
| 2 | 2A | Houston, TX | Bush Intercontinental Airport |
| 3 | 2B | Phoenix, AZ | Sky Harbor International Airport |
| 4 | 3A | Atlanta, GA | Hartsfield-Jackson International Airport |
| 5 | 3B | Los Angeles, CA | Los Angeles International Airport |
| 6 | 3B | Las Vegas, NV | McCarran International Airport |
| 7 | 3C | San Francisco, CA | San Francisco International Airport |
| 8 | 4A | Baltimore, MD | Baltimore-Washington International Airport |
| 9 | 4B | Albuquerque, NM | Albuquerque International Airport |
| 10 | 4C | Seattle, WA | Seattle-Tacoma International Airport |
| 11 | 5A | Chicago, IL | Chicago-O'Hare International Airport |
| 12 | 5A | Boston, MA | Logan International Airport |
| 13 | 5B | Denver, CO | Denver International Int'l Airport |
| 14 | 6A | Minneapolis, MN | Minneapolis-St. Paul International Airport |
| 15 | 6B | Helena, MT | Helena Regional Airport |
| 16 | 7 | Duluth, MN | Duluth International Airport |
| 17 | 8 | Fairbanks, AK | Fairbanks International Airport |

The baseline building energy model for this playbook was a single-story 47,000-$ft^2$ supermarket based on a Food Marketing Institute study indicating that the average supermarket floor area from 2004 through 2010 was 46,980 $ft^2$. The baseline building was divided into five zones: sales floor, refrigeration area, backroom, offices, and service departments. It was modeled to comply with the requirements of ASHRAE 90.1-2004 Appendix G, with exceptions or modifications as noted. This code was selected because it was assumed to represent the bulk of the existing U.S. building stock.

## Baseline Energy Model: Building Constructions

Building envelopes for each climate zone were modeled to meet the baseline requirements established by ASHRAE 90.1-2004 Tables 5.5-1 through 5.5-8 and Appendix G. **Exception:** ASHRAE 90.1-2004 Appendix G indicates that exterior walls shall be steel-frame construction; however, solid-grouted concrete masonry unit walls with continuous insulation were used to be more representative of common supermarket construction practices. Building envelope material data were obtained from NREL's Building Component Library. If materials were unavailable from the Building Component Library, they were selected from the provided EnergyPlus materials data set or were created from material tables provided in the *ASHRAE Fundamentals 2005 Handbook*.

## Baseline Energy Model: Zone Loads and Schedules

Zone occupancy, lighting and load profiles, and related mechanical system and building schedules were set to match the profiles established by the *ASHRAE 90.1-2004 User Manual* Table G-B and Tables G-E through G-N. As the building is considered mixed-use, each modeled zone was independently considered to best match the building types per Table G-B.

## Baseline Energy Model: Zone Mechanical Systems

HVAC systems were modeled to meet Baseline System 3 (packaged single-zone air conditioning) per ASHRAE 90.1-2004 Tables G3.1.1A and G3.1.1B. **Exception:** air distribution systems were sized to provide 1 CFM/ft² for supermarkets per the *ASHRAE Applications 2003 Handbook*. Mechanical equipment capacities and efficiencies were sized per ASHRAE 90.1-2004 Appendix G with a heating sizing factor of 1.25, cooling sizing factor of 1.15, and heating and cooling supply temperature differentials of 20°F between the supply air temperature and thermostat set point. Mechanical capacity and EIR curves were modeled using OpenStudio defaults. Economizers were not included because of refrigerated equipment in spaces per ASHRAE 90.1-2004 Section G3.1.2.6(b).

Mechanical heating was modeled as natural gas and mechanical cooling as direct expansion. Fans were set to operate at constant volume during occupied hours and cycle as required to maintain setback set points during unoccupied hours. Zone heating and cooling set points were input per the *ASHRAE Refrigeration 2002 Handbook* for the sales zone and *ASHRAE Applications Handbook 2003* for all other zones. Dehumidification controls were established to limit each air-conditioned zone to 50°F maximum dew point temperature to minimize the impact to building refrigeration systems. All conditioned zones were programmed via the EnergyPlus energy management system to activate night-cycle operation for space temperature and dehumidification control during unoccupied periods.

Infiltration air leakage rates were estimated using Chartered Institution of Building Service Engineers TM23 Building Tightness Specifications for Supermarkets with infiltration to meet the "Good Practice" qualification. The pressure drop coefficient per Chartered Institution of Building Service Engineers TM23 in CFM/ft$^2$ was corrected to match the reference wind speed of the BLAST coefficients (7.5 mph) using Infiltration Modeling Guidelines for Commercial Building Energy Analysis. Air changes per hour were then calculated volumetrically per zone and scheduled to reduce infiltration to 25% during zone occupied hours to reflect building pressurization.

