AN ANALYSIS OF ENERGY CONSUMPTION IN GROCERY STORES

IN A HOT AND HUMID CLIMATE

A Dissertation

by

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ABSTRACT

The intent of this study was to investigate the efficient use of energy by developing an energy efficient grocery store combined with cogeneration. This study demonstrated the potential to reduce the energy use in buildings, by implementing a decentralized source of energy generation that allowed for the use of a portion of the energy generated to be shared across building boundaries.

This study considered a high energy use building such as a grocery store to be a part of a residential community, which could potentially participate in the sharing of energy across building boundaries. To better utilize energy resources the study proposed the implementation of a cogeneration facility to supply energy primarily to the store. Surplus energy generated by this cogeneration system was then shared with the requirements of the surrounding residential community. Finally, in order to better account for energy consumption of these buildings both site and source energy was considered. The study focused on hot and humid climates. This study was presented in two parts: Analyzing conventional grocery store systems to determine the maximum savings possible; and examining the option of co-generation systems to provide power to grocery stores and a portion of the community in order to reduce source energy use for the grocery store and a portion of the surrounding community.

Source energy savings were in the range of 47% to 54% depending on the energy efficiency measures selected and the cogeneration configuration determined in the grocery store. Economic payback periods in the range of 4 to 7 years (time until zero net present value) were observed. The selection of appropriate options was narrowed down to two options that utilized more thermal energy within the boundaries of the store and generated more amount of surplus energy to be absorbed by the neighboring residential buildings.

DEDICATION

To my mother and father

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CHAPTER I

INTRODUCTION

1.1 Background

Climate change and the rate of depletion of fossil fuels have currently become major topics of discussion in both academic and political circles worldwide (Kerr 2007, Fletcher 2007). The focus of these discussions is on efficient use of energy to alleviate the current rate at which fossil fuels are being consumed. The building sector accounts for one-fifth of the world's total energy consumption (US EIA 2010). In order to substantially improve the energy efficiency of buildings in the United States, the U.S. Department of Energy has initiated a drive towards reducing energy consumption in buildings US DOE 2011). Currently, improvements to large, internal load dominated buildings are still predominantly accomplished by conventional mechanical environmental controls on a building-by-building basis (Cole 2004). The potential symbiotic relation between such buildings and communities has rarely been explored in previous studies on reducing energy consumption.

1.2 Problem Statement

Approximately 3% of the commercial building energy consumption in the United States is attributable to food sales (US DOE 1986). Although a small percentage, it is still a significant amount of consumption (Cox 1993). Currently, a typical grocery store consumes approximately 52.5 kWh/ ft^2-year of energy, which is almost twice that consumed by a typical office building at 22.5 kWh/ ft^2-year (US EIA 2005) making it clear that energy consumption in grocery stores needs to be further researched for potential areas where it can be reduced.

Energy reduction in grocery stores is currently being performed on a system-by-system basis for individual buildings, considering primarily reductions in site energy use. On using such an approach alone it becomes very challenging to reduce and offset the typically high energy consumption levels of a grocery store. On the other hand, if a grocery store could be considered as part of a larger network of interactions within a community then other opportunities may appear. Furthermore, by shifting the focus from individual buildings to buildings as part of a larger community the potential to explore a symbiotic relationship between the building and the community, which includes residential, commercial, industrial and transportation activities, can be explored particularly in terms of improved energy production and distribution. For example, combined heat and power (CHP) generation or cogeneration[1] has long been used as an alternate to providing electricity and thermal power to industrial applications, large commercial establishments and university campuses. The potential for CHP generation for communities exists and has been explored in climates with high heating demands in form of district heating[2]. However, this potential has not been fully explored for communities in cooling dominated climates due to the inability to utilize the waste heat associated with power production during summer months. A similar concept of combined cooling, heating and power generation (CCHP) or tri-generation[3] has been explored for the operation of grocery stores. In addition, the absorption refrigeration and dehumidification systems in a grocery store provide a year-round thermal sink for waste heat resulting from power generation. Therefore, this study will explore the potential of a grocery store as part of a community and will explore community-based CHP technologies to reduce energy levels for the store as well as the community.

1.3 Purpose and Objective

The intent of this study is to test the proposition that claims that more efficient use of energy resources can be obtained by considering a decentralized approach to electricity generation versus a centralized approach that wastes substantial amounts of energy in the generation as well as transmission of electricity. This wasted energy becomes very evident when considering energy consumption at the source[4] as opposed to considering energy consumption at the site[5]. Therefore, a potential way to reduce the energy use, especially of high energy consuming buildings, is to implement a decentralized source of electricity generation and to better absorb the thermal waste by sharing energy across the building boundaries.

To prove this proposition, the study considers a high energy use building such as a grocery store and considers it to be a part of a residential community which can potentially participate in sharing energy across the store boundaries. To better utilize energy resources the study proposes the implementation of a CHP facility to supply energy primarily to the store.

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[1] CHP can be defined as the production of electrical power and capture of co-existing thermal energy for useful purposes (Caton 2010).
[2] District heating can be defined as th e supply of heat to a number of residential or commercial buildings from a centralized heat source though a network of pipes carrying either hot water or steam (DECC 2012).
[3] CCHP or tri-generation can be defined as the production of electrical power and capture of co-existing thermal energy used for heating and cooling (Hyman and Meckler 2010).
[4] Source energy is the equivalent units of raw fuel consumed to generate that one unit of energy consumed on-site (U.S. EPA n.d).
[5] Site energy is the amount of heat and electricity consumed by a building as reflected in utility bills (U.S. EPA n.d).

Surplus energy generated by this CHP system could then potentially be shared with the needs of the surrounding residential community. Finally, in order to better account for energy consumption of these buildings both as site as well as source energy levels are considered and examined. The study will focus on hot and humid climates.

In order to carry out the study, the study is divided into two parts: Analyzing conventional grocery store systems to determine the maximum savings possible; and examining the option of co-generation systems to power grocery stores and a portion of the community in order to reduce source energy use for the grocery store and a portion of the surrounding community.

1.4 Organization of the Dissertation

The dissertation is organized in nine chapters:

- The first chapter provides a background for the research and defined the problem.
- The second chapter describes the literature review that was conducted by this study. This chapter introduces the reader to conventional as well as alternative energy efficiency strategies and CHP technologies that can potentially be used in the grocery store to reduce energy usage. The review also assesses the tools that can be used to assess the impact of implementing the energy efficiency strategies and CHP technologies.
- The third chapter provides the objective and scope of this study.
- The fourth chapter describes the overall methodology used by this dissertation to address the problem.
- The fifth chapter dwells on the procedure adopted to calibrate the simulation model of the grocery store.
- The sixth chapter examines the energy efficiency measures that were selected to reduce the energy consumption in the grocery store.
- The seventh chapter describes the analysis for the potential CHP systems that can be implemented to provide the energy requirements for the grocery store.
- The eighth chapter assesses the economics of the implementing the options of the CHP system.
- Finally the ninth chapter provides a conclusion to the study, and thoughts about future work.

CHAPTER II

LITERATURE REVIEW

2.1 Introduction

In order to develop the proposed research, a literature review was conducted that was broadly divided into the following relevant categories: 1) Defining a net-zero building ; 2) Functioning of a typical grocery store; 3) Strategies to reduce energy consumption for individual components in a grocery store and the impact of these strategies on whole-building energy consumption; 4) Investigating the potential of cogeneration systems as a method to reduce source energy use in order to approach net-zero energy consumption in both grocery stores and residential buildings; 5) Methods to assess the impact of efficiency measures and the implementation of cogeneration systems on the energy consumption in grocery stores; and 6) Economic analysis of cogeneration systems used in buildings such as grocery stores.

Numerous sources have been reviewed to obtain information for this literature review. These include information from publications by research laboratories such as Oak Ridge National Laboratory (ORNL), Florida Solar Energy Center (FSEC), National Renewable Energy Laboratory (NREL) and Lawrence Berkeley National Laboratory (LBNL); Publications published by the American Society of Heating Refrigeration and Air-conditioning Engineers (ASHRAE) such as the ASHRAE Transactions, the ASHRAE Journal and ASHRAE sponsored research projects; Peer-reviewed journals such as the International Journal of Refrigeration, Applied Energy, Energy and Building; Government agencies such as US Department of Energy; Studies and Reports by the Electric Power Research Institute; Food Marketing Institute; Gas Research Institute; Information from public databases such as the 2003 Commercial Buildings Energy Consumption Survey (CBECS) (US EIA 2005); the US DOE Commercial Building Research Benchmarks for Commercial Buildings (Torcellini et al. 2008) and the Methodology for Modeling Building Energy Performance Across the Commercial Sector (Griffith et al. 2008) have been reviewed to better understand the simulation and energy consumption in a typical grocery store. Information provided by the World Alliance for Decentralized Energy (WADE 2003), Oak Ridge National Lab (2003a, b), US Environment Protection Agency (US EPA 2008b) and the International Energy Agency (IEA) on distributed generation and CHP technologies was also found to be useful to review cogeneration technologies and their applications in commercial buildings.

2.2 Defining a Net-Zero Building

In order to initiate the investigation, it is important to understand the current definitions of the net-zero buildings. A net-zero building can be defined in several ways. The National Renewable Energy Laboratory (NREL) has developed a number of specific definitions for net-zero energy buildings (NZEBs) (Torcellini, et al. 2006), including: 1. *Net-Zero Site Energy* — A building that produces and exports at least as much renewable energy as the total energy it imports and uses in a year, when accounted for at the site[1]. 2. *Net-Zero Source Energy* — A building that produces and exports at least as much renewable energy as the total energy it imports and uses in a year, when accounted for at the source[2]. 3. *Net-Zero Energy Costs* — A building where the amount of money a utility pays the building's owner for the renewable energy the building exports to the grid is at least equal to the amount the owner pays the utility for the energy services and energy used over the year. 4. *Net-Zero Energy Emissions* — A building that produces and exports at least as much emissions-free renewable energy as it imports and uses from emission-producing energy sources annually.

From the above definitions, it can be implied that in order to attain the net-zero status it is necessary for the building to generate on-site renewable energy. Currently, the concept net-zero energy consumption has been demonstrated in residential and some commercial buildings (US DOE 2011) by offsetting the energy requirements with energy efficiency measures and renewable on-site energy resources. However, in the case of a grocery store with a typical energy consumption[3] of 250 kBtu/ft^2-yr versus less than 100 kBtu/ft^2-yr as reported for office buildings (Griffith et al., 2008), the viability of such an approach may not work.

2.3 Functioning of a Typical Grocery Store

Baxter (2003), in a review of grocery stores, identifies the typical store sizes in the U.S. to range from 10,000 ft^2 to 100,000 ft^2 The report also identifies the typical grocery store electric power requirements to range from 30 kW to 400kW and the typical annual energy

[1] “Site energy” refers to the energy being consumed within the boundaries of the building site.
[2] "Source energy" refers to the primary energy required to generate and deliver the energy to the site (i.e., coal, oil and natural-gas) (Torcellini et al. 2006). Therefore, to calculate a building's total source energy, imported and exported energy is multiplied by the appropriate site-to-source conversion multipliers.
[3] Reported as total site energy consumption.

consumption to ranges from 100,000 kWh/yr to 1.5 million kWh/yr. Several interdependent systems can be identified in a typical grocery store. These systems include: the refrigeration, the HVAC, the building envelope and the lighting system (Cox 1993; Leach et al. 2009). Over the years a range of activities have been included as part of the grocery store often placing contradictory demands on these systems. The national average for the energy end-use of a typical grocery store is approximately 39% for refrigeration, 23% for lighting, 11% for cooling, 13% for heating, 5% for cooking and 9% for other uses (US EPA 2010). This indicates that refrigeration, cooling, and lighting represent almost 73 % of the grocery stores energy use with the refrigeration load representing the single largest electric load. Clearly, keeping a continuous stream of products at near-uniform cold temperatures consumes substantial energy and makes it extremely challenging to achieve net-zero energy consumption in a grocery store.

Grocery stores in hot and humid climates present an added challenge. This is because space humidity levels need to be controlled not only because it affects indoor air quality and comfort, but it also impacts the energy use of the refrigeration systems (Henderson 1996; Stoecker 1998). Therefore, this study explores grocery stores operating in hot and humid climates with a goal of reducing the total store energy use. The study will proceed to identifying energy efficiency measures for each of the above mentioned categories using a whole-building grocery store simulation analysis in order to select the best set of viable options.

2.4 Efficiency Measures for the Grocery Store

Several sources were reviewed to compile a list of energy efficiency measures for the grocery store. These include ASHRAE's Advanced Energy Design Guides (AEDGs) for different building types and their related technical support documents; Standards such as ASHRAE Standard 90.1[4], ASHRAE Standard 189.1, the International Green Council Code (IGCC), California Energy Commission's (CECs) Title-24 and related technical support documents; Websites of product rating councils such as the Cool Roof Rating Council (CRRC), the National Fenestration Rating Council (NFRC) and the Air-conditioning, Heating and Refrigeration Institute (AHRI); and several individual research documents which will be cited appropriately in the sections that follow. Efficiency measures were divided into the following categories:

- Envelope,

[4] ASHRAE Standard 90.1-1989,1999, 2001, 2004, 2007 and 2010.

- Lighting and daylighting,
- Heating ventilation and air conditioning systems,
- Service hot water systems, and
- Refrigeration systems.

2.4.1 Efficiency Measures for the Building Envelope

Grocery stores are usually considered to be 'internal load dominated' buildings where the energy consumption is primarily driven by process loads, which are mainly imposed by the refrigeration system, lighting loads, and the resultant loads from heating and cooling systems. Common construction practices dictate that a compact layout of a store be used in order to minimize the layout configuration for refrigeration pipelines and HVAC ductwork, if any. Variations in the building envelope components such as the building shape, orientation and window shading, and thermal properties of the opaque and glazing portions of the envelope potentially have far less impact on the annual energy consumption of a typical store than variations in the refrigeration or HVAC systems.

2.4.1.1 Improved Insulation

Typical grocery store construction consists of mass walls, opaque doors in loading zones, slab-on-grade floors and roofs with insulation entirely above the structural deck. ASHRAE Standard-90.1[5] sets the minimum requirements for exterior walls, opaque doors, roofs and slab insulation depending on the climate zone in which the building is located. Improved insulation values for mass walls, opaque doors, roofs with insulation entirely above the deck and slab-on-grade floors can be found in the ASHRAE AEDGs for achieving 50% above ASHRAE Standard-90.1[6] requirements for retail buildings (AEDG 2011).

Insulation requirements for coolers, freezers and food preparation rooms specific to grocery stores are also important. Types of insulation materials that are used for low-temperature spaces include rigid insulation such as polystyrene, polyisocyanurate, polyurethane and phenolic materials; panel insulation, which includes prefabricated panels constructed from rigid insulation, foam-in-place insulation; and precast insulation concrete panes (Becker and Fricke 2005). Other insulation materials that are not so frequently used include: mineral rock fiber,

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[5] All the ASHRAE 90.1 Standards (i.e. 1989, 1999, 2001, 2004, 2007 and 2010) have provisions for opaque building envelope specifications.

[6] The corresponding year of the ASHRAE Standard 90.1 cited in the AEGD depends on the year in which the AEDG was published.

cellular glass and glass fiber insulation (Becker and Fricke 2005). Characteristics of the popular board insulation, which includes R-value specifications and the relative cost data, can be found in the industrial refrigeration handbook authored by Stoecker (1998). Improved specifications for these components can be found in the ASHRAE Handbook for Refrigeration (ASHRAE 2006)[7] and work published by Becker and Fricke (2005), which discusses design essentials for refrigerated storage facilities.

2.4.1.2 Cool Roofs

Based on CBECS (2007) data and observations of a case-study store, grocery stores are typically constructed with built-up insulated roofs, concrete or concrete block exterior walls, and slab-on-grade floors. Most grocery stores are single-story buildings with large footprints. Having an almost 1:1 ratio of roof area to total facility square footage makes the grocery stores good candidates for cool-roof solutions in cooling dominated climates (US EPA 2008a).

White color or some other highly reflective color when applied to the roof surface can minimize the amount of heat that the roof absorbs. In a cooling load dominated climate this change can reduce peak cooling demand and total cooling energy use, depending on climate zone in which the building is located (US EPA 2008a). Konopacki and Akbari (2001) determined a summertime daily measured space cooling energy savings of 11% and average space cooling demand reduction of 14% by installing a reflective roof membrane at a large retail store in Austin, Texas. New roof coating products such as Cool Roof Colored Materials (CRCMs) made of complex inorganic color pigments are currently being researched. The Oak Ridge National Lab and the heat island group at Lawrence Berkeley National Lab are currently investigating the performance of CRCMs under laboratory conditions and in the field (Akbari and Rainer 2000; Akbari and Konopaki 2004; Akbari and Levison 2008; Levison and Akbari 2010a; Levison et al. 2010b).

More recently, ASHRAE Standard 90.1-2010 has made it mandatory for building roofs in cooling dominated climate zones 1 through 3 to implement cool roofs with a minimum three-year-aged solar reflectance of 0.55 and a minimum three-year-aged-thermal emittance of 0.75. A list of rated cool roof products can be found on the website of the Cool Roof Rating Council (CRRC 2012).

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[7] A discussion on insulation materials used for low-temperature spaces in grocery stores is also included in the more recent editions of the ASHRAE Handbook of Refrigeration (i.e. ASHRAE 2010c).

2.4.1.3 Skylights

The introduction of skylights in grocery stores situated in cooling dominated climates can effectively be implemented to reduce the energy consumption due to electric lighting (Liu et al. 2007) which is discussed in another section of this chapter. Increasing the area of skylights as an independent efficiency measure may not be considered an effective strategy, although a reduction in heating energy due to increase in passive solar gains is a possibility.

ASHRAE Standard 90.1 sets the minimum specifications for skylights depending on the climate zone in which the building is located. The NFRC (NFRC 2012) provides a list of certified fenestration products available in the market that meet or exceed the U-value, SHGC and transmittance (thermal and visible) specifications as set by the ASHRAE Standard 90.1-2010.

2.4.1.4 Building Infiltration

Infiltration can be categorized into three components (Liu et al., 2007) which includes general infiltration through the building envelope cracks; air leakage from relief dampers (when not in operation); and infiltration through the loading dock doors and entrance doors. The high traffic of customers and products entering and leaving a grocery store throughout the day makes infiltration an important issue to address in designing energy efficient envelopes for grocery stores.

Reduced infiltration can be attained by the installation of an air barrier and vestibules (Hale et al., 2008, ASHRAE 2010a), reducing air leakage through air dampers (Liu et al., 2007), improving the leakage area across the dock doors and around the dock doors, and by adding air curtains (Berner 2008).

An air barrier is a continuous membrane over the entire building enclosure that is impermeable to air-flow (US EPA 2008a). Properly installed air barriers can reduce air leakage in a building due to envelope cracks by up to 83% (Woods 2006). Air barriers can also double up as vapor barriers to provide addition protection for the building against moisture penetration through the building envelope (ASHRAE 2006). Vestibules or revolving doors are often considered as design measures to decrease air infiltration through high traffic doors (Cho et al. 2010). The ASHRAE 90.1 codes[8] require vestibules to be installed in certain situations. Another

[8] All the ASHRAE 90.1 Standards (i.e. 1989, 1999, 2001, 2004, 2007 and 2012) have provisions for vestibules. However, specifications for vestibules in these codes have become more stringent over the years.

measure to reduce infiltration through doors is the air curtain system which is essentially a jet of air from a fan that creates an air barrier from the vertical wall of pressurized air. Usually such units are located over the main entrances to the building and at loading docks. The use of air curtains can be especially beneficial when used in combination with either vestibules or physical doors for high traffic areas such as the main entrance of stores or loading docks (Berner 2008).

Infiltration can also play a significant role in low temperature spaces such as the product cooler and the preparation room where a constant interaction takes place with the ambient store conditions via open service windows and pedestrian traffic to and from the spaces. Infiltration is the largest source of humidity in these spaces, which translates into more frost over refrigeration coils (Jekel 2001). Most of the infiltration takes place through operable doorways. Efficiency measures for freezer doorways such as strip curtains, air curtains and fast sliding doors can be implemented to curtail infiltration to a great extent (Downing and Meffert 1993, SCE 2012). This translates into reduced refrigeration loads resulting in reduced compressor run times and finally reduced energy consumption.

2.4.2 Efficiency Measures for the Lighting Systems and Equipment

Lighting represents approximately 23% of annual energy use in a typical grocery store and provides a good opportunity to improve efficiency, while increasing the quality and productivity in most facilities (US EPA 2008a). Lighting energy can generally be reduced by up to 30% to 50% by installing more efficient lighting fixtures, improved lighting controls and taking advantage of daylight where available (US EPA 2008a). In addition to the lighting electricity savings, saving of 10% to 20% of the cooling energy can also be saved by implementing efficient lighting technologies (US EPA 2008a), although efficient lighting can impose a penalty on the store space heating, which is needed to replace the heat gain from the inefficient lights. Paybacks for lighting projects are usually within a few years because of the long hours of operation, much of it during the utility’s peak cooling periods (Fedrizzi and Rogers 2002). In addition, the implementation of efficient lighting technologies and digital controls has proven to improve demand responsiveness in including grocery stores (Rubinstein and Kiliccote 2007).

2.4.2.1 Lighting Standards

The quality of lighting is an important aspect to be considered in the store to allow the customer to examine features and qualities of the merchandise, while minimizing glare and avoiding large differences in brightness. Lighting qualities include luminance and illuminance, diffusion, uniformity, chromaticity and color rendering index (CRI) (Grondzik et al. 2010). IESNA standard requirements for illuminance levels in grocery stores, as quoted in Grondzik et al. (2010), are in the range of 90 fc (1,000 lux) for general merchandise areas for grocery stores. The requirements are not uniform for all the areas in the grocery store. Loading areas require lower values of 14 fc (150 lux) and storage requires a value of 9 fc (100 lux).

Specifications are also set for the energy consumption of lighting systems. In ASHRAE Standard 90.1- 2007, (ASHRAE 2007) the lighting requirements are presented in terms of lighting power density (LPD) specifications with 1.5 W/sq. ft. as the requirements for retail establishments when considering the building area method for calculations. LPD requirements are further reduced to 1.4 W/sqft. in ASHRAE Standard 90.1- 2010. The codes also require the use of time switches and occupancy sensors in buildings. Almost all the ASHRAE 90.1 standards[9] indirectly require the use of time switches and occupancy sensors in buildings to meet the ever decreasing LPD.

2.4.2.2 Lighting Technologies

There are essentially four types of lighting systems to be considered in a grocery store, which include: high bay lighting, ceiling height or aisle lighting, display-case lighting and product spot lighting. Display case lighting will be covered separately under the section focusing on efficiency measures for refrigerated display cases. High bay lighting typically uses High Intensity Discharge (HID) fixtures or florescent lighting (Knowhow -Highbay Industrial Lighting, 2000). Fluorescent lighting has also been the favored choice for ceiling height lighting (Theobald, 2007). Product spot lighting is obtained by using incandescent, tungsten-halogen lamps or more recently Light Emitting Diodes (LEDs).

The diffuse characteristics of fluorescent lighting makes it suitable for illuminating or washing large areas such as the ceiling planes and aisles where products are displayed (Grondzik et al. 2010) and hence are widely used for ambient lighting in grocery stores. T8 lamps are currently the standard used in stores for ceiling lighting. T5 technology offers twice the lumen output in the same length as its T8 counterpart with a lamp efficacy that is attractive in meeting project energy goals (Grondzik et al. 2010). However, T5 lamps can only be used with the newer, more expensive electronic ballasts (Rea 2000) making their installation more expensive

[9] ASHRAE 90.1 Standards include 1989, 1999, 2001, 2004 , 2007, and 2010.

than T8 lamps. High Output (HO) alternatives of T5 lamps are also available. T5 HO lamps are used where high output is required from limited-size source, which is typical for grocery stores. T5 HO lamps can also be used in cold environments that prevent proper operation of standard output 430 mA lamps. However, these lamps have lower luminous efficacy than standard output 430 mA lamps (Grondzik et al. 2010). According to the Energy Star Building Manual, for storewide ambient lighting either T5 lamps or high-performance T8 lamps, can reduce lighting energy consumption by 35 % or more when compared to T12 lighting (US EPA 2008a).

Magnetic ballasts are generally used for T8 or T12 fluorescent lamps. However, electronic ballasts are more efficient than magnetic ballasts having less power loss. The use of electronic ballasts also increases the lamp efficacy by approximately 10% to 25% (Grondzik 2010) when compared to the use of magnetic ballasts. Finally, the ballasts are lighter, more energy efficient, generate less heat, no flicker and are virtually silent (Grondzik 2010).

Hemispherical metal halide fixtures are at times used for ambient lighting in grocery stores. Metal halide lamps are part of the HID group of lamps[10]. These lamps are characterized by high efficacies[11]. However, metal halide lamps have poor color rendering capabilities (Grondzik 2010). In addition, metal halide lamps are not instant-starting, requiring approximately 2 to 3 minutes on initial startup and 8 to 10 minutes for restrike. Ceramic metal halide lamps are an improvement over metal halide lamps and are currently the industry standard offering a high color rendition index of 80 to 90, a color temperature of 3000 K or 4100 K, improved lumen maintenance, and stable color consistency (Grondzik 2010).

Grocery stores can also save energy by reducing ambient lighting levels and using product spotlighting on product displays (US EPA 2008a). Product spotlighting can be accomplished with incandescent, compact fluorescent, tungsten halogen lamps or LED lamps and spot reflectors to direct the light on to the product. Characteristics of tungsten halogen lamps include longer life and lower lumen depreciation than incandescent lamps, excellent color rendering characteristics (CRI>100) and a smaller envelope for a given wattage due to the high temperature requirement of the halogen cycle (Grondzik 2010). As a result the lamp is effectively a point source. Tungsten halogen lamps have found wide acceptance in all types of

[10] Lamps in this category include mercury vapor, metal halide and high-pressure sodium.
[11] Typical luminous efficacies of halide lamps are in the range of 50 – 90 lm/W, depending on the wattage and lamp type (EERE, 2009). These efficacies are comparable to those of linear fluorescent lamps.

display and accent lighting applications and are used as spotlighting to accent products or areas of the store. However, these lamps are not very efficient.

The induction lamp is one of the latest developments in lighting technology which can be effectively implemented in grocery stores. The extraordinarily long life of 100,000 hours along with a 70% lumen maintenance at 60,000 hours and instantaneous restrike time make the induction lamp technology a good choice in areas where relamping and maintenance are difficult or hazardous such as in high-ceiling portions of the store and parking areas (Grondzik 2010) . In addition, they offer compact construction, and vibration resistance. However, limited availability in higher wattages, unsuitability for use with dimmers and high first costs are factors that deter the widespread application of this technology (Induction Lamps 2010).

In recent years, electronic ballasts for fluorescent lamps have been developed that allow for dimming capabilities in addition to improving the efficiency of the lamp. Dimming can be used in conjunction with skylights throughout the store and can be installed with motion sensors or daylighting sensors in order to optimize control lighting. Recently, dimming ballasts have been developed for HID lighting, allowing them to be dimmed up to 50% (Grondzik 2010). However, dimming or reduced output operation of HIDs is usually not recommended because of the very noticeable color shift to a lower CRI that occurs when the lamp is dimmed which is not good for representing colors (Grondzik 2010). Technologies are also available for dimming tungsten halogen lamps (Grondzik 2010).

2.4.2.3 Daylighting

With an almost 1:1 roof-to-floor area ratio and high ceilings grocery stores lend themselves ideally to skylights to offset space lighting requirements during daylight hours. A cross-sectional field study by the Heschong Mahone Group (1999) statistically demonstrated that diffusing sky-lights can improve retail sales by 40 % compared to retail stores without daylight. Several grocery store chains such as Walmart, Target, and Bighorn Home Improvement are incorporating skylights in their typical new store layouts. On the other hand, introducing daylighting in stores imposes a penalty on the store space cooling and heating due to incoming solar radiation, reduced U-values of the skylights, and can pose a security risk.

2.4.2.4 Lighting Control Technologies

Lighting control systems play an important role in the reduction of energy consumption of lighting without impeding the comfort levels in the space (Halonen et al. 2010). Suitable lighting control strategies include predicted control strategies, real occupancy control strategies, constant illuminance control strategies and daylighting harvesting control strategies (Halonen et al. 2010). Predicted control strategies are used to control the lights on a preset daily time schedule. Another type of predicted control strategy – dusk or dawn control strategy controls the lights so that they are switched on automatically when it gets dark outside. The basic components include implementation of a scheduler, a timeclock, switches and dimmers. 10% to 20% in lighting energy savings have been observed from implementing thess strategies (Halonen et al. 2010). Real occupancy control strategies on the other hand limit the operating time of the lighting system based on the actual occupancy of a space. The basic components include implementation of occupancy sensors, switches and dimmers. 20% to 50% in lighting energy savings are observed on the implementation of this strategy (Halonen et al. 2010). The constant illuminance control strategy employs a photocell to measure the lighting levels and in turn control the lumen output of the light sources when ambient conditions are appropriate. The basic components include the implementation of a photosensor and a dimmer. In general, 5% to 15% in lighting energy savings are observed from the implementation of this strategy (Halonen et al. 2010). Finally, a daylight harvesting control strategy reduces the lighting energy consumption by incorporating daylight to maintain the required lighting levels in a space. This strategy also implements a photocell to measure the lighting levels in the space and to control the lumen output of electric lighting to maintain required lighting levels. Basic components include the implementation of photosensors, dimmers and switches with multi-level lighting. The potential savings vary from 20% when implementing daylight harvesting alone to 50% saving in lighting energy consumption when implemented in conjunction with real occupancy control strategies (Halonen et al. 2010).

2.4.2.5 Exterior Lighting Technologies

Parking light technologies that offer energy savings over mercury vapor lamps include high intensity discharge (HID) light sources such as metal halide (MH) or high pressure sodium (HPS) lamps (US EPA 2008a). Recently, Light Emitting Diode (LED) lights have been developed that can provide significant energy savings over conventional strategies (PG&E 2009). Other advantages include improved directionality and improved distribution of lighting levels, better color rendition, longer lamp life and instant-on capability (US DOE 2008a). However this technology is still very expensive when compared to the conventional technologies. Bi-level controls and motion sensors can also be implemented to control exterior lighting systems. Bi-level controls allow reduction in lighting levels in parking lots when not in use (US DOE 2009b).

2.4.2.6 Equipment Power Density

Miscellaneous equipment loads accounts for approximately 6% of the total energy consumption of the grocery store[12] (US EPA 2008a). These include loads from cooking equipment computer and cash register operation, cleaning equipment, photo and pharmaceutical equipment, stand-alone vending machines and gondola receptacles, compactors and cardboard balers.

Cooking equipment consumes 13% of the natural gas purchased by grocery stores (US EPA 2008a). ENERGY STAR qualified cooking equipment can save 10 to 50% of energy consumed than conventional models (US EPA 2008a). Reductions in operation time of the cooking equipment when not in use can cut the cooking related energy consumption by 60 percent (EPA 2008a). Proper maintenance and strategic placement of equipment can also play an important role in reducing energy costs. Some relevant ENERGYSTAR products for grocery stores include commercial fryers, steam coolers and hot-food holding cabinets, commercial dishwashers, commercial icemakers, computers and monitors and vending machines.

In another study on reducing energy consumption in grocery stores by Leach et al. (2009), the authors describe the lack of credible sources to assess the implementation and performance of plug loads in grocery stores. Thornton et al. (2010) in their report on energy reduction in small office buildings discuss that schedules can be modified using power management software particularly at the network level, occupancy sensor controls of computer monitors and other equipment, vending misers, and time switches for coffee makers and water coolers. The authors conclude that an additional 20% of total plug loads can be reduced by the implementation of additional controls.

2.4.3 Efficiency Measures for the Heating Ventilation and Air-Conditioning Systems

According to an assessment by the US EPA (2008a) approximately 23% of the energy consumption in grocery stores is from HVAC systems. HVAC systems perform the task of maintaining indoor air quality, reducing the latent load on refrigeration and maintaining the temperature and humidity control for thermal comfort (Henderson et al. 1996). Moreover,

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[12] The energy consumption for miscellaneous equipment reported in this review includes 2% of energy consumption for cooking activities.

grocery stores are unique in that indoor humidity levels are generally maintained at or below a set point to ensure optimum operation of refrigerated cases (Henderson et al. 1996). The sub-sections below describe the issues that have to be considered for designing and optimum HVAC system for the grocery store. These include addressing issues for humidity control, heat recovery technologies, and various alternates to the constant volume system such as packaged variable air volume system (PVAVS), dedicated outdoor air system (DOAS) and demand controlled ventilation (DCV) operation.

2.4.3.1 Humidity Control

Humidity control is strongly related with ventilation requirements set by ASHRAE Standard 62.1-2004 for buildings including grocery stores. A study by Henderson et al. (1996) simulated several conventional and alternative space-conditioning systems in order to evaluate optimal humidity levels that resulted in the lowest annual operating costs. The study considered conventional, single-path system; single-path with low supply air flow; single-path with heat pipes (HPs); dual-path system; dual-path with heat pipes (HPs) ; thermal energy storage(TES), dual-path (44°F Chilled Water Temperature CWT); thermal energy storage, dual-path (37°F CWT); thermal energy storage, dual-path with HPs (44°F CWT); thermal energy storage, dual-path with HPs (37°F CWT); gas-fired desiccant system (silica gel) and single-path with demand ventilation that are CO_2-controlled.The technologies were also simulated to assess their ability to mitigate the impact of fresh air requirements set by ASHRAE Standard 62 - 1981.

The results generally showed that the more complex systems that use desiccants, heat pipes, or TES yield the largest reduction in annual operating costs. Generally, the optimal humidity set point determined for each system was higher than has typically been assumed in the industry. Humidity set points of 50% to 55% provided the lowest annual operating costs and life-cycle costs for most systems. The optimal humidity set points are higher because more realistic trends of anti-sweat heater performance as a function of store humidity were assumed in this study based, in part, on results from field tests. The realistic anti-sweat heater control scenario involves the use of partial cycling of anti-sweat heaters rather than the on-off cycling of the heaters as assumed by more conservative studies. These more realistic assumptions for refrigerated display-case performance reduced the benefits of maintaining lower store humidity. This, in turn, reduced the efficacy of improved dehumidification technologies such as heat pipes, desiccants, and TES. The study concluded that the demand-controlled ventilation system was generally the most cost-effective of all the systems that were studied. While this system did not have the lowest operating costs, the relatively low installed costs made it the most cost-effective alternative (i.e., highest life-cycle savings). The dual- path system was also relatively cost effective compared to the other systems since it had a relatively low operating costs as well as low installed costs. The desiccant system had the lowest operating costs, yet was not cost effective due to the higher installed costs and higher maintenance costs.

Controlling the ambient temperature and humidity in the grocery store plays a key role in the energy consumption of open faced display-cases (Howell 1993a, 1993b; Tassou et al. 1999; Henderson et al. 1999). The effect of humidity was found to be more pronounced than that of temperature in hot and humid climates (Tassou et al. 1999). Howell (1993a,b) observed that a 5% reduction in store relative humidity resulted in a total store energy reduction of nearly 5% as a result of more efficient operation of open faced display-cases. Other studies conducted by Henderson, Khattar and Faramarzi made similar observations (as cited in Kosar et al. 2005). Hence, controlling the ambient humidity in a grocery store can lead to lower humidity levels in the operation of open faced display-cases, which in turn leads to lower energy consumption and lower operating costs for the grocery store.

Finally, it was demonstrated by the initial results in ASHRAE Research Project 1467 (Brandemuehl 2010) that different layouts of refrigeration system components such as display cases, freezers and coolers have an impact on optimizing the design and operation of the combined HVAC and refrigeration systems. The preliminary results of the research project suggested that maintaining lower humidity levels around refrigerated cases reduced refrigeration system energy use without requiring dehumidifying the entire grocery store. However, it should be noted that the simulation program utilized in this analysis is not sensitive to the variations in layout of the components of the refrigeration system. Hence, this aspect of grocery store design was not explored by this study.

2.4.3.2 Heat Recovery Technologies

In many grocery stores integrating HVAC systems with refrigeration systems allows the high temperatures discarded by the refrigeration systems to be used as a heat source for space heating and in some cases to supplement service water heating (Baxter 2003). Several studies assessed the impact of the implementation of heat recovery technologies in grocery stores and several recommendations have been proposed to improve the utilization of waste heat from grocery store refrigeration systems. These include studies by Baxter (2003), Arias and Lundqvist

(2006), Genest and Minea (2006), Minea (2006), Cecchinato et al. (2010), Minea (2010), Sawalha and Cheng (2010) and Fricke (2011).

In the IEA Annex 26 report on advanced grocery store refrigeration heat recovery systems Baxter (2003) discusses various approaches that can be used for heat recovery in grocery stores. Heat recovery strategies include the use of heat pumps; direct heat recovery from the compressed refrigerant for space and service hot water heating purposes; and combined cooling heating and power (CCHP) systems that integrate electric power production for the store with refrigeration and HVAC systems. Baxter noted that heat-pump-based heat recovery does not require the refrigeration system condensing temperature to be maintained at artificially high levels to facilitate heat recovery. The study observed savings of over 10% of annual energy consumption depending on the system used. Baxter also noted that when considering heat recovery from refrigerant the amount of waste heat that is effectively applied to the space and water heat requirements at a given site depended upon the size of the coincident refrigeration load and the refrigeration/HVAC control system's ability to effectively manage the heat recovery process. Recovery of refrigeration system rejected heat was shown to be able to provide from 40% to 100% of the space and water heating needs depending on the climate for the test stores examined in this Annex.

Similar results have been reported in the study by Arias and Lundquist (2006) for heating dominated grocery stores in Sweden. The study shows that heating requirements can be covered completely by heat reclaim from the condenser. The authors conclude that the highest potential of energy saving was achieved from using a system with both heat recovery and floating condenser head refrigeration systems. However, the authors observe that practical installations are less efficient due to poor system design solutions and / or control strategies. Hence, to realize the maximum benefits of installing the heat recovery system, the system must be effectively designed and integrated into the HVAC system using sophisticated control systems.

Minea (2010) in a study on heat pumps for energy recovery in grocery store refrigeration systems compares the basic heat recovery method in direct-expansion grocery store refrigeration systems with other concepts such as two-stage systems, refrigeration systems with warm heat rejection loops, totally secondary fluid systems and CO_2 refrigeration systems. The study concludes that waste recovery with heat pumps is potentially an efficient strategy.

Fricke (2011), in a report on waste heat recapture from supermarket refrigeration systems, presented a number of existing and advanced waste heat recovery systems for grocery stores. These include heat reclaimed by de-superheating[13] the compressor discharge, heat recovered by partial or full condensing of the refrigerant, and the use of heat pumps in the heat recovery process. The report concluded that the implementation of simple heat recovery systems to either space heating or process water can save up to 50% of heating energy for typical grocery stores. On the other hand, when using heat pumps for heat recovery up to 100% of fuel for heating can be saved. In addition, to provide greater savings, the refrigeration system may employ a floating head pressure[14] strategy without negatively affecting the ability of heat pumps to recover useful waste heat from the refrigeration system.

Finally, Sawalha and Cheng (2010) present a number of heat recovery system solutions for grocery stores. These include fixed head pressure heat recovery, de-superheater, heat pump cascade, heat pump cascade for subcooling. Information regarding the de-superheater as is presented in Sawalha and Cheng (2010) is assessed by this analysis. A schematic diagram presenting the working of a de-superheater is presented in Figure 2-1.

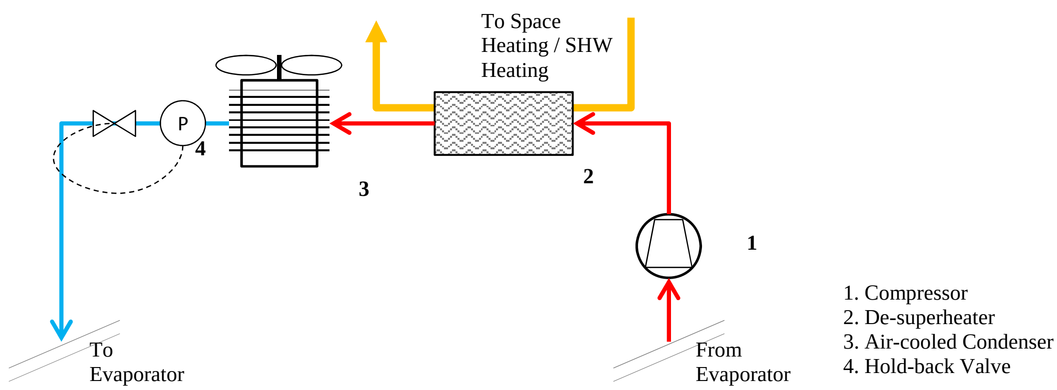


Figure 2-1: Schematic Diagram of the Heat Recovery at the De-Superheater (Source: Sawalha and Cheng 2010)

[13] A de-superheater is a device that is used to cool the superheated refrigerant gas coming from compressors before it enters the air-cooled condenser. The heat from the gases rejected to the de-superheater can then be used for space heating and service water heating purposes.

[14] Floating head pressure is used as a control strategy for the condensers of a refrigeration system. In this strategy, the pressure and hence the corresponding temperature of the condenser is allowed to float according to the ambient temperature conditions, thus saving compressor energy that is required to maintain a fixed head pressure at all times to operate the condenser. This concept is discussed later, in the section on refrigeration of this literature review.

2.4.3.3 Packaged Variable Air Volume Systems

Packaged variable air volume systems (PVAVs) are also a viable alternative to packaged single zone systems which are constant volume systems for commercial buildings. Several manuals and reports examining the various aspects of the performance of the PVAV system were considered by this study. These include: Winkelman et al. (1993) and Hirsch (2006) for the basic system operation; Thornton et al. (2010), Liu et al. (2007), and the Advanced Energy Design Guide for Retail (AEDG 2011) for energy savings obtained; and Wei et al. (2000) for optimizing the performance of the PVAV system.

The basic configuration of the PVAV system consists of a compressor, an air-cooled condenser, an evaporator, a baseboard heater or reheat heater, a filter, variable volume control boxes and a thermostat (Winkelman et al. 1993). A schematic diagram presenting the working of a de-superheater is presented in Figure 2-2.

An important benefit of PVAV systems is the potential for reduced operating costs at part load conditions. The lower operating costs result from supply fans having to move less air and thus requiring less fan horsepower. Lower operating costs are also from the elimination of mixing hot and cold air streams at off-peak conditions to meet space loads.

The air flow in PVAV systems is controlled by either installing inlet vanes or discharge dampers which control the air flow by change in position, or by installing variable speed drives (VSDs) on fan motors which regulate the speed of the fan motors (Hirsch 2006). However, recent applications of PVAVs have seen a consistent use of VSDs for fan motors.

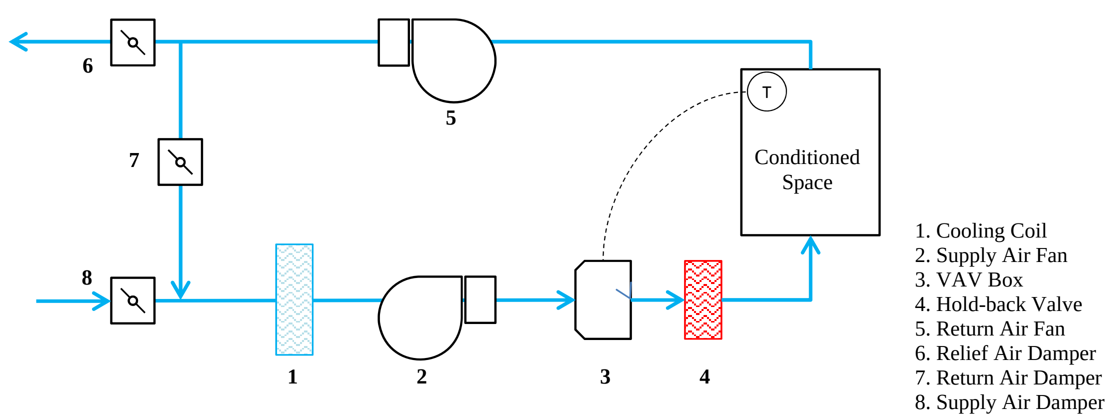


Figure 2-2: Schematic Diagram of the Packaged Variable Air Volume (Constant Temperature) System (Source: Birdsall et al., 1994)

Thornton et al. (2010) in a study of energy efficient measures for small office buildings report that for certain climate zones the PVAV option for HVAC systems can achieve close to 50% energy savings. The authors also report that installing a PVAV system may also be a better choice than installing heat pumps in terms of initial costs, and maintenance. PVAV systems can be used to serve either a single zone or multiple zones. The use of single zone units is recommended for a more effective control of large open sales floors (AEDG 2011).

To optimize the performance of PVAV system and ensure no overcooling of spaces, various strategies of controlling supply air temperature can be implemented. The strategies include reset controls that reset the supply air temperature either with the outdoor temperature conditions or with space loads (Winkelman et al. 1993). Significant energy savings can be seen by installing appropriate reset schedules for supply air temperature in single duct variable air volume systems (Wei et al. 2000).

2.4.3.4 Dedicated Outside Air Systems

A general overview for Dedicated Outdoor Air Systems (DOASs) is obtained from Mumma and Shank (2001), Thornton et al. (2010) discusses the benefits of implementing DOAS, different configurations of DOAS were discussed by Mumma and Shank (2001) and Emmerich and McDowell (2005). A schematic representation of the DOAS system is presented in Figure 2-3.

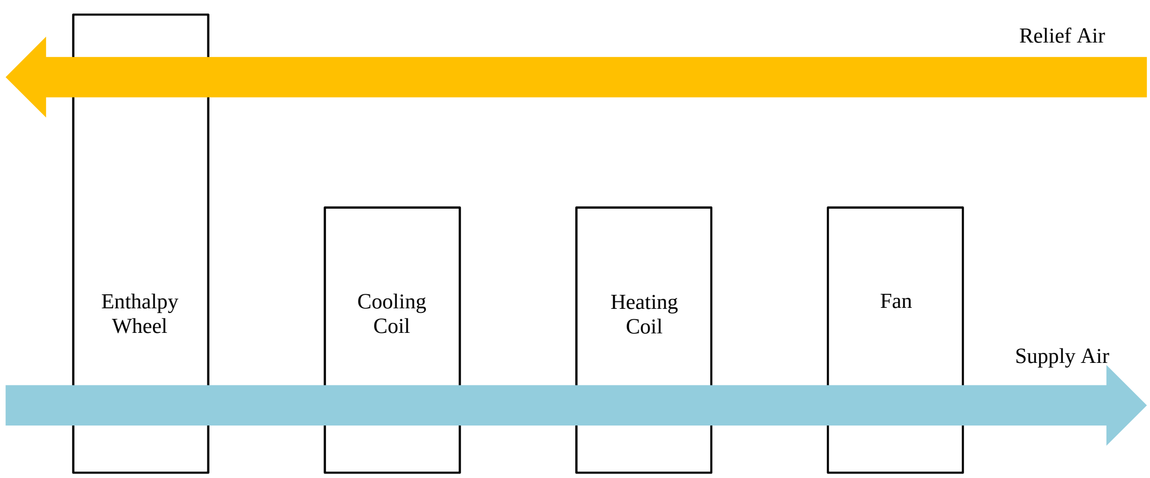


Figure 2-3: DOAS with Enthalpy Wheel, Conventional Cooling Coil and Heating Coil (Source: Thornton et al. 2010)

DOASs separate the delivery of ventilation air from air streams used to meet space conditions. The system is designed to remove the latent loads from ventilation air as well as all the latent loads associated with the space (Mumma and Shank 2001). This is done by setting the supply air dewpoint temperature lower than the room air dewpoint temperature which is typically 45°F to 52°F. Terminal equipment operating in parallel with the DOAS system is then required to remove only the sensible loads that remain after the dry ventilation air has been introduced into the space (Shank and Mumma 2001). Maximizing the savings from a DOAS, requires the need for an energy recovery system, modulation of supply air temperature and improved control strategies.

Thornton et al. (2010) showed that DOASs provide a number of benefits when compared to systems that mix the outdoor air with return air for each rooftop unit, including:

- The DOASs with a centralized location of the outdoor air intake, which requires only one energy recovery ventilator (ERV) to pretreat all the outside air. This can reduce the first cost of the system.
- Meeting the loads from outside air with DOAS allows the individual systems that the DOAS serves to be downsized.
- Meeting the ventilation loads from a central source means that only the DOAS fan needs to be run continuously during unoccupied hours. The zonal fans are then allowed to cycle when meeting the zone heating and cooling loads.

Different configurations of DOAS are available. Six arrangements were compared by Mumma and Shank (2001) and arranged in terms of their energy performance. The authors concluded that the best performing DOAS consists of a preheat coil, an enthalpy wheel, a deep cooling coil and a sensible heat exchanger. In another study comparing different configurations of DOASs, Emmerich and McDowell (2005) investigated the energy performance of two DOAS configurations for a building with water source heat pump system for different climate zones in the U.S. One DOAS simply consists of a preheat coil and an enthalpy wheel, while the other DOAS has a preheat coil, an enthalpy wheel, a deep cooling coil, a sensible wheel and a fan. They concluded that although the more complex arrangement of the DOAS system performed better than the simple arrangement, the improvement was in the range of 1% - 7%.

2.4.3.5 Demand Control Ventilation

For the Demand Control Ventilation (DCV) strategies publications from the Federal Energy Management Program (FEMP) (US DOE 2004a) and US EPA (2008a) were reviewed to provide an overview. The ASHRAE Standard 62.1 requires that a certain amount of minimum fresh air should be provided in occupied spaces in order to maintain adequate air quality. In order to comply with this standard, fixed ventilation rates are usually implemented, which are based on peak occupancies. This often results in more fresh air than necessary coming into the building, which results in increased sensible as well as latent loads. DCV strategies using CO_2 sensing technology regulates the amount of ventilation air admitted in a conditioned space based on occupancy of the space as measured by the CO_2 sensor. This is because the CO_2 production in space closely tracks occupancy. Indoor CO_2 measurement can be used to measure and control the amount of outside air, which is typically at a lower CO_2 concentration than the indoor air (US DOE 2004a). This is done whenever the CO_2 levels in space reaches a predetermined level that represents a differential between the indoor and outdoor CO_2 levels. Potentially large savings can occur with CO_2-based DCV technologies in spaces where there is a high fluctuation of occupancy levels. The benefits of DCV systems include avoiding heating, cooling and dehumidification of more outside air than is actually needed, improved indoor air quality by increasing outdoor air ventilation if CO_2 levels rise and finally improved humidity control (US DOE 2004a). Grocery stores are ideal candidates for fluctuating occupancy patterns. Grocery stores also have large refrigeration loads which potentially could benefit from reduced space humidity loads that the display cases would otherwise remove. More recently, declining costs for implementing DCV technologies have made this measure very attractive (US EPA 2008a).

2.4.3.6 Capturing Cold Air Spills from Open Display Cases

The operation of open display cases in a typical supermarket causes a continuous spill of cold air into adjacent aisles causing the “cold aisle” effect. This cold air spillage contributed by the display cases, also known as case credits, proves to be beneficial to lowering the demand put on space cooling systems in the store and needs to be accounted for. This phenomenon creates a need for specialized air distribution to compensate for resulting “cold aisle” effect. Three studies have been reviewed by this literature which account for this effect and propose solutions. These include a study by Pitzer and Malone (2005) for case credits and return air paths for supermarkets and a study by Deru et al., (2011b) describing a case study grocery store for the Whole Foods Market.

Pitzer and Malone in their study on case credits and return air paths for supermarkets provide solutions which include providing air ducts below the display cases and fans on the back side of the cases to ensure removal of cold air collected near the floor of the aisle and directing it to the return air ducts of the stores HVAC system. Providing below / behind the cabinet return air ducts around open refrigerated cases is a measure that has been successfully implemented in several large grocery store chains (Pitzler and Malone 2005). In this measure the return air ducts are placed below and behind the display cases to "drain" the cold air that potentially pools around a display case especially open display cases (Hirsch 2006). The return air can potentially be collected in ducts above the display cases and be directed as return air of the HVAC system. Implementing such a measure ensures that the return air is cooler and drier than what is obtained with the conventional return paths (Pitzer and Malone 2005). Pitzer and Malone concluded that optimum implementation of return air methods to account for case-credits reduces the capacity of the space cooling requirements by from 144 to 80 tons and reduces the fan motor from 50 to 40 hp. At the same time the space heating capacity requirement is increased from 1,037 to 1,322 MBtu/hr. A schematic diagram of the system configuration is presented in Figure 2-4.

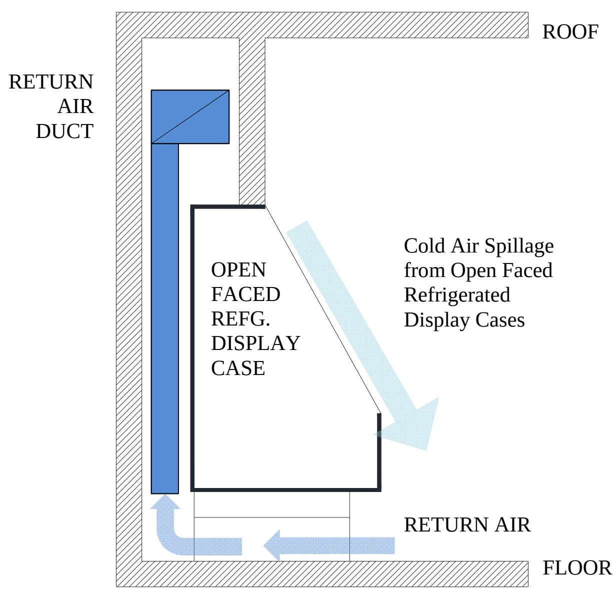


Figure 2-4: Schematic Sectional View of Open Faced Refrigerated Display Case with Under-Case Return-Air Path (Source: Pitzer and Malone 2005)

2.4.4 Efficiency Measures for the Service Water Heating

A number of references were reviewed to assess various efficiency measures for energy requirements for service water heaters in grocery stores. These include the design guide for efficient water heating delivery and use (Fisher-Nickel 2010), E Source (2006), and AHRI directory of certified hot water heating equipment (AHRI 2012).

Fisher-Nickel (2010) categorize conventional hot water systems into three fundamental groups. These include appliances that use hot water, distribution piping and hot water heaters. Preheating is sometimes considered using free-heating from heat recovery technologies or from solar water heating technologies.

Reducing hot water consumption by specifying high-performance equipment and accessories that use less hot water is an effective way of reducing water heating energy. Improving the efficiency of point of use equipment includes installing low-flow, high performance fixtures such as faucets, pre-rinse spray vales and dishwashers (Fisher-Nickel, 2010).

When considering service water heaters, gas hot water heaters are typically preferred to meet the requirements of the activities in the grocery store due to lower fuel costs. Water heaters can include storage, tank-less and boiler based. Fisher-Nickel (2010) observes that certain service areas in the grocery store require large quantities of hot water at some point of time. This is due to the intensive daily washdown periods typically seen in the bakery and other food preparation areas of the store. This practice makes the installation of tankless water heaters not very practical (Fisher-Nickel 2010). Installation of these heaters does however have certain advantages in terms of space saving. These heaters can also be used in restrooms and other service spaces such as the pharmacy and photo processing where not so much water consumption is required. E Source (2006) points out that although tankless water heaters have a drawback of providing hot water more slowly than conventional tank water heaters, which can slow the performance of flow based equipment, this disadvantage can be overcome by connecting multiple tankless units in parallel to provide the desired flow rate.

Gas-fired condensing water heaters are 10 – 20% more efficient than non-condensing models (Fisher-Nickel 2010). Condensing water heaters extract more energy from combustion gases by means of condensing water vapor contained in the combustion products. Condensing water heaters can attain efficiencies over 95%. The website of the American Heating and

Refrigeration Institute (AHRI 2012) has a directory of commercially available condensing water heaters that could be implemented in energy efficient grocery store designs.

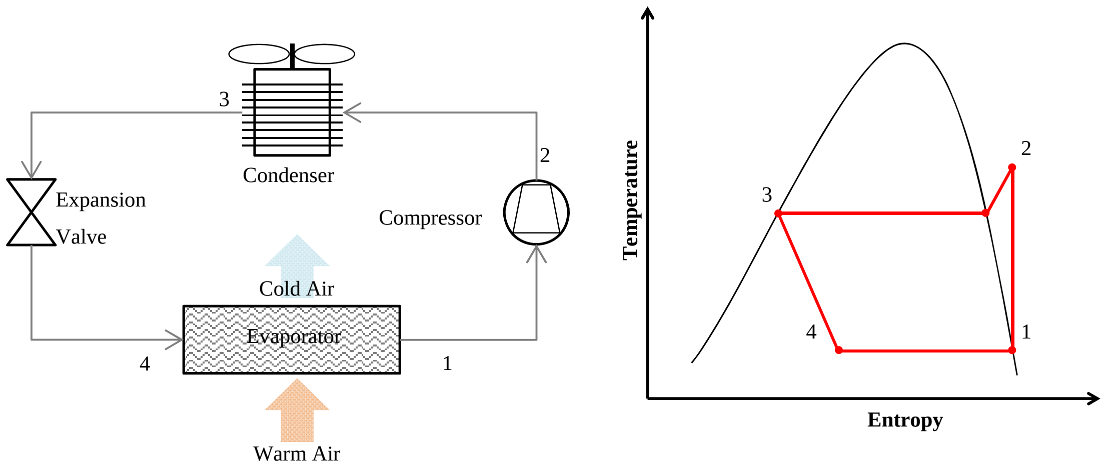
The use of solar thermal strategies for water heating in grocery stores is best implemented for preheating purposes. This is because of the high minimum water temperatures (i.e. 120 F – 140 F) required by commercial food preparation establishments (Fisher-Nickel 2010). Refrigerant heat recovery systems may be used to provide hot water heating or preheating. These systems work by harvesting the heat that would otherwise be rejected from the refrigerants in the condensers (Fisher-Nickel 2010).

2.4.5 Efficiency Measures for the Refrigeration Systems

The typical refrigeration systems installed in grocery stores operate on the thermodynamic principles of a vapor-compression cycle. The basic vapor-compression cycle consists of an evaporator, a compressor, a condenser and an expansion valve. A schematic diagram presenting the working of the vapor-compression cycle is presented in Figure 2-5.

As mentioned in the earlier section, refrigeration systems can account for about 39% of the energy consumption in grocery stores. Compressors and condensers account for 60-70% of refrigeration energy consumption (Baxter 2003). This amounts to 23% - 27% of the total energy use of the grocery store. The remainder is consumed by evaporator fans, display-case lighting evaporator defrosting, and for anti-sweat heaters (Baxter 2003) which are components of refrigerated display-cases and low temperature storage areas.

Efficiency measures that can be applied to the different components of the refrigeration system include measures in publications by Author Little Inc. (Westphalen 1996) , the U.S. Department of Energy Building Technologies Program (Goetzler et al. 2009), the California Energy Commission’s recommendations for energy efficiency standards for refrigeration systems in the Title-24 building code (PG&E 2011) and energy saving measures and strategies provided by the Carbon Trust good practice guides for refrigeration systems (Carbon Trust 2012).



Note: Numbers on diagram indicate the position of the refrigerant on the temperature entropy graph.

Figure 2-5: Schematic Diagram of the Vapor-Compression Refrigeration System

2.4.5.1 Refrigeration Compressors

Grocery stores typically use several medium and low temperature[15,16] compressor racks to meet the different temperature refrigeration loads. A typical refrigeration rack consists of several compressors connected in parallel, refrigerant piping, electronic controls and thermal insulation (Goetzler et al. 2009). A schematic diagram of a refrigeration compressor rack is presented in Figure 2-6. Finally, in order to meet the changing refrigeration loads, the operation of compressors is typically controlled by cycling varying number of compressors to meet the loads. A larger range of refrigeration loads can be served by incorporating compressors of different capacities. Energy efficient measures for compressors include improvements to compressor efficiency and improvements to the suction pressure and control strategies implemented to operate the compressors.

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[15] Medium temperature refrigeration cases operate within a temperature range between 0°F and 40°F. Low temperature refrigeration cases operate within a temperature range of -40°F to 0°F (ASHRAE 2006).
[16] In many instances, a single split-suction compressor rack is installed to serve both medium and low temperature refrigeration cases.

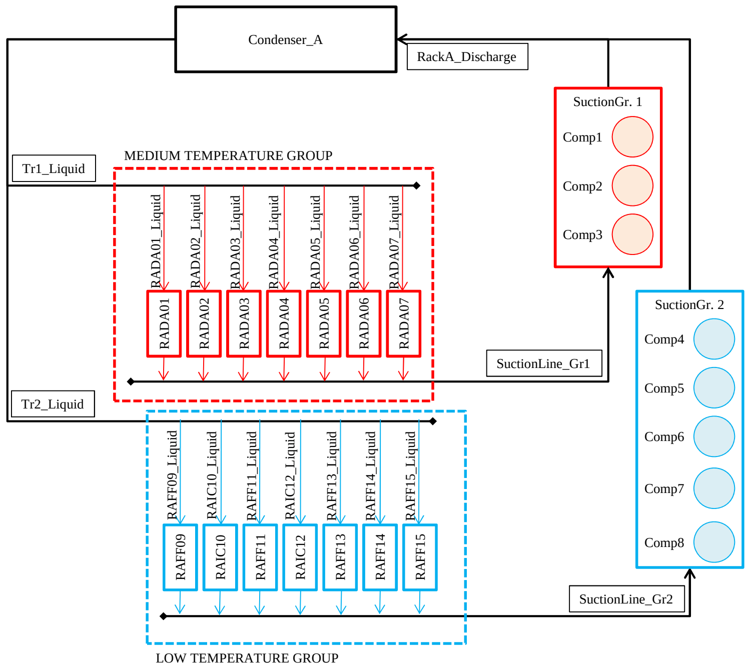

Figure 2-6: Schematic Diagram of the Refrigeration a Split-Suction Compressor Rack

2.4.5.1.1 Compressor Efficiency

Compressor efficiency can be gauged on the ability of compressors perform at optimum efficiencies over a range of refrigeration loads. Typically, compressors are cycled on and off to meet the varying load requirements. Cylinder unloading[17] for reciprocating compressors is another measure that is typically implemented to modulate compressor capacity but does not provide as much savings (Goetzler et al. 2009). Other measure incorporate variable speed drive[18] (VSD) motors to achieve greater efficiency over systems that rely on cycling to meet the loads. Compressor motors with VSDs vary the capacity of the compressor by regulating the speed of the motor. The motors enable the compressors to have higher efficiencies over a wide operating range (Tassou et al. 1994, Goetzler et al. 2009).

[17] Reciprocating compressors have many cylinders that are utilized to move large quantities of refrigerant. At reduced refrigerant flow rates, lesser number of cylinders is required to move the refrigerant.
[18] Variable Speed Drives (VSDs) adjust the speed of the motor to meet fluctuating load requirements by varying the motor's supplied voltage and the frequency of power.

Other improvements to compressor efficiency the use of ECM motors[19] (Goetzler et al. 2009) and the use of different types of compressor technologies such as scroll compressors[20]. The design of scroll compressors provides 70% fewer moving parts than reciprocating compressors. Advantages of using scroll compressors over reciprocating compressors include reductions in noise, cost and reliability (Goetzler et al., 2009). However, high efficiency reciprocating compressors are as efficient as, or more efficient than scroll compressors.

2.4.5.1.2 Suction Pressure and Temperature Control

Typical grocery store refrigeration systems implement fixed suction controls[21] to operate the refrigeration compressors. By installing floating suction pressure controls compressor rack suctions pressures are allowed to float upwards provided all the display-case temperatures are met (Goetzler et al. 2009, PG&E 2011). Energy savings result from operating at higher saturated suction pressures, which in turn reduce lift and compressor power (PG&E 2011).

Floating suction pressure controls can be achieved by electronic evaporator pressure controls (EEPRs) that use digital reading of temperatures in the display case as a basis to automatically adjust and control the suction pressure (Singh n.d.). As a result, the display case is held at the exact same temperature, at the same time the suction pressure set-points of the compressors are continually readjusted to meet the evaporator requirements. According to Singh (Singh 2006) up to 8% of compressor energy savings are seen as a result of implementing EEPRs. Other advantages of implementing EEPRs include remote access; lower pressure drops across the valve than when compared to the EPR; and tighter fixture temperature control, which results in reduced product shrinkage and increased shelf life.

2.4.5.2 Refrigeration Display Cases

2.4.5.2.1 Operation and Categories of Display Cases

One of the primary components in the operation of the grocery store is the refrigerated display-case. These cases are used for display of products to consumers and to store perishables

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[19] Electronically Commutated Motor is a brushless DC motors with a permanent magnet rotor that is surrounded by stationary motor windings. The power provided to the motor is pulsed on and off electronically with semi-conductor devices or transistor to several coil groups within the motor. By varying the time and duration of pulses, the electronic controller can control the speed of the motor and maintain a high torque at start over a broad range of speeds.

[20] Scroll compressors compress gas between two spirals, which is different than the method implemented in reciprocating compressors.

[21] In this control strategy, the fixed set point of suction pressure of compressors is selected. This set point in turn requires the temperature of the refrigerant leaving the refrigerated display cases (evaporator) to be the same regardless the load on the display cases.

such as meats, produce and dairy products under controlled temperature. These display-cases operate at different temperatures ranging from -40°F to 35°F (ASHRAE 2006) depending on the requirements of the products and can be categorized as low and medium temperature cases.

Display-cases can also be categorized based on their configuration. The four basic configurations are: a) open vertical, b) tub, c) closed vertical and d) service cases. Walk-in storage coolers and freezers form another category of refrigerated product storage found in the store whose function is similar to that of the display-cases. However, walk-in coolers and freezers are used to store food products after receiving them from delivery trucks. Walk-in coolers must then hold large batches of products for long periods of time until these products are placed out for sale.

2.4.5.2.2 Loads on Display Cases

In order to establish energy efficiency measures for display-cases, it is important to assess the loads on the refrigerated cases. Loads on the display-cases include: infiltration, heat conduction, thermal radiation, and internal thermal loads (ASHRAE 2006; Walker et al. 2004; Faramarzi et al. 2000; Faramarzi and Kemp 1999; Faramarzi 1999). It is also important to note that the case type plays an important role in calculating the impact of each component on overall loads for display-cases. Laboratory tests conducted by Faramarzi (1999) and Walker et al. (2004) showed that infiltration constitutes the largest cooling load for open vertical display-cases. Radiation and internal loads are the next largest constituents. For tub cases and closed vertical cases, radiation constitutes the largest load.

Infiltration in refrigerated display cases primarily takes place though case openings in case of open display cases. As noted in the previous paragraph, infiltration forms the largest percentage of loads for open vertical display cases[22]. For such display cases, infiltration can be reduced by an air curtain that separates the refrigerated air of the case from the ambient air of the store. Given the enormous amount of energy being wasted due to infiltration, designing open display-cases with more efficient air curtains is crucial. Several studies have investigated the performance of display-case air curtains (Howell 1993a &1993b; Howell et al. 1999; Walker et al. 2004; Amin et al. 2009). These studies conclude that up to 10% - 15% reduction in infiltration be achieved by implementing appropriate measures such as the design of an improved discharge air grill and improved air-flow control. The use of low-emissivity aluminum shields over open

[22] According to Faramarzi (1999), infiltration accounts for 70 to 80% of the total cooling loads on open display cases.

display cases during unoccupied periods of the grocery store results in the reduction of radiative and convective heat transfer into the cases and hence reducing the energy consumption (SCE 1997). The installation of glass doors for display cases has proven to be more energy efficient than open display cases due to a drastic reduction in infiltration loads (Fricke and Becker 2010, Faramarzi et al. 2002). Installing glass doors on an open vertical refrigerated display case reduced the refrigeration load by 68%, resulting in an 87% reduction in compressor power demand (Faramarzi et al. 2002). Furthermore, it was also observed that the average temperature of food products was reduced by 6%.

Conduction in refrigerated display cases takes place primarily through the physical envelope of the fixture. Conduction loads on display cases can be reduced by the use of insulation with higher R-values such as vacuum panels (Walker et al. 2004) and by installing more efficient doors with triple-pane glass and insulating gases encased between the panes (Goetzler et al. 2009).

2.4.5.2.3 Evaporator Fans

The function evaporator fans in display-cases is to move air across the evaporator coil and to circulate the chilled air within the refrigerated space in order to maintain near uniform temperatures within the space. Typically, these fans are installed with shaded pole (SP) [23] or permanent split capacitor (PSC)[24] induction type electric fan motors and operate at a single speed.

Improvements to evaporator fans can be made by improving the efficiency of fan motors using ECM motors[25] instead of SP motors or PSC induction type electric fan motors (Walker et al. 2005, Karas 2006, Goetzler et al. 2009). The use of ECM motors instead of standard SP motors can reduce the fan power consumption by 67% (Karas 2006). Savings from installing high efficiency fans result in less consumption of electrical power to operate the motors.

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[23] Shaded Pole (SP) motor is a single-phase induction motor having one or more auxiliary short-circuited windings acting on only a portion of the magnetic circuit; generally, the winding is a closed copper ring embedded in the face of a pole; the shaded pole provides the required rotating field for starting purposes.

[24] Permanent Split Capacitor (PSC) motors implement smaller start-up winding in addition to the main winding. The start-up winding is connected in parallel with the main winding and in series with a capacitor. At start-up the interactions between the magnetic field generated by the start-up winding and that generated by the main winding in due rotation (Goetzler et al., 2009).

[25] Electronically Commutated Motor is a brushless DC motors with a permanent magnet rotor that is surrounded by stationary motor windings (US DOE 2008b).

Reductions are also due to the reduced heating of the fan motor,[26] which results in reduced refrigeration loads and less compressor energy required to remove the heat from the cabinet (Goetzler et al. 2009).

Improvements to evaporator fans also can be made by installing fan motor controllers such as variable speed drives (VSDs) instead of operating the fans at one speed, which is standard practice for many grocery stores (Walker et al. 2004, Goetzler et al. 2009) . The installation of VSD controls saves energy because a fan with a VSD can vary the energy consumption and associated fan motor heat loads according to the change in cooling capacity requirements (Goetzler et al. 2009).

Finally, an improvement to the design of fan blades helps in the more efficient movement of air, which translates into reductions in energy consumption (Walker et al. 2004, Goetzler et al. 2009). In some cases installation of tangential evaporator fans[27] can also be considered to reduce energy consumption (Walker et al. 2004, Goetzler et al. 2009). The long thin impellers of these fans provide improved distribution of airflow at the same time requiring only one high efficiency fan motor for the entire display case (Goetzler et al. 2009).

2.4.5.2.4 Expansion Valves

Temperature in refrigerated display cases is controlled by means of controlling the flow of refrigerant. The refrigerant flow in the evaporator coil is regulated by an expansion valve. Mechanical thermal expansion valves (TXVs)[28] are commonly used in refrigerated display cases to meter the refrigerant into the evaporator coils. They do so by adjusting the refrigerant flow in the evaporator to control the refrigerant superheat[29] leaving the evaporator (Goetzler et al., 2009). This is accomplished by an appropriate selection of the charge within the thermostatic sensing bulb of the TXV located at the exit of the evaporator and adjustment of a counterbalance spring in order to achieve the desired level of superheat over a range of

[26] The evaporator fan motor is usually installed within the refrigerated enclosure. Depending on the efficiency of the fan, electric power consumed by the fan which is not converted to mechanical energy, gets dissipated as heat in the refrigerated enclosure.

[27] Tangential fans consist of a broad cylindrical rotor with many forward curved blades. The fan blades are thin and long, which make the fan longer in length than in diameter. The flow of air across the fan is two dimensional moving transversely across the impeller passing the blading twice.

[28] TXVs work by application of two pressure balancing forces on opposite side of the valve plunger, the evaporating pressure and the pressure of a thermostatic sensing bulb in contact with the evaporator exit piping (Goetzler et al., 2009).

[29] Superheat refers to the difference in temperature between the vapor at the entrance and exit of the evaporator as seen by the sensing bulb of the TXV valve (Althouse et al., 1996).

evaporating pressures and conditions (Goetzler et al., 2009). These valves have numerous limitations, which include complicated manual control, narrow range of refrigerant flow; and high pressure drop across the valves, all of which increase compressor energy consumption.

Electronic Expansion Valves (EEVs) are an improvement over the TXVs. These valves are controlled electronically by information provided by temperature sensors at entrance and exit of the evaporator and by a small motor that opens and closes the valve accordingly, thus offering a more precise control of the superheat. Other advantages include a possibility of wider flow range of refrigerant and more precise display case temperature control, both of which reduce compressor energy consumption.

2.4.5.2.5 Defrosting

Defrosting is a necessary process to remove frost built-up from evaporator coils of refrigerated display-cases and food storage areas that operate at or below 32 F. This is done to maintain proper food storage conditions, which would otherwise deteriorate over time with the build-up of frost. Common methods of defrost in refrigerated cabinets include: a) off-cycle defrost, b) electric defrost, c) hot or cool gas defrost, and d) modular defrost. In the off-cycle defrost mode, the frost over coils is allowed to melt away naturally. Electric defrost is the most reliable but the most energy consuming, while hot gas defrost, although the most energy efficient, is the most costly method to implement because of the extensive additional piping, controls and additional maintenance. Therefore, considerable opportunity exists for the application of more sophisticated defrost control strategies both to save energy without increasing first costs, and to improve temperature control, when compared to electric defrost if these constraints can be overcome (Walker et al. 2004). On the other hand, use of hot gas defrost requires an increase in refrigeration charge as well as an increase in potential for leaks due to installation of additional piping and valves (PG&E 2011). In the modular defrost, the refrigerated display case is connected to two evaporators with each evaporator being independently defrosted.

Humidity also impacts the energy consumed by defrosting (Henderson et al. 1999) as well as the pull-down loads associated with the defrost cycle and product loading. Controlling store-wide humidity levels can reduce the energy consumed to defrost the ice accumulated around the evaporator coil in the refrigerated cases. However, it has been observed that the impact of reduced store-wide humidity decreases drastically for closed door cases (Howell

1993a, b; Kosar et al., 2005). Hence the impact of store-wide humidity decreases on the installation of doors for refrigerated display cases.

Several defrost control methods are available, and can be broadly categorized into two categories: a) Timer controlled, and b) demand controlled.

Advantages of using timer controlled defrosting is simplicity, reliability and low cost. On the other hand, time clock initiated defrost controls may cause the cabinets to defrost too infrequently at high relative humidity resulting in momentary high product temperature, or too frequently at low humidity resulting in excessive energy consumption (Tassou et al., 1999). Tassou noted that with such strategies the refrigerant temperatures can become unstable after an extended length of time before the defrosting mode is turned on indicating the complete air-side blockage of the evaporator coil. In addition, the authors also note that this condition can happen for different periods of time and for different humidity levels.

Demand defrost controls initiate and vary the time of the defrost mode according to the actual need of the refrigerated display cases. A number of demand defrost techniques are available which include air pressure differential sensing across the evaporator, sensing the temperature difference between the air and the evaporator surface, fan power sensing, variable time defrost based on relative humidity and air differential across the coil (Dutta and Tassou 1997) Advantages of implementing such type of controls is to reduce supermarket refrigeration system energy. Studies demonstrated reduction in defrost heater operation by 50% to 75% on the implementation of demand defrost controls (Tassou and Datta, 1999; Lawrence and Evans, 2008; Hindmond and Henderson 1998, as referenced in Fricke and Sharma, 2011). Disadvantages include poor reliability and high capital costs (Datta and Tassou, 1997).

2.4.5.2.6 Anti-Sweat Heaters

Another component of the display-cases that consumes substantial energy is the anti-sweat heater. The purpose of the anti-sweat heaters is to hold the exposed surfaces above the dew point so that condensate and frost do not form on the surfaces. Anti-sweat heaters are also used on glass surfaces to prevent fogging and to keep door gaskets from sticking. A typical grocery store can have 10 kW to 20 kW of electricity operated anti-sweat heaters installed (Henderson et al. 1999) that operate continuously or are constantly pulsated for all hours of the year (Henderson et al. 1999, Faramarzi et al. 1999).

Anti-sweat heaters are affected by store-wide humidity levels. Henderson et al. (1999) observed a 3.5 – 7.8 kWh/day reduction in energy use of anti-sweat heaters for every 1% drop in relative humidity of the store depending on the control system used or about 0.07 – 0.15% of annual energy consumption of the store.

In order to reduce the energy consumption of anti-sweat heaters, several commercially available technologies are often appropriated. These include pulse modulating controls (PMC) that modulate the frequency of heating pulses per unit of time as a function of relative humidity (Faramarzi et al. 1999, Goetzler et al. 2009, Hirsch 2006); anti-fog films and special polymer doors (SPD), which are doors that do not require a heater for the glass part of the door assembly; multiple panes of glass separated by inert gas fill and use of door frame construction materials with low thermal conductivity (Faramarzi et al., 1999). Electric anti-sweat heaters can also be replaced with hot refrigerant gas lines (Goetzler et al. 2009). However this measure would require additional piping and increase in the quantity of refrigerant charge and the possibility of leaks.

Research performed at the Southern California Edison (Faramarzi et al., 1999) concluded that varying the intensity of pulse modulations on varying the store-wide humidity from 55% to 35% resulted in 17.4 % drop in total cooling loads of refrigerated display-cases under test conditions[30]. The research also concluded that anti-fog films and SPDs [31] have also shown promising results and performed comparably to the PMC technologies in reducing the overall energy consumption of the grocery store. However, the main drawback observed was the increased fog recovery time (Faramarzi et al., 1999) implying that SPDs are better suited for climates with low humidity levels.

2.4.5.2.7 Display-Case Lighting

Display-case lighting consumes approximately 16% of the energy consumed by product display-cases in grocery stores (Faramarzi 2006). T8 fluorescent lighting is typically used in display-cases. New LED technologies offer an effective energy efficient alternative to the conventional T8 fluorescent lighting. LEDs have traditionally been used in small lighting situations such as exit signs and neon signs for years. More recently they are beginning to be used in refrigerated display-cases. LED technology has improved over the years. Narendran et al. (2006) in a study on energy efficient lighting alternatives for commercial refrigeration report that in 2003, typical efficacies for LED fixtures was measured to be 25 lm/W. By 2006 the efficacy

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[30] Testing conditions met the requirements of ASHRAE Standard 72-83.
[31] For SPD's, the glass portion of the door does not require heating. However, the frame portion of the door requires the use of anti-sweat heaters to prevent the formation of frost on these surfaces.

of LEDs improved to 45 lm/W. More recently EERE reported improved efficacies in the range of 60-92 lm/W for cool white LEDs and in the range of 27-54 lm/W for warm white LEDs as compared to linear fluorescent lamps, which provided efficacies in the range of 50 -100 lm/W (US DOE 2009b).

A study performed by Pacific Gas and Electric Company (Theobald 2007) observes that not only are LEDs more economical to operate (43% power reduction and 33% in luminance reductions), they also last as much as ten times longer than the fluorescent lamps they replace. LEDs have a fast response rate and are not affected by low temperatures like fluorescent lamps. LEDs also lend themselves to more flexible configurations than that provided by fluorescent lamps. Fluorescent lamps must be placed vertically[32] at the hinge of the door from where the light is unevenly distributed. On the other hand, LEDs can be placed under shelves for better distribution of light. The resultant savings on installing LEDs are not only from reduced lighting energy consumption, but also due to reduced heat loads resulting from reduced production of radiant heat, making the compressors run less often (US DOE 2008a; Grondzik 2010). In addition, LEDs can also be combined with motion sensors to further save energy consumption due to lighting (Pandharipande et al. 2010). Finally, LEDs are more environmentally friendly due to absence of mercury content (Narendran et al 2006).

Fiber-optic lighting is currently being investigated for low-temperature reach-in cases as well as for perishable goods (Faramarzi 2006). In this technology a single centrally-located, high-efficiency light source feeds multiple cases remotely via fiber-optic cables. Although the systems can be extremely costly, the ability to eliminate infrared radiation heat gain away from the freezer cases, while still illuminating the goods in an efficient, manner is a big plus. A study performed by the Southern California Edison Institute observed about 49% reduction in energy consumption from lighting, 17% reduction in energy consumption of compressors, and 25% reduction in overall energy consumption of the test-case grocery store when comparing the performance fiber-optic lighting with T8 fluorescent lighting in low temperature freezer cases (SCE, 2006).

[32] Vertical positioning of fluorescent lamps causes non-uniform distribution of gases in the lamp resulting in a reduction in light output and uniformity. In addition, the vertical positioning of the lamp also allows the accumulation of mercury droplets near the lower cathode, which has a detrimental effect on the life of the lamp.

2.4.5.1 Refrigeration Condensers

In typical arrangements of the refrigeration systems in grocery stores, one compressor rack is served by one dedicated refrigeration condenser unit. These condensers are typically air cooled and remotely located. The fixed head pressure[33] system is a commonly implemented control strategy. Efficiency measures for refrigeration condensers include the installation of evaporative condensers, floating head pressure control, and improving fan efficiency.

2.4.5.1.1 Evaporative Condenser

Evaporative condensers[34] operate by using evaporative cooling to condense the refrigerant. The condenser coils are housed within a structure and water is sprayed on these coils. The cooling process is a result of water being evaporated on coming in contact with the tubes and the outside air that is forced through the structure. Baxter in his assessment of distributed refrigeration systems in the grocery store pointed out that 8% in overall savings were linked to implementing evaporative condensers when compared to the base-case multiplex system with an air-cooled heat rejection unit (Baxter 2003). However, it was noted that evaporative condensers impose greater maintenance efforts and costs (Baxter 2003, Goetzler et al., 2009). In addition, evaporative condensers may not be effective in humid climates.

2.4.5.1.2 Floating Head Pressure Control

Floating head pressure control in condensers takes advantage of low outside air-temperatures to reduce the work for the compressor by allowing the head pressure to vary with the outdoor air conditions (Thornton 1991). The floating head pressure system was found to reduce the electric energy consumption of the refrigeration system in cool climates due to a decrease in condensing pressure at low ambient temperatures. By removing the fixed head pressure mechanism, the floating head pressure control eliminates the inefficiencies associated with fixed head pressure operation at low ambient temperatures, which include increased subcooling of the refrigerant, increasing the capacity and efficiency of both the condenser and evaporator, which improved overall COP of the refrigeration system (Thornton 1991, Scott

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[33] In a fixed head pressure control, the condensing pressure is maintained at a fixed set-point regardless of the system load by controlling the condenser fan operation.

[34] Evaporative condensers are not be confused with cooling towers, in which the condenser and tower are separate structures with water being pumped from the condenser to the tower where it is cooled via the evaporative process (Whitman et al., 2005).

2007, PG&E 2011). In addition, floating head pressure reduces the work to be done by the compressor (i.e., lift) at non-peak loads (Scott 2007).

Scott (2007) in his presentation on refrigeration control for operating cost reduction points out that 12 – 20% savings in compressor and condenser energy can be seen by implementing floating head pressure control. Scott also points out that no savings may be seen without the implementation of appropriate control strategies. On the other hand, Thornton (1991) observed that a floating head pressure system did not reduce electrical demand and achieved only minimal savings against the fixed head pressure systems in warm climates.

2.4.5.1.3 Efficient Fan Control

Measures to improve condenser fan efficiency are similar to that for evaporator fans and include high efficiency fan motors such as ECM motors, variable speed control of fan motors, and improved design of fan blades to improve efficiency. These measures have been discussed in detail in the Section 2.4.5.2.3 on evaporator fans.