Refrigerated display case end-use loads located within the retail sales area were separated into a dedicated zone that shares the HVAC system with the sales floor zone. The purpose was to simulate the microclimate effect in a supermarket where the refrigeration effect of the display cases removes heat from the surrounding environment, causing "cold spots" on the sales floor where cases are located. The driving thermostat for the shared HVAC system is located within the sales floor zone, which causes the temperature in the refrigeration area zone to remain relatively low as the sales floor zone is satisfied. The result is a simulation of the microclimate and, incidentally, a reduction in required refrigeration load as the temperature difference of the case operating temperatures and their surrounding environment is reduced from their rated conditions.

## Baseline Energy Model: Service Hot Water

The baseline model water heater component was sized to accommodate an assumed 2,700 gallons per day of water consumption at a 140°F set point and an 80% combustion fuel efficiency. The load profiles for the SHW system were applied from the *ASHRAE 90.1-2004 User Manual* Tables G–L. The domestic cold water temperature profile is generated using the EnergyPlus correlation method which estimates entering cold water temperatures based on outside db temperatures using a method defined in "Towards Development of an Algorithm for Mains Water Temperature (Burch and Christensen 2007)."

## Baseline Energy Model: Refrigeration System Overview

A representative set of input assumptions for the baseline supermarket refrigeration systems was adapted from an existing supermarket in western Montana. This supermarket was selected because its zone types and areas were similar to the intended baseline model. Many of the modeled assumptions established for the baseline refrigeration systems were based on industry standards, common practices, and internal discussions with NREL to confirm and approve these assumptions.

Baseline refrigerated display case capacities were modeled to adhere to the DOE 2012 Standards for Commercial Refrigeration (DOE 2012). The selected standard identifies end-use equipment categorically and identifies the maximum allowable daily energy consumption for each connected refrigeration system. The maximum allowable daily energy consumption for each equipment class in DOE 2012 consolidates all energy required to drive the indicated refrigeration system (compressor power, internal display loads, condenser fan energy, etc.) per each unit of total display area. Technical Support Documents for the DOE 2012 ruling were used to isolate performance characteristics for individual components for each equipment class that comprises the maximum allowable daily energy consumption. This enabled NREL researchers to appropriately input refrigeration systems into EnergyPlus in a manner that complied with the categories defined within DOE 2012.

Baseline refrigerated walk-in coolers and freezers were modeled to adhere to the Energy Independence and Security Act of 2007, which established mandatory federal requirements for minimum insulation values of walk-in cooler and freezer panels and freezer floors, evaporator fan selection, lighting efficacy, glazing construction, and anti-sweat heater power (see Table C-2).

**Table C-2. DOE 2012 Governed Baseline Refrigeration Systems**

| DOE 2012 Equipment Category | Condensing Unit Configuration | Equipment Family | Equipment Class Designation | Quantity (Linear Feet) |
|---|---|---|---|---|
| Commercial Refrigerators and Commercial Freezers | Remote (RC) | Vertical Open | VOP.RC.M | 288 |
| | | Semivertical Open | SVO.RC.M | 56 |
| | | Horizontal Open | HZO.RC.M | 36 |
| | | Vertical Closed Transparent | VCT.RC.L | 136 |
| | | Service Over Counter | SOC.RC.M | 28 |
| Commercial Ice Cream Freezers | Remote (RC) | Vertical Closed Transparent | VCT.RC.I | 144 |
| | | Horizontal Open | HZO.RC.I | 8 |
| **Other Connected Refrigeration Systems** | | | | |
| **Equipment Type** | **Condensing Unit Configuration** | **Equipment Family** | **Equipment Class Desig.** | **Quantity (Square Feet)** |
| Medium-Temperature Walk-Ins | Remote (RC) | n/a | n/a | 2,795 |
| Low-Temperature Walk-In | Remote (RC) | n/a | n/a | 120 |
| Ice Cream Walk-In | Remote (RC) | n/a | n/a | 520 |

## Baseline Energy Model: Refrigeration Compressor Systems

The refrigeration compressor systems were modeled to represent a typical supermarket refrigeration system installation. The refrigerant for each system was modeled as R-404a, which is a low-glide zeotropic refrigerant that is commonly used in the supermarket industry. Compressor racks were equipped with semihermetic reciprocating compressors. The baseline refrigeration system comprised four compressor systems: Rack A, Rack B, Rack C, and Rack D. Racks A and B were low-temperature systems that operated at –25°F saturated suction temperature. Racks C and D were medium-temperature racks that operated at +21°F saturated suction temperature. End-use loads assigned to the modeled racks were selected to correspond to the connected loads of the reference supermarket template.