2.4.5.2 Mechanical Subcooling

Mechanical subcoolers are heat exchangers that use the cooling effect of refrigerant from the medium temperature rack to provide subcooling to low temperature racks (Thornton 1991, Khattar and Henderson 2000). Mechanical subcoolers can also be separate units. A schematic diagram of a mechanical subcooler is presented in Figure 2-7. Thornton, Khattar and Henderson noted that the addition of a heat exchanger downstream of the condenser, that rejected excess heat to the ambient, resulted in a lower quality at the evaporator inlet. This subsequently resulted in an increase of the COP of the system (Thornton 1991). It was also noted that since the subcooler loads are met with more efficient medium temperature compressors this resulted in decreased annual compressor energy usage (Khattar and Henderson 2000). The dedicated subcooling cycle was found to perform best at high ambient temperatures and low refrigeration temperatures because of the larger amount of subcooling provided at higher ambient temperatures as compared to the amount of subcooling provided at lower ambient temperatures (Thornton 1991). Couvillion et al. (1988) used a computer model to predict improvements in COP that ranged from 6% to 82% and improvements in capacity that ranged from 20% to 170% depending on conditions at which the subcooler model operated.

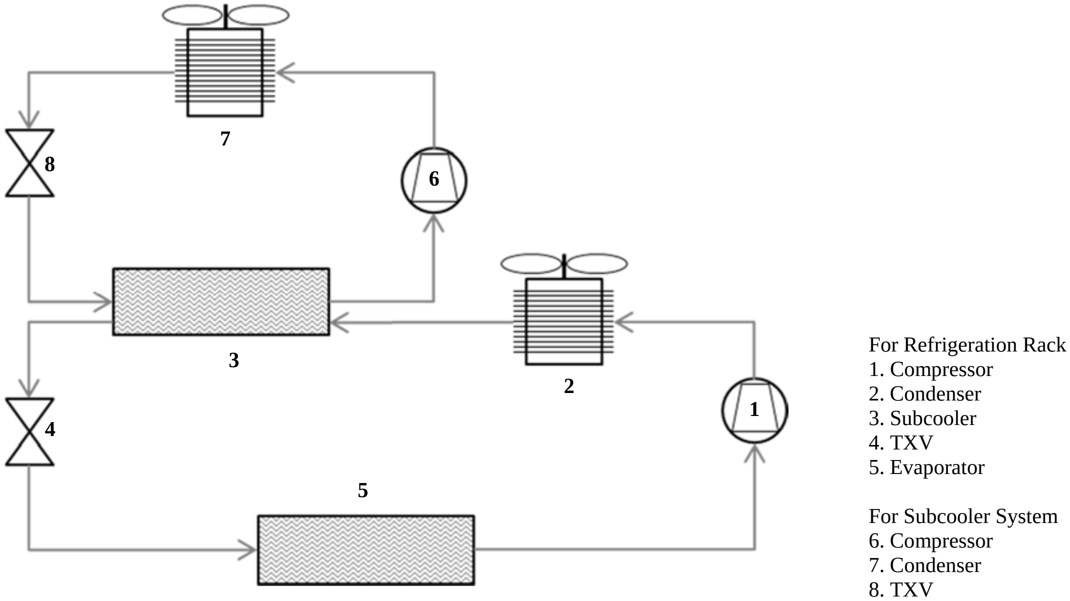

Figure 2-7: Schematic Diagram of a Mechanical Subcooler Attached to a Vapor-Compression Refrigeration System (Source: Thornton et al. 1991)

In order to perform effectively, the subcooling cycle evaporator temperature must lie between the evaporating temperature and the condensing temperatures of the main cycle (Thornton 1991). In practical situations it was noted that the subcooling cycle temperature is approximately 50 degrees higher than the main cycle evaporating temperature (Thornton 1991). Thornton (1991) notes that if the subcooling evaporator temperature is located at the upper extreme, the main cycle condensing temperature, the combined system effectively becomes one system. The sub cooling cycle does not provide any subcooling to the main cycle because there is no temperature difference in the subcooler between the refrigerant flows. With no heat transfer in the subcooler, there is no work supplied to the subcooling cycle compressor. The COP of this system is identical to the COP of the standard floating head pressure system operating between the same extremes. On the other hand, if the subcooler evaporator temperature is located at the lower extreme of the main cycle evaporator temperature, the dedicated subcooling system effectively becomes two systems. The reason that this operation degrades the system performance is due to the effectiveness of the subcooling. Both systems provide the extra capacity at approximately the same COP. The optimum point at which to set the subcooling cycle evaporator temperature must balance the two effects previously mentioned – the increased subcooling to the main cycle and the greater subcooling cycle thermal lift. Thornton et al. (1994) observed that the optimum subcooling temperature was strongly dependent on the heat sink and refrigerated space and weakly dependent on the subcooler heat exchanger parameters.

In summary, the vapor-compression refrigeration systems present a viable solution to refrigeration loads in stores. Over the years several variations in the basic cycle, such as introduction of floating head pressure, ambient subcooling and dedicated mechanical subcooling, have been used to improve the efficiency of the overall system, making this technology well established in the food refrigeration industry. However, the rising costs of electricity and pressure to reduce the environmental impact of refrigeration operations has renewed the interest in thermal- driven technologies and development of new and innovative technologies that could prove to be advantageous as alternates to the vapor-compression system.

2.4.5.3 Alternative Refrigeration – Emerging Technologies

Over the years different alternative technologies to the concepts of vapor-compression refrigeration have been developed to improve the performance and efficiency of refrigeration systems. The key drivers to encourage the development of these alternative refrigeration technologies is the phase-out of the use of chlorofluorocarbons (CFCs), hydrochlorofluorocarbons (HCFCs) and hydrofluorocarbon (HFCs). The main challenge for most of these refrigeration cycles is the improvement of the COP to make them competitive with the vapor-compression systems, more commercial availability and to lower first costs.

In their review of emerging technologies for refrigeration applications Tassou et al. (2010) reviewed several emerging technologies including ejector refrigeration systems, air cycle refrigeration for low temperature refrigeration, Stirling cycle refrigeration, thermoelectric refrigeration, thermoacoustic refrigeration and magnetic refrigeration. The use of solar energy to drive refrigeration systems was also investigated by Klein and Reindl (2005), and Kim and Infante Ferriera (2008). The use of natural refrigerants such as CO_2 was examined by Girotto et al. (2004) and Sawalha, (2008a & 2008b). Ammonia as a refrigerant provides more refrigerating effect per unit mass flow than any other refrigerant used in vapor-compression cycles making it a good candidate to adopt as a refrigerant (Presotto and Suffert 2001; Pearson 2008). However safety concerns have prevented the widespread use of ammonia in refrigeration installations where refrigerant leaks could expose the customers to ammonia gas.

2.4.6 Whole Building Energy Efficiency in Grocery Stores

Very few studies have looked at energy efficiency for the entire grocery store. The report by Leach et al. (2009) at the National Renewable Energy Laboratory (NREL) is one of the recent studies that have implemented a whole store energy analysis. It documented the technical analysis and design guidelines to achieve whole-building energy savings of at least 50% over ASHRAE Standard 90.1-2004. To accomplish this, a set of energy efficiency measures was compiled for each of the sixteen climate locations in the United States as specified by ASHRAE. Efficiency measures were applied to lighting systems, plug and process loads, fenestration, building envelope, and HVAC systems. Specifications for refrigeration equipment were determined from industry standards and established references. The Energy-Plus whole-building energy simulation software was used for the analysis. The study provided a discussion on the inputs to the various parameters in the grocery store. However, an in-depth discussion on modeling these parameters in EnergyPlus was not presented. The study went on show a potential 50% reduction for each of the simulated climate zones. An important omission in the report was that there is no documented evidence regarding the calibration of the simulation model used for the analysis. In addition, potential technologies such as alternative HVAC systems, solar thermal technologies, advanced humidity control, strategies to use waste heat from equipment, tri-generation technologies, multiple compressor types, and under-case HVAC return air systems were omitted due to modeling constraints and a lack of reliable input data.

A report by Southern California Edison Institute investigated the impact of lighting and refrigeration technologies on a small grocery store with an area of 2000 ft^2 located in southern California (Sarhadian, 2004). Energy efficient measures that were assessed included retrofitting the store lighting system with high efficiency T8 lamps and low ballast factor electronic ballasts, replacing self-contained refrigeration units with multiplex parallel systems that were controlled using a central processing unit (CPU) backed rack controller and an energy management system (EMS); installing ECM motors for evaporator fans and using liquid-to-suction heat exchangers in walk-in coolers; using PSC evaporator fan motors in display cases; using special doors with low wattage anti-sweat heaters for display cases; implementing time initiated and time terminated defrost mechanisms[35]; implementing floating head pressure controls for refrigeration condensers; installing low temperature differential condenser coils; and implementing capacity

[35] As discussed earlier, using timer controlled defrost method is not the most energy efficient option available. However, this option is widely used due to low first costs and durability.

modulation control in compressors. The analysis was performed by monitoring the power requirements of the grocery store during pre-retrofit and post-retrofit period[36] of operation. The report found that implementation of efficient lighting technologies resulted in savings of 11% of lighting power demand and corresponding 5% reduction in lighting energy. While the implementation of emerging refrigeration technologies resulted in savings of 20% to 30% in refrigeration power demand and corresponding 23% reduction in refrigeration energy usage. Although two largest categories for energy consumption in the grocery store - lighting and refrigeration were addressed, the study did not perform a whole building energy analysis which included addressing the energy consumption of other systems in the store such as the performance of the HVAC system. In addition, in larger grocery store where there is an increase in the complexity of systems, the resultant interactions between these systems becomes more complex and hard to evaluate separately. This complexity makes the whole-building analysis, which is inclusive of all systems, a more viable approach to reduce energy consumption.

From the above discussion it can be concluded that grocery stores are large energy consumers or energy sinks. Several studies that were reviewed demonstrated that the energy consumption of grocery stores could be reduced by as much as 50%. However, even if the grocery store energy use could be reduced from 52.5 kWhr /sq. ft. (US EIA 2005) to 26.25 kWh / sq. ft., this still represents a very large electric load to be met by an on-site renewable energy system. Therefore, in order to approach net-zero energy consumption for a grocery store, one possibility would be to consider the store as part of a community. In the next section the study the review goes on to examine the options for the grocery store to be a part of such a community.

As seen in the above section, implementing energy efficiency measures in a high energy consuming building such as the grocery store can save up to 50% of the energy consumption. In the next section the possibility of cogeneration is explored to further bring down the source energy consumption of the grocery store. The discourse presented in the next section also discusses sharing of energy across building boundaries to maximize the cost-effective utilization of all energy (electricity and thermal) generated by the cogeneration system.

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[36] The store was monitored for a period of 16 days during both pre-retrofit and post retrofit periods.

2.5 The Option of Cogeneration

2.5.1 Overview of Cogeneration Systems

Cogeneration or Combined Heat and Power (CHP)[37] can be defined as a simultaneous production of electric power and thermal energy from a single fuel source (Caton 2010). A major benefit of implementing a CHP system is reducing source energy use. Other benefits include increased total system thermodynamic efficiency, lowered overall facility energy consumption costs, improved facility reliability, reduced electricity demand on constrained utility grid, reduced total CO_2 emissions and the ability to use biofuels[38] (Hyman and Meckler 2010). In many cases however, a CHP facility depends on the utility grid for either supplemental or standby power (Orlando 1996). In other instances, to ensure the viability of the CHP system, surplus power generated by the CHP system may need to be exported back to the utility. Hence an interconnection between the two sources is essential and the flow of power needs to be accommodated in both directions (Orlando 1996).

Several types of facilities can potentially benefit from the implementation of CHP systems. These include: district energy systems, universities and colleges, hospitals, municipal centers, commercial campuses, large commercial buildings, data centers, jails and prisons, oil refineries, wastewater treatment plants, pharmaceutical industries, industries requiring heating processes and residential systems. This analysis focuses on the implementation of CHP systems in large commercial buildings such as a grocery store and the potential of using surplus energy generated from the CHP system to power surrounding residential buildings.

In most applications, the main factor that determines the economic viability of the CHP scheme is a simultaneous and high utilization of both the thermal energy and electric power that are produced by the CHP system (Hyman and Meckler 2010). This concern can be offset to an extent by the use of thermal storage technologies in conjunction with CHP technologies[39]. Most of the literature indicates that the CHP plant needs to be fully utilized, providing heat and electric power for a period of at least 4,500 hours per year, to be viable for a 4 to 5 year payback period (ETSU 1995). The main difficulty in achieving high annual utilization is providing a heat

[37] From this point onwards, cogeneration is referred to as CHP.
[38] Biofuels are liquid fuels that have been created from biomass feedstock, which is a renewable energy resource (NREL 2012). The two most common types in use today are ethanol and biodiesel. These fuels can be used to power CHP systems thus creating a potential of reaching net-zero levels. However, this possibility has not been explored in this study.
[39] The use of thermal storage provides the possibilities of storing thermal energy during periods of low thermal demand for later use when needs arise (Dorgan and Elleson 1993).

demand during the summer months (Maidment and Tozer 2002). This becomes even more apparent in hot and humid climates.

Current CHP technologies can be applied to a variety of prime movers, including steam and gas turbines, IC engines, fuel cells[40] and more recently micro-turbines[41] and Stirling engines[42] (Borbely and Krieder 2001; WADE 2003; Caton 2010). Waste heat generated from powering the prime movers is either diverted to a heat exchanger or used directly to meet the thermal loads of the facility such as space heating and service water heating or used to run thermal-driven technologies such as absorption chillers or desiccant dehumidifiers. An assessment of selected prime movers followed by a discussion on the various options for heat recovery is provided in the following sections.

2.5.2 Prime Movers

Several documents and reports were reviewed to better understand the characteristics and performance of prime movers. These include information from EPA-sponsored studies for CHP technologies (US EPA 2008b), the CHP resource guide prepared by the World Alliance for Decentralized Energy (WADE 2003), a review on combined cooling, heating and power authored by Wu and Wang (2006), the CHP design guide for cogeneration by Orlando (1996) and several other sources which are cited accordingly.

2.5.2.1 Internal Combustion (IC) Reciprocating Engines

IC engines include two basic types of engines – spark ignition engine and the compression ignition engine. The spark ignition engine operates on the Otto cycle, which involves using a spark plug to ignite a pre-mixed air fuel mixture introduced into the cylinder (US EPA 2008b). The compression ignition engines operates on the Diesel cycle, which involves compressing the air introduced into the cylinder to a high pressure, and in the process raising its temperature to auto-ignite the fuel that is injected at high pressure (US EPA 2008b). Available sizes for spark ignition engines range from 15 kW to 10 MW (WADE 2003, Wu and Wang 2006). Available sizes for compression ignition engines range from 75 kW to 20 MW (WADE

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[40] Fuel cells are electrochemical devices that convert the chemical energy in fuels into electricity. In these technologies, the process of converting fuel to electricity is not limited by the thermodynamic restrictions imposed on the production of heat and mechanical work typical of most power generating equipment and hence do not fall into the category of prime movers(US DOE 2004b). These technologies will not be discussed in this study.

[41] Micro-turbines are small high-speed gas turbines with capacities in the range of 25 – 500 kW (WADE 2003).

[42] A Stirling engine is an external combustion engine (heat engine). In this type of engine heat is supplied from an external source to expand the working fluid which in turn moves a piston, thus producing work (WADE 2003).

2003). IC engine efficiencies are relatively high when compared to gas and steam turbines and can reach 45%.

The spark ignition engine can operate on natural gas, propane, gasoline or landfill gas as the source fuel (US EPA 2008b). The compression ignition engine can operate on diesel fuel, heavy oil or in a dual fuel configuration (US EPA 2008b). The compression ignition engines can also operate on bio-diesel (Turner 2006).

IC engines are characterized by compression ratios, size of bores and strokes, displacement capacity, and the type of aspiration system. IC engines are also characterized by the number of strokes on which the engine operates, with four-stroke engines being the most relevant to stationary power generation application (US EPA 2008b). Compression ratios for compression ignition engines are in the range of 20:1. On the other hand, compression ratios for natural gas spark ignition engines range between 9:1 and 12:1 depending on use of mode of aspiration and engine design. The modest compression ratios for the spark ignition engines are to prevent knocking[43]. Knocking in spark ignition engines is also prevented by operating the natural gas spark ignition engines at lower Brake Mean Effective Pressure (BMEP)[44] and peak pressure levels. However, lower compression ratios and operating spark engines at lower BMEP impact the efficiencies of natural gas spark ignition engine efficiencies making them less efficient than corresponding compression ignition engines (WADE 2006, US EPA 2008b). When considering type of aspiration systems in IC engines, in the simplest natural gas engines, the suction of the intake stroke provides a natural aspiration of air and fuel into the cylinder. On the other hand, more air is forced by the process of turbocharging[45] high performance natural gas engines.

The performance of IC engines is rated at ISO[46] conditions of 77°F and 100 kPa. Similar to gas turbines, the efficiency of gas engines degrades on an increase in ambient temperature or site elevation. However, the impact on efficiencey is not as significant as that for gas turbines (Caton 2010). Electric efficiencies of natural gas spark ignition engines range from

[43] Knocking is produced when a portion of the fuel in the cylinder is explosively auto-ignited due to compression and heating of the mixture before reaching the flame front from the spark.
[44] Brake Mean Effective Pressure (BMEP) is the average cylinder pressure on the piston during the power stroke. BMEP is a measure of engine power output or mechanical efficiency.
[45] Turbocharging is the process of forcing a large amount of intake air into the IC engine using a compressor driven by exhaust gases. To further increase the intake of air, the compressed air, which has been heated by compression, is cooled afterwards (WADE 2006).
[46] ISO Standard 3046/1.

30 percent LVH[47] for small stoichiometric engines[48] (<100 kW) to 40 percent LVH for large lean-burning[49] engines. These efficiencies are higher than the efficiencies of gas turbines of comparable size. Hence the fuel related operating costs are lower (US EPA 2008b). The electrical efficiency of a CHP system improves on increase in engine size. However, an increase in electrical efficiency implied that the absolute quantity of thermal energy available to produce useful thermal energy decreases per unit of power output. Efficiency of IC engines decrease at part-load conditions. However, this reduction in efficiency compares favorably to gas turbines, which typically experience steeper efficiency drops at corresponding part-load conditions. Occasionally, multiple-staged engines are preferred to avoid efficiency penalties.

Advantages of IC engines include lowest first capital costs, fast start-up capability and good operating reliability, high efficiency at partial load operation and they are ideal applications for electricity demands below 1 MW (Wu and Wang, 2006; US EPA 2008b). IC engines also have excellent load following characteristics (US EPA 2008b). Drawbacks include relatively high vibrations which require shielding measures to reduce acoustic noise, a large number of moving parts, frequent maintenance and locally high emissions which include NO_2 generation (Wu and Wang, 2006).

The four sources of usable waste heat in an IC engine include: exhaust gas, engine jacket cooling water, lube oil cooling water and turbocharger cooling. Engine exhaust heat represents 30 to 50 percent of the available waste heat. Exhaust temperatures between 850°F and 1200°F are typical. Engine jacket coolant accounts for up to 30 percent of the energy input and is capable of producing 200°F to 210°F of hot water. The percent depends on the type of engine cooling method being used[50]. It should be noted that in order to minimize the thermal stresses, the temperature differential across the circulating engine jacket water is limited to a maximum of

[47] Lower Heating Value is determined by subtracting the heat of vaporization of the water vapor from the Higher Heating Value (HHV) of the fuel (MECA 1997).
[48] Stoichometric engine operation is characterized to have the chemically correct amount of air in the combustion chamber during combustion (MECA 1997).
[49] Lean burning engine operation is characterized by excess air in the combustion chamber during combustion resulting in an oxygen rich exhaust (MECA 1997).
[50] Types of engine cooling methods include closed loop cooling systems and ebullient cooling systems. In closed loop cooling systems a forced circulation of a coolant occurs through the engine passage and an external heat exchanger. Coolant temperature is usually between 190°F to 210°F (US EPA 2008b). Depending on the engines requirement the lube oil cooling and turbocharger after-cooling may be used either separately or as part of the jacket cooling system. In ebullient cooling systems, the engine is cooled with a natural circulation of boiling coolant. This system is typically used with exhaust heat recovery for the production of low-pressure steam. Temperature of the coolant is usually in the range of 250 F. Turbocharged engines having greater percent of thermal energy in the exhaust gases and naturally aspirated have a greater percent of thermal energy in the jacket water.

15°F[51] (Hyman and Landis 2010). In some cases, increasing the temperature of the jacket water is a possibility especially if operating a hot water absorption chiller. Lube oil heat represents about 5 percent of the fuel energy and is typically rejected via the engine thermostat at about 130°F[52]. Approximately, 70 to 80 percent of the fuel's energy can be utilized by the recovery of thermal energy from the cooling and exhaust processes as well as power generation. Recovered thermal energy is either in the form of hot water or low pressure steam. The recovered thermal energy is usually used for low temperature process needs, space heating, potable water heating, and to drive absorption chillers for both air-conditioning as well as refrigeration (US EPA 2008b).

2.5.2.2 Gas Turbines

Gas turbines operate on the principles of the Brayton cycle, where air is compressed in a compressor, heated in a generator and then expanded through a turbine thereby producing power. The power generated by the gas turbine and consumed by the compressor is proportional to the temperature range (in terms of absolute temperature) within which the gas turbine is operating. (i.e. cooler inlet temperatures and higher exhaust temperature result in higher efficiencies).

Performance of a gas turbine system is particularly sensitive to contaminants in fuels, hence only clean fuels such as natural gas and clean liquid fuels are used. Gas turbines primarily operate on natural gas, synthetic gas, landfill gas and fuel oils. Gas turbines are one of the cleanest means of generating electricity (US EPA 2008b). When operated along with appropriate emission control technology, NO_x emissions from gas turbines are in single-digit parts per million range especially from larger turbines (US EPA 2008b). In addition, natural gas used in the gas turbines emits substantially less CO_2 per kWh generated than any other fossil technology in general commercial use (US EPA 2008b).

Gas turbine efficiencies are rated at ISO inlet conditions[53] of 59°F and 60% RH. Efficiencies for simple cycles range from 28% to 42%, while for combined cycles[54] range from 45% to 60%. However, gas turbines loose efficiency and capacity rather rapidly as the

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[51] Temperature differences greater than a delta T of 15°F can result in thermal shocking of the engine.
[52] 130°F water can be used for various low temperature uses such as domestic hot water heating, space heating and swimming pool heating.
[53] ISO Standard 3977-2, 1997.
[54] Combined cycle power plants combine a steam turbine in a bottoming cycle with a gas turbine. The steam generated in the heat recovery steam generator (HRSG) of the gas turbine is used to power the steam turbine to provide additional electricity.

temperature of inlet air rises above the rated levels (Caton 2010). The use of the chilled water[55] to cool the gas turbine inlet air particularly in humid climates can substantially improve the efficiency of the gas turbine and in turn improve the CHP economics (Hufford, 1992). Part load conditions also profoundly affect the performance of the gas turbine with the efficiency of the gas turbine decreasing rapidly at part load conditions (US EPA 2008b).

Typical efficiencies for gas turbines have improved over the years with the introduction of cooling systems[56] as well as material improvements[57]. Moreover, the introduction of processes such as recuperation[58], intercooling[59], inlet air cooling[60], and reheat steam injection[61] (US EPA 2008) have further improved the performance of gas turbines.

Gas turbines are available in sizes ranging from 500 kW to 250 MW (US EPA 2008b). For capacities less than 1MW, the installation of gas turbines proves to be uneconomical because of low electrical efficiency and consequent high cost per kW output (Wu and Wang, 2006) with smaller gas turbines, in the power range to 500 kW, having an electrical efficiency of about 18%. This fact is mainly due to the scale effect on the aerodynamic components, the assembly clearances and the relatively low turbine inlet temperature compared with large gas turbine (Cengel 2006). Also, for smaller sizes, gas turbines compete with IC engines, which are cheaper and more efficient.

Effective use of thermal energy contained in the exhaust gases of the gas turbine is the key to justifying its use. Thermal energy in exhaust gases represent approximately 60 to 70 percent of the inlet fuel's energy use. The most common uses of this energy are for steam generation in Heat Recovery Steam Generators (HRSGs). Thermal energy from the exhaust

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[55] Chilled water can be produced by absorption chillers that are driven by waste heat produced by the gas turbine.
[56] Cooling technologies involving circulation of cooled air through and around turbine blades and vanes. This allowed increase in firing and rotor inlet temperatures (Unger et al., 1998).
[57] Material improvements include improved steel alloys used in turbine vanes, blades and inlet blocks allowing for higher temperatures as well as increase in rotor life and reliability. Alloys include varying quantities of cobalt, nickel and chromium. Cobalt alloys are preferred in stationary blades and vanes due to higher heat tolerance and better welding characteristics. On the other hand rotating turbine blades use nickel alloys (Unger et al., 1998).
[58] Recuperators are heat exchangers that utilize the hot turbine exhaust to preheat the compressed air entering the combustor by means of a heat exchanger (US EPA 2008b).
[59] Intercoolers are used to cool the compressed air exiting the first section of the compressor before it enters into the second section, hence reducing the overall power consumption of the compressors in the gas turbine (US EPA 2008b).
[60] Inlet air cooling is the process of cooling the inlet air entering the turbine by 40 to 50 F. This cooling is typically done using absorption cooling that uses waste heat from the gas turbine. Other popular cooling methods include evaporative cooling and thermal-energy storage systems (US EPA 2008b).
[61] Reheat steam injection is a process in which combustion products are re-fired in a reheat combustor after partial expansion in the turbine, in the process increasing the turbines power output and efficiency (Engineering Review 2010).

gases can also be used as a source of direct process energy for unfired or fired process fluid heaters, or as preheated combustion air for boilers (US EPA 2008b). In many instances, supplemental firing is done by means of duct burners to increase the temperature of exhaust gases entering the HRSG in order to increase the steam production[62]. Supplemental firing can raise the temperature of exhaust gases up to 1,800°F. In addition, due to already elevated temperatures of the exhaust air entering the HRSG, the fuel consumed by duct burners is less than what would be required by a corresponding stand-alone boiler providing the same increment in steam generation (Unger et al. 1998). Supplemental firing also increases system flexibility by providing the ability to control steam production (Unger et al. 1998).

2.5.2.3 Steam Turbines

Steam turbines operate on the principle of the Rankine cycle, which is the basis for conventional power generating stations. The cycle consists of a boiler that converts water to high pressure steam. The pressurized steam is then expanded to a lower pressure in a turbine and finally exhausted to a condenser. The condensate from the condenser is then returned to a pump that pumps the water to the appropriate pressure required by the boiler, which heats the water converting it back to steam. Depending on the exit pressure of the steam, the steam turbines can be categorized as back pressure turbines and condensing turbines. Condensing turbines are usually used for central power generation while non-condensing and extraction turbines are used for CHP purposes. The non-condensing turbine exhausts its entire flow of steam for process heating at a pressure at or above atmospheric pressure (i.e., 50, 150 and 250 psig are commonly used). Power generation capabilities are reduced when exiting steam pressures are increased (US EPA 2008b). Variations to the original Rankine cycle such as the organic Rankine cycle (ORC) use organic working fluids such as iso-butane or propane in place of water (WADE 2003).

Unlike gas turbines and IC engines, steam turbines generate electricity as a byproduct of heat (steam) generation with no direct conversion of fuel to electricity (US EPA 2008b). This gives the steam turbine the flexibility to operate with a large variety of fuels. These include all types of coal, wood, wood waste and agricultural byproducts (US EPA 2008b).

Steam turbines capacities are available in the range of a few hundred kilowatts to above 1,000 MW. Overall system efficiencies for condensing turbines range from 20% to 38%, and for

[62] Supplemental firing is possible due to the high oxygen content in the exhaust gases of gas turbines, which is not always the case when considering IC engines.

backpressure turbines are between 7% and 20%. However, the efficiencies of these turbines are limited by Carnot's efficiencies and are reduced by mechanical efficiencies, steam losses, and imperfections in the flow path (Orlando 1996). Efficiencies can be improved by increasing the energy content at the turbines inlet steam conditions, increasing turbine speed, increasing the number of turbine stages, and finally increasing inlet pressure and decreasing condensing pressure. The use of combined cycle[63] also improves the efficiencies of the steam turbines. Steam turbines are rated at ISO conditions[64] with optimum performance occurring at 95% of the rated load (Orlando 1996). Unlike combustion turbines, steam turbines can require lengthy start-up period which includes a warm-up of boilers and can take several hours (Orlando 1996).

The advantages of steam turbines include extremely long life and reliability, production of large amount of thermal energy and the use of a wide range of possible fuels. Disadvantages include low electrical efficiency, slow start-up time, poor partial load performance and added expenses due to high pressure boilers and other equipment (Wu and Wang 2006). As a result steam turbines are more popular in large central plant utilities or industrial CHP than in distributed generation applications (WADE 2003, Wu and Wang 2006). Steam turbines are also advantageous where steam requirements are relatively high to the power needs within the facility (US EPA 2008b). Applications of steam turbines include in industrial processes, combined cycle power plants and district heating systems.

Heat recovery options for steam turbines include providing high-grade heat for industrial process plants and providing heat for district heating systems. Heat recovery possibilities for steam turbines are greatly enhanced with the possibility of multiple extraction ports which allow the turbine to satisfy different steam requirements and multiple induction ports which allow the use of process by-product steam for power generation (Orlando 1996).

2.5.3 Heat Recovery Options and Devices for CHP Systems

Several sources were reviewed to compile the literature on heat recovery options and devices. These include discussions provided in the review on sustainable on-site CHP systems (Meckler and Hyman 2010), CHP class notes provided by Caton (2010), design guide for cogeneration by Orlando (1996), and several other sources which are cited accordingly.

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[63] In combined cycle applications, a gas turbine is the main drive for power generation. To improve the overall efficiency, the exhaust heat from the gas turbine driven is used to produce steam for generation of additional electricity by a steam turbine (Siemens 2012).

[64] ISO Standard 14661, 2000.

According to Foley (2010), heat recovery options are dependent on the type and quality of thermal energy[65] required by the facility's needs and the type of prime mover used in the CHP system. The types of thermal energy typically required for a facility include high and low pressure steam, hot water and chilled water. When considering the selection of an appropriate prime mover, it should be noted that IC engines generate hot water[66] as well as exhaust heat, gas turbines generate high volumes of high temperature exhaust, and steam turbines generate large quantities of steam that is available for heat recovery. Another factor as noted by Foley (2010) is that the quality[67] of thermal energy required by the facilities has a significant effect on heat recovery capabilities especially in the case of IC engines where thermal energy is obtained at different temperatures. For example, in case the facility requires high pressure steam or high temperature water, then heat recovered from the jacket loop coolant can at best be used for preheating, hence only meeting the loads only partially.

The recovered heat can typically be extracted using a heat exchanger, which depends on the prime mover used. Knight and Ugursal (2005) in a review of residential CHP systems pointed out that heat recoveries from IC engine based CHP systems cannot be made directly to a building's heating medium because of problems associated with pressure, corrosion, and thermal shock. Therefore, shell and tube heat exchangers or plate heat exchangers are recommended to transfer heat from the engine cooling medium to the building's heating medium.

For exhaust based systems, Heat Recovery Steam Generators (HRSGs) are typically used to produce steam. In order to properly size an HRSG, the exhaust gas temperature, the minimum allowed exhaust stack temperature, the exhaust gas mass flow rate, the required steam pressure and flow rate and the condensate return temperatures need to be specified (Hyman and Landis 2010). The exit temperature of the heat recovery device should be no less than 250°F–300°F to avoid condensation and formation of acid in the exhaust stack (Caton 2010) with higher exit temperatures offering more flexibility for part-load operations without allowing the formation of condensate in the exhaust stack (Foley 2010). Exhaust heat can also be recovered through gas-to-air heat exchangers or used directly to drive a solid or liquid desiccant. In certain cases the recovery system is built in directly into the thermal conversion devices such as hot

[65] Recovered thermal energy can be categorized into low, medium and high temperature.
[66] Hot water from the IC engine is obtained by the recovery of heat from the jacket loop coolant. In a jacket loop the coolant is circulated in the jacket of the IC engine to maintain efficient operating temperatures and to avoid engine failure due to high temperatures in the combustion chambers of the IC engine.
[67] The quality of thermal energy refers to the temperature of thermal energy available from the prime mover.

water absorbers used in absorption chillers. Depending on the use and relative demands of the recovered heat, designs for CHP systems typically split potential heat recovery from jacket loop and exhaust heat recovery into two separate systems, with lower quality heat requirements such as space and water heating being met by heat recovered from the jacket coolant and the higher quality heat requirements of the facility such as operation of absorption chillers being met by the higher quality heat obtained from the engine exhaust.

Foley (2010) notes that is important to consider the prime mover operating conditions. On operating the prime mover at part-load conditions, the volume of heat recovery is predictably reduced in a linear pattern. Degradation in temperature of recovered heat should also be considered and will have an impact on reducing the output from the thermal conversion device.

Finally, in order to maintain high usability, CHP systems may need to consider heat rejection options to systems other than those used for heating only. Thermal technologies are ideal for this purpose. These technologies can potentially utilize the heat rejected from CHP systems to provide cooling. Thermal technologies include absorption chillers, adsorption chillers, desiccant dehumidifiers and steam turbine -driven chillers[68]. These technologies can be harnessed for a variety of cooling applications which include space cooling and refrigeration. Absorption, adsorption and desiccant dehumidifier technologies are discussed in detail in the following section on tri-generation.

2.5.4 Applications of CHP Systems

Distributed power utility seems to have evolved in four directions – Large scale electric power generation systems (sizes ranging 400 to 1,000 MW), district energy and industrial / agricultural[69] CHP systems (sizes ranging 3 to 50 MW), CHP in buildings(sizes in the range of 50 kW to 3 MW); and micro-CHP systems (sizes in the range of 3 to 20 kW) (Maor and Reddy 2010). Of particular interest to this study is the implementation of CHP in buildings and in residential communities. The following sections provide examples of the variety of such applications. Several sources were referenced to compile the review on applications of CHP in buildings and residential communities. These include discussions provided in the review on sustainable on-site CHP systems (Meckler and Hyman 2010), studies on the performance of

[68] Steam turbine chillers are vapor-compression chillers that are driven with steam turbines instead of electric motors, which are typically used to drive conventional vapor-compression chillers.
[69] The industrial sector is characterized by continuous operation of the facility and coincident electrical and thermal loads, are are excellent candidates for the implementation of cogeneration systems. However, the cogeneration applications in this sector will not be described in this study.

CHP systems in grocery stores which include research performed by Prosser and Maidment (1999a), Maidment et al. (1999b), Maidment et al. (2001), Maidment and Tozer (2002), Sugiartha et al. (2009), and Ge et al. (2009) and research on residential CHP technologies performed by the group led by Beausoleil-Morrison which culminated in several publications of the International Energy Agency's Annex 42.

2.5.4.1 CHP in Buildings

In order for CHP systems to be successful in buildings, the following conditions should be met (Maor and Reddy 2010):

- A good coincidence between electric and thermal loads[70],
- The building thermal energy requirements in the form of hot water or steam,
- Electric demand to thermal demand ratios ranging from 0.5 to 2.5,
- Cost difference between electricity (total cost) and natural gas (total cost) greater than $12/MMBtu,
- Moderate to high operating hours, and
- Electric power quality and reliability are important considerations.

Keeping the above mentioned factors in mind, good candidates for CHP applications include commercial and institutional buildings; hospitals and other healthcare facilities; hotels; universities and educational facilities; supermarkets; large residential buildings or complexes; research and development and laboratory buildings; large office buildings; military bases and district energy systems. It is important to note that the larger sized building and facilities allow lower initial cost of CHP in terms of $/ft^2 of built area and larger annual savings.

Applications in grocery stores are particularly interesting. The simultaneous demand of electricity and thermal energy[71] in the grocery store makes this building type an ideal candidate for CHP application. Several studies in the past have undertaken to explore the potential of implementing CHP in grocery stores especially in conjunction with thermal technologies[72] for cooling. These include studies by Prosser and Maidment (1998), Maidment et al. (1999), Maidment et al. (2000), Maidment and Tozer (2002), Sugiartha et al. (2009), and Ge et al. (2009).

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[70] A good coincidence between thermal and electrical loads is not always the case for many commercial buildings where there is a strong dependency on seasonal variations and scheduled operating hours.
[71] If considering the option of absorption refrigeration.
[72] Details are provided in the section on thermal- driven cooling technologies.

Prosser and Maidment (1998) investigated the viability of using CHP and absorption cooling in cold storage. The system investigated provided both power and chilled glycol to a cold storage facility. In its conclusions, the study reiterated the need for a high utilization time in order to make the CHP economically viable. In the study a source energy savings of 10% was observed when comparing the energy consumption against a conventional heating and cooling systems. The study also noted that an increase in the COP[73] of the absorption chiller from the 0.35 to 0.4 yielded a significant reduction in source energy consumption which is in the range of 12- 15%.

Maidment et al. (1999a) theoretically investigated the viability of implementing a CHP system with absorption cooling in a grocery store in the UK using mathematical models. Their study concluded that the installment of CHP with absorption cooling for medium temperature refrigeration resulted in source energy savings of 20%, with a payback period of 5 years. The study also pointed out that low temperature refrigeration systems using absorption refrigeration was not practical due to the requirement of extremely high temperatures to run these systems, which were not possible with the commercial CHP systems available at the time of the study[74].

In another research paper Maidment et al. (2001) investigated the thermodynamic and economic viability of implementing a gas engine driven CHP system in a grocery store in the UK. The study explored the integration of the gas engine in a novel system which included the operation of a direct-drive screw compressor to operate the refrigeration system of the store and using the heat rejected from the engine for space and service water heating purposes. Results indicate that a gas engine may be used in this configuration to provide both heating and cooling in a supermarket. The study also reported a pay-back period of 4.2 years.

In yet another study Maidment and Tozer (2002) reviewed the potential of a number of CHP options in a grocery store in the UK involving the use of different cooling and engine technologies. The study used validated mathematical models to carry out the analysis. The five options examined the resulting energy usage in the grocery store. The option implementing a standard LiBr/Water chiller provided optimum energy savings. The study concluded that, on a short term basis, CHP could offer source energy savings of 15 %. However, the study also noted

[73] Coefficient of Performance (COP) is a unit of efficiency for heating and cooling systems. COP can be defined as the ratio of the heating or cooling provided by the energy consumed.

[74] The current availability of improved absorption chiller technologies such as improved COPs and direct fired absorption chillers can potentially improve the performance of low temperature absorption chillers. However, these chillers are still not available for small scale commercial purposes.

that on a long term basis the CHP options may have to compete against more efficient grid-generated efficiencies.

Sugiartha et al. (2009) reviewed the use of CHP technologies in retail food facilities in terms of energy efficiency, economic and environmental performance. The analysis concluded that benefits for both energy reduction and environment preservation could be obtained by implementing CHP systems in grocery stores. It was also concluded that operating the CHP facility continuously at full electrical output mode provided higher energy savings than when operating at part-load electricity generating conditions. The study also noted that the economic viability was sensitive to the relative prices of natural gas and grid electricity, with payback periods improving as the gap increased between natural gas and electricity prices.

Ge et al. (2009) investigated the performance CHP system using a test rig with a validated simulation model. The test rig contained a power component consisting of a micro-turbine, a refrigeration unit consisting of an absorption chiller, and a supermarket section consisting of a display case cabinet. The nalysis was performed for different operating and design conditions. The analysis aimed to establish optimum operating conditions given the design conditions for typical grocery stores in the UK. Using the test rig validated simulation model, the study was able to provide an initial design for the application of the tri-generation systems in a typical grocery store in the UK.

From the studies described above it can be concluded that implementing a CHP system with the option of absorption cooling is indeed a viable proposition to conserve source energy in a grocery store. However, it should be noted that this viability depends on several factors such as the run-time of the CHP system, the operating conditions (i.e. full-load electric versus part-load electric operation) and the temperatures and efficiencies at which the absorption chillers operate. The viability of implementing a CHP also depends on a favorable economic analysis with low payback periods. It is also important to note that the studies mentioned above were primarily carried out for cold climates. In hot and humid climates the challenges are different. For example, not as much energy is needed for space heating, whereas dehumidification and cooling becomes a major concern. Hence priorities for absorbing the thermal energy rejected from the prime mover has to be considered in terms of assessing the performance of cooling and dehumidification technologies.

2.5.4.2 CHP in Residential Communities

Although primary used in industrial and commercial scenarios, CHP technologies can also be used to power residential and mixed land-use communities (Dorer 2007; Beausoleil-Morrison 2008). Residential communities can either be served by large district energy systems that provide energy for the entire community or by micro-CHP systems that are suited for individual residential applications. However, in the case of residential and mixed land-use communities the design and implementation of a viable CHP system poses a significant technical challenge due to the potential non-coincident thermal and electrical loads of residential/ mixed land-use energy usage (Knight and Ugursal 2005). Such non-coincident loads require energy storage, since the energy production is not always synchronized with the energy consumption patterns. Although the storage of energy (both electrical and thermal) is important, a complete review of the available storage technologies is outside the scope of this study. Another difficulty seen in residential CHP systems is how to achieve high annual utilization of the heat rejected during the summer months (Maidment et al. 2002). In the past, this has restricted the use of such systems to heating dominated climates.

In district heating and cooling systems, a central plant generating electricity is usually located on-site. The waste thermal energy exhausted from the central plant is utilized to power absorption chillers for cooling and to generate hot water and steam for heating. Chilled water and hot water/steam are then transported to the entire community via a network of pipelines and pumps. The advantage over electricity supplied by a distant power plant includes improved efficiency of the entire power generating system, reliability, safety, and in many situations improved economics. Currently in the U.S., not many facilities exist that service residential communities. Furthermore, such facilities are usually most viable in heating dominated climates where there is a greater requirement for heating over a sustained period and hence a higher degree of utilization of the CHP system.

More recently, the use of CHP systems has also been demonstrated for individual single-family and multi-family units (Knight and Ugursal 2005). IC engines, fuel cell and Stirling engine based CHP technologies are typically used (Knight and Ugursal 2005).

The main difficulty in justifying the viability of using CHP in hot and humid climates is providing a heat demand during the summer months. Fortunately, with a near constant demand for cooling power all year round, grocery stores provide an ideal sink to absorb the waste heat from the CHP system through the use of technologies such as absorption refrigeration and desiccant cooling technologies. The next section examines the various approaches that can be adopted by grocery stores to implement these technologies.

2.5.5 Technical Design Issues for CHP Systems

Proper CHP sizing is critical to the viability of installing and operating a CHP system. For example, if a CHP is oversized, it is likely that the facility will not fully be able to utilize the waste heat, heat dumping will occur, which causes overall system efficiencies to be low, and therefore economic expectations will not be utilized. Therefore, the approach to the proper sizing of a CHP system includes estimating the quantity and quality of the building loads and establishing the building load profiles; selection of an appropriate prime mover and heat recovery equipment to provide for the above mentioned loads; and finally optimizing the operation of the CHP system by selecting an appropriate size and operating strategy. Several sources were referenced to compile the review on heat recovery options and devices. These include discussions provided in the review on sustainable on-site CHP systems (Meckler and Hyman 2010), CHP class notes provided by Caton (2010), and design guide for CHP by Orlando (1996). The Handbook on Energy Management compiled by Turner (2006) was found to be useful in the review for CHP screening technologies as well.

2.5.5.1 Estimating Building Loads

In order to properly size the CHP system it is essential to obtain and characterize the buildings electric and thermal loads. This characterization includes identifying the base electric load and multiple thermal loads as well as to understand the type and quality of thermal loads (Foley 2010). Historical energy use data is either obtained or calculated to estimate electrical and thermal usage. To properly size the CHP system, these loads are needed on an hourly, daily, monthly and yearly basis (Caton and Turner 1997). Information regarding monthly and annual building energy usage can be obtained from utility bills. In addition, computerized load management systems can be a good source of daily and hourly energy consumption profiles (Orlando 1996). Alternatively, building load data and resultant energy can be obtained from a calibrated simulation model of the building (Orlando 1996).

It also becomes important to evaluate the daily profile of energy use (i.e. energy use versus time) with the relationship between electric energy demand and coincident thermal energy demand being especially critical[75] (Orlando 1996). Under typical circumstances, CHP favors facilities that have coincident electrical and thermal loads or large, constant thermal loads.

Finally, the heat-to-power ratio of the facility needs to be calculated. This number will eventually be used to determine the prime mover ideal for the facility (Caton 2010).

2.5.5.2 Selecting and Sizing the Prime Mover

Since perfectly matching the building loads with the performance of the CHP system is not always possible, matching the thermal-electric ratio of the CHP system to that of the buildings energy consumption profile should be considered. Steam turbines and combustion gas turbines have an advantage of generating higher amounts of thermal waste than IC engines, which implies the ability to generate high pressure steam. This makes steam and gas turbines attractive in industrial facilities with high thermal energy needs. On the other hand, in commercial applications, high electrical generation efficiency is typically more desirable, which makes IC engines more applicable and more cost effective for these applications. In addition, in contrast to industrial applications, which need high pressure and / or low pressure steam, commercial buildings typically need hot water or low pressure steam for space heating and service water heating, which can be effectively met by IC engines.

A rule-of-thumb is recommended when selecting an appropriate prime mover for the CHP system. The rule-of-thumb recommends the use of engines when the heat-to-power ratio of the facility is 0.5 to 1.5, gas turbines are recommended when the heat-to-power ratios of the facility is between 1.0 to 3.0, and steam turbines are recommended when the heat-to-power ratio of the facility is between 4.0 to over 10.0 (Caton 2010).

CHP systems can be sized using several criteria. Turner (2006) summarizes the different types of CHP design options. These include sizing for isolated operation[76] ; sizing using electric base load; sizing using thermal base load; sizing using intermediate loads; and sizing using peaking loads.

Using thermal energy storage can help to optimize the efficiency of a CHP system, especially when there is a high variation in electric and thermal loads (de Wit 2007). Thermal energy storage provides a transfer and retrieval of thermal energy to a storage medium (Foley

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[75] For example, a facility that has high electricity use during the daytime and little electricity use at night, and has a high thermal usage at night with little thermal use during the day is usually a poor candidate for cogeneration unless a thermal storage strategy is incorporated.
[76] In this option the site is stand-alone, with no connection to the electrical grid.

2010). Media such as water, ice, rock, brick, thermal oils and chemicals are good candidates for thermal storage (Foley 2010). Methods include storing excess electric power[77] or thermal energy[78] in form of chilled water or ice when thermal demands exceeds the coincident power demand, and storing excess thermal production of heat when power demand exceeds the heat demand. In either case, cool or heat storage must be able to productively discharge most of its energy before it is dissipated to the environment; and finally sell excess power or heat through approved protocols to a user outside the host facility (ASHRAE 2008).

Finally, to optimize the CHP system, feasibility analysis can be performed by three types of design tools which include manuals and nomograms, software screening tools and hourly simulation programs. These tools will be discussed in the section on simulation programs.

2.6 Thermal-Driven Cooling Technologies

Deng et al. (2010), in their review of thermal-driven cooling technologies for CHP systems, concluded that the thermal-driven cooling technologies are viable alternatives to conventional vapor compression operated cooling technologies because these technologies provide for air-conditioning, refrigeration, dehumidification and enhancement of the overall thermodynamic efficiency of the CHP systems as well as meet the demand for energy conservation and environmental protection. The Combined Cooling, Heating and Power (CCHP) achieved by using thermal-driven technologies in combination with CHP systems is also known as tri-generation, Building Cooling Heating and Power (BCHP), or Integrated System Design (ISD). The steady demand of cooling energy year around in grocery stores for refrigeration and space cooling justifies the viability of using CHP in combination with thermal-driven cooling technologies (Ge et al. 2000; Bassols et al. 2002).

The most common methods to produce cooling using a heat source use sorption technology. Sorption may either be adsorption or absorption depending on the nature of the process. Low-grade or medium-grade heat sources are typically used in the regeneration process of the sorption material. Different types of absorption chillers are available based on thermal energy sources. These include single and two stage hot water fired LiBr absorption chillers, single and two stage steam fired LiBr absorption chillers, two stage exhaust fired LiBr absorption chillers, single and two stage hot water or steam fired NH_3 absorption chillers, two

[77] Electric power can be converted to chilled water by employing vapor-compression chillers.
[78] Thermal energy can be converted to chilled water by employing absorption chillers.

staged exhaust fired NH_3 absorption chillers and hybrid single / two staged LiBr and NH_3 absorption chillers (Foley 2011). In addition, sorption chillers have fewer moving parts than chillers that use a compressor. Therefore these chillers are quieter than electric chillers, use almost no electricity and can be used for waste heat recovery applications and finally, because they do not contain chlorofluorocarbon (CFC) refrigerants they do not have an ozone-depleting potential (Ziegler,1999; Atta 2006; Caton 2010; Deng et al. 2010).

The components of a typical CCHP system include a prime mover, a heat recovery component and a sorption chiller. The prime mover is driven by fuels such as natural gas. The resultant mechanical work obtained from the combustion process is converted to electric power. Simultaneously, the sorption chiller[79] is driven by the heat recovered from the exhaust gases or hot water generated by the prime mover. Waste heat from various prime movers can be in the form of steam, hot water or exhaust gas falling into different temperature categories. Thermal-driven cooling technologies, on the other hand, have their own optimum operating temperatures. Therefore, for the operation of a CCHP system to be successful, the temperature requirements of thermal-driven cooling and other heat recovery technologies are to be well matched with the operating specifications of the prime movers.

Besides sorption cooling technologies, desiccant cooling also make use of the waste thermal energy available from the prime mover. This technology works by incorporating desiccant dehumidification in the cooling unit. In such technologies, latent cooling loads are removed by the desiccant, the traditional expansion-type cooling unit removes the sensible loads and reheating of air is avoided (Sweetster 1996). In this case, thermal energy in form of low grade heat from the prime mover can be used to dry and reactivate the desiccant (Katipamula and Brambley 2010).

Detailed information on CCHP technologies can be obtained from Zeigler (1999), Srikhirin et al. (2001), and Heroid et al. (1996) for absorption refrigeration; Wang et al. (2005), and Critoph et al. (2005) for adsorption refrigeration; Wurm et al. (2002), Daou et al. (2006), Öberg et al. (1998) and Sweetser (1996) for desiccant cooling. The reviews by Wu (2006) and Petchers (2003) are recommended for an assessment of current technical characteristics of CCHP systems. For a review of thermal-driven cooling technologies for combined cooling, heating and power systems the paper by Deng et al. (2010) is considered as an important reference. Finally,

[79] The working of the sorption chiller is described in the sections on LiBr/ Water , Water/NH_3 chillers and adsorption chillers.

information regarding the specifications and performance of absorption cooling can be obtained in an application guide for absorption cooling and refrigeration by Dorgan et al. (1995).

2.6.1 LiBr / Water Absorption Chillers

The working of an absorption chiller is based on the relation between the absolute pressure and the boiling point of the refrigerant[80] which in the case of a LiBr / Water chiller is water (Foley 2010). The following description of the operation of the LiBr/Water absorption chiller has been adopted from the ASHRAE Handbook –Refrigeration (2006). A schematic layout of a single-effect indirect-fired absorption chiller is provided in Figure 2-8. During the operation of the LiBr/Water absorption chiller, heat is supplied to the *generator* in form of hot fluid, steam, or exhaust gas causing the dilute absorbent solution to boil. The desorbed refrigerant vapor (water) flows to the *condenser*, where it condenses. Both boiling and condensing occur in the same vapor space at a pressure of about 0.9 psia. Dorgan et al. (1995) point out that heat rejection for absorption refrigeration is usually 1.2 to 2 times more than a corresponding vapor-compression systems. The condensed refrigerant then enters the *evaporator*, in which the liquid refrigerant boils as it get in contact with the outside surface of the tubes that contain a flow of water from the building load. The dilute absorbent solution, which is weak in absorbing power, that enters the generator increases in concentration as it boils and releases water vapor. The resulting strong absorbent solution leaves the generator and flow down a *solution heat exchanger*, where it cools as it heats a stream of weak absorbent solution passing on the other side of the solution heat exchanger on its way to the generator. The cooled strong absorbent solution then flows to a solution distribution system located above the absorber tubes and drips over the absorber tubes in the *absorber*. The absorber and the evaporator share the same vapor space at a pressure of about 0.1 psia. This allows the refrigerant vapor, which is evaporated in the evaporator to be readily absorbed in the absorbent solution flowing over the absorber tubes. The absorption process releases heat of condensation and heat of dilution, which are removed by the cooling water flowing the condenser tubes which run through the absorber.

[80] As the pressure increases the boiling point is raised. As the pressure decreases the boiling point is lowered.

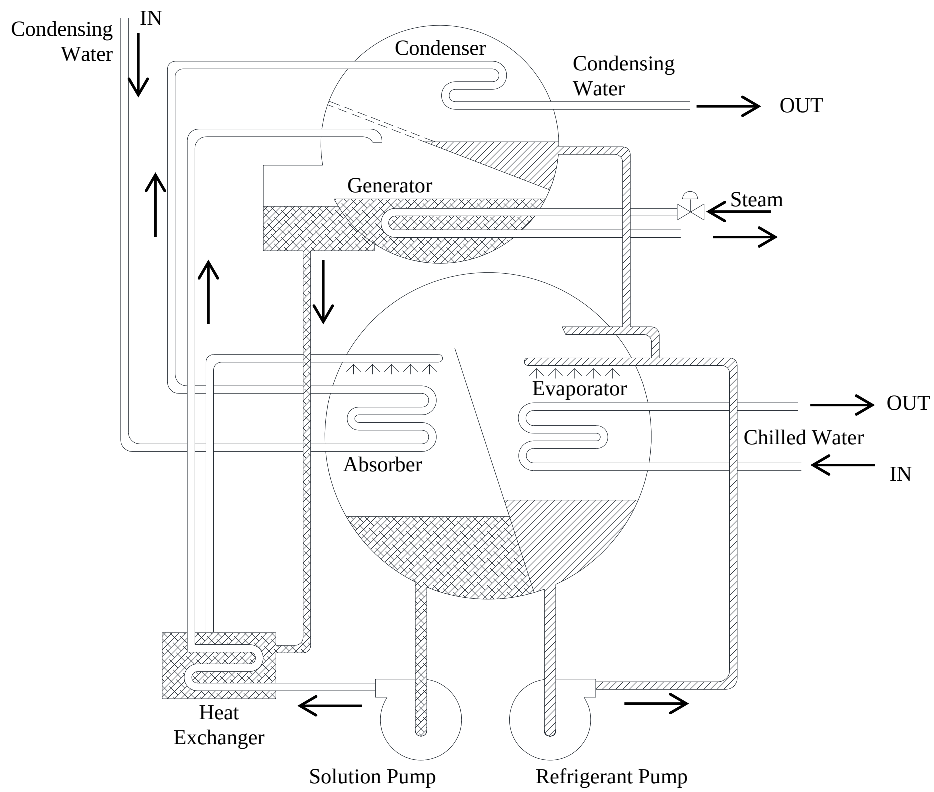

Figure 2-8: Single-Stage Absorption Chiller (Source: ASHRAE Handbook - Refrigeration 2006)

The evaporator temperature of these systems is in the range of 40°F – 50°F making these systems suitable for air-conditioning or selected types of product cooler applications. The required heat source temperature for the LiBr/water chillers range is from 175°F – 440°F depending on the quality of heat and the number of times this heat is supplied to the generator. Depending on the number of times the heat supply is utilized within the chiller, absorption chillers are divided into single-effect[81], double-effect and triple-effect[82] chillers. Currently, single and double-effect LiBr/Water absorption chillers are commercially available. Triple-effect absorption chillers are in the experimental stage of development and are available for small commercial installations. Single-effect chillers work with lower temperatures of waste heat when

[81] In a single-effect cycle, the heat rejected from a the generator is not reused (Srikhirin et al., 2001)
[82] In a double-effect and triple-effect cycles, the heat rejected from a higher-temperature stage in the generator is re-used as heat input in a lower-temperature stage for the generation of addition cooling effect in the low-temperature stage (Srikhirin et al., 2001).

compared to the requirements of double and triple-effect chillers. Given that high-pressure steam is not needed in most commercial buildings, the ability to better utilize low-grade heat source makes single-effect systems attractive choice over the use of double-effect absorption systems (Ryan 2004).

In addition, absorption chillers can also be categorized as indirect-fired [83] and direct-fired[84] depending on the quality of heat supplied to the generator. The available cooling capacity for absorption chillers is in the range of 20 kW – 11,630 kW (i.e., 6 tons to 3,300 tons). The thermal COP is in the range of 0.5 to 0.7 for single-effect chillers, 1 – 1.4 for double-effect chillers and 1.4 to 1.7 for triple -effect chillers. (Ziegler 1999; Srikhirin et al. 2001; Deng et al. 2010). Although the use of direct-fired eliminates the need of extra heat recovery equipment that generates steam or hot water, Dorgan et al. (1995) point out that the cons of direct-fired chillers include the drop in operating efficiency if fouling agents in source heat stream are allowed to build up in the absorption machine.

The main drawbacks of the LiBr/Water absorption chillers include corrosion from the LiBr solution at higher temperatures; operation of the system at vacuum pressures[85] that can cause problems of air-entrainment if the system is not well sealed; a narrow solution concentration range of the LiBr/Water solution that is limited by crystallization, which limits the absorber temperature; and finally high first costs and maintenance costs (Deng et al. 2010).

2.6.2 Water /NH_3 Absorption Systems

The basic principal of operating Water/NH_3 systems is similar to that of the LiBr / Water systems with the difference being in the addition of a rectifier[86] to the absorption process. The Water/NH_3 system is used primarily for industrial refrigeration with the evaporator temperatures as low as -75°F. Some units also offer an evaporator temperature in the medium temperature range of 40°F to 50°F. The heat source temperature for low cooling temperature technologies is in the range of 212°F – 392°F. The heat source for medium cooling temperatures is in the range

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[83] Indirect-fired chillers utilize hot water or steam to provide heat in the generator of the absorption chiller.

[84] Direct-fired absorption chillers utilize thermal energy directly combustion of natural gas or other fuel eliminating the need for additional heat exchangers. These chillers can be modified to accept to exhaust gases from prime movers (Deng et al. 2010).

[85] The operating pressures in the generator and evaporator section of the absorption chiller were noted to be 0.9 psia. While the operating pressures in the absorber and evaporator section of the absorption chiller were noted to be 0.7 psia.

[86] The Water/ NH_3 absorption process utilizes a rectifier to separate the ammonia from water in the absorption process. Without a rectifier, the water content would join the ammonia water to condense blocking the throttling valve in the chiller. In addition, the water entering the evaporator could raise the chilling temperatures (Deng et al. 2010).

of 176°F – 248°F[87]. Ammonia has a lower vaporization heat than water. Hence Water/NH_3 units incorporate a rectifier to remove water from the refrigerant vapor. The introduction of the rectification process lowers the COP of the basic Water/NH_3 cycle than that of the corresponding LiBr/Water cycle operating under similar temperature conditions. For lower cooling temperature ranges (-75°F) the Water/NH_3 system has a thermal COP in the range of 0.25 – 0.6, while for medium cooling temperature ranges (40°F – 50°F) the Water/NH_3 system demonstrates a thermal COP of 0.5 – 0.6. The available cooling capacity for low cooling temperature systems are 10 – 6,500 kW (i.e., 3 tons – 1,850 tons), while the available cooling capacity for medium temperature are 10 – 110 kW (i.e., 3 tons – 30 tons) (Srikhirin et al. 2001; Ryan 2002; Deng et al. 2010).

The advantages of the Water/NH_3 chillers over LiBr/Water units is the low freezing point of pure ammonia (-75°F) providing lower refrigeration temperatures. This trait makes the Water/NH_3 system primarily applicable to the refrigeration industry (Ziegler 1999; Deng et al. 2010). Other advantages over LiBr/Water chillers include a more compact unit that is possible because ammonia is a high-pressure refrigerant with a low specific volume; no problems with crystallization; elimination of an air purge system due to operation under positive pressure; and the ability of locating the unit outdoors (Deng et al. 2010). The limitations of the Water/NH_3 absorption technology include the high pressures of operation, the toxicity and flammability of ammonia, the lower efficiency compared with LiBr/water units (due to necessary rectification), incompatibility of ammonia with the use of copper and brass materials resulting from corrosion, and relatively high investment costs (Deng et al. 2010).

2.6.3 Adsorption Chillers

Recently there has been an interest in developing adsorption cooling technologies. The main difference when compared to absorption chillers is that adsorption systems have two or more adsorbent beds in order to provide continuous operation. Each of the adsorbent beds alternates between being the generator and absorber function due to the difficulty of transporting solid sorbent between the two components (Deng et al. 2010). A schematic layout of the basic cycle for the adsorption refrigeration is presented in Figure 2-9 (a. Basic absorption refrigeration system[88]. b. continuous absorption refrigeration system). Low grade heat sources (140°F –

[87] It should be noted that a higher heat source temperatures are required to produce lower output temperatures.
[88] The basic adsorption process is intermittent due to the sequence of the adsorption process.

250°F) can be used to drive the chillers (Wang et al. 2005). Also, neither a liquid pump nor a rectifier for the refrigerant is required in the operation of adsorption chillers. Finally, corrosion is not an issue due to the composition of the working pairs that are typically used in the operation of the chiller. Hence, an adsorption chiller is considerably simpler, has a quieter operation, and requires no lubrication of machine parts and has less maintenance (Deng et al. 2010). The disadvantages include a somewhat lower COP [89]compared to absorption systems operating under the same conditions. Other disadvantages as noted by Wang et al. (2005) are size and high first costs.

2.6.4 Desiccant Cooling Systems

As mentioned in the earlier sections of this literature review, in hot and humid climates the dehumidification of grocery stores is a major concern. Using desiccant dehumidification in conjunction with cooling technologies has the advantage of removing latent heat without using CFCs, HCFCs and HFCs while lowering electricity usage. Solid desiccants that are commonly used include lithium chloride, calcium chloride, silica gels, zeolites or molecular sieves and aluminum oxides (Deng et al. 2010). Additionally, solid desiccant systems are compact and are less subject to corrosion and carry-overs (Deng et al. 2010). For such systems the required regeneration temperatures in the range of 140°F – 300°F depending on the desiccant material used (Deng et al. 2010). The use of compound adsorbents[90] as desiccants further reduce regeneration temperatures, thereby facilitating the use of low grade heat sources (Aristov et al., 2007, Aristov et al., 2008, as cited in Deng et al. 2010). Optimum system configurations such as staged regeneration[91] (Collier and Cohen, 1991, as cited in Deng et al. 2010) and isothermal dehumidification[92] can be used to improve the performance of the desiccant systems (Meckeler, 1989, as cited in Deng et al. 2010).

[89] Lower COP is due to low thermal conductivity of the absorbent, relatively low cycle mass and inability to apply internal solution heat exchange due to non-fluidity of solid absorbent (Wang et al. 2005).
[90] The use of compound adsorbents such as silica gel impregnated with LiCl or $CaCl_2$ can obtain better dehumidification at lower driven temperatures (Deng et al., 2010).
[91] In staged regeneration, the regeneration section of desiccant is divided into two parts. The latter fraction subjected to the desorption air stream which is heated to desired temperatures.
[92] Isothermal dehumidification involves the dispersal of water vapor to the environment after having been created by boiling water, in the process increasing the temperature of the surrounding air. The process requires input from an external source of energy which in this case is the hot exhaust gas from the CHP system.

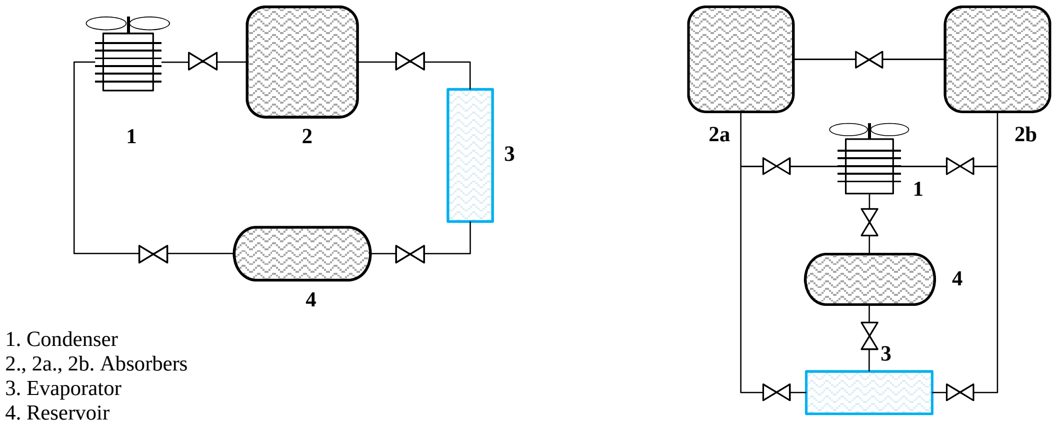

Figure 2-9: Conventional Adsorption Refrigeration Systems (Source: Deng et al., 2010)

Desiccant systems integrate well with CHP systems. In such arrangements, the waste heat in form of either hot water, or a steam heat recovery system, or by using direct exhaust gases from the prime mover of the CHP system heat is delivered to the desiccant process. Advantages of such system include significant energy savings and CFC-free characteristics are the (Deng et al. 2010). Disadvantages include the large size of the dehumidification equipment, high first investment costs, excessive fan energy requirements, poor dynamic properties, and in some cases the exhaust gas of CHP being too hot for direct use without dilution (Deng et al. 2010). In addition, Deng et al. note that the cooling effect brought about by the implementation of desiccant cooling system can be limited by humidity levels in outside air (Deng et al. 2010). A schematic diagram describing the operating principle of solid desiccant cooling is provided in Figure 2-10.

For liquid desiccant systems, possible configurations include finned tubed surfaces, coil type absorbers, spray towers, and packed towers (Deng et al. 2010). Advantages include low regeneration temperature (212°F) and the ability to serve as air purifiers (Deng et al. 2010). Major problems on implementing these systems include corrosion caused by inorganic salts and carry-over of liquid desiccant into the air stream, fan and pumping requirements, high equipment costs and complexity and maintenance issues (Deng et al. 2010).

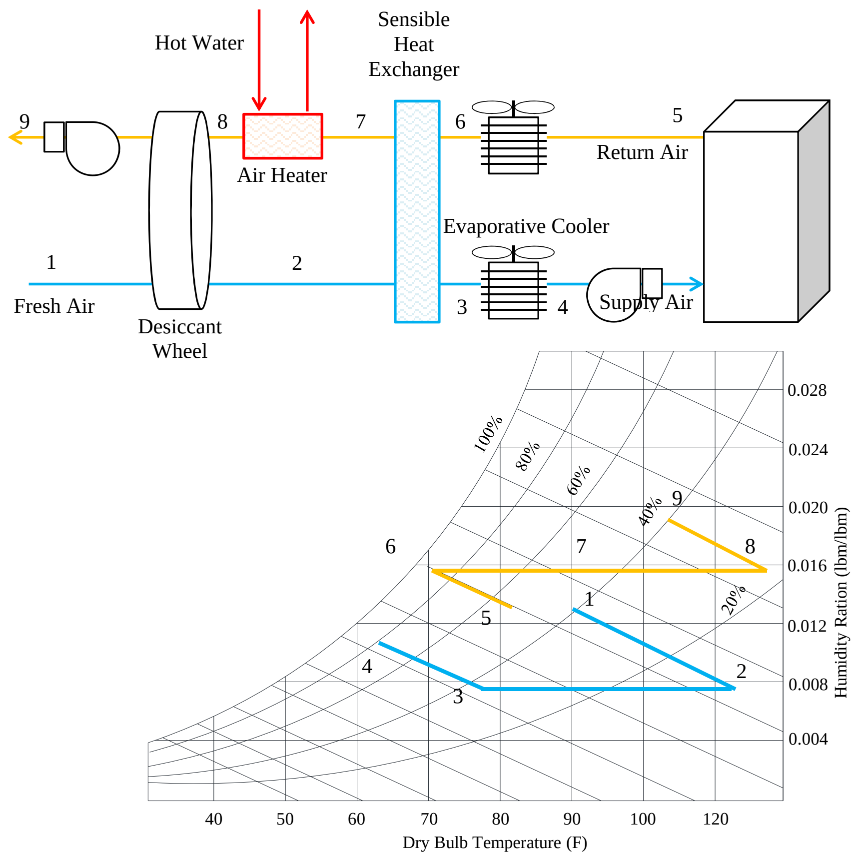

Figure 2-10: Schematic Layout of Desiccant Dehumidification System (Source: Deng et al., 2010)

From the above discussion it can be concluded that CCHP technologies have been well established. The discussion has also described the benefits and advantages of implementing these technologies. However, in recent years these technologies have lost their popularity to more efficient vapor compression technologies. By considering the interaction of building and community to approach net-zero source energy consumptions, it is proposed to re-examine the potential and viability of technologies. The next section will discuss the various options available to assess the technologies investigated in the above sections.

2.7 Simulation Programs used for the Building Energy Analysis

Numerous tools and techniques for building energy analysis have been developed since the 1960s. These tools and techniques include databases, spreadsheets and simulation programs (US DOE 2006). Certain programs specialize in limited applications such as analysis of specific building component and system, ventilation/air flow, daylighting and solar/climate analysis. On the other hand, certain other programs have the capabilities to perform whole-building energy analysis. An up-to-date comparison of the various whole-building energy simulation programs is provided in Crawley et al. (2008). Five programs for whole-building analysis were reviewed for this study. The programs include: DOE-2.1e (Winklemann et al. 1993), eQUEST (LBNL & Hirsch Associates 2004), TRNSYS (Klein et al. 2004), and EnergyPlus (Crawley et al. 2004). A brief description is provided in the section that follows. In addition, several simulation programs have been reviewed for assessing the performance and feasibility of co-generation systems.

2.7.1 Whole Building Energy Analysis Programs

DOE-2.1e is a fixed-schematic, whole-building energy simulation program that predicts hourly energy use and energy cost of the building using hourly weather data inputs. The program uses one subprogram for translation of inputs (BDL Processor) and four simulation subprograms (LOADS, SYSTEMS, PLANT and ECONOMICS) executed in sequence to perform the simulation and economic analysis. The program performs the calculation of thermal loads by using the weighting factor method[93,94]. The weighting factor either implements custom weighting factors or uses pre-determined ASHRAE weighting factors (Winklemann et al. 1993). DOE-2.1e has the capability of simulating a wide range of design features and has been widely used to evaluate the energy performance of buildings. The accuracy and consistency of DOE-2.1e has been extensively validated by tests such as those the ASHRAE Standard Method of Test for Evaluation of Building Energy Analysis Computer Program (BESTEST) (Judkoff et al. 1995). A refrigeration module is provided in DOE-2.1e that allows for the simulation of refrigerated case-work. However, several limitations to this module restrict the use of this software program to model buildings such as the grocery store, which have complex refrigeration systems.

The Quick Energy Simulation Tool (eQUEST) is software that uses the DOE-2.2 simulation program, combined with a Graphical User Interface (GUI) that includes a building creation wizard, an energy efficiency measures wizard, industry standard input defaults, and a graphical results display module (LBNL & Hirsch 2004), that can be used to perform detailed analysis of building energy performance. The DOE-2.2 program is based on the earlier versions

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[93] Weighting factors are a set of parameters that quantify how much energy that enters the room should be stored and how fast this energy should be released (Winkelmann et al. 1993).
[94] Weighting factor method, which is one of the several methods that have been used in building energy analysis, represents a compromise between the simpler methods such as the steady-state methods and more complex methods such as complete energy-balance calculations (Winkelmann et al. 1993).

of DOE-2 (Hirsch 2008). A number of other commercial versions of DOE-2 are also available including EZDOE, DOE-Plus^{TM}, and VisualDOE3.1. These programs also use the DOE-2.1 simulation program with specially developed user interfaces to simplify data input procedures. Similar to DOE-2.1e, a basic refrigeration module is provided in this simulation program too, which cannot be used in the analysis of complex refrigeration systems.

eQUEST-Refrigeration (ver. 3.61) is a version of eQUEST that has been exclusively developed for refrigeration in commercial facilities (grocery stores and food services) and industrial facilities (warehouses, food processing). The program was developed as part of the Energy Design Resources program, which is funded by the California utility customers. The program is useful in assessing the impact of refrigeration systems in whole-building simulations by providing the ability to model refrigeration systems in detail. Capabilities of the refrigeration version include modeling of components such as display fixtures, compressors, condensers, subcoolers, refrigerants etc. (Hirsch 2008). Unlike the parent eQUEST program, the algorithms in the refrigeration version are component based, allowing the users to build an entire system out of individual components. Each major device in the refrigeration system such as the refrigeration circuit, display fixture, compressor, condenser etc. can be is specified separately and subsequently connected to each other. A library of refrigeration components is included to help the user select appropriate components in order to build an entire system.

Developed by the Solar Energy Laboratory at the University of Wisconsin TRNSYS (TRaNsient SYstem Simulation Program) was originally used as a program for simulating solar thermal systems (Klein 1973). In subsequent versions, the program incorporated general HVAC and refrigeration system simulation routines. The modular structure of TRNSYS configures and assembles a series of smaller components to facilitate the simulation of complex energy systems (Klein et al. 2004). The subroutines representing the physical components are combined and solved simultaneously with a building envelope thermal balance and an air network model at each time step (Crawley et al. 2004). The TRNSYS library includes components for multi-zone building models, low energy buildings, HVAC systems, renewable energy systems, including passive solar, active solar thermal and photovoltaic systems, wind energy, fuel cells, CHP and refrigeration etc. The modular nature of TRNSYS also facilitates the creation and use of new mathematical models to the program (Klein et al. 2004), a task that cannot be easily accomplished with DOE-2.

EnergyPlus is a modular, structured code that combines selected features of BLAST and DOE-2.1e. Similar to BLAST and DOE-2.1e, EnergyPlus also uses the response factor method for transient heat transfer through multilayered opaque building envelope components. The simulation uses a heat balance method based zonal simulation. In contrast to BLAST and DOE-2.1e, EnergyPlus allows user-specified time steps of less than an hour. The program then performs the load calculation and the simulation of the response of the systems and plant for each time step[95]. This integrated solution provides more accurate space temperature predictions crucial for more accurate system and plant sizing as well as other features such as designing for occupant comfort and occupant health calculations. The program also allows users to evaluate realistic system controls, moisture absorption in the building envelope and desorption in building elements, radiant heating and cooling systems, and inter-zonal air flow, photovoltaic systems and fuel cells (Crawley et al. 2009). The EnergyPlus (ver. 7.2.0) has been updated to include simple and detailed models of refrigeration system components and refrigerants. Capabilities of the refrigeration model include calculating the electric consumption of refrigerated cases, impact of the refrigerated cases on zone cooling and humidity conditions, calculating the electric consumption of the compressor rack including auxiliary energy consumptions, and determine the total amount of heat rejected by the compressor racks condenser which potentially can be used in heat reclaim models (EnergyPlus 2012). The models account for nearly all performance aspects of typical refrigerated case equipment.

All of the programs mentioned above have different levels of capability for the analysis of refrigeration, absorption systems and co-generation systems. Being modular in structure, eQUEST-Refrigeration, EnergyPlus and TRNSYS are probably the best suited to simulate refrigeration components of whole building energy simulationDOE-2.1e and eQUEST can handle limited options of refrigerated display-cases and refrigeration systems and hence can only approximately model a complex refrigeration entity such as a grocery store.

2.7.2 Analysis of CHP Systems

CHP systems require careful evaluation in order to be deemed thermodynamically and economically feasible for a given site. For this purpose a number of computer programs and simulation models have been developed. This study summarizes the software that is used to

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[95] In contrast to EnergyPlus, DOE-2 has four simulation sub-programs (i.e. LOADS. SYSTEMS, PLANT and ECONOMICS) that are executed in sequence, with the output of one becoming the input for the next (Winkelmann et al.,1993).

evaluate CHP applications for buildings, campuses, industries and district systems. CHP assessment programs can be categorized into detailed engineering analysis models, combined thermodynamic and economic analysis models, financial analysis models and forecasting models (Baxter 1997). CHP programs can also be categorized by intended use, type of calculations, analysis duration and time step, CHP technologies, data libraries, type of CHP processes and cost and availability (Hudson 2003). This study categorizes the available CHP software considering the time step provided by the software for analysis.

To perform a feasibility analysis using only monthly thermal and electricity data, several software are available, which provide both energy as well as economic analysis of the installed CHP systems. The list of software includes (but is not limited to) CHP Ready Reckoner, RECIPRO, BCHP screening tool, Building Energy Analyzer (BEA), D-Gen Pro and RETScreen.

- CHP Ready Reckoner was developed for the Australian Department of Industry, Science and Resources by Sinclair Knight Merz (2002). The software is used primarily in the screening on industrial applications and is free of cost. In addition, the software provides a baseline comparison and has access to equipment data library for gas turbine and IC engines.
- RECIPRO is a program developed by Thermoflow, Inc. (2010) and is primarily used in the screening of small commercial and industrial CHP applications. The program is designed as an add-in module to Microsoft Excel 2000 spreadsheet. The program is used to primarily assess the performance of IC engines ranging from 70 kW to 11MW. The program can also analyze the impact of using absorption chillers instead of electric chillers.
- BCHP is a screening tool that has been at the ORNL (MacDonald 2007). The tool is primarily used in screening CHP applications that use DOE-2 simulation engine. The software provides a grid-based baseline comparison for the assessment of CHP options. Data libraries provided by this software include generation equipment, HVAC equipment, utility rates, weather, and certain specific building types.
- Building Energy Analyzer (BEA) was developed by the InterEnergy Software and Gas Technology Institute (InterEnergy 2004) and is primarily used in screening of CHP applications in commercial buildings using DOE-2 simulation engine. The program includes capabilities of quick economic analysis of cooling, heating, thermal storage in addition to the analysis of components such as on-site power generation capabilities and life cycle cost analysis.

- D-Gen Pro has been developed by the Architectural Energy Corporation and the Gas Technology Institute and is primarily used in the preliminary screening of CHP heating applications in commercial buildings. The program provides a baseline comparison of grid electricity and separate steam boiler. D-Gen Pro consists of improved on-site generation modeling capabilities (weekend operation, part load efficiency, thermal recovery, automatic generator deployment) and enhanced rate-handling capability.
- GT Pro is a program developed by Thermoflow, Inc.(2010) and is primarily used in the detailed design of industrial gas turbine applications with/without HRSG and / or combined cycles.
- The CHP model of RETScreen (NRC 2012) developed by the Natural Resources Canada (NRC) evaluates energy production and savings, costs, emission reductions, financial viability and risk for central-grid, isolated-grid and off-grid CHP (CHP) projects on a monthly basis. The software tool is Excel-based and can model a wide variety of projects ranging in size from large scale coal-fired steam turbine central plants or natural gas-fired gas turbine - combined cycle central plants connected to district energy networks, to biomass-fired distributed energy systems providing cooling, heating and power to institutional and commercial buildings and industrial facilities, to stand-alone energy supplies for commercial and institutional buildings, to small-scale remote IC engine CHP systems. The software can also be used to incorporate a variety of power, heating and cooling equipment operating under design as well as part-load conditions. The program can also analyze a wide range of renewable and conventional fuels.

When considering hourly simulation tools, several programs have been identified to do the job. These include HeatMap CHP, CHP Capacity Optimizer and HOMER.

- HeatMap was developed by the Washington State University Cooperative Extension Energy Program to assess the use of CHP systems in conjunction with district heating and cooling (DHC) and thermal storage (Bloomquist and O'Brien 2000). The program provides a detailed 3-D design simulation of both proposed and existing CHP systems using DOE-2 simulation engine. The program provides a baseline comparison with the existing grid-operated system. The program provides various data libraries, which include libraries for weather, building loads, production equipment, fuels and piping.
- CHP Capacity Optimizer (Hudson 2005) is an automated standalone spreadsheet program that assesses the cost of the CHP system provided the electrical and thermal load behavior of

the facility, the tariff structure, the price of primary fuel the operating strategy and characteristics of the CHP system, the installed capacity of the prime mover and the absorption chillers are known. Using an hour-by-hour operation simulation the program is designed to compute the optimal capacities of prime movers and chillers that will maximize the life cycle, net present value savings from the CHP system. The program has been designed to provide guidance on proper installation of properly sized prime movers and absorption chillers in commercial applications.

- HOMER (Lambert et al. 2005) a program developed by the NREL evaluates alternative off-grid and grid connected system options for a variety of applications using hourly simulations. The program allows results to be compared on economic as well as technical merits. In addition to modeling IC engine generators and microturbines, HOMER can also model other micropower systems such as wind turbines, fuel cells and hydrogen storage.
- Finally, CHP models have also been incorporated in whole building simulation programs such as DOE-2.1e and eQUEST and more recently in TRNSYS, ESP-r and EnergyPlus (Beausoleil-Morrison 2008).

From the above discussion it can be concluded that no single simulation program has the potential of providing a complete assessment of this interaction between a grocery store, a co-generation system and the surrounding community. Therefore, in this study several existing simulation programs will be used in conjunction with each other to provide a complete assessment of the following: 1) Whole building energy reduction strategies in grocery stores; and 2) Whole building energy reduction in grocery store on implementing CHP systems.

It is also necessary to determine whether the inputs provided for the simulation model are correct and whether the model is performing the way it should. Hence, it is required to calibrate the simulation models to better represent the facility. Over the years several calibration methods have evolved to validate the simulation process. Several of these methods are elaborated in the next section.

2.8 The Calibration Process

The calibration process usually is initiated with a starting base-case model that is constructed using various specifications and assumptions. The model is then modified by adjusting individual parameters, selected by the user until the difference between the measured data and simulated data is within acceptable limits. Procedures for calibrating simulations have evolved in the 1990s (Claridge 1998). Guidelines and procedures have been developed that help identify probable errors and assess the resultant changes to correct the errors. A discussion on various calibration guidelines and processes considered by this study is discussed in the sections that follow. A discussion on selecting an appropriate weather file for the purpose of calibration is also provided. Other aspects of the calibration process include selecting a suitable time period for calibration (Kaplan et al., 1990a) and establishing daytyping procedures (Hadley 1993, Akbari et al. 1988) to determine lighting, equipment and occupancy schedules in the simulation model. Although important, these issues have not been discussed or addressed by this study.

2.8.1 Calibration Guidelines and Procedures

Calibration guidelines and procedures examined by this study include work by Hsieh (1988), Kaplan et al. (1992), Carroll et al. (1993), and Clarke et al. (1993). These guidelines are discussed in the section that follows. The procedures and guidelines implement a variety of assessment techniques that are used to match simulated results with corresponding measured consumption. Assessment techniques including both statistical and graphical techniques have been the most prominent. Several works are cited to describe these indices that include work done by Krieder and Haberl (1994), Bronson (1992), Abbas (1993), Bou-Saada (1994), Thamilseran (1999), Wei et al. (1998) and Bensouda (2004).

2.8.1.1 Calibration Guidelines

Hsieh (1988) proposed a sensitivity analysis that helps to determining which building and HVAC component have the greatest influence on energy use. Although primarily designed for the DOE-2 simulation program, this set of guidelines may also be used for other simulation programs. The sensitivity analysis was performed by varying each parameter independently, then combining the parameters into one simulation, and finally adjusting the parameters as the results dictate. Hsieh proposed a format that embodied three broad categories based on decreasing energy impact on a building. The first category included building tenant energy consumption, the second category contained HVAC schedules, and the third category contained information related to HVAC equipment and building shell performance. Hsieh also showed that it was important to measure factors such as occupant use of lights and equipment, HVAC schedules and thermostat settings, HVAC and building shell performance, and fan curves. It was also observed that tenant use was most difficult to simulate due to unpredictable daily habits. Finally Hsieh noted that using default values of the DOE-2 program (which was used in Hsieh's analysis) may introduce significant modeling errors. Hsieh's analysis was found to be important because it provided guidelines to the identification of simulation parameters that potentially could play an important role in the calibration process.

In addition to the research performed by Hsieh, Kaplan et al. (1992) also provided guidelines to aid DOE-2 users in calibrating building models. The guidelines included a discussion on strength and weaknesses of computer models and their value in energy conservation programs, simulation reliability improvement, and differences between predicted and actual savings from energy conservation options. The study also provided a protocol for error checking and how modeling errors can be avoided by focusing on inputs, outputs and energy use indices. The study points out to sources of errors, which includes assumptions for equipment power density, unoccupied equipment and lighting schedules, window shading, window and wall U-values and operation assumptions. The model also encouraged to focus on significant inputs which include zoning, infiltration, window units U-values, thermal mass, interior walls, weather files, internal loads, HVAC system selection, HVAC controls, simulation of multiple zone systems and fan schedules. Other sources of discrepancy pointed out by this study include exterior energy consumption and including energy consumption from mainframe computer systems.

2.8.1.2 Statistical Calibration Indices and Techniques

The percent difference was most widely used to assess monthly calibrations and the RMSE, CV(RMSE) and MBE were used to assess hourly calibrations. The percent difference is a simple calculation method that can be used to calibrate monthly energy consumption. In this method the difference between each monthly measured and simulated consumption value was taken and divided by the measured monthly total consumption. The equation for calculating the percent difference is provided in Section C-4.2.1 of Appendix C. This method however can be very misleading. Since these calculations are usually shown for monthly or sometimes even annual calculations, strong possibility exists of errors cancelling out and not being taken into account in the reporting of the percentage difference (Bou-Saada 1994). As seen in Bou-Saada (1994), actual hourly patterns could have a very different trend from the trends reported on a monthly basis.

The indices such as RMSE, CVRMSE and MBE are used to assess hourly calibrations. These were first used for calibration in studies by Krieder and Haberl (1994). The Root Mean Square Error (RMSE) index is a measure of variability in the data. Claridge notes that RMSE is a good measure of the overall magnitude of errors, but does not give any reflection of bias, since no indication is made whether the errors are positive or negative (Claridge 2008). The formula to calculate the RMSE is provided in Section C-4.2.2 of Appendix C.

The Mean Bias Error (MBE) is a method which is used to determine a non-dimensional bias between the simulated data and the measured data for each individual hour. The total difference between the predicted data and the simulated data is divided by the total number of hours considered in the calculation thus rendering a mean bias. The MBE is an overall measure of how biased the data is, since positive and negative errors cancel each other out (Claridge 2008). The formula to calculate the MBE is provided in Section C-4.2.3 of Appendix C.

The Coefficient of Variation Root Mean Squared Error CV(RMSE) is the root mean squared error divided by the measured mean. CV(RMSE) is a non-dimensional value and reflects how well a model fits the data. Lower the CV(RMSE) the better the calibration. The CV(RMSE) is calculated for hourly data and presented on both a monthly summary and total data period. The formula to calculate the CV(RMSE) is provided in Section C-4.2.4 of Appendix C.

The present analysis uses the statistical indices described above to carry out the sensitivity analysis during the calibrating the model. Values of selected parameters[96] were varied within an acceptable range. The final selected value of the parameter was based on the lowest value of RMSE. Unfortunately, natural gas consumption was available on a monthly basis only. Hence, the percentage difference was calculated on a monthly basis for each iteration.

2.8.1.3 Graphical Calibration Indices and Techniques

Graphical indices or graphical displays of building energy data can show important features of a building's energy consumption behavior. Graphical techniques used for calibration range from conventional methods such as X-Y scatter plots, two dimensional line graphs and bar charts to unconventional methods such as box-whisker plots and three dimensional surface plots. Seminal references on presenting data in a graphical format include work by Cleveland (1985), Tufte (1983) and Tukey (1977). These works provide a detailed discussion on the issues encountered when using graphical methods of presenting data and provide comprehensive solutions and methods to enhance the ability of the graph to convey the required information.

Two dimensional time series plots and scatter plots are typically used for calibration procedures on a short term basis (Hseih 1988). The plots include compiling the hourly values of variables – measured and simulated as well as the resultant residuals. Advantages of using such

[96] A list of selected parameters is available in another section of this report.

plots include observing and understanding the general trends and standard errors that become apparent in this arrangement of data. Disadvantages include data overlap especially if a large number of points exist making it impossible to conduct an hourly comparison.

For non-weather dependent loads Bronson (1992) developed a graphical procedure to visualize differences between simulated and measured data over the entire simulation period in terms of three dimensional plots. The three dimensional format implemented by this method permits the visualization of small differences between the simulated and measured data in terms of a positive / negative residual plots.

To assist the calibration process of weather dependent loads, Bronson also proposed a method using a nine graph carpet plot to improve the representation of the space conditioning energy consumption in the simulation model. The graphs present energy consumption versus weather data on a single page to effectively present simulation results.

Newer techniques include binned temperature box whisker mean plots which alleviate the data overlap problem seen in conventional scatter plots. As an improvement to the scatter plots proposed by Bronson, Abbas (1993) proposes the use of temperature binned plots in form of 52 week box-whisker mean plots. The plots essentially eliminate the potential of data to overlap allowing the user to view data over a period of one year. The concept was originally developed by Tukey (Cleveland 1985). Taking a cue from a concept proposed by Cleveland (1985), Abbas (1993) also proposed the juxtapositioning and juxtapaging of plots in order to compare data.

Bou-Saada (1994) improved the procedure described in Bronson (1992) by providing a method of calibration using temperature binned data in a box-whisker plot format to improve the x-y scatter plots by addressing the issue of data overlap. In Bou-Saada’s plots the measured data was superimposed in the simulated data to aid the comparison. Several statistical goodness of fit measures were also considered for a more detailed comparison of the simulated and measured data.

Thamilseran (1999) used residual plots and comparison plots to test an inverse bin method for providing a baseline energy use. Thamilseran’s residual plots are time series plots that show simulated data, measured data and the differences between them (residuals). The comparison plots are x-y plots. In these plots simulated data is plotted in the y axis as a function of measured data plotted on the x axis. The resultant cloud of data points is compared to a line with a property of x = y.

In yet another development of graphical indices, Wei et al. (1998) developed graphical signatures of heating and cooling energy consumed by HVAC systems. The signatures were developed to help make quick decisions during the calibration process. A set of parameters was selected, which produced an impact within ± 10% over the entire ambient temperature and 'characteristic signatures' were developed. The 'characteristic signature' of each parameter is defined as the ratio of the changes in the AHU heating / cooling energy consumption when the parameter was altered by a certain value. The development of these 'characteristic signatures' were based on the fact that both heating and cooling energy consumption of building AHUs are not only dependent on system type and weather, but also dependent on other parameters involved in the functioning of the building such as occupancy, internal load and outside air intake. The calibration procedure was then performed by generating 'calibration signatures', which is a normalized difference between the simulated and measured data as a function of ambient temperature. A pair of graphs was developed for change in both chilled water and hot water. The pair of graphs is then compared to 'characteristic signatures' for the corresponding AHU types, to see what change of parameters from the set of 'characteristic signatures' most closely resemble the patterns obtained from the 'calibration signature'. The parameter is changed accordingly in the simulation model.

This process of calibration using graphical calibration procedures was later refined and extended by Bensouda (2004) to calibrate a case-study building in three climate zones of California for four major system types. Bensouda also provided a step-by-step description of the procedure which is to be adopted during the process of calibration.

2.8.2 Selection of the Weather Files for the Calibration Procedure

Bronson (1992) used site monitored weather data that was reformatted using a Test Reference Year (TRY) weather file. Bronson (1994) noted that the use of average data may be acceptable as long as the building experiences average weather conditions during the monitoring period. However, the comparison of results could become unsatisfactory during periods of abnormally hot or cold weather conditions. In a study similar to Bronson, Haberl et al. (1994) demonstrated the value of preparing site specific data onto a TRY weather tape. The study compared the simulations performed using both TMY2 and TRY weather data and found that the TRY weather formats improved cooling energy use predictions.

The impact of using different types of standard weather data on cooling and heating energy consumption was investigated by Haberl et al. (1995), and Huang and Crawley (1996).

Haberl et al. (1994, 1995) found that using on-site measured weather data improved the simulation. However, standard weather data may be used when no site measured data is available. Huang and Crawley (1996) also assessed the impact of various weather data sets, including: TRY, TMY, TMY2, WYEC (Weather Year for Energy Calculations), and WYEC2, on simulated annual energy use and energy cost. The authors recommended the use of TMY2 weather data set in building energy simulations where solar radiation is critical to the analysis.

The papers cited above primarily discuss various aspects of calibrating the simulation model. In general, these studies primarily discuss the calibration of commercial office buildings. The task of this study would be to adapt the recommendations presented in these studies to the calibration of a grocery store.

Until this point the discussion presented different strategies to reduce energy consumption in grocery stores, implement modeling these strategies using simulation programs and validating whether these strategies were being modeled correctly by discussing relevant calibration techniques. The discussion also presented an overview of CHP technologies and methods available to successfully implement such technologies in buildings. However, a decision to install CHP system is often based primarily on economic considerations. The next section of this literature review presents a discussion on the economic assessment of CHP systems.

2.9 Economic Assessment

As pointed out by Atta (2006), economic assessment is a basic component of the evaluation of energy efficiency technologies, equipment and systems. According to ASHRAE Handbook of HVAC Applications (ASHRAE 2003), the economic assessment can be categorized into two broad categories: simple payback and detailed economic analysis[97]. Typically, methodologies used to provide this economic assessment include costs incurred by the project over an estimated project life cycle. These costs can be categorized as ownership costs[98] and operating costs[99] (ASHRAE 2003). The ASHRAE Applications Handbook (2003) identifies and provides equations for a number of such economic analysis simple and detailed methodologies, which include simple payback, present worth analysis, single payment present

[97] Often, both simple and detailed methodologies are used together with simple payback providing an initial viability assessment and the detailed methodologies assessing the strength of the viable options.
[98] Ownership cost includes owning costs such as first costs, periodic costs such as taxes, replacement costs and salvage costs.
[99] Operating costs include utility costs, maintenance costs and administrative costs.

value analysis, improved payback analysis, savings to investment ration, life cycle costs, internal rate of return, uniform annualized cost method, and cash flow analysis method.

When considering the economic performance of CHP in buildings, Caton (2010) noted that monetary earnings and savings obtained from the installation of a CHP system had to be sufficient to justify the capital investment. Caton (2010) argued that several factors affected the economic performance of a CHP system, which include its initial capital cost as well as by the magnitude and profile of the building loads; the cost of electricity and fuel; and the utility rate structure, which may include demand (peak) charges and/or back-up or standby-power charges. Hence, in order to assess CHP systems, Caton recommended the use of simple payback period, investor's rate of return, annualized costs, annualized worth, net present value, and internal rate of return.

Ellis (2002), on the other hand, recommended two approaches for the economic analysis of CHP in building applications. The first approach compares the cost of electricity generated from the CHP system to that purchased from the utility. The second approach is an hourly system modeling, which accounts for part-load performance of CHP systems and the utility rate structure by determining the electricity generated by the CHP system and that purchased from the utility for each hour of the year.

When considering the economic performance of CHP in district heating and cooling networks (DHC), Fleming (1997) noted the difficulty in modeling such behavior because of the complex interactions between the different system components. Phetteplace (1995) in his work on DHCs, developed a design method for the optimal sizing of pipes that takes into consideration all major network costs. However, Phetteplace (1995) while emphasizing DHC systems potential for energy conservation and reduced environmental impact identified the high cost of piping as a major barrier for the widespread use in the US. Finally, the International District Heating Association (IDHA) also published a "*District Heating Handbook*" (IDHA, 1983), which analyses various topics related to DHC networks including system consideration, distribution systems, metering, and economic and financial analyses.

When discussing methodologies of economic assessment for CHP systems, several important items had to be defined. Park (2007) provided definitions, pros and cons of several economic indices proposed by Caton (2010) for the economic assessment of CHP systems. These are briefly described below[100]:

- Simple payback analysis determines the number of years required for the invested capital to be recovered from the net cash flows that are generated. The calculation does not include the time value of money. Hence, the calculation can be inaccurate when looking at payback periods longer than three years.
- Investors Rate of Return (IROR) is simply the inverse of the payback calculation with similar advantages and disadvantages.
- Net Present Value (NPV) is one of the discounted cash flow techniques that accounts for the time value of money. NPV represents the current value of the project for a specific year based on the initial capital investment, cash flow up to that specific year and discount rates[101]. The method includes the initial investment, all cash flows, and the time value of the investment and the economic life of the project to perform the calculations. A NPV value greater than zero makes the project worthwhile.
- Internal Rate of Return (IRR) calculation determines the equivalent interest rate for the initial capital investment based on series of non-discounted cash flows to that year. IRR is used to avoid making a decision on the discount rate, which in many cases is subjective. In this method, the earning rate of a project is determined by converting all the cash flows to present values which equal the initial investment (Park 2007).

The final section provides a brief discussion of the relevant energy code standards for the commercial and residential buildings considered by this analysis as well as federal regulations related to the installation and operation of CHP systems.

2.10 Energy Codes and Regulations

2.10.1 Energy Conservation Codes for the Grocery Store

For most locations in the U.S., the energy efficiency requirements of the grocery store buildings have to comply with the requirements for commercial buildings provided in either the ASHRAE Standard-90.1 or the International Energy Conservation Code (IECC). However, these

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[100] Equations for the economic indices described in this section are provided in Appendix C of this study.

[101] The discount factor or discount rate is the interest which the firm wants to earn on its investments. The value is decided by the firm’s management (Park 2007). For this study the discount factor is obtained from a technical support document for energy efficiency standards for commercial and industrial equipment published by the Department of Energy (US DOE 2008b).

specifications do not include process loads[102]. Process loads such as energy consumed by refrigeration systems, which constitute a major portion of the grocery store are not considered in ASHRAE Standard -90.1 specifications. The energy consumption requirements for this type of equipment are governed by the stipulations in Energy Policy Act of 2005 and Energy Independence and Security Act of 2007.

The EPACT of 2005 directed the Department of Energy (DOE) to issue specific energy conservation standards for several residential and commercial equipment. The commercial refrigeration equipment included solid door reach-in refrigerators, freezers and refrigerator-freezers; glass-door refrigerators and freezers; and automatic commercial ice machines manufactured on or after January 1, 2010. The Energy Independence and Security Act of 2007 included prescriptive standards for walk-in coolers and freezers manufactured on or after January 1, 2009. The standards include requirements for automatic door closers, methods to minimize infiltration through doors, wall and floor insulation requirements, provisions for evaporator and condenser fan motors, light fixtures and lighting control, as well as anti-sweat heater specifications. More recently the DOE published energy conservation standards for equipment sold on or after January 1st 2012, which includes commercial ice-cream freezers, self-contained commercial refrigerators, freezers, and refrigerator-freezers without doors, remote condensing commercial refrigerators, freezers, and refrigerator-freezers.

In addition to the federal energy conservation standards, state standards such as California Energy Commission (CEC) provide certain supplementary regulations on energy consumption of products that are not regulated by the federal law. In addition to the federal and state energy conservation standards, several voluntary programs promote the adoption of high-efficiency equipment that is available on the market. These include ENERGYSTAR[103], Consortium for Energy Efficiency Commercial Kitchens Initiative (CEE)[104], and the Federal Energy Management Program (FEMP)[105].

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[102] Exceptions being process loads for data centers.

[103] ENERGYSTAR is a joint labeling program of the US EPA and the US DOE designed to identify and promote energy and water-efficient products to reduce greenhouse gas emissions.

[104] CEE develops initiatives to promote the manufacture and purchase of energy efficient products and services. The goal of the CEE initiative is to provide clear and credible definitions as to what constitutes highly efficient energy and water performance in refrigeration equipment, as well as cooking and sanitation equipment, and then help streamline the selection of products through a targeted market strategy.

[105] FEMP works to reduce the cost and environmental impact of federal government by promoting the use of distributed and renewable energy, and improving utility management decisions at federal sites.

2.10.2 Energy Conservation Codes for Residential Buildings

In the U.S., building energy codes for residential buildings[106] are primarily guided by the requirements in the International Residential Code (IRC) or the International Energy Conservation Code (IECC). Compliance with the codes is required to achieve the energy and economic goals set by these codes. The IECC provides two paths for code compliance: the prescriptive path and the performance path. The current residential code for Texas is the 2009 IRC (ICC 2009b)[107]. In Texas, compliance with performance path can be attained by using code compliance software such as the International Code Compliance Calculator (IC3), which is a software developed by the Energy Systems Laboratory. The residential model developed for this software (Kim 2007, Malhotra 2009) is utilized by this analysis.

2.10.3 Regulations for CHP Systems

According to Foley (2010b) regulations for CHP systems can influence the sizing, selection and cost of the CHP system, system configuration, accessory equipment requirements, minimum efficiencies, financial support, allowable emissions and interconnection methods.

The development of CHP facilities got a big boost with the passing of the Public Utility Regulatory Policy Act PURPA of 1978. The act required the utilities to purchase power generated by PURPA qualified facilities and allowed CHP plant access to the whole sale power market. The utilities were required to provide backup power service at a reasonable cost.

The Energy Policy Act of 1992 deregulated parts of the U.S. electric market making it more competitive. Subsequent lowering of electricity costs made PURPA qualified facility contracts less attractive to generators (Chittum and Kauffman 2011). Recognizing this trend towards deregulation, the Energy Policy Act of 2005 (EPACT 2005) removed the requirements of the utilities to purchase power from PURPA qualified facilities in regions were the power market was deregulated (Chittum and Kauffman 2011). This change in policy has forced many CHP plants to shut down. Foley (2010) noted that post-PURPA CHP systems generally do not export electricity back to the grid, nor are these systems sized to meet the power peak requirements of the facility because of lack of economic feasibility. On the other hand, the EPACT 2005 provided certain incentives, which included 30% tax credit for fuel cells and 10%

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[106] Residential buildings include single family detached dwelling units, two-family attached units and multi-family units that are up to three stories tall.
[107] Section N1101.2 in the 2009 IRC provides the option of meeting the requirements of the 2009 IECC. For this study the specifications provided in the 2009 IECC are used.

tax credit for micro-turbine based power generation systems and extending federal energy procurement contract to 10 years. Certain additional incentives were also added by the introduction of the Energy Improvement and Extension Act of 2008, which included investment tax credit for CHP through 2016 and allowing a 5 year accelerated depreciation schedule for CHP systems. The benefits of CHP have also been recognized by the state governments and have led to creation of incentive programs by some states, which include capital grants and tax credits.

In addition to the above mentioned acts, the Federal Clean Air Act of 1970 required the U.S. Environmental Protection Agency to establish ambient air quality standards that in turn regulated the air emissions from CHP systems. The standards prescribed limits for six pollutants of which NO_x, CO and SO_2, and particulate matter (PM) are of greatest relevance to CHP systems.

2.11 Conclusions from the Literature Review

In order to investigate the viability of reducing energy consumption in grocery stores to near net-zero levels, it was important to first define net-zero energy buildings. To accomplish this, several definitions were discussed. These definitions were developed by the NREL and involve defining the net-zero energy consumption of buildings in terms of site energy, source energy, energy costs and emissions. While these levels have been demonstrated in residential and some commercial buildings, the viability of such an approach in the case of a grocery store is debatable. The second section of the review discussed the functioning of a typical store. Using the previous literature, the energy end-use of each building system operating in a typical store was identified and assessed. It was concluded that the largest load comes from year-round refrigeration requirements followed by lighting and space conditioning.

In the third section, the literature review examined energy efficiency measures for individual systems involved in operating the grocery store followed by a discussion on whole-building energy performance on implementing various combinations of energy efficiency measures. This section of the review included discussions on improving the performance of vapor-compression refrigeration cycles, display-case design, HVAC systems, building envelope and lighting measures. The impact of each system on site energy savings was also discussed. The review established that a 50% reduction (over ASHRAE 90.1-2004 base-case) in site energy consumption of grocery stores could be achieved by increasing the efficiency of the refrigeration, HVAC, building envelope, and the lighting systems. However, even the reduced levels of energy consumption in the grocery store were substantial. Hence, the review concluded that it would be prohibitively expensive to meet the reduced store energy consumption by renewable energy systems. Therefore, in order to approach net-zero energy consumption, one possibility would be to consider the implementation CHP systems. In this way, the store would be able to reduce its electricity demand with thermal energy recovered from the CHP plant.

In the fourth section, the review discussed the concept of CHP. Several issues such as the selection of appropriate CHP technologies were reviewed. The feasibility of implementing CHP technologies in a typical grocery store was also reviewed. It was concluded that with a near constant demand for cooling power year around, grocery stores provide an ideal thermal sink to absorb the waste heat from the CHP system through the use of technologies such as absorption refrigeration and desiccant cooling technologies.

In the fifth section of the review several CCHP technologies were investigated. The review concludes that a number of tri-generation technologies that include absorption cooling and dehumidification systems exist, which are potentially beneficial for running the grocery store. The review concluded that absorption chillers using the Water/NH_3 working fluid are best suited to achieve the low temperatures required for product storage at grocery stores. Other technologies such as adsorption chillers and solid desiccant cooling systems are still being developed which may prove to be useful in future. By using waste heat generated from a proposed CHP plant used to power the community for running absorption chillers at grocery stores the demand of electricity from the grid can be considerably reduced and the total source energy use reduced. However, current studies also show that the implementation of such technologies has mostly been limited to cold climates.

In the seventh section of the literature review various simulation programs were discussed that could be used to assess the viability of implementing energy efficiency measures as well as assessing the performance of CHP and CCHP systems for a grocery store and the surrounding community. It was concluded that no single simulation program had the potential of providing a complete assessment of this interaction between the grocery store and the community. Hence, several simulation programs would have to be used together to provide a complete assessment

In the eighth section of the review presents a brief discussion on the various calibration procedures available to calibrate energy simulation models. Guidelines for calibration as proposed by various authors were presented and assessed. In addition, both statistical and graphical techniques for carrying out the calibration were reviewed. Finally, the use of weather files in simulation models was discussed. The use of TRY weather files was determined to be suitable for the calibration process. On the other hand, the use of TMY files was determined to be most suitable for efficiency strategies evaluation purposes.

In the last section of the review, an overview of the economic assessment is presented. A number of methodologies for economic analysis were briefly reviewed. The parameters that impact the economic viability of CHP systems was presented and discussed. Finally, several economic indices that would be used in this study were identified and described.

CHAPTER III

SIGNIFICANCE AND SCOPE OF THE STUDY

3.1 Significance of the Study

This work intends to provide the following benefits towards the design of buildings demonstrating high energy consumption levels with an aim to reduce energy consumption:

- The development of a calibrated building model of a grocery store.
- The use of the calibrated building model to assess the implementation of energy efficient measures in the grocery store.
- Examining the option of CHP facility to further reduce the energy consumption of the grocery store.
- Investigating the potential of sharing energy across the boundary of the grocery store with surrounding residential buildings to reduce energy consumption in the community.

3.2 Scope and Limitations of the Study

The scope of this study is limited to assessing energy consumption of grocery stores that can be reduced. Attainment of net-zero levels in the grocery store is neither attempted nor discussed. In addition, the study is limited to a discussion on grocery store centric options for cogeneration. Other scales of community based cogeneration exist but have neither been attempted nor discussed. Finally, the economic analysis is limited to the assessment of cogeneration systems. An economic assessment for the energy efficiency measures proposed for the grocery store was not discussed.

The limitations of this study can be categorized into the following categories:

- Limitations due to the design of building systems, which include limitations of the building system configuration being assessed and the level of detail of building system design proposed by this analysis.
- Limitations due to selection of individual design software, which include the selection of methods and software for assessing energy efficiency measures as well as CHP options in the grocery store.

- Limitations due to the integration process, which included simplification of assessing building system components due to the integration of outputs from the eQUEST-Refrigeration program and the results from the spreadsheet analysis.

CHAPTER IV

METHODOLOGY

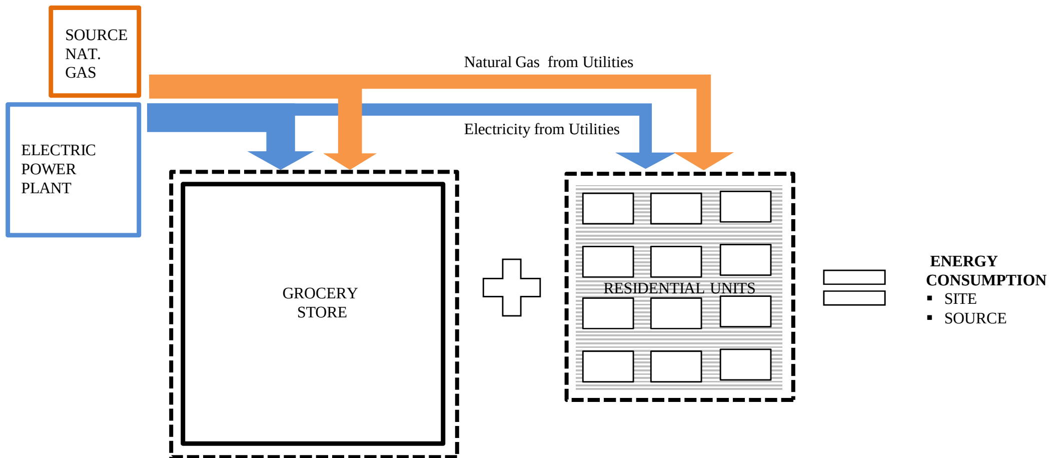
4.1 Overview

In order to proceed with the analysis, the study considered two building types –a grocery store and a multifamily building, which consisted of multiple units that could be scaled up or down to absorb the surplus electricity and thermal energy generated by an on-site CHP system. The study was broadly divided into two parts. In the first part the study evaluated the grocery store at the building level and investigated measures to reduce the overall energy consumption of the grocery store. The study evaluated the impact of implementing the measures in terms of site and source energy consumption. The second part of the analysis involved reducing the energy consumption in the grocery store by using appropriate CHP systems. Surplus energy was then shared across the boundary of the store with the surrounding residential community, which in this case, were multiples of an 8-unit multifamily building. The resultant energy consumption for the energy efficient store and the surrounding residential units were accounted for at the site as well as source levels. Figure 4-1 presents a diagrammatic view of this concept. A discussion of part 1 and part 2 are presented in the second and third section of this chapter. A detailed methodology and results from each part are presented in separate chapters of the study.

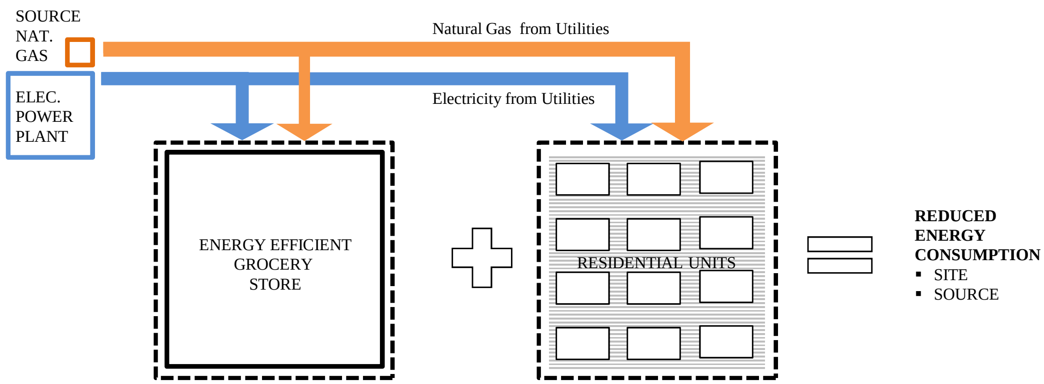
4.2 Part 1: Reducing Energy Consumption in a Grocery Store on a Building Level

The first part of the analysis was conducted in two steps. In the first step, a simulation model of a grocery store was constructed and calibrated to data from an existing store in central Texas. The case-study store provided information to describe the characteristics of a base-case store in terms of building size and usage, as well as specifications for the building envelope, lighting and equipment, HVAC systems and refrigeration systems. The resultant grocery store model was constructed using eQUEST-Refrigeration (Version 3.61)[1] whole building energy simulation program specifically created for supermarket and industrial refrigeration. The model used information from the case-study store and certain other assumptions from a literature review as well as reasonable default values from the simulation program.

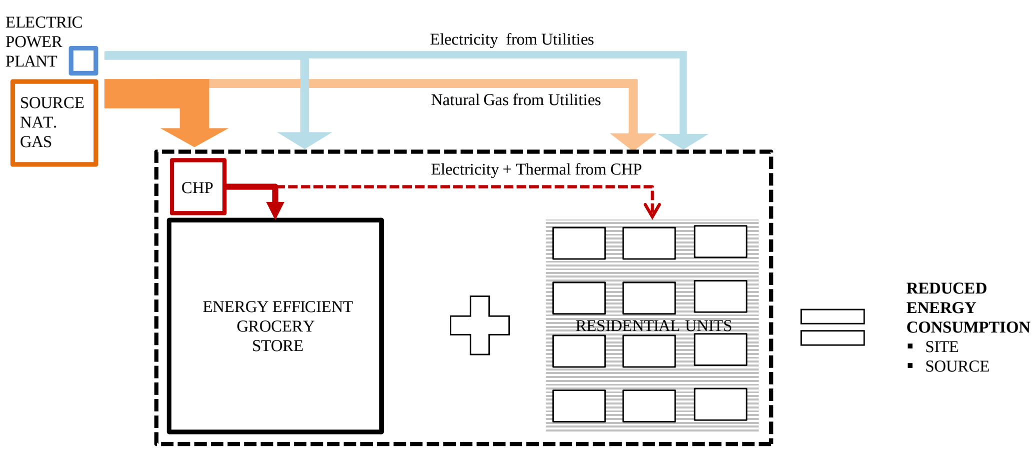
[1] Development on the eQUEST Refrigeration software program was discontinued in 2006 due to lack of funding with the latest version being Ver. 3.61.



BASE-CASE



PART 1: Reducing Energy Consumption in a Grocery Store at a Building Level



PART 2: Reducing Energy Consumption in a Grocery Store as Part of a Community using CHP

Figure 4-1: Research Methodology – Overview

To ensure that the simulation model was performing correctly it was found necessary to validate the model using measured data obtained from the grocery store. For this study the simulation model was validated using both hourly and monthly measured data obtained from the grocery store for a time period of one year. The time period selected for the analysis was 2009. A suitable weather file was selected to run the simulation, which was compiled using 2009 measured weather data[2] from a nearby weather station (Kim and Baltazar-Cervantes 2010).

The calibration process was conducted using both statistical and graphical indices. Graphical indices included time series plots, bin plots and scatter plots. Statistical indices include RMSE, CV(RMSE) and MBE values[3]. Parameters selected for the iterative process represent the building systems operating in the store, which includes the building envelope, lighting and equipment, HVAC systems and refrigeration system. The overall process was cumulative with a change made to the model after each iteration. Details of this step are provided in Chapter 5 of this study.

In the second step, the modified base-case was used in the assessment of efficiency measures that could potentially be used to reduce the energy consumption in the grocery store. The literature review presented a discussion on a number of efficiency measures that could be used in grocery store to reduce energy consumption. These measures were grouped together under several subcategories of building systems that operate in the grocery store. The subcategories included building envelope, lighting and equipment, HVAC systems and the refrigeration system. The list of energy efficiency strategies was narrowed down considering the modeling constraints imposed by eQUEST Refrigeration whole building energy consumption software that was selected for this analysis. The efficiency strategies were first assessed on an individual basis. After reviewing the results from individual runs, several high performing strategies were grouped under the above mentioned categories. The simulations were then cumulatively performed. The final simulation model included the efficiency measures for the envelope, lighting and equipment, HVAC and refrigeration. Details of this step are provided in Chapter 6 of this study. A flowchart diagram for Part I of the analysis is presented in Figure 4-2.

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[2] A TRY formatted weather file for College Station, TX was used for the calibration analysis (Kim and Baltazar-Cervantes 2010). The TRY weather file was packed using data downloaded from National Climatic Data Center website (NCDC), and solar radiation data (Global solar radiation) downloaded from Texas Commission on Environmental Quality (TCEQ).

[3] A detailed discussion of these statistical indices is provided in the literature review of this study.

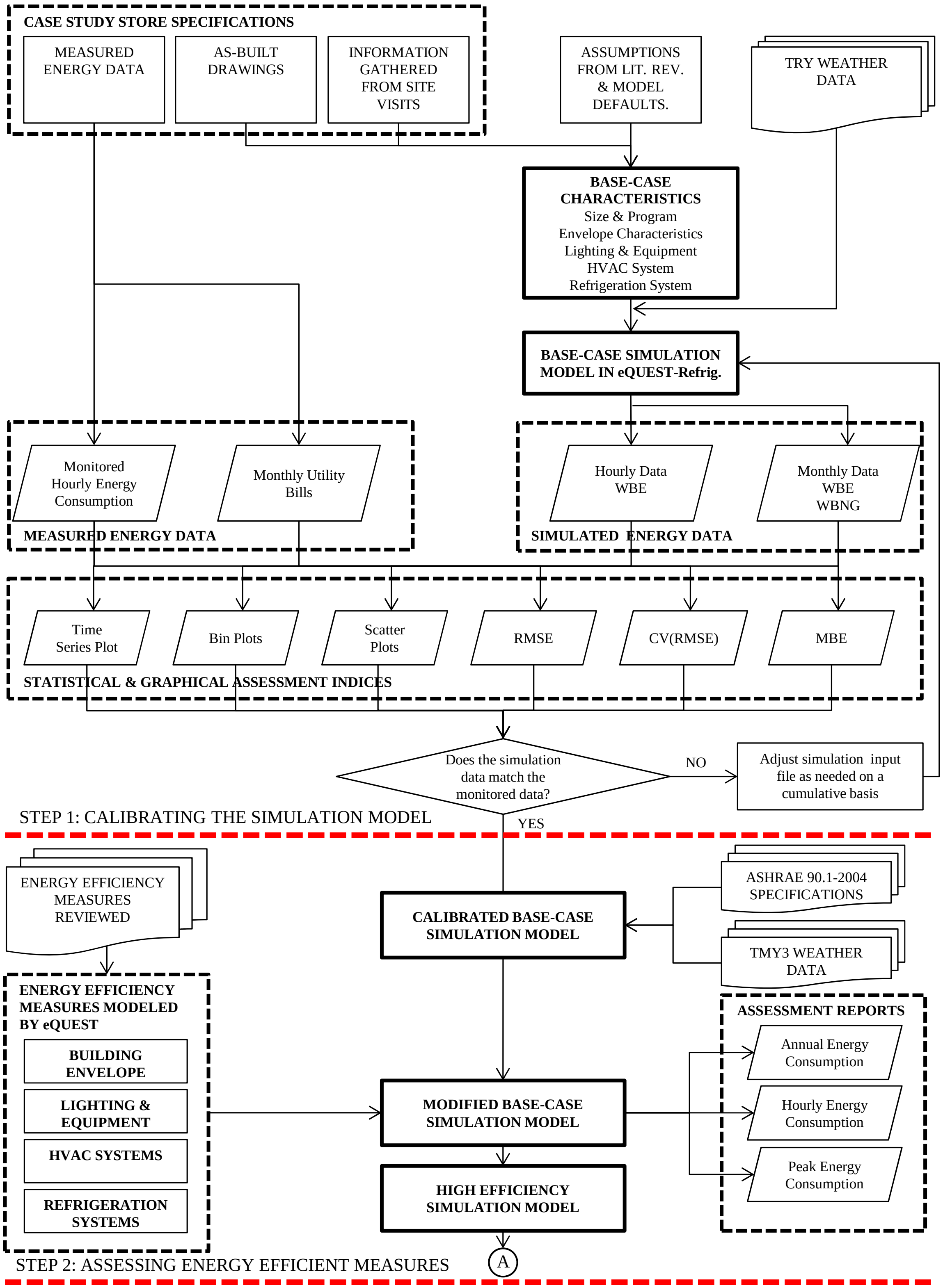

Figure 4-2: Research Methodology – Part I

4.3 Part 2: Reducing Energy Consumption in a Grocery Store using CHP

The second part of the analysis involved the assessment of installing a CHP system both in terms of the resultant impact on energy consumption as well as on economic terms. The analysis was performed by considering the grocery store as part of a residential community in terms of sharing of energy across the grocery store building boundary. As in the first part, the second part was also divided into two steps. The first step dealt with the assessment of the CHP system in terms of the impact on energy consumption. In the second step, an economic assessment of the CHP system was conducted.

In the first step of the second part of the analysis, an appropriate CHP system was installed to service the electricity and thermal requirements of the grocery store. Surplus energy generated was then matched with the electricity and thermal requirements of the neighboring residential community. To carry out the analysis for this section two models - a CHP model and a multi-family residential model were created in addition to the grocery store model.

The CHP model was used to assess an appropriate CHP system that would serve the needs and requirements of the grocery store. The CHP model provided options for selecting an appropriate prime mover and its mode of operation, thermal cooling technologies and heat recovery strategies. The CHP model was created using Microsoft Excel for Windows (MS-Excel 2010). The multi-family residential model was adopted from the simulation model[4] created for the IC3 code compliance calculator developed by the Energy Systems Laboratory (Kim 2006, Malhotra 2009). The model was constructed using specifications prescribed in the International Energy Conservation Code (IECC 2009) for Texas Climate Zone 2. In addition to the above mentioned models, the grocery store simulation model created in eQUEST Refrigeration was modified to better accommodate the operation of the CHP system. Modifications include provisions for hot water boilers, absorption chiller model for space cooling as well as medium and low temperature absorption refrigeration and a sub-cooler model.

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[4] An input file created using DOE-2.1e (Wikelmann et al. 1993) whole building energy simulation software program is used to run the IC3 calculator. This study used this DOE-2 input file to conduct the analysis.

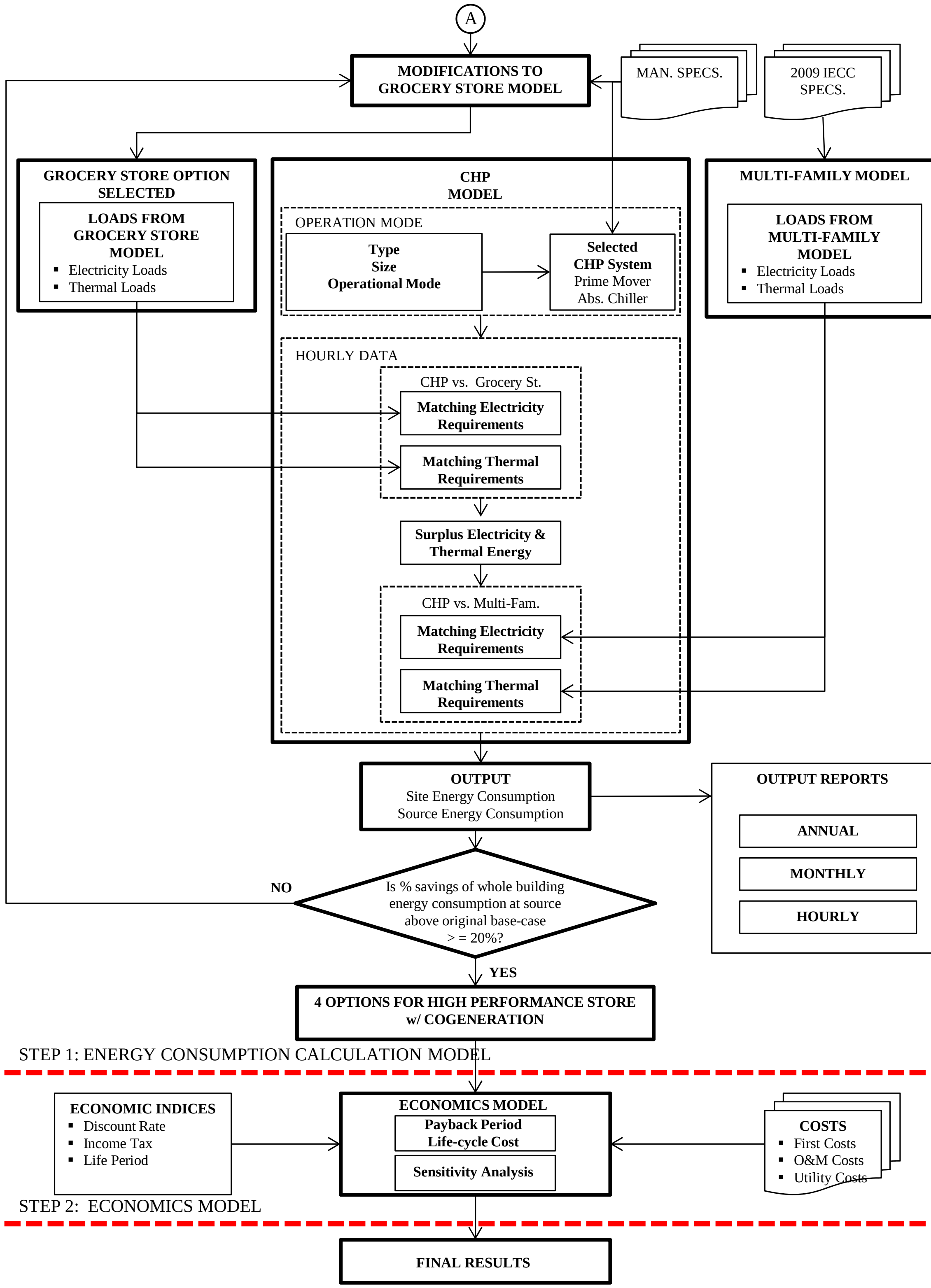

Figure 4-3: Research Methodology – Part II

In the CHP model, several options for a grocery store were selected, which presented a reasonable amount of both electricity and thermal energy consumption. The options were selected to present varying degrees of usage of waste thermal energy generated by the CHP system installed in the grocery store. The building loads obtained from the selected option were matched with the electric power and waste thermal energy generated by a prime mover selected in the CHP model. The surplus loads were then matched to the loads from the multi-family model and the resultant number of dwelling units being served by the surplus energy was determined. The output was assessed in terms of annual, monthly and hourly energy consumption. The output was also evaluated in terms of site and source energy consumption. A site to source energy factor of 3.15 was used for electricity and 1.1 for natural gas. These factors are obtained from the Annual Energy Review published annually by the U.S. Energy Information Administration (EIA 2010). The site to source conversion for electricity also accounts for 7% transmission losses. The objective of the analysis was to determine the highest percentage reduction above the base-case model, which included the energy consumption of the unmodified grocery store with implemented energy efficiency measures and the multi-family units that could potentially be served by the surplus energy from the CHP system implemented in the grocery store. An arbitrary cutoff percentage of 20% above base-case was selected. Finally, four options satisfying the criteria for reduction in energy consumption were selected to proceed to the second step. Details of this step are provided in Chapter 7 of this study.

In the second step, an economic assessment was performed. The analysis implements two typically used methods for the assessment - life-cycle cost analysis and payback period analysis. Several metrics that are commonly used for economic assessment were used to conduct this analysis. The measures include assessing the simple payback period and investor's rate of return (IROR) when performing the payback period analysis; and assessing the net present value (NPV), time until net present value and internal rate of return (IRR) of the project when performing the life-cycle cost analysis. Details of this step are provided in Chapter 8 of this study. A flowchart diagram for Part II of the analysis is presented in Figure 4-3.

4.4 Summary

This chapter provides an outline of the methodology used in this study. The methodology used in this study was developed to assess the reduction of energy consumption in grocery stores in a hot and humid climate. Hence, a single location representing the hot and humid climate was selected for the analysis. The assessment is then divided into two parts. In the first part, the energy reduction of the grocery store was assessed on a building level. In this part a calibrated base-case building was modeled. Several energy efficiency measures were evaluated using this model. The energy reduction was monitored at both site and source levels. In the second part, reducing energy consumption in the grocery store was addressed by considering appropriate CHP systems. Surplus energy was then shared across the boundary of the store with the surrounding residential community, which in this case, were multiples of an 8-unit multifamily building. Here too, the energy reduction was monitored at both site and source levels. An appropriate economic analysis was subsequently performed. Finally, the energy consumption of the base-case grocery store building, and results from the two parts were compared side-by-side and the total reduction in energy consumption at site and source levels was determined.

CHAPTER V

CALIBRATING A GROCERY STORE SIMULATION MODEL

5.1 Overview

In this chapter a grocery store model was developed using information from a case-study grocery store, which is situated in a hot and humid climate of central Texas. Several other references and assumptions were also used in the development of the simulation model. The model was then used to assess various measures proposed for the grocery store in order to reduce the energy consumption levels of the store. The implementation of the energy efficiency measures and the corresponding results will be discussed in the next section of this study (Section 6). The whole-building energy consumption model was developed in eQUEST-Refrigeration (Version 3.61) whole-building energy simulation software. The simulated model was calibrated using measured data for whole-building electricity and natural gas consumption obtained from the case-study store.

The first section of this chapter provides an overview. The second section of this chapter provides a general description of measured data that was used in the calibration procedure. The third section of this chapter provides a general description of the simulation data that was used in the calibration procedure. The fourth section of this chapter provides a detailed description of the base-case simulation model for the grocery store. The fifth section describes the calibration procedure adopted by this study to calibrate the grocery store model. Finally, the sixth section presents a summary of the result and the conclusions.

5.2 Measured Data

5.2.1 Time Period of Data Available for Calibration

Both monthly and hourly data was used to calibrate the store. This information was made available for the year 2009. Monthly data was made available from utility bills. Hourly measured data was available from an on-site monitoring system installed at the store. Hourly data was made available for the HVAC, lighting, refrigeration system and overall power consumption in the building. However, gas data was only available on a monthly basis. Hence only monthly calibration could be carried out for natural gas related energy consumption patterns. Natural gas primarily is used for water heating and space heating purposes.

5.2.2 On-Site Monitoring System

The on-site monitoring system MT Alliance[1] from Micro-Thermo Technologies (MT Alliance 2003) was used in the case-study store. The on-site monitoring system has numerous channels which record data at one minute interval for the parameters of the building systems. Measured data recorded by these channels for temperatures, pressures, humidity levels, lighting levels and operation logs of HVAC, refrigeration and lighting systems was available on a one minute basis and was converted to hourly data. However, data for power consumption was available only for the whole- building electricity consumption and for four refrigeration racks installed in the grocery store. A complete list of these channels has been provided in Section B-3, Appendix B in this report.

The information regarding temperature that was obtained from the on-site monitoring system was compared against independent measurements performed with calibrated instruments[2]. Calibration of other parameters measured by the on-site monitoring system in the store was not carried out due to time constraints.

5.3 Simulation Data

5.3.1 Simulation Model of the Grocery Store

eQUEST-Refrigeration Version 3.61 (Hirsch, 2008) was used to model the grocery store. The refrigeration version of the program enables the user to model the performance of refrigeration systems in the store and analyze these systems as part of whole-building energy performance. A significant change from the parent eQUEST program was the addition of a refrigeration module capable of simulating supermarket and industrial refrigeration in detail (Hirsch, 2008).

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[1] Micro Thermo Technologies is a manufacturer of refrigeration controls for supermarket applications. The MT Alliance information and control system developed by Micro Thermo Technologies is software that provides full monitoring and control of refrigeration, HVAC, lighting and energy sub systems that are all run on the same network and interface (MT Alliance 2003).

[2] Details of the calibration procedure with HOBO data loggers are presented in Section B-4, Appendix B of this study.

5.3.2 Weather Data

According to the literature review carried out by this study, on-site weather data measured for the simulation period has been shown to significantly improve the simulation (Haberl et al. 1995). Hence, a TRY[3] formatted weather file for College Station, Texas was used for the calibration analysis[4]. The data was compiled using 2009 measured weather data from a nearby weather station (Kim and Baltazar-Cervantes 2010). The TRY weather file was packed using data downloaded from National Climatic Data Center website (NCDC), and solar radiation data (Global solar radiation) downloaded from a test bench installed on Texas A&M University campus. It should be noted that since the weather file does not account for daylight saving time, the addition / subtraction of one hour to adjust for daylight savings had to be manually done to the weather file in one of the iterations of the calibration process.

5.4 Base-Case Model for the Grocery Store

5.4.1 Initial Setting for the Base-Case Simulation Model of the Grocery Store

The base-case model of the store was developed using information primarily from the case-study store. The information is documented in the building drawings of the grocery store. Certain other references were used to input information that could not be obtained from the case-study store. This information includes rates and schedules for infiltration and occupancy (Hale et al. 2008); equipment power density and schedules (Deru et al. 2011a)[5]; for lighting power density[6] and initial calculations of HVAC system sizing (ASHRAE Standard 90.1-2004); initial calculations of outdoor air requirements (ASHRAE Standard 62.1-2004): service hot water demand (ASHRAE Applications Handbook 1999); and service hot water schedule in the grocery store (Hale et al. 2008).

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[3] TRY (Typical Reference Year) is a weather file format that contains the weather for a specific location and particular year of interest.

[4] US Climate Zone 2A (warm and humid), as specified in ASHRAE Standard 90.1-2004 (Appendix B, Pg 109).

[5] Although information regarding the electric power consumption of the miscellaneous and lighting equipment in the store was available from the documented specifications, the information represented the maximum power that could be provided for the installed equipment. The actual number of equipment installed and the operating schedule would result in a lower consumption of electricity than what was estimated in the specifications of the store.

[6] Although information regarding the electric power consumption of the lighting fixtures in the store was available from the documented specifications, the original lighting was changed out in certain sections of the store for more efficient lighting systems. The management could only provide an overall LPD value for the store. Hence the study assumed a LPD as specified in the ASHRAE 90.1-2004 and varied it in subsequent iterations.

5.4.2 Location, Building Form and Building Program (Table 5-1)

5.4.2.1 Building Location and Total Floor Area

The location of the base-case building was determined to be in central Texas. This location represents the hot and humid climate. The location of the base-case building was selected to be the same as that of the case-study grocery store.

The floor area of the case-study building was determined to be 92,952 ft^2. This floor area approximately matches that of the case-study store. This decision was made as it would be possible to validate the base-case building with measured data from the case-study store.

5.4.2.2 Building Form

The specifications of the built volume of the base-case simulation model were adopted from the case-study store. The building is a single-story structure with an aspect ratio of 1: 1.5. The floor-to-ceiling height is, on average, 20 ft. The building front faces the north-west orientation.

5.4.2.3 Thermal Zoning

There are 23 zones in the case-study store, each of which is conditioned by a packaged roof top HVAC unit. In order to simplify the modeling process, similar zones were combined into one zone resulting in the creation of 5 zones (Appendix B, Section B-1) in the simulation model. The zones are: a) General merchandise zone; b) Display case zone; c) Bakery zone; d) General loading dock zone; and e) Produce loading dock zone.

There are 16 additional spaces that support the functioning of the grocery store. The temperatures of these spaces are maintained below the normal store temperatures. The temperature of each zone was determined by the activities performed in that zone. These spaces include: preparation rooms, cooler rooms and freezer rooms. The conditioning of these spaces is provided by the refrigeration system of the store. In order to simplify the modeling process, spaces with similar temperatures were grouped together. The final simulation model includes three spaces that are kept at temperatures lower than the normal store temperature. These include: a) Cooler zone, b) Freezer zone, and c) Preparation zone

The criteria for grouping of thermal zones in the simulation model are provided in Table B-1 and Figure B-1. The original and modified configurations of thermal zones are presented in Figure 5-1 and Figure 5-2. The front and back view of the simulation model is presented in Figure 5-3.

5.4.2.4 Space Conditions

The environmental conditions in the base-case model were set to match the observed conditions in the case-study store. From the monitoring system in the case-study store it was observed that the space temperature was set at 72°F. The environmental conditions for coolers, freezers and preparation areas were based on a weighted average obtained from the specifications in the case-study store. The temperature conditions in these spaces are presented in Table B-2.

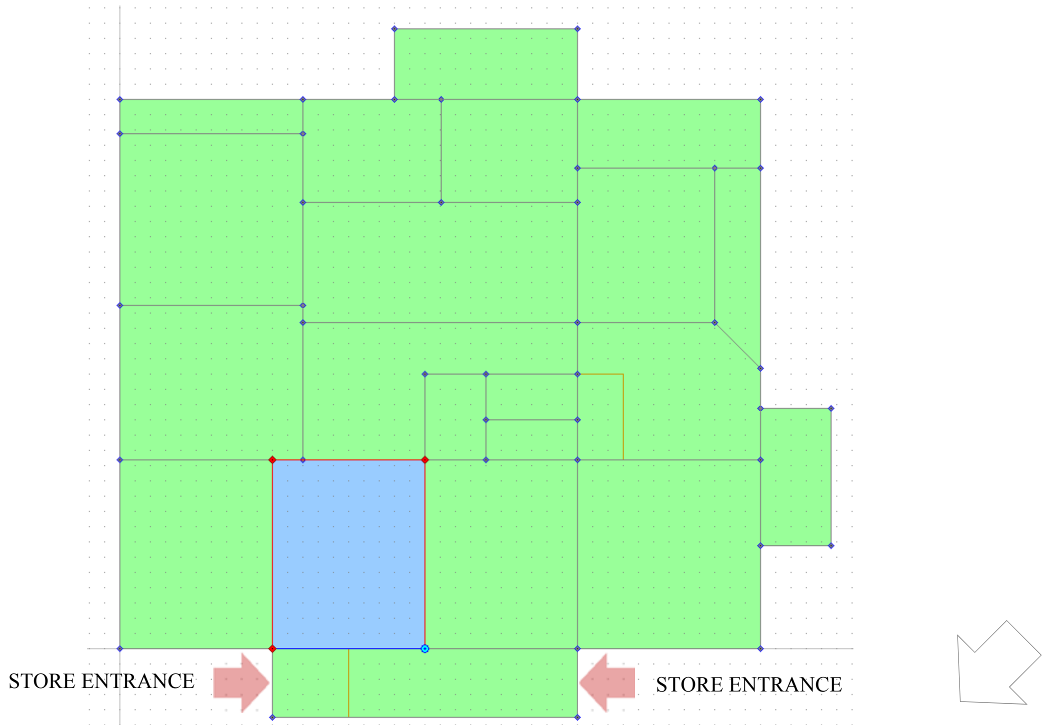

Figure 5-1: Case-study Store with Original Thermal Zones

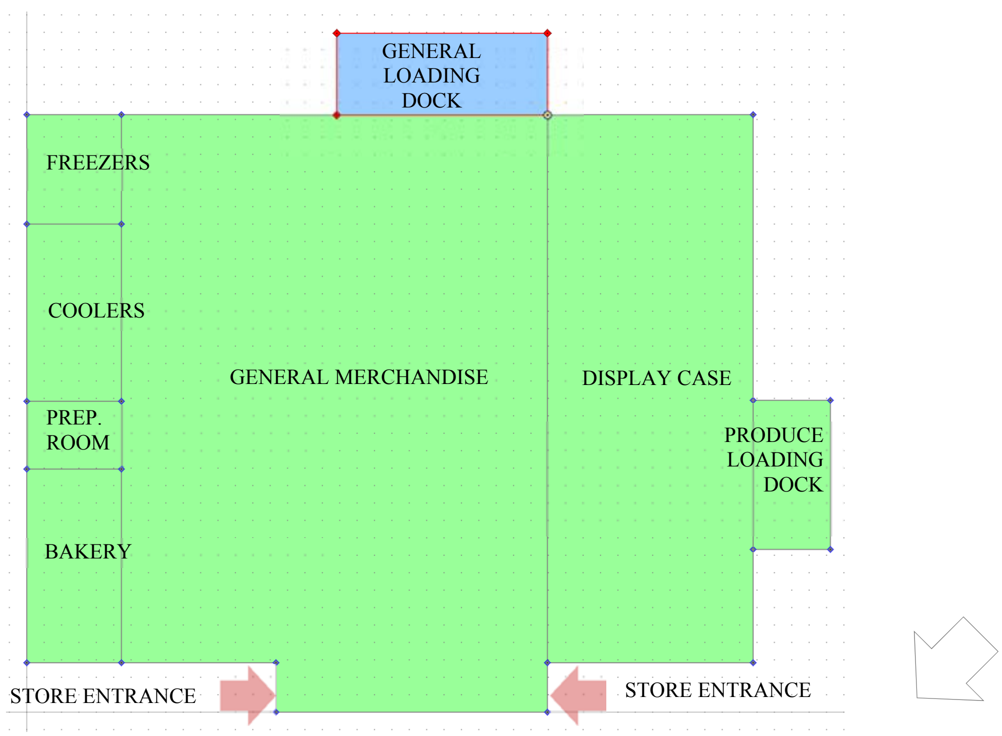

Figure 5-2: Base-case Store with Modified Thermal Zones

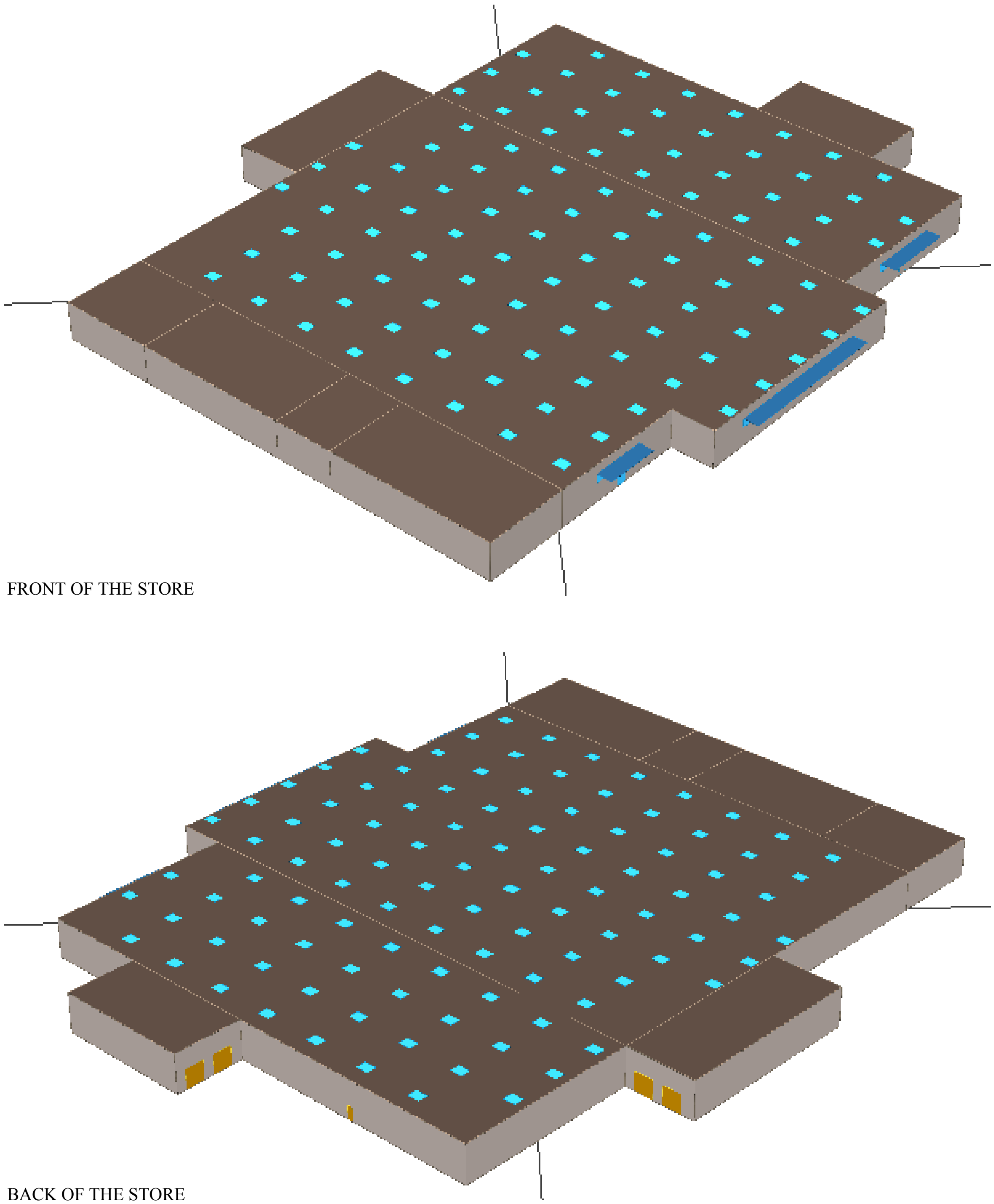

Figure 5-3: Front and Back View of the Base-Case Simulation Model of the Grocery Store

5.4.3 Building Envelope (Table 5-2)

5.4.3.1 Exterior Walls

Most of the information for the building envelope was adopted from the case-study store. For simulating exterior walls, as noted from the case-study store, the cross section of exterior included insulated pre-cast concrete panel[7,8], a 1 ¾" airspace sandwiched in between, a layer of building paper and finally 3 5/8" stone veneer mounted on the exterior surface of the wall section. No information was provided from the case study store regarding the insulation used in the insulated concrete panels. Moreover, ASHRAE Standard 90.1-2004 does not have any requirements for insulation when installing mass walls for this particular Climate Zone. eQUEST-Refrigeration does not have provision of modeling pre-cast insulated concrete panels. Hence, each section of the insulated concrete panel had to be modeled separately. A diagrammatic view of the typical cross-section of the exterior wall is presented in Figure 5-4a.

5.4.3.2 Roof and Floor

The roof of the store simulation model consists of polyvinyl chloride (PVC) single ply, membrane system on top of a rigid, R-19 continuous insulation, which is fastened to a galvanized metal roof. The floor of the store is a slab-on-grade construction with a 4" concrete slab poured over 12" thick compacted soil. No insulation is provided, which is typical of buildings built in this Climate Zone (i.e., Climate Zone 2A). In the base-case model, the floor was modeled using the procedure proposed by Winkelmann (1998) for modeling underground floors[9]. A diagrammatic view of the typical cross-section of the roof is presented in Figure 5-4a.

5.4.3.3 Freezer, Cooler and Preparation Room Envelope

Freezers, coolers and preparation rooms are maintained at lower temperatures than other spaces in the grocery store. Hence, freezer, cooler and preparation room walls and roof require special treatment in order to maintain tightly controlled lower temperatures within these spaces. According to the specifications of the case-study grocery store, the wall and roof of the coolers

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[7] Pre-cast concrete insulated panels consist of two reinforced concrete panels with a continuous layer of rigid insulation, which is typically extruded polystyrene (CPCI n.d).

[8] Since no dimensions were provided in the case-study drawings, the panels were assumed to be 6" thick with a 2" of insulation sandwiched between two 2" thick concrete panels.

[9] Winkelmann (1998) proposes a method to provide a more accurate method to calculate the underground surface heat transfer. In this method the authors propose the use of an effective U-value to calculate the heat transfer between the slab and the ground, which is much less than the raw U-value of the slab. The paper describes the procedure to determine appropriate effective U-values for surfaces in contact with the ground under different conditions.

and freezers are 5” thick consisting of insulation sandwiched between two metal sheets, and are assembled by the freezer manufacturers. The wall insulation panels are usually four to ten inches away from the exterior wall. The model recreated the construction specifications of the case-study grocery store. A diagrammatic view of the typical cross-section of the freezer wall and floor is presented in Figure 5-4b.

Freezer and cooler floors were simulated according to the specifications provided in the case-study store. The floors were simulated with a four inch thick extruded polystyrene insulation boards[10] placed under a 5” thick heavy weight concrete slab. A diagrammatic view of the typical cross-section of the freezer wall and floor is presented in Figure 5-4b.

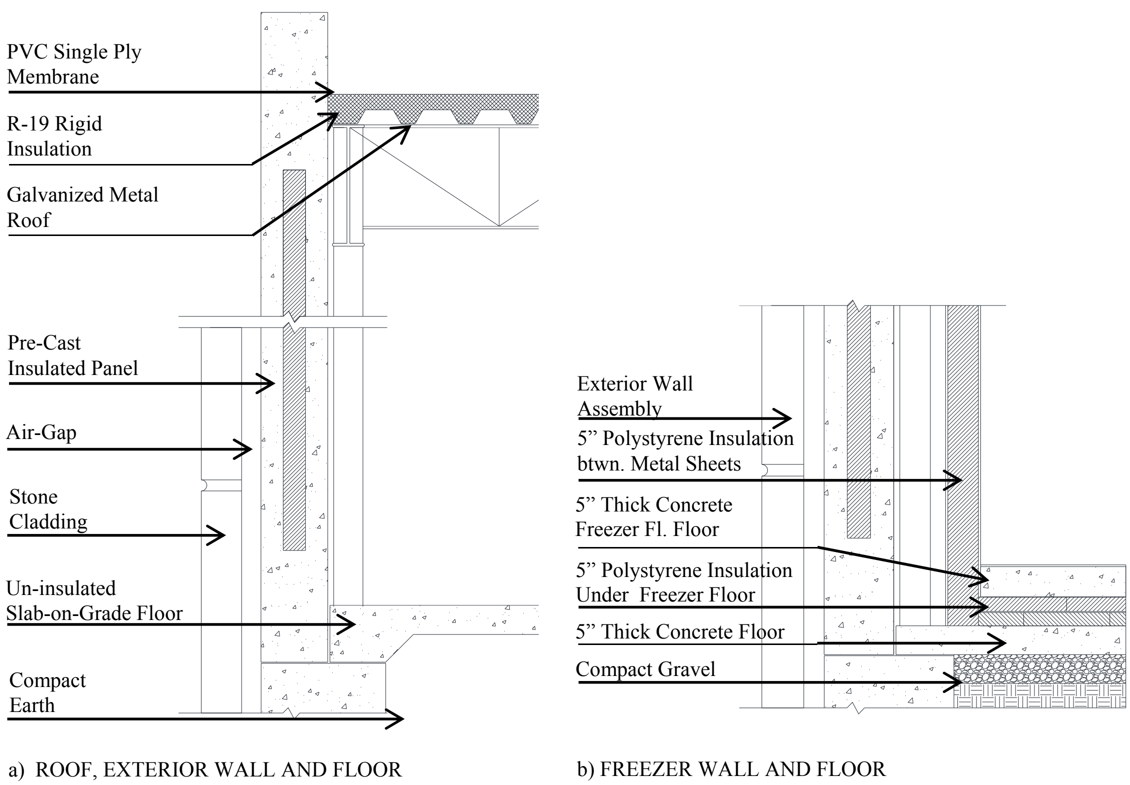

Figure 5-4: Typical Cross Sections of the Grocery Store

[10] The extruded polystyrene rigid foam insulation is manufactured as 2” and 3” thick boards by the Dow Chemical Company under the brand name of ‘FREEZERMATE™’ (Dow Building Solutions 2013).

5.4.3.4 *Window, Skylights and Door Openings*

The window and door area of the model was set at 10% of the front wall area, which corresponds to the areas of windows and entrance doors in the case-study store. No specifications were provided regarding the U-value and the SHGC of these windows and doors in the drawings documenting the case-study store. Hence, specifications for windows prescribed in ASHRAE Standard 90.1-2004 were considered[11]. The specifications included a U-value of 1.22 and an SHGC of 0.25 for all vertical glazing except north and an SHGC of 0.61 for glazing facing the north. Loading dock doors are modeled as un-insulated metal doors with an overall U-value of 2.08 Btu/hr ft^2 F. This specification was selected from the construction library provided by eQUEST-Refrigeration.

122 acrylic skylights[12] are installed on the roof of the case-study store covering an area of 3,959 ft^2 of the roof[13]. This amounts to 4.55% of the roof area. This area is estimated from the construction drawings provided for the case-study store. Specifications for the center of glass U-value of 0.71, an SHGC of 0.61 and a visible transmittance of 60% were provided in the case-study drawings and were adopted in the simulation model.

5.4.3.5 *Infiltration*

Infiltration or air leakage is a difficult quantity to measure in large buildings. At best guidelines for individual envelope components are available in the ASHRAE Standard 90.1 codes[14]. In this study infiltration has been modeled in terms of air changes per hour (ACH)[15] for all zones except the freezers, coolers and preparation rooms. The value for infiltration is set at 0.161 ACH and has been adopted from Hale et al. (2008).

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[11] Table 5.5-2, Building Envelope Requirements for Climate Zone 2 (A, B), 10% Vertical Glazing, ASHRAE Standard 90.1-2004.

[12] The acrylic skylights were provided by Naturalite (Model No. CWTL6274). This model has been discontinued. Similar models can be found from Wasco Sentinel Series (Model No. DDCAFP6476) specifications of which have been used in the simulation model.

[13] Daylighting sensors were installed in the grocery store, below these skylights to take advantage of the light coming into the interior space. However, it was observed that the daylighting sensors did not work and the lighting system is on all the time.

[14] Specifications for air leakage are provided in all the ASHRAE Standard 90.1 – 1989, 1999, 2001, 2004, 2007 and 2010 codes. However, the specifications are limited to specifications for sealing the building envelope, air leakage rates for materials and assemblies, and in some cases the requirement of loading dock weather seals and vestibules. No information is provided to determine the permissible infiltration rates for the spaces in the building.

[15] DOE-2.2 provides three methods of inputting infiltration in the simulation model. These include the air change, residential, Sherman-Grimsrud, and crack method. The air-change method requires either air changes per hour (ACH) or CFM/ft2 to be specified for the space. The ACH input incorporates the used of wind speed, while the CFM/ft2 method does not incorporate wind speed into the calculations for infiltration (Hirsch et al., 2006).

According to ASHRAE Handbook of Refrigeration, heat gain from infiltration air and associated equipment loads can amount to more than half the total refrigeration load of warehouses and similar applications such as walk-in freezers, coolers and preparation rooms (ASHRAE 2006). Infiltration has been modeled in terms of CFM/ft^2 for the freezers, coolers and preparation rooms. The CFM/ft^2 method was chosen to represent infiltration for the freezers, coolers and preparation rooms as they can be considered as interior zones and hence not affected by wind speed. The value for infiltration is set at 0.07 CFM/ft^2 which is the default value used in the sample eQUEST-Refrigeration file.

According to the report by Hale et al. (2008), a constant infiltration rate was initially assumed for all hours assuming that the HVAC system was always enabled and pressurizing the building. The same schedule was used for all the spaces in the model.

5.4.4 Occupancy, Lighting and Plug Loads (Table 5-3)

For the initial base-case model, the number of people in the general merchandise and display case zone was adopted from Hale et al. (2008). The number of people in other zones was determined after discussions with the store management. Occupancy schedules for the general merchandise zone and the display case zones were also adopted from Hale et al. (2008) and were been modified to reflect the operation hours of the store. The occupancy schedules for other zones such as the bakery, loading docks, freezers, coolers and preparation rooms were determined after discussions with the store management. Table B-3, in Appendix B provides the number of people in each zone of the grocery store simulation. The occupancy schedules are presented in Table B-16 in Appendix B of this study.

Lighting energy consumption was input in the model in terms of W/ ft^2. In the initial model the space-by-space method provided in ASHRAE Standard 90.1-2004 was used to determine the lighting power levels. The specifications were changed in subsequent iterations to provide a closer match to the specifications prescribed in the case-study store. The lighting power values assumed for the different spaces are provided in Table B-4. Lighting schedules were assumed from the case-study store and are presented in Table B-17. Exterior lighting in the initial simulation model was arbitrarily set at 0 kW. This value was subsequently changed to 2 kW to match the specification in the case-study store. The schedule required the exterior lights to operate for 12 hours per day all year round.

The initial values for plug and process loads were adopted from Deru et al. (2011a) and a sample eQUEST-Refrigeration input file for a grocery store. These values were incorporated in the simulation model both in terms of W/ft^2 as well as total wattage for each space. Table B-5 in Appendix B provides the plug and process loads assumed for each space in the simulation base-case model. The schedule for plug and process loads were assumed from Deru et al. (2011a) and are presented in Table B-18 of Appendix B.

5.4.5 HVAC Systems of the Grocery Store (Table 5-4)

5.4.5.1 HVAC System Efficiency

The case-study store is served by 15 roof top units (RTUs) and 8 air-cooled condensing units (ACCUs). Each of the RTUs is provided with a gas-fired furnace to provide for the space heating needs. However, in the simulation model similar zones were combined into one zone resulting in 5 thermal zones in the simulated base-case which are served by packaged single zone (PSZ) units. As in the case-study store, the PSZ units in the simulation model are installed with individual gas-furnaces to provide space heating. The HVAC systems were sized in the base-case simulation model using the sizing runs[16, 17]as proposed in ASHRAE Standard-90.1 2004[18]. A sizing ratio of 1:1.2 was for the HVAC systems as defaulted in the eQUEST-Refrigeration program. Minimum Energy Efficiency Ratio (EER) and heating efficiency for the base-case model were set according to the defaulted values provided in eQUEST-Refrigeration program Cooling and heating equipment specifications are provided in Table B-6.

5.4.5.2 Supply and Exhaust Fans

The supply fans in the initial base-case simulation model were constant-volume fans modeled with an assumed exterior static pressure of 1.25 inch WG[19]. The total efficiency of each fan unit was assumed to be 53%. The supply fan specifications mentioned above were adopted from the specifications prescribed in the sample file for a grocery store eQUEST-Refrigeration.

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[16] eQUEST-Refrigeration allows the user size the cooling and heating systems by performing sizing runs. The use of this option requires the input of design-day conditions for the given location of the building. Two separate design days are required to be input one for heating and one for cooling. The program then determines design peak loads by simulating the buildings for a 24-hour period on each of the design days. Special schedules have been specified for estimating the size of the heating and cooling systems. These include scheduling zero occupancy and lights on the heating design day and maximum occupancy and lights on the cooling design day.

[17] Weather conditions used in the sizing runs was adopted from design-day specifications for College Station Texas provided in ASHRAE Handbook of Fundamentals (2009), Chapter 14. The design days were developed using 99.6% heating design temperatures and 1% dry-bulb and 1% wet-bulb cooling design temperatures.

[18] Section 11.3.2 (i), HVAC Systems, ASHRAE Standard 90.1-2004.

[19] eQUEST-Refrigeration requires entering the total HVAC system static pressure. On the other hand, the case-study drawings provided only external static pressure of 0.8 inch WG. The internal static pressure for each unit would need to be determined from manufacturers' data on the brake horsepower of the fan at the estimated external static pressure. This calculation was performed by this study in a subsequent iteration in which the fan static pressure was adjusted.

Exhaust fans were modeled in the "Bakery" zone of the base-case model. In this zone the exhaust fan was modeled with an assumed flow of 5,800 CFM. The default values of the eQUEST simulation program were used to model the total static pressure across the exhaust fan and the overall efficiency. The default values are 0.3 inch WC for static pressure across the fan and an efficiency of 40%[20]. Exhaust fan specifications are provided in Table B-7 of Appendix B in this study.

5.4.5.3 Supply and Outdoor Air Flow Requirements

For the initial base-case run the design flow rate was arbitrarily selected to be 0.5 CFM/ft^2. This number was later adjusted in a subsequent iteration to include the air-flow rates provided in the specifications for the case-study store. For the initial base-case run the outdoor air requirements are modeled according to the requirements stated in ASHRAE Standard 62.1-2004[21]. Specifications are provided in Table B-6, of Appendix B in this study. This number was later adjusted in a subsequent iteration to include the outside air-flow rates provided in the specifications for the case-study store.

5.4.6 Service Hot Water Equipment (Table 5-5)

The initial base-case had two gas-fired water heaters, which represent the same number of gas-fired water heaters in the case-study store[22]. The water heaters in the base-case model are located in the interior of the store (i.e., the bakery zone and the general merchandise zone of the simulation model). 119 and 40 gallon tank sizes were modeled to match the specifications from the case-study store. Hot water temperatures in the tank were set at 140°F[23] (Fisher-Nickel 2010) for the larger tank and 125°F (case-study store specifications) for the smaller tank. In the initial model the peak hot water demand was set at 1.93 gpm based on provisions in the ASHRAE HVAC Applications Handbook (ASHRAE 1999)[24]. The specifications for SHW equipment is

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[20] eQUEST-Refrigeration requires an input of combined efficiency of the zone exhaust fan and motor at design conditions.

[21] ASHRAE Standard 62.1 -2004, Table 6-1, Minimum Ventilation Rates in Breathing Zone. Specifications for supermarkets :7.5 CFM/person and 0.06 CFM/ft^2.

[22] One of the service water heater installations in the case-study store had a boiler with a 119 gallon hot water storage tank. However, this configuration could not be modeled in eQUEST-Refrigeration. A 119 gallon service water heater was modeled instead.

[23] No information was available from the case-study store for the tank water temperature of the 119 gallon water heater. Information from a publication by Fisher-Nickel inc. on efficient water heating in commercial kitchens was used instead.

[24] According to ASHRAE HVAC Applications Handbook (ASHRAE 1999), Chapter 48, Pg.48.12, grocery stores typically use 300 -1000 gallons of hot water per day.

provided in Table B-8 of Appendix B in this study. The schedule for hot water usage is adopted from Hale et al. (2008). The schedule is provided in Table B-20 of Appendix B in this study.

Two recirculation pumps were modeled to operate along with the 119 gallon tank. These pumps have a flow rate of 2.5 gpm and a 25 ft. of pressure rise across the pump when the pump is running. The power consumption of the recirculation pumps was modeled to be 1/8 HP per pump. The pumps were assumed to operate continuously in the base-case model as no information could be obtained regarding the operation of these pumps from the case-study store.

eQUEST-Refrigeration also requires the input of standby loss coefficient for the modeled water heater. The standby heat loss coefficient (UA) of the gas-fired heater was determined by using the following equation adopted from Thornton et al. (2010):

$$UA = (SL \times RE)/70$$

Where, UA = Standby heat loss efficiency (Btu/hr F)

SL = Standby heat loss[25] (Btu/hr)

RE = Recovery efficiency (assumed to be 0.95 for 119 gallon gas-fired storage water heaters[26])

70 = Difference in temperature between stored water thermostat setpoint and ambient air temperature at the test condition (°F).

Appropriate UA for the gas-fired water heaters was input in a subsequent iteration of the calibration process.

5.4.7 Refrigeration System (Table 5-6)

eQUEST-Refrigeration provides a component approach to model refrigeration systems. In this approach individual components are modeled separately and then connected to each other to build one or more systems (Hirsch et al., 2008). In order to simulate the refrigeration system components such as the liquid, suction and discharge circuits, condensers and suction groups, which consist of one or more compressors and are first defined. Then, each of the demanders on the system is defined. These include display-cases and walk-in spaces. Control parameters such as compressor sequencing, suction pressure control, condenser temperature control method and usage profiles are also defined along with the components. A sample component configuration

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[25] The appropriate standby losses were calculated using the following equation (ASHRAE 2010, Table 7.8, Performance Requirements for Water Heating Equipment, Gas storage water heaters >75,000Btu/hr): $SL=Q/800+110\sqrt{V}$. Where SL = standby heat loss (Btu/hr), Q = rated input power (Btu/hr), and V = rated storage tank volume (gallons).

[26] The 0.95 RE was estimated by averaging the different RE values obtained for 119 gallon gas-fired water heaters in the AHRI directory of certified product performance (AHRI 2008).

for a refrigeration system is provided in Figure 5-5. The program then uses this information along with other information required to simulate the building, such as building envelope, HVAC systems and weather data to derive hourly profiles and subsequently the energy consumption of the whole building (Hirsch et al., 2008).

5.4.7.1 Refrigeration Compressors

The refrigerant used in the base-case store is R-22, which is the same as what is used in the case-study store. The refrigerated cases and areas served by four split-suction temperature parallel compressor racks[27] were chosen to best represent the case-study store, which includes semi-hermetic reciprocating compressors with 7 to 8 compressors per rack. As seen in the case-study store, the compressors in the base-case model are cycled on and off to match the varying loads of the refrigeration system. The capacity of each compressor in the initial simulation run was allowed to be defaulted by the simulation program. The compressor COPs in the initial run were also allowed to be defaulted by the simulation program[28]. Figure 5-6 below provides a schematic view of the compressor rack layout in the case-study store.

5.4.7.2 Refrigeration Condensers

In the case-study store there is one air-cooled condenser assigned for each refrigeration rack. In total, there are four air-cooled condensers operating in the case-study store. The condensers were operated using a fixed head pressure control[29]. A 10°F temperature differential is assumed in the initial run of the base-case[30]. The condenser capacity is controlled by cycling the condenser fans. Design condenser temperatures are assumed from the specifications of the case-study store and are set at 115°F and 120°F depending on which compressor rack the condenser serves. The specifications for the air-cooled condensers used in the case-study store are provided in Table B-15 and are incorporated to the base-case model.

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[27] Parallel compressor systems consist of two or more compressors operating between a common suction head, common discharge head and a common receiver. The main advantage of using such systems is their ability to nearly exactly match refrigeration loads. Split-suction parallel compressor racks have the ability to handle both medium and low temperature applications. Installation of split-suction parallel compressor racks has become a standard practice in the grocery store industry. Details of the four compressor racks installed in the case-study store and modeled in the base-case are presented in Table B-9, Appendix B.

[28] Specifications for the refrigeration rack compressor were obtained at a later date. Hence, these specifications were incorporated in a subsequent iteration.

[29] Fixed head pressure is a control strategy used in condensers where the condensing setpoint is set at a constant value at all hours.

[30] This throttling range defines the points at which the fans cycle. The 10°F ΔT was later changed to match the specifications obtained from the case-study store as seen in Table B-15, Appendix B.