Each refrigeration compressor system was enabled to simulate floating suction pressure control. To perform this simulation, EnergyPlus calculated the maximum allowable evaporator temperature for each connected system and set the system temperature to the lowest calculated evaporator temperature each time step.

Each refrigeration compressor system incorporated a “dummy load” that was created to set the system operating temperature to account for suction line pressure drop. The compressor system calculations inherent within EnergyPlus set each system operating temperature to 1°K below the lowest connected evaporator temperature. The dummy loads further reduced low-temperature systems by 1.2°F and medium-temperature systems by 0.2°F to bring the systems in line with conventional temperature reductions of 3°F for low-temperature systems and 2°F for medium-temperature systems accounted for during system design. The assigned dummy loads did not add or remove evaporator loads from the system and only impacted each system’s operating temperature.

Compressors were selected to maintain a constant return gas superheat setting of 40°F for low-temperature systems and 30°F for medium-temperature systems. EnergyPlus maintains a constant 7°F evaporator superheat throughout the simulation, which was included in the compressor superheat constant. Each rack included a liquid suction heat exchanger to transfer heat from the liquid line to the suction line. Each liquid suction heat exchanger was modeled to provide a 50°F subcooled liquid temperature at the minimum condensing condition.

Compressors were input into EnergyPlus using compressor coefficients adhering to the format defined by ANSI/AHRI Standard 540-2004. EnergyPlus accepts these compressor coefficients to generate capacity and power performance curves based upon the saturated suction temperature entering and the saturated discharge temperature leaving the compressor (see Table C-3). The coefficient formula defined by ANSI/AHRI Standard 540-2004 for calculating compressor performance and compressor coefficients used in the baseline energy model is indicated in this equation:

$$X = C1 + C2(S) + C3(D) + C4(S2) + C5(S \cdot D) + C6(D2) + C7(S3) + C8(D \cdot S2) + C9(S \cdot D2) + C10(D3)$$

Where,

| | | |
|---|---|---|
| D | = | condensing dew-point temperature in °C |
| S | = | suction dew-point temperature in °C |
| X | = | compressor capacity or power input in Watts |

**Table C-3. Low- and Medium-Temperature Compressor Coefficients**

| Low-Temperature Compressor Coefficients (SI Units*) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| **Input Type** | $C_1$ | $C_2$ | $C_3$ | $C_4$ | $C_5$ | $C_6$ | $C_7$ | $C_8$ | $C_9$ | $C_{10}$ |
| Capacity (W) | 170,752 | 6,103 | –2,959 | 78.26 | –90.96 | 19.79 | 0.436 | –0.609 | 0.459 | –0.041 |
| Power (W) | 29,669 | 1,028 | –442.2 | 18.11 | –20.38 | 11.13 | 0.140 | –0.218 | 0.204 | –0.059 |
| **Medium-Temperature Compressor Coefficients (SI Units*)** | | | | | | | | | | |
| **Input Type** | $C_1$ | $C_2$ | $C_3$ | $C_4$ | $C_5$ | $C_6$ | $C_7$ | $C_8$ | $C_9$ | $C_{10}$ |
| Capacity (W) | 151,166 | 5,266 | –1,852 | 60.67 | –46.93 | 5.731 | 0.136 | –0.403 | –0.039 | –0.036 |
| Power (W) | 7,439 | –269.6 | 553.6 | –9.856 | 14.87 | –3.386 | –0.101 | 0.097 | –0.040 | 0.010 |

* Note: EnergyPlus accepts coefficient inputs in SI units only.

## Baseline Energy Model: Refrigeration Condensers

The refrigeration system condensers were modeled as air-cooled for all locations with one condenser assigned per refrigeration compressor system. Condensers operated using constant-volume fans that cycled on load. This was simulated in EnergyPlus by using the FixedLinear condenser fan control method which calculated part-load fan power as a linear function of the rejected heat load. Condensers serving low-temperature systems were sized for a saturated condensing temperature at 10°F over the ambient db temperature. Condensers serving medium-temperature systems were sized for 15°F over the outside temperature. All condensers were controlled to hold a minimum 70°F saturated condensing temperature to maintain required pressures at mechanical thermal expansion valves. Condenser temperature controls were managed using the EnergyPlus energy management system.

Baseline condenser efficiencies were set to provide a minimum 50 Btu/h/W of fan power for low-temperature condensers and a minimum 75 Btu/h/W of fan power for medium-temperature condensers as recommended by NREL.

## Resources


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