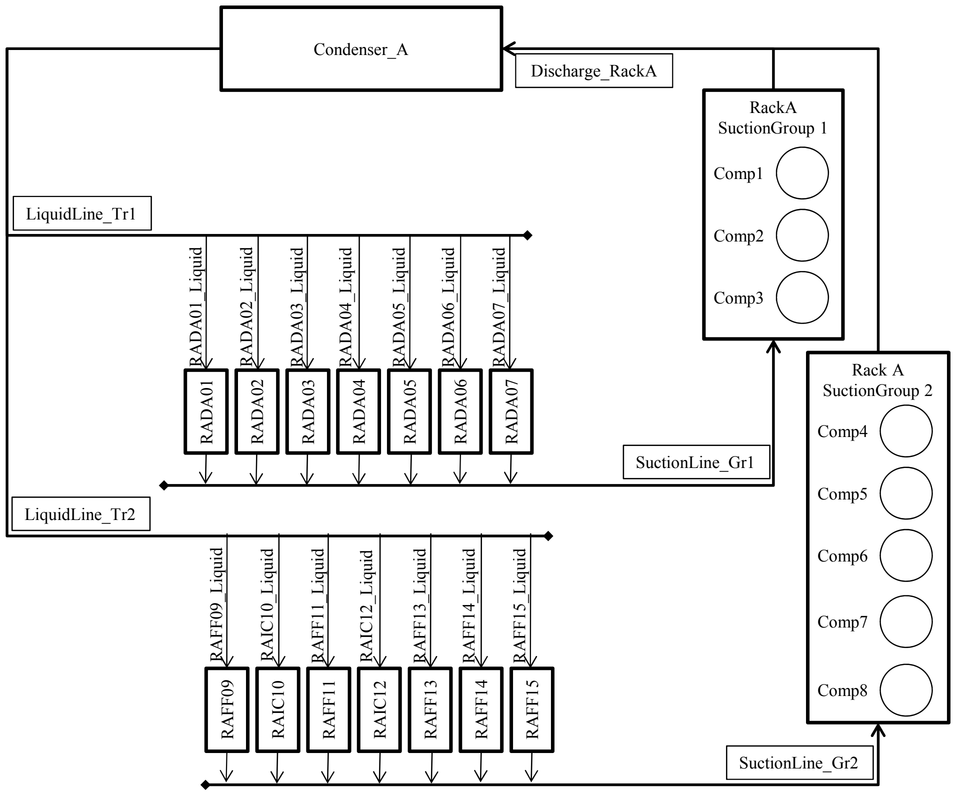
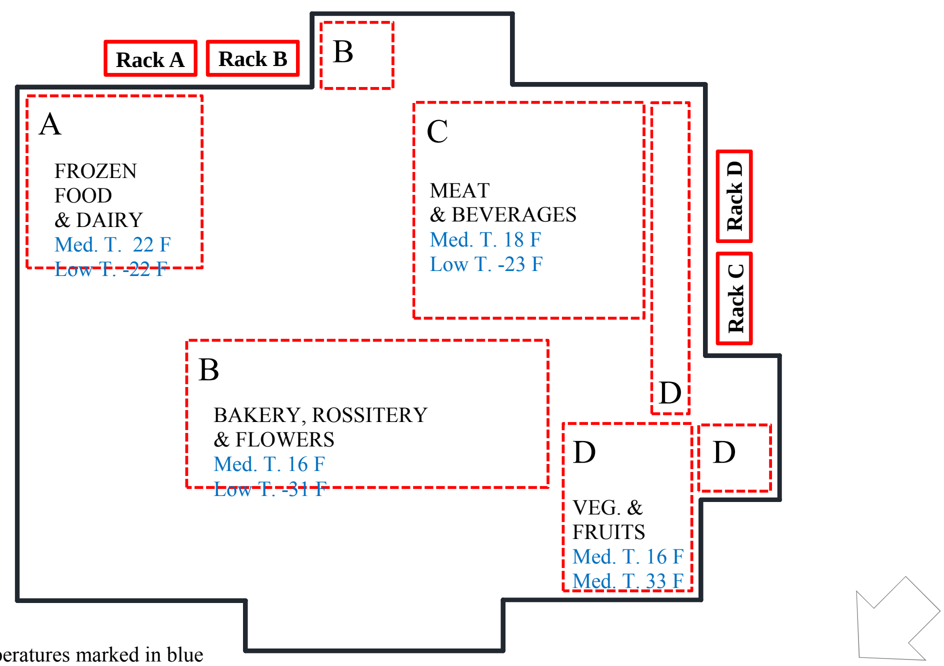


Figure 5-5: Schematic Configuration for the Refrigeration System in eQUEST-Refrigeration (For Compressor Rack A)



Note: Suction temperatures marked in blue

Figure 5-6: Schematic View of the Compressor Rack Layout in the Case-Study Store

5.4.7.3 Refrigerated Display Cases and Spaces

There are several types of display cases in the case-study store that are modeled in the base-case simulation model (Table B-10). The types of refrigerated display cases include: open vertical cases, vertical cases with doors and coffin-type cases. The number / length of each display-case type are provided in Table B-11. Refrigerated spaces in the case-study store also include coolers, freezers and preparation rooms. These spaces are also modeled in the base-case simulation model. The areas of these spaces are also provided in Table B-11.

The display cases in the case-study store operate at different temperatures. The different temperatures are incorporated in the base-case simulation model. A complete list of display cases in the store and the corresponding temperatures is provided in Table B-12. Several of these display cases are connected to the same suction group of compressors and hence operate between common liquid and suction lines. In order to maintain the desired temperature in the display cases temperature control needs to be provided. Temperature control in the display case units of the case-study store are provided by evaporator pressure regulators (EPRs) [31] used in conjunction with the thermostatic expansion valves (TXVs). In more recent installations of display cases electric evaporator pressure regulators (EEPRs)[32] have been used. Figure 5-7 below provides a schematic view of Rack A, suction group 2 in which EPRs are installed.

Off-cycle defrost was used for the medium temperature display cases and electric defrost was used in the low temperature display cases. In both cases the defrosting mode is initiated by a timer and terminated using a temperature set point. Schedules for defrost initiation in the base-case model were set according to the specifications in the grocery store. Power requirements for electric defrost in the base-case model were also adopted from the specifications in the case-study store. The defrost power requirements, schedules and termination temperatures for the display-cases that have been adopted from the case-study store and used in the base-case model are documented in Table B-13.

[31] The EPR prevents the refrigerant in the evaporator from going below a particular pressure. The bellows in the EPR valve senses evaporator pressure and throttles the suction gas to the compressor, allowing the evaporator pressure to go as low as the pressure setting on the valve (Whitman and Johnson, 1991).

[32] EEPR valves are designed for a closer temperature control in display cases than what is shown by EPR valves.

Some display cases have installed lighting systems. Lighting power for the various display cases was provided from the specifications for case-study store and documented in Table B-14. Lighting schedules were set as ‘always-on’ to match the current operation in the case-study store, which was reduced to 50% of full power during unoccupied hours.

Low temperature display cases require the installation of anti-sweat heater controls to prevent condensation on the surface of the case. Therefore, the base-case store has anti-sweat heater controls installed with rated power to match specifications in the case-study store. The specifications for anti-sweat heaters in the case-study store are provided in Table B-14. The heaters are modeled to operate at all times.

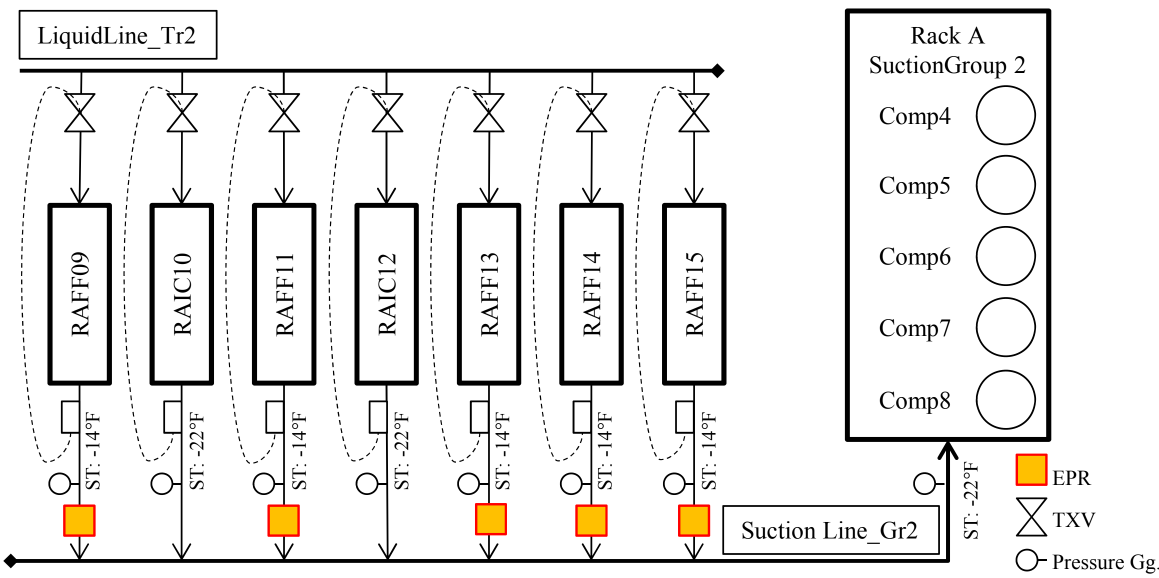


Figure 5-7: Schematic View of EPRs Installed in Rack A, Suction Group 2

Table 5-1: Specifications for Building Program and Building Form in the Base-Case Grocery Store

Base-Case Parameters	Units	Case-Study Store	Initial Input Value	Reference
BUILDING PROGRAM				
Location	-		Climate Zone 2A	-
Total Area	ft²	93,876	92,952	-
Zoning	-	23 Zones	8 zones, Table B-1	-
Space Env. Conditions	°F	72 / 75	72	Initial assumption
BUILDING FORM				
Number of Floors	-	1	Case-study store	-
Aspect Ratio	-	1:1.5	Case-study store	-
Floor to Floor Height	ft	20 (Average height)	Case-study store	-
Orientation	-	Front facing north-west	Case-study store	-

Table 5-2: Specifications for the Building Envelope in the Base-Case Grocery Store

Base-Case Parameters	Units	Case-Study Store	Initial Input Value	Reference
BUILDING ENVELOPE				
Exterior Walls				
Construction	-	Stone veneer 3 5/8"(OUT) Air space 1 ¾" Insulated concrete panels 6" Gypboard (IN)	Case-study store	-
Insulation	hr.ft².F / Btu	No information	R-2 rigid polystyrene	Canadian Pre-cast Concrete Institute (CPCI) n.d.
Roof				
Construction	-	Single ply membrane (OUT) Cont. rigid insulation Gal. metal roof (IN)	Case-study store	-
Surface Properties	-	No information	Solar reflectance: 0.4 Thermal emittance: 0.9	eQUEST model default
Insulation	hr.ft².F / Btu	19	Case-study store	-
Slab-on-Grade Floor				
Construction	-	12 " compacted soil 4 "conc. HW	Case-study store	-
Insulation	hr.ft².F / Btu	0	Case-study store	-
Freezer and Cooler Interior Walls and Roof				
Construction	-	Steel siding (OUT) 5" FREEZERMATE™ insulation Steel siding (IN)	Case-study store	-
Insulation	hr.ft².F / Btu		5" 'FreezerMate' insulation (R-25)	Dow Building Solutions 2013
Freezer Floor				
Construction	-	12" soil (OUT) 4" rigid insulation 5" conc. HW (IN)	Case-study store	-
Insulation	hr.ft².F / Btu	20	Case-study store	-

Table 5-2: Continued

Base-Case Parameters	Units	Case-Study Store	Initial Input Value	Reference
BUILDING ENVELOPE				
Fenestration				
Area & Location	-	10% WWAR for front wall (Doors + Windows)	Case-study store	-
U-value	hr.ft².F / Btu	No information	Single pane tinted - 1000 From WINDOW-5 library	ASHRAE Standard 90.1-2004 specifications for Climate Zone 2A
SHGC	-			
Window Shading	ft	4' horizontal interior shades	Case-study store	-
Skylights				
Area & Location	ft²	Unit: 62-5/8" x 74-5/8" Total: 3959.12 (122 Units) (4.22% of roof area)	Unit: 5.19 x 6.30 Total: 3433.19 (105 Units) (3.66% of roof area)	Initial assumption
U-value	hr.ft².F / Btu	0.71	Case-study store	-
SHGC	-	0.61		
Visible Transmittance	%	60	50	Initial assumption
Infiltration				
Rates	ACH CFM/ft^2	No information	All others: 0.161 Freezer/cool./ prep.: 0.07	Hale et al., 2008 eQUEST model default
Infiltration Schedule	-	No information	Constant, set at design value Same schedule for all zones	Deru et al., 2011

Table 5-3: Specifications for Occupancy, Lighting and Plug Loads in the Base-Case Grocery Store

Base-Case Parameters	Units	Case-Study Store	Initial Input Value	Reference
OCCUPANCY, LIGHTING & PLUG LOADS				
Occupancy				
Occupant Density	-	Customers: 3000 persons/day Service personel: Fixed number	Various, Table B-3	Initial assumption -
Occupancy Schedule	-	Customers: Monthly variations Service personel: Fixed schedule	Various, Table B-16	Hale et al., 2008 -
Lighting				
Int. Lighting Power Density	W/ft²	Information not available	Various, Table B-4	Calculated from ASHRAE Standard 90.1-2004
Int. Lighting Schedule	-	Occupied: 95% Unoccupled: 25%	Case-study store, Table B-17	-
Ext. Lighting	kW	2	0	Arbitrary selection
Ext. Lighting Schedule	-	Astronomical clock	Case-study store	-
Plug & Process Loads				
Equip. Power Density	W/ft²	Various	Various, Table B-5	Deru et al., 2011 Sample eQUEST file
Equip. Schedule	-	No information	Various, Table B-18	Deru et al., 2011

Table 5-4: Specifications for HVAC Systems in the Base-Case Grocery Store

Base-Case Parameters	Units	Case-Study Store	Initial Input Value	Reference
HVAC SYSTEM				
System Specifications				
Cooling Type	-	15 PSZ Units 8 ACC Units	5 PSZ Units, Table B-1	-
Heating Type	-	Gas furnace	Gas furnace	-
System Size	-	Various	Design-day calculation	ASHRAE 90.1-2004
Cooling Efficiency	EER Btu/Btu	Various	All others, Table B-6 Freezer / cooler: 0.36	eQUEST model default
Heating Efficieny	HIR	Various	All others, Table B-6 Freezer/cooler/bakery: None	eQUEST model default
Supply Fan Specifications				
Ext. Static Pressure	in. WG	0.8	1.25	Initial assumption
Total Efficiency	%	No information	53	Initial assumption
Mech. Efficiency	%	No information	55	Initial assumption
Fan Control	-	Constant volume	Constant Volume	Case-study store
Exhaust Fan Specifications				
Flow	CFM	6,763, Table B-7 (For both "Bakery" and "General Merchandise"	5,800, Table B-7 (For both "Bakery" zone only)	Initial assumption
Static Pressure	in. WG	0.8 (Ext. pressure)	0.3, Table B-7	eQUEST model default
Total Fan Efficiency	%	No information	40	eQUEST model default
Ventillation Requirements				
Design Flow	CFM/sqft	Various, Table B-6	0.5, Table B-6	Initial assumption
Outdoor Air Req.	CFM/person	Various, Table B-6	14.6, Table B-6	Initial assumption

Table 5-5: Specifications for Service Hot Water Heating in the Base-Case Grocery Store

Base-Case Parameters	Units	Case-Study Store	Initial Input Value	Reference
SERVICE WATER HEATER				
Hot Water Heater / Boiler				
Number of Heaters	-	2	Case-study store	-
Location of Heaters	-	Interior zones	Tank A: Bakery Tank B: Gen. merchandise	-
Fuel Type	-	Gas	Gas	-
Capacity	-	Heater A: 750,000 Btu/hr Hearer B: No information	Tank A: 750,000 Btu/hr Tank B: Auto-sized	- eQUEST model default
Demand	gpm	No information	1.5 0.15	1999 ASHRAE Applications Handbook
Tank Temperature	°F	Tank A: No information Tank B: 125	Tank A: 140 Tank B: 125	Hale et al.2008 -
Tank Size	Gallons	Tank A: 119 Tank B: 40	Tank A: 119 Tank B: 40	-
Tank UA	Btu/f-F	No information	Tank A: 35.7 Tank B: 12	eQUEST model default
HW Schedule		No information	See reference	Hale et al. 2008
Recirculation Pumps (For Tank A)				
Number	-	2	2	-
Flow	gpm	2.5	2.5	-

Table 5-6: Specifications for Refrigeration System in the Base-Case Grocery Store

Base-Case Parameters	Units	Case-Study Store	Initial Input Value	Reference
REFRIGERATION SYSTEM				
Refrigerant		R-22	Case-study store	-
System				
Rack Type		Split suction temperature Parallel compressor racks Table B-9	Case-study store	-
Configuration		4 split temperature racks	Case-study store	-
Size of Racks		Various, Table B-9	Case-study store	-
Compressor number		7-8 per rack	Case-study store	-
Compressor type		Semi-hermetic (Copeland Discus)	Case-study store	-
Cap. /Comp. Unit	Btu/hr	Various, Table B-9	eQUEST-Refg. model default	-
Compressor COP		Various Calculated from Table B-9	eQUEST-Refg. model default	-
Display Cases				
Type and Number		Various, Table B-10 & 11	Case-study store	-
Temp. Ctrl.		Thermostat / EPR / EEPR Table B-12	Case-study store	-
Defrost Type		Med. temp.: Off-cycle Low temp.: Electric Table B-13	Case-study store	-
Defrost Control		Initiation: Time Termination: Time / Temperature Table B-13	Case-study store	-
Defrost Initiation Schedule		Various, Table B-13	Various	eQUEST-Refg. model default
Defrost Energy		Various, Table B-13	Case-study store	-

Table 5-6: Continued

Base-Case Parameters	Units	Case-Study Store	Initial Input Value	Reference
REFRIGERATION SYSTEM				
Display Cases				
Lighting		Various, Table B-14	Case-study store	-
Lighting Schedule		Always on	Case-study store	-
Anti-Sweat Heater		Various, Table B-14	Case-study store	-
Anti-Sweat Heater Ctrl.		Pulsating	Always on	eQUEST-Refg. model default
Condensers				
Configuration	-	One per refrigeration rack	Case-study store	-
Type	-	Air-cooled	Case-study store	-
Temp. Differential	°F	Various, Table B-15	10 °F	Initial assumption
Control	-	Fixed head pressure control	Case-study store	-
Subcooling Effect	-	0.26	Case-study store	-
Electric Input Ratio x TD	-	Various Calculated from Table B-15	0.55	
Capacity Control	-	Cycle fans	Case-study store	-
Design Cond. Temp.	°F	Rack A, B, C: 115 Rack D: 120	Case-study store	-

5.5 Calibration Procedure

The calibration procedure involved comparing information obtained from the simulation model with the utilities bills of the store as well as the information obtained from the on-site monitoring system. The process was divided into 3 sections comparing:

- Energy use intensities (EUI)
- Monthly utility bills
- Hourly data from on-site monitoring system

5.5.1 Energy Use Intensity (EUI) Check

After developing the initial model the resultant EUI was compared with that of the case-study store. The measured EUI of the store for the year 2009 (January to December) was 211 kBtu/ft^2-year. The EUI for the base-case model was 191.6 kBtu/ft^2-year. The numbers, though not the same are within 10 % range of each other, which was found acceptable by this study. Corresponding Commercial Building Energy Consumption Survey (CBECS) numbers for the year 2009 for Climate Zone 2 was not available. However these numbers were well within the national average EUI of 199.7 kBtu/ft^2-year provided by the 2003 CBECS (US EIA 2012). The electricity EUI for the case-study store and the base-case model were 49.4 and 50.0 kWhr/ft^2-year respectively. These numbers were well within the national average electricity EUI of 49.4 kWhr/ft^2-year provided by the 2003 CBECS (US EIA 2012).

5.5.2 Monthly Calibration

Electricity data obtained from the monitoring system at the case-study store was compared to the monthly consumption information from the utility bills. A difference of 657,140 kWh was observed between the electricity use from the utility bills and the corresponding data from the monitoring system. This value translates to 75 kW of constant power usage throughout the year, which was not accounted for by the monitoring system. The monitoring system underestimated the electricity consumption by approximately 14%. A calibration error was assumed in the monitoring system and partially confirmed by the store management[33]. The results of the monthly calibration of the initial model are presented in Figure 5-8 through Figure 5-13.

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[33] The store management confirmed that an approximate 5% difference was observed in each of the three current transducers that are set up to measure power for the monitoring system from each of the three power phases that supply power to the store.

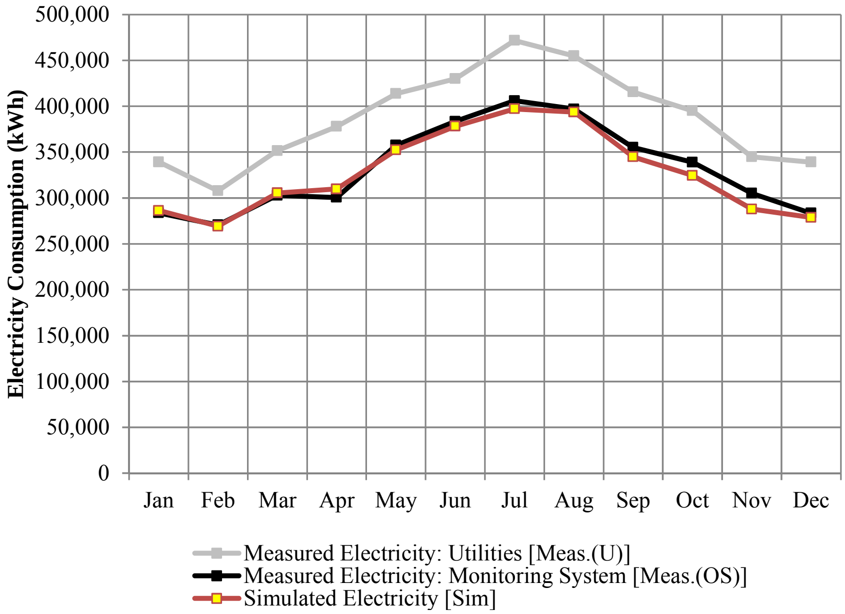

Figure 5-8: Results from Monthly Calibration of the Initial Base-Case Run with Measured Data[34] [35]for Whole-Building Electricity Usage –January to December 2009

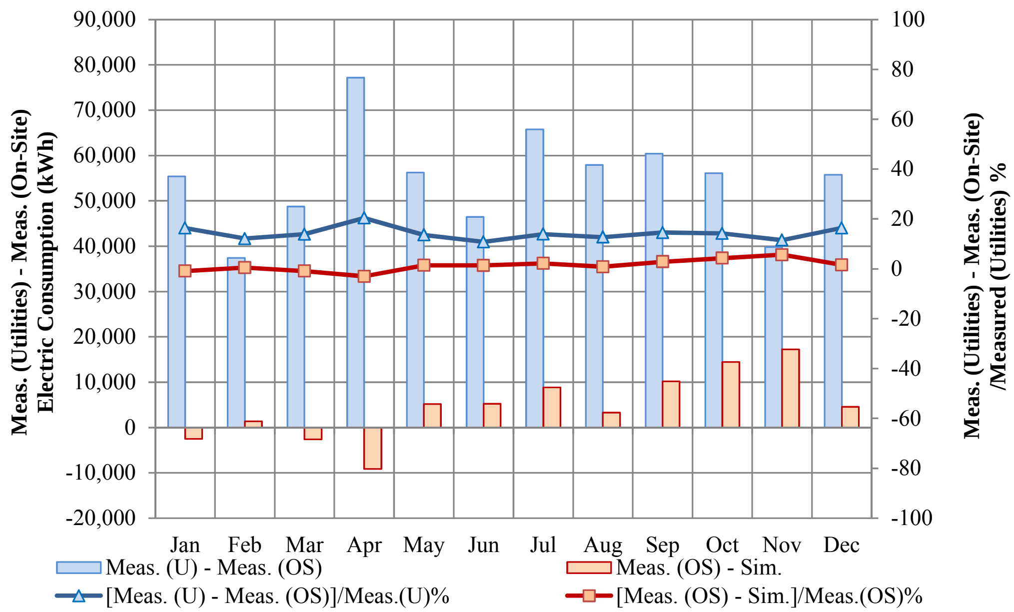

Figure 5-9: Residuals Comparing Simulated Data from the Initial Base-Case Run and Measured Data for Whole-Building Electricity Usage - January to December 2009

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[34] For on-site measurement option, it was assumed that data was reported at the end of every hour.

[35] For the on-site measurement option, only 20 days were available for December. Hence, data had to be extrapolated to obtain results for the entire month.

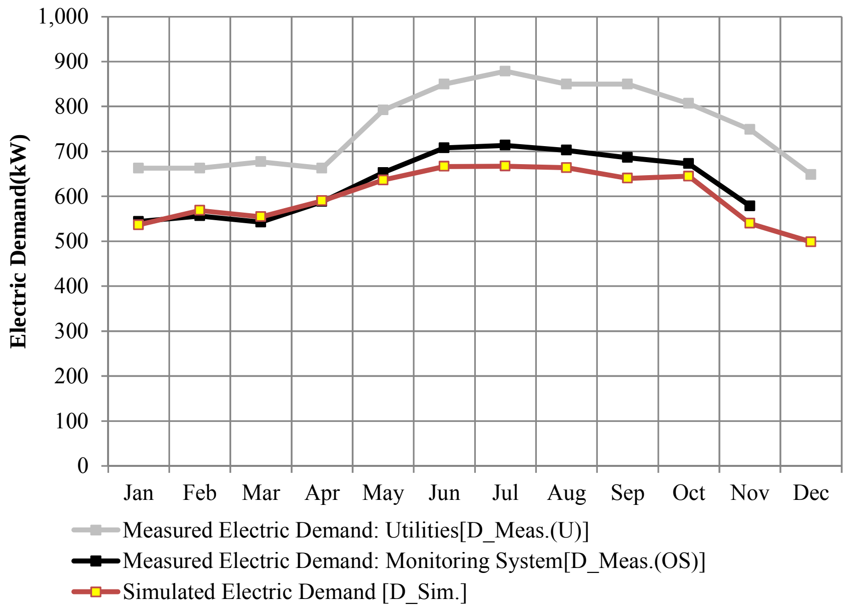

Figure 5-10: Results from Monthly Calibration of the Initial Base-Case Run with Measured Data for Whole-Building Electric Demand - January to December 2009

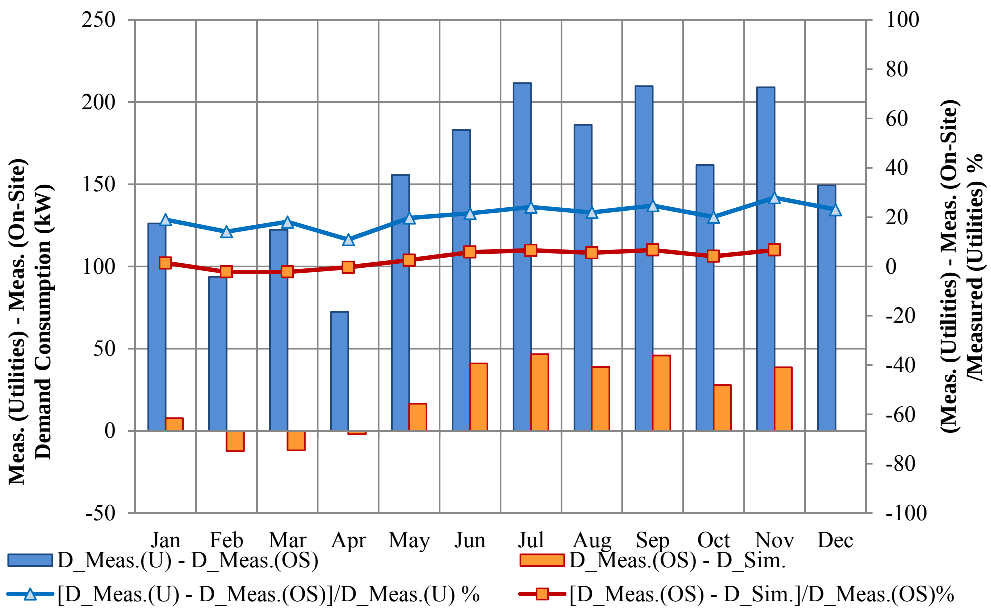

Figure 5-11: Residuals from Comparing the Initial Base-Case Run with Measured Data for Whole-Building Electric Demand - January to December 2009

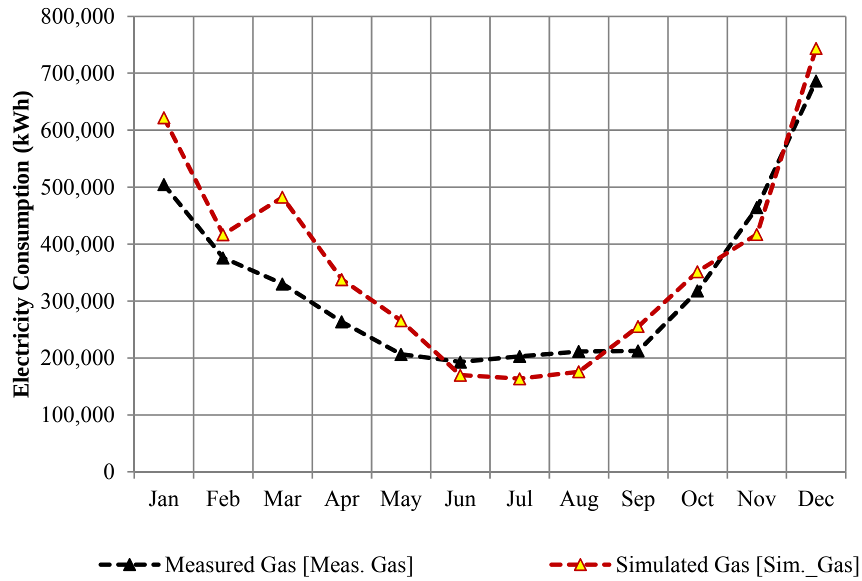

Figure 5-12: Results from Monthly Calibration of the Initial Base-Case Run with Measured Data for Gas Energy Usage –January to December 2009

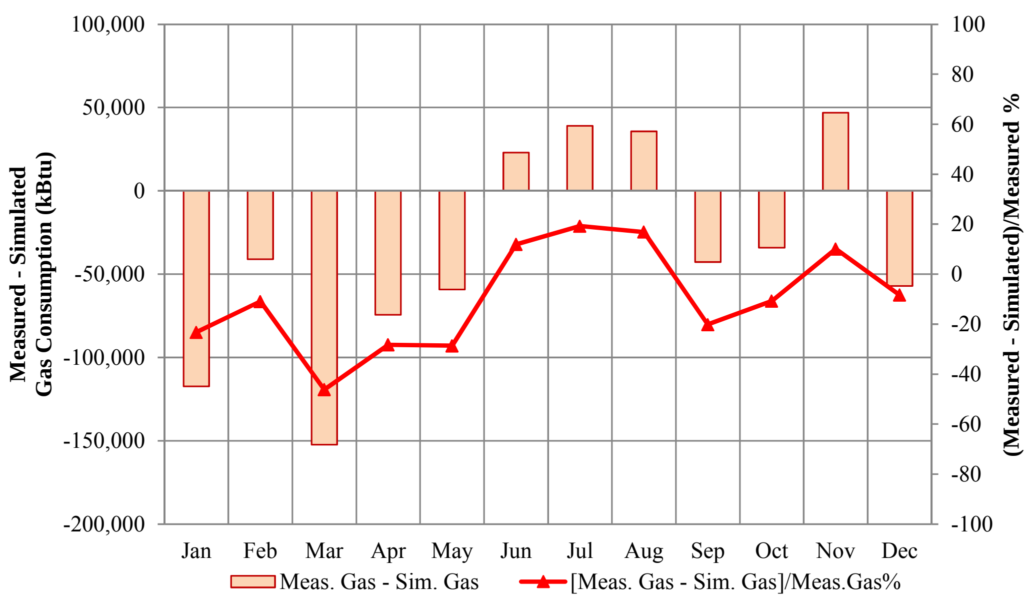

Figure 5-13: Residuals from Comparing Initial Base-Case Run with Measured Data for Gas Energy Usage - January to December 2009

No clues are provided regarding the pattern in which this extra energy is consumed. However, it was decided to move forward with the hourly calibration of electricity usage using the data from the monitoring system and account for the extra electricity by increasing the equipment power usage at the end of the hourly calibration process. There was no hourly information for the energy consumption resulting from usage of natural gas. Hence, calibration for energy consumption from gas is performed using monthly data from the iterative runs.

5.5.3 Hourly Calibration

In the final leg of the calibration process hourly data obtained from the simulation model was calibrated with measured hourly data obtained from the monitoring system of the store. Only whole-building electricity usage was calibrated using hourly information. Other information such as temperatures of spaces and refrigerated display cases, and power consumption from refrigeration racks were examined for additional insight into how the store operated. Due to the lack of hourly gas consumption data, calibration of corresponding gas energy consumption for the each iteration is performed on monthly basis. A monitoring diagram is presented in Figure 5-14.

5.5.3.1 Structure of Hourly Analysis

The calibration procedure included an analysis of the whole building electricity (WBE) consumption patterns with respect to the outdoor air temperatures. The procedure also compared measured and simulated data with respect to each other by means of scatter plots; with respect to hourly consumption patterns by means of time series plots (Hsieh 1988), with respect to distribution of WBE data by means of an inter-quartile analysis using box-whisker plots (Bou-Saada 1994, Haberl and Bou-Saada 1998).

When constructing the scatter plots, both the measured and simulated WBE data of the grocery store for the entire year was plotted against hourly outdoor air temperatures. Sample plots of WBE data for four months representing summer (July), winter (January), spring (March) and fall (October) seasons were also examined. In addition, corresponding residuals were plotted as scatter plots for the four sample months selected. In another variation of the scatter plot analysis, simulated hourly data for WBE consumption was plotted against measured hourly WBE consumption.

Time series plots for the sample months were also constructed to observe the diurnal and monthly consumption patterns of the WBE usage. The resultant residuals were also plotted as time series plots.

To better view the distribution of data, a statistical box plot analysis of the hourly power consumption was also created to determine the trend for minimum, maximum, 25%, 75% and median value of the WBE consumption. In order to create the box plot, the WBE data were sorted into 5°F temperature bins. The resultant data collected for the entire year was then sorted into minimum, maximum, 25%, 75% and median values of WBE consumption. This procedure was then repeated for the measured data. Finally, information from the simulated base-case run and subsequent iterations were then juxtaposed with the trends obtained from measured data.

5.5.3.2 Results and Observations

In the first set of iterations a sensitivity analysis was conducted where each parameter was adjusted and checked for reasonableness of the input value. The number of runs conducted to gauge the sensitivity of each of the components and the resultant variation in the values being changed are documented in Table 5-8. Each iteration was analyzed in terms of statistical quantities which include: root mean square error (RMSE), coefficient of variation of the root mean square error CV(RMSE) and mean bias error (MBE) (Krieder and Haberl 1994a and 1994b). A log recording the changes to the statistical quantities of RMSE, CV(RMSE) and MBE was maintained to track the calibration process. The log is presented in Table 5-7 through Table 5-10, Figure 5-21 and Figure 5-22. The first set of iterations regarded changes to the general assumptions made in the model such as observed holidays, daylight savings, and changes to the envelope, space conditions, HVAC systems and refrigeration systems.

5.5.3.2.1 Observations from the Initial Run

The first run incorporates the values of all the parameters noted in the section on the base-case description. The results are presented in Figures 5-15 through Figure 5-20. Figure 5-15, Figure 5-16 and Figure 5-17 provide a scatter plot analysis' Figure 5-18 and Figure 5-19 provide a time series plot analysis and Figure 5-20 provides a bin analysis of the hourly measured data obtained from the on-site monitoring system and data obtained from a simulation model of the grocery store.

On inspecting the time series plots (Figure 5-18 and Figure 5-19) the following discrepancies were observed:

- Simulated electricity consumption is under-predicted during unoccupied periods for the sample months of January, March and October. While in the sample month of July the simulation over-predicted the electricity usage.
- The simulation program accounts for several holidays, whereas the measured data did not.
- In the sample months of July and October spikes in the residual trends are observed. These are due to the mismatch in daylight savings assumptions between the simulation and the measured data. On closer examination it was observed that the spikes in residual data started on the 5th of April[36] and ended on 25th of October, which corresponded to the daylight saving schedule for 1998[37].

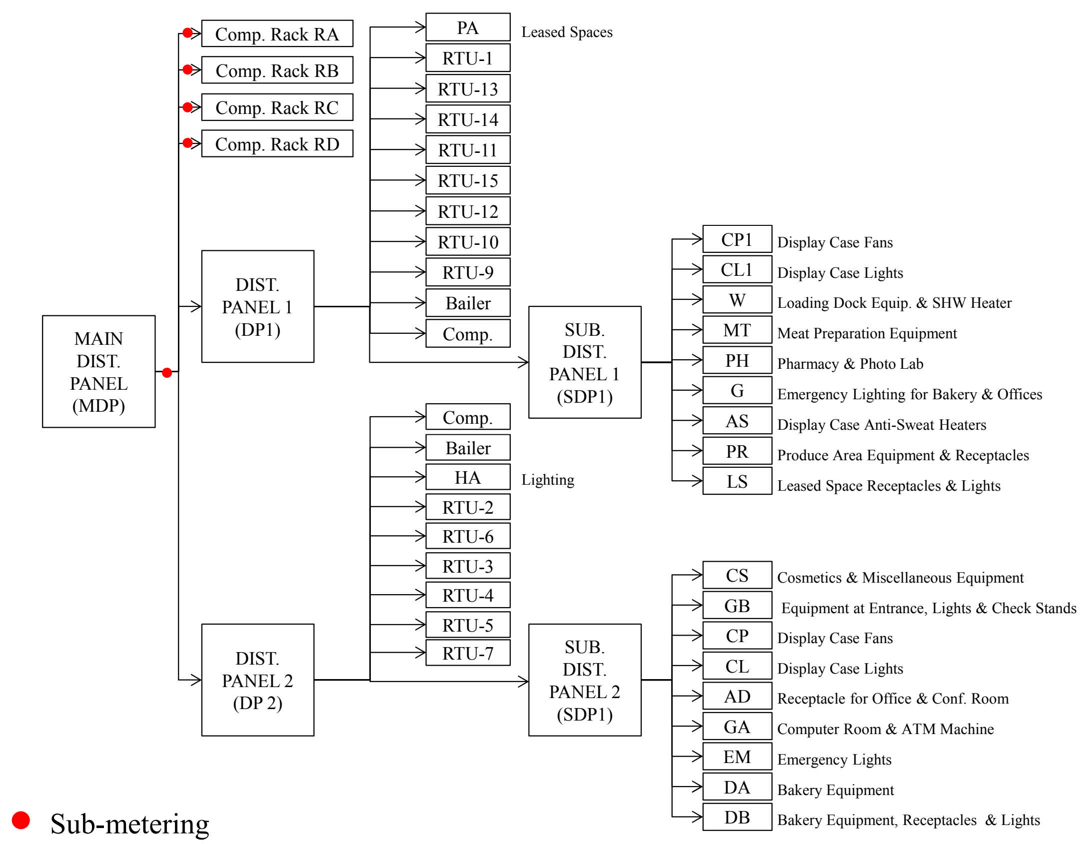

Figure 5-14: Schematic Electric Circuit Diagram of the Case-Study Store with Sub-Metering

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[36] The information from this month is not shown in this study.
[37] On discussing this issue with the store management, it was concluded that there was a possibility for the timer for daylight settings in the on-site monitoring system to not be set correctly.

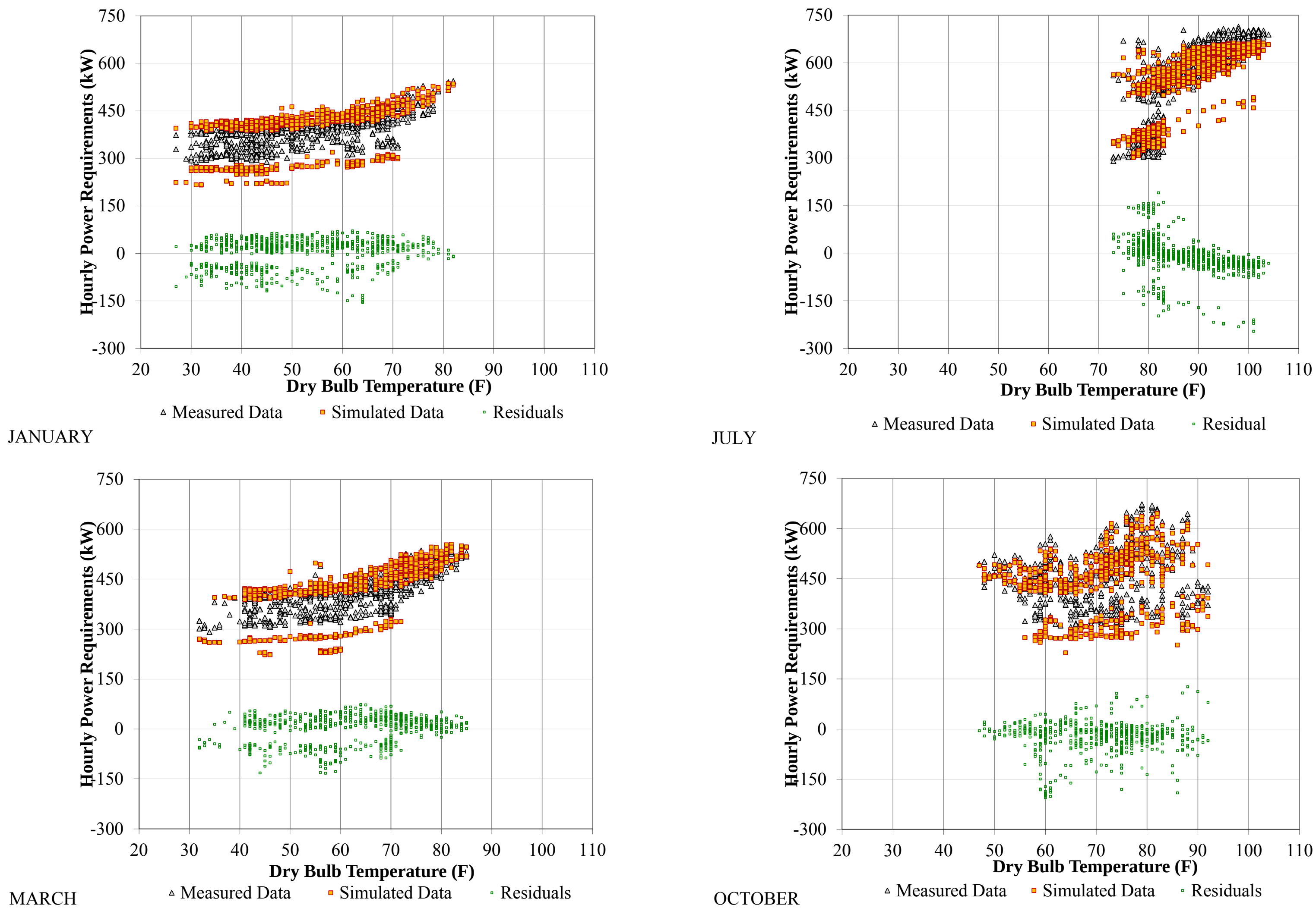


Figure 5-15: Results of the Scatter Plots for the Initial Run and Measured Data for Whole-Building Electricity Consumption for January, July, October and March 2009 versus Ambient Temperature

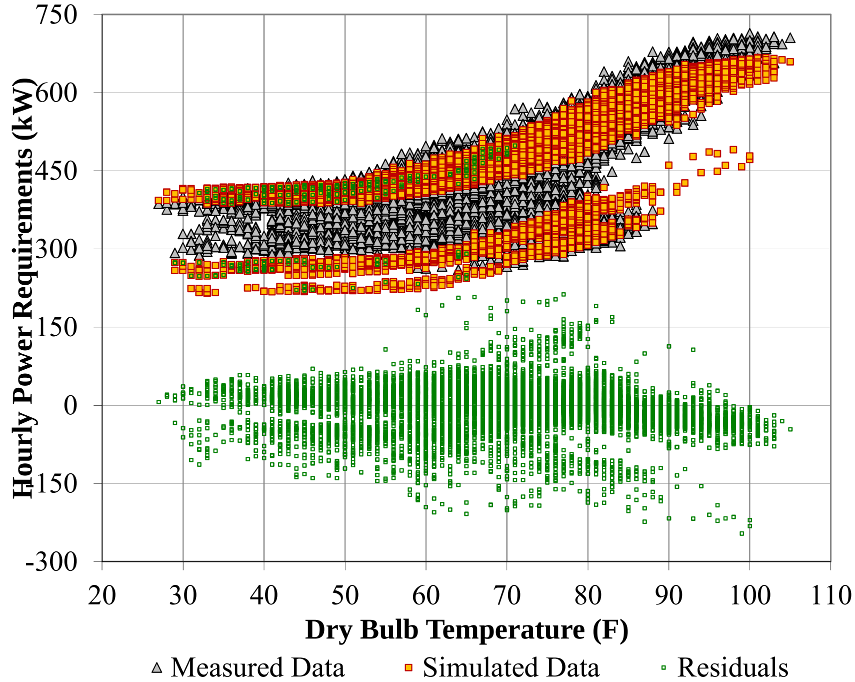


Figure 5-16: Comparing Whole-Building Electric Requirements of the Initial Run and Measured Data with Dry Bulb Temperature between 1st January to 20th December 2009

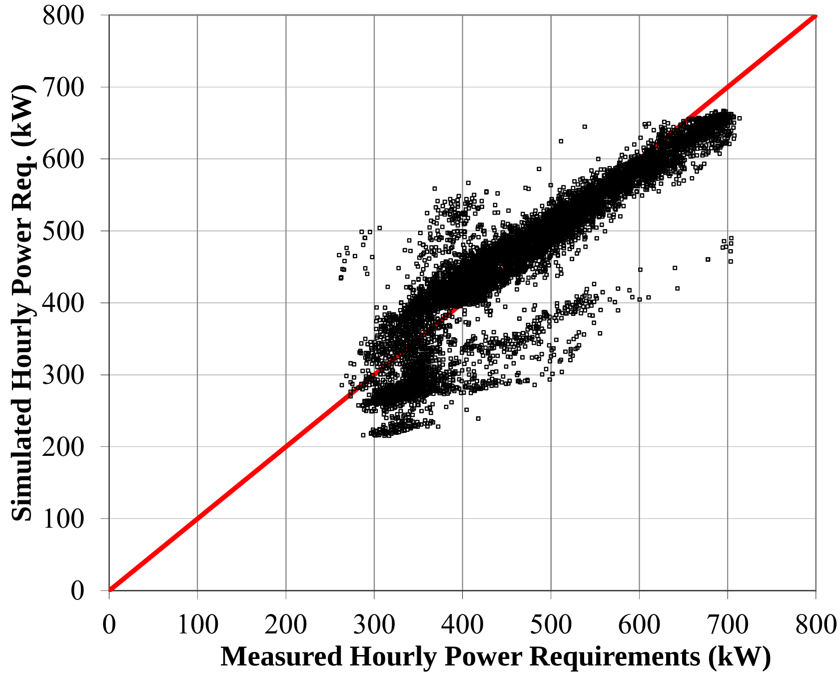
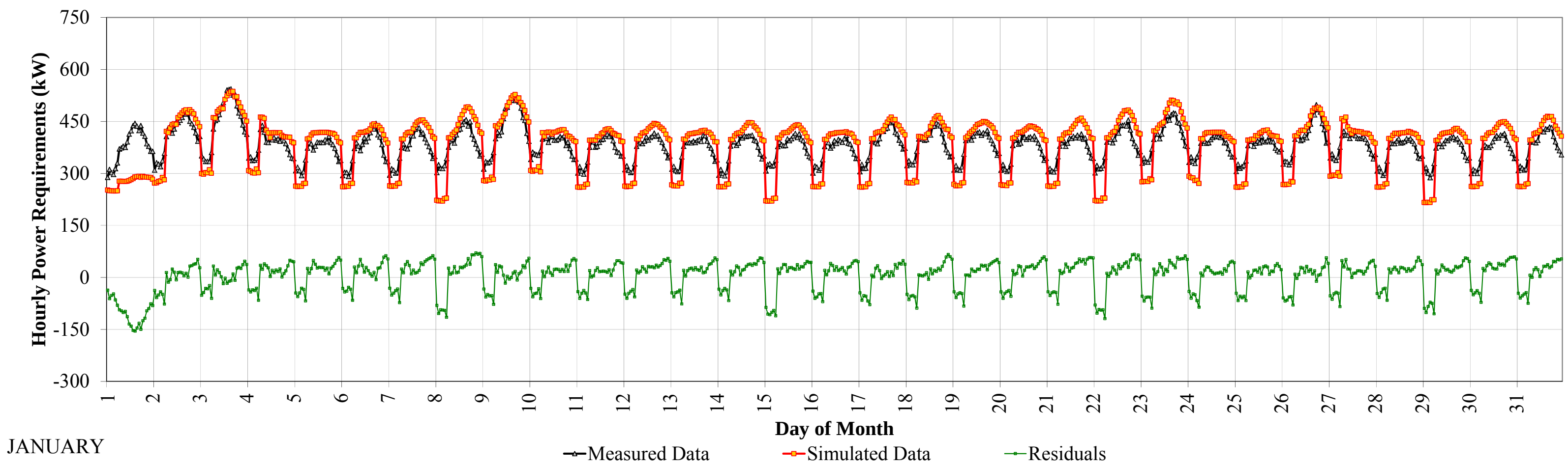


Figure 5-17: Comparison of the Simulated with Measured Whole-Building Electricity Consumption between 1st January to 20th December 2009



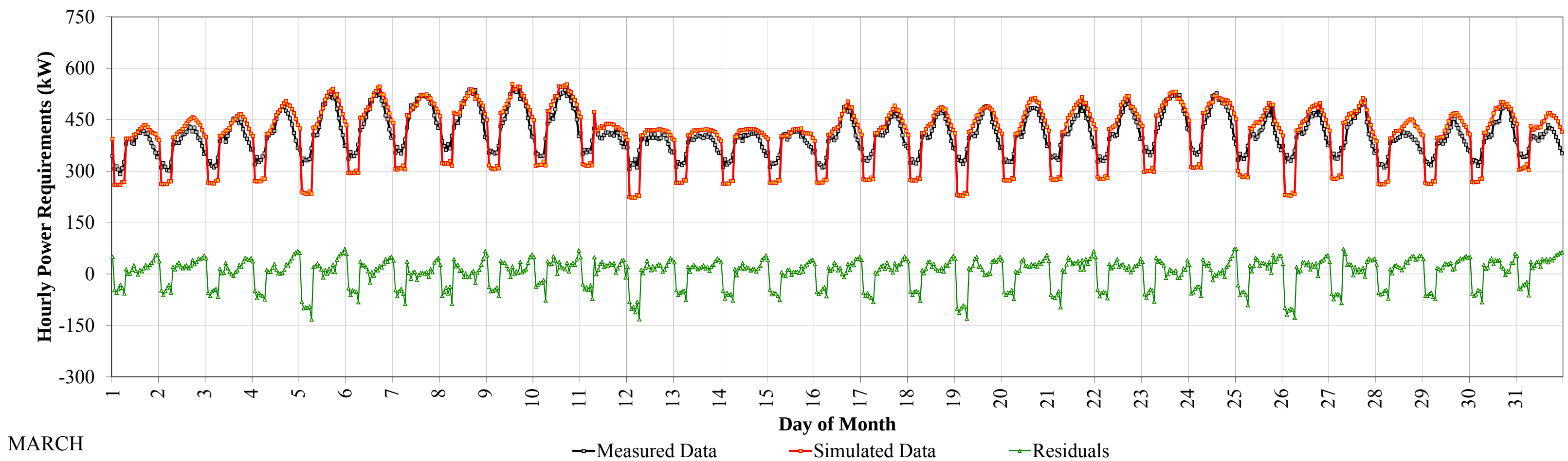


Figure 5-18: Time Series Plots of Whole-Building Electricity Consumption for the Initial Run for Months of January and March 2009

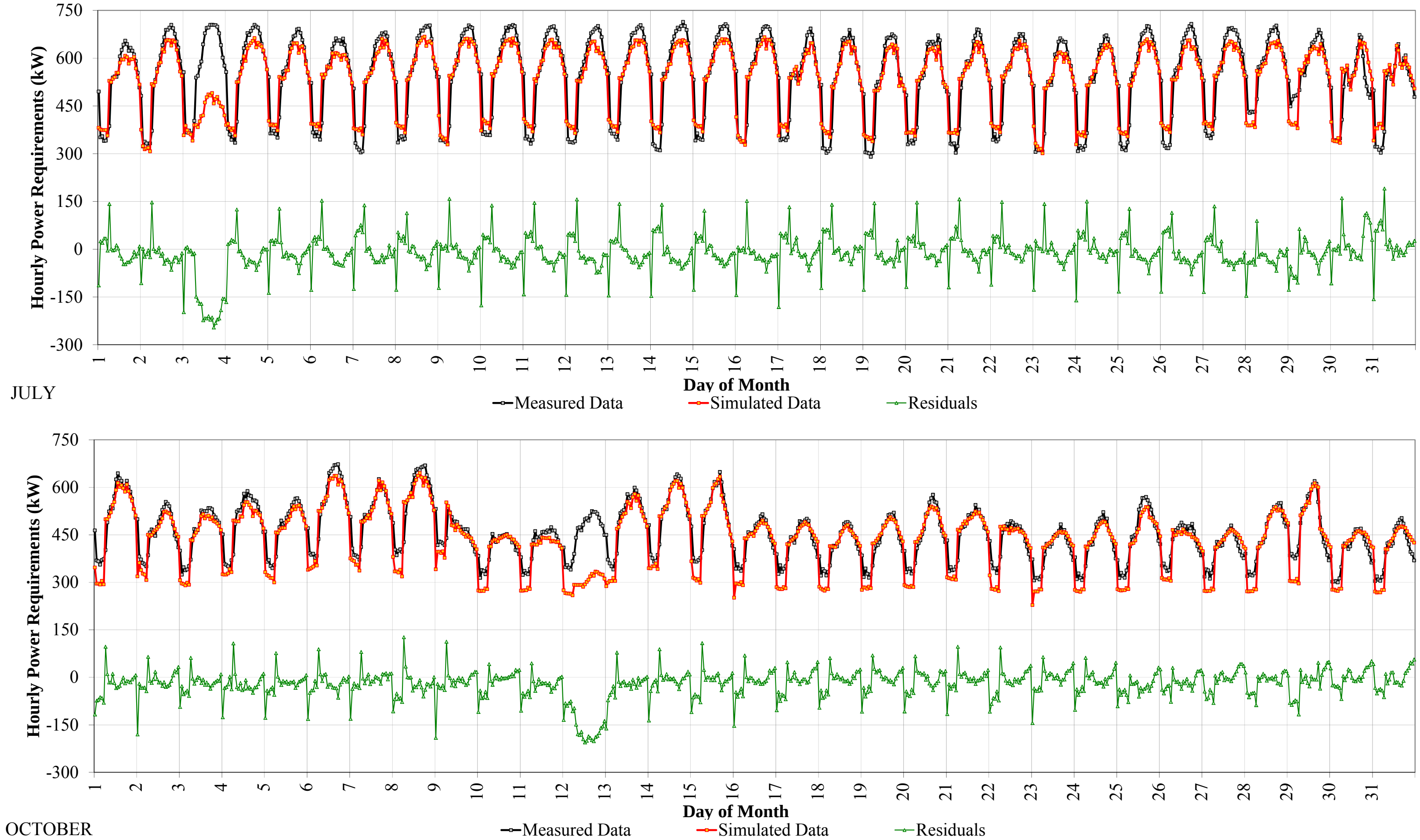
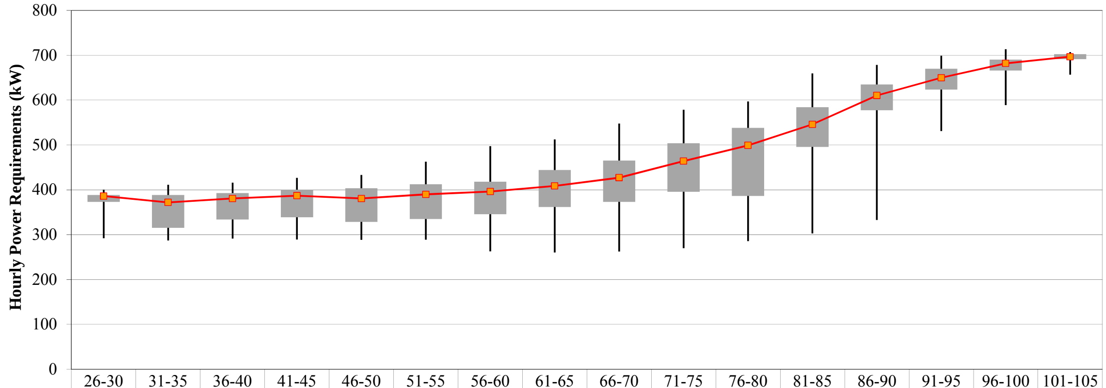
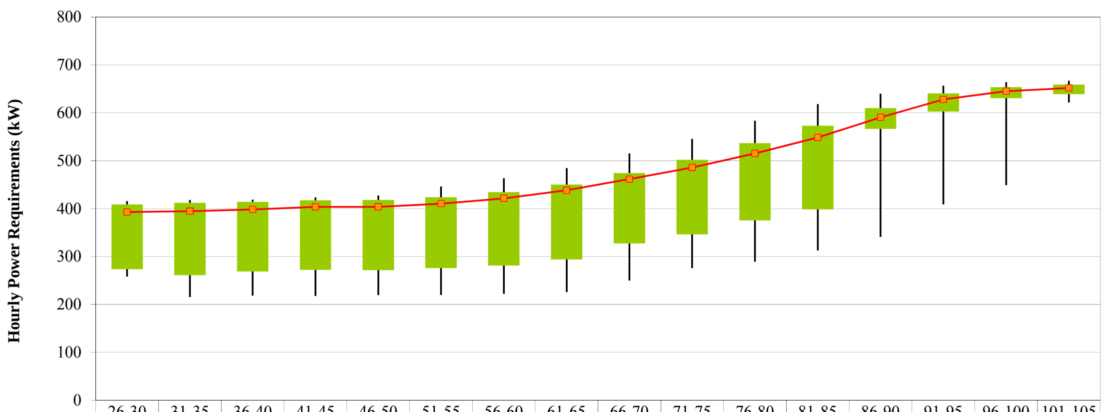


Figure 5-19: Time Series Plots of Whole-Building Electricity Consumption for the Initial Run for Months of July and October 2009



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
25%	373.3	315.2	333.8	338.9	328.6	334.9	345.6	361.8	372.9	395.7	386.1	495.7	577.4	623.4	665.8	691.4
Min	291.9	287.3	291.1	289.3	288.6	289.0	262.7	260.3	262.2	269.9	285.8	302.7	332.8	531.1	588.9	656.7
Max	400.4	411.3	416.0	426.5	433.3	463.1	497.6	512.4	547.9	578.6	597.0	659.7	678.3	698.9	713.6	707.2
75%	388.5	388.6	393.0	400.2	403.8	412.4	418.2	444.3	465.3	503.9	538.0	584.2	635.0	670.3	690.3	702.3
Median	386.1	372.1	380.6	387.0	380.6	389.9	396.3	408.8	427.1	464.0	499.2	546.0	610.4	649.9	681.9	696.7

Dry Bulb Temperature Bins (F)



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
25%	273.5	261.1	268.9	272.3	271.4	276.0	281.1	293.8	327.3	346.5	375.3	398.3	566.4	602.4	630.5	638.9
Min	258.3	215.2	218.2	217.6	219.5	219.8	222.2	225.7	249.9	275.7	289.1	312.7	340.9	408.8	448.6	621.3
Max	415.6	418.0	419.0	423.2	427.4	446.2	463.4	484.5	515.5	545.8	583.6	618.1	640.0	656.8	663.9	666.9
75%	408.8	412.3	414.0	417.3	418.2	423.8	434.7	450.3	474.0	501.8	536.6	573.1	610.0	640.5	653.8	658.9
Median	392.9	394.7	398.3	403.5	403.7	410.2	421.5	438.2	461.6	486.0	515.2	548.5	590.5	627.8	645.0	651.7

Dry Bulb Temperature Bins (F)

Figure 5-20: Results of Annual Bin Analysis of Measured Whole-Building Electric Power Data and Corresponding Simulated Data for the Initial Base-Case Run between 1st January to 20th December 2009

5.5.3.2.2 Observations from the 1st Set of Iterations

In this set of iterations several simulation inputs were re-examined with information based on inputs from the grocery store management, references from the literature review and reassessing the soundness of the assumptions made in the initial base-case run. Since the first set of iterations involved correcting the simulation inputs, the inputs were varied and changed in a cumulative fashion. For the initial base-case an RSME of 47.71, CV(RMSE) of 0.10 and MBE of -6.41 were recorded. The results of the first set of iterations are provided in Table 5-7.

Run 1 – 2: General Conditions

In the first run the base-case file was corrected for daylight savings. Adjusting for daylight savings reduced the statistical indices to an RMSE of 42.27, CV(RMSE) of 0.09 and an MBE of -6.39. In the second run the holiday schedule defaulted in the eQUEST-Refrigeration input file was modified to match the holiday schedule of the case-study store[38]. The initial and the modified holiday schedule of the store are presented in Table B-21 of Appendix-B. Adjusting the holiday schedule reduced the statistical indices to an RMSE of 37.39 and an MBE of -4.31. The results are provided in Table 5-7 below.

Run 3 – 6: Building Envelope

A sensitivity analysis was conducted varying each of the envelope components in the store. On varying several entries for the building envelope in the simulation model it was observed that changes to the building envelope did not significantly impact the energy consumption patterns of the grocery store[39]. Hence, only four building envelope parameters were selected which include changing the dimension of skylights; visible transmittance of skylights; infiltration values; and infiltration schedule. These components were selected because of their potential impact on the performance of other systems such as lighting and HVAC systems.

For Run 3 the area of the skylights was increased from 5.19 x 6.30 ft^2 to 5.70 x 6.61 ft^2 in order to correctly match the specifications of the store[40]. Increasing the area of skylights reduced the RMSE to 37.34, CV(RMSE) to 0.08 and the MBE to -3.58. For Run 4 the visible transmittance was raised from 50% to 60% to match the specifications provided in the

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[38] The case-study store was only closed to two days in 2009 – Easter and Christmas. The store resumed operating regular operating hours during all other days of the year.

[39] This is due to the fact that grocery stores are process dominated buildings where the major component of electricity consumption is from refrigeration and HVAC systems.

[40] It was noted that the specifications of skylights was incorrectly input in the initial base-case model.

construction drawings of the store. No changes were observed in the resultant statistical indices for this run. Changing the visible transmittance does not affect the thermal model but will be used when considering daylighting control options as efficiency measures for the store. For Run 5 the infiltration rates were changed from 0.161 ACH to 0.268 ACH. The value of 0.268 ACH was considered from the report on 50% reduction in energy consumption for grocery stores (Leach et al. 2009). A set of twenty one runs varying from 0.161 ACH to 0.268 ACH were executed to find the best fit for the reported statistical indices. The selected value of 0.268 ACH provided the lowest values for the statistical indices used in this analysis. Infiltration values for preparation rooms, coolers and freezers were also varied and it was determined that the infiltration assumed in the initial base case (i.e. 0.07 CFM/ft^2) was suitable for all the low temperature spaces in the store. Increasing the infiltration in the base-case model reduced the RMSE to 37.05, and the MBE to -1.12. In Run 6 the infiltration schedule was changed from a constant value assumed in the initial base-case for all the zones (Hale et al. 2008) to a schedule that increase the infiltration in the loading docks at selected time of day to reflect the unloading process, which includes opening the service doors for the entry of goods being unloaded. Infiltration schedules for the loading docks were obtained from the log maintained by the store management. The updated infiltration schedules can be found in Table B-19. Changing the infiltration schedule reduced the RMSE to 37.04, and the MBE to -1.11. The results are provided in Table 5-7 below.

Run 7 – 12: Space Conditions

In Run 7 the number of people in the "General Merchandise" zone and "Display Case" zone were varied. Six runs were conducted varying the number of people from 40 ft^2 / person to 140 ft^2 / person in order to obtain the best fit for the statistical indices. The number of people in service spaces such as the bakery, preparation room, loading docks, cooler and freezer were established after discussions with the store management. Hence, the number of people for these spaces was not changed. Changing the number of people in the "General Merchandise" zone and "Display Case" zone from the original number of 100 ft^2 / person to 80 ft^2 / person reduced the RMSE to 36.93 and the MBE to -0.28. In Run 8 the schedule of occupants was adjusted[41].

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[41] Based on discussions with the store management, the occupancy schedules in the main areas ("General Merchandise" and "Display Case" zones) change from month-to-month, season-to-season as well as day-to-day. Unfortunately, it is extremely difficult to capture these variations in the occupancy schedule implemented in the model. Hence, it was decided to go with numbers published by reliable sources such as Leach at al. (2009) and ASHRAE Standard 90.1-1989.

Values from Leach at al. (2009) and ASHRAE Standard 90.1-1989 were used. Although the implementation of the adjusted schedule increased the statistical indices of RMSE to 37.31 and MBE to 1.19, it was decided to go ahead with the new schedules. The new schedules are presented in Table B-16.

In Run 9 the lighting power density (LPD) was modified for all the spaces. However, after varying the LPD from 0.9 W/ft^2 to 2.4 W/ft^2 (six runs), it was found that the original settings of 1.8 W/ft^2 provided the best values for the statistical indices. In Run 10, the lighting power schedule was modified based on discussions with the store management. In the initial run 75% of all the lamps in the store were switched off during unoccupied hours. Whereas in the updated schedule 50% of all the lamps in the store were switched off during unoccupied hours. Changing the lighting schedule reduced the RMSE to 34.98 and increased the MBE to 11.93.

In Run 11 the exterior lighting power was added to the simulation model. Exterior lighting power was not considered in the initial base-case model. Four runs were conducted to establish appropriate exterior lighting power wattage. Lighting power wattage of 2 kW which was calculated from specifications provided in the drawings of the case-study store was used. These lights were set to operate for an 11 hour time period which takes into account the length-of-day for the entire year[42]. The introduction of increased the RMSE to 35.61 and reduced the MBE to 13.60.

In Run 12 the equipment power density was varied from 0.5 W/ft^2 to 2.3 W/ft^2 for all zones except the bakery zone. A value of 0.5 W/ft^2 provided the lowest values for the statistical indices observed and hence selected. For the bakery zone the equipment power was input in terms of power (kW). The power was varied from 3 kW to 10 kW in a series of seven runs. An equipment power of 3 kW provided the lowest values for the statistical indices observed and hence selected. Different equipment power densities were specified for the different zones of the store. No information could be obtained regarding the equipment power schedule. Hence it was decided to keep the original schedule which is referenced from Deru et al. (2011) and Hale et al. (2008). The introduction of decreased the RMSE to 34.37 and reduced the MBE to 8.93. The results are provided in Table 5-7 below.

[42] Based on the discussions with the store management it was noted that energy consumption from the lights in the surrounding parking lots were not monitored by the on-site monitoring system in the case-study store. Hence, energy consumption from parking lights were not considered by this analysis.

Table 5-7: Description of Calibration Runs for Building Envelope and Space Conditions and Corresponding Statistical Indices

Run No.	Description	Electricity (kW)			No. of Iterations per Run
		RMSE	CV(RMSE)	MBE	
0	**BASE-CASE**	47.71	0.10	-6.41	
1	**Corrected for daylight savings** Removed the mismatch between measured data and simulation.	42.27	0.09	-6.39	*
2	**Adjusting the holiday schedules to match the case-study store operation.** Changed from defaulted schedule in eQUEST-Refg. input file to match holiday schedule of the case-study store. Refer to Table B-21, Appendix B for updated holiday schedules.	37.39	0.09	-4.31	*
3	**Changed area of skylight** Changed skylight area from 5.19 x 6.30 ft^2 to 5.70 x 6.61 ft^2 to match specs. of case-study store.	37.34	0.08	-3.58	*
4	**Changed visible transmittance (VT) of skylights** Changed VT of skylights from 50% to 60%.	37.34	0.08	-3.58	*
5	**Changed infiltration values for normal temperature spaces** Changed infiltration values for normal temperature spaces from 0.16 ACH to 0.268 ACH.	37.05	0.08	-1.12	21
6	**Changed infiltration schedules** Changed infiltration schedules from Hale et al. (2008) to match Leach et al. (2009).	37.04	0.08	-1.11	4
7	**Changed the number of people in the general area and display case area** Changed the number of people in the general area and display case area from 100 ft^2 per person to 80 ft^2 per person.	36.93	0.08	-0.28	6
8	**Changed occupancy schedules** Changed occupancy schedules in "General Merchandise" and "Display Case" areas from Hale et al. (2008) to match schedules Leach et al. (2009) and ASHRAE Standard 90.1-1989.	37.31	0.08	1.19	*
9	**Lighting power density (LPD)** Retained LPD for all zones at 1.8 W/ft^2.	37.31	0.08	1.19	6
10	**Changed lighting power schedules** Changed lighting power schedules for all zones from 75% switch off during unoccupied hours to 50% switched off during unoccupied hours to match the schedules in the case-study store.	34.98	0.08	11.93	*
11	**Exterior lighing** Introduced exterior lighing of 2kW operating on a schedule during night time to match the specifications in the case-study store.	35.61	0.08	13.60	4
12	**Reduced equipment power density** Reduced equipment power density for all spaces except "Bakery" from specifications in Table B-5 to 0.5 W/ ft^2 for all zones.	34.37	0.08	8.93	7

Run 13- 22: HVAC Systems

The system sizing numbers were adjusted in Run 13 after a discussion with the store management regarding the sizing of HVAC systems. In the initial base-case model the HVAC systems were oversized by 1.2 times as defaulted in the eQUEST-Refrigeration program. The sizing ratio was removed in this run. Changing the sizing ratio increased the RMSE to 34.57 and reduced the MBE to 5.88.

In Run 14 the pressure drop across the supply fans for the packaged single zone systems is reduced from of 1.25 in.WG as defaulted in eQUEST-Refrigeration program to an arbitrary selection of 0.8 in.WG[43]. Changing the pressure drop across the HVAC system reduced the RMSE to 34.04, reduced the CV(RMSE) to 0.07 and reduced the MBE to 0.31.

In Run 15 the fan efficiency was changed from 53% to 56%. No specifications were provided for the case study store. Hence, three runs were conducted to determine the best fit. Changing the fan efficiency across the HVAC system reduced the RMSE to 34.03, and reduced the MBE to -0.22.

In Run 16 the EIRs for individual systems were changed from values defaulted in the initial base-case run to the values provided in the specifications of the base-case store and appropriately modified for the base-case simulation model[44]. The EER specifications for the simulation model are presented in Table B-6. Changing the EERs to match the specifications presented in the case-study store reduced the RMSE to 34.01, and reduced the MBE to 0.01.

In Run 17 design flow rates were changed from 0.5 CFM/ft^2 to reflect the supply air flow rates based on the specifications provided in the case-study store. Although the implementation of the specified design flow rates increased the statistical indices of RMSE to 35.06, CV(RMSE) to 0.08 and MBE to 12.72, it was decided to go ahead with the new schedules. The specifications for design flow rates which are implemented in this run are provided in Table B-6.

In Run 18 the outside air intake was changed from the values specified in the initial base-case model that were calculated using ASHRAE Standard 62.1 to better match the specifications provided for the case-study building. Changing the outside air intake specifications to match the specifications presented in the case-study store reduced the RMSE to

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[43] This value was later changed to a more realistic values in the subsequent runs.
[44] The EERs and the heating efficiencies for the consolidated thermal zones in the base-case simulation model were determined from the original equipment specifications by performing weighted average calculations.

34.33, and reduced the MBE to 10.19. The specifications for outside air intake rates, which have been adopted from the case-study store and implemented in this run, are provided in Table B-6.

In Run 19 the exhaust fan rates were changed to match the rates provided in the specifications for the case-study building. The value for the kitchen exhaust airflow in the “Bakery” zone was changed from 5,600 CFM to 5,561 CFM to match the specifications in the case-study store. Additional 1,200 CFM of exhaust air were added to the “General Merchandise” zone to reflect the operation of restroom fans. The exhaust air values have been adopted from the specifications in the case-study store. Changing the exhaust air specifications to match those presented in the case-study store reduced the RMSE to 34.29, reduced the CV(RMSE) to 0.07 and reduced the MBE to 10.19.

In Run 20 the zone temperatures for space heating were varied to provide the best fit as indicated by the statistical indices. It was observed that the setting for zone temperature during the heating season was different from what was recorded as set point temperatures by the on-site monitoring system. The heating set point temperature was changed from the 71°F to 70°F. Changing the space heating temperatures by one degree reduced the RMSE to 33.61, but increased the MBE to 19.45.

In Run 21 the process loads modeled in the freezers, coolers and preparation rooms[45] were removed, which resulted in lowering of the statistical indices. In addition, the design air flow rate was changed from 0.5 CFM/ft^2 as initially set in the simulation model to 1.0 CFM/ft^2. This change also resulted in an improvement to the reported statistical indices. Changing the specifications to match those presented in the case-study store reduced the RMSE to 33.40 and reduced the MBE to 2.24.

Run 22 involved changing the furnace efficiency from the values defaulted in the eQUEST-Refrigeration program to the values specified in the case study store. Since the changes impacted the energy consumption of furnaces which operate using natural-gas, no change was observed in the statistical indices tracking electricity consumption in the store. The HIR values implemented in the final base-case model are presented in Table B-6. The results are provided in Table 5-8 below.

[45] Processes loads account for machinery such as forklifts operating in these spaces (i.e. that contribute to the space heating but not as plug loads).

Run 23- 24: Service Hot Water Heaters

In Run 23 the process loads were changed to 1.5 GPM and 0.15 GPM for the two gas heaters modeled to reflect the specifications from the store. This change did not impact the statistical indices. In Run 24 two electric water heaters were added to the model based on the specifications in the case-study store. Specifications for the service hot water heaters are provided in Table B-8. This addition increased the RMSE to 36.75 and MBE to -14.98. These results are documented in Table 5-8 below.

Run 25- 32: Refrigeration Systems

In Run 25 the throttling range for condensing temperature was changed from a default value of 10°F to the temperature ranges specified in the specifications for the case-study store[46]. This change in the throttling range decreased the RMSE to 34.98 and MBE to -12.24.

In Run 26 the ratio of the electric input of the condenser fan to nominal capacity of the condenser fan, which is expressed as a ratio[47] in the simulation model, was changed from a default value of 0.55 to values calculated from the specifications provided for the case-study store. The values from the case-study store used in the calculation of the condenser fan power ratio of the condensers simulated in this run are presented in Table B-15. This change in the ratio of the electric input of condenser fans decreased the RMSE to 34.48 and MBE to -10.74.

In Run 27 the capacity of condensers was input into the model as per specifications provided for the case-study store. Prior to this run, condenser capacities defaulted by eQUEST-Refrigeration program were used. Condenser specifications for the case study store that have been used in the simulation model are presented in Table B-15. This change in condenser capacity decreased the RMSE to 33.75, decreased CV(RMSE) to 0.07 and decreased MBE to -5.14.

[46] For the base-case model this throttling range defines the points at which the fans cycle.
[47] Condenser Fan Power Ratio = Fan Electric Power (Btu/h)/ Temperature Difference (F)

In Run 28 the display case lighting was reduced from 50% of the total output being turned off during unoccupied hours to 75% of the total output being turned off during unoccupied hours. This change in the schedule of display case lighting decreased the RMSE to 33.51 and decreased MBE to -6.77.

In Run 29 the defrost schedules were changed to match the information provided from the case-study store. This change in the defrost schedules decreased the RMSE to 33.22 and increased the MBE to -6.89.

In Run 30 night covers over open refrigerated display cases were simulated in the model reflecting the practice currently implemented at the case study store[48]. The installation of night covers in the base-case simulation model decreased the RMSE to 33.2 and decreased the MBE to 6.09. In Run 31 a number of compressors that were found to be missing in the model were added to the base-case simulation model. The installation of compressors increased the RMSE to 33.36 and changed the MBE to -4.56. In Run 32, after confirming with the store management, a pulsating control was added to operation of anti-sweat heaters[49]. The installation of pulsating controls for anti-sweat heaters increased the RMSE to 33.41 and changed the MBE to -4.86. Finally, in Run 33 corrections were made to the installation of EPR and EEPR controls in display cases. The modification of EPR and EEPR controls increased the RMSE to 33.44 and changed the MBE to -4.60. The results are provided in Table 5-9 below.

[48] The night covers were pulled down over the open refrigerated display cases during unoccupied hours and retracted during hours when the case-study store was open to public.

[49] In the case-study store the anti-sweat heaters are controlled by pulsing the power flow through the heaters to reduce energy consumption. This can't be modeled in eQUEST. Instead, a lower power rating, which integrates the average energy use, was used to power the anti-sweat heaters.

Table 5-8: Description of Calibration Runs for HVAC and SHW Systems, and Corresponding Statistical Indices

Run No.	Description	Electricity (kW)			No. of Iterations per Run
		RMSE	CV(RMSE)	MBE	
0	**BASE-CASE**	47.71	0.10	-6.41	
12	**CALIBRATING BUILDING ENVELOPE AND SPACE CONDITIONS**	34.57	0.08	8.93	
13	**Changed System sizing ratio** Changed System sizing ratio from 1.2 to 1.	34.57	0.08	5.88	*
14	**Changed static pressure of supply fans** Changed static pressure of supply fans from 1.25 in. WG. to 0.8 in. WG.	34.04	0.07	0.31	*
15	**Changed supply fan efficiency** Changed supply fan efficiency from 53% to 56%.	34.03	0.07	-0.22	3
16	**Changed EER of packaged units** Changed EER of packaged units from default values in eQUEST-Refg. to match specifications provided in case-study store (Table B-6).	34.01	0.07	0.01	*
17	**Changed zone design flow rates** Changed zone design flow rates from 0.5 CFM/ft^2 to match specifications provided in case-study store (Table B-6).	35.06	0.08	12.72	*
18	**Changed outside air quantities** Changed outside air quantities from quantities calculated using ASHRAE 62.1 to match specifications provided in the case-study store (Table B-6).	34.33	0.08	10.19	*
19	**Changed exhaust fan specifications** Changed exhaust fan specifications from 5,600 CFM to 5,561 CFM in the "Bakery" zone. Added 1,200 CFM in the "General Merchandise" to match specifications from the case-study store.	34.29	0.07	9.97	*
20	**Changed Design Heating Temperature** Changed Design Heating Temperature from 71°F to 70° F	33.61	0.07	19.45	40
21	**For freezers, coolers and preparation room** Removed the extra process loads. Changed design air flow rate from 0.5 CFM/ft^2 to 1 CFM/ft^2.	33.40	0.07	2.24	3
22	**Changed furnace efficiency** Changed furnace efficiency from defaulted values to match specifications of the case-study store (Table B-6).	33.40	0.07	2.24	*
23	**Changed process flow rates for gas-fired water heaters** Changed process flow rates for gas SWH to 1.5 gpm and 0.15 gpm to match specs	33.89	0.07	-5.36	8
24	**Adding electric water heaters** Added two electric water heaters as per specifications	36.75	0.08	-14.98	*

Table 5-9: Description of Calibration Runs for Refrigeration Systems and Corresponding Statistical Indices

Run No.	Description	Electricity (kW)			No. of Iterations per Run
		RMSE	CV(RMSE)	MBE	
0	**BASE-CASE**	**47.71**	**0.10**	**-6.41**	
12	**CALIBRATING BUILDING ENVELOPE AND SPACE CONDITIONS**	**34.57**	**0.08**	**8.93**	
24	**CALIBRATING BUILDING HVAC AND SHW SYSTEMS**	**36.75**	**0.08**	**-14.98**	
25	**Changed SCT throttling range** Changed SCT throttling range from 10 F to match specifications in case-study store (Table B-15).	34.98	0.08	-12.24	*
26	**Changed fan-EIR-TD of condensers** Changed fan-EIR-TD of condensers from 0.55 to match specifications in case-study store (Table B-15).	34.48	0.08	-10.70	*
27	**Changed condenser capacities** Changed condenser capacities to match specifications in the case-study store (Table B-15)	33.75	0.07	-5.14	*
28	**Changed the display case lighting schedule** Changed the display case lighting schedule from 50% turned off to 75% turned off during unoccupied hours to match specifications in the case-study store.	33.51	0.07	-6.77	*
29	**Changed defrost schedules** Changed defrost schedules to as per observation from measured data	33.39	0.07	-6.89	*
30	**Installed night covers**	33.22	0.07	6.09	*
31	**Added compressors that were found missing**	33.36	0.07	-4.56	*
32	**Controls for Anti-sweat heaters** Added pulsating control for antisweat heaters	33.41	0.07	-4.86	*
33	**Corrected certain display cases by installing EPR controls**	33.44	0.07	-4.60	*

5.5.3.2.3 Observations from the 2nd Set of Iterations

In the second set of iterations several of the inputs to the building simulation model were revisited and fine-tuned. In general, changes were made to the model as improved information became available and incorrect assumptions regarding inputs to the simulation model were corrected.

Run 34- 51

In Run 34 the exterior lights schedule was found to be incorrect and was corrected to reflect the 11 hour schedule on which the exterior lights operated. Correcting the exterior lighting schedules increased the RMSE to 33.89 and changed the MBE to -5.36.

In Run 35 the lighting power density was lowered from 1.8 W/ft^2 to 1.6 W/ft^2 in the "General Merchandise" and "Display Case" zones based on input from the store management, and the lighting schedule was changed from 0.95 to 1 during the occupied times of the day. The updated lighting power density is presented in Table B-4. Updated lighting power schedules are presented in Table B-17. Correcting the lighting power density and the corresponding schedules increased the RMSE to 36.75 and changed the MBE to -14.98.

In Run 36 the infiltration was changed from 0.27 ACH in all normal temperature spaces to 0.5 ACH in the "General Merchandise", "Display Case" and "Bakery" zones; to 0.8 ACH in the "Loading Dock Produce" zones and to 1 ACH in the "Loading Dock General" zones[50]. The values were based on observing the schedule of activities in these spaces. In addition, a commercial exhaust fan was installed in one of the leased spaces next to one of the entrance doors of the grocery store. The operation of the exhaust fan during operating hours of the grocery was assumed to increase the infiltration in the building by drawing in outside air through the entrance door. The changes decreased the RMSE to 34.98 and decreased the MBE to -12.24.

In Run 37 the outdoor air quantities were increased to 15% of the values provided in the construction drawings of the store. In Run 38 the supply air quantities for all zones were increased by 20%. It should be noted that although changing the outdoor air and supply air quantities in the simulation model decreased the values of the statistical indices for electric

[50] The product delivery schedules for the two loading docks in the case-study store were provided by store management. The schedules were used to develop infiltration for the loading docks modeled in the base-case model. During a delivery the doors of the loading zone were opened allowing a huge amount of outside air to infiltrate inside. This trend was also observed when looking at the trends in interior temperature recorded for these spaces. Interior temperatures in these spaces tended to float at several periods during the day.

consumption, this change had a detrimental impact on the calibration of natural gas usage in the store[51]. These changes were removed in the subsequent runs.

In Run 39 the fan static pressure was arbitrarily changed from 0.8 in.WG to 1.1 in.WG to account for both the internal and external static pressure drops across the supply air fans. Changing the fan static pressure decreased the RMSE to 33.04 and decreased the MBE to 0.13.

In Run 40 the equipment schedule for "General Merchandise" zone was corrected to match the equipment schedules of the other zones. The final values of the equipment power schedules are presented in Table B-18. Correcting the equipment schedule decreased the RMSE to 32.14 and changed to MBE to -2.96.

In Run 41 the fan power for exhaust fans in the "Bakery" zone was corrected from 0. 41 W/CFM to 0. 245 W/CFM as specified in the drawings for the case-study store. In addition, the static pressure across the exhaust fans were increased from 0.7 tin WC to 1.2 in WC to account for internal as well as external static pressure drop across the fan (Thornton et al. 2010). Correcting the specifications decreased the RMSE to 32.09 and changed to MBE to 10.12.

In Run 42 the roof surface properties of solar reflectance and thermal emittance were changed from 0.4 and 0.9 initially assumed by the simulation model to 0.23 and 0.87 respectively to provide a better representation of the case-study roof surface. The numbers were obtained from the defaulted values of the eQUEST-Refrigeration model and were thought to best represent the light colored roof of the case-study store. Correcting the specifications decreased the RMSE to 32.04 and changed to MBE to -1.04.

In Run 43 the space heating equipment was removed from the "Bakery" zone. In Run 44 the UA specifications for the gas water heaters in the store were recalculated using the equations provided in Section 5.4.6 of this chapter. Changes made in Run 43 and Run 44 proved to be beneficial to the calibration of natural gas usage in the simulation model.

In Run 45 the specifications for freezers, coolers and preparation rooms were changed to better match the specifications provided by the store management and standard practice. Changes included setting the evaporator fan controls to constant volume, recalculating the evaporator fan power with information from the construction drawings for the store and auto-sizing the supply flow air allowing the program to calculate design supply air flow rate based on the temperature difference in these zones. These changes decreased the RMSE to 32.01 and changed the MBE to 1.85. In Run 46 the infiltration in the preparation rooms, freezer rooms and

[51] Monthly values were compared for natural gas consumption usage.

coolers was assessed. A total of 17 iterations were performed for the assessment. The infiltration was retained at 0.05 CFM / ft^2 for preparation rooms and coolers. On the other hand infiltration was increased to 0.09 CFM / ft^2 for freezers. The specifications for the initial and final base-case are provided in Table B-22. These changes decreased the RMSE to 31.90 and changed the MBE to 0.22.

In Run 47 the equipment schedule was simplified to reflect days when the store is open and holidays. These changes decreased the RMSE to 31.34 and changed the MBE to 1.96. In Run 48 the values for supply air and outdoor air intake were returned back to the original values as specified in the construction drawings of the store. These changes decreased the RMSE to 31.34 and changed the MBE to 1.96.

In Run 49 the input for the cooling EIR was corrected to account for the fan energy. The EER values provided in Table B-6 are inclusive of indoor fan efficiency[52]. However, inputs for EER to the eQUEST-Refrigeration simulation model are considered in terms of Electric Input Ratio (EIR) and should exclude indoor fan efficiency to avoid being counted twice in the simulation. Hence, in order to adjust the EER value to exclude the indoor fan energy, the EIR is obtained by using the following equation (Thornton et al. 2010):

$$1/EIR = (EER\ /\ 3.413\ +\ R)/(1 - R)$$

Where R is the ratio of the supply fan power to total equipment power at rating condition and is assumed to be 0.12 (Thornton et al. 2010) for this analysis. Changing the EERs to match the specifications presented in the case-study store increased the RMSE to 34.71, and changed the MBE to -12.98.

In Run 50 the fan static pressure was changed from 1.1 to 1.53 as recommended in Thornton et al. (2010) to reflect the overall static pressure difference across the supply fan for a typical roof top unit. Changing the fan static pressure decreased the RMSE to 32.45 and changed the MBE to -2.84. In Run 51 the overall fan efficiencies were changed from 56% assumed in an earlier run to 60% with corresponding motor efficiencies of 85.5%. (Thornton et al. 2010). Changing the fan static pressure decreased the RMSE to 32.15 and changed the MBE to 0.84. The results are provided in Table 5-10 below. Trends in the RSME, CV(RMSE) and the MBE are provided in Figure 5-21 and Figure 5-22 Results from monthly as well as hourly final calibration plots are provided in Figures 5-23 through Figure 5-34.

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[52] Manufacturer ratings for packaged HVAC units are typically include indoor fan efficiency.

Table 5-10: Description of Calibration Runs for the 2nd Iteration and Corresponding Statistical Indices

Run No.	Description	Electricity (kW)			No. of Iterations per Run
		RMSE	CV(RMSE)	MBE	
0	**BASE-CASE**	**47.71**	**0.10**	**-6.41**	
12	**CALIBRATED BUILDING ENVELOPE AND SPACE CONDITIONS**	**34.57**	**0.08**	**8.93**	
24	**CALIBRATED BUILDING HVAC AND SHW SYSTEMS**	**36.75**	**0.08**	**-14.98**	*
33	**CALIBRATED REFRIGERATION SYSTEM**	**33.44**	**0.07**	**-4.60**	
34	**Corrected exterior lights schedule to reflect 11 hours of operation**	33.89	0.07	-5.36	*
35	**Lowered LPD and changed lighting schedules** Lowered LPD from 1.8 W/ft^2 to 1.6 W/ft^2 in "General Merchandise" and "Display Case" zones. Changed lighting schedule from 0.95 to 1 during occupied time of the day.	36.75	0.08	-14.98	*
36	**Changed infiltration** Changed infiltration from 0.27 ACH in all spaces, to 0.5 ACH in "General Merchandise", "Display Case" and "Bakery" zones, to 0.8 ACH in "Loading Dock Produce" zone, to 1 ACH in "Loading Dock General" zone.	34.98	0.08	-12.24	*
37	**Increased outdoor air intake** Increased outdoor air intake for all the zones by 20%.	34.48	0.08	-10.70	4
38	**Increased design flow rate** Increased design flow rate by 15%.	33.86	0.07	-7.71	4
39	**Changed fan static pressure** Changed fan static pressure from 0.8 in. WG to 1.1 in. WG.	33.04	0.07	0.13	*
40	**Corrected equipment schedule** Corrected equipment schedule for "General Merchandise" zone to match equipment schedules of other spaces.	32.14	0.07	-2.96	*
41	**Increased exhaust fan power** Increased exhaust fan power in "Bakery" zone from 0.041kW/1000 CFM to 0.245kW/1000 CFM. Increased static pressure from 0.7 in. WG to 1.2 in. WG. Increased total static pressure of exhaust fan in "General Merchandise" zone from 0.35 in. WG to 1.2 in. WG.	32.09	0.07	10.12	*

Table 5-10: Continued

Run No.	Description	Electricity (kW)			No. of Iterations per Run
		RMSE	CV(RMSE)	MBE	
0	**BASE-CASE**	**47.71**	**0.10**	**-6.41**	
12	**CALIBRATED BUILDING ENVELOPE AND SPACE CONDITIONS**	**34.57**	**0.08**	**8.93**	
24	**CALIBRATED BUILDING HVAC AND SHW SYSTEMS**	**36.75**	**0.08**	**-14.98**	*
33	**CALIBRATED REFRIGERATION SYSTEM**	**33.44**	**0.07**	**-4.60**	
42	**Changed roof surface properties** Changed roof surface properties of solar reflectance from 0.4 to 0.23 and thermal emittance from 0.9 to 0.87.	32.04	0.07	-1.04	*
43	**Removed heating equipment from "Bakery" zone**	32.04	0.07	-1.06	*
44	**Corrected UA specifications for gas water heater**	32.04	0.07	-1.22	*
45	**Changed freezer, cooler & preparation room fan specifications**	32.02	0.07	1.85	*
46	**Changed freezer, cooler & preparation room infiltration specifications** Retained infiltration in prep rooms and coolers to 0.07 CFM/ft^2 Increased infiltration in freezers to 0.09 CFM/ft^2.	31.90	0.07	0.22	17
47	**Simplified equipment schedules**	31.34	0.07	1.96	*
48	**Reverted to original supply air and outdoor air intake values**	31.22	0.07	-3.61	*
49	**Changed cooling EIR values** Changed cooling EIR values to reflect input without fan power.	34.71	0.08	-12.39	*
50	**Changed fan static pressure** Changed fan static pressure from 1.1 to 1.53 in. WG.	32.45	0.07	-2.84	*
51	**Changed fan efficiencies** Changed fan efficiencies from 56% to 60% with motor efficiency of 85.5% to match specifications provided by the case-study store.	32.15	0.07	0.84	6

5.6 Summary of Results and Conclusions

In this chapter a base-case simulation model of the grocery store was created using eQUEST-Refrigeration (version 3.61) simulation program. The initial base-case was modeled using information provided by a case-study store, which was situate in a hot and humid climate of central Texas. Several other references and assumptions were made in the development of the base-case model. Some assumptions include the defaults provided by the eQUEST-Refrigeration program.

The simulation model was calibrated against measured data for electricity and gas usage obtained from the grocery store. Both monthly and hourly calibrations were conducted. However, due to the unavailability of hourly natural gas consumption hourly calibrations were conducted for electricity usage only.

Several parameters were selected for the calibration procedure which includes parameters for the building envelope and space conditions, HVAC, service water systems and refrigeration systems. The impact of each of the selected parameter was assessed on a cumulative basis by examining the statistical indices RMSE, CV(RMSE) and MBE. In general, changes were made to the model as improved information became available and incorrect assumptions regarding inputs to the simulation model were corrected.

A set of 51 iterations was performed. The initial RMSE, CV(RMSE) and MBE values were 47.71, 0.10 and -6.41 respectively. The RMSE, CV(RMSE) and MBE values of the final run were established to be 32.15, 0.07 and 0.84 respectively.

The changes made to the initial model are recorded in Table 5-11. The final base-case that will be used to assess the energy efficiency measures in the next chapter includes information from Table 5-1 through Table 5-6 and the changes recorded in Table 5-11.

This study confirms the conclusions in Bou-Saada and Haberl (1995) regarding the calibration procedure. However, in this case the recommendations are specific to the calibration of a grocery store model. Information for improving the simulation model includes both on-site observations as well as independent measurements in the grocery store. The recommendations are listed below:

- A complete set of construction drawings, which includes architectural and mechanical drawings, needs to be available for the calibration process.
- Measured zone air temperature as well as supply and return air temperatures of HVAC systems have to be documented by independent measurements in the grocery store.

- An equipment and light fixture count as well as corresponding power consumption requirements needs to be performed to provide an accurate estimate of equipment and lighting Wattage in the grocery store.
- Various hourly schedules in the building such as those for equipment, lighting and occupancy have to be documented by on-site observations.
- Blower door tests for the whole building are recommended to check and confirm the infiltration rates in the building.
- Infiltration schedules should be determined by on-site observations of opening and closing of doors as well as operation of equipment such as HVAC systems and exhaust fans.
- Correct input of equipment performance specifications need to be performed. This includes both design performance specifications as well as performance specifications at part-load conditions.
- Defaulted values used in the simulation model should be checked for reasonableness. In addition, limitations of the simulation model should also be accounted for in the assessment of the calibrated model.
- Measurement and use of local weather data, which includes relative humidity, dry-bulb temperature, wind speed and global horizontal solar radiation should be used.
- Finally, a year's worth of measured data that includes whole-building electricity consumption, and sub-metered data that includes energy consumption from space cooling, lighting and equipment, refrigeration compressors and condensers, and various components of display cases should be made available in order to improve the calibration procedure.

Table 5-11: List of Changes Made to the Initial Base-Case Simulation Model

	No.	Description
BUILDING ENVELOPE & SPACE CONDITION	1	**Roof surface properties:** Solar reflectance - 0.23, Thermal emittance - 0.87
	2	**Skylight area:** 5.70 x 6.61 ft^2
	3	**Skylight visible transmittance:** 60%
	4	**Infiltration values:** 0.5 ACH in "General Merchandise", "Display Case" and "Bakery" zones 0.8 ACH in "Loading Dock Produce" zone 1 ACH in "Loading Dock General" zone 0.07 CFM/ft^2 for preparation room and cooler 0.09 CFM /ft^2 for freezer
	5	**Infiltration schedules:** Table B-19
	6	**Occupancy in "General Merchandise" and "Display Case" zones:** 80 ft^2/person
	7	**Occupancy schedules:** Table B-16
LIGHTING & EQUIPMENT	8	**Lighting power density "General Merchandise" and "Display Case" zones:** 1.6 W/ft^2
	9	**Lighting power schedules:** Table B-17
	10	**Exterior lighing power:** 2kW
	11	**Exterior lighting schedule:** From sunset to sunrise
	12	**Equipment power density for all spaces except "Bakery":** 0.5 W/ft^2
	13	**Equipment power schedules:** Table B-18
HVAC & SHW SYSTEMS	14	**EER of packaged units:** Table B-6
	15	**Furnace efficiency:** Table B-6
	16	**Design supply air flow rates:** Table B-6
	17	**Design outside air flow rates:** Table B-6
	18	**Supply fan total static pressure:** 1.53 in. WG
	19	**Supply fan overall efficiency:** 60% (motor eff. 85.5%)
	20	**Exhaust fan specifications:** Table B-7
	21	**Design heating temperature:** 70 F
	22	**For freezers, coolers and prep. rooms removed the extra process loads defaulted in the model.**
	23	**Gas-fired SWH specifications:** Table B-8
	24	**Electric water heater specifications:** Table B-8
REFRIGERATION SYSTEM	25	**SCT throttling range:** Table B-15
	26	**Condenser fan power:** Table B-15
	27	**Condenser capacity:** Table B-15
	28	**Display case lighting schedule:** Reduced to 75% of full power during unoccupied period
	29	**Defrost schedules:** Table B-13
	30	**Night covers over open display cases**
	31	**Anti-sweat heater power consumption:** Table B-14
	32	**Freezer, cooler & preparation room specifications:** Table B-12, Table B-13, Table B-14 and Table B-22

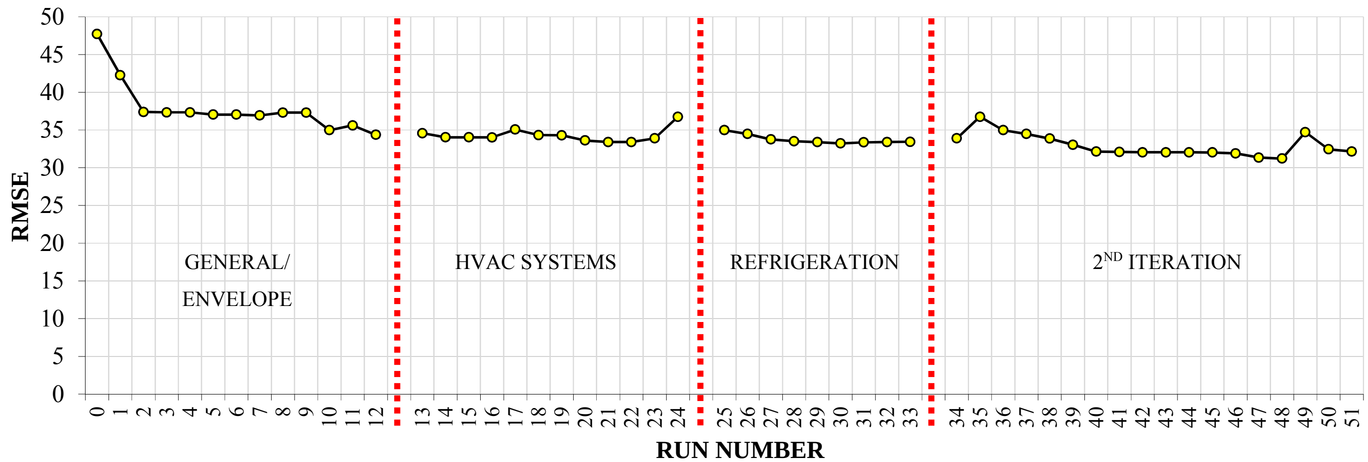


Figure 5-21: Trends in RMSE for Whole-Building Electricity Consumption with Change in Specifications for Parameters

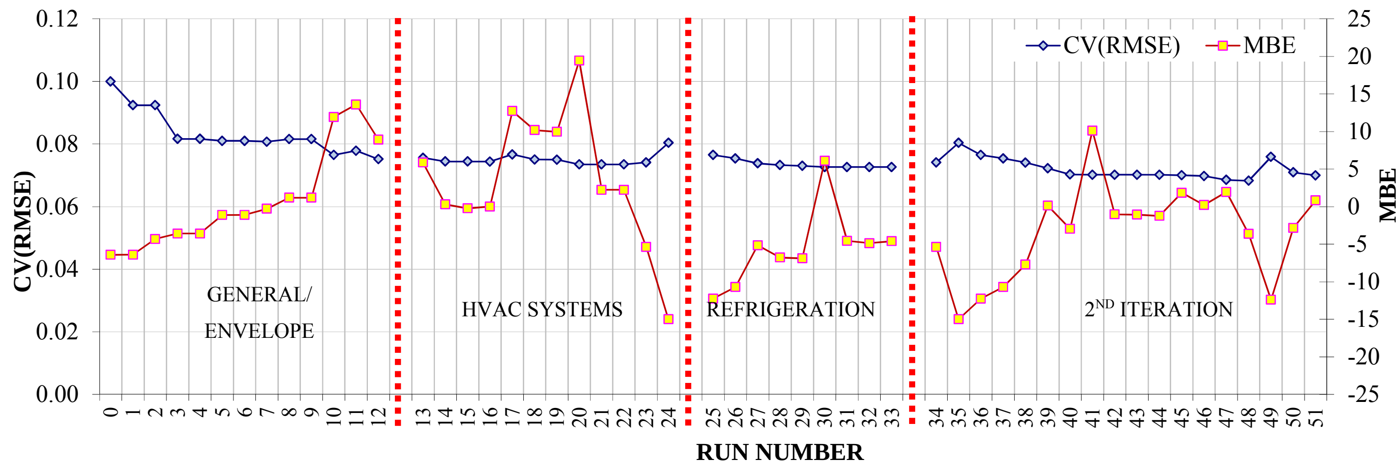


Figure 5-22: Trends in (CV)RMSE and MBE for Whole-Building Electricity Consumption with Change in Specifications for Parameters

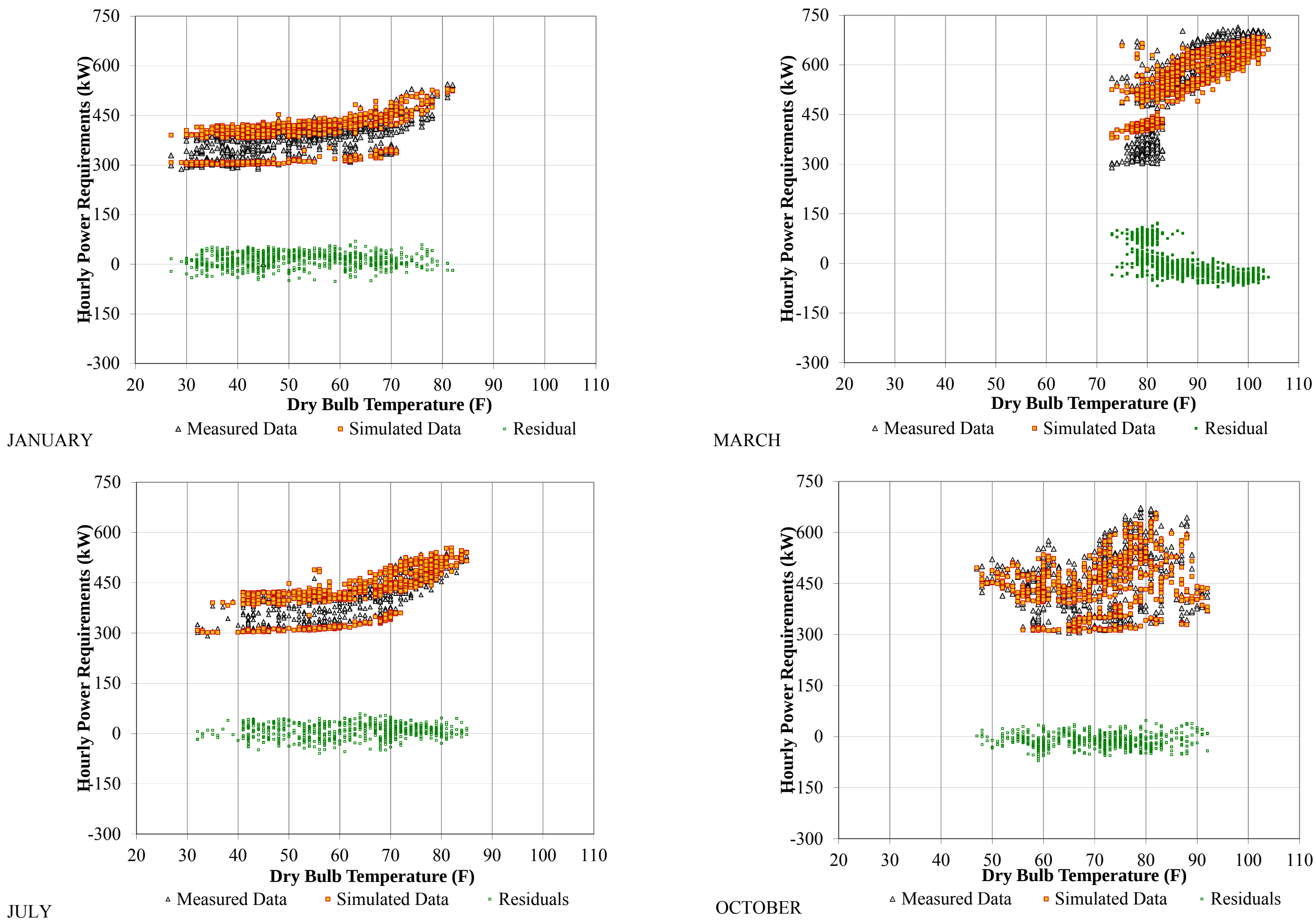


Figure 5-23: Results of the Scatter Plots for the Final Run and Measured Data for Whole-Building Electricity Consumption for January, July, October and March 2009 versus Ambient Temperature

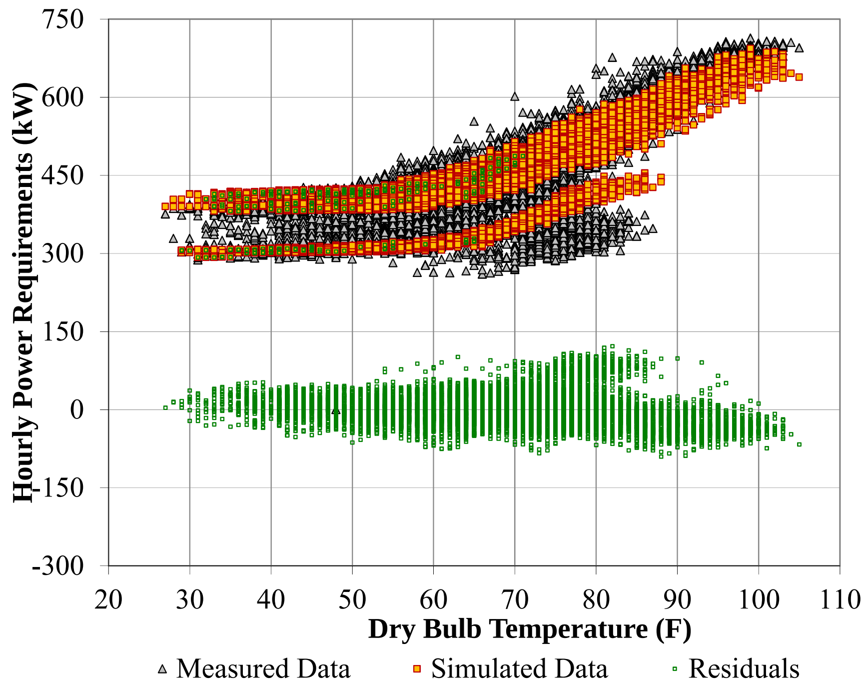


Figure 5-24: Comparing Whole-Building Electric Requirements of the Final Run and Measured Data with Dry Bulb Temperature between 1st January to 20th December 2009

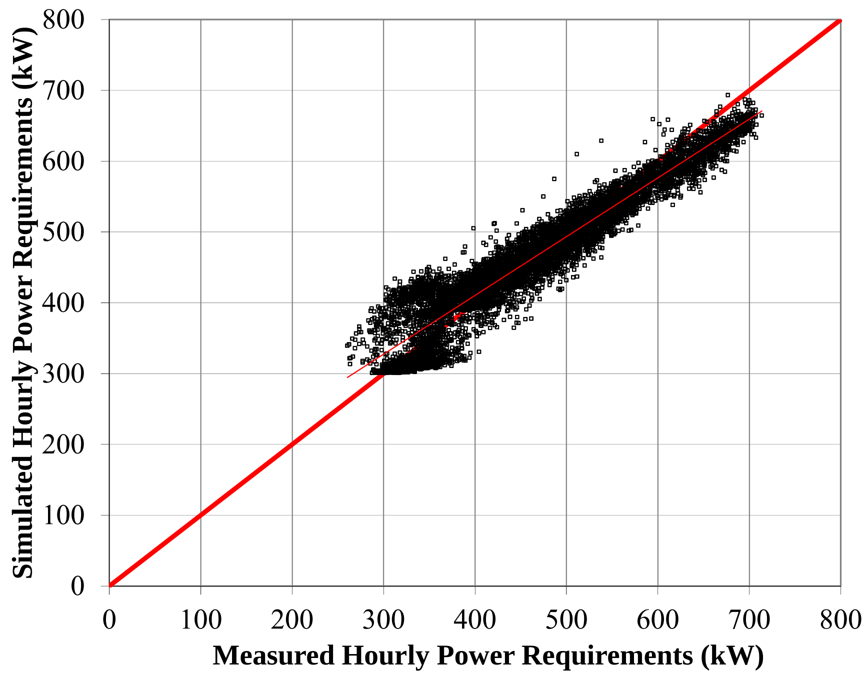
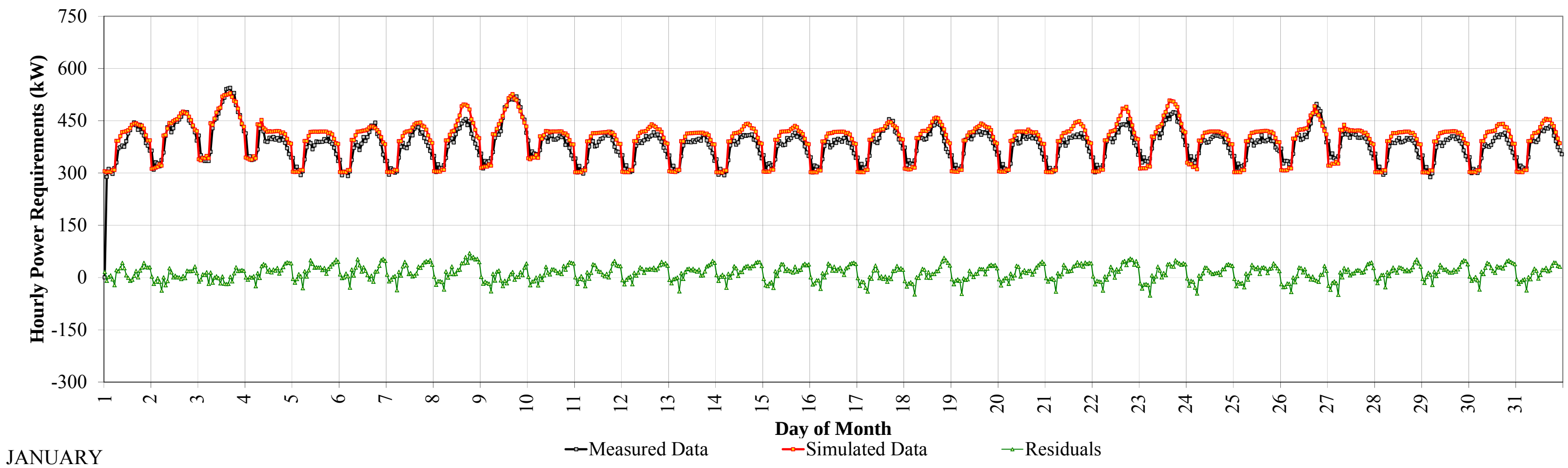


Figure 5-25: Comparison of the Simulated with Measured Whole-Building Electricity Consumption for the Final Run between 1st January to 20th December 2009



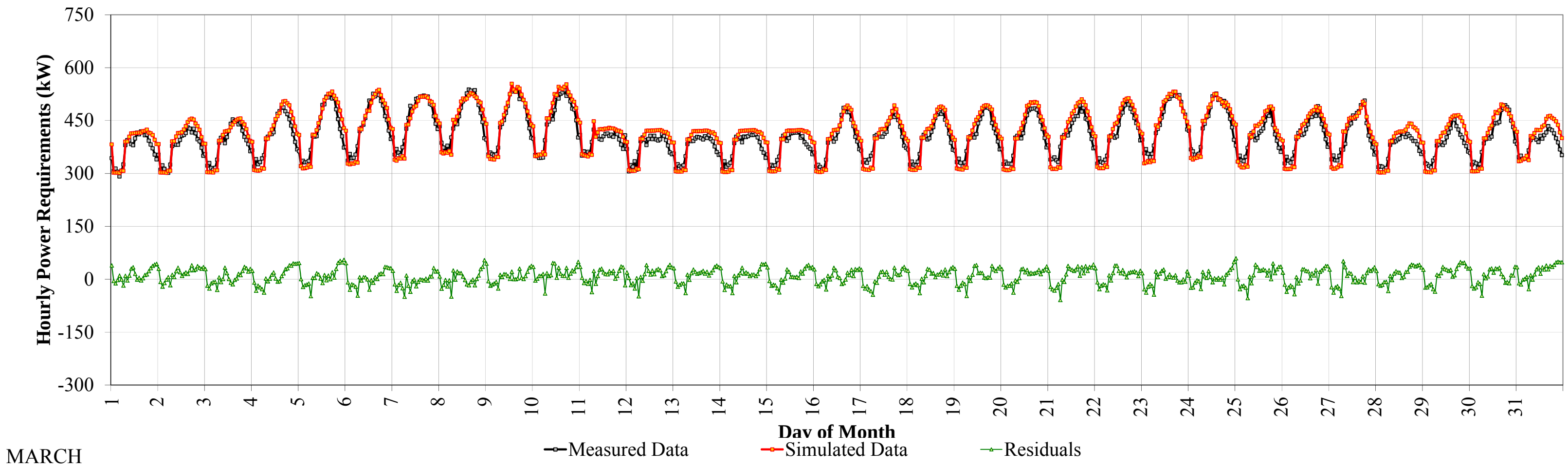
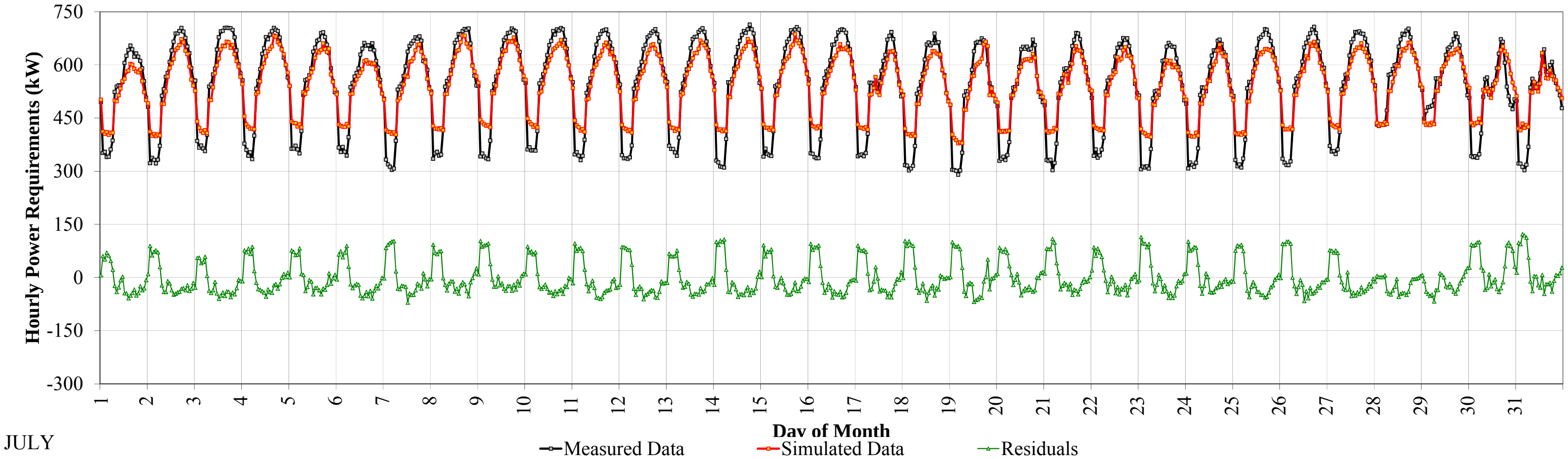
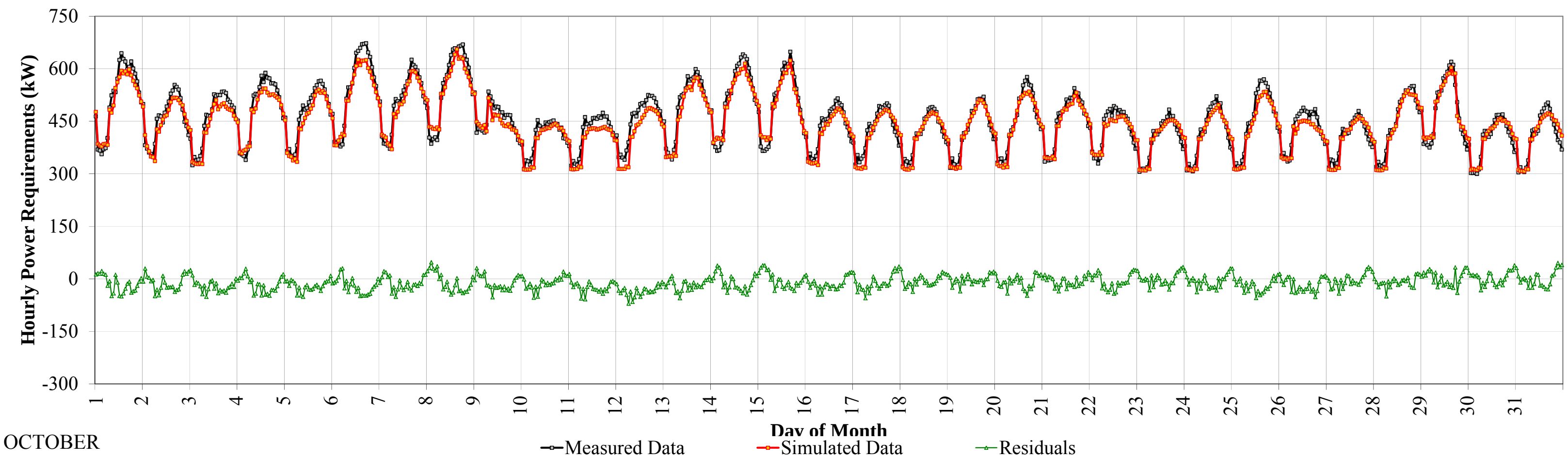


Figure 5-26: Time Series Plots of Whole-Building Electricity Consumption for the Final Run for Months of January and March 2009

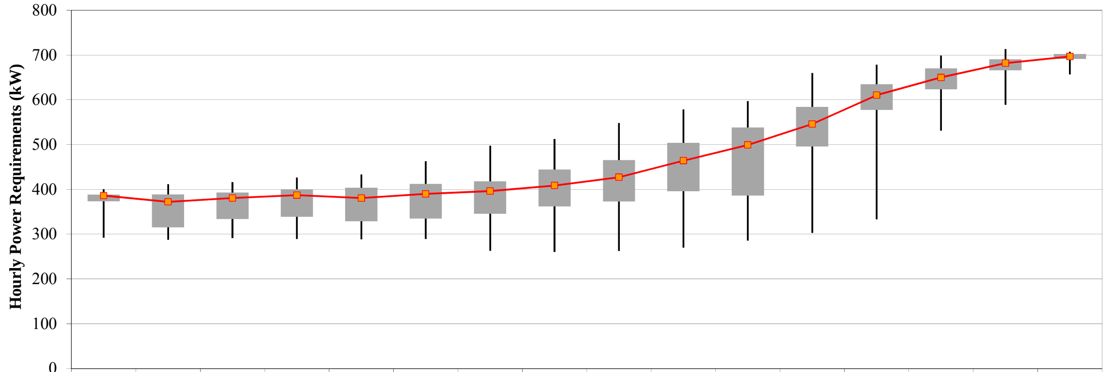


JULY



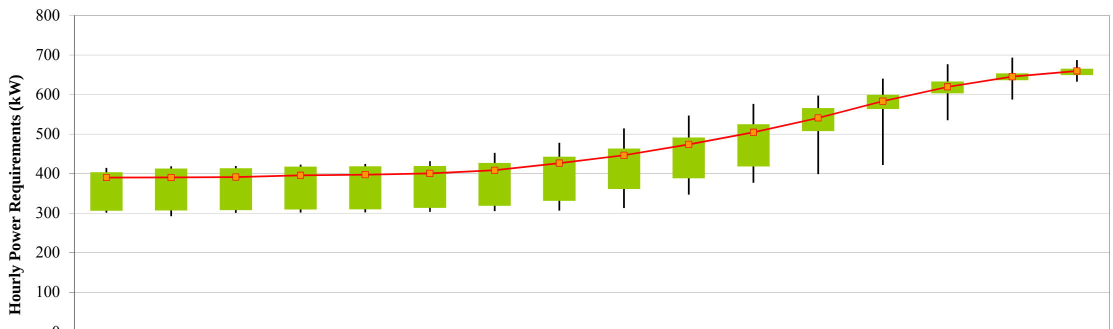
OCTOBER

Figure 5-27: Time Series Plots of Whole-Building Electricity Consumption for the Final Run for Months of July and October 2009



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
25%	373.3	315.2	333.8	338.9	328.6	334.9	345.6	361.8	372.9	395.7	386.1	495.7	577.4	623.4	665.8	691.4
Min	291.9	287.3	291.1	289.3	288.6	289.0	262.7	260.3	262.2	269.9	285.8	302.7	332.8	531.1	588.9	656.7
Max	400.4	411.3	416.0	426.5	433.3	463.1	497.6	512.4	547.9	578.6	597.0	659.7	678.3	698.9	713.6	707.2
75%	388.5	388.6	393.0	400.2	403.8	412.4	418.2	444.3	465.3	503.9	538.0	584.2	635.0	670.3	690.3	702.3
Median	386.1	372.1	380.6	387.0	380.6	389.9	396.3	408.8	427.1	464.0	499.2	546.0	610.4	649.9	681.9	696.7

Dry Bulb Temperature Bins (F)



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
25%	305.9	306.9	307.5	309.0	309.4	313.2	318.6	330.8	361.1	388.0	418.0	507.5	563.4	603.0	636.2	649.2
Min	301.2	292.4	300.9	301.5	302.0	303.1	304.9	306.4	312.7	346.9	376.6	398.6	422.0	534.9	587.4	632.7
Max	414.4	418.4	419.5	422.5	424.3	431.5	452.5	478.0	514.5	546.9	576.3	597.6	640.2	677.0	693.4	687.0
75%	403.8	412.7	413.6	417.6	418.4	419.3	426.8	442.7	463.2	491.4	525.0	565.8	599.5	633.1	653.8	665.3
Median	389.9	390.2	391.3	395.7	397.3	400.5	408.3	426.4	446.3	473.8	504.5	540.8	583.1	619.4	645.2	659.6

Dry Bulb Temperature Bins (F)

Figure 5-28: Results of Annual Bin Analysis of Measured Whole-Building Electricity Consumption and Corresponding Simulated Data for the Final Run between 1st January to 20th December 2009

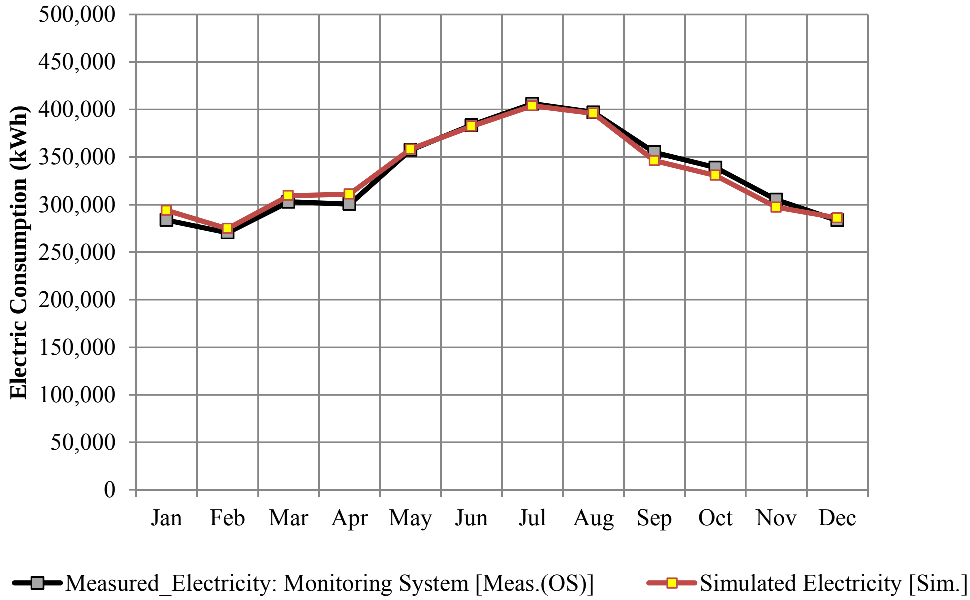


Figure 5-29: Results from Monthly Calibration of the Final Run with Measured Data for Whole-Building Electricity Usage - 1st January to 20th December 2009

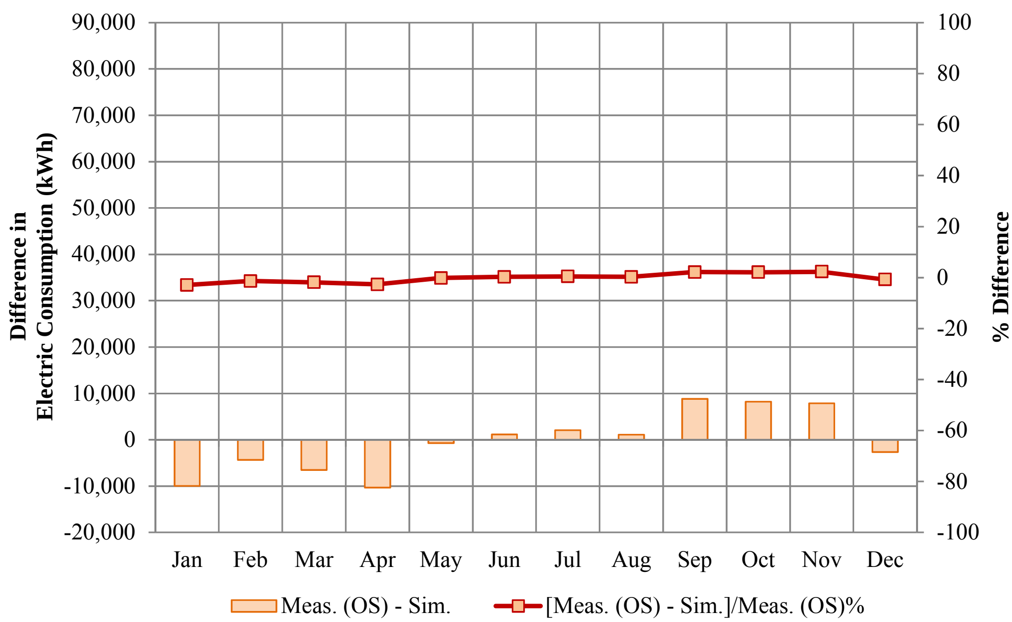


Figure 5-30: Residuals from Comparing Measured Data from Utilities and Measured Data from On-Site System for Whole-Building Electricity Usage for Final Run - 1st January to 20th December 2009

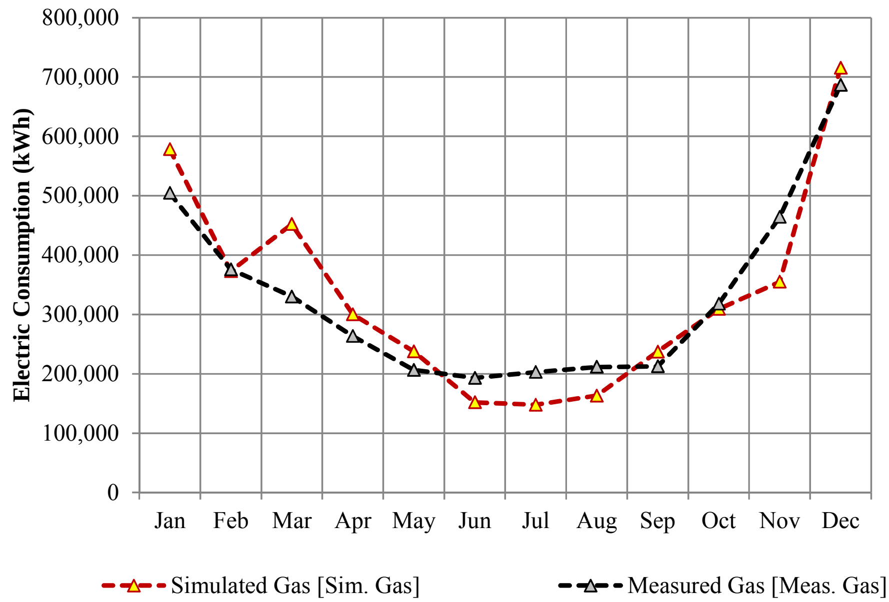


Figure 5-31: Results from Monthly Calibration of the Final Base-Case Run with Measured Data for Gas Energy Usage –January to December 2009

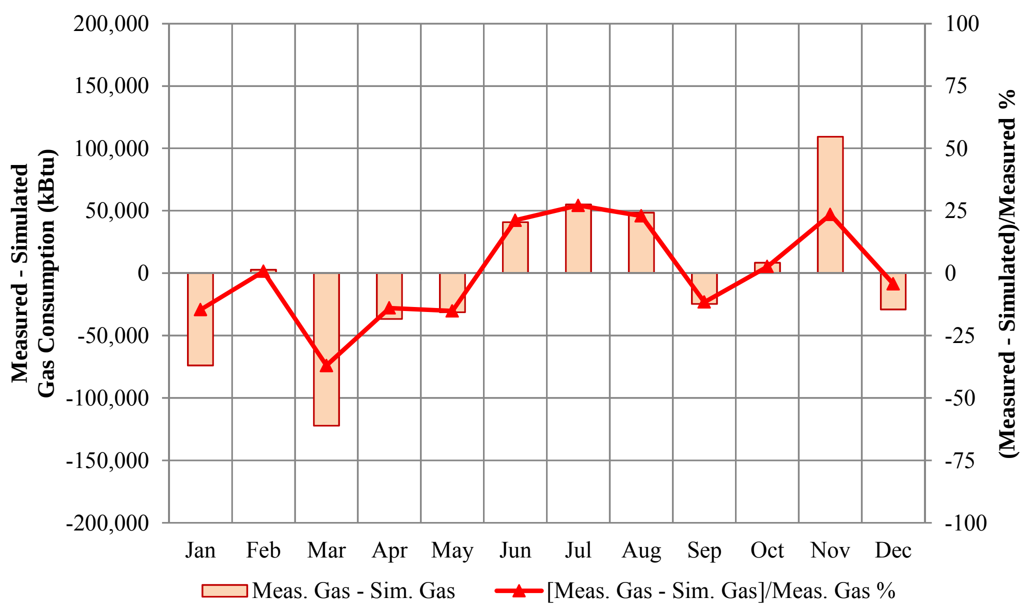


Figure 5-32: Residuals from Comparing Final Base-Case Run with Measured Data for Gas Energy Usage - January to December 2009

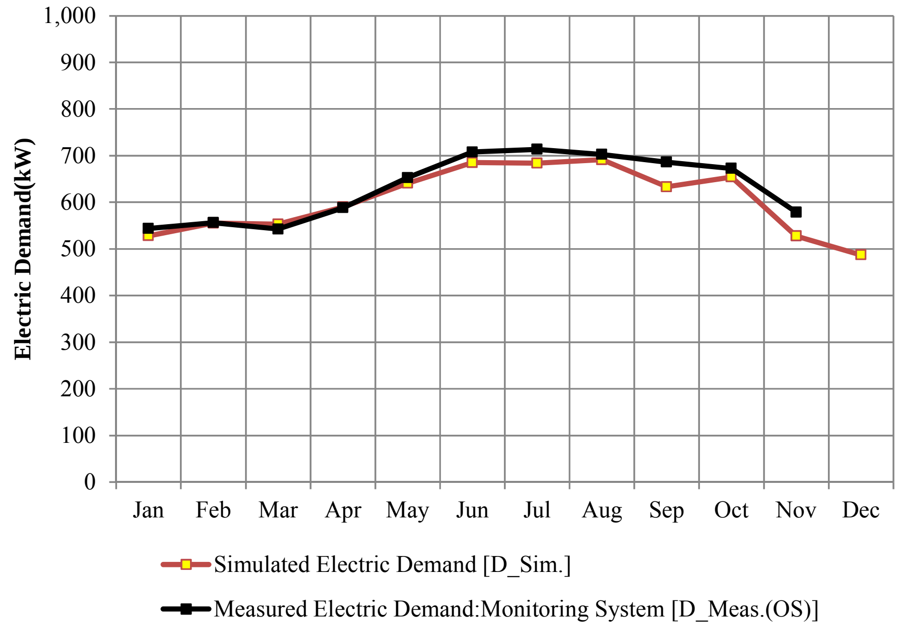


Figure 5-33: Results from Monthly Calibration of the Final Run with Measured Data for Whole-Building Electric Demand - 1st January to 20th December 2009

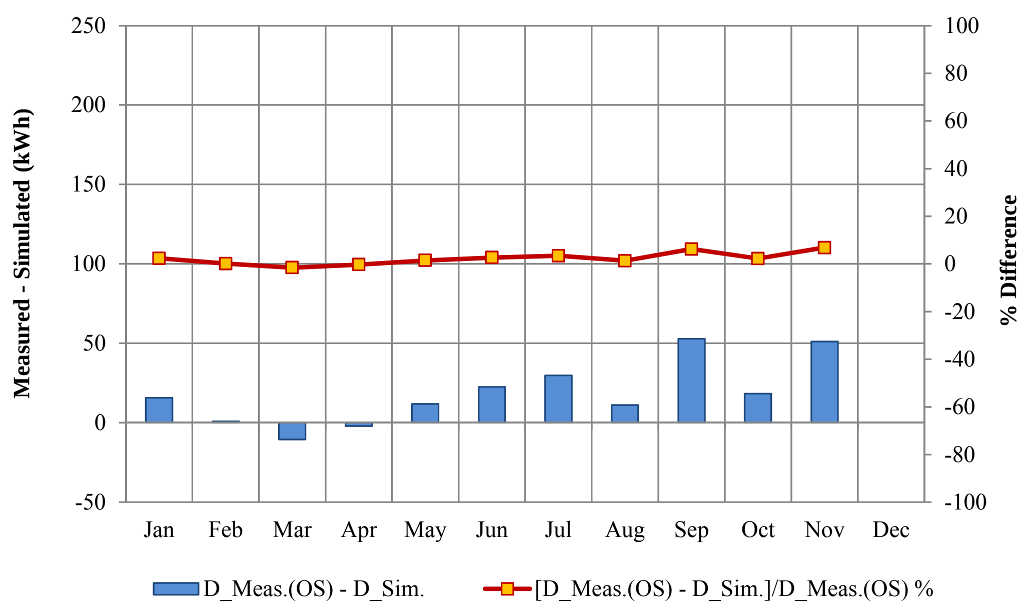


Figure 5-34: Residuals from Comparing Measured Data from Utilities and Measured Data from On-Site System for Whole-Building Electric Demand for Final Run - 1st January to 20th December 2009

CHAPTER VI

ENERGY EFFICIENCY MEASURES (EEMs) FOR THE GROCERY STORE

6.1 Overview

In this chapter several measures were examined for the grocery store to reduce the overall energy consumption. The calibrated grocery store model presented in the previous section was used to carry out this analysis. The measures were categorized into the following subsections. These subsections include: Envelope, lighting and daylighting, heating, ventilation and air-conditioning and service water systems; and refrigeration systems.

A comprehensive discussion of several measures is presented in the literature review of this study. However, only certain measures were selected under each category. The final selection process was based on whether the measure could be simulated in eQUEST – Refrigeration.

The first section of this chapter provides an overview of the efficiency measures considered for the analysis. The second and third sections describe the changes that were made to the calibrated building model to carry out this analysis. The fourth, fifth, sixth, seventh and eighth sections provide a description of the energy efficiency measures (EEMs) for the building envelope, lighting system, HVAC system, service water heaters and the refrigeration system. The impact of implementing individual measures on annual heating, cooling and total energy consumption is discussed in the ninth section. Finally, based on the results of individual assessments, certain measures were shortlisted, grouped and re-simulated to provide results for overall savings attainable in the grocery store that are presented in the tenth section of this chapter.

6.2 Changes to the Calibrated Base-Case Model

Unfortunately, the calibrated base-case grocery store model proved to be more efficient than a typical store in the U.S. as projected by CBECS (2007) in the nationwide survey of commercial buildings. In addition, the energy code in Texas for commercial buildings[1] during

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[1] The 2000 International Energy Conservation Code (IECC) with the 2001 supplement for residential, industrial and commercial buildings has been adopted by the Texas State Energy Conservation Office in 2001. One of the compliance paths prescribed in the 2001 IECC (Section 801.2) is to meet the requirements of the ASHRAE Standard 90.1-2001.

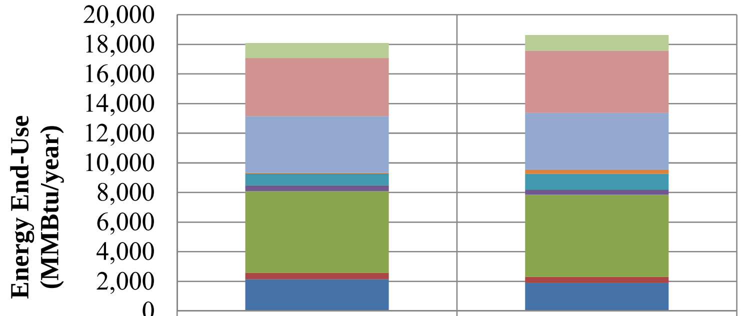
the time this store was built[2] was IECC 2000 with the 2001 supplement (ICC2000). However, in order to establish a baseline for this analysis the base-case grocery store model incorporated specifications from ASHRAE Standard 90.1-2004[3]. In addition, energy consumption from parking lights[4] was introduced in the base-case model. A complete list of changes that have been made in the calibrated model to make it suitable for this analysis is presented in Table 6-1. A graph presenting the end-use energy consumption for the calibrated base-case model and the modified base-case model of the store is presented in Figure 6-1.

Table 6-1: Modifications to the Calibrated Base-Case Building Model to Match ASHRAE Standard 90.1-2004 Specifications

Characteristics	Modifications	Source (ASHRAE 90.1-2004)
Construction		
Roof Insulation (Insulation entirely above deck)	R-15	Table 5.5-2 Building Envelope Requirements for Climate Zone 2 (A,B)
Roof Surface Properties	Solar Reflectance: 0.7 Emittance: 0.75	Section 5.5.3.1.1 High Albedo Roofs
U-factor of Glazing (Vertical glazing, 0 – 10% of wall)	U – 1.22	Table 5.5-2 Building Envelope Requirements for Climate Zone 2 (A,B)
Solar Heat Gain Coefficient (Ver. glazing, 0 – 10% of wall, facing north)	SHGC – 0.61.	Table 5.5-2 Building Envelope Requirements for Climate Zone 2 (A,B)
Skylight U-factor (Plastic w/ curb, 2.1% - 5%)	U_{all} - 1.90	Table 5.5-2 Building Envelope Requirements for Climate Zone 2 (A,B)
Skylight Solar Heat Gain Coefficient (W/ curb, 2.1% - 5%)	SHGC - 0.34	Table 5.5-2 Building Envelope Requirements for Climate Zone 2 (A,B)
Space Conditions		
Lighting Power Density (LPD) (Retail)	Sales Area: 1.5 W/ft^2 Food Prep.: 1.2 W/ft^2 Storage: 0.9W/ft^2	Table 9.5.1 Lighting Power Densities using Space-by-Space Method
Automated Lighting Controls in Space [a]	Modification to lighting schedule	Section 9.4.1.2 Space Controls
Exterior Lighting	Façade: 0.2 W/ft^2 (4.8 kW) Parking: 0.15 W/ft^2 (18 kW)	Table 9.4.5 Lighting Power Densities for Building Exteriors
HVAC and SHW System		
Air-Conditioner Efficiency (Air Conditioners, Air Cooled)	For "General" zone: 9.5 EER For all other zones: 10 SEER	Table 6.8.1A Electronically Operated Unitary Air Conditioners
Supply Fan Efficiency	55%	Thornton et al. 2010 (Standard practice)
Service Water Heater Efficiency (Gas Storage >75 kBtu/h)	80% E_t	Table 7.8 Performance Requirements for Water Heaters

Notes:
a. Although the grocery store model is open to public between 6:00 AM to 12:00 AM, the grocery store can be designated to be operating for 24 hours because of the constant delivery of products to the store.
b. Energy conservation standards for selected refrigeration equipment in the grocery store have been established by the federal and state governments of the US. Federal energy conservation standards include the Energy Policy Act of 2005 (EPACT), which provided minimum standards solid-door reach-in refrigerators, freezers, and refrigerator-freezers, glass-door refrigerators and freezers, and automatic commercial ice machines manufactured on or after January 1, 2010 (Goetzler et al., 2009).The case-study store was constructed in 2003 and hence exempt from the regulations imposed by the EPACT.

[2] The case-study store was built in 2003.
[3] The use of R-22 refrigerant for operating the refrigeration system in the grocery store was retained.
[4] In the calibrated model the energy consumption from parking lights was not considered.



	Calibrated Store Model	ASHRAE 90.1-2004 Compliant Store Model
Misc. Equip. (E+G)	1,013	1,069
Space Heat. (G)	3,924	4,180
Lighting (E)	3,836	3,822
Ext. Light. (E)	27	312
Pumps & Fans (E)	803	1,047
SWH (E+G)	383	355
Refg. (E)	5,537	5,539
Heat Rej. (E)	410	385
Space Cool (E)	2,154	1,927

Note: Both the simulations were performed using TMY2 weather file for College Station, TX.

Figure 6-1: Comparison of End-Use Energy Consumption from Calibrated and Modified Base-Case Models

6.3 Weather Data

A TMY3 weather file for College Station, Texas was used for the EEM assessment process. This weather file replaced the TRY weather file with measured data, which was used for calibrating the simulation model. Although the TMY3 is not designed to provide meteorological extremes, the data set in the TMY3 file represents a year of typical climatic conditions for the location (Wilcox and Marion, 2008). This representation of typical climatic conditions was considered appropriate in the assessment of the EEMs in the grocery store.

6.4 Efficiency Measures for the Envelope

As seen in the energy end-usage of the base-case grocery store, a major portion of the energy consumption in grocery stores is from refrigeration systems and lighting, both of which are not strongly dependent on ambient conditions. Hence efficiency measures for the building envelope of the grocery store, whose primary task is to insulate the building interiors for ambient conditions do not have a major impact on reducing the overall energy consumption. The measures selected for this study included:

- Enhanced insulation for exterior walls and roof,
- High albedo roof,
- Enhanced insulation for walls and roofs of spaces at low temperature conditions (i.e. freezers, coolers and preparation rooms),
- Improved insulation for loading dock doors,
- Increased area for skylights,
- Improved specifications for skylights, and
- Reduced infiltration in the general spaces and low temperature spaces in the store.

6.4.1 Enhancing Insulation R-values for Walls and Roof

For exterior walls, a continuous layer of 1 inch thick expanded polyurethane insulation with an insulation value of R-5.7 was added to the exterior surface of the pre-cast concrete panel of the exterior wall construction in the base-case model of the grocery store[5]. Similarly, for the roof, a continuous layer of 4 inch thick expanded polyurethane foam with an insulation of R-25 was used. The roof insulation of the base-case building is R-15 continuous insulation. The base-case building had a polystyrene insulation of R-2 sandwiched between the pre-cast concrete panels. The improved values for exterior walls have been adapted from ASHRAE Standard 90.1-2010[6] and the improved values for the roof insulation have been adapted from the Advanced Energy Design Guideline for Medium to Big Box Retail Buildings (ASHRAE 2011).

6.4.2 Installation of High Albedo Roof

A white polyvinyl chloride (PVC) coating was used to create a high albedo roof surface[7]. The PVC coating has solar reflectance of 0.9 and thermal emittance of 0.9. This product was selected from the rated products directory provided by the Cool Roof Rating Council (2012).The roof surface of the base-case model had a solar reflectance of 0.7 and an emittance of 0.75.

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[5] The polyurethane insulation was placed in the air gap between the exterior stone layer and the pre-cast concrete panel exterior wall.
[6] Table 5.5-2 Building Envelope Requirements for Climate Zone 2(A,B), ASHRAE Standard 90.1-2010.
[7] High-albedo roof coatings have high solar reflectance (both in visible and near-infrared bands), have high infrared emittance, hence lowering the absorption of solar energy and reducing the heat transfer into the interior of the building (Bretz and Akbari 1994).

6.4.3 Enhancing Insulation R-values for Walls and Roof of Low Temperature Spaces

The insulation values of walls and roof of the freezer, cooler and preparation room were improved to R-50 and R-60 respectively. These values have been adopted from the specifications for blast freezers (Becker and Fricke 2005). The insulation values for the walls and roof of the freezer, cooler and preparation room in the base-case model was set at R-31. An R-25 insulation was assumed for the walls and roof of the freezer, cooler and preparation room in the base-case.

6.4.4 Improving Insulation for Loading Dock Doors

Loading dock doors represent a very small component of the grocery store. Nevertheless, the impact of implementing an improved insulation value for these doors was assessed. An improved insulation value of R-4.75 was considered for this analysis. This value was obtained from the AEDG for Medium to Big Box Retail Buildings (ASHRAE 2011)[8] and the technical support document for the AEDG for small warehouse and self-storage buildings (Liu et al. 2007). Un-insulated loading dock doors were implemented in the base-case model. The base-case loading dock door assembly had an overall U-value of 2.08 Btu/h ft^2 °F.

6.4.5 Improving the Specifications for Skylights

In the first efficiency measure for skylights the area was changed from 4.2% as implemented in the base-case model to 7% of the total roof area as recommended in the technical support document for the AEDG for small warehouse and self-storage buildings (Liu et al. 2007)[9]. In the second efficiency measure for skylights specifications for U-value, SHGC and visible transmittance were altered according to product specifications provided in the National Fenestration Rating Council (NFRC) certified product directory (2012)[10]. Glazing U-value was improved to 0.53 from 1.90 Btu/hr ft^2 °F as assumed in the base-case model; the SHGC was improved to 0.27 from 0.34 assumed in the base-case; and the visible transmittance was increased to 62% from 60% as assumed in the base-case. Improved specifications for skylight frames were also examined in this measure. Vinyl skylight frames with improved U-value[11] of

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[8] In this AEDG refer to specifications for Climate Zone-2.

[9] This number was later changed when the energy saving potential of daylighting controls was assessed. An optimum percentage for skylight area was established by performing a number of iterations assessing different percentage of skylight area in combination with daylighting controls.

[10] eQUEST-Refrigeration accepts center of glass values when inputting specifications for glazing. The impact of the frame on the U-value of the skylight assembly is accounted for separately in the program.

[11] eQUEST-Refrigeration accepts the input for frame conductance, which is reported by excluding the outside air film resistance. The frame conductance is calculated from the corresponding U-value by the following equation: Frame – Conductance = (U-value) -1-0.197 Btu/h ft^2 °F.

0.30 Btu/hr ft^2 °F were modeled along with the improved skylight glazing specifications (Hirsch 2006). The skylight frames in the base-case model were of thermally unbroken aluminum with a U-value of 1.90 Btu/hr ft^2 °F.

6.4.6 Reducing Infiltration

Specifications for reduced infiltration rates have been provided by the technical support document for the AEDG for medium box retail buildings (Hale et al., 2008) and the technical support document for the AEDG for small warehouse and self-storage buildings (Liu et al., 2007). Using information from the report on medium box retail buildings, the infiltration rate was reduced to 0.104 ACH [12] for all spaces in the model except loading docks. The infiltration rate in the base-case simulation model for these spaces was set at 0.5 ACH.

Using information from the report on small warehouses and self-storage buildings, for "General Loading Dock" zone, infiltration rates were reduced to 0.6 ACH and 0.15 ACH [13] for loading dock doors in open and closed positions respectively. For "Produce Loading Dock" zone the infiltration rates were reduced to 0.34 ACH and 0.127 ACH for loading dock doors in open and closed positions respectively. The peak infiltration rate in the base-case model is assumed to be 1 ACH in the "Loading Dock General" zone and 0.8 ACH in the "Loading Dock Produce" zone. These rates vary according to a schedule implemented in the base-case model[14].

Infiltration levels for freezers, coolers and preparation rooms are reduced to 0.04 cfm/ft^2 (Downing and Meffert 1993)[15]. The infiltration of freezers, coolers and the preparation room was set to be to 0.07 cfm/ft^2 for the cooler and preparation room, and 0.09 for the freezers in the base-case model. The efficiency measures for the envelope are provided in Table 6-2, which follows.

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[12] According to the report on medium box retail buildings (Hale et al. 2008), the infiltration rate in the space is reduced by applying an envelope air barrier or a front entrance vestibule. The installation of the air barrier reduces the envelope infiltration to 0.05. The installation of a vestibule is assumed to reduce the front door infiltration to 0.054.

[13] According to the report on small warehouses and self-storage buildings, a reduced infiltration of 22.4 cfm / door is calculated for closed dock doors and a reduced infiltration of 203 cfm per door is calculated for open dock doors. As in the general spaces, the installation of the air barrier in these zones reduces the envelope infiltration to 0.05.

[14] When calculating the infiltration values of the loading docks, this study assumes the doors to be open for 75% of the time for the "General Loading Dock"; and the doors to be open for 25% of the time for the "Produce Loading Dock". These assumptions were based on the consultations with the store management.

[15] The study by Downing and Meffert considered eight protective devices for the analysis of infiltration reduction in cold storage spaces. The study concluded that between 82.5% to 97.8% infiltration can be reduced through an open unprotected door with the installation of strip curtains. Strip curtains are currently implemented in the case-study store and hence modeled in the base-case model. Hence, to implement this efficiency measure infiltration is further reduced by the implementation of improved strip curtain design.

Table 6-2: Energy Efficiency Measure for the Building Envelope

EEM No.	Base-Case Parameters	Units	Efficiency Measure Description	Reference
BUILDING ENVELOPE				
1	**Exterior Walls**			
	Insulation	hr.ft².F / Btu	R-5.7 c.i.	ASHRAE 90.1-2010
	Roof			
	Insulation (For entirely above deck)	hr.ft².F / Btu	R-25 c.i.	ASHRAE 90.1-2010
2	Roof Surface		Solar reflectance = 0.90 Thermal emittance = 0.90 (White PVC)	CRRC, 2012
3	**Freezer and Cooler Walls and Roof**			
	Wall Insulation	hr.ft².F / Btu	R-50 c.i.	Becker and Fricke, 2005
	Roof Insulation	hr.ft².F / Btu	R-60 c.i.	
4	**Doors**			
	Insulation for Loading Dock Doors	hr.ft².F / Btu	R-4.75	AEDG, 2011 Liu et al., 2007
5	**Skylights**			
	Area	%	7 (Unit size: 6.58' x 7.65')	Liu et al., 2007
6	U-Value	hr.ft².F / Btu	0.53	NFRC, 2012
	SHGC	-	0.27 (SC: 0.31)	
	Visible Transmittance	%	62	
7	**Infiltration**			
	Infiltration Rates	ACH CFM/ft^2	For all spaces except docks: 0.2 For general dock: 0.15 / 0.6 (wt. av. 0.5) For produce dock: 0.127 / 0.34 (wt. av.0.2) For freezers, cooler and prep. room: Reduce infiltration by 80%	Hale et al.,2008 Liu et al., 2007 Downing and Meffert, 1993

Note: The 'EEM No.' assigns a unique number to the efficiency measure simulated by this study. Some measures that were considered in the initial list of EEMs were removed either due to poor energy savings or due to technical issues involved in modeling these measures. However, the unique 'EEM No.' for each measure was retained from the initial list of measures to avoid discrepancy in presenting and describing the results.

6.5 Efficiency Measures for the Lighting Systems and Plug Loads

In the next section, efficiency measures for lighting systems and plug loads were examined. Long operating hours of a typical grocery store require lighting systems to be operational almost continuously. This makes the implementation of the measures discussed in this section especially viable as potential energy reduction strategies. The measures selected for this study include:

- Reducing internal lighting power density,
- Implementing time switches and occupancy sensors,
- Implementing daylighting controls,
- Reducing exterior façade lighting,
- Reducing selected parking lighting levels,
- Reducing equipment power density levels, and
- Optimizing equipment operation schedules.

6.5.1 Reducing Internal Lighting Power Density (LPD)

In the first efficiency measure for lighting, the lighting power density (LPD) in the sales areas (‘General Merchandise’, and ‘Display Case’ zones) was reduced to 1.15 W/ft^2. While for storage spaces, which include the general and produce loading docks, the cooler, the freezer and the preparation room areas, the lighting power density is reduced to 0.6 W/ft^2. These values were adapted from the AEDG for Medium to Big Box Retail Buildings (ASHRAE 2011). In addition, the LPD of the ‘Bakery’ and the preparation rooms was reduced to 0.83 W/ft^2, which were adopted from the technical support document for 50% energy savings in quick service restaurants (Zhang et al. 2010). The LPD in the base-case building are set according to ASHRAE Standard 90.1-2004[16], which include 1.6 W/ft^2 for sales area (‘General Merchandise’ and ‘Display Case’ zones), 1.2 W/ft^2 for preparation areas (‘Bakery’ and ‘Preparation Room’ zones) and 0.9 W/ft^2 for storage and subservient areas (loading docks, ‘Freezer’ and ‘Cooler’ zones).

6.5.2 Implementing Time Clocks and Occupancy Sensors

The implementation of time switches and occupancy sensors is introduced in this measure to electric reduce lighting loads. This measure was introduced in addition to the reduced

[16] Table 9.6.1, Lighting Power Densities Using Space-by-Space Method, for retail (sales area), food preparation and for warehouse (Medium/Bulky material storage).

lighting power density measure discussed in the previous sub-section. In this measure time clocks were implemented to reduce the LPD during stocking and unoccupied periods. Time clocks were simulated by altering the lighting schedules for the appropriate zones to incorporate stocking period and unoccupied period. During the stocking period in the sales areas the LPD was reduced from 1.15 W/ft^2 to 0.6 W/ft^2 to match the LPD requirements of stockrooms assumed in the previous efficiency measure which describes the implementation of reduced LPD in the grocery store. During unoccupied periods for all zones, the LPD was reduced to 5% of the total LPD of the zone (ASHRAE 2010b). Time clocks were not implemented in the base-case simulation model of the store.

Occupancy sensors were implemented in freezers and coolers. Since no information was found in the literature review on implementing the occupancy sensors in these spaces, the schedules were arbitrarily modified to best suit the occupancy patterns of these spaces as observed in the case-study store. A 5% reduction in the LPD was arbitrarily assumed in the loading dock zones to account for the implementation of occupancy sensors in these areas. The sensors were also implemented in certain areas of the 'General Merchandise', 'Display Case' zones, which include restrooms, offices and computer rooms as well as lighting for display-cases in the cosmetics section of the grocery store[17]. These areas comprise a small percentage of the total area of the parent zone and have lower LPDs than that of the parent zone. The resultant LDP of the entire zone was reduced by 1% assuming that the sensors achieve 10% savings in the areas in which they were implemented (Hale et al. 2008)[18]. Occupancy sensors were not modeled in as the 'Bakery' and the 'Preparation Room' zones. Figure 6-2 provides a sample comparison of the schedules of lighting levels in the 'General Merchandise' zone of the base-case model with and without the implementation of occupancy sensors. The modified lighting schedules used in the base-case simulation model to simulate the implementation of time clocks and occupancy sensors for all the zones are provided in Table B-27 of Appendix B.

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[17] These spaces are not accounted for in the ASHRAE 90.1-2004 base-case building.

[18] The study by Hale et al., assumed a 1% reduction in LPD to include the performance of occupancy sensors in various spaces of the grocery store (i.e., active storage, office, lounge, restroom and electrical / mechanical spaces). The reduction of 1% was calculated by assuming that sensors achieve 10% savings in areas where they are installed. Because those areas comprise just 11/3% of the building and have lower LPDs than the sales floor, the whole-building LPD was estimated to be reduced by 1%.

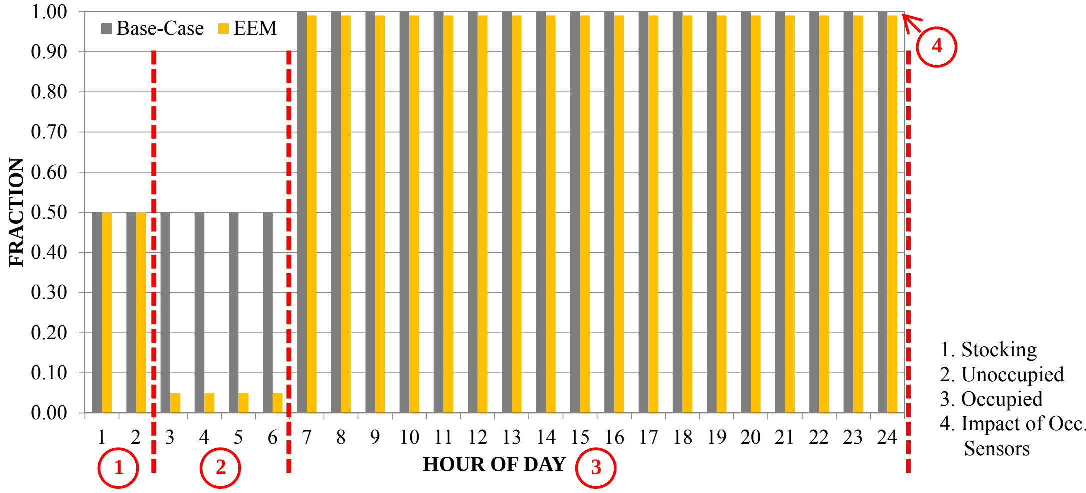

Figure 6-2: Lighting Schedules With and Without the Implementation of Time Clocks and Occupancy Sensors for the "General Merchandise" Zone

6.5.3 Implementing Daylighting Controls

Daylighting controls were introduced in the daylit zones of the simulation model of the grocery store (i.e., 'General Merchandise' and 'Display Case' zones). Specifications for the daylighting controls are adapted from the technical support document for 50% savings in grocery stores (Leach et al., 2009). Continuous dimming controls were modeled by implementing two sensors in each day lit zone at a height of 2.95 feet from the floor. The controls were set to linearly decrease the lighting levels till the lighting set point was met or the input power decreased to 30% of the maximum power levels which corresponds to lighting levels decreases to 20% of the maximum lighting levels. A schematic diagram for continuous controls is presented in Figure 6-3. The assessment was restricted to 2 sensors per daylit zone due to modeling constraints of the eQUEST-Refrigeration software. The lighting set point was set at 46.5 fc. The layout of the daylighting sensors used in the simulation of the daylighting efficiency measure is presented in Figure 6-4. In addition to implementing daylighting controls, the ceiling surface reflectance of the daylit zones was increased to 80% from 70% implemented in the base-case model. This measure was adopted from the AEDG for Small Warehouses (2008).

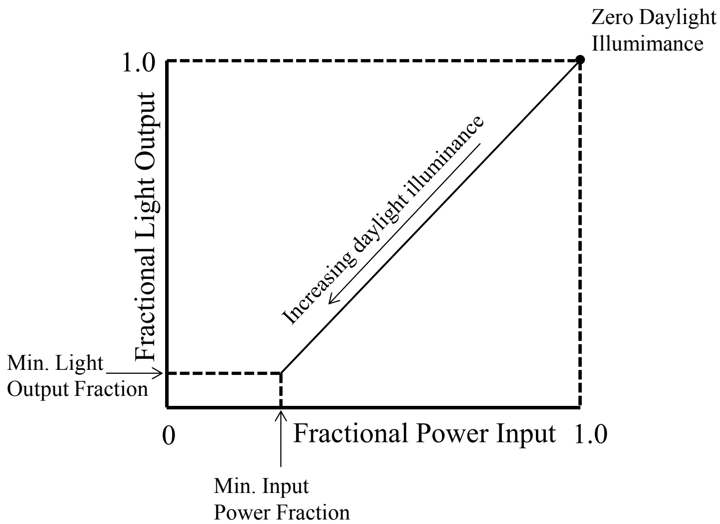

Figure 6-3: Schematic Diagram of Continuous Dimming Daylighting Controls (Source: Hirsch et al., 2006)

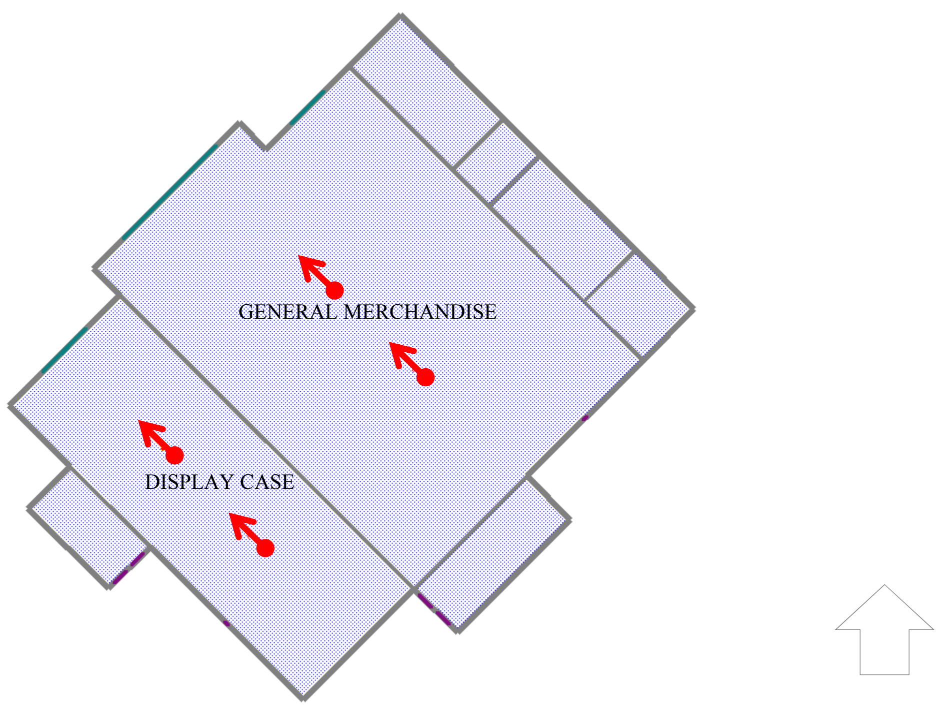

Figure 6-4: Layout of Daylighting Sensors in the eQUEST-Refrigeration Mode

6.5.4 Implementing Exterior Façade and Parking Lighting Systems

The exterior façade lighting was reduced to 0.05 W/ft^2 of façade area as prescribed in the AEDG for Medium to Big Box Retail Buildings (ASHRAE 2011). As a result, the total power allocated to exterior façade lighting was reduced to 1.3 kW from 4.8 kW as originally described in the base-case model of the store. Lighting power densities of exterior parking lights were reduced from 0.15 W/ft^2 as described in ASHRAE Standard 90.1-2004[19] to 0.05 W/ft^2 [20] (ASHRAE 2011). In addition, the schedule of operation for all exterior lighting was reduced to 25% of the full capacity from 12:00 AM to 6:00 AM (ASHRAE 2011). The exterior lighting schedules were unaltered in the base-case model. The modified exterior lighting schedules used in the simulation model are provided in Table B-28 of Appendix B.

6.5.5 Reducing Equipment Power Density Levels (EPD)

Two measures were considered in this category. The first measure involved the installation of energy efficient equipment. Using the recommendations proposed by the EPA for load reduction in grocery stores using ENERGYSTAR equipment[21] (US EPA 2008a) and the recommendations provided in the technical support document for 50% energy savings in small office buildings (Thornton et al., 2010)[22] the EPD in the simulation model was reduced by 25%. As a result, the total power allocated to miscellaneous equipment for all zones except the bakery was reduced to 0.375 W/ft^2 from 0.5 W/ft^2 as described in the base-case model (Table B-29). For the bakery, the EPD was reduced to 2.25 kW from 3 kW as originally described in the model. In addition, energy consumption for equipment operated by natural gas was reduced to 43,500 Btu/h from 58,006 Btu/h as provided in the original base-case model.

In the second measure, the equipment schedule was optimized to incorporate power management software, occupancy sensor controls for computer monitors and other equipment and simple practices such as turning off equipment when not in use (Thornton et al., 2010). To accomplish this measure the equipment schedule was modified to incorporate 20% reductions in the total plug loads. The efficiency measures for the lighting and equipment are summarized in Table 6-3, which follows.

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[19] Table 9.4.5 Lighting Power Densities for Building Exteriors, Uncovered Parking Areas, ASHRAE 90.1-2004.

[20] It was assumed that the grocery store was situated in Lighting Zone 2, which represents areas predominantly consisting of residential zoning.

[21] According to this document, installation of ENERGY STAR qualified cooking equipment uses 10 to 50% less energy than conventional models.

[22] As seen in this report, office buildings have miscellaneous equipment similar to some of the equipment found in a typical grocery store (i.e. desktop computers and printers).

Table 6-3: Energy Efficiency Measure for Lighting, Daylighting and Equipment

EEM No.	Base-Case Parameters	Units	Efficiency Measure Description	Reference
LIGHTING & EQUIPMENT				
	Lighting & Daylighting			
8	Int. Lighting Power Density (LPD)	W/ft²	Sales areas: 1.15 Storage areas: 0.6 Food Preparation: 1.2	AEDG, 2011
8 + 9	Time Switches & Occupancy Sensors	-	For Time Switches: Altered schedules for all zones to incorporate stocking period and unoccupied period. For Occupancy Sensors: 1% reduction in LPD in "General Merchandise" zone. Alterered schedules in subservient zones. Modified schedules for time switches and occupancy sensors provided in Table B-27, Appendix B.	AEDG, 2008 AEDG, 2011 Leach et al., 2009 Hale et al., 2008 ASHRAE, 2010b
10	Daylighting Controls	-	2 sensors for each daylit zone. Daylighting setpoint - 46.5 fc Lighting setpoint = 20% Power setpoint = 30% Continious dimming controls Z-axis = 2.95 Glare index: 22	Leach et al., 2009
10+11	Ceiling Surface Reflectance	%	80	AEDG, 2008 AEDG, 2011 Leach et al., 2009 Hale et al., 2008 ASHRAE, 2010b
12	Ext. Façade Lighting	kW	0.05 W/ft^2 and 0.1 footcandle Total Power: 1.3 kW Schedule reduced to 25% of full capacity from 12:00 AM to 6:00 AM Exterior lighting schedule, Table B-28	AEDG, 2011
13	Ext. Parking Lighting	W/ft²	0.06 (For Lighting Zone 2: Predominantly residential, neighborhood business districts, ligth industrial with limited nighttime use and residential mixed use areas) Exterior lighting schedule, Table B-28	AEDG, 2011
14	**Plug & Process Loads**			
	Equip. Power Density	W/ft²	25% reduction in plug loads on implementing ENERGYSTAR rated equipment in the office and the bakery	Thornton et al., 2010 US EPA, 2008a
15	Equip. Schedule	-	Reducing from 40% to 15% during unoccupied hours Reducing from 65% to 50% during transition hours Modified schedule provided in Table B-29	Hale et al., 2008 Thornton et al., 2010 Arbitrary selection

6.6 Efficiency Measures for the HVAC Systems

In this section the efficiency measures for HVAC systems are described. The efficiency measures include:

- Improved cooling and heating efficiency,
- Installation of economizers,
- Implementation of heat recovery from refrigeration coils,
- Installation of packaged variable air volume systems (PVAVS),
- Installation of dedicated outdoor air system (DOAS),
- Improved the efficiency of supply fans and exhaust fans, and
- Installation of demand control ventilation (DCV) strategies.

6.6.1 Improving Cooling and Heating Efficiency

In this measure the cooling efficiency[23] of the packaged units was improved by 20% as described in the technical support document for 50% energy savings in grocery stores (Leach et al., 2009). The installation of condensing furnaces[24] improved the heating efficiency of the packaged units to 92% (US EPA 2008a). The cooling and heating efficiencies of packaged units in the base-case model are provided in Table B-6 of Appendix B in this study.

6.6.2 Installation of Economizers

The base-case simulation model implements the ASHRAE Standard 90.1 – 2004 code. Hence, according to the ASHRAE standard no economizer is required[25] in the base-case model. More recently, for buildings located in Climate Zone 2a ASHRAE Standard 90.1-2010 requires enthalpy based economizers to be implemented in cooling units with cooling capacity greater than or equal to 54,000 Btu/h[26]. However, the use of economizers for grocery store is exempted in the code as the use of outdoor air may have a detrimental impact on products contained in the open refrigerated casework systems[27]. Nevertheless, this measure was simulated to generate an academic discussion regarding the impact of installing economizers in grocery stores. The

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[23] As described in the technical support document the improved in efficiency of cooling units include the compressor and condenser but exclude the supply fans.

[24] Condensing furnaces work on the principle of capturing the latent heat that is released when the hot flue gases condense by implementing a corrosion-resistant secondary heat exchanger (Sachs 2005).

[25] Table 6.5.1, minimum system size for which an economizer is required, for climate zone 2a, ASHRAE Standard 90.1-2004.

[26] Table 6.5.1A, minimum fan-cooling unit size for which an economizer is required for comfort cooling, for climate zone 2a, ASHRAE Standard 90.1-2010.

[27] Section 6.5.1, Economizers, Exception h., ASHRAE Standard 90.1-2010.

enthalpy based economizers were selected and modeled by setting the enthalpy high limit to 28 Btu/lb as prescribed in ASHRAE Standard 90.1 – 2010 code[28].

6.6.3 Heat Recovery from the Refrigeration System

For a typical grocery store the year around cooling effect from refrigerated display cases creates heating loads for all or a substantial portion of the year (Hirsch 2006, Khattar and Henderson 2000). In many instances, the heat available in the hot gas discharged from the refrigeration system compressor can be readily recovered for space heating and service water heating (Baxter 2003) to meet these requirements. In this study, heat recovery from refrigeration systems was simulated by recovering a portion of energy available in the superheated refrigerant (Hirsch 2006, Sawalha and Cheng 2010, Fricke 2011).

The simulated strategy considered for this analysis involved the installation of a de-superheater[29] right before the air-cooled condenser into which the heat is rejected. A schematic diagram of the system configuration is presented in Figure 6-5.

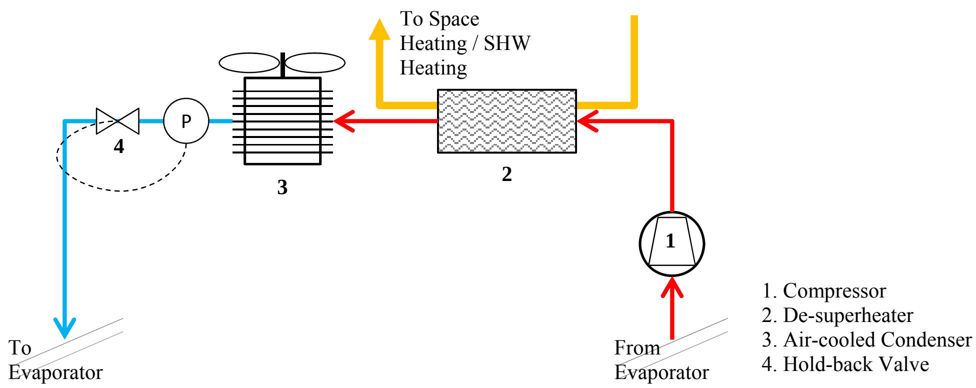

Figure 6-5: Schematic Diagram of the Heat Recovery at the De-Superheater (Source: Sawalha and Cheng 2010)

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[28] Table 6.5.1.1.3B, High-limit shutoff control settings for air economizers, for fixed enthalpy, ASHRAE Standard 90.1-2010.

[29] A de-superheater is a device that is used to cool the superheated refrigerant gas coming from compressors before it enters the air-cooled condenser. The heat from the gases rejected to the de-superheater can then be used for space heating and service water heating purposes.

For air-cooled condensers implementing floating condenser pressure control, the condensing pressure in the system is controlled by a regulating valve in response to the heating capacity of the de-superheater. The regulating valve, which is known as the holdback valve, controls the discharge pressure from the condenser according the required capacity from the de-superheater. On the other hand, for condensers implementing fixed condenser pressure control, no hold-back valves need to be installed. Although this strategy is a potential energy saving measure, Minea in his study on using heat pumps for heat recovery in supermarket refrigeration systems pointed out that the electricity consumption of the refrigeration system at the higher condensing pressure set to improve heat recovery may be greater than the amount of useful heat recovered (Minea 2010).

In eQUEST-Refrigeration this measure was modeled by activating the refrigeration heat reclaim option in the specifications of HVAC units and by pointing to different discharge circuits from the compressor racks in the refrigeration system. Since the base-case was simulated with a fixed condenser pressure control, no holdback control option was simulated when implementing this measure. However, a holdback control option was simulated when opting for a floating pressure control option for the condenser (PG&E 2011). The holdback control was simulated as active only during periods when heat recovery (i.e. need for space heating) was required from the circuit. The holdback setpoint temperature was set at 95°F. A design temperature difference of 40°F was assumed between the reclaim condensing temperature and the design return air temperature.

6.6.4 Packaged Variable Air Volume Systems (PVAVS)

Packaged variable air volume (PVAV) systems with reheat were examined as an alternate to the constant air volume (CAV) systems used in the base-case building. In a typical PVAV system, the system responds to decreasing heat gain in the space by reducing the cold air supply to the space. However, to maintain suitable air quality in the space, it is necessary to set the cold air supply to a minimum quantity. Either reheat or baseboard radiation has to be used to offset the cooling effect of the minimum allowable air supply and to supply heat to offset losses. A schematic diagram of the system configuration is presented in Figure 6-6 below.

In the eQUEST-Refrigeration model, each thermal zone in the grocery store was installed with an individual PVAV system. The minimum supply air temperature was set at 55°F. In the cooling mode, the supply air volume was controlled each hour to adequately cool the zone. The minimum supply air flow was set at 30% of the zone peak air flow according to the requirements in ASHRAE Standard 90.1-2010[30]. In addition the standard requires the temperature of the supply air to be reset higher by 5°F under minimum cooling load conditions[31]. In eQUEST-Refrigeration this condition was modeled by setting the cooling coil temperature each hour to adequately cool the zone with the highest temperature, which in this case was set at 65°F.

When in heating mode, the PVAV system operates like a constant volume system with the supply air flow set at 30% of the zone peak air flow. Supply air temperature is varied by the means of a reheat coil. A reheat delta T of 20°F above space temperature setpoint was assumed according to the specifications in ASHRAE Standard 90.1-2010[32]. In the loading zones[33], heating capacity of the reheat coil had to be supplemented by additional heating from baseboard heaters.

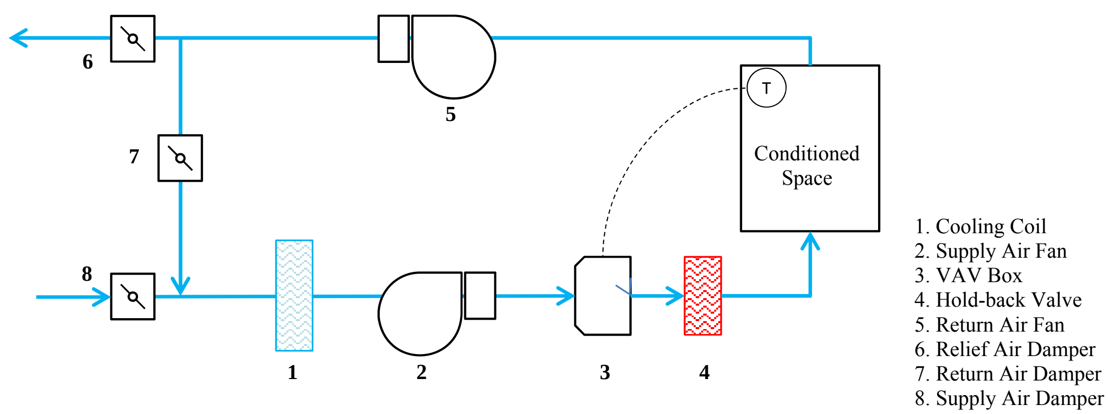

Figure 6-6: Schematic Diagram of the Packaged Variable Air Volume (Constant Temperature) System (Source: Birdsall et al., 1994)

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[30] Table 11.3.2A, Budget system description, note b: VAV with reheat, ASHRAE Standard 90.1-2010.

[31] Table 11.3.2A, Budget system description, note b: VAV with reheat, ASHRAE Standard 90.1-2010.

[32] Section 6.5.2.1.1, Supply air temperature reheat limit, ASHRAE Standard 90.1-2010. According to this section, when reheating is permitted , zones that have both supply and return/exhaust air openings greater than 6 ft above floor shall not supply heating air more than 20 F above the space temperature setpoint. In the eQUEST model the reheat temperature is modeled above supply air temperature.

[33] Loading zones in the grocery store model have high infiltration rates and require additional heating than what is provided by the reheat coil installed in the PVAV system.

6.6.5 Dedicated Outdoor Air System (DOAS)

The use of DOAS system has notable advantage over the conventional system implemented in the base-case model. Thornton et al. in their investigation on energy savings in small office buildings pointed out certain advantages of using DOAS over the conventional system, which include downsizing of zonal systems because loads from outside air are being met by DOAS; installation of a single ERV to pretreat the outdoor air instead of multiple ERVs; and zonal fans being run only to meet zonal heating and cooling loads (Thornton et al. 2010).

In this study, a dedicated outdoor air (DOAS) system was modeled in eQUEST-Refrigeration, which provided the outdoor ventilation requirements of the store. The method of modeling the DOAS system was adopted from a technical support document for 50% energy savings in small office buildings for reducing energy consumption in small offices (Thornton et al., 2010)[34]. The DOAS configuration implemented in this study includes an enthalpy wheel, a cooling coil, a heating coil and a supply fan as proposed by Thornton et al. A schematic diagram of the system configuration is presented in Figure 6-7 below.

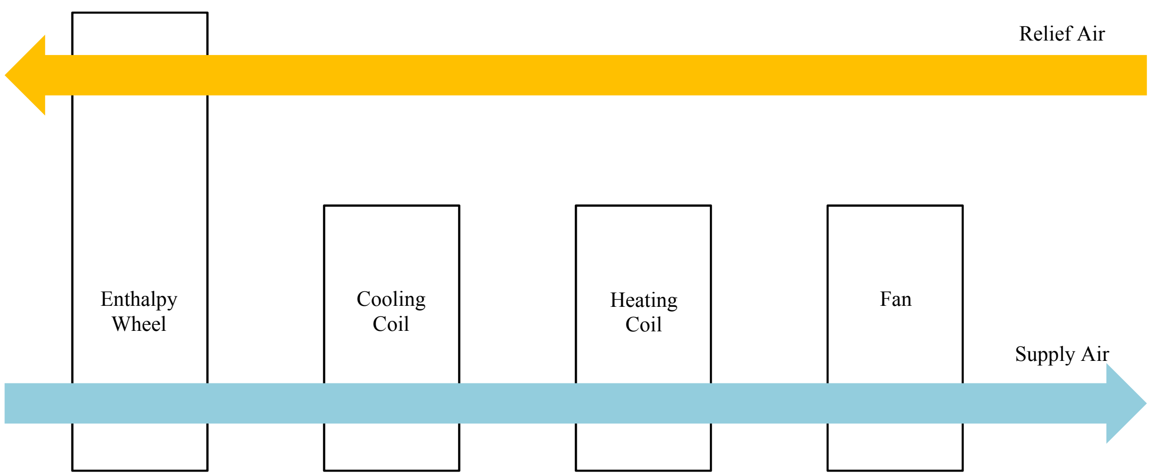

Figure 6-7: DOAS with Enthalpy Wheel, Conventional Cooling Coil and Heating Coil (Source: Thornton et al. 2010)

[34] The study by Thornton et al. assessed the DOAS using EnergyPlus. In order to assess the DOAS in eQUEST-Refrigeration certain other assumptions had to be made by this study in addition to the assumptions adopted from Thornton et al.

The DOAS unit was modeled as a single constant air volume unit with 100% outside air and a fixed outdoor airflow rate. The system was sized to handle all the outdoor air requirements of the grocery store model. A constant speed supply fan was implemented for the DOAS with a total supply static (external + internal) of 1.93 in. WG. This value incorporates the static pressure increase due to ducts required to provide outdoor air to different zones. The overall fan efficiency was set at 65%.

In order to extract humidity from moist outside air, the DOAS supply air temperature was maintained at 55°F while the supply air temperature of the other zones served by the DOAS was allowed to be maintained at 65°F. In addition, the outside air ventilation for all zones except the DOAS zone was shut-off.

According to ASHRAE Standard 90.1 – 2010, systems such as the DOAS with outside air intake equal to 100% have to be installed with energy recovery ventilation (ERV) systems[35]. To accomplish this, a sensible heat recovery system was modeled in eQUEST-Refrigeration to reclaim energy from the exhaust airflow. The runtime status of the heat recovery system was controlled by an outdoor temperature of 60°F. The energy savings from the implementation of heat recovery system were offset by increased fan energy requirements to overcome the additional static pressure of the device and the parasitic energy that was consumed by the system when it is operational. A parasitical power is assumed to be 50 W which was accounted for when the system was operational. Only sensible effectiveness of the ERV could be modeled. A sensible effectiveness of 75% for the ERV was assumed.

Condensing furnace with an efficiency of 95% was used for heating the outside air through the DOAS. The cooling efficiency of the DOAS was similar to that of the other cooling systems in the grocery store. An 11.5 EER was selected to represent the cooling efficiency of the DOAS. Certain issues have been encountered while modeling the DOAS in eQUEST. These are discussed below:

Issue 1: The DOAS was modeled using a dummy zone wherein the entire supply air consists of outside air. The dummy zone has zero heat transfer which is obtained by deleting all the surfaces.

Issue 2: The zone also has the entire supply air pushed through the exhaust in order to activate the heat transfer between the outside air and the relief air in the ERV system.

[35] Table 6.5.6.1, Energy recovery requirement, for % outdoor air at full design airflow rate at ≥ 80%, ASHRAE Standard 90.1-2010.

Issue 3: Internal heat loads were added to the dummy zone in order for this zone to see the return air temperatures that are approximately similar to that of the other zones in the grocery store. The resultant energy consumption due to the internal loads is assigned to a dummy meter that is read separately in the BEPS report.

Issue 4: Static pressure difference across the ERV could not be modeled and hence was added to the total static pressure of the supply fan.

6.6.6 Improving the Efficiency of Supply Air Fans, Exhaust Air Fans and Exhaust Fan Schedules

Two measures were considered in this category. In the first measure, the overall efficiency of supply air fans was improved from 55% as assumed in the base-case to 65% as described in a technical support document for 50% energy savings in small office buildings for reducing energy consumption in small offices (Thornton et al. 2010)[36].

In the second measure, the schedule of operation for the exhaust fan was changed from fans being set at ON at all the time to following the operation schedule of equipment installed in the "Bakery" zone of the grocery store (Bohlig and Fisher 2004)[37]. The schedule for implementing demand control ventilation for exhaust fans in the 'Bakery' zone is presented in Figure 6-8[38] below.

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[36] Fan efficiency can be improved by altering the fan design (i.e. design of fan blades), type of motor implemented as well as the operation mode of the fan (i.e. variable speed drive fans)..

[37] According to Bohlig and Fischer, state-of-the-art technologies are equipped with microprocessor-based controls with sensors that vary the fan speed based on cooking load and / or time of day. The demand control ventilation strategies include monitoring the energy input to cooking appliances, monitoring the energy and effluent output from cooking appliances, installation of time clocks and manual operation of appliances that in turn are coupled with the operation of exhaust system.

[38] The numbers for this graph are provided in Table B-30 of Appendix-B in this report.

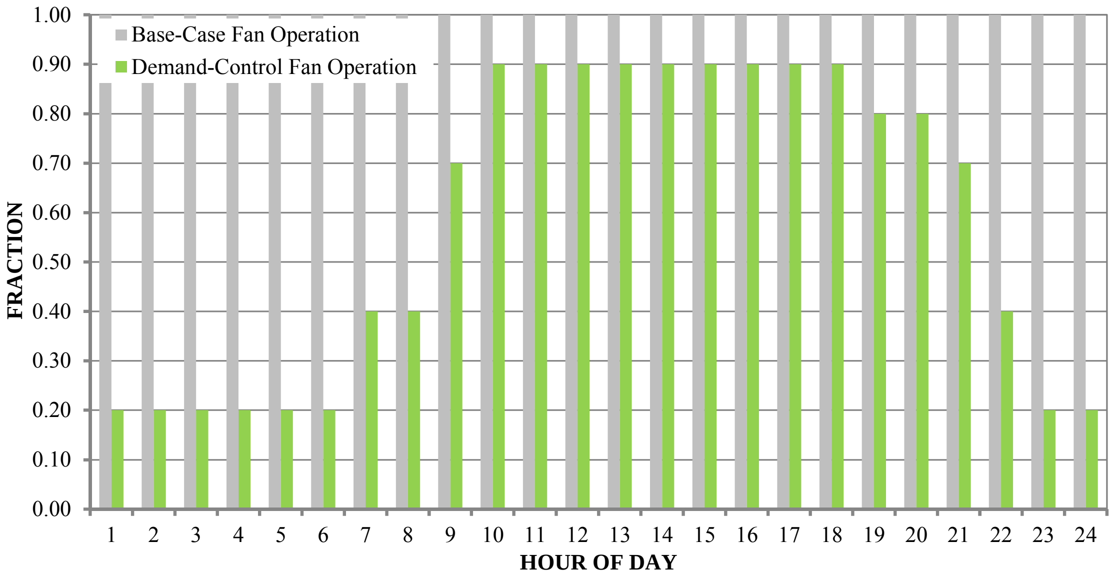

Figure 6-8: Schedule for Exhaust Fan in 'Bakery' Zone Implementing Demand Control Ventilation Strategy

6.6.7 Demand Control Ventilation (DCV)

Demand Control Ventilation modulates the amount of outdoor ventilation in response to the actual occupancy in a zone as it varies throughout the day. DCV can be accomplished by the use of CO_2 sensors that measure the changes in CO_2 concentration in occupied space.

eQUEST-Refrigeration Version 3.61 does not have the capability to directly model DCV for the HVAC system. Hence, in order to assess the impact of implementing the DCV system, a minimum outside air ratio was calculated for each zone of the grocery store by dividing the outdoor airflow requirements by supply air requirements for each space. Finally, a minimum air schedule was provided for each zone based on the observed occupancy schedules of the different zones in the grocery store. The schedule varies the minimum outside air requirements for each zone on an hourly basis depending on the hourly variation in the number of occupants in each zone. The sample schedule implementing DCV strategy in the 'General Merchandise' zone is presented in Figure 6-9 below. Efficiency measures for HVAC systems are summarized in Table 6-4[39].

[39] The numbers for this graph are provided in Table B-31 of Appendix-B in this report.

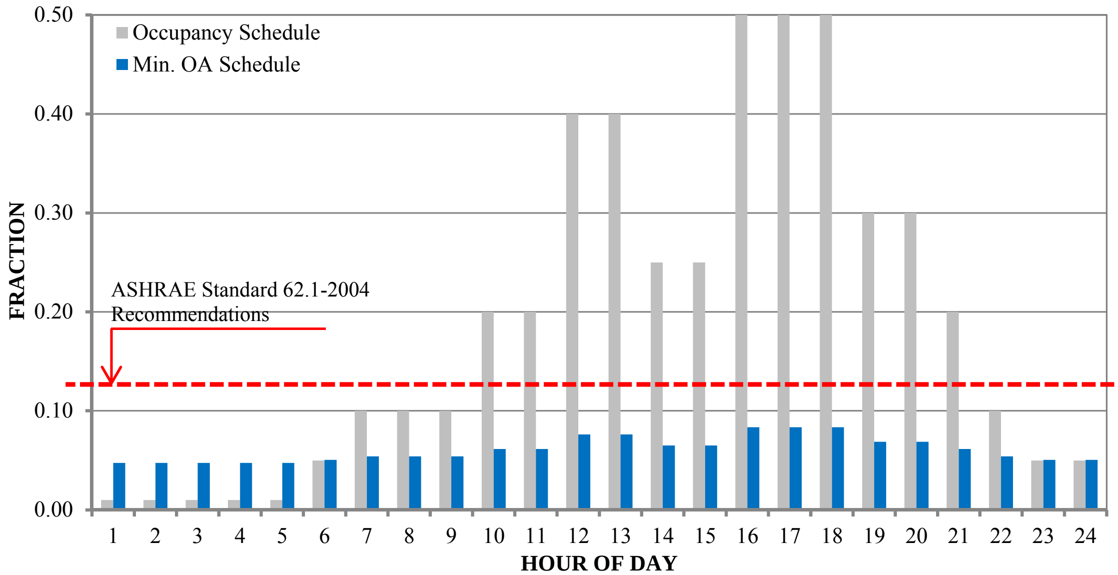

Figure 6-9: Sample Schedule for Demand Control Ventilation in "General Merchandise" Zone

6.7 Efficiency Measures for the Service Hot Water Heating Systems

Energy consumed by service hot water systems constitutes a small portion of the total energy consumption in the grocery store. Only one measure was considered by this analysis.

6.7.1 Efficient Gas Water Heaters

The efficiency of gas service water heaters was improved to 95% from 78% as assumed in the base-case model. The improved efficiency was obtained from the information on appropriate products listed in the Air-Conditioning, Heating and Refrigeration Institute (AHRI) directory of certified products (AHRI 2012). This efficiency measure for the gas water heaters is summarized in Table 6-5.

Table 6-4: Energy Efficiency Measure for HVAC Systems

EEM No.	Base-Case Parameters	Units	Efficiency Measure Description	Reference
HVAC SYSTEMS				
16	**System Specifications**			
	Cooling Efficiency	-	Improving COP by 20%	Hale et al., 2008 Leach et al., 2009
17	Heating Efficieny	-	Improving AFUE to 92%	US EPA, 2008a
18	Economizer	-	Installation of economizers Enthalpy economizers h_{OA} > 28 Btu/lb	AEDG, 2011 ASHRAE 90.1-2010
19	Heat Recovery from Refrigeration Coils	-	Reclaiming heat from refrigeration system compressors for space heating	Hirsch, 2006 Sawalha and Chen, 2010 PG&E, 2011
21	Packaged Variable Air Volume Systems		Installing Packaged Variable Volume (PVAV) system	AEDG, 2008 Thornton et al., 2010
22	Dedicated Outdoor Air System	-	Installing Dedicated Outdoor Air System	
23	**Supply Fan Specifications**			
	Total Efficiency	%	Improved to 65%	Thornton et al., 2010
25	**Exhaust Fan Specifications**			
	Exhaust Fan Schedule	-	Demand controlled exhaust Modified schedule provided in Table B-30	Bohlig and Fisher 2004
26	**Ventillation Requirements**			
	Demand Controlled Ventilation	-	Installation of CO_2 sensors Resultant schedule provided in Table B-31	Hale et al., 2008 Thornton et al., 2011 AEDG, 2011

Table 6-5: Energy Efficiency Measure for Service Hot Water Systems

EEM No.	Base-Case Parameters	Units	Efficiency Measure Description	Reference
SERVICE HOT WATER SYSTEMS				
	Hot Water Heater / Boiler			
27	Efficient Gas Heaters (For 119 & 40 gallon heaters)	E_t % SL Btu/hr	E_t = 95% for both heaters For 119 gallon tank: SL = 1020 For 40 gallon tank: SL = 410	AEDG, 2011 AEDG, 2008 AHRI, 2012

6.8 Efficiency Measures for the Refrigeration Systems

Energy consumption from refrigeration systems forms the largest end-use representing nearly 31.4% of the overall end-use energy consumption in the grocery store. Numerous measures can be adopted to reduce the resultant energy consumption. However, the selection of measures examined this study is restricted those that can be modeled in eQUEST Refrigeration. The measures address the performance of the various components of the refrigeration systems, which include measures for the compressors, condensers and evaporators (display cases). Efficiency measures for compressors include:

- Implementing floating suction pressures, and
- Implementing compressor capacity control,

Efficiency measures for display cases include:

- Installation of glass doors on all cases,
- Installation of vacuum insulated panels and doors to reduce conduction loads,
- Improving efficiency of evaporator fans,
- Introducing hot gas defrost for low temperature cases and implementing demand based defrost control,
- Using hot-gas from compressors to operate anti-sweat heaters and implementing humidity control for anti-sweat heaters,
- Allocating a fraction of conductive and infiltrative zone heat losses directly to return air of the HVAC system, and
- Switching to high efficiency lighting in display cases and implementing motion sensors to reduce lighting loads in display cases.

Efficiency measures for condensers include:

- Implementing load reset condenser head pressure control,
- Implementation of mechanical subcooling, and
- Implementation of high efficiency fan motors.

6.8.1 Implementing Floating Suction Pressures in Compressor Racks

In the base-case model the refrigeration compressor racks were modeled with fixed suction pressure control. This control maintained a constant pressure setpoint for each compressor rack, which was minimum suction pressure required meeting the maximum fixture cooling loads or peak loads for walk-in refrigerators (PG&E 2011). In eQUEST-Refrigeration this fixed suction pressure in compressor racks was modeled by setting the control strategy for compressor racks to 'FIXED'. The fixed temperature setpoint was determined by suction temperature of the compressor rack[40].

In order to improve the performance of compressors in the refrigeration system the suction pressure control for each of the compressor racks was changed from fixed suction pressure to floating suction pressure. In eQUEST-Refrigeration the temperature control for compressor racks was changed from 'FIXED' mode to 'LOAD-RESET' mode. Changing to 'LOAD-RESET' control allows the suction temperature setpoint to vary as a function of the worst case demand on the compressor group thus saving energy at lower loads (Hirsch 2006). Energy savings were obtained from operating the compressors at higher suction temperatures on an average by reducing the lift and resultant compressor power (PG&E 2011). However, the suction temperature was restricted to a specified temperature range. The minimum allowed suction temperature was assumed to be the base-case suction temperature setpoint, with a maximum float of 5°F (Hirsch 2006, PG&E 2011).

6.8.2 Implementing Compressor Capacity Control

The base case refrigeration system does not allow for the compressors to have any capacity control mechanism. To modulate the total capacity of the compressor rack, the suction group cycles individual compressors depending on the load from the corresponding evaporators (Hirsch 2006). In eQUEST-Refrigeration the capacity control for each compressor in the base-case model is set at 'NONE'.

Setting the capacity control to 'EXTERNAL-SIGNAL' in eQUEST-Refrigeration enables the compressors to have a capacity control mechanism such as cylinder unloaders or a variable speed drive that are controlled by the parent suction group (Hirsch 2006)[41]. This setting enables the compressors to be modulated to meet a range of loads before being cycled off by the parent suction group[42]. A linear unloading curve was implemented for the compressors[43,44]

[40] Specifications for the suction temperatures in each of the compressor racks modeled in the base-case can be found in Table B-12, Appendix B.
[41] eQUEST-Refrigeration does not require specific input for capacity control mechanisms such as cylinder unloaders or speed. The effect of these devices in included in the capacity performance curves (Hirsch 2008).
[42] Default values presented in eQUEST-Refrigeration were utilized to simulate these control mechanisms. These include setting the minimum capacity of compressors to 20% before letting the suction group cycle off the compressors.
[43] Part load operating conditions for reciprocating compressors.

(Manske et al., 2000). In addition, to take advantage of the improved modulating mechanisms, the throttling temperature range of suction groups was lowered[45] from 10°F as set in the base-case model to 5°F.

6.8.3 Installing Glass Doors on All Display Cases

As an efficiency measure for refrigerated display cases, glass covers and doors were installed on all open faced refrigerated display cases in the base-case model. The installation of doors modifies the interaction the display case has with the thermal zone in which the display-case is located. Corresponding conduction and infiltration loads of these refrigerated display cases were modified accordingly to account for the installation of doors.

For the base-case model the loads from open faced display cases are documented in Table B-12, Appendix B. These loads were divided to represent loads from infiltration, conduction and internal thermal loads[46] (Walker et al. 2004). In the base-case model for open vertical display cases 80% of the loads were allocated to infiltration and 3% of the loads were allocated to conduction. On the installation of doors for these display cases, the loads of the cases were reduced by 68% (Faramarzi et al., 2002). In addition, the percentage of load allocation in these cases was changed with 3% of the loads being allocated to infiltration loads and 80% of these loads being allocated to conduction (Walker et al., 2004).

6.8.4 Installing Vacuum Insulated Panels and Doors

In another efficiency measure for refrigerated display cases, the heat losses due to conduction were reduced by installing vacuum insulated panels and "low heat" doors that use triple pane glazing with insulating gases encased between the panels (Goetzler et al. 2009). In a study on the performance of energy efficient display case refrigerator Tao et al. (2004) determined that the installation of such panels and doors reduced the conduction loads in refrigerated cases by 20%. The impact of improved insulation for display cases could only be approximately modeled in the eQUEST-Refrigeration simulation program. Accordingly, the conduction loads of refrigerated cases specified in the simulation model were reduced by 20%.

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[44] As pointed out by the authors, the linear part-load characteristics of the reciprocating compressors does not pass through the origin because of the additional compressor power requirements to overcome the parasitic losses (approximately 3%) associated with the unloading of cylinders.
[45] With the installation of modulation mechanisms, smaller throttling range can be used without short cycling the compressors.
[46] Internal thermal loads include the use of lights, evaporator fans, periodic defrosts and anti-sweat heaters.

6.8.5 Improving Efficiency of Evaporator Fans

For display cases, the evaporator fan efficiency can be improved by the installation of electronically commutated motors (ECM) (Karas 2006, Goetzler et al. 2009). According to the study on evaporator fans by Karas (2006) the evaporator fan power is reduced by 67% by means of installing ECMs. Modeling improved efficiency of evaporator fans in the eQUEST-Refrigeration model is at best approximated by reducing the fan power consumption of the evaporator fans for each display case. Standard shaded pole (SP) motors were assumed to be used in the base-case model. Power consumption of evaporator fans used in the base-case model have been documented in Table B-14, Appendix B.

The performance of variable speed drives installed for evaporator fans in walk-in refrigerators was also evaluated. The measure assumes the modulation of the speed of the walk-in air unit fans as the primary means of temperature control (PG&E 2011). In the eQUEST-Refrigeration model this option was modeled by implanting fan control at 'SPEED'[47]. This setting implies that the fan was controlled by varying the speed of the fan motor. No fan control was assumed in the base-case model. The fan control for the walk-in coolers, freezers and preparation room of the base-case model was set at 'CONSTANT VOLUME'.

6.8.6 Improving Defrost Methods and Schedules

As pointed out in the literature review the hot-gas involves diverting hot refrigerant gases from the compressor for defrosting refrigerated display cases. The implementation of hot-gas defrost can potentially save energy. A schematic diagram of the system configuration is presented in Figure 6-10 below.

Electric defrost was implemented for low temperature display cases in the base-case model. Defrost specifications for these cases are provided in Table B-13, Appendix B. As an efficiency strategy the method of defrost for low temperature display cases was changed from electric to hot-gas method. This option was modeled by selecting "HOT GAS" instead of "ELECTRIC" in eQUEST-Refrigeration[48,49]. The discharge circuit from which the hot gas used for defrost also had to be specified.

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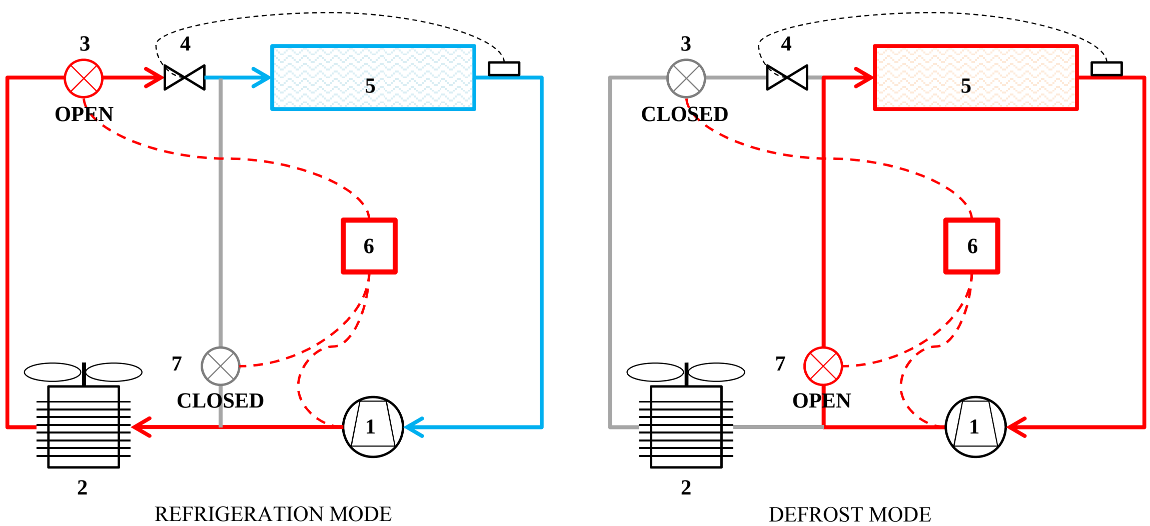
[47] Assumptions for the variation in the fan speed were adopted from the default values provided in eQUEST-Refrigeration.

[48] For this selection to be effectively modeled in eQUEST-Refrigeration, it had to be ensured that there was more than one low-temperature display case fixtures were attached to the discharge circuit of the compressor rack. It also had to be ensured that the display cases attached to the same discharge circuit had relatively non-coincident defrost cycles.

In addition, defrost initiation settings were changed as an efficiency strategy. The defrost initiation control was changed from ‘TIME’, which was assumed in the base-case building to ‘DEMAND’ indicating a demand-based defrost initiation schedule. By specifying the defrost cycle to be initiated on demand, the defrost cycle initiates when the amount of ice on the evaporator exceeds the maximum defrost limit, which in turn is dictated by the number of defrosts per day and the evaporator design entering and leaving conditions (Hirsch 2006). The specifications for the maximum frost are left to be defaulted by the simulation program.

6.8.7 Improving Anti-Sweat Heaters

Two measures were considered to improve the performance of anti-sweat heaters implemented in the base-case model. In the first measure the implementation of hot-gas operated anti-sweat heaters was considered. In the second measure the use of relative humidity to activate anti-sweat heaters was considered.



1. Compressor
2. Condenser
3. Solenoid Valve 1
4. TXV
5. Evaporator
6. Defrost Clock
7. Solenoid Valve 2

Figure 6-10: Schematic Diagram of the Hot-Gas Bypass from Compressor to Evaporator

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[49] It should be noted that unlike the case of electric defrost where the fans continue to operate, the fans in hot-gas defrost are defaulted to stay off during the defrost periods, hence contributing to the energy savings that can be attained from the implementation of this measure.

Electric resistance anti-sweat heaters are currently implemented for low-temperature display cases in the base-case model. The power consumption of the installed anti-sweat heaters is provided in Table B-14, Appendix B. As an efficiency strategy the use of hot gases coming off the refrigeration compressors was considered for use in the anti-sweat heaters of the base-case model. The use of hot-gas operated anti-sweat heaters makes use of the free energy available from the refrigerant gas thus saving electricity, which is currently being spent in the base-case model to operate the electric resistance anti-sweat heaters. Since it is not possible to model this measure in eQUEST-Refrigeration an alternate method of calculating savings was used to evaluate the impact of using hot-gas operated anti-sweat heaters. Anti-sweat heaters powered by hot-gas do not have any electricity consumption requirements. However, the procedure assumed that the impact of hot gas anti-sweat heaters on evaporator and zone loads would be the same as that imposed by electric anti-sweat heaters. Hence, in order to assess this measure the electricity consumption of the electric resistance anti-sweat heaters per door / unit length of display cases was manually calculated on an annual basis and then subtracted from the base-case annual electricity consumption[50].

In the second measure for improving the performance of anti-sweat heaters, the use of humidity levels to activate the heaters was considered. In eQUEST-Refrigeration this can be modeled by setting the heater control option to 'RELATIVE-HUMIDITY' to specify the anti-sweat heater controls to the relative humidity levels of the adjacent space[51]. The maximum setpoint of relative humidity at which the heater was at full output was specified to be 70% and the minimum setpoint of relative humidity was specified to be 45% (Hussman 2012)[52]. In the base-case model the anti-sweat heaters were operated on a schedule, which required the heaters to be on all the time.

6.8.8 Capturing Cold Air Spills from Open Display Cases

As noted in the literature review, open faced refrigerated display cases are associated with cold air spillage that tends to accumulate at the bottom of the display cases causing what is known as the cold aisle effect. A study by Pitzer and Malone has pointed out that this cold air can be captured and reused, hence lowering energy consumption of HVAC system in the grocery

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[50] The calculations included the pulsating effect of the anti-sweat heaters as discussed in Chapter 6 of this study.
[51] The specifications for relative humidity pertain to the zone in which the display case is located.
[52] These numbers are adapted from manufactures' specifications for Door Anti-Sweat Heater (DASH) controls by Hussman (Hussman 2012).

store. This study assesses the impact of implementing this measure on the overall energy consumption of the store. This measure was simulated in eQUEST-Refrigeration by specifying the fraction of conductive and infiltration zone heat losses that are allocated to each display case directly to the HVACs return air rather than impacting the zone. This was accomplished by manipulating the 'FRAC-TO-RETURN' parameter for open display cases. According to the eQUEST-Refrigeration users' manual the defaulted value is 0 which implies that none of the conductive infiltration zone heat losses go directly to the return, all losses impact the zone. This value was changed to 1 to approximate the effect of ducting the return air from under and behind the display cases (Hirsch et al., 2006). A schematic diagram of the system configuration is presented in Figure 6-11 below.

6.8.9 Improving Lighting Efficiency for Display Cases

As pointed out in the literature review display case lighting can be a major source of energy consumption in grocery stores. Efficiency measures that were evaluated by this study include the use of LED lighting, fiber-optic lighting systems and assessing the impact of installing motion sensors to control lighting in display cases. The base-case model implements T-8 fluorescent lamps. The lights are scheduled to be turned off to 75% of the full output during unoccupied hours of the store. The power consumption of display case lighting is provided in Table B-14, Appendix B.

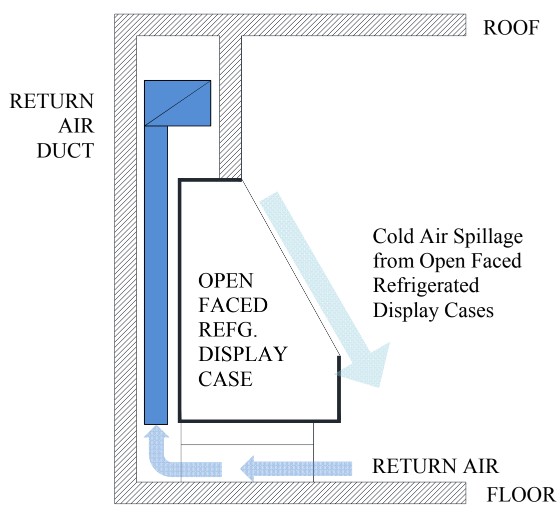


Figure 6-11: Schematic Sectional View of Open Faced Refrigerated Display Case with Under-Case Return-Air Path

For the first measure, in order to model the LED lighting, the lighting energy consumption of the display cases was reduced by 60% from the base-case display case lighting system. The percentage reduction was obtained from manufacturer's literature (GE Lighting 2009).

For the second measure, the performance of fiber-optic lighting systems was evaluated. This measure was modeled using the same energy consumption as that of LED lighting. However, in this case the lamps were placed outside the refrigerated display cabinet. In order to model this strategy all the lighting loads on the refrigeration system were removed from the display cases and added as an internal load to the space in which the display cases were modeled.

For the third measure involving lighting power reductions in display cases, motion sensors were modeled. Implementing this measure enables the lighted in the display cases to turn on when detecting movement. The performance of motion sensors were modeled using the a ratio based on occupancy schedules of the base-case grocery store.

6.8.10 Implementing Floating Condenser Head Pressure

The refrigeration condensers modeled in the base-case simulation model were air-cooled with a fixed head pressure control. In order to improve the efficiency of the refrigeration condensers, floating head pressure controls[53] were evaluated.

The assumptions implemented in the base-case to model this strategy were adopted from the report on performance of efficiency strategies in supermarkets by Pacific Gas and Electric Company (PG&E 2011) and with some modifications. According to the report, this strategy of control required controls to float the saturated condenser temperature (SCT) of the refrigeration system to 70°F during low-ambient temperature conditions with an ambient temperature following control logic and variable speed condenser fans. The ambient following control logic set the target SCT by adding a fixed control temperature difference (TD) to the ambient temperature (dry-bulb temperature difference for air cooled condensers). The TD for condensers serving low temperature refrigeration systems was assumed to be 8°F and for condensers serving medium temperature refrigeration systems was assumed to be 10°F. The condenser fan speeds were continuously adjusted to maintain the target SCT, with an override minimum SCT of 70°F and an override maximum of 95°F in hot climate.

[53] This strategy allows the condensing pressure of the refrigerant to float with low ambient conditions instead of retaining the condensing pressure at a fixed set point as seen in the fixed head control strategy.

In the eQUEST-Refrigeration model implemented by this study, floating head pressure control for condensers was modeled setting the SCT control strategy of the refrigeration rack to 'LOAD-RESET'. Variable speed condenser fans were modeled as per recommendations of the PG&E report as well as TDs for low temperature and medium temperature condensers. However, the recommendations in the PG&E report were for Californian climates. Hence, in this study the override minimum SCT of 75°F and an override maximum of 115°F were modeled instead.

6.8.11 Improving Condenser Fan Efficiency

Improvement of condenser fan efficiency was also considered as a potential efficiency measure. The base-case simulation model in this study implements the use of ECM motors by reducing the fan power by 67% as recommended by Karas (Karas, 2006). In the eQUEST-Refrigeration model implemented in this study, the condenser fan power was modified by altering the input to the electric input ratio[54] of the condenser fan.

6.8.12 Implementing Mechanical Sub-Cooling

As pointed out in the literature review, mechanical sub-cooling involves the cooling of liquid refrigerant after it has been condensed. This is usually accomplished by either the medium temperature suction group or by the installation of a separate refrigeration system dedicated for this purpose. In this study the mechanical sub-cooling was modeled as an efficiency measure by installing a separate refrigeration system operating at medium temperature conditions, which provided sub-cooling to both medium and low temperature suction groups installed in the base-case model. To accomplish this, the design temperature for the evaporator of this sub-cooler was modeled at 40°F, while the design temperature for the sub-cooler condenser was modeled at 115°F. A schematic diagram of the mechanical sub-cooler is presented in Figure 6-12 below. In the eQUEST-Refrigeration model of the sub-cooler, the condenser and compressor capacities of the sub-cooler were defaulted to eQUEST defaults. The efficiency measures for the refrigeration systems are summarized in Table 6-6 below.

[54] This variable can be defined as the ratio of the fan power to the nominal capacity for the condenser fan.

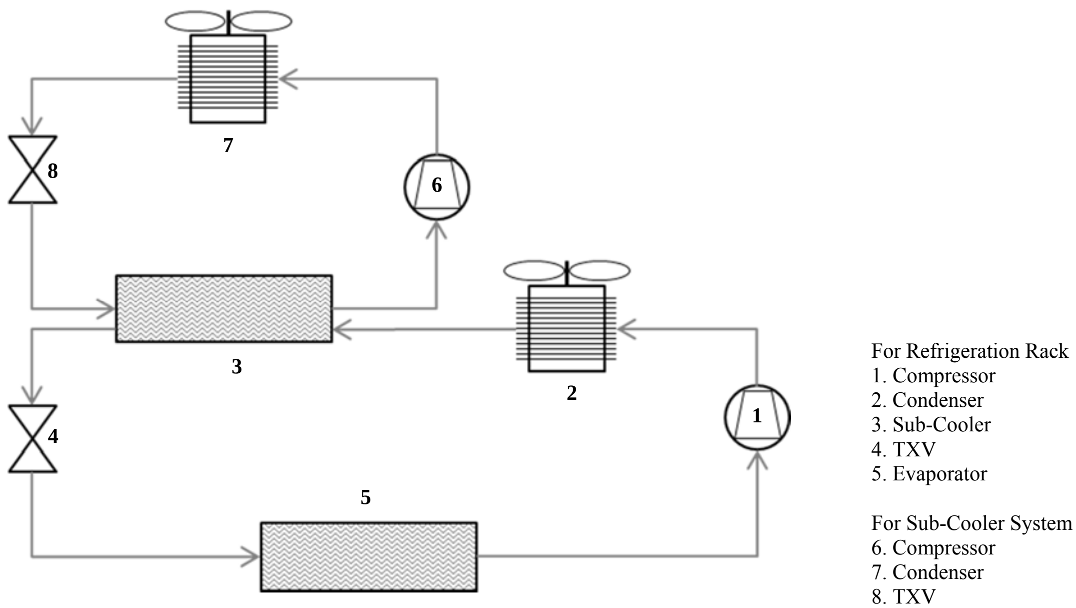


Figure 6-12: Schematic Diagram for a Dedicated Sub-Cooling Refrigeration System (Source: Thornton 1991)

Table 6-6: Energy Efficiency Measure for Refrigeration Systems

EEM No.	Base-Case Parameters	Units	Efficiency Measure Description	Reference
REFRIGERATION SYSTEM				
	Compressors			
28	Suction Group Temperature Control	-	Changed from "FIXED" to "LOAD-RESET" in eQUEST-Refrigeration	PG&E, 2010
29	Compressor Capacity Control	-	Changed from NONE to EXT-SIGNAL in eQUEST-Refrigeration Part load curve for power cunsumption x=y	Goetzler et al., 2009 Energy Center of Wisconsin, 2001

Table 6-6: Continued

EEM No.	Base-Case Parameters	Units	Efficiency Measure Description	Reference
REFRIGERATION SYSTEM				
	Display Cases			
31	Type	-	Glass doors on all cases	Leach et al., 2009
32	Reduced Conduction	-	Vaccuum insulated panels Reduced conduction loads by 20%	Goetzlet et al., 2009 Tao et al., 2004
			Implementing low heat doors	Goetzler et al., 2009
35	Fans		Using Electronically Commutated Motors (ECM) Base-case fan power reduced by 67% in eQUEST-Refrg.	Leach et al., 2009 Goetzlet et al., 2009 Karas, 2006
36	Defrost Type		Low-temp. cases: Hot-gas defrost	Leach et al., 2009 Goetzler et al., 2009
36 + 37	Defrost Control		Initiation: Demand	Fricke and Sharma, 2011
38	Anti-Sweat Heater Type		Use of hot-gas from compressors to run anti-sweat heaters. Electricity reduction due to installation of hot-gas anti-sweat heaters = 713 MMBtu/year	Goetzler et al., 2009
39	Anti-Sweat Heater Control		Using humidity levels to activate anti-sweat heaters	Leach et al., 2009 Hussmann, 2012
40	Direct Return		Allocating a fraction of conductive and infiltrative zone heat losses directly to the return air of the HVAC system.	Pitzer and Malone, 2005
41	Lighting	kW/ft kW/door	Switching to high efficiency fluorescent lighting Reducing LPD by 10% in eQUEST-Refrg.	Goetzler et al., 2009
42			Switching over to LED Reducing LPD by 60% in eQUEST-Refrg.	Leach et al., 2009 GE Lighting, 2009
47		kW	Fiber optic lighting	Goetzler et al., 2009
43	Lighting Schedule	-	Motion / occupancy sensors	Leach et al., 2009
	Condensers			
44	Control	-	Change from FIXED to LOAD-RESET head pressure control in eQUEST-Refrg. Changed from CYCLE-FAN to VARIABLE-SPEED mode in eQUEST-Refrg.	Goetzler et al.,2009 VaCom Engineering,2007
45	Fan Motor	-	High efficiency Electronically commutated motor (ECM) Base-case fan power reduced by 67% in eQUEST-Refrg.	Goetzler et al., 2009 Karas, 2006
46	Subcooling Effect	-	Change from ambient subcooling to mechanical subcooling	Thornton, 1991

6.9 Results

6.9.1 Results from the Implementation of EEMs for the Building Envelope

As expected, almost all the efficiency measures for the building envelope in the grocery store did not provide any substantial energy savings in terms of annual energy consumption. Almost all the strategies in this category provided savings within one percent. A notable difference was seen in the implementation of strategies to reduce infiltration. Energy savings from EEMs for the building envelope are presented in Table 6-7 and Figure 6-13 below.

When considering site energy consumption:

- Within 1% savings are achieved with the implementation of improved insulation for walls and roofs; freezer walls and roofs, and improved specifications for skylight.
- Negative savings are observed on the implementation of high-albedo roofs and increasing the area of skylights.
- A savings of 3.10% in annual energy consumption was seen on reducing infiltration values.

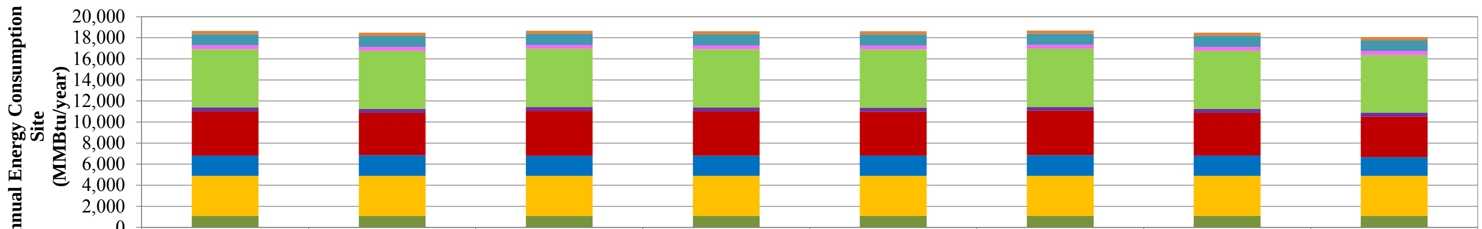
When considering source energy consumption:

- Within 1% savings are achieved with the implementation of most of the efficiency measures.
- Negative savings are observed on the increasing the area of skylights.
- Reduced infiltration rates provided a saving of 2.34%.

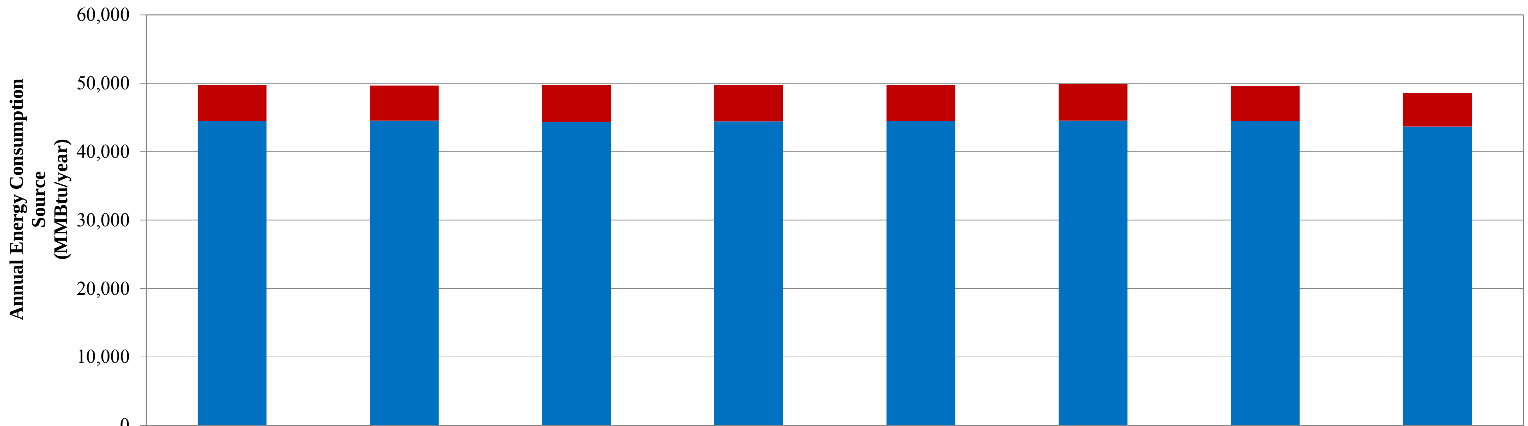
Table 6-7: Results from the Implementation of EEM's for the Building Envelope

		SITE ENERGY (MMBtu/yr)				SOURCE ENERGY (MMBtu/yr)	
		Elec.	Nat. Gas	Total	% Above BC	Total	% Above BC
BASE-CASE		13,809	4,828	18,637	-	49,776	-
EEMS							
1	Improved Insulation for Ext. Walls and Roof	13,832	4,653	18,485	0.82	49,658	0.24
2	Implemented High-Albedo Roof	13,774	4,897	18,671	-0.18	49,740	0.07
3	Improved Insulation for Freezer Wall and Roof	13,795	4,828	18,623	0.08	49,730	0.09
4	Improved Insulation for Loading Dock Doors	13,803	4,811	18,614	0.12	49,738	0.08
5	Increased Area of Skylights	13,832	4,854	18,686	-0.26	49,878	-0.20
6	Improved Specifications for Skylights	13,807	4,681	18,488	0.80	49,607	0.34
7	Reduced Infiltration Rates	13,560	4,499	18,059	3.10	48,613	2.34

Note: To assess the energy consumption at source levels, a conversion factor of 3.15 for electricity and 1.1 for natural gas was used. The site to source energy conversion for electricity also accounts for the 7% transmission losses.



	Run 0	Run 1	Run 2	Run 3	Run 4	Run 5	Run 6	Run 7
Total	18,637	18,485	18,671	18,623	18,614	18,686	18,488	18,059
Ext. Light. (E)	312	312	312	312	312	312	312	312
Pumps & Fans (E)	1,047	1,045	1,045	1,043	1,047	1,047	1,047	1,016
SHW (E+G)	355	355	355	355	355	355	355	355
Refg. (E)	5,539	5,540	5,539	5,539	5,539	5,540	5,539	5,482
Heat Reject. (E)	385	385	385	385	385	385	385	378
Space Heat. (G)	4,180	4,005	4,249	4,180	4,163	4,206	4,034	3,851
Space Cool. (E)	1,927	1,951	1,894	1,916	1,921	1,949	1,924	1,773
Lighting (E)	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822
Misc. Equip. (E+G)	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069



	Run 0	Run 1	Run 2	Run 3	Run 4	Run 5	Run 6	Run 7
Natural Gas	5,311	5,119	5,387	5,311	5,292	5,339	5,150	4,949
Electricity	44,465	44,540	44,354	44,419	44,445	44,538	44,457	43,664

Figure 6-13: Annual Site and Source Energy Consumption of EEMs for the Building Envelope

6.9.2 Results from the Implementation of EEMs for Lighting and Equipment

Efficiency measures for lighting and equipment fared reasonably well in terms of annual energy savings. Energy savings from EEMs for lighting and equipment are presented in Table 6-8 and Figure 6-14 below.

When considering site energy consumption:

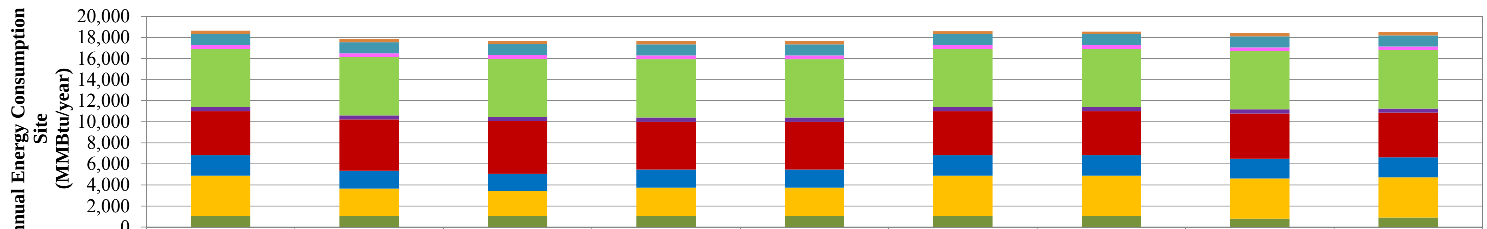
- Reducing the lighting power density of the ambient lights in the grocery store provided a savings of 4.22%. Implementing time switches and occupancy sensors in addition to reducing the lighting power density further increased the savings to 5.11%.
- Implementing daylighting controls saved 5.29% in annual energy consumption. Changing the reflectance of the interior surfaces in addition to daylighting controls did not make any substantial change to the energy savings.
- Within 1% savings was achieved on reducing the LPD of the façade lighting and parking lighting.
- Reducing equipment loads by means of installing energy efficiency equipment in the store provided a modest saving of 1.16%.
- While improving the schedule of equipment operation provided a saving within 1%.

When considering source energy consumption:

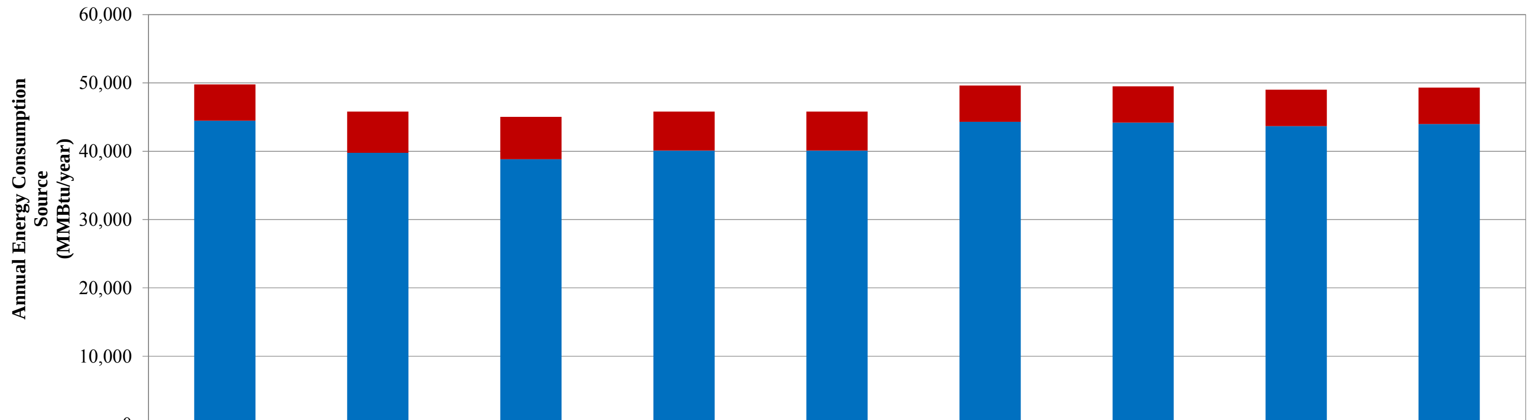
- Reducing the lighting power density of the ambient lights in the grocery store provided a savings of 7.96%. Implementing time switches and occupancy sensors in addition to reducing the lighting power density further increased the savings to 9.55%.
- Implementing daylighting controls saved 7.95% in annual energy consumption. Changing the reflectance of the interior surfaces in addition to daylighting controls did not make any substantial change to the energy savings.
- Less than 1% savings was achieved on reducing the LPD of the façade lighting and 0.55% savings was achieved on reducing the LPD of the parking lights.
- Reducing equipment loads by means of installing energy efficiency equipment in the store provided a modest saving of 1.54%.
- While improving the schedule of equipment operation provided savings within 1%.

Table 6-8: Results from the Implementation of EEM's for Lighting and Miscellaneous Equipment

		SITE ENERGY (MMBtu/yr)				SOURCE ENERGY (MMBtu/yr)	
		Elec.	Nat. Gas	Total	% Above BC	Total	% Above BC
BASE-CASE		13,809	4,828	18,637	-	49,776	-
EEMS							
8	Reduced Lighting Power Density	12,348	5,502	17,850	4.22	45,813	7.96
8,9	Implemented Time Switches & Occ. Sensors	12,061	5,624	17,685	5.11	45,023	9.55
10	Implemented Daylighting Controls	12,455	5,195	17,651	5.29	45,821	7.95
10,11	Increased Ceiling Surface Reflectance	12,455	5,196	17,651	5.29	45,821	7.95
12	Reduced Ext. Façade Lighting	13,653	4,828	18,589	0.26	49,622	0.31
13	Reduced Ext Parking Lighting Power	13,832	4,828	18,552	0.46	49,503	0.55
14	Reduced Equipment Power Density	13,560	4,861	18,421	1.16	49,011	1.54
15	Improved Equipment Schedule	13,654	4,852	18,505	0.71	49,302	0.95



	Run 0	Run 8	Run 8 + 9	Run 10	Run 10 + 11	Run 12	Run 13	Run 14	Run 15
Total	18,637	17,850	17,685	17,651	17,651	18,589	18,552	18,421	18,505
Ext. Light. (E)	312	312	312	312	312	264	227	312	312
Pumps & Fans (E)	1,047	1,046	1,046	1,047	1,047	1,047	1,047	1,046	1,047
SHW (E+G)	355	355	355	355	355	355	355	355	355
Refg. (E)	5,539	5,538	5,538	5,537	5,537	5,539	5,539	5,539	5,539
Heat Reject. (E)	385	384	383	385	385	385	385	385	385
Space Heat. (G)	4,180	4,854	4,976	4,547	4,548	4,180	4,180	4,287	4,250
Space Cool. (E)	1,927	1,707	1,654	1,721	1,720	1,927	1,927	1,872	1,893
Lighting (E)	3,822	2,585	2,351	2,678	2,678	3,822	3,822	3,822	3,822
Misc. Equip. (E+G)	1,069	1,069	1,069	1,069	1,069	1,069	1,069	802	900



	Run 0	Run 8	Run 8 + 9	Run 10	Run 10 + 11	Run 12	Run 13	Run 14	Run 15
Natural Gas	5,311	6,052	6,186	5,715	5,716	5,311	5,311	5,347	5,337
Electricity	44,465	39,761	38,836	40,107	40,106	44,311	44,192	43,664	43,965

Figure 6-14: Annual Site and Source Energy Consumption of EEMs Implemented for the Building Lighting

6.9.3 Results from the Implementation of EEMs for HVAC Systems and Service Hot Water Systems

Efficiency measures for the building HVAC system also provided good savings in terms of annual energy consumption. Energy savings from EEMs for lighting and equipment are presented in Table 6-9 and Figure 6-15 below.

When considering site energy consumption:

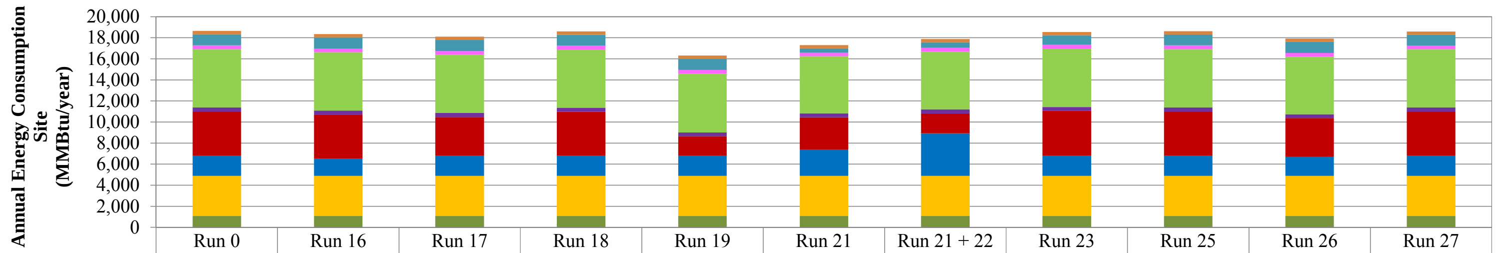
- Improving the efficiency of the packaged cooling and heating systems provided savings of 1.58% and 2.87% respectively.
- Implementing heat recovery from refrigeration coils provided exceedingly good savings of 12.51%.
- Installing a PVAV system provided a saving on 7.15%.
- Installing DOAS system provided a savings of 4.06%.
- Implementing demand control ventilation using CO_2 sensors provides savings of 3.88%.
- Implementing measures such as enthalpy based economizers, improved fan efficiencies and improved exhaust fan schedule provided savings within 1% of the whole building energy consumption.
- Within 1% savings were obtained on improving the efficiency of service water heaters.

When considering source energy consumption:

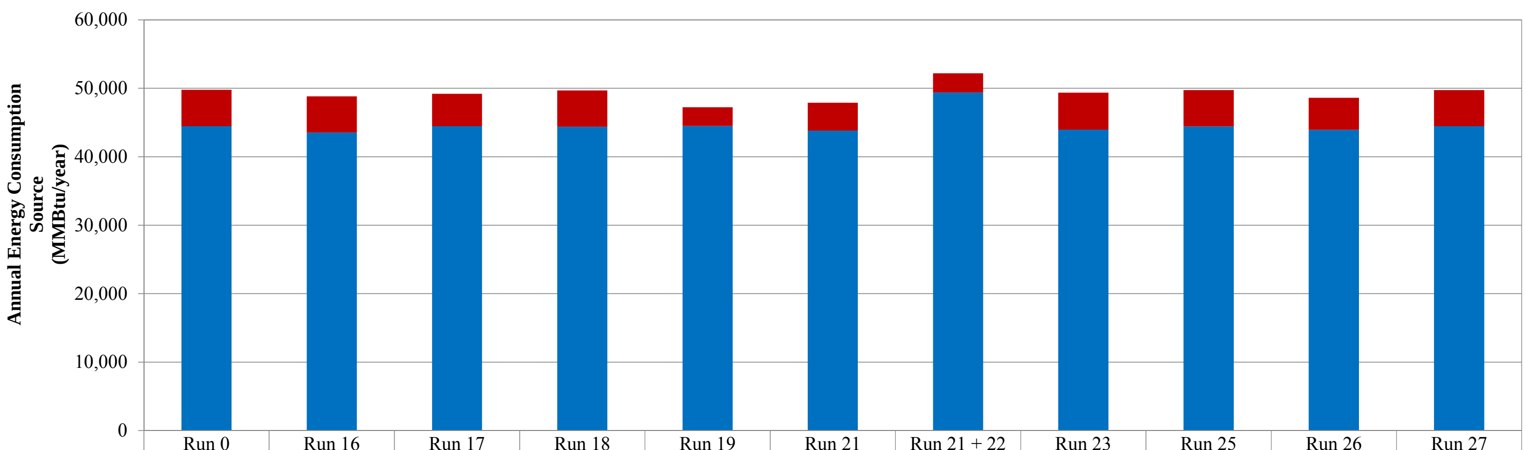
- Improving the efficiency of the packaged cooling and heating systems provided savings of 1.91% and 1.18% respectively.
- Implementing heat recovery from refrigeration coils provided savings of 5.13%.
- Installing a PVAV system provided a saving on 3.82%.
- Installing DOAS system provided a savings of -4. 84%.
- Implementing demand control ventilation using CO_2 sensors provides savings of 2.33%.
- Implementing measures such as enthalpy based economizers, improved fan efficiencies and improved exhaust fan schedule provided savings within 1% of the whole building energy consumption.
- Within 1% savings were obtained on improving the efficiency of service water heaters.

Table 6-9: Results from the Implementation of EEM's for HVAC and SHW System

		SITE ENERGY (MMBtu/yr)				SOURCE ENERGY (MMBtu/yr)	
		Elec.	**Nat. Gas**	**Total**	**% Above BC**	**Total**	**% Above BC**
BASE-CASE		13,809	4,828	18,637	-	49,776	-
EEMS							
16	Improved Cooling Efficiency	13,514	4,828	18,343	1.58	48,827	1.91
17	Improved Heating Efficiency	13,809	4,293	18,102	2.87	49,188	1.18
18	Installed Economizer	13,779	4,828	18,607	0.16	49,678	0.20
19	Implemented Heat Recovery from Refrigeration Coils	13,815	2,490	16,305	12.51	47,224	5.13
21	Installed PVAV System	13,603	3,701	17,304	7.15	47,874	3.82
22	Installed DOAS System	15,337	2,544	17,881	4.06	52,184	-4.84
23	Improved Fan Efficiency	13,648	4,899	18,548	0.48	49,337	0.88
25	Improved Exhaust Fan Schedule	13,795	4,828	18,623	0.08	49,731	0.09
26	Implemented Demand Control Ventilation	13,638	4,276	17,914	3.88	48,618	2.33
27	Improved Efficiency of Gas Service Hot Water Systems	13,810	4,773	18,583	0.29	49,718	0.12



	Run 0	Run 16	Run 17	Run 18	Run 19	Run 21	Run 21 + 22	Run 23	Run 25	Run 26	Run 27
Total	18,637	18,343	18,102	18,607	16,305	17,304	17,881	18,548	18,623	17,914	18,583
Ext. Light. (E)	312	312	312	312	312	312	312	312	312	312	312
Pumps & Fans (E)	1,047	1,047	1,047	1,047	1,047	392	531	913	1,033	1,047	1,047
SHW (E+G)	355	355	355	355	355	355	355	355	355	355	301
Refg. (E)	5,539	5,539	5,539	5,539	5,584	5,418	5,482	5,539	5,539	5,458	5,539
Heat Reject. (E)	385	385	385	385	344	378	383	385	385	379	385
Space Heat. (G)	4,180	4,180	3,645	4,180	1,843	3,053	1,896	4,251	4,180	3,629	4,180
Space Cool. (E)	1,927	1,632	1,927	1,896	1,929	2,505	4,031	1,900	1,927	1,843	1,927
Lighting (E)	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822
Misc. Equip. (E+G)	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069



	Run 0	Run 16	Run 17	Run 18	Run 19	Run 21	Run 21 + 22	Run 23	Run 25	Run 26	Run 27
Natural Gas	5,311	5,311	4,723	5,311	2,739	4,071	2,798	5,389	5,311	4,704	5,251
Electricity	44,465	43,516	44,465	44,367	44,484	43,802	49,386	43,948	44,420	43,914	44,467

Figure 6-15: Annual Site and Source Energy Consumption of EEMs Implemented for HVAC and SHW Systems

6.9.4 Results from the Implementation of Measures for Refrigeration Systems

Finally, several refrigeration measures that could be modeled in eQUEST – Refrigeration (Version 3.61) simulation program were assessed. Energy savings from EEMs for refrigeration equipment are presented in Table 6-10 and Figure 6-16 below.

When considering site energy consumption:

- Changing the suction group temperature / pressure control to floating mode based on loads on the compressor provided savings of 1.46%.
- Changing the compressor capacity controls from cycling to cylinder unloading control provided savings of 1.08%.
- Installing glass doors and covers on all display cases saved 16.90% in annual energy consumption.
- Improving the insulation for display cases by installation of vacuum panels and triple glazed insulated glass doors provided savings of 2.83%.
- Installing electronically commutated motors (ECMs) for evaporator fans saved 6.45%.
- Using hot gas from compressors to operate anti-sweat heaters provided savings of 4.64%.
- Using humidity levels to control the performance of anti-sweat heaters provides a saving of 3.08%.
- Implementing direct return ducts in the “Display-Case” zone to take advantage of the cooling effect provided by air spilling out from display cases saves 6.08% in annual energy consumption.[55]
- Switching from T8 lighting to T5 lighting in display cases provides savings of 1.72%. Switching to LED lighting systems provided a savings of 4.89%. Switching to fiber-optic lighting for display cases provides a saving of 7.19% in annual energy consumption. Implementing motion sensors for display case lighting provides savings of 4.78%.
- For improving the performance of refrigeration condensers implementation of floating head pressure control was considered. Implementing this measure saved 1.36% in annual energy consumption. Installing ECMs for condenser fans provided savings of 1.68%.
- Installing mechanical sub-cooling provided savings of 3.17%.

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[55] On looking at the results obtained from output reports for this efficiency measure, it was observed that this measure was not performing as originally intended. Instead, the eQUEST model was implementing this measure as strategy for heat removal from the fans and lighting system installed in the display cases of the grocery store.

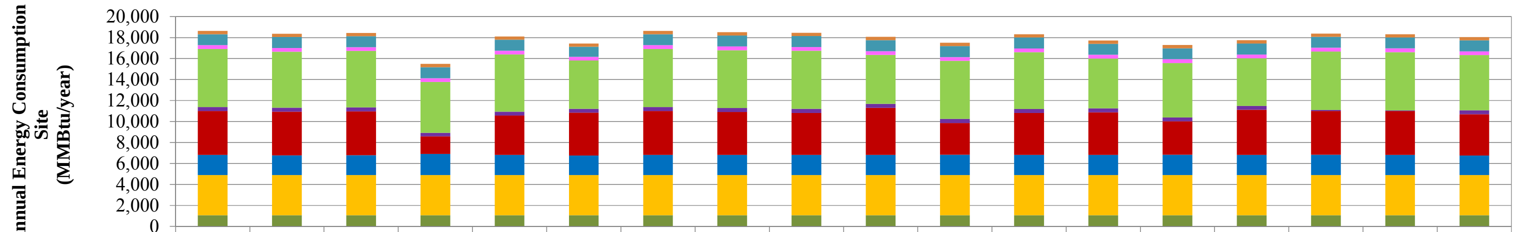
- Finally, installing hot gas defrost in low temperature display cases and implementing demand based defrost control for all display cases provided a saving of less than 1%.

When considering source energy consumption:

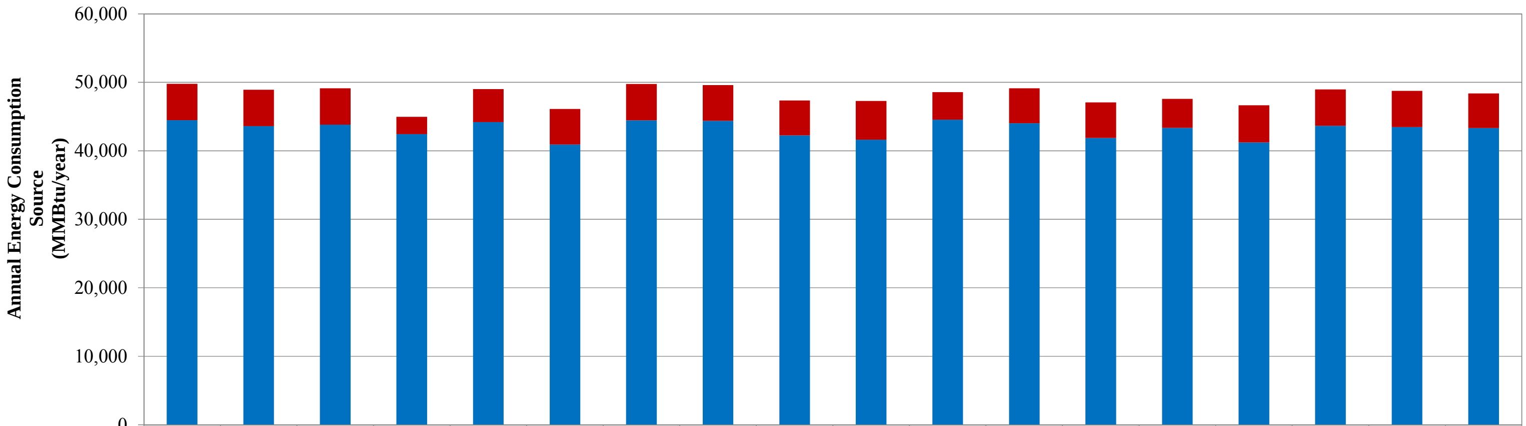
- Changing the suction group temperature / pressure control to floating mode based on loads on the compressor provided savings of 1.75%.
- Changing the compressor capacity controls from ‘cycling’ to ‘cylinder unloading’ control provided savings of 1.29%.
- Installing glass doors and covers on all display cases saved 9.64% in annual energy consumption.
- Improving the insulation for display cases by installation of vacuum panels and triple glazed insulated glass doors provided savings of 1.53%.
- Installing electronically commutated motors (ECMs) for evaporator fans saved 7.36%.
- Using hot gas from compressors to operate anti-sweat heaters provided savings of 4.85%.
- Using humidity levels to control the performance of anti-sweat heaters provides a saving of 5.02%.
- Implementing direct return ducts in the “Display-Case” zone to take advantage of the cooling effect provided by air spilling out from display cases saves 2.40% in annual energy consumption.
- Switching from T8 lighting to T5 lighting in display cases provides savings of 1.27%. Switching to LED lighting systems provided a savings of 5.41%. Switching to fiber-optic lighting for display cases provides a saving of 4.42% in annual energy consumption. Implementing motion sensors for display case lighting provides savings of 6.27%.
- For improving the performance of refrigeration condensers implementation of floating head pressure control was considered. Implementing this measure saved 1.63% in annual energy consumption when accounted for at source. Installing ECMs for condenser fans provided savings of 2.02% in annual energy consumption when accounted for at source.
- Installing mechanical sub-cooling provided savings of 2.81%.
- Finally, hot gas defrost in low temperature display cases and implementing demand based defrost control for all display cases provided a saving of less than 1%.

Table 6-10: Results from the Implementation of EEM's for Refrigeration System

		SITE ENERGY (MMBtu/yr)				SOURCE ENERGY (MMBtu/yr)	
		Elec.	**Nat. Gas**	**Total**	**% Above BC**	**Total**	**% Above BC**
BASE-CASE		13,809	4,828	18,637	-	49,776	-
EEMS							
28	Changed Suction Group Temperature Control	13,539	4,826	18,365	1.46	48,904	1.75
29	Changed Compressor Capacity Controls	13,610	4,826	18,436	1.08	49,133	1.29
31	Installed Doors / Covers on All Display Cases	13,179	2,308	15,488	16.90	44,976	9.64
32	Improved Insulation for Display Cases	13,724	4,385	18,109	2.83	49,016	1.53
35	Installed ECM Motors for Evaporator Fans	12,705	4,731	17,436	6.45	46,115	7.36
36	Installed Hot-Gas Defrost in Low Temperature Display Cases	13,806	4,828	18,633	0.02	49,763	0.03
37	Improved Defrost Control to "Demand Control"	13,803	4,731	18,516	0.65	49,591	0.37
38	Used Hot-Gas to Operate Anti-Sweat Heaters	13,119	4,653	17,773	4.64	47,363	4.85
39	Used Humidity Levels to Activate Anti-Sweat Heater Controls	12,927	5,135	18,063	3.08	47,275	5.02
40	Implemented Direct Return Ducts in "Display-Case" Zone	13,833	3,672	17,505	6.08	48,580	2.40
41	Switched to T5 Display Case Lighting	13,676	4,641	18,317	1.72	49,142	1.27
42	Switched to LED Display Case Lighting	13,011	4,714	17,725	4.89	47,082	5.41
47	Implemented Fiber Optic Lighting for Display Cases	13,467	3,830	17,297	7.19	47,576	4.42
43	Implemented Case Lighting Schedule	12,800	4,946	17,746	4.78	46,657	6.27
44	Changed Condenser Controls to LOAD-RESET	13,558	4,826	18,384	1.36	48,967	1.63
45	Installed ECM Motors for Condenser Fans	13,496	4,828	18,324	1.68	48,769	2.02
46	Installing Mechanical Sub-cooling	13,455	4,592	18,047	3.17	48,377	2.81



	Run 0	Run 28	Run 29	Run 31	Run 32	Run 35	Run 36	Run 37	Run 38	Run 39	Run 40	Run 41	Run 42	Run 47	Run 43	Run 44	Run 45	Run 46
Total	18,637	18,365	18,436	15,488	18,109	17,436	18,633	18,516	17,773	18,063	17,505	18,317	17,725	17,297	17,746	18,384	18,324	18,047
Ext. Light. (E)	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312	312
Pumps & Fans (E)	1,047	1,047	1,047	1,047	1,047	959	1,047	1,047	1,045	1,047	1,047	1,047	1,047	1,047	1,047	1,047	1,047	1,047
SHW (E+G)	355	355	355	355	355	355	355	355	355	355	355	355	355	355	355	355	355	355
Refg. (E)	5,539	5,326	5,376	4,856	5,457	4,595	5,535	5,515	5,540	4,669	5,549	5,408	4,751	5,202	4,543	5,573	5,543	5,260
Heat Reject. (E)	385	380	381	338	381	371	385	384	385	376	385	383	375	369	372	96	68	370
Space Heat. (G)	4,180	4,178	4,178	1,660	3,737	4,083	4,180	4,083	4,005	4,487	3,024	3,993	4,066	3,182	4,298	4,178	4,180	3,944
Space Cool. (E)	1,927	1,875	1,895	2,027	1,929	1,868	1,927	1,927	1,927	1,925	1,940	1,927	1,926	1,938	1,927	1,932	1,927	1,867
Lighting (E)	3,822	3,823	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822	3,822
Misc. Equip. (E+G)	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069	1,069



	Run 0	Run 28	Run 29	Run 31	Run 32	Run 35	Run 36	Run 37	Run 38	Run 39	Run 40	Run 41	Run 42	Run 47	Run 43	Run 44	Run 45	Run 46
Natural Gas	5,311	5,309	5,308	2,539	4,823	5,204	5,311	5,204	5,119	5,649	4,039	5,105	5,186	4,213	5,441	5,309	5,310	5,051
Electricity	44,465	43,596	43,825	42,437	44,193	40,911	44,452	44,386	42,244	41,626	44,541	44,037	41,896	43,362	41,216	43,658	43,458	43,326

Figure 6-16: Annual Site and Source Energy Consumption of EEMs Implemented for the Refrigeration Systems

6.9.5 Cumulative Assessment of Results from the Implementation of EEMs

The measures that were assessed individually were grouped together to assess the cumulative impact on implementing efficiency measures for the building envelope, lighting, HVAC and refrigeration system in the grocery store. Energy savings from the consolidated EEMs are presented in Table 6-11 and Figure 6-17 below.

However, certain measures assessed as individual entities were not considered for the cumulative assessment. The combined list of measures excluded measures for economizers, dedicated outdoor air systems, changing out the display case suction temperature controls, hot-gas defrost options, EEMs for open display cases, and improvements to lighting system using T5 systems. The list is presented in Table 6-12 below.

- The use of economizers was excluded because less than 1% savings is seen from the results. In addition, ASHRAE Standard 90.1-2010 exempts grocery stores from the implementation of economizers because the use of outdoor air is detrimental to the operation of open refrigerated cases.
- The use of DOAS system although found to be a viable alternative to the packaged single zone systems being implemented in the base-case model did not show effective results. The ineffectiveness in results can be attributed to limitations in eQUEST-Refrigeration program. The exclusion of display case suction temperature controls was also excluded for similar reasons.
- The use of hot-gas defrost was not considered a viable measure because the practical issues associated with the implementation of this measure far exceed its energy saving potential.
- EEMs incorporating improvements to lighting system using T5 systems were not considered because the implementation of EEMs incorporating LEDs was found to provide greater savings.
- Finally, the implementation of fiber optic lighting for display cases although provided good savings was not considered because this measure is still in development stages. Hence, as of today, the practical issues associated with the implementation of this measure can far exceed the energy saving potential.

When considering site energy consumption:

- Consolidated envelope measures provide a savings of 5.60%.
- In addition to the envelope measures consolidated lighting measures provide a saving of 15.58%.

- In addition to the envelope and lighting measures, consolidated EEMs for HVAC system provide a saving of 30.71%.
- Finally the inclusion of refrigeration EEMs in addition to envelope, lighting and HVAC EEMs provide a cumulative savings of 57.99%.

When considering source energy consumption:

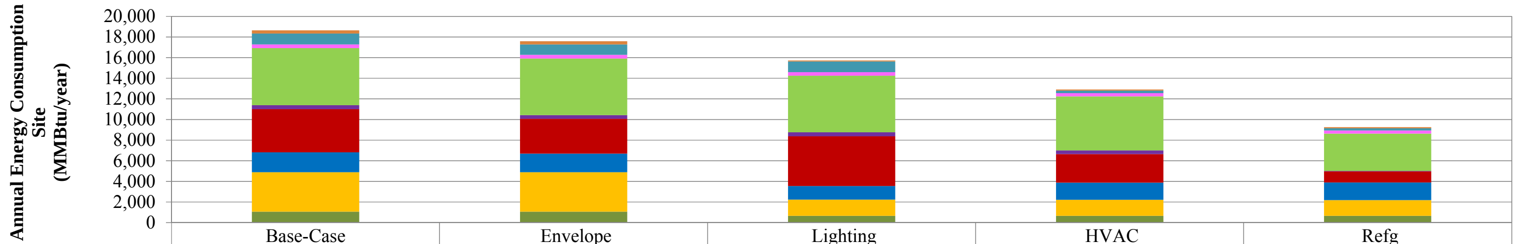
- Consolidated envelope EEMs provide a savings of 3.26%.
- In addition to the envelope EEMs, consolidated lighting measures provide a saving of 21.09%.
- In addition to the envelope and lighting EEMs, consolidated EEMS for HVAC systems provide a saving of 30.32%.
- Finally the inclusion of refrigeration EEMs to envelope, lighting and HVAC EEMs provide a cumulative savings of 56.00%.

Table 6-11: Results from the Implementation of EEM's for the Building Envelope

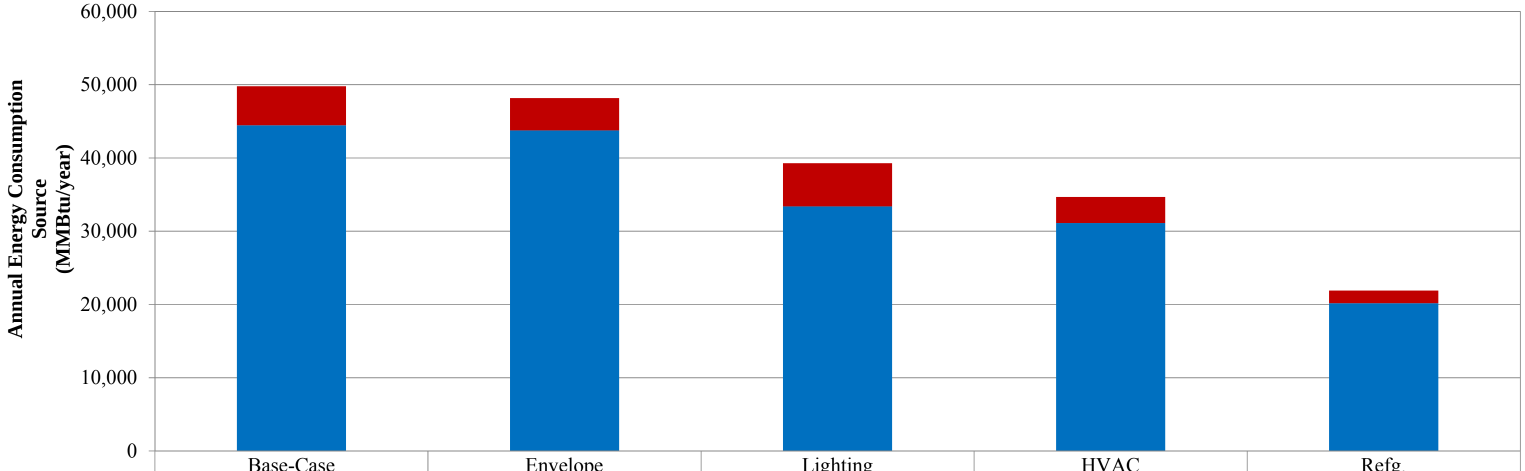
	SITE ENERGY (MMBtu/yr)				SOURCE ENERGY (MMBtu/yr)	
	Elec.	**Nat. Gas**	**Total**	**% Above BC**	**Total**	**% Above BC**
BASE-CASE	13,809	4,828	18,637	-	49,776	-
Consolidated EEMs						
Envelope	13,586	4,008	17,594	5.60%	48,156	3.26%
Envelope + Lighting	10,364	5,370	15,734	15.58%	39,280	21.09%
Envelope + Lighting + HVAC	9,660	3,254	12,914	30.71%	34,685	30.32%
Envelope + Lighting + HVAC + Refrigeration	6,268	1,562	7,830	57.99%	21,900	56.00%

Table 6-12: Energy Efficiency Measures Selected for the Final Simulation Run

EEM No.	Energy Efficient Measures
Building Envelope	
Run 1	Improved insulation for walls and roof
Run 2	Improved solar reflectance and thermal emittance of roof
Run 3	Improved insulation of walls and roof for freezer and cooler
Run 4	Improved loading dock door insulation
Run 5	Increased area for skylights
Run 6	Improved U-value, SHGC and transmittance for skylights
Run 7	Reduced infiltration
Lighting and Daylighting Systems	
Run 8	Reduced lighting power density
Run 8+9	Implemented time switches and occupancy sensors
Run 10	Implemented of daylighting controls
Run 12, 13	Improved façade and parking lighting systems
Run 14, 15	Reduced equipment power density and implemented schedules to turn off equipment
HVAC Systems	
Run 16	Improved cooling efficiency of packaged rooftop units
Run 17	Improved heating efficiency of packaged rooftop units
Run 19	Implementing heat recovery from refrigeration coils for space heating
Run 21	Implementing package variable air volume (PVAV) system
Run 23	Improved supply fan efficiency
Run 25	Installed demand based exhaust fan schedule
Run 26	Demand control ventilation
	Service Water Systems
Run 27	Improved service hot water heater efficiency
Refrigeration Systems	
Run 28	Implemented floating suction group temperature controls
Run 29	Implemented compressor capacity controls
Run 31	Installed glass doors on all refrigerated display cases
Run 32	Improved insulation for display case walls and doors
Run 35	Improved display case evaporator fan efficiency
Run 38	Implemented hot gas anti-sweat heaters
Run 39	Implemented humidity levels to activate anti-sweat heater controls
Run 40	Implemented direct return ducts
Run 42	Implemented LED lighting in all display cases
Run 43	Implemented occupancy sensors to activate display-case lighting
Run 44	Implemented floating saturated condenser pressure controls
Run 45	Implemented mechanical subcooling
Run 46	Implemented high efficiency motors for condenser fans



	Base-Case	Envelope	Lighting	HVAC	Refg
Total	18,637	17,594	15,734	12,914	7,830
Ext. Light. (E)	312	312	106	106	106
Pumps & Fans (E)	1,047	1,011	1,034	263	209
SHW (E+G)	355	355	355	301	301
Refg. (E)	5,539	5,483	5,480	5,240	3,628
Heat Reject. (E)	385	376	375	354	36
Space Heat. (G)	4,180	3,360	4,831	2,769	1,077
Space Cool. (E)	1,927	1,804	1,326	1,654	1,716
Lighting (E)	3,822	3,822	1,552	1,551	1,507
Misc. Equip. (E+G)	1,069	1,069	675	675	675



	Base-Case	Envelope	Lighting	HVAC	Refg.
Natural Gas	5,311	4,409	5,907	3,579	1,719
Electricity	44,465	43,747	33,373	31,106	20,181

Figure 6-17: Annual Site and Source Energy Consumption of Consolidated EEMs for Envelope, Lighting, HVAC and Refrigeration System

6.10 Summary and Conclusions

In this chapter, several EEMs for the grocery store were considered and assessed. These included EEMs for the building envelope, lighting, HVAC and refrigeration systems of the grocery store. The measures were first assessed individually and then combined to provide a cumulative energy savings.

EEMs for the building envelope were least effective of all the EEMs considered by this study. This is because the grocery store is an internal load dominated building with the primary sources of energy consumption being the lighting and refrigeration system. Savings for site energy consumption were in the range of -0.26% to 3.10%; and for source energy consumption were in the range of -0.20% to 2.34%. Biggest savings were seen on implementing reduced infiltration rates.

EEMs for the ambient lighting systems were one of the big-ticket items that decreased the energy consumption from the implementation of individual measures. This is because of the round the clock operating hours of the grocery store. Savings for site energy consumption were in the range of 4.22% to 5.29%; and for source energy consumption were in the range of 7.95% to 9.55%. Biggest savings were seen on implementing the time-switches and occupancy sensors in addition to reduced lighting power density.

EEMs for the HVAC systems fared reasonably well. Savings for site energy consumption were in the range of 0.02% to 16.9%; and for source energy consumption were in the range of 0.03% to 9.64 Biggest savings were seen on implementing the heat recovery from refrigeration coils. However, this measure may not be as viable in real life scenario due to practical issues involved in implementing this EEM. Issues such as increase refrigeration charge[56], which is associated with the heat reclaim strategy (PG&E 2011), have not been covered in this study. The assessment of dedicated outdoor air system (DOAS) although attempted could not be correctly assessed due to the limitations in the eQUEST-Refrigeration model.

EEMs for the refrigeration systems also fared reasonably well. Savings for site energy consumption were in the range of 0.1% to 16.9%; and for source energy consumption were in the range of 0.1% to 9.8%. Most successful EEMs in this category included the installation of glass

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[56] According to the recommendations prepared by Pacific Gas and Electric Company for supermarket refrigeration codes in California, refrigerant heat recovery was assumed to increase charge size by up to 20%. The study also observed that refrigerant heat recovery could increase a grocery store's annual refrigerant rate by 5% as a result of the additional equipment and piping required.

doors on all display cases in the grocery store model (site energy savings of 16.9% and source energy savings of 9.8%); using EEM motors for evaporator fans in the display cases (site energy savings of 6.4% and source energy savings of 7.3%); and using LEDs for display case lighting (site energy savings of 7.2% and source energy savings of 4.5%). The installation glass door on display cases is commonly misunderstood by grocery store management to have a negative impact on sales. However, studies have pointed out to the contrary (Fricke and Becker 2010).

Finally, assessing the impact of including all the EEMs in the grocery store model provided site energy savings of 57.99% and source energy savings of 56.00%. The reported savings are in agreement with the savings reported from reputed sources such as refrigeration road map presented by the Carbon Trust (2010), Leach et al. (2009) and several other sources as cited in the literature review section of this study. However, it should be noted that several assumptions had to be made in the eQUEST-Refrigeration model to simulate some of the EEMs discussed above, which could have impacted the results. In future, it is recommended to use component based simulation tools such as TRNSYS, which would allow more flexibility in modeling innovative efficiency strategies for the grocery store.

CHAPTER VII

CHP OPTIONS FOR THE GROCERY STORE

7.1 Overview

The last chapter demonstrated savings in building energy use that could be achieved by considering the grocery store as an individual entity. In this section the impact of installing a CHP system to provide electricity and thermal energy to the grocery store was analyzed. In addition, instead of being considered as an individual entity, the grocery store was considered as part of the community in terms of sharing of energy across the boundaries of the grocery store. Any surplus energy generated on site within the boundary of the grocery store was exported to the surrounding, which in this study was assumed to be the surrounding residential community. This part of the study assessed the installation of CHP facilities to primarily facilitate the requirements of the grocery store. Surplus thermal energy was then utilized by the surrounding residential community. Four options for CHP were selected and analyzed. The selection process was based on the varying the utilization of thermal energy being generated by the CHP facility.

In order to proceed with the analysis, in addition to the grocery store model discussed in the previous sections of this study, an additional residential model and a CHP model were designed. Modifications to the grocery store model also had to be made in order to better integrate the CHP facility in the grocery store. Details of the modifications to the grocery store model as well as detailed description of the residential model and the CHP model are described in the sections that follow. Finally, the four selected options are described and analyzed in terms of energy consumption of the grocery store and the potential of sharing surplus energy from the store with energy requirements of a neighboring residential community.

7.2 The Grocery Store Model

The base-case grocery store model and the corresponding energy efficiency measures (EEMs) that have been recommended for the grocery store have been discussed in the last two chapters of this study. In this chapter, an energy efficient grocery store model was used to assess several options of CHP for the store. The inclusion of an appropriate set of EEMs in the base-case grocery store model was based on the option of CHP facility being analyzed. These measures were presented along with the discussion on each of the CHP options being analyzed.

In addition, for the analysis of the CHP options, the base-case grocery store model was modified to utilize the thermal output from the CHP facility. The modifications included incorporating:

- An absorption chiller,
- A central chilled water and hot water distribution system, and
- An auxiliary hot water boiler.

However, these modifications were not modeled using eQUEST-Refrigeration. The models for these modifications were developed as part of the CHP model, which is presented in Section 7.3 of this chapter. In order to integrate the grocery store model with the CHP model, depending on the CHP option selected, hourly loads for space cooling and heating; service hot water heating, medium and low refrigeration, as well as electricity loads were extracted from the eQUEST-Refrigeration output file to an excel spreadsheet and matched with the hourly calculations from the CHP model.

The next section discusses the residential model that was created to absorb the excessive electricity and thermal energy generated by the CHP facility at the grocery store. The residential model was also modified to accommodate the surplus energy provided by the CHP facility installed in the grocery store.

7.3 The Residential Model

Several research studies and sources (Phetteplace 1995, Atta 2006, ASHRAE 2008[1]) recommend that high density building clusters be used to more effectively absorb waste thermal energy from a centralized CHP facility[2]. Hence, this study focuses on the performance of the multifamily units to absorb the waste thermal energy from the CHP facility installed in the grocery store.

This section of the study describes the residential model for a multi-family residential building that was used in the analysis. The model is based on the DOE-2.1e (Winkelmann et al. 1993) simulation model[3] developed at the Energy Systems Laboratory (Kim 2006, Malhotra 2009). The residential model used in this study adheres to the specifications for Climate Zone 2

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[1] Chapter 11, District Heating and Cooling.
[2] Although other examples (Knight and Ugursal 2005) demonstrate some successful stories for low-density residential areas utilizing centralized CHP facilities.
[3] BDL version 4.01.08.

provided in the 2009 IECC and certain assumptions adopted from other sources[4] (ICC 2009a). The model uses natural gas heating and electric cooling.

7.3.1 Multi-Family Model Specifications

The base-case simulation model is a two-story building consisting of 8 units with two bedrooms per unit. There are four units on each floor that are arranged in sets of 2 units sharing a common wall. A breezeway is located between the two sets of units. Each unit in the multi-family building was simulated as a single-zone building in delayed construction mode[5] to take into account the thermal mass of the construction materials[6]. Two occupants were assumed for each unit. A summary of the multi-family model is provided in Figure 7-1 and Table 7-1.

7.3.1.1 Building Envelope and Space Conditions

The base-case units are square shaped, one storied, with a conditioned floor area of 1009 ft^2, and a floor to ceiling height of 8 ft. (NAHB 2003). The units on the second floor have a ventilated attic with a roof pitched at 23 degrees. The units have a brick finish on the exterior walls and asphalt shingle roofing. The wall construction was made from light weight wood frame with 2x4 studs at 16” on-center spacing (NAHB 2003). The floor of the first floor units is of slab-on-grade construction in accordance to standard practice for residential buildings in Climate Zone 2. The wall insulation is R-13 and the ceiling insulation is R-30 as recommended in the 2009 IECC for Climate Zone 2.

The window area for each multi-family unit is 8% of the total conditioned space area (NAHB 2003), and is distributed equally on two of the exterior walls of the unit. No exterior shading was provided to either the windows or the walls of each unit. The window U-value was set at 0.65 Btu/hr ft^2 °F and the window SHGC was set at 0.3. The fenestration characteristics were simulated by creating custom windows with double pane, low-e glazing and vinyl frames. A single door 20 ft^2 in area is assumed to be on the front of each unit. The door has the U-value of 0.65 Btu/hr ft^2 °F, which is same as that for the windows of the unit.

The total internal heat gain is specified by the 2009 IECC is assumed to be 0.25 kW for lighting and 0.36 kW for equipment. The lighting and equipment schedules were adopted from Building America Benchmark Definition (Hendron 2008) and are presented in Figure 7-2. The

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[4] Sources for the base-case other than the 2009 IECC have been cited separately.
[5] Delayed construction mode is used in DOE-2.1e to account for the thermal mass properties of the building materials used in the simulation model, which has an impact on the calculated space heating and cooling.
[6] This was accomplished using the DOE-2.e Custom Weighting Factors.

infiltration rate in the house is provided by specific leakage area (SLA) of 0.00036 ft^2/ft^2 of conditioned floor area. The infiltration rate of the vented attic is modeled with a SLA of 0.0033 ft^2/ft^2 of conditioned floor area.

7.3.1.2 HVAC and Domestic Hot Water System Characteristics

In the original setting of the base-case simulation model, each of the apartment units included a central air-conditioning unit and a heating system. The efficiency of the central electric air-conditioning system was set to be at SEER 13. The efficiency of the central furnace operated on natural gas was set to be at AFUE 0.78. The efficiencies are mandated by the National Appliance Energy Conservation Act (NAECA). The system was sized at 2.0 tons using 500 ft^2 conditioned space per ton. For units on the first floor, the mechanical equipment[7] was located in the conditioned space. For units on the second floor, the mechanical equipment was located in the unconditioned attic. Duct insulation is set at R-6 for both supply and return ducts. A 5% leakage was assumed for both the supply and return ducts.

The base-case domestic hot water (DHW) system used a 30 gallon storage type water heater. The energy factor (EF) of the DHW heater was calculated to be 0.613. The daily hot water usage was calculated to be 50 gallons. Schedules for DHW usage are adopted from the Building America Benchmark Definition (Hendron 2008), and are presented in Figure 7-2.

[7] Mechanical equipment includes air-handler units, supply ducts and return ducts.

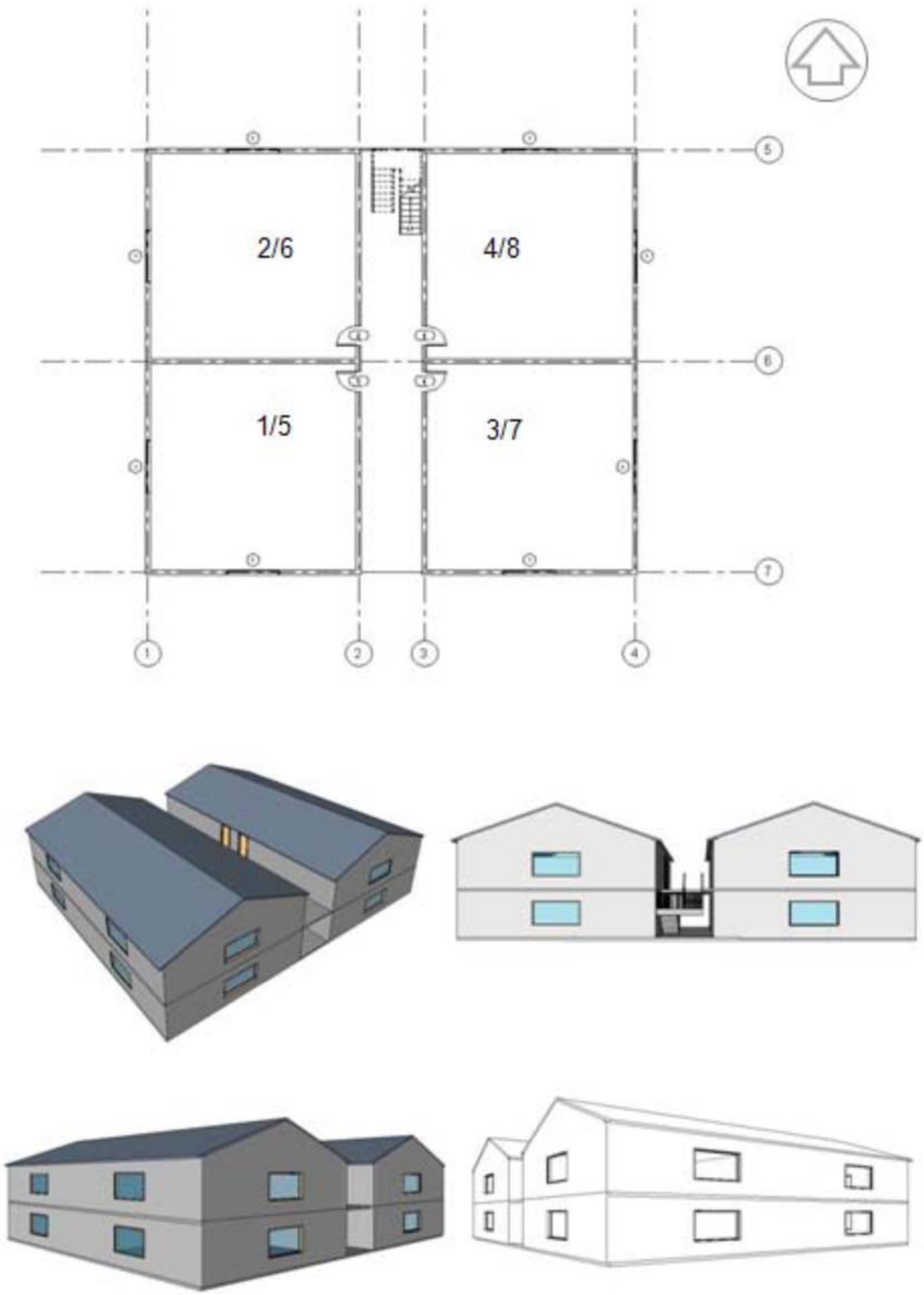
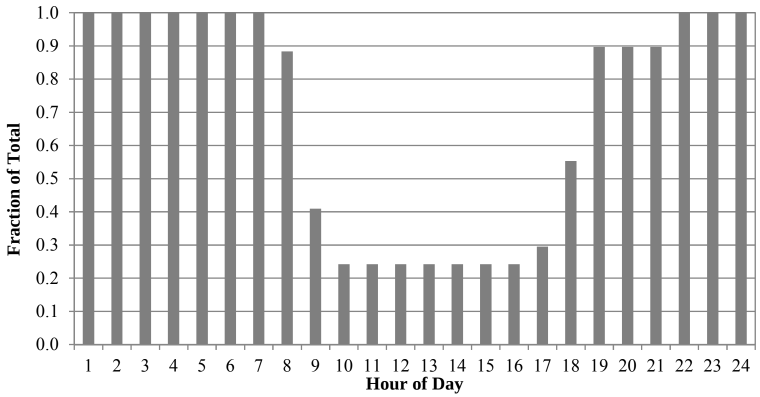


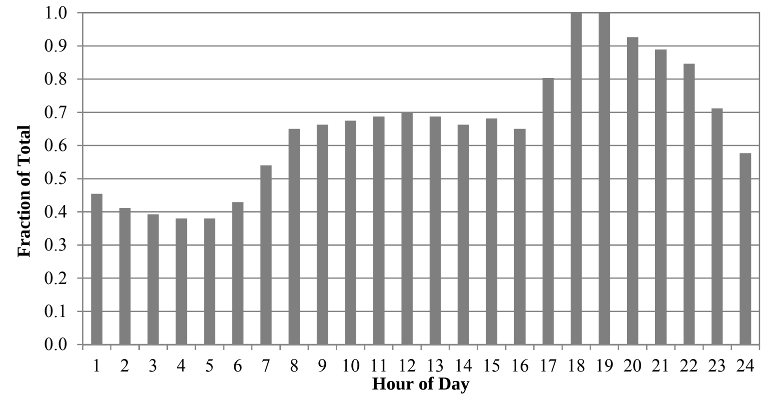
Figure 7-1: Schematic Layout of the Eight Units for the Multi-Family Model

Table 7-1: Input for the Multi-Family Building Model

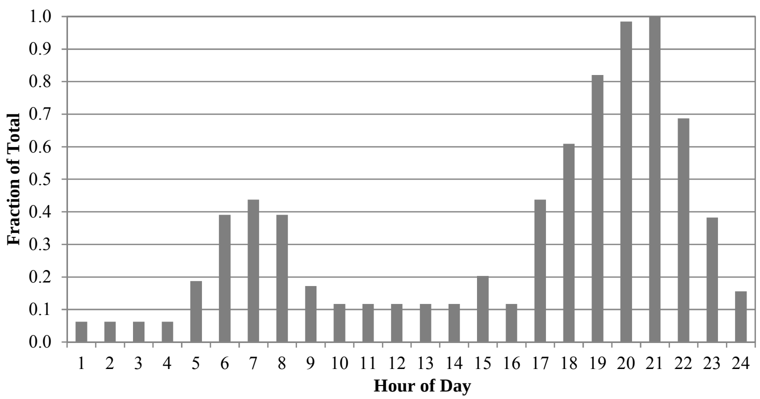
CHARACTERISTICS	ASSUMPTIONS AND SPECIFICATIONS	INFORMATION SOURCES	COMMENTS
BUILDING			
Building Type	Multifamily unit 8 units per building		
Gross Area	1,006 sq. ft. (31.6 ft. x 31.6 ft.)	NAHB (2003)	
Number of Floors (Entire building)	2 floors 4 units per floor	NAHB (2003)	
Floor to Floor Height (ft.)	8	NAHB (2003)	
Number of Exposed Walls (Per unit)	3 exposed walls		
Number of Bedrooms	2		
Number of Occupants	2		
CONSTRUCTION			
Construction	Light-weight wood frame with 2x4 studs spaced at 16" on center	NAHB (2003)	
Floor	Slab-on-grade floor for lower units	NAHB (2003)	
Roof Configuration	Unconditioned, vented attic	NAHB (2003)	
Roof Absorptance	0.75	2009 IECC, Table 405.5.2(1)	
Ceiling Insulation (hr-sq.ft.-°F/Btu)	R-30	2009 IECC, Table 402.1.1	Assuming insulation on ceiling.
Wall Absorptance	0.75	2009 IECC, Table 405.5.2(1)	
Wall Insulation (hr-sq.ft.-°F/Btu)	R-13	2009 IECC, Table 402.1.1	
Slab Perimeter Insulation	R-0	2009 IECC, Table 402.1.1	
U-Factor of Glazing (Btu/hr-sq.ft.-°F)	0.65	2009 IECC, Table 402.1.1	
Solar Heat Gain Coefficient (SHGC)	0.3	2009 IECC, Table 402.1.1	
Window Area	8% window to floor area ratio (WFAR) distributed equally on 2 orientations	NAHB (2003)	This amounts to 79.38 sq. ft. of total window area with 15.7% window to wall area ratio for two exterior walls.
Exterior Shading	None	2009 IECC, Table 405.5.2(1)	
Roof Radiant Barrier	No		
Space Conditions			
Space Temperature Set point	72°F Heating, 75°F Cooling, No set-back	2009 IECC, Table 405.5.2 (1)	
Internal Heat Gains	Igain = 17,900 + 23.8xCFA + 4104 x N_{br}	2009 IECC, Table 405.5.2 (1)	This assumes heat gains from lighting, equipment and occupants. The % breakdown of the lighting and equipment component is adopted from Hendron et al. 2008.
Number of Bedrooms	2		Calculated from the area assigned to each unit.
Number of Occupants	2		
Mechanical Systems			
HVAC System Type	Electric cooling (air conditioner) and Natural gas heating (gas fired furnace)		
HVAC System Efficiency	AC: SEER 13 Furnace: 0.78 AFUE	2009 IECC, Table 503.2.3 (2), 503.2.3 (4)	
Cooling Capacity (Btu/hr)	24,216		Assuming 500 ft^2/ton.
Heating Capacity (Btu/hr)	24,216		1.0 x cooling capacity.
DHW System Type	30-gallon tank type gas water heater	ASHRAE Applications Handbook (2003)	
DHW Heater Energy Factor	Gas EF: 0.613	2009 IECC, Table 504.2	Gas: 0.67-0.0019 V EF Where V=storage volume (gal.).
Duct Location	Top Floor: Unconditioned attic Lower Floor: In conditioned space	NAHB (2003)	
Duct Leakage (%)	Top Floor: 5.56% (supply & return)	2009 IECC, Sec. 403.2.2	8 CFM/100 ft^2 of CFA to outdoors.
	Lower Floor: 0%		
Duct Insulation (hr-sq.ft.-°F/Btu)	Top Floor: R-8 (supply)/R-6 (return) Lower Floor: N.A.	2009 IECC, Sec. 403.2.1	
Supply Air Flow (CFM/ton)	360		
Attic Infiltration	0.0033 (Top floor only)	2009 IECC, Table 405.5.2 (1)	1 ft^2 per 300 ft^2 of ceiling area.
Infiltration Rate (SG)	SLA= 0.00036	2009 IECC, Table 405.5.2 (1), ASHRAE 119 Section 5.1	



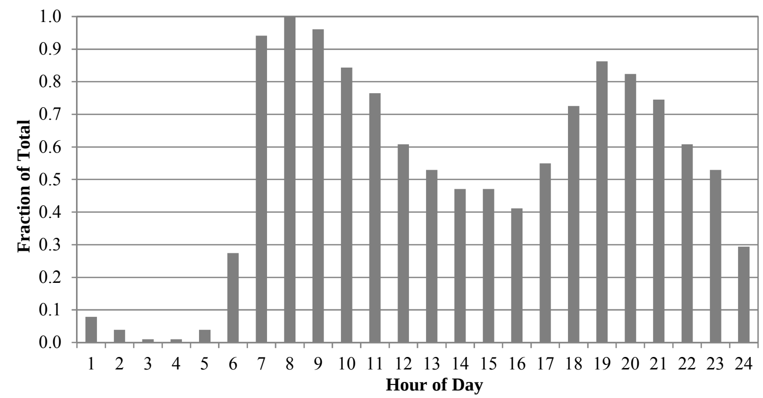
OCCUPANCY PROFILE



MISC. EQUIPMENT PROFILE



LIGHTING. EQUIPMENT PROFILE



DHW USAGE PROFILE

Figure 7-2: Occupancy, Miscellaneous and Lighting Equipment, and DHW Usage Profiles for the Multi-Family Building Model (Source: Hendron et al., 2008)

7.3.2 Energy Usage

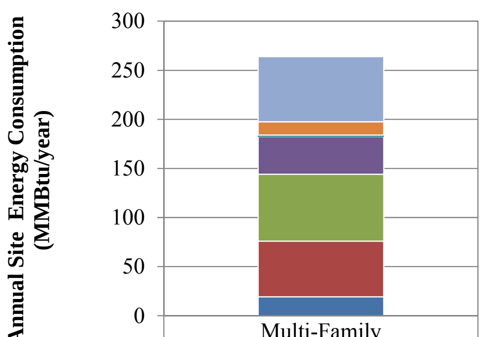
7.3.2.1 Annual Energy Usage

Annual energy usage for an 8-unit multi-family building operating on electricity and natural gas are presented in Figure 7-3. When considering site energy consumption:

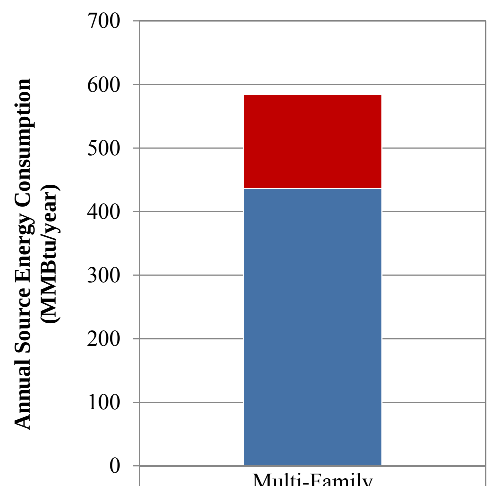
- The total annual energy consumption was predicted to be 263.9 MMBtu/yr with electricity consumption of 129.5 MMBtu/yr and a natural gas consumption of 134.4 MMBtu/yr.
- Lighting and equipment consumption accounted for 28.8% of the total energy consumption, followed by space heating consumption at 25.8% and DHW energy consumption at 25.2%. Space cooling accounted for 14.6% of the total energy consumption while 5.7% of the total energy consumption can be attributed to the operation of pumps, miscellaneous and ventilation fans.

When considering source energy consumption:

- The energy for electricity generation was predicted to be 436.5 MMBtu/yr, and
- The natural gas consumption was predicted to be 147.8 MMBtu/yr.[8]



	Multi-Family
TOTAL	263.9
DHW	66.4
Vent. Fans	13.4
Pumps & Misc.	1.6
Space Cooling	38.5
Space Heating	68.0
Equipment	56.8
Lights	19.2



	Multi-Family
TOTAL	584.3
Natural Gas	147.8
Electricity	436.5

Figure 7-3: Annual Energy Consumption for the Multi-Family Model (Site and Source)

[8] A site to source energy factor of 3.15 was used for electricity and 1.1 for natural gas. These factors are obtained from the Annual Energy Review published annually by the U.S. Energy Information Administration (EIA 2010). The site to source conversion for electricity also accounts for 7% transmission losses.

7.3.2.2 Typical Hourly Building Loads

Typical hourly electricity and thermal energy requirements for the original multi-family model considered for this analysis is presented in Appendix D[9] of this study. Hourly electricity loads include requirements from lighting and miscellaneous systems, space cooling, pumps and fans to operate the space cooling and heating systems. Thermal energy requirements include space heating loads, and DHW heating loads. Assessment of the hourly energy consumption of the grocery store is assessed in terms of analysis of temperature bin distribution and analysis of typical daily profiles. The assessment is presented in the sub-sections below.

7.3.2.2.1 Typical Daily Profile of Loads

To assess typical daily profiles for multi-family energy usage the months of January, March, and July are selected to represent winter, spring, summer and fall respectively. Hourly trends projecting the maximum, 75th percentile, 50th percentile, 25th percentile and the minimum loads for electrical and thermal loads are provided for four sample months. The typical daily profiles of electricity consumption, space heating ad DHW loads are presented in Figure D-2, Appendix D.

For electricity consumption the following trends were observed:

- The highest consumption happens during summer and the lowest consumption happens during winter.
- In January, the electricity consumption pattern mostly follows the consumption pattern of lighting and miscellaneous equipment.
- On the other hand in the month of July, electricity consumption from space cooling is a major factor contributing to the typical electricity load profiles.
- The range of electricity consumption is narrow in most cases except in October. This is because of the ambient temperature swings which drive the space conditioning system into either heating or cooling mode.

For thermal energy consumption an inverse pattern was observed:

- Highest consumption patterns of thermal energy are seen in the month of January with both space heating and domestic hot water heating requirements contributing to the hourly trends.

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[9] Hourly electricity and thermal energy requirements for each multi-family unit were extracted from hourly DOE-2.1e output reports. Values for electricity and natural gas consumption were extracted from the hourly end-use reports for electricity and natural gas consumption in the PLANT section of the DOE-2.1e program. Values for hourly space heating loads and DHW loads were extracted from the SYSTEM section of the DOE-2.1e program. The hourly values obtained from each unit were then added to present the energy consumption for the entire multi-family building.

- On the other hand, lowest consumption patterns are in the month of July with the consumption patterns primarily driven by DHW usage pattern.

7.3.2.2.2 Temperature Bin Distribution of Loads

Electricity and thermal loads are also analyzed using temperature bins. The electricity consumption and heating loads are plotted against the average ambient temperature bins on the horizontal axis. The temperature bin distribution of loads is presented in Figure D-1, Appendix D.

For electricity consumption the following trends were observed:

- The electricity consumption was constant below the temperature bin of 61-65°F. Below this temperature point electricity consumption for space cooling is no longer required and the electricity requirements are reduced to lighting and miscellaneous loads only.

For thermal energy consumption an inverse pattern was observed:

- As the ambient temperature increased the heating load decreases to a point where it remained fairly constant. This temperature was observed to be the 55 – 60°F temperature range. Eventually, all that was left is the heating load from DHW consumption during summer months.

7.3.3 <u>Modifications to the Space Heating and DHW Systems</u>

Surplus thermal energy for space heating and DHW heating purposes from the grocery store was available at a temperature of 180°F. In order to absorb this surplus thermal energy from the grocery store and to utilize this energy for space heating and DHW purposes, certain modifications had to be made. The modifications are listed below:

- The natural gas operated furnace in the original model of the multi-family unit were replaced with an appropriate hydronic system for space heating.
- Direct-fired water heater in the original model of the multi-family unit was replaced with an indirect-fired hot water heater.
- The modified base-case multi-family building was also equipped with a hot water storage tank for each multi-family building. Hot water was supplied to the storage tank at a temperature of 160°F and return hot water was set at a temperature of 140°F[10].

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[10] In this arrangement, also known as combination hydronic system (Butcher 2011), hot water obtained from a single hot water storage tank is used to meet the requirement of space heating and DHW heating of each unit.

Thermal energy requirement for space heating is recovered from the 20°F temperature difference between the supply (160°F) and the return hot water temperature (140°F). A natural gas burner provided supplemental thermal energy to the space heating loads that were not met by the thermal energy obtained from the CHP facility. The DHW tank temperature was set at 120°F (ICC 2009a). Here too, the hot water tank was supplemented with a backup natural gas burner to maintain the setpoint temperature.

A hot water (HW) circulation pump circulated hot water from the thermal storage tank to an air-to-water heat exchanger centrally located in each apartment unit. Another hot circulation pump is required to circulate hot water from the thermal storage tank to the point of hot water storage for each unit. The pumps were assumed to operate at design conditions and consume a constant amount of energy whenever operational. Energy consumption of the hot water circulation pumps operating at part load conditions was ignored by this analysis.

The amount of hot water delivered for space heating and DHW storage tank is controlled by a thermostat located within the space or tank. The thermostat also controls the operation of aux. burners and pumps. A schematic diagram of the modified residential heating system is provided in the Figure 7-4 below.

In order to match the energy consumption of the multi-family units with the surplus thermal energy provided from the CHP facility, the hourly space heating and DHW loads were obtained from the DOE-2.1e output files of the original base-case model. These loads were then matched on an hourly basis by the surplus thermal energy available from the CHP facility. Energy consumption of natural gas burners along with the electricity consumption of HW pumps was added to the total hourly energy consumption of the multi-family building. Finally, hourly energy end-use for space heating and DHW of the original base-case were identified and subtracted from the hourly total energy usage of the multi-family building.

In summary, the above paragraphs have described the simulation model for multi-family residential units and the resulting electricity and thermal loads of each unit. Several such building models were considered when matching the surplus electricity and thermal energy generated by the CHP facility installed in the grocery store. In the next section the CHP model is discussed.

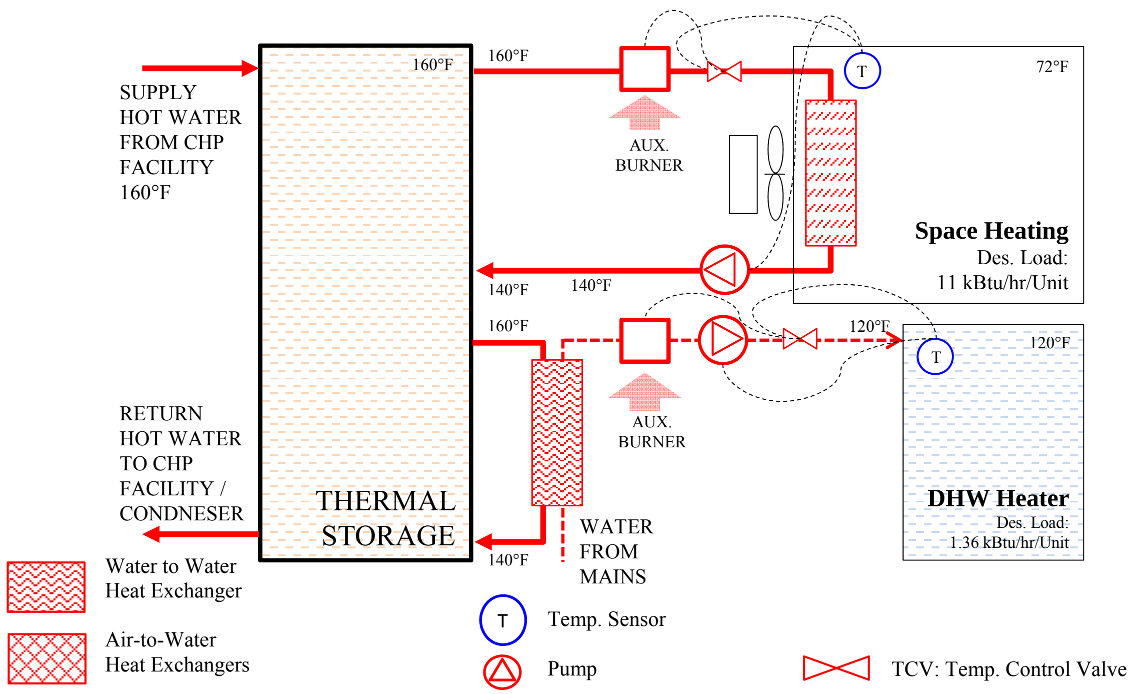

Figure 7-4: Schematic Layout of the MF Space Heating and DHW System

7.4 The CHP Model

7.4.1 Working of the CHP Model

The CHP model in this study assessed the impact of using the CHP facility to provide electricity and thermal energy to the grocery store[11]. A flow chart presenting the overall configuration of the model is presented in Figure 7-5 and Figure 7-6. The CHP model required an hourly input of loads for electricity consumption and thermal energy consumption from the grocery store. These hourly loads were obtained from the eQUEST-Refrigeration model of the grocery store. Depending on the options selected for the use of waste thermal energy generated from the prime mover, electricity consumption of the grocery store can include meeting loads from space cooling, refrigeration, lighting systems and/or miscellaneous equipment. Potential candidates for thermal energy usage can include loads from absorption refrigeration, space cooling, space heating and service water heating.

[11] The CHP model was created for this study using Microsoft Excel 2010 spreadsheet.

Electricity requirements were calculated by matching the power generated by the prime mover with the electric load requirements of the grocery store. Power generating capabilities of the prime mover depend on the number and type of engine / turbine, the operating mode[12] selected in the CHP model as well as ambient temperature conditions[13]. The CHP model then checked whether the building electric load requirements were met with the power generated from the prime mover selected in the model. Building electric load requirements not met by electricity generated by the prime mover were supplemented by electricity from the utilities.

Along with the generation of power, waste thermal energy was also produced with the operation of the prime mover. The amount of waste thermal energy available for recovery depends on manufacturers specifications for the prime mover such as temperatures and flow rates for exhaust air and engine jacket water coolant. These specifications were required to be input in the model. The CHP model then matched the thermal loads from the building with the thermal energy being generated by the prime mover. Modulation of thermal loads was performed by the use of supplementary boilers and duct burners, heat dumping[14] or the use of thermal storage[15].

The available thermal energy is also dependent on the quality of thermal energy required by the building loads and the heat recovery device selected as well as the operation mode of the prime mover. The CHP model was programmed to prioritize meeting the high temperature loads such as thermal energy required to operate absorption chillers first, followed by meeting the space heating loads and finally the service hot water loads.

In the first step for matching the thermal loads, the CHP model checked whether the building absorption chiller load requirements were met with the maximum thermal energy available from the prime mover at the required conditions. Building loads not met by the waste thermal energy from the prime mover were met by either an auxiliary boiler (i.e., when using an HRSG or exhaust gas to water heat exchangers) or by an auxiliary natural gas burner (i.e., when using direct fired absorption chiller technologies). Any surplus thermal energy was cascaded to the next level of thermal energy requirement. In the second step and third step, this matching process was performed for space heating requirements and then for the service hot water heating requirements of the grocery store model.

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[12] Operating mode of a CHP system have been described in the literature review of this study.
[13] Gas turbines are more susceptible than IC engines to variations in ambient temperature.
[14] Heat dumping refers to discharging the thermal energy when not being absorbed by the building loads.
[15] In such systems thermal energy is stored during periods of excessive production of thermal energy (Caton, 2010)

Finally, any surplus electrical and thermal energy remaining after meeting the electricity and service hot water requirements respectively of the grocery store were then 'wheeled' across the boundaries of the grocery store to the neighboring residential community. Surplus thermal energy was either retained by thermal storage tanks or dumped by heat rejection units. A flow chart presenting the overall configuration of the model is presented in Figure 7-5 and Figure 7-6 for hot water fired and direct fired absorption chillers respectively.

7.4.2 Assumptions in the CHP Model

Certain assumptions were made in order to run the CHP model. These include:

- The building systems in the analysis were assumed to be operating at steady-state conditions.
- Thermal energy from the exhaust gases generated by the IC engine was primarily made available to the absorption chillers. However, any excess energy not utilized by the absorption chillers was captured to be used for space heating purposes.
- Thermal energy obtained from the jacket coolant was made available to space heating and service hot water heating respectively.
- The CHP model is limited to assess a single configuration that had all the equipment connected in parallel to the hot water loops was considered for the analysis. Hot water provided for the operation of the system was assumed to be flowing at a pressure of 30 psia[16].
- 90% of the water to be heated for both absorption chillers and space heating is assumed from return line with corresponding return water conditions. The remaining 10% of water to be heated was at feed water conditions[17].
- The absorption refrigeration system operated for all the hours of the year (i.e., 8760 hours).
- All the surplus thermal energy is assumed to get absorbed either by the various thermal requirements of the grocery store or by the surrounding residential community. Thermal energy not being utilized by the residential community[18] is dumped to the surrounding by heat rejection units.
- Hot water pumps and heat rejection equipment were modeled to consume energy at design conditions.

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[16] Water supply from the municipality was assumed to be at a pressure of 30 psia.
[17] Feed water temperature is assumed to be the same as the ground temperatures obtained from the TMY-2 file for College Station, TX. The monthly ground temperatures are presented in Figure A-8, Appendix A.
[18] This includes thermal energy storage units.

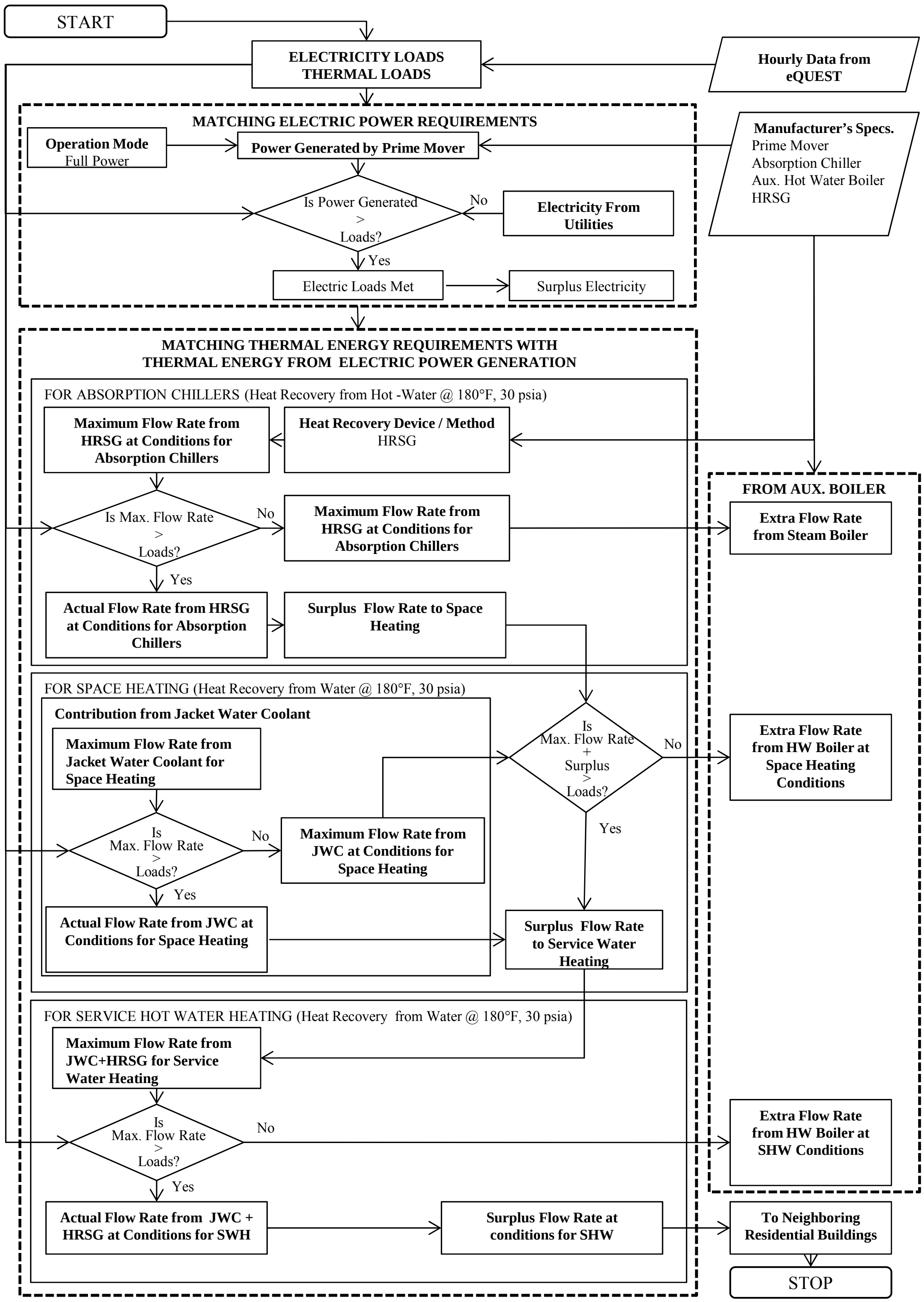

Figure 7-5: Flow Chart for the CHP Model (w/ Hot Water Fired Abs. Chiller)

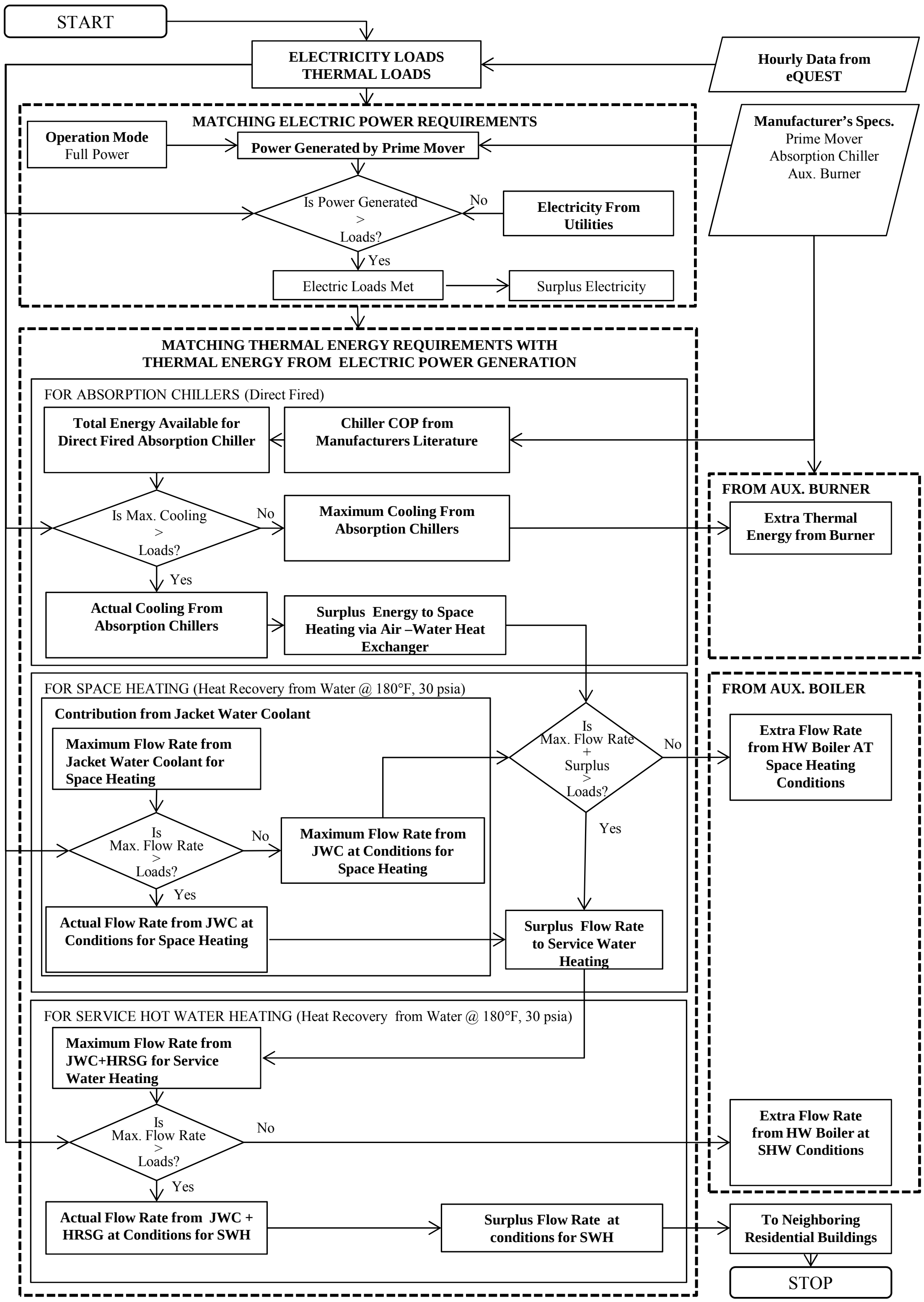

Figure 7-6: Flow Chart for the CHP Model (w/ Direct Fired Abs. Chiller)

7.4.3 Components of the CHP Model

Components of the CHP model include a prime mover, heat exchangers for space heating and service hot water heating, and a thermal storage system for storing surplus thermal energy from the grocery store. The components are described briefly below. Detailed descriptions of the equations that govern the working of the CHP model are provided in Appendix C of this study.

7.4.3.1 The Prime Mover

The main component of the CHP model is the prime mover, which is responsible for the generation of on-site electric power. Electrical equipment and balance of plant equipment The CHP model designed for this analysis requires the input of certain specifications for the prime mover. The specifications include:

- Net power (kW),
- Fuel rate (MMBtu/hr),
- Mass flow rate of exhaust air (lbm/s),
- Temperature of exhaust air (°F),
- Minimum temperature setting of the exhaust air at both inlet and outlet conditions (°F), and
- Mass flow rate of fuel (lbm/s).

Since CHP system was intended to primarily serve the grocery store, the type of prime mover was selected by calculating the heat-to-power[19] ratio of the store. For all the options considered by this analysis, the heat-to-power ratio obtained was below 1.5, which suggested the use of an IC engine[20] for electric power production. Specifications for the IC engines also include those for the jacket water coolant, which are:

- Coolant flow (lbm/s),
- Inlet and outlet temperatures of the jacket coolant (°F), and
- Recoverable thermal energy from the jacket water coolant at the determined temperature conditions (Btu/hr).

The specifications for the IC engine were obtained and compiled from manufacturer's literature. A list of specifications from different manufacturers for different sizes of IC engines is provided

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[19] As determined in the literature review section of this analysis, the heat-to-power ratio can be used as a preliminary indicator for the feasibility of the cogeneration system. The heat-to-power ratio is also used in the preliminary selection of an appropriate prime mover for the facility.

[20] As determined in the literature review section of this analysis, for facilities with heat-to-power ratios lower than 1.5, the selection of an IC engine is appropriate to meet the electric and thermal requirements.

in Appendix F of this study. The appendix includes information from IC engine manufacturers such as Man, Cummins, Caterpillar and Waukesha. In this study it was also determined that operating the CHP model at full mode provided the best results in terms of maximizing efficiency in terms of electricity generation. Hence, only results from the full power mode were considered for this analysis.

7.4.3.2 Absorption Chillers

Absorption chillers were modeled to utilize the thermal energy recovered from the CHP to meet cooling loads in the grocery store. In the case of a grocery store, absorption chillers can potentially be implemented in one or more of the following scenarios:

- Space cooling,
- Sub-cooling for vapor compression refrigeration systems, and
- A replacement for the medium and low temperature vapor refrigeration system.

LiBr/Water absorption chillers provide chilled water in the range of 40 – 50°F and are suited for space cooling and sub-cooling purposes. On the other hand, Water/NH_3 chillers can generate cooling temperature as low as – 60°F and can potentially be used to provide low and medium temperature refrigeration to the grocery store. Manufacturer's specifications for both the LiBr/Water[21] and Water/NH_3[22] chillers are provided in Appendix F of this study. Specifications are provided for both direct-fired and indirect-fired as well as single-stage and double-stage categories of absorption chillers. Specifications include rated COP, outlet chilled water temperature, electricity consumption for parasitic loads[23], cooling capacity and the quality of thermal energy required for the operation of the chiller.

The absorption chiller model implemented in this analysis modifies the COP of the absorption chillers at part load conditions of chiller capacity. The capacity of absorption chillers may either be controlled by flow rate of the hot media or its temperature, or the flow rate or temperature of heat rejection (Leavell, personal communication, July 2012). Specifications for the COP of absorption chillers at part load conditions are provided in Appendix F of this study.

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[21] For LiBr/Water chillers information from manufacturers include Broad, Yazaki, Carrier, Thermax, York, McQuay and Trane were assessed and presented in Appendix F.

[22] Water/ NH_3 chillers are typically custom built for each facility. As a result limited information was available from manufacturers' literature. Therefore, this study relied heavily on research performed by Dorgan et al. (1995) to obtain information for these chillers to be utilized in the simulation model. Robur was the only manufacturer which produced packaged Water/NH_3 chillers in the size range suitable for the purpose of this study.

[23] Parasitic loads for absorption chillers include the operation of solution and condensing water pumps as well as burners (as in the case of direct-fired absorption chillers.

The absorption chiller model implemented in this study utilizes a part-load ratio curve specified in the DOE-2.1e supplement (Winkelamnn et al. 1993)[24].

In addition, absorption chillers required a modest amount of electricity consumption in the operation of the absorption chiller machinery and associated electricity consumption of the heat rejection equipment. The operation of absorption chiller machinery included the energy consumption of the absorbent pumps, refrigerant pumps, purge pumps and burner blowers in the case of direct fired chillers. For this study, the energy consumption of all the machinery operating in the absorption machine was consolidated together and modeled as a constant quantity, which was added to the electricity requirements of the grocery store. Details of the electricity usage of absorption machinery were obtained from Dorgan et al., (1995) and are provided in Appendix F of this study.

For heat rejection machines, typically, the total amount of heat removed from absorber and condenser of the absorption chiller is 1.5 to 2 times the heat rejected from a corresponding vapor compression chiller (Dorgan et al. 1995). Quite often water-cooled condensers are used for a more efficient rejection of heat. Therefore, in this study water-cooled condensers replaced the air-cooled condensers operating in the grocery store base-case model to operate absorption chillers. For this study, the cooling tower in the absorption chiller model was modeled to consider the energy consumed by condenser water pumps and cooling tower fans. These fans and pumps were sized according to ARI conditions[25] (ARI 2000). Condenser water pumps are considered to operate under design load conditions, operation of these pumps at part load conditions is ignored in this analysis. On the other hand, cooling tower fans do not operate continuously, so a part-use factor (as reported in Dorgan et al. 1995) is used for the calculations. Details for sizing and subsequent energy consumption of the condenser fans and pumps are provided in Appendix F of this study.

Depending on the firing mode[26], the operation of the absorption chiller may require a back-up provision of thermal energy using either a natural gas burner, or a hot-water boiler, or a

[24] Part-load curves for both single-stage indirect fired and double-stage direct fired chillers have been presented in Appendix F. It should be noted that these specifications are LiBr/Water absorption chillers. Due to lack of information for Water/NH_3 chillers, the same specifications were assumed for Water/NH_3 chillers.

[25] ARI Standard 560, Absorption Water Chilling and Water Heating Packages (ARI 2000)

[26] Indirect firing modes implement the use of an HRSG to generate steam which in turn is directed to the generator section of the absorption machine. Direct firing modes involve the direct use of exhaust gas in the generator section of the absorption machine.

steam boiler[27]. In this study, a natural gas burner was specified along with direct-fired absorption chillers to provide backup energy for the chiller. A hot-water boiler was specified along with an indirect-fired absorption chiller to provide backup energy for the chiller. . These specifications were obtained from manufacturers' literature and are provided in Appendix F of this study. Parasitic energy consumption was modeled whenever the burner or boiler was in use.

In order for the absorption chiller model to meet the cooling loads from the grocery store model designed in eQUEST-Refrigeration, the hourly cooling loads for cooling and refrigeration (depending on the CHP option selected) first had to be identified. Hourly loads for space cooling, sub-cooling and refrigeration were extracted from the eQUEST-Refrigeration output file and matched with the hourly cooling capacity of the absorption chiller model. Then, using the COP[28] of the selected absorption chiller, an appropriately sized absorption chiller was selected to meet these loads on an hourly basis. In the CHP model adopted by this analysis, absorption chillers were simplistically modeled by multiplying the thermal energy available for cooling with a COP. When examining the performance of absorption chillers with smaller capacities, the COP was set at a constant value. For absorption chillers with larger capacities, the COP was modified using part load conditions. This decision reflected the specifications provided by the manufacturers of absorption chillers. Depending on the temperature range of the hourly loads required to be satisfied by the chiller, either LiBr/Water or Water/NH_3 absorption chillers were considered. LiBr/ Water chillers were considered when meeting space cooling and sub cooling loads that did not require temperature to be below freezing levels. On the other hand, when considering meeting the requirements from medium and low temperature refrigeration units that operate under sub-freezing conditions, Water/NH_3 absorption chillers were considered. In addition, both hot water-fired and direct-fired absorption chillers were considered for the analysis. Operating the chiller by the use of hot water was considered when implementing a single-effect LiBr/Water absorption chiller. On the other hand, operating the chiller by using the energy from the exhaust gases directly was considered when implementing double-effect[29]

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[27] Steam boilers have issues regarding maintenance and operation and are usually not considered on building scale level. Hence steam boilers are not considered by this study.

[28] The COP of the absorption chiller determines how much cooling is available from an input of thermal energy into the machine to meet the hourly loads established from the simulation model.

[29] Double-effect chillers require higher temperature than single effect absorption chillers.

LiBr/Water absorption chillers and Water/NH_3 absorption chillers[30]. Finally, the hourly end-use energy consumption of the original equipment in the eQUEST-Refrigeration model, which would be replaced by the absorption chiller was identified and subtracted from the total hourly electricity consumption of the grocery store model[31].

7.4.3.3 Hydronic Space Heating and Cooling Model[32]

Depending on the CHP option selected, the packaged air-conditioner and furnace model installed in the base-case grocery store model were replaced with a central hot water and chilled water piping system to permit the utilization of thermal energy obtained from the CHP installation in the store.

The hydronic model was designed to meet all the space heating and cooling loads by either utilizing thermal energy from the CHP facility or from auxiliary sources such as a hot water boiler. The heating and cooling loads as well as the energy end-use of the cooling and heating equipment for all the zones were obtained from the grocery store building modeled in eQUEST-Refrigeration. Appendix H provides additional information about the hourly reports extracted from eQUEST-Refrigeration program which were used in the analysis.

The space cooling loads on the chilled water system were met on an hourly basis by absorption chillers as discussed in the sub section on absorption chillers. Space heating loads were met by a hot water loop with a supply temperature of 180°F and a return temperature of 140°F. Hot water was provided from the waste energy generated by the CHP facility, which in this case was the thermal energy available from the engine jacket coolant. Hot water and chilled water pumps were required to be included in the simulation in order to circulate the hot water and chilled water to different terminals in the grocery store. Electricity consumption of these pumps was considered in the assessment of the total energy consumption. In calculating the power requirements of the pumps in the hydronic system, certain assumptions had to be made. The efficiency of the pumps was assumed to be 65%, which is in line with the standard practice. A pump head of 5ft of water was assumed. The pump installed in the in the hot water loop that was coupled with the jacket coolant loop was modeled to operate at all times. The hot water loop

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[30] Water/NH_3 chillers are typically used to provide cooling at lower and below freezing temperatures. When considering absorption chillers lower cooling temperatures require higher temperatures in the generator section of the absorption chiller.
[31] Refer to Appendix H to identify the hourly reports extracted from eQUEST-Refrigeration simulation program.
[32] Hydronic heating and cooling systems use water to move thermal energy from where it is produced to where it is needed (Siegenthaler, 2004).

that supplied hot water to the space heating terminal and to meet the hot water requirements was modeled to operate when heat was required. Pump performance curves were not considered for this analysis and the pumps were modeled to operate at design flow conditions.

Hourly energy end-use for space heating in the original base-case model was substituted with the thermal energy obtained from the CHP facility. Hourly end-use energy consumption for space heating and space cooling of the packaged roof-top air-conditioning units in the grocery store model were subtracted from the overall hourly electricity consumption of the grocery store. The additional electricity use associated with the operation of chilled water and hot water pumps was added to the overall hourly electricity consumption of the grocery store.

7.4.3.4 Heat Exchangers

The generation of electricity using an IC engine creates waste heat in form of exhaust gases and the jacket coolant. Exhaust gases are used to extract medium to high grade thermal energy such as the energy required to operate absorption chillers. On the other hand, energy extracted from jacket coolant falls in the range of medium to low quality, which is usually used for space heating or service hot water heating purposes. In this analysis, thermal energy from the waste heat generated by the prime mover was transferred using air-to-water heat exchangers[33] as well as by means of water-to-water heat exchangers[34]. Heat exchangers implemented in this analysis were modeled using basic energy and mass balance equations executed on an hourly basis. Details of these equations are presented in Appendix C.

In addition to the mass flow rate of the water circulating between the supply and return temperatures in the heat exchangers of the grocery store, a small stream of water is provided to supply the feed water at the temperature of the water mains to compensate for any leaks the system may have as well as to maintain an acceptable mineral composition of circulating water[35]. This stream of water accounts for 10% of the total mass flow rate of water flowing through the heat exchangers.

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[33] As in the case of heat transfer from exhaust gas to a coolant circulated in a coil inserted into the exhaust stream.
[34] As in the case of heat transfer from jacket coolant to hot water.
[35] Feed water contains impurities. If the suspended solids are allowed to concentrate beyond certain limits, deposits on surfaces of pipes and heat exchangers will form, which will retard the performance of the heat exchanger equipment and in some cases failure of the equipment. The concentration of suspended solids is controlled by replacing some amount of water circulating in the system with makeup water with lower concentration of impurities. (Payne and Thompson 1996). However, it should be noted that hot water systems experience less issues with increased concentrations than steam systems.

Space cooling, sub-cooling and refrigeration loads were primarily met by absorption chillers that utilized the thermal energy obtained from the exhaust gas generated by the IC engine. Space heating loads and service hot water heating loads in the grocery store were primarily met by the heat rejected from the engine to the jacket water and oil cooler. The CHP model was configured to prioritize thermal loads in the order of highest to lowest temperature with higher temperature loads gaining priority over lower temperature loads. Thermal energy was first used to meet the loads for absorption chillers, then used to meet loads for space heating and finally to meet loads for domestic hot water heating. Any surplus energy that was obtained after meeting the requirements of the grocery store was redirected to meet the space heating and domestic hot water heating requirements for residential multi-family units. Space heating was provided at a temperature of 160°F. DHW storage tank was heated to a temperature of 120°F.

7.4.3.5 Auxiliary Hot Water Boilers

A natural gas fired hot-water boiler was modeled in the CHP model to provide backup thermal energy to the operation of the proposed hot water fired absorption chillers and the hot water system implemented for space heating. The boiler was sized to meet the loads that were not met by the waste thermal energy generated by the CHP system implemented in the grocery store. To accomplish this, a condensing hot water boiler with an efficiency of 95% was specified for this purpose[36]. The boilers operated on a curve-fit that corrects the Heat Input Ratio (HIR)[37] of the boilers as a function of part load ratio (Hirsch 2008). The equation utilized in the curve-fit is described in Appendix C.

7.4.3.6 Thermal Storage

Finally, on meeting all the requirements of the grocery store, any surplus thermal energy available from electricity and thermal energy was wheeled across the boundary of the grocery store to meet the thermal energy requirements of multi-family units. Due to the non-coincidence of surplus thermal energy available from the grocery store and the energy requirements of the residential units an appropriate thermal storage device was modeled in form of a hot water thermal storage tank. Modeling storage for electricity, although feasible, was outside the scope of this study. The hot water thermal energy storage tank was modeled to store the surplus

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[36] The efficiency was selected from a boiler listed in the AHRI directory of certified product performance (AHRI 2012). The capacity of the boiler was assumed based on the option selected for the analysis.
[37] Heat Input Ratio (HIR) = Energy Input /Heat Output

thermal energy available from the CHP facility when not being used by either the grocery store or for residential purposes. The thermal storage system is designed to retain thermal energy from the surplus hot water available from the grocery store for up to 24 hours. The capacity of the storage tank was optimized by considering various factors such as the number of units being served, the coincidence of surplus thermal energy and the requirements of the residential units, and energy required from the supplemental natural gas burner used when energy requirements of the residential units exceed the provision of surplus thermal energy from the grocery store. Details of the assumptions and calculations are provided in Appendix C of this report.

7.5 Options for the CHP Model

Based on the options available for utilizing the waste thermal energy generated as a byproduct of electricity generation by the on-site cogeneration facility, four options were considered for the analysis. The four options were grocery store centric, which implies that the requirements of the grocery store were always considered first. Surplus energy was then exported across the boundary of the store to be used for residential energy consumption.

7.5.1 Description of the Options Considered by the Analysis

7.5.1.1 Option 1: Using LiBr / Water Absorption Refrigeration for the Sub-Cooling of the Refrigerant

7.5.1.1.1 Base-Case Description

In Option 1, the loads from the following components in the grocery store model were served by the waste thermal energy generated by the CHP system:

- Mechanical sub-cooler,
- Space heating, and
- Service water heating.

The base-case grocery store model for the analysis of Option 1 included most of the efficiency measures that were incorporated in the cumulative assessment of the energy efficiency measures in Section 6.3.5 of this study. The excluded EEM is that for the improved efficiency of packaged furnaces. This is because space heating was provided by the installed CHP system. The resultant electricity and thermal energy consumption patterns of the grocery store as reported for the end-uses are presented below:

- For electricity[38] consumption of the grocery store, the base-load (50th percentile) was determined to be 230 kW. The peak electricity consumption was determined to be 342 kW.
- For the mechanical sub-cooler, the base-load (50th percentile) was 63.6 kBtu/hr and the maximum load was 160.6 kBtu/hr.
- For space heating, the base-load (50th percentile) was 86.9 kBtu/hr and the maximum (99th percentile) load was 471.6 kBtu/hr.
- For service water heating, the base-load (50th percentile) was 14.2 kBtu/hr and the maximum load was 39.6 kBtu/hr.
- The heat-to-power ratio of the grocery store with this mode of operation was calculated to be 0.25.

Details of the load profiles for electricity and thermal energy consumption of the store resulting from the implementation of this option are reported in Appendix F of this analysis and are discussed in the Section 7.5.2, which presents the results for this analysis.

7.5.1.1.2 CHP System Specifications

The heat-to-power ratio for this option was calculated to be 0.25, implying that an IC engine would be best suited for this application. A 300 kW (Cummins, CUM SCG300)[39] capacity engine was selected for the simulation. The IC engine selected provided 94% of the electricity required by the store with a potential of exporting electricity to the surrounding residential units when not utilized by the store. Electricity requirements include space cooling, lighting and miscellaneous equipment, as well as loads from medium and low temperature refrigeration. The waste thermal energy generated from the engine met all the designated thermal energy requirements of the grocery store, which in this case was to provide sub-cooling to medium and low temperature absorption refrigeration units, space heating and service hot water heating.

According to the specifications for the IC engine, exhaust gas was generated at a temperature of 1,202°F and a mass flow rate of 0.84 lbm/s. The exit temperature of exhaust gases was set at 250°F[40] .The exhaust was driven through an air to water heat exchanger to

[38] It should be noted that since the loads from the mechanical sub-cooler were met by the absorption chiller operated by the CHP, the electricity consumption of the mechanical sub-cooler in the base-case simulation model were first extracted from the hourly output report and then subtracted from the hourly whole building electricity consumption.
[39] Specifications of the IC engine are provided in Appendix G of this study.
[40] Exhaust gas exit temperatures lower than 250°F can cause condensate on the walls of the exhaust flue that cause corrosion.

produce hot water at 180°F and 30 psia[41]. A hot-water fired single-effect LiBr/Water absorption refrigeration system[42] was installed alongside an air-to-water heat exchanger to take advantage of the hot water produced by the heat exchanger. The LiBr/Water absorption chiller provided sub-cooling to the refrigerant from medium and low temperature refrigeration units. The LiBr/Water absorption chiller has an inlet hot water temperature of 180°F and outlet hot water temperature of 140°F. The mechanical sub-cooler provided chilled water at 45°F. According to the manufacturer's data[43,44], the COP of the absorption chiller is 0.7 and the capacity of the absorption chiller is estimated to be 15 tons. After meeting the requirements of the sub-cooler, surplus hot water is used for space heating and service hot water (SHW) heating respectively.

Space heating loads and service hot water heating loads in the store are primarily met by the heat rejected from the engine to the jacket water and oil cooler. In the manufacturer's specifications for the jacket water coolant of the IC engine, the estimated coolant flow for the selected engine is 6.3 lb/s. The operating range of the coolant is 203°F (exiting temperature) and 188°F (entering temperature). For the IC engine selected in this option, the manufacturer also provided the heat rejection to jacket water and oil cooler which is given at 341,214 Btu/hr. Heat obtained from the jacket water coolant in the IC engine is used generate hot water at 180°F for space heating. Thermal energy required for space heating is obtained from water circulating between 180°F and 140°F. Hot water at 180°F not utilized by absorption chillers and space heating is then diverted to meet the service water heating loads of the grocery store. The temperature of service hot water provided to the store is set at 120°F. Surplus electricity and thermal energy from the grocery store is wheeled over the boundary of the store to meet the requirements of neighboring multi-family units. Calculations for electricity consumption from pumps and heat rejection units are provided in Appendix C of this study. A diagram of the proposed option is presented in Figure 7-7 below.

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[41] The 30 psia pressure is assumed to be the pressure at which the water is supplied from the municipality.
[42] Specifications of the LiBr/Water absorption chiller are provided in Appendix X of this study.
[43] See Appendix G for specifications of absorption chillers.
[44] See Appendix G for corresponding electric consumption of absorption chillers.

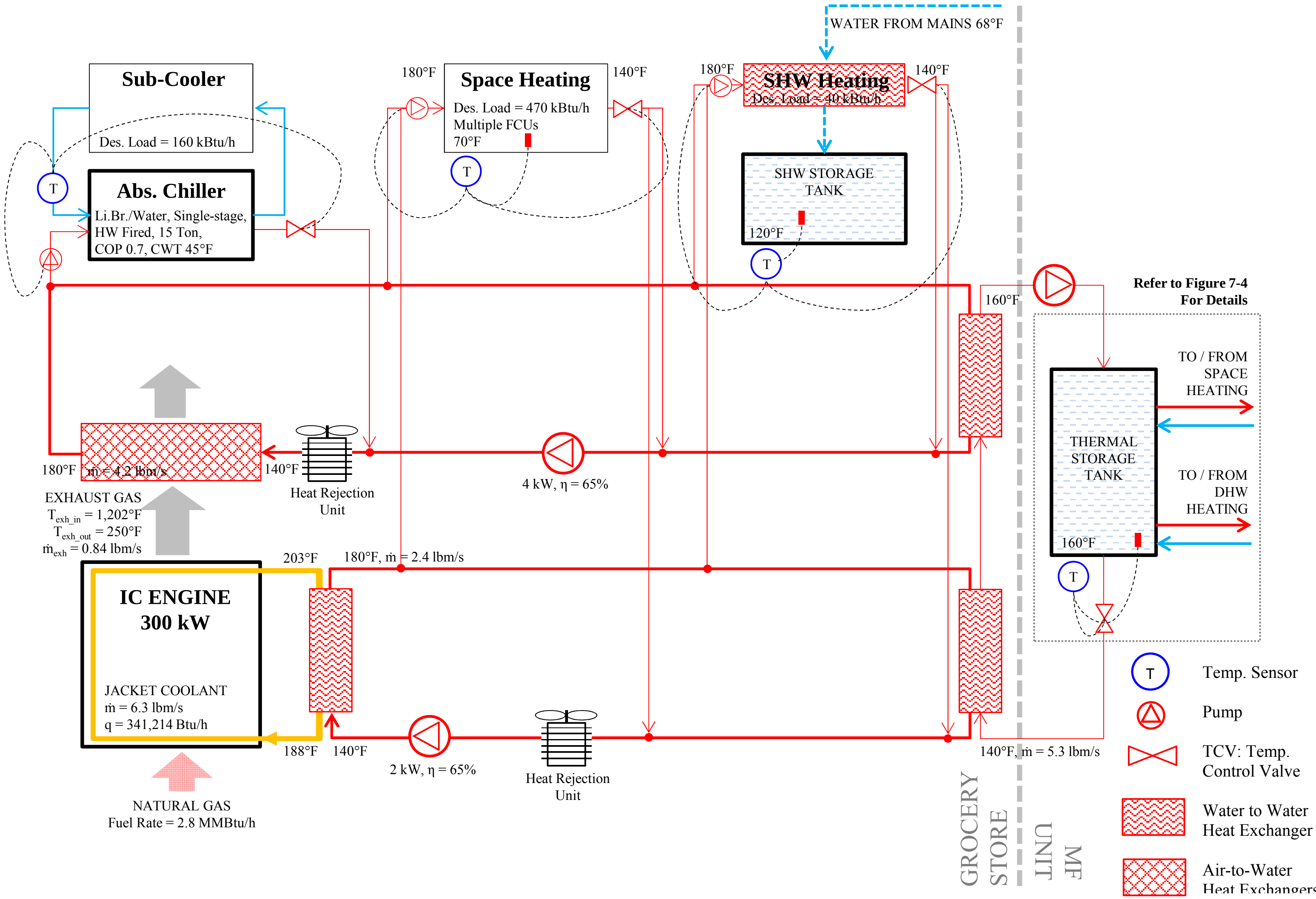

Figure 7-7: Thermal Energy Distribution Diagram for Option 1

7.5.1.2 Option 2: Using LiBr/Water Absorption Refrigeration for Sub-cooler Operation and Space Cooling

7.5.1.2.1 Base-Case Grocery Model Description

In Option 2, the loads from the following components in the grocery store model were met by the waste thermal energy generated by the CHP system:

- Mechanical sub-cooler,
- Space cooling,
- Space heating, and
- Service water heating.

The base-case grocery store model was altered to exclude the EEM that addressed the improvement of cooling and heating efficiency in the cumulative assessment of the EEMs in Section 6.3.5. The excluded EEMs are listed below:

- Improved efficiency of packaged air-conditioning units, and
- Improved efficiency of packaged furnaces.

The resultant electricity and thermal energy consumption patterns of the grocery as reported for the end-uses in the building model are presented below:

- For the electricity consumption of the grocery store, the base-load (50^{th} percentile) was determined to be 183 kW. The peak electricity consumption was determined to be 286 kW.
- For the absorption chiller operating the sub-cooler, the base-load (50^{th} percentile) was 68.6 kBtu/hr and the maximum load was 160.6 kBtu/hr.
- For the absorption chiller operating the space cooling system, the base-load (50^{th} percentile) was 355.6 kBtu/hr and the maximum load was 1,544.7 kBtu/hr.
- For space heating, the base-load (50^{th} percentile) was 86.9 kBtu/hr and the maximum (99^{th} percentile) load was 471.6 kBtu/hr.
- For service water heating, the base-load (50^{th} percentile) was 14.2 kBtu/hr and the maximum load was 39.6 kBtu/hr.
- The heat-to-power ratio of the grocery store with this mode of operation was calculated to be 1.03.

Details of the load profiles for electricity and thermal energy consumption of the store resulting from the implementation of this option are reported in Appendix H of this analysis and are discussed in the subsections on annual and hourly results.

7.5.1.2.2 CHP System Specifications

The heat-to-power ratio of 1.03 implied that an IC engine would be best suited for this application. A 300 kW (Cummins, CUM SCG300) capacity IC engine was selected for the simulation. This engine provided 100% of the electricity required by the store with a potential of exporting electricity to surrounding residential units when not utilized by the store. Electricity requirements include the energy requirements for medium and low temperature refrigeration, lighting and miscellaneous equipment loads, space cooling and heating fans and the auxiliary power requirements for absorption chillers. The thermal waste energy generated from the engine meets all the thermal energy requirements of the grocery store. This includes providing sub-cooling to the refrigerant from medium and low temperature units as well as provisions for space cooling.

A direct-fired, double-effect LiBr / Water absorption chiller was installed alongside the IC engine[45] to take advantage of the thermal energy wasted in form of exhaust gas generated by the engine. The chiller provided chilled water at 45°F, which was used for sub-cooling the refrigerant and for space cooling purposes. According to the manufacturer's data, the COP of the absorption chiller is 1.4. The capacity of the absorption chiller is determined to be 140 tons. The COP of the absorption chiller is modified as a function of part-load conditions.

In the specifications for the IC engine, exhaust gas is generated at a temperature of 1,202°F at a mass flow rate of 0.84 lb/s. The exhaust gas is first driven through the generator section of the direct-fired absorption chiller. After meeting the requirements of the absorption chiller, the exhaust gas is then driven through an air-to-water heat exchanger that produces hot water at 180°F. The hot water is then used for space heating and service water heating.

[45] Specifications of the IC engine and the direct-fired double effect LiBr/Water chiller are provided in Appendix F and Appendix G respectively.

Space heating loads and service hot water heating loads in the store are primarily met by the heat rejected from the engine to the jacket water and oil cooler. In the manufacturer's specifications for the jacket water coolant of the IC engine, the estimated coolant flow for the selected engine is 6.3 lb/s. The operating range of the coolant is 203°F (exiting temperature) and 188 °F (entering temperature). For the IC engine selected in this option, the manufacturer also provided the heat rejection to jacket water and oil cooler which is given at 341,214 Btu/hr. Heat obtained from the jacket water coolant in the IC engine is used generate hot water at 180°F for space heating. Thermal energy required for space heating is obtained from water circulating between 180°F and 140°F. Hot water at 180°F not utilized by absorption chillers and space heating is then diverted to meet the service water heating loads of the grocery store. The temperature of service hot water provided to the store is set at 120°F.

Finally, on meeting all the requirements of the grocery store, any surplus thermal energy available from either the exhaust gas or the jacket water coolant is wheeled across the boundary of the grocery store to meet the thermal energy requirements of multi-family units. Surplus hot water is supplied at a temperature of 180°F with a return temperature of 140°F. Space heating for multi-family units is provided at a temperature of 160°F. Domestic hot water is heated to a temperature of 120°F. Hot water not utilized by residences is stored in thermal storage tanks before being returned to the CHP facility. Calculations for electricity consumption from pumps and heat rejection units are provided in Appendix C of this study. A diagram of the proposed option is presented in Figure 7-8 below.

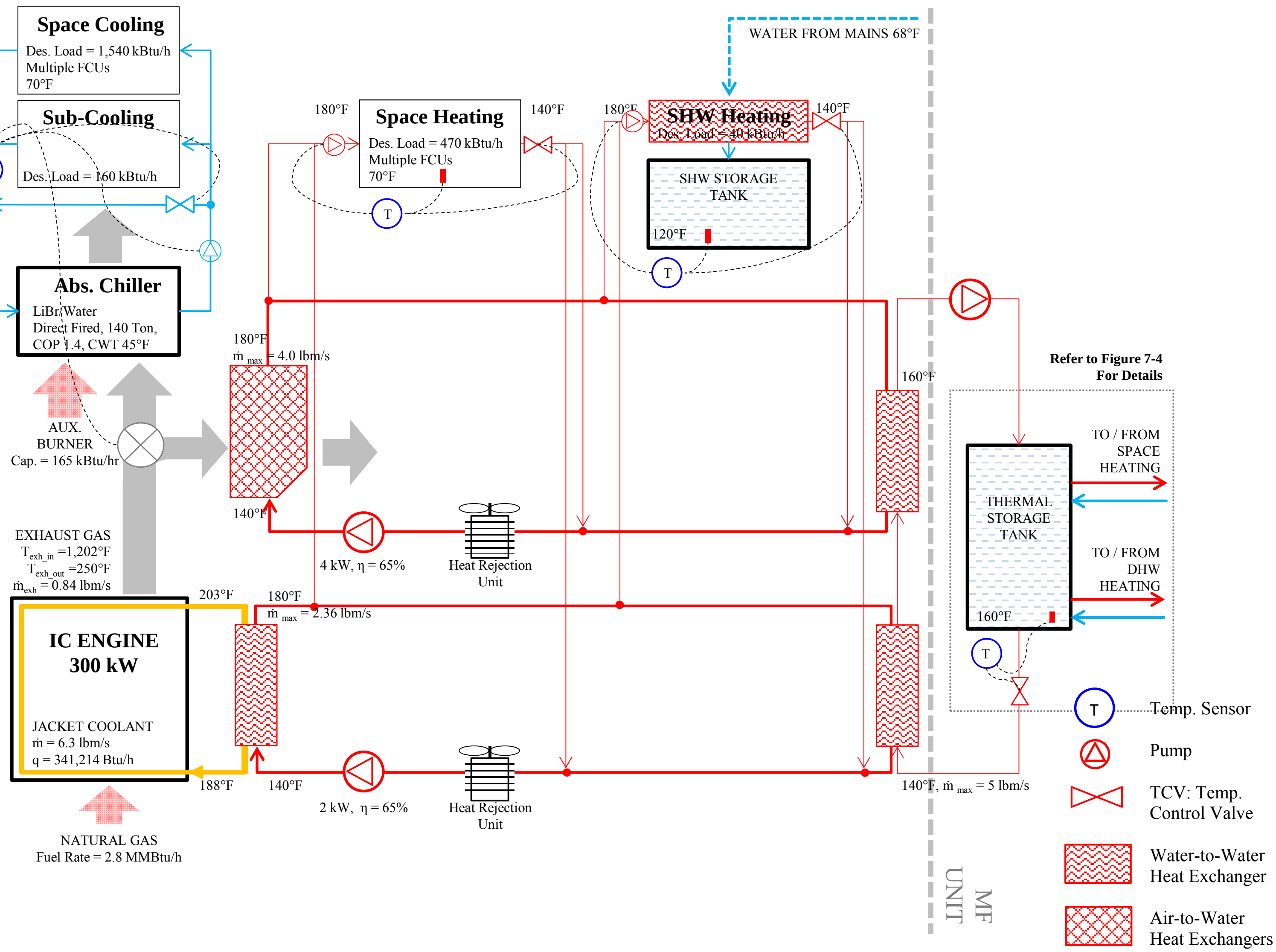

Figure 7-8: Thermal Energy Distribution Diagram for Option 2

7.5.1.3 Option 3: Using Water/NH_3 Absorption Refrigeration for Meeting Medium Temperature Refrigeration Loads

7.5.1.3.1 Base-Case Grocery Store Model Description

In Option 3, the loads from the following components in the grocery store model were met by the waste thermal energy generated by the CHP system:

- Mechanical sub-cooler (for low temperature refrigeration),
- Medium temperature refrigeration,
- Space heating, and
- Service water heating.

The base-case grocery store model was altered to exclude EEMs that addressed the medium temperature refrigeration system. The excluded EEMs are listed below:

- Medium temperature suction group temperature control,
- Medium temperature suction group compressor capacity control,
- Implementing floating temperature setpoints in condensers, and
- Implementing improved fan motors in condensers.

The resultant electricity and thermal energy consumption patterns of the grocery store were calculated to be:

- For electricity consumption of the grocery store, the base-load (50^{th} percentile) was determined to be 209 kW. The peak electricity consumption was determined to be 324 kW.
- For the sub-cooler the base-load (50^{th} percentile) was 20.1 kBtu/hr and the maximum load was 41.0 kBtu/hr.
- For medium temperature absorption chillers the base-load (50^{th} percentile) was 406.2 kBtu/hr and the maximum load was 556.3 kBtu/hr.
- For space heating, the base-load (50^{th} percentile) was 86.9 kBtu/hr and the maximum (99^{th} percentile) load was 471.6 kBtu/hr.
- For service water heating, the base-load (50^{th} percentile) was 14.2 kBtu/hr and the maximum load was 39.6 kBtu/hr.
- The heat-to-power ratio of the grocery store with this mode of operation was calculated to be 0.76.

Details of the load profiles for electricity and thermal energy consumption of the store resulting from the implementation of the sub-cooler are reported in Appendix H of this analysis and are discussed in the subsection on results.

7.5.1.3.2 CHP System Specifications

The heat-to-power ratio of the loads met by this option were calculated to be 0.76, implying that an IC engine would be best suited for this application. A 300 kW (Cummins, CUM SCG300) capacity engine was selected for the simulation. This engine provided 99% of the electricity required by the store with a potential of exporting electricity to surrounding residential units when not utilized by the store. The grocery store electricity requirements included space cooling, medium temperature refrigeration and sub-cooling, lighting and miscellaneous equipment loads. The waste thermal energy generated from the engine met all the thermal energy requirements of the grocery store. These included meeting the medium temperature loads and sub-cooling requirements of the low temperature refrigeration system as well as the space heating and service hot water heating loads. For this case too, there was the potential of exporting thermal energy to the surrounding residential units when not utilized by the store. For maximum efficiency, the IC engine was operated at full power mode.

A direct-fired single-effect Water/NH_3 absorption chiller was installed alongside the IC engine[46] to take advantage of the thermal energy wasted in the form of exhaust gas generated by the engine. Lower working temperatures in the range of 16°F were required in this option. As a result the COP of the absorption chiller was calculated to be 0.6 and the capacity of the absorption chiller is determined to be 50 tons. The COP of the absorption chiller varied as a function of part-load conditions[47].

According to the specifications for the IC engine, exhaust gas was generated at a temperature of 1,202°F and at a mass flow rate of 0.84 lb/s. The exhaust gas was first directed through the direct-fired absorption chiller. The lowest refrigerant temperature provided by the Water / NH_3 absorption chiller selected for this option is set at 16°F. After meeting the cooling load requirements, the remaining quantity of the exhaust gas is driven through an air-to-water

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[46] Specifications of the IC engine and the direct-fired single effect Water/NH3 chiller are provided in Appendix F and Appendix G respectively.

[47] Since no specifications could be obtained for Water/NH3 absorption chillers at part-load conditons, part-load curves similar to single-effect LiBr/Water chiller curves have been used.

heat exchanger with a resultant production of hot water at 180°F. The hot water was used for the space heating and service water heating loads.

Space heating loads and service hot water heating loads in the store were primarily met by the heat rejected from the engine to the jacket coolant loop and the oil cooler. The thermal energy required for space heating and service water heating was obtained from water circulating between 180°F and 140°F. Heat obtained from the jacket water coolant in the IC engine was used to generate hot water at 180°F for space heating. According to the manufacturer's specifications for the jacket water coolant of the IC engine, the estimated coolant flow for the selected engine was at 6.3 lb/s. The coolant temperature change was estimated to be from 188 °F to 203°F. For the engine selection, the manufacturer also provided the heat rejection to jacket water and oil cooler which was given at 341,215 Btu/hr. Surplus hot water not utilized by space heating was then diverted to meet the service hot water heating loads of the grocery store. The temperature of hot water provided to the store was set at 120°F.

Finally, on meeting all the requirements of the grocery store, any surplus thermal energy available from either the air-to-water heat exchanger or the jacket water coolant was wheeled across the boundary of the grocery store to meet the thermal energy requirements of multi-family units. Calculations for electricity consumption from pumps and heat rejection units are provided in Appendix C of this study. A diagram of the proposed option is presented in Figure 7-9 below.

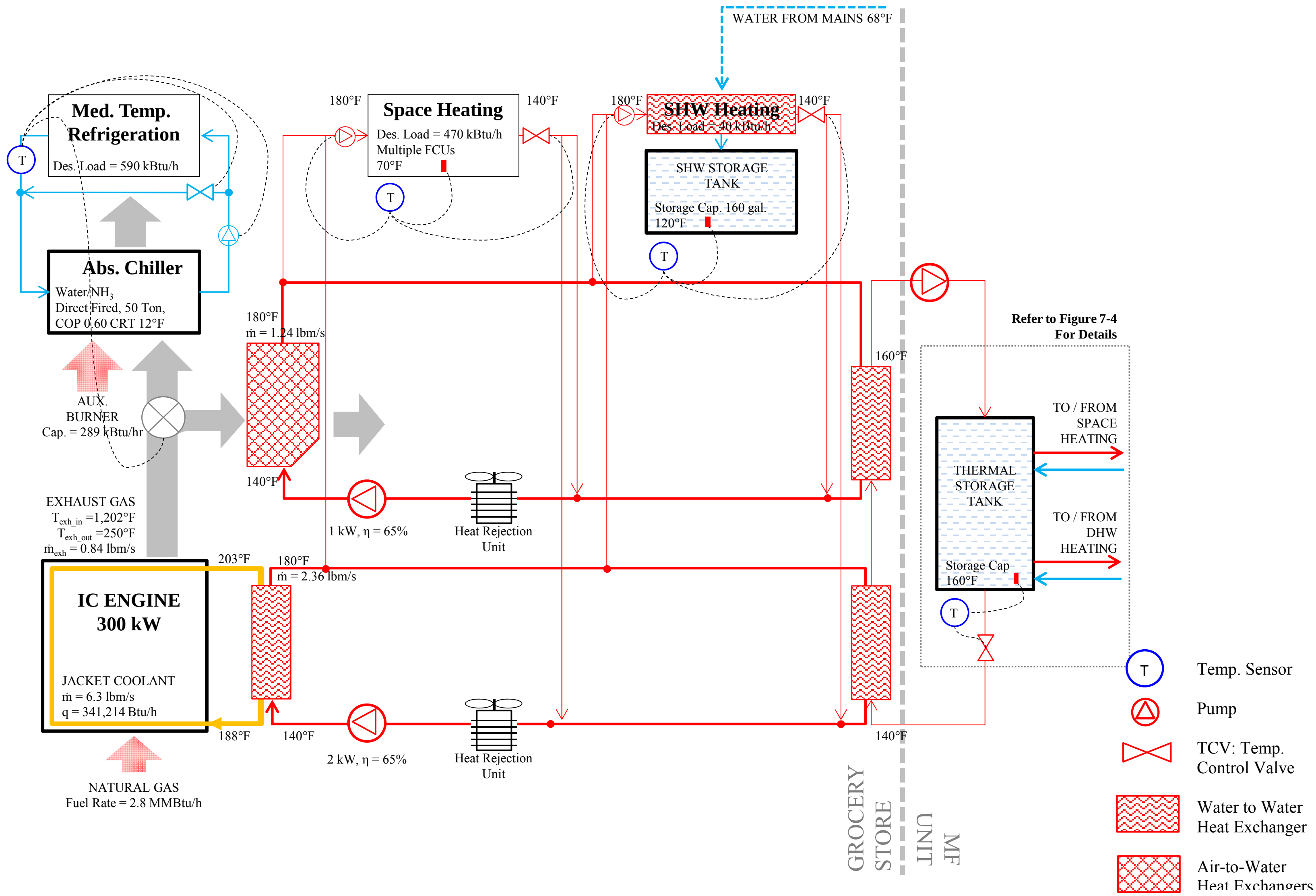

Figure 7-9: Thermal Energy Distribution Diagram for Option 3

7.5.1.4 *Option 4: Using Water/NH_3 Absorption Chillers for Medium and Low Temperature Refrigeration Units*

7.5.1.4.1 Analysis of Building Loads and Building Operation

In Option 4, the loads from the following components in the grocery store model were met by the waste thermal energy generated by the CHP system:

- Medium temperature refrigeration,
- Low temperature refrigeration,
- Space heating, and
- Service water heating.

The base-case grocery store model was altered to exclude EEMs that addresses medium and low temperature refrigeration systems. The excluded EEMs are listed below:

- Improved efficiency of packaged furnaces,
- Installing a sub-cooler,
- Suction group temperature control,
- Compressor capacity control,
- Floating temperature setpoints in air-cooled condensers, and
- Improved fan motors in air-cooled condensers.

The resultant electricity and thermal energy consumption patterns of the grocery store were assessed and reported below:

- For electricity consumption of the grocery store, the base-load (50th percentile) was determined to be 187 kW. The peak electricity consumption was determined to be 287 kW.
- For low temperature suction groups the base-load (50th percentile) was 0.14 MMBtu/hr and the maximum load was 0.18 MMBtu/hr.
- For medium temperature suction groups the base-load (50th percentile) was 0.41 MMBtu/hr and the maximum load was 0.56 MMBtu/hr.
- For space heating the base-load (50th percentile) was 0.09 MMBtu/hr and the maximum load was 0.79 MMBtu/hr.
- For service water heating the base-load (50th percentile) was 0.01 MMBtu/hr and the maximum load was 0.04 MMBtu/hr.
- The heat to power ratio of the grocery store with this mode of operation was calculated to be 1.09.

The refrigeration demand and space conditions were set to be for 24 hours a day year around. The load profiles for electricity and thermal energy consumption of the store are reported in Appendix H of this analysis and are discussed in the subsection on results.

7.5.1.4.2 CHP System Specifications

The heat-to-power ratio of the loads met by this option was calculated to be 1.09, implying that an IC engine would be best suited for this application. A 300 kW (Cummins, CUM SCG300) capacity engine was selected for the simulation. This engine provided 99% of the electricity required by the store with a potential of exporting electricity to surrounding residential units when not utilized by the store. The waste thermal energy generated from the engine met most of the thermal energy requirements of the grocery store. In addition, for this case, there was a potential of exporting thermal energy to the surrounding residential units when not utilized by the store. For maximum efficiency, the IC engine was operated at full power mode.

A direct-fired single-effect Water / NH_3 absorption chiller was installed alongside the engine to take advantage of the waste thermal energy in form of exhaust gas generated by the engine. The COP of the absorption chiller was calculated to be 0.47[48] and the capacity of the absorption chiller was determined to be 65 tons. Electricity requirements for auxiliary equipment such as heat rejection equipment, solution pumps and condensing water pumps are calculated and added to the overall electricity consumption of the grocery store[49]. In addition, an auxiliary burner was installed to supplement the energy requirements of the absorption chiller when the loads could not be met by energy captured from the exhaust gases.

According to the specifications for the IC engine, exhaust gas is generated at a temperature of 1,202°F at a mass flow rate of 0.84 lb/s. The exhaust gas was driven through the direct-fired absorption chiller. At design conditions the absorption chiller provides refrigerant at -24°F[50] for low and medium temperature refrigeration units in the grocery store.

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[48] Manufacturer's data was not available for Water / NH_3 chillers as these chillers are usually custom built on site. Therefore, the calculation of the COP was based on the temperatures required for low and medium temperature refrigeration systems. The calculations have been adopted from Dorgan et al. (1995) and are reported in Appendix G of this study.

[49] These calculations are adopted from Dorgan et al. (1995) and presented in Appendix G of this study.

[50] Suction temperatures of low and medium temperature refrigeration racks for the base-case grocery store are provided in Table A-9 of Appendix A. The refrigerant was required to maintain the set-point temperature of the refrigerated display cases in the grocery store. Both suction and evaporator temperatures of the display-cases served by the refrigerant are provided in Table A-12 of Appendix-A.

Space heating loads and service hot water heating loads in the store were primarily met by the heat rejected from the engine to the jacket water and oil cooler. Thermal energy required for space heating and service water heating was calculated from water circulating between 140°F and180°F. Heat obtained from the jacket water coolant in the IC engine was used generate hot water at 180°F for space heating. According to manufacturer's specifications for the jacket water coolant of the IC engine, the estimated coolant flow for the selected engine was 6.3 lbs. /s. The coolant temperature change is provided to be from 188 °F to 203°F. For this engine selection, the manufacturer also provided the heat rejection to jacket water and oil cooler, which is given at 341,214 Btu/hr. Surplus hot water not utilized by space heating was then diverted to meet the service hot water heating loads of the grocery store. The temperature of hot water provided to the store was set at 120°F. Electricity requirements for pumps and heat rejection units were calculated on an hourly basis and added to the overall electricity consumption of the grocery store.

Finally, on meeting all the requirements of the grocery store, any surplus thermal energy available from either the exhaust gas or the jacket water coolant was wheeled across the boundary of the grocery store to meet the thermal energy requirements of multi-family units. A diagram of the proposed option is presented in Figure 7-10 below.

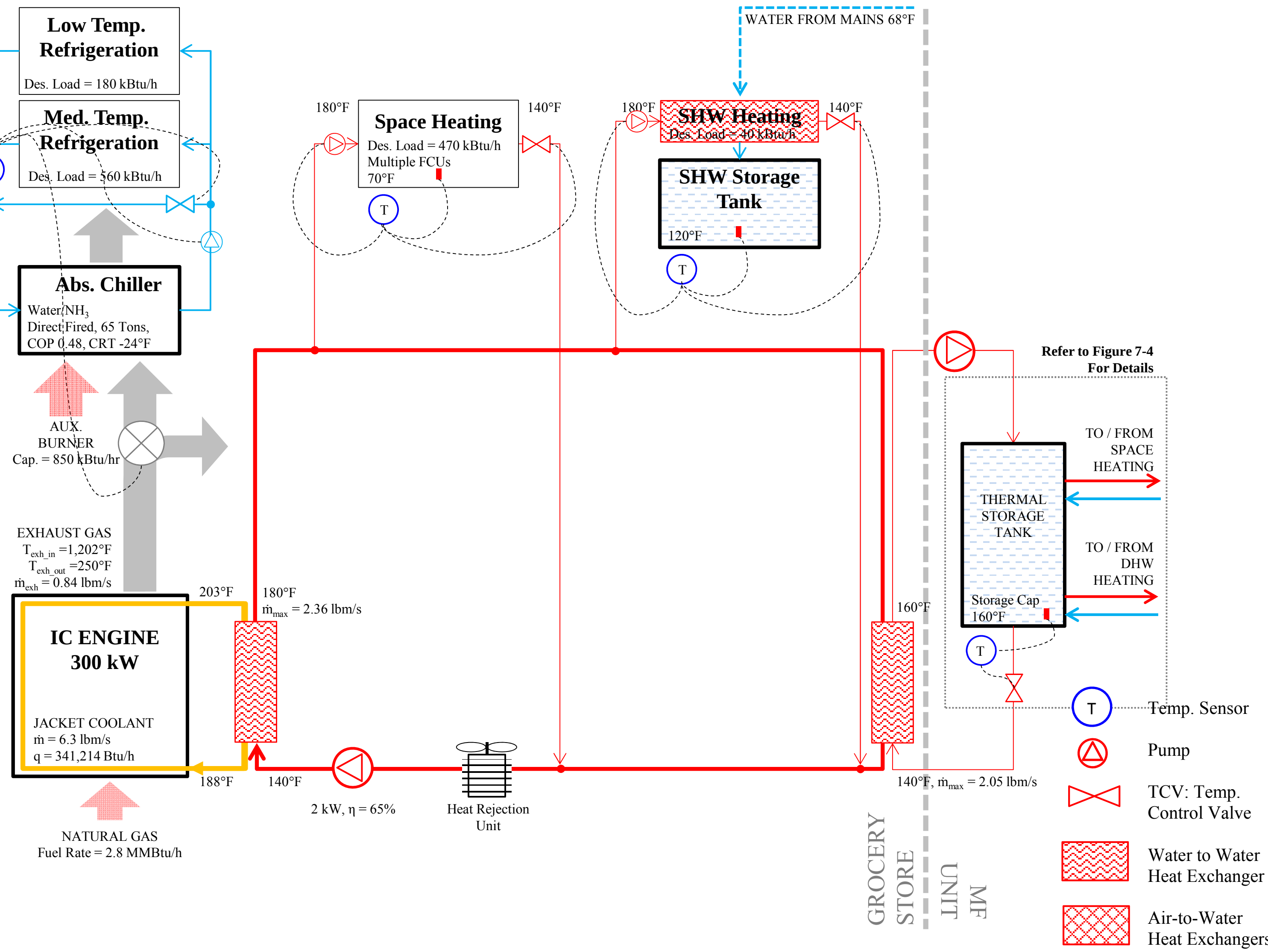

Figure 7-10: Thermal Energy Distribution Diagram for Option 4

7.6 Results

The analysis was performed for the four options described above by calculating the annual energy consumption, using a temperature bin analysis and using typical daily profiles of electricity and thermal energy usage in the grocery store. The utilization of surplus energy from the grocery store for residential energy consumption was also assessed.

7.6.1 Annual Energy Consumption

The results showing annual energy consumption for the four options are presented in Figure E-1, Figure E-6, Figure E-11, and Figure E-16 of Appendix E in this report. A summary of the annual energy consumption for the base-case scenarios and the CHP scenarios for the four options is provided in Table 7-2 and Table 7-3 respectively.

In order to calculate the performance of the four options, when considering the base-case, the annual energy consumption for both the grocery store and the multi-family units had to be considered. The amount of energy considered from the multi-family units was different for each option and was determined from the surplus electricity being generated by the CHP system implemented in the grocery store. For each base-case scenario in Table 7-2, energy consumption of the base-case grocery store and the multi-family buildings, which potentially will be served by the surplus energy generated by the CHP scenario, is presented in terms of electricity and natural gas[51] consumption. Then the electricity and natural gas consumption is summed up and presented as total site energy consumption[52]. Resultant source energy consumption is calculated by using appropriate conversion factors[53]. Finally, the energy wasted in generation and transmission is presented in the last column of this table. Table 7-3 provides the site and source energy consumption for the CHP scenario of the four options considered for this analysis. As mentioned in the earlier section, these options implement different configurations of CHP in the grocery store. Similar to Table 7-2, information is presented in terms of site energy consumption, source energy consumption as well as the energy wasted in generation and transmission. Finally, the last column in this table presents the percent difference in energy consumption with the corresponding base-case energy consumption when accounted for at the source.

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[51] It is assumed that 75% of the energy content in natural gas gets converted to useful thermal energy due to equipment inefficiencies.
[52] Energy Wasted = Source Energy – Site Energy
[53] A factor of 3.15 was used to convert site electricity consumption to source electricity generation. In addition, electricity transmission losses were assumed to 7%. A factor of 1.1 was used to convert site natural gas consumption to source natural gas production (EIA, 2010).

Table 7-2: Base-Case Site Energy Consumption

	Case	FOR GROCERY STORE		FOR 8-UNIT M.F. BUILDING		Total Site Energy Consumption (MMBtu/yr)	Total Source Energy Consumption (MMBtu/yr)	Energy Wasted in Generation and Transmission (MMBtu/yr)
		Electricity (MMBtu/yr)	Natural Gas (MMBtu/yr)	Electricity (MMBtu/yr)	Nat. Gas[e,f] (MMBtu/yr)			
BASE-CASE	Option 1[a]	6,950	1,810	1,800	5,469	16,028	37,498	21,469
BASE-CASE	Option 2[b]	7,535	1,761	2,680	4,824	16,800	41,296	24,496
BASE-CASE	Option 3[c]	7,307	1,819	2,411	1,828	13,364	36,425	23,061
BASE-CASE	Option 4[d]	7,487	1,820	3,006	1,954	14,268	39,098	24,830

Table 7-3: Site and Source Energy Consumption for CHP Test Cases

	Case	SITE ENERGY			SOURCE ENERGY		Percentage Difference at Source w/ Base-Case [g] %
		Electricity (MMBtu/yr)	Natural Gas (MMBtu/yr)	Total Energy Consumption Site (MMBtu/yr)	Total Energy Consumption Source (MMBtu/yr)	Energy Wasted in Generation and Transmission (MMBtu/yr)	
CHP-CASE	Option 1	68	24,959	25,026	27,683	2,657	26%
CHP-CASE	Option 2	1	25,123	25,124	27,639	2,515	33%
CHP-CASE	Option 3	12	25,451	25,462	28,035	2,573	23%
CHP-CASE	Option 4	0	29,098	29,098	32,008	2,910	18%

Notes;
a. Option 1 utilizes thermal energy to operate absorption chillers that are used for mech. sub-cooling, space heating and service hot water heating. H/P = 0.25
b. Option 2 utilizes thermal energy to operate absorption chillers that are used for mech. sub-cooling and space cooling, space heating and SHW heating. H/P = 1.03
c. Option 3 utilizes thermal energy to operate absorption chillers that are used for med. temperature refrigeration, space heating and SHW heating. H/P = 0.76
d. Option 4 utilizes thermal energy to operate absorption chillers that are used for med. and low temperature refrigeration, space heating and SHW heating. H/P = 1.09
e. Assuming 75% of energy content in natural gas gets converted to useful thermal energy due to equipment efficiencies.
f. Piping heat losses of 10% were assumed when accounting for delivery of surplus thermal energy generated at the grocery store to the surrounding multi-family buildings.
g. Percentage Difference at Source w/ Base-Case = (Base-Case – CHP-Case)/Base-Case %.

As seen in Table 7-2, different heat to power ratios for the four options assessed provide different amounts of surplus electricity and thermal energy to be utilized by residential units. In Option 1, for the base-case scenario:

- Electricity consumption for the base-case grocery store was 6,950 MMBtu/yr,
- Natural gas consumption for the base-case grocery store was 1,810 MMBtu/yr,
- Electricity consumption for the multi-family units was 1,800 MMBtu/yr,
- Natural gas consumption for the multi-family units was 5,469 MMBtu/yr,
- The total energy consumption when considered at site was calculated to be 16,028 MMBtu/yr,
- The total energy consumption when considered at source was calculated to be 37,498 MMBtu/yr, and
- Energy wasted in generation and transmission was calculated to be 21,469 MMBtu/yr.

In Option 2, for the base-case scenario:

- Electricity consumption for the base-case grocery store was 7,535 MMBtu/yr,
- Natural gas consumption for the base-case grocery store was 1,761 MMBtu/yr,
- Electricity consumption for the multi-family units was 2,680 MMBtu/yr,
- Natural gas consumption for the multi-family units was 4,824 MMBtu/yr,
- The total energy consumption when considered at site is calculated to be 16,800 MMBtu/yr,
- The total energy consumption when considered at source was 41,296 MMBtu/yr, and
- Energy wasted in generation and transmission was calculated to be 24,462 MMBtu/yr.

In Option 3, for the base-case scenario:

- Electricity consumption for the base-case grocery store was 7,307 MMBtu/yr,
- Natural gas consumption for the base-case grocery store was was 1,819 MMBtu/yr,
- Electricity consumption for the multi-family units was 2,411 MMBtu/yr,
- Natural gas consumption for the multi-family units was 1,828 MMBtu/yr,
- The total energy consumption when considered at site is calculated to be 13,364 MMBtu/yr,
- The total energy consumption when considered at source was 39,425 MMBtu/yr, and
- Energy wasted in generation and transmission was calculated to be 23,061 MMBtu/yr.

In Option 4, for the base-case scenario:

- Electricity consumption for the base-case grocery store was 7,487 MMBtu/yr,

- Natural gas consumption for the base-case grocery store was 1,820 MMBtu/yr,
- Electricity consumption for the multi-family units was 3,006 MMBtu/yr,
- Natural gas consumption for the multi-family units was 1,954 MMBtu/yr,
- The total energy consumption when considered at site was 14,268 MMBtu/yr,
- The total energy consumption when considered at source was 39,098 MMBtu/yr, and
- Energy wasted in generation and transmission was calculated to be 24,830 MMBtu/yr.

As seen in Table 7-3, when considering the energy consumption of the CHP scenario, a major portion of the energy consumption of the store was dependent on the supply of natural gas. Electricity from the utilities was required only when the electricity generated by the IC engine did not meet the requirements of the grocery store. In Option 1, for the CHP scenario:

- Electricity consumption when considered at site was 68 MMBtu/yr,
- Natural gas consumption when considered at site was 24,959 MMBtu/yr,
- Total energy consumption when considered at site was 25,026 MMBtu/yr,
- Total energy consumption when considered at source was 27,683 MMBtu/yr,
- Energy wasted in generation and transmission was calculated to be 2,657 MMBtu/yr, and
- A 26% reduction over the base-case scenario when considering source energy consumption.

In Option 2, for the CHP scenario:

- Electricity consumption when considered at site was negligible,
- Natural gas consumption when considered at site was 25,123 MMBtu/yr,
- Total energy consumption when considered at site was 25,124 MMBtu/yr,
- Total energy consumption when considered at source was 27,639 MMBtu/yr,
- Energy wasted in generation and transmission was calculated to be 2,515 MMBtu/yr, and
- A 33% reduction over the base-case scenario when considering source energy consumption.

In Option 3, for the CHP scenario:

- Electricity consumption when considered at site was 12 MMBtu/yr,
- Natural gas consumption when considered at site was 25,451 MMBtu/yr,
- Total energy consumption when considered at site was 25,462 MMBtu/yr,
- Total energy consumption when considered at source was 28,035 MMBtu/yr,

- Energy wasted in generation and transmission was calculated to be 2,573 MMBtu/yr, and
- A 23% reduction over the base-case scenario when considering source energy consumption.

In Option 4, for the CHP scenario:

- Electricity consumption when considered at site was 0 MMBtu/yr,
- Natural gas consumption when considered at site was 29,098 MMBtu/yr,
- Total energy consumption when considered at site was 29,098 MMBtu/yr,
- Total energy consumption when considered at source was 32,008 MMBtu/yr,
- Surplus energy available to residential buildings was 4,472 MMBtu/yr,
- Energy wasted in generation and transmission was calculated to be 2,910 MMBtu/yr, and
- An 18% reduction over the base-case scenario when considering source energy consumption.

As seen from the percentage difference above base-case for source energy consumption, even though Option 2 and Option 4 had similar heat-to-power ratios, Option 2 provided maximum savings and Option 4 provided minimum savings above the corresponding base-case. This is because of the difference in efficiency of the equipment being used in the two options to capture the waste heat generated by the CHP facility. As seen in Option 3 and Option 4, where direct fired Water/NH_3 absorption chillers were used to meet medium and low temperature refrigeration requirements of the grocery store, higher temperatures of thermal energy provided by the IC engine were required to operate the low-temperature absorption chillers. It was also noted that the performance of absorption chillers depended on the temperature required by the cooling load. For example, when considering lower cooling temperatures as in Option 3 and Option 4 more thermal energy was required by the absorption chillers to provide a unit of cooling than what was required in Option 1 and Option 2, resulting in a lower COP.

7.6.2 Hourly Energy Consumption of the Grocery Store

The hourly energy consumption of the grocery store was assessed using a temperature bin distribution and an analysis of typical daily profiles. The assessment is presented in the sub-sections below. Graphs for the hourly energy consumption of the four options are provided in Appendix E of this study.

7.6.2.1 *Temperature Bin Distribution Analysis*

As seen in Figure E-2a, Figure E-7a, Figure E-12a and Figure E-17a (Electricity Requirements), when considering the temperature bin distribution of the hourly electricity requirements of the grocery store, the electricity consumption requirements for all the options gradually increased till the 76-80°F temperature bin after which a step increase in electricity consumption was observed. This trend is typical for internal load dominated building such as the grocery store, which has large non-weather dependent electricity loads such as loads from lighting and refrigeration compressors.

When considering the electricity generated by the 300 kW IC engine operating at full power mode, for Option 1, most of the electricity requirements of the grocery store were met by the CHP facility. However, all the electricity requirements could not be met by the CHP facility in temperature bins of 96-100°F and above. Electricity loads unmet by the CHP facility were supplemented by electricity purchased from the utilities. For all other options, the generated electricity met all the requirements of the facility for all the temperature bins except in a few instances where maximum electricity requirements had to be supplemented by electricity from the utilities. Since, electricity consumption at higher temperature bins is associated with peak demand, a drastic reduction in peak demand was observed on the implementation of the CHP system.

As seen in Figure E-2b, Figure E-7b, Figure E-12b and Figure E-17b (Abs. Chiller Loads), when considering loads for absorption chillers, for Option 1, which addressed the energy requirements of the mechanical sub-cooler, an almost flat profile of loads was observed from the lowest temperature bin to the 76-80°F temperature bin after which the load profile became gradually steeper. The slope for the loads was due to the fact that when the air is warmer, refrigeration compressors tend to run for a longer period of time over a greater temperature difference, to achieve the desired cooling effect. This resulted in greater loads for the mechanical sub-cooler in Option 1. On the other hand, for Option 2, when considering the loads from the mechanical sub-cooler and space cooling, cooling loads were almost constant at lower temperature bins till the 51-55°F after which the loads increase when the space cooling loads started to gain precedence over the mechanical sub-cooling loads. For Option 3 and Option 4, which addressed the cooling loads from the mechanical sub-cooler and medium temperature refrigeration units, and loads from medium temperature and low temperature refrigeration units respectively, the loads gradually increased from the lowest temperature bin all the way up to the highest temperature bin.

As seen in Figure E-2b, for Option 1, the captured heat from exhaust gas generated from the operation of the IC engine was in form of hot water at 180°F and was used to meet the requirements of the absorption chiller for all the temperature bins. All cooling requirements were met by the captured waste heat. As seen in Figure E-7b, Figure E-12b and Figure E-17b for Option 2, Option 3 and Option 4 respectively, the requirements for the mechanical sub-cooler and space cooling (as in Option 2); mechanical sub-cooler and medium temperature refrigeration loads (as in Option 3); and medium and low temperature refrigeration (as in Option 4) were met by a direct-fired absorption chillers. As seen in Figure E-2b and Figure E-12b, for Option 2 and Option 3, the captured waste heat met most of the requirements for only the lower temperature bin. An auxiliary burner[54] operating on natural gas provided supplementary thermal energy to meet the remaining cooling loads. As seen in Figure E-17b for Option 4, because of the much lower temperature of -36°F was imposed by this configuration of refrigeration loads on the absorption chillers, the captured waste heat could meet only a portion of cooling loads. An auxiliary burner operating on natural gas was installed to provide supplementary thermal energy to meet the remaining cooling loads.

As seen in Figure E-2c, Figure E-7c, Figure E-12c and Figure E-17c (Space Heating Loads), an inverse trend was observed for all options when considering the hourly space heating requirements of the grocery store. In all the options, the space heating loads were the highest in the 26-30°F temperature bin and decreased until the 91-95°F temperature bin. No space heating was required in temperature bins that were above 95°F[55]. These complementary trends proved to be beneficial for the waste thermal energy generated from the IC engine that was utilized to provide both heating and cooling for the grocery store.

Space heating for all the four options was primarily provided by thermal energy recovered from the jacket coolant loop of the IC engine. In addition, surplus energy from the exhaust gases not utilized by the absorption chillers was used to supplement the heat extracted from the jacket coolant to meet the space heating requirements. As seen in Figure E-2c, Figure E-7c, Figure E-12c and Figure E-17c, for all the four options, this arrangement met all the loads

[54] An efficiency of 100% was assumed for the auxiliary burner.

[55] In grocery stores, space heating is occasionally required at high ambient temperatures to offset the cooling effect created by the spilling over of cold air from open refrigerated display cases. This can be reduced by the use of glass doors on all refrigerated cases.

for higher temperature bins. Occasional heating requirements of some of the lower temperature bins as seem in Option 4, had to be met by an auxiliary hot-water boiler operated by natural-gas[56]. After meeting the space heating requirements, hot water at 180°F that was heated by both the air to water heat exchanger coupled with the exhaust gases and the jacket coolant was diverted to meet the SHW heating requirements.

As seen in E-2d, Figure E-7d, Figure E-12d and Figure E-17d (Service Hot Water Heating Loads), the energy consumption of SHW usage was based on a usage schedule, hence no apparent pattern was observed across the temperature bins. The provision of hot water to the grocery store was assumed to be at a temperature of 120°F. The thermal energy required to heat water to this temperature was provided by energy available from the jacket coolant loop of the IC engine. After meeting the space heating requirements, hot water at 180°F, which is heated by the jacket coolant loop, is diverted to meet the SHW heating requirements. Most of SHW requirements are met by this arrangement. However, supplemental energy from an auxiliary hot water boiler is occasionally required especially for last two temperature bins.

As seen in Figure D-3a, Figure D-5a, Figure D-7a and Figure D-9a (Surplus Electricity from CHP), the availability of surplus electricity gradually increased for temperature bins below the 76-80°F temperature bin. For temperature bins above the 76-80°F temperature bin, the availability of surplus electricity tapered off as the ambient temperatures increase with less surplus electricity made available to be wheeled across the boundary of the grocery store to the surrounding residential units.

As seen in Figure D-3b, Figure D-5b, Figure D-7b and Figure D-9b (Surplus Thermal Energy from CHP), when considering the availability of thermal energy, an inverse trend was observed when considering the distribution of the surplus thermal energy available at 180°F for different temperature bins for Option 1, Option 3 and Option 4. As the ambient temperature increased, more surplus thermal energy was made available to be wheeled across the boundary of the grocery store to the surrounding residential units. The availability of surplus thermal energy remained more or less constant for temperature bins above 86-90°F. On the other hand for Option 2, the availability of surplus thermal energy started at a low point with the lowest temperature bin of 26-30°F. The availability curved up to reach a point at the 66-70°F temperature bin after which it curved down again and assumes an almost flat profile at higher

[56] A condensing boiler with an efficiency of 95% was assumed for the analysis.

temperature bins above 91-95°F. The lowest possible available thermal energy remained at a constant level above the 91-95°F temperature bin.

7.6.2.2 Typical Daily Profile Analysis

Typical hourly profiles for the four options are obtained for the month of January, March, July and October. Hourly trends projecting the maximum, 75th percentile, 50th percentile, 25th percentile and the minimum loads for electrical and thermal loads are provided for four sample months.

As seen in Figure E-3a, Figure E-8a, Figure E-13a and Figure E-18a (Electricity Requirements) when considering electricity consumption, it is observed that for all the months the electricity consumption was relatively flat during the unoccupied hours between 12:00 AM and 6:00 AM. The slight increase in electric load profiles between 12:00 AM and 2:00 AM was due to the 25% of the lights in the main areas of the grocery store being left on for maintenance and stocking of products. There was a sharp increase in electricity consumption between 6:00 AM and 7:00 AM. This coincided with the opening hours of the store. This increase was also due to the store lights being switched on in the morning, there being not enough daylight to activate the daylight sensors. The electricity consumption then increased during the afternoon hours. This coincides with the increase in ambient temperature during the day[57] as well as increase in the number of occupants in the grocery store. Another sharp increase was seen during the late afternoon and evening hours depending on the month selected. This was due to the lighting being switched on in the main areas due to the unavailability of daylight to illuminate the main spaces in the grocery store. Peak daily electricity consumption now shifted from late afternoon hours, which is typical of commercial buildings, to between 7:00 PM and 8:00 PM depending on the month. Electricity generated by the CHP system installed in the grocery store almost met all the electricity requirements for the four options.

Depending on the option being assessed, the cooling load requirements of the mechanical sub-cooler, space cooling, low and medium temperature refrigeration units were met by absorption chillers. As seen in Figure E-3b, Figure E-8b, Figure E-13b and Figure E-18b (Absorption Chiller Requirements for Option 1, Option3 and Option 4 respectively) when considering the daily load patterns for the absorption chillers, cooling loads were almost constant for all months with a slight increase in loads during afternoon and evening hours. On the other

[57] See Appendix A for daily ambient temperature profiles for College Station, TX.

hand as seen in Figure E-8b (Absorption Chiller Requirements for Option 2), a sharp increase in cooling loads occurred during late afternoon and early evening hours. This is because Option 2 incorporated space cooling loads, which is weather dependent, to be met by absorption chillers. For all options although the daily profile of loads for absorption chillers were similar for all months the magnitude of the loads increased during summer months. This is because more energy was required during summer months to maintain the specified temperatures of the cooling loads being met. Occasional requirement of supplementary thermal energy from a natural gas burner was required for Option 2 and Option 3 to meet the cooling loads. On the other hand as seen in Figure E-18b, Option 4 required substantial amount of thermal energy from the natural gas burner to meet the requirements of the medium and low temperature cooling loads.

As seen in Figure E-4a, Figure E-9a, Figure E-14a and Figure E-19a (Space Heating for Option 1, Option 2, Option 3 and Option 4 respectively), when considering space heating loads in the grocery store, some space heating was required during the summer months especially during the hours when the store was closed and morning hours. As expected, during winter months, the magnitude of loads was higher. Also, a greater variation in the range of hourly consumption was observed during winter months implying a strong dependence on ambient temperatures. As seen in Figure E-4a, Figure E-9a, for Option 1 and Option 2 all space heating loads are met. However, as seen in Figure E-14a, Figure E-19a, for Option 3 and Option 4, during the winter seasons, supplemental heating is required from an auxiliary hot water boiler.

Finally, as seen in Figure E-4b, Figure E-9b, Figure E-14b and Figure E-19b (SHW Heating for Option 1, Option 2, Option 3 and Option 4 respectively),when considering the SHW heating loads (hot water at 120°F) the consumption profiles were the same for all months[58]. However, the magnitude of consumption changes with larger consumption was observed in the winter and spring months. This is due to the variation in the temperature of water at the mains[59].

As seen in Figure E-5a, Figure E-10a, Figure E-15a and Figure E-20a (Surplus Electricity for Option 1, Option 2, Option 3 and Option 4), maximum surplus electricity was available for all months during night time when the store is closed, tapering off when moving towards afternoon and evening hours. In addition, as seen in Figure E-5a and Figure E-15a for Option 1 and Option 3 respectively, no surplus electricity was available during afternoon hours

[58] Refer to Figure 7-3 for the profile of SHW usage.
[59] Monthly record water mains temperatures used in this analysis is provided in Appendix A of this study.

during the summer season. The generation of surplus electricity during night time when the store is closed indicates the potential of using a storage arrangement for electricity[60]. Potential methods however, are not explored in this study.

Finally, as seen in Figure E-5b, Figure E-10b, Figure E-15b and Figure E-20b (Surplus Thermal Energy at 180°F for Option 1, Option 2, Option 3 and Option 4), when considering surplus thermal energy consumption the provision was almost constant for summer months. On the other hand, the availability of surplus thermal energy exhibited a wide variation during winter months. In addition, as seen in Figure E-5b (Surplus Thermal Energy at 180°F for Option 1) the provision of surplus thermal energy was almost constant for all the months. As seen in Figure E-10b (Surplus Thermal Energy at 180°F for Option 2), the provision of surplus thermal energy was almost constant for winter months. However, the availability of surplus thermal energy dipped during the afternoon hours of summer months. Finally, as seen in Figure E-20b (Surplus Thermal Energy at 180°F for Option 4), no surplus energy was available during the morning hours of the winter months. Surplus thermal energy is a good candidate for thermal storage, in which hot water can be stored in insulated tanks whenever available and used later whenever required. The viability of this possibility is explored in a subsection presented below.

7.6.3 Comparison with Residential Energy Consumption

Surplus electricity and thermal energy (50th percentile) from the grocery store were superimposed on typical hourly electricity and thermal energy consumption of several 8-unit multifamily buildings[61]. As in the hourly assessment of the energy consumption in the grocery store, hourly energy consumption of the multi-family residential buildings was analyzed in terms of a temperature bin distribution and an analysis of typical daily profiles. The assessment is described in the sub-sections below.

7.6.3.1 Temperature Bin Distribution Analysis

When considering the temperature bin distribution of the hourly electricity requirements of the residential buildings, as seen in Figure D-3a, Figure D-5a, Figure D-7a and Figure D-9a

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[60] Potential arrangements of using surplus electricity include operating vapor compression chillers during off-peak hours and storing chilled water for use during hours when peak demand occurs.

[61] The number of buildings was obtained by dividing the annual surplus electricity and thermal energy available from the grocery store by the annual electricity consumed by one 8-unit multi-family building. The number of multi-family buildings was calculated on the basis of surplus annual electricity and thermal energy available from the grocery.

(Electricity Requirements[62]) it was observed that the electricity loads were constant till the 66-70°F temperature bin after which an increase in electricity consumption was seen in the subsequent temperature bins. The surplus electricity available from the grocery store meets all the requirements till the 76-80°F temperature bin for Option 1(Figure D-3a), 86-90°F temperature bin for Option-2, Option 3 and Option 4 (Figure D-5a, Figure D-7a and Figure D-9a), after which the availability of surplus electricity tapers off.

As seen in Figure D-3b, Figure D-5b, Figure D-7b and Figure D-9b (Thermal Energy Requirements[63]) the thermal energy requirements had an inverse trend with maximum requirements occurring at the lowest temperature bins and reaching a minimum at the 71-75°F temperature bin after which the trend remains constant. For Option 1 (Figure D-3b), the surplus thermal energy available from the grocery store meets most of the thermal energy requirements till the 51-55°F. For Option 2 (Figure D-5b), the surplus thermal energy available from the grocery store meets most of the thermal energy requirements till the 36-40°F. For Option 3 and Option 4 (Figure D-7b and Figure D-9b), the surplus thermal energy available from the grocery store met most of the thermal energy requirements till the 46-50°F. In general, the surplus thermal energy obtained from the store did not meet all the requirements of the lower temperature bins. On the other hand surplus thermal energy was not being utilized in the higher temperature bins.

7.6.3.2 Typical Daily Profile Analysis

Similar to the analysis conducted for the grocery store, sample months of January, March, July and October were considered. For the four options, a general trend in the availability of surplus electricity from the grocery store was observed for all seasons. As seen in Figure D-4a, Figure D-6a, Figure D-8a and Figure D-10a (Electricity Requirements), the availability peaked during early morning and late evening hours. This profile however, did not coincide with the electricity consumption profile of the multifamily units in which peak consumption occurred during late afternoon and early evening hours. For all options considered for January, all the electricity loads were met except for a few hours during the afternoon. Similar trends were observed in the swing season months of March and October. However, in this case the electricity

[62] The number of houses assumed to be served by surplus electricity from the store was calculated by dividing the total surplus electricity available from the store by the annual electricity consumption of a single multi-family building.
[63] The number of houses assumed to be served by surplus thermal energy from the store was calculated by dividing the total surplus thermal energy available from the store by the annual thermal energy consumption of a single multi-family building.

consumption for residential units was higher. For the month of July, the electricity consumption of multifamily units increased even further. On the other hand, during this month, no surplus electricity was available from the grocery store especially during afternoon and early evening hours for Option 1 (Figure D-4a). Small amount of surplus electricity was available during some periods in the afternoon for Option 2 Option 3 and Option 4 (Figure D-6a, Figure D-8a and Figure-D-10a). It should be noted that these hours tend to coincide with the peak demand for electricity. During these hours electricity loads had to be met with supplementary electricity purchased from the utilities. In addition, the availability of surplus electricity was restricted to early morning hours which coincide with the lowest residential electricity requirements. Hence, most of the residential electricity requirements for the month of July for all options are provided from the utilities and a large quantity of surplus electricity during morning and evening hours was left unutilized.

As seen in Figure D-4b, Figure D-6b, Figure D-8b and Figure D-10b (Thermal Energy Requirements), when considering thermal energy consumption without the implementation of thermal storage, an almost constant pattern of provision of surplus energy is observed on a daily basis in all the sample months considered for this analysis. However, the availability of thermal energy tends to vary from month to month with more thermal energy available in summer months than during winter months. During the summer months, almost all the thermal loads for selected multi-family buildings are met with the surplus thermal energy from the store. However, when looking at the winter and spring month's non-coincidence of loads and the availability of surplus energy was observed. The non-coincidence prompted large amount of surplus thermal energy to be wasted when not in use in certain instances and in other instances prompting the use of hot water boilers to provide supplemental energy. This non-coincidence of loads was removed to an extent by the introduction of thermal storage.

7.7 Summary and Conclusions

In this chapter, the impact of using a CHP facility in a grocery store was assessed. The CHP model that was developed and used to accomplish this analysis was discussed. Four options for CHP operation in the grocery store were considered for the analysis. The impact of implementing these options on the annual and hourly energy consumption of the grocery store was presented and discussed in this chapter.

When assessing the annual energy consumption of the four options, it was observed that the total energy consumed by the implementation of these options is greater than the total annual energy consumed by the corresponding base-case buildings when accounted for at the site. However, the electricity consumption from the utilities was drastically decreased. On the other hand, when assessing the annual energy consumption of the four options at the source, the percentage savings from implementing the CHP scenarios above the corresponding base-case scenarios was within the range of 18% and 33%. In addition, it should also be noted that on the implementation of the CHP facility energy spent in generation and transmission was reduced by approximately 87 - 89%.

It was noted that in all the four options, the IC engine was operated at full power mode and was sized to meet more than 95% of the electricity loads, thus drastically reducing levels of electricity consumption. Sizing the prime-mover to match 95% of electricity requirements led to the generation of a large quantity of surplus electricity and thermal energy during off-peak hours that could not be absorbed by the grocery store. The generation of surplus electricity and thermal energy served a greater number of surrounding residential buildings than what would have been the case with a smaller sized IC engine. This resulted in an increase in the percentage difference with the corresponding base-case scenario leading to a conclusion that oversized systems make better options. However, at this point in the analysis there was no check on the impact of system sizing. Hence, an economic assessment of the four options was required for a conclusion regarding an optimal system size for the facility.

The analysis also examined hourly energy consumption profiles for the four options and compared these profiles to the electricity and thermal energy available from the IC engine to meet these requirements.

When considering the distribution of hourly electricity requirements into temperature bins, it was also observed that peak electricity requirements associated with higher temperature bins were rarely met by the four options implemented. However, the overall electricity requirements from utilities were greatly reduced due to the on-site generation of electricity. As noted earlier in this discussion, the same thing cannot be said about natural gas consumption because the overall consumption of natural gas was greatly increased due to the installation of the IC engine for power generation. When considering the distribution of hourly thermal energy requirements into temperature bins, the trend in the thermal energy requirements for space heating and SHW heating were complemented by the trend in the thermal energy requirements for absorption chillers. This complimentary juxtaposition of cooling and heating loads provided optimum conditions for the absorption of waste thermal energy generated as a byproduct of electricity generation.

Despite the creation of optimum conditions for the absorption of thermal energy in the grocery store, a large amount of surplus thermal energy was available to be absorbed by the surrounding residential buildings. The amount of surplus energy available depended on the option selected. The least amount of surplus thermal energy was available on implementing Option 3 and Option 4. On the other hand, for Option 1 and Option 2, the number of multifamily units served by the surplus thermal energy from the store far exceeded the number of multifamily units served by the surplus electricity from the store. When the assessment was carried out in terms of energy consumption only, the generation of a greater quantity of surplus thermal energy improved the percentage of energy savings above the corresponding base-case.

Finally, when comparing the profiles of surplus electricity and thermal energy from the grocery store to the typical profiles of residential energy consumption, a non-coincidence of surplus electricity and thermal energy available to residential units and the electricity and thermal energy consumption profiles of these units for all the options was observed. Providing storage for surplus electricity, when not in use, although feasible, was not been assessed by this study. However, for thermal energy, an appropriate thermal storage capability was designed to relieve the frequent non-coincidence of loads and surplus thermal energy available from the store.

CHAPTER VIII

ECONOMIC EVALUATION

8.1 Overview

This chapter presents an economic evaluation of the four options discussed in Chapter 7 of this study. As seen from the conclusions of Chapter 7, an economic evaluation could be a deciding factor in the selection of an optimum CHP system for the grocery store. When performing the economic evaluation it should be noted that in order for the project to be feasible, monetary savings from installing the CHP system must be sufficient to justify the capital investment (Caton 2010). In addition, the project must also meet specific values set by the client and reflect the quality of investment (Baxter 1997).

This study provides an assessment of the payback period and lifecycle cost analysis associated with the installation and operation of each of the four CHP options described in Chapter 7. Figure 8-1 below provides a diagram for determining the payback periods and the lifecycle cost associated with each option. The diagram provides a graphical overview of the relationship between the installed cost and operating costs for calculating the lifecycle cost and payback period for each option.

The second section of this chapter discusses the basic economic concepts used and appropriate inputs that were required to assess payback period and lifecycle cost analysis. This section also includes a discussion on the capital and operating costs incurred with the installation and operation of the CHP system. The third section of this chapter presents an economic assessment of the four options of CHP installation in the grocery store that have been described in Chapter 7 of this study. In the fourth section, a sensitivity analysis of a selected option was conducted to assess the impact of the various parameters involved in the assessment. Finally, in the fifth section, conclusions drawn from the economic analysis of the four options are presented and discussed.

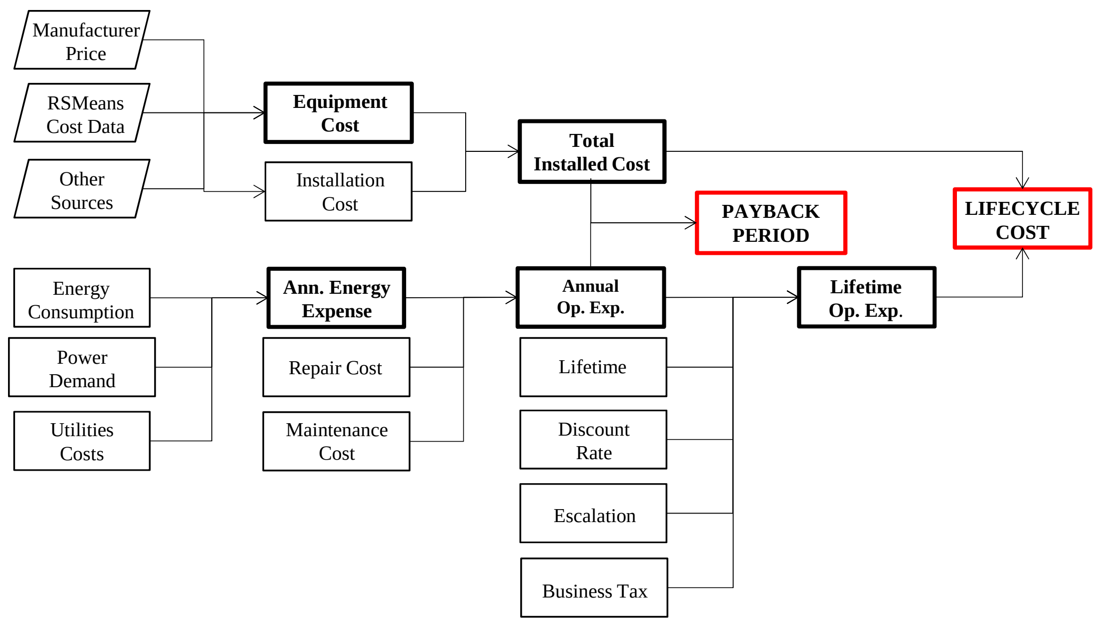

Figure 8-1: Flow Diagram of Inputs for Determining Lifecycle Costs and Payback Period (Source: US DOE 2008b)

8.2 Criteria Implemented for Economic Assessment

The subsections presented below describe the criteria adopted to perform the lifecycle cost and the payback period assessment. These include a description of the assessment measures, economic indices implemented in these assessment measures, capital and installation costs, operation costs, and cost of utilities.

8.2.1 Economic Assessment Measures

Calculations were performed using conventional methods of assessment. These methods have been discussed in detail by Caton (2010), ASHRAE Handbook of HVAC Applications (1999b) and Park (2007). A brief description of the method used for this analysis is presented in the Chapter 2, which presents the literature review performed by this study. The formulae used in the calculation of these measures are presented in Appendix C of this study.

Four measures were selected by this study to conduct the assessment and the selection of an optimum CHP system in the grocery store. The measures are:

- For payback period analysis
    - Simple payback period, and
    - Investors rate of return (IROR),

- For lifecycle cost analysis
    - Net present value (NPV), and
    - Internal rate of return (IRR).

Each of these measures quantifies different aspects of the investment. Hence, these measures were used in combination with one another (Caton 2010) to provide a comprehensive economic assessment of the four options.

8.2.2 Economic Indices

Economic indices such as economic discount rate, income tax rate, lifetime and depreciation are required by the above mentioned economic measures to perform the assessment. Most of the assumptions for the economic indices implemented in this study are adopted from an analysis performed by the U.S. Department of Energy (DOE) for commercial refrigeration equipment (US DOE 2008b). The DOE study evaluated economic impact on individual customers of possible energy efficiency standards developed for commercial refrigeration equipment. Other assumptions made by this study follow standard practice and are cited accordingly below.

The discount rate can be defined as a rate at which future expenditures are discounted to establish their present value (US DOE 2008b). For this analysis the discount rate of 5.9% was assumed from the analysis conducted by the DOE (US DOE 2008b). A tax rate of 29.0% was assumed for the same analysis. The economic lifetime can be defined as the number of years the CHP unit is in operation before it is retired from service (US DOE 2008b). This study based its assumptions on discussions with industry experts (Johnson, personal communication, July 2012), and concluded that a typical CHP system operates for approximately 25 years. In addition to the economic indices described above, a straight line depreciation schedule is used for this analysis. A depreciation value of 10 %is assumed over a period of 10 years. Table 8-1 presents a summary of these indices.

Table 8-1: Economic Indices Implemented in the Analysis

Economic Parameters			
Economic Index	**Rate / Value**	**Unit**	**Ref/Notes**
Discount Rate	5.9	%	US DOE 2008b
Income Tax	29.0	%	US DOE 2008b
Economic Life	25	Years	Johnson 2012, personal communication
Depreciation	10.0	%	Standard Practice Straight line - Over a period of 10 years

8.2.3 Capital Costs

Capital costs used in this analysis included equipment costs well as the installation of the equipment on site. This assessment included costs of purchase and installation of CHP systems implemented in the grocery store. The costs were included either as first costs or incremental costs depending on whether the equipment replaced existing equipment or is installed in addition to the existing equipment.

Information for capital costs related to the installation of a CHP system was adopted from various sources such as relevant manufacturer's literature (Trane 2007), reports and publications (Westphalen et al. 1996, ORNL 2004, MCHPAC 2004, Caton 2010, De Wit 2007), and cost data published by RSMeans (RSMeans 2009, 2012). The capital costs did not include adjustments to equipment performance such as improvements in prime movers, absorption chillers, heat exchangers or cooling towers.

Costs for major equipment packages include the cost of the prime mover, which in this case was the IC engine; absorption chillers and auxiliary boilers and burners (if required). Costs for balance of plant equipment included controls, emergency devices, exhaust systems and stacks, natural gas compressors, any thermal storage equipment, water treatment devices, concrete bases or pads, fuel supply system components, any necessary building modifications, other piping fittings, mechanical system interfaces, condensers, cooling systems, feed-water tanks, deaerators, feed-water pumps, other pumps, flue gas bypass valves, dampers and ducts, and other such equipment. In addition, engineering and construction costs as well as other costs pertaining to permitting contingency and other miscellaneous items were included either as first costs or as incremental costs. A typical division of capital costs in a CHP project are provided in Table 8-2 below. Unit costs for the certain key components of the CHP system are provided in

Table 8-3. Since the installation of a CHP facility is an incremental cost, the analysis modified the information from Table 8-2 to calculate the lump-sum costs for engineering and construction (from 15% to 5%) and ‘other’ entities (from 20% to 10%). The analysis utilized cost specifications from Table 8-3 to calculate the costs of the major equipment packages and balance of plant equipment implemented in the four options. A comprehensive list of the capital costs are provided in Appendix H.

All four options considered for this analysis implement a 300 kW IC engine, which generated electrical power. In order to absorb waste thermal energy generated by the engine, all four options implement a space heating system for the grocery store. The space heating system operates using waste thermal energy obtained from the jacket water coolant of the installed IC engine. An auxiliary boiler had to be installed to meet space heating loads that could not be met by the energy from the IC engine.

An absorption chiller was implemented in each option to meet designated cooling loads. However, the type of chiller, capacity and the operating temperature of the absorption chiller were different for each case, depending on the temperature and type of cooling loads the chiller was designated to serve. As seen in Table 8-3 the cost of the chiller not only depended on the type of chiller and the capacity installed, but also the temperature at which the chiller was operating; Water/NH_3 chillers tend to cost more than the LiBr/Water chillers; Chillers used for low temperature refrigeration applications were more expensive than chillers installed for medium temperature refrigeration systems or for space cooling purposes. In addition, direct fired chillers were found to be more expensive than hot water driven chillers. In some instances, an auxiliary burner was installed along with the direct fired absorption chiller to meet cooling and refrigeration loads that could not be met by the CHP system. The auxiliary burner was sized to meet all the cooling load requirements designated for the absorption chiller to meet all the loads that could not be met by the CHP system.

Table 8-2: A Typical Division of Capital Costs in a CHP Project (Source: Caton 2010)

Capital Costs	Percentage Breakdown
Major Equipment Packages	40%
Balance of Plant Equipment	25%
Engineering and Construction	15%
Other (Interest during construction, permitting, contingency, etc.)	20%

Table 8-3: Unit Capital Costs of Components of the CHP Scenario[1]

Item	Unit Costs		Reference
	Unit	Cost ($)	
Reciprocating Engine [a] (100 – 500 kW)	$/kW	$1,400 – $1,800	MCHPAC 2004
Absorption Chiller[b] (100 tons)			
LiBr/Water, single-effect, hot water driven	$/ton	$1,000	MCHPAC 2004
LiBr/Water, double-effect, direct-fired	$/ton	$1,200	MCHPAC 2004
Water/NH3, direct-fired (med. temp.)	$/ton	$1,600	MCHPAC 2004
Water/NH3, direct-fired (low temp.)	$/ton	$2,000	MCHPAC 2004
Heat Rejection Unit			
Cooling Tower	$/ton	$203	RSMeans 2012
Pumping	$/ton	$100	Trane 2007
Water –to– water Heat Exchangers [c]	$/GPM	$85 – $86	RSMeans 2012
Aux. Boiler [d] (350 – 500 MBTUH)	$/MBTUH	$28 – $53	RSMeans 2012
Thermal Storage Tank[e]	$/gallon	$0.47 – $4.5	De Witt 2007
Residential Hot Water Loop Piping & Pumping Costs[f]	$/dwelling unit	$900	Boulter 2012

Notes:

a. Smaller IC Engines are more expensive to manufacture. The costs are inclusive of a medium sized heat exchanger for transfer of heat from exhaust gases to another medium.
b. The $/ton costs for the absorption chillers have been derived from the specifications of a 100 ton absorption chiller.
c. The $/GPM costs for heat exchangers are based on the costs of 1200 GPM and 1800 GPM plate heat exchangers.
d. The $/MBTUH costs for boilers are derived from the costs of 350 – 500 MBTUH condensing boilers.
e. The range of costs corresponds to the size of thermal storage tank with smaller tanks costing more to build.
f. Residential hot water loop piping and pumping costs have been derived from the equation described in Section 8.3.2 of this chapter and certain assumptions regarding the layout of the residential units. The costs include the costs for pumps.

[1] Costs include installation costs.

Surplus electricity and thermal energy generated in the store was exported to the surrounding multifamily residential buildings via means of a hot-water loop system. A thermal storage tank was also installed to reduce the asynchronous nature of the availability and demand of thermal energy for residential use. For the residential buildings the analysis included the cost of installing a thermal storage tank and the installation of hot water pipes and pumps which distribute surplus thermal energy from the store to surrounding residential units. The costs for insulated free standing steel thermal storage were obtained from literature on heat storage for CHP plants (De Wit 2007). The costs for piping and pumping thermal energy in the residential community varied greatly depending on the specific applications (e.g. greenfield vs. existing downtown scenario) as well as the system operating parameters (i.e. supply temperatures, temperature drops and pressures). The assumption used in this analysis was for shallow buried, pre-insulated bonded thin-walled steel hot water piping. The cost for piping can be calculated by the formula (Boulter, personal communication, September 2012):

$$Cost[\$(CAD) per\ trench\ meter] = 4.76 \times DN\ + 371$$

Where $(CAD) is cost in Canadian dollars[2] per meter of trench, and DN is the internal pipe diameter in mm. It is noted that this is the total trench cost – supply and return pipe, installation, excavation & backfilling, etc. Costs of pumps were included in the overall costs for piping and pumping.

8.2.3.1 Relative Capital Costs for the Base-case Scenario of the Four Options

For the four options a corresponding set of base-case capital costs of replaced and modified equipment were created. These costs were then deducted from the equipment costs for the respective option. The equipment of the corresponding base-case scenarios and the CHP scenarios for the four options are tabulated in Table 8-4. Detailed breakdown of these estimates are provided in Appendix H of this study. The following points present an overview of the relative base-case costs used for the analysis:

- The relative base-case capital cost for Option 1 included the cost of a sub-cooler unit run as part of the vapor compression refrigeration system. The sub-cooler was sized at 15 tons. The corresponding air-cooled heat rejection unit was sized at 19 tons[3]. The costs of the base-case equipment were estimated to be $11,500.

[2] Assume $ 1 CAD = $1 US for the purpose of calculating costs for this analysis.
[3] A heat rejection factor of 1.25 was assumed for the air-cooled condenser.

- The relative base-case capital cost for Option 2 included the cost of a sub-cooler unit run as part of the vapor compression refrigeration system and packaged space cooling units with gas furnaces. The sub-cooler was sized at 15 tons. The corresponding air-cooled heat rejection unit was sized at 19 tons. The packaged space cooling unit inclusive of the gas furnace was sized at 129 tons[4]. The total cost of the base-case equipment that was replaced was estimated to be $113,088.
- The relative base-case capital costs for Option 3 included medium temperature vapor-compression refrigeration racks, a sub-cooler unit run as part of the low temperature vapor compression refrigeration system and associated air-cooled heat rejection equipment. The vapor compression unit for the refrigeration system was sized at 45 tons. The sub-cooler was sized at 5 tons. The corresponding air-cooled heat rejection unit was sized at 63 tons. The total cost of the base-case equipment that was modified or replaced by equipment in Option 3 was estimated to be $56,249.
- The relative base-case capital costs for Option 4 included a medium and low temperature vapor-compression refrigeration system and associated air-cooled heat rejection equipment. The vapor compression refrigeration system was sized at 62 tons. The corresponding air-cooled heat rejection unit was sized at 78 tons. The total cost of the base-case equipment that was modified or replaced by the CHP equipment in Option 4 was estimated to be $84,757.

8.2.3.2 Capital Costs for the CHP Scenario of the Four Options

The equipment costs of the CHP scenarios for the four options are tabulated in Table 8-4. Detailed breakdown of these estimates are provided in Appendix H of this study. The following points present an overview of the cost breakdown for the CHP scenario of the four options:

- The costs for Option 1 included those for the 300 kW IC engine, 15 ton absorption chiller with a water cooled heat rejection unit and hot water loop for space heating. A hot water loop network and appropriate heat rejection equipment was installed to deliver surplus thermal energy generated at the grocery store to 41 multi-family buildings (328 multi-family dwelling units) surrounding the grocery store[5]. A thermal storage tank of 8,500 gallon capacity was installed to reduce the asynchronous nature in the availability and demand of

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[4] The cost of the packaged space cooling and heating units was based on the cost of multiple 10 ton units.
[5] The number of multi-family buildings was determined by dividing the annual surplus energy available from the grocery store by the annual energy consumption of each multi-family building.

thermal energy for residential use[6]. The total equipment and installation cost of implementing Option 1 was estimated to be $913,498.

- The costs for Option 2 included those for the 300 kW IC engine, a 142 ton direct-fired, two-stage LiBr/Water absorption chiller with a water cooled heat rejection unit and hot water loop for space heating. The absorption chiller was used for mechanical sub-cooling and for space cooling. A chilled-water loop was installed in addition to the hot-water loop to provide space cooling and heating. A hot-water loop network and appropriate air-cooled heat rejection equipment was installed to deliver surplus thermal energy generated at the grocery store to 36 multi-family buildings (312 multi-family dwelling units) surrounding the grocery store. A thermal storage tank of 33,500 gallon capacity was installed to reduce the asynchronous nature in the availability and demand of thermal energy for residential use. The total equipment and installation cost of implementing Option 2 was estimated to be $1,175,543.
- The costs for Option 3 included those for the 300 kW IC engine, a 50 ton Water / NH_3 absorption chiller with a water cooled heat rejection unit and hot water loop for space heating. The absorption chiller was used to operate the mechanical sub-cooler and medium temperature refrigeration system. A hot-water loop was installed to provide space heating for the grocery store. A hot-water loop network and an appropriate air-cooled heat rejection equipment was installed to deliver surplus thermal energy generated at the grocery store to 14 multi-family buildings (112 multi-family dwelling units) surrounding the grocery store. A thermal storage tank of 3,800 gallon capacity was installed reduce the asynchronous nature in the availability and demand of thermal energy for residential use. The total equipment and installation cost of implementing Option 3 was estimated to be $759,711.
- The costs for Option 4 included those for the 300 kW IC engine, a 62 ton Water / NH_3 absorption chiller with a water cooled heat rejection unit and hot-water loop for space heating. The absorption chiller was used to operate the medium and low temperature refrigeration system. The hot-water loop was installed to provide space heating for the grocery store. A hot-water loop network and appropriate heat rejection equipment was installed to deliver surplus thermal energy generated at the grocery store to 15 multi-family

[6] The thermal storage tank was sized using 25th percentile of the demand for the selected number of multifamily buildings. This is done to avoid oversizing the tank during summer months when thermal energy available from the grocery store is not fully utilized by the residential units.

buildings (112 multi-family dwelling units) surrounding the grocery store. A thermal storage tank of 5,500 gallon capacity was also installed to reduce the asynchronous nature in the availability and demand of thermal energy for residential use. The total equipment and installation cost of implementing Option 4 was estimated to be $1,031,545.

8.2.4 Maintenance Costs

Maintenance costs for this analysis included maintenance and operating costs for prime movers, both vapor compression and absorption refrigeration systems, space heating and space cooling equipment, both air-cooled and water cooled heat rejection equipment, auxiliary boilers, air to water and air to air heat exchangers and hot water and chilled water piping systems.

Maintenance costs were primarily adopted from the RSMeans facilities maintenance and repair cost data catalogue (RSMeans 2009) and other relevant sources. RSMeans provides estimates on person-hours, labor rates and materials required to maintain building systems. Because data available from RSMeans was for a period of several years, an annual cost for maintenance had to be interpolated. Also, because of this reason it was decided not to incorporate an escalation rate to assess the maintenance costs. Unit maintenance costs for the certain key components of the CHP facility are provided in Table 8-5. Unit maintenance costs for the corresponding base-case scenarios are provided in Appendix H.

Table 8-4: Capital Costs for the Four Options

Option No.	Capital Costs	
	Relative Base-Case Scenario ($)	CHP Scenario ($)
Option 1	$11,500	$913,498
Option 2	$113,088	$1,175,543
Option 3	$56,249	$759,711
Option 4	$84,757	$858,273

Table 8-5: Unit Maintenance Costs of Components of the CHP Scenario

Item	Unit Costs		Reference
	Unit	Cost ($)	
IC Engine	$/MWh	$8	ORNL 2004
Absorption Chiller	$/ton/yr	$18	RSMeans 2009
Air-cooled Heat Rejection	$/ton/yr	$4	RSMeans 2009
Water-cooled Heat Rejection	$/ton/yr	$2	RSMeans 2009

Table 8-6: Maintenance Costs for the Four Options

Option No.	Maintenance Costs	
	Relative Base-Case Scenario ($)	CHP Scenario ($)
Option 1	$2,065	$26,101
Option 2	$20,789	$26,480
Option 3	$6,882	$24,816
Option 4	$8,532	$25,390

8.2.4.1 Maintenance Costs for the Relative Base-case Scenario of the Four Options

Maintenance costs for the base-case of each option were different and depended on the system selected for the analysis. The costs primarily involved the maintenance of a vapor compression refrigeration system and accompanying air-cooled heat rejection units, which would be replaced in the four options. The maintenance costs for the four base-case scenarios are summarized in Table 8-6. The costs for the base-case are summarized in the following points:

- The relative base-case maintenance costs for Option 1 included maintenance costs for a 25 ton vapor compression refrigeration unit and an accompanying air-cooled heat rejection equipment. The total maintenance costs for the base-case were estimated to be $2,065.
- The relative base-case maintenance costs for Option 2 included maintenance costs for a 15 ton mechanical sub-cooler and an accompanying air-cooled heat rejection unit. The costs also included maintenance charges for 127 ton packaged space cooling and heating units that are replaced by alternate systems in this option. The total maintenance costs for base-case equipment in Option 2 are $20,798.

- The relative base-case maintenance costs for Option 3 included maintenance costs for a 50 ton vapor compression medium temperature refrigeration system, and a corresponding air-cooled heat rejection equipment. The total maintenance costs for base-case equipment in Option 3 are $6,882.
- The relative base-case maintenance costs for Option 4 included maintenance costs for a 62 ton vapor compression medium and low temperature refrigeration system, and a corresponding air-cooled heat rejection equipment. The total maintenance costs for base-case equipment in Option 4 are $8,532.

8.2.4.2 Maintenance Costs for the CHP Scenario of the Four Options

Maintenance costs for the CHP scenario of each option were different and depended on the system selected for the analysis. The maintenance costs for the four options are tabulated in Table 8-6. The maintenance costs for the four options are summarized in the following points:

- For Option 1, maintenance costs include those for a 300 kW IC engine, a 15 ton absorption chiller and an appropriately sized water and air cooled heat rejection units. The total maintenance cost for the CHP scenario in Option 1 was estimated to be $26,101.
- For Option 2, maintenance costs include those for a 300 kW IC engine, a 142 ton absorption chiller and an appropriately sized water and air cooled heat rejection unit, and heat exchangers. The total maintenance cost for the CHP scenario in Option 2 was estimated to be $ 26,480.
- For Option 3, maintenance costs include those for a 300 kW IC engine, a 50 ton absorption chiller and an appropriately sized water and air cooled heat rejection unit, an auxiliary hot water boiler, and heat exchangers. The total maintenance cost for the CHP scenario in Option 3 was estimated to be $ 24,816.
- For Option 4, maintenance costs include those for a 300 kW IC engine, a 62 ton absorption chiller and an appropriately sized water and air cooled heat rejection unit, an auxiliary hot water boiler, and heat exchangers. The total maintenance costs for the CHP scenario in Option 4 are $ 25,390.

8.2.5 Operation Costs

In general, the electrical utilities have four categories for their rate structure (Caton 2010). These include: generation costs, transmission costs[7], distribution costs[8] and customer costs[9]. The above-mentioned costs are reflected in the customers' bills as electric demand costs, electricity energy costs and customer service costs. Every utility has a unique rate structure. Hence, to perform an economic assessment the electric rate structure of the utility serving the store was needed. The utility structure of Bryan Texas Utilities for commercial and residential customers was adopted for this analysis (BTU 2012). For the natural gas rates the analysis adopted the rate structure implemented by Atmos Energy (Atmos Energy 2012). Atmos Energy is the largest natural gas distributor in the State of Texas.

When considering the utility costs of electricity the rates vary from month to month. Hence, average electricity rates for the State of Texas published by the EIA in 2011 were considered instead (US EIA 2011). For the State of Texas, in 2011 the electricity prices for residential sector were reported to be 11.4 cents/kWh and for the commercial sector they were reported to be 8.8 cents/kWh (US EIA 2011). Electricity demand charges were obtained from Bryan Texas Utilities and were reported to be 7.03 $/kW. It should be noted that demand charges were billed to commercial customers only. For residential customers a monthly customer charge[10] of $7.50 and for commercial customers a monthly customer charge of $22.12 was used. The rates are reported in Table 8-7 below. In addition to the above mentioned rates an escalation rate of 1% was selected on an arbitrary basis and implemented in the analysis.

Natural gas utilities have a simpler rate structure with the customer charged for monthly consumption as well as a monthly service charge. For residential customers the cost of natural gas consumption was reported to be $2.51 per Mcf. While for commercial customers, the cost of natural gas consumption was estimated[11] to be $1.02 per Mcf. The monthly customer charges for natural gas usage by residential customers were reported to be $7.50. The monthly customer charges for natural gas usage by commercial customers were reported to be $16.75. The rates are reported in Table 8-7 below. In addition to the above mentioned rates an escalation rate of 1% was selected and implemented in the analysis.

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[7] Transmission costs include first costs and operating expenses of transmission equipment.
[8] Distribution costs include costs of distributing power from distribution points to customers.
[9] Customer costs include costs for customer service, advertising, accounting, electric lines and metering.
[10] Monthly customer charge includes a fixed monthly amount that covers the cost of providing service to the location such as maintenance of electric lines, meter reading and other costs.
[11] Assuming 1Mcf = 1 MMBtu

Table 8-7: Utility Costs Implemented in the Analysis

Item	Unit Cost ($)	Unit	Reference
Electricity & Natural Gas Costs			
For Grocery Store			
Electricity Costs			
Demand	$7.03	$/kW	Bryan Texas Utilities
Energy	$0.09	$/kW-hr	US EIA 2011
Monthly Charges	$22.12	$/month	Bryan Texas Utilities
Natural Gas Costs			
Energy	$1.03	$/10^6Btu	Atmos Energy
Monthly Charges	$16.75	$/month	
For Residential Buildings			
Electricity Costs			
Energy	$0.1139	$/kW	US EIA 2011
Monthly Charges	$8.33	$/month	Bryan Texas Utilities
Natural Gas Costs			
Energy	$2.5	$/10^6Btu	Atmos Energy
Monthly Charges	$7.5	$/month	
Escalation Rates			
For Residential	1%		Assumption
For Commercial	1%		Assumption

8.3 Economic Assessment

Using the economic measures and assumptions stated above, an economic assessment was performed for the four options of CHP facility being proposed at the grocery store.

8.3.1 Analysis for the Four Options

When assessing the four options of the CHP systems installed in the grocery store, it was convenient to record and report the economic assessment on an annual basis. The annual totals for electricity and thermal loads were obtained by summing the hourly energy-use, which resulted from implementing these systems at the grocery store as well as the energy-use of the surrounding residential units.

In the first step, the capital costs and maintenance costs for base-case scenario and for the four options were determined. These costs included capital costs and maintenance costs for the replaced or modified equipment for the base-case store. These costs also included capital costs and maintenance costs for the option of the CHP systems implemented in the grocery store as well as the modifications that had to be performed to the grocery store and the residential units. These costs have been discussed in Section 8.2 of this chapter. These costs are presented in Table 8-8 and Table 8-9.

In the second step, operation costs were determined on an annual basis for the base-case scenario. This included determining the annual electricity and natural gas costs for the base-case grocery store and the corresponding number of residential units that would potentially be served by the CHP system. These costs are presented on an annual basis in Table 8-8 of this chapter. Corresponding calculations for operation costs of the four CHP scenarios were also performed. The calculations also include the amount of energy (both electricity and thermal energy) that is generated as surplus and can potentially be exported to the surrounding residential community. These costs have been presented on an annual basis in Table 8-9 of this chapter. The numbers from the base-case scenario and the corresponding CHP scenario were compared and the resultant savings were determined. These amounts were then spread over a period of 25 years, which was assumed to be the economic life of the CHP installation. An escalation of 1% was included for the electricity and fuel costs of both the base-case scenario and the CHP scenario. Finally, annual savings for a period of 25 years were calculated on the implementation of the CHP scenario.

In the third step, savings from the CHP scenario were modified to account for depreciation and income tax over the period of 25 years, which is assumed to be the lifespan of a CHP system. A straight-line depreciation schedule is used and is calculated to be 10% over a period of 10 years. An income tax rate of 29% is assumed for this analysis. In addition, a discount factor of 5.87% was also applied to the calculations. Finally, the net present values (NPV), internal rate of return (IRR), simple payback and investors simple return on investment (IROR) were calculated and assessed[12]. The summary of the results is presented in Table 8-10 below.

[12] These assessment indices have been described in the literature review of this study. The equations used to calculate these indices have been presented in Appendix C of this study.

8.3.2 Conclusions from the Economic Assessment of the Four Options

From the above results it can be seen that Option 4 performed the best with the greatest values for NPV of 2,747,398 and an IRR of 30%. The option also provided the smallest time until zero NPV of 4.0 years, a simple payback period of 2.7 years, and an IROR of 57%. As seen in Table 8-9 this option also had the second lowest capital cost when compared to the other options. Option 3 performed similar to Option 4 and proved to be the next best option with an NPV of 2,429,407 and an IRR of 30%. The option provided a time period of 4.1 years to reach zero NPV, a simple payback period of 2.7 years, and an IROR of 56%. As seen in Table 8-9, Option 3 had the lowest capital cost when compared to the other options. For both Option 3 and 4, most of the thermal energy generated by the CHP system in the grocery store was absorbed all year round within the store itself via means of installing an appropriate absorption refrigeration system to meet the medium and low temperature refrigeration requirements of the grocery store. These options decreased the amount of surplus thermal energy that was available for consumption by the surrounding residential units. This in turn decreased the capital costs of the equipment required for transporting and storing thermal energy for residential thermal energy usage. On the other hand, as seen from Table 8-8 and Table 8-9, since most of the refrigeration end-use energy consumption was met by thermally driven refrigeration equipment the amount of surplus electricity available for consumption by the surrounding units was increased. This increased the revenue obtained from the sale of surplus energy to residential customers with cost of electricity being higher than the cost of thermal energy.

Option 2 was ranked third and provided an NPV of 2,399,552 and an IRR of 22%. The option provided a time period of 6.0 years to reach zero NPV, a simple payback period of 3.7 years, and an IROR of 42%. As seen in Table 8-9, this option had the highest capital cost when compared to the other options. In this option the absorption refrigeration system was configured to meet the sub-cooling as well as the space cooling loads of the grocery store. Given the nature of space cooling loads, the thermal energy required by the chillers is maximized during summer months. On the other hand, during the winter months when there is minimal requirement for space cooling, the waste thermal energy is available as surplus for the surrounding residential community as most of it cannot be absorbed by the grocery store. Hence, the resultant capital costs have to incorporate a substantial absorption refrigeration system to meet the space cooling loads of the grocery store. In addition, in order to absorb the surplus thermal energy during times when space cooling is not required, the option has to incorporate the costs for equally substantial thermal energy distribution and storage system to meet the requirements of the surrounding residential units. Hence although the surplus energy sold to residential customers was the second highest among the four options, high first costs incurred in this option provided less economic benefit.

Option 1 was ranked last. The option provided an NPV of 1,727,169 and an IRR of 19%. The option provided a time period of 7.1 years to reach zero NPV, a simple payback period of 4.1 years, and an IROR of 39%. As seen in Table 8-8 and Table 8-9, this option had the second highest capital cost when compared to the other options. In this option the absorption refrigeration system was configured to meet the requirements of the mechanical sub-cooler. Hence, only a small portion of the thermal energy generated by the IC engine was consumed within the boundaries of the grocery store to operate the absorption chiller and to meet the space heating requirements of the grocery store. As a consequence, the surplus thermal energy available for residential consumption greatly exceeds the surplus thermal energy obtained from all other options. Hence, while the capital costs for the grocery store itself were low, overall capital costs were high to account for the equipment implemented to absorb this thermal energy in the surrounding residential units. In addition, since most of the electricity generated by the IC engine was consumed within the boundaries of the grocery store. Hence, smaller amount of surplus electricity available from the two options resulted in lower revenues being generated on selling surplus energy to residential customers with cost of electricity being higher than the cost of thermal energy.

Table 8-8: Energy Costs, Equipment and O&M Costs for the Corresponding Base-Case Scenarios in the Four Options

Option No.	Base-Case					
	Grocery Store Energy Costs		Residential Energy Costs		Rel. Equipment, Rel. Operation & Maintenance Costs	
	Electricity ($)	Nat. Gas ($)	Electricity ($)	Nat. Gas ($)	Rel. Equip. Costs ($)	Rel. O&M Costs ($)
1	207,312	2,048	60,259	13,713	11,500	2,065
2	224,323	1,999	89,560	12,202	113,088	20,798
3	218,051	2,058	80,566	4,632	56,249	6,882
4	223,382	2,059	100,451	3,774	84,757	8,532

Table 8-9: Energy Costs, Equipment and O&M Costs for the Four Options

Option No.	CHP Facility							
	Energy Costs			Equipment & Maintenance Costs				
	IC Engine + Boiler Fuel Costs ($)	Electricity from Utilities ($)	Surplus Energy to Community ($)	First Cost of CHP System ($)	IC Engine O&M Costs ($)	HEX, Heat Rej. O&M Costs ($)	Boiler O&M Costs ($)	Abs. Chiller O&M Costs ($)
1	25,533	2,009	73,919	913,498	21,024	4,673	0	404
2	25,678	26	101,586	1,175,543	21,024	3,908	0	1,549
3	26,036	567	85,060	759,711	21,024	2,499	130	1,164
4	29,737	266	105,264	858,273	21,024	2,475	185	1,706

Table 8-10: Summary of the Economic Assessment of the Four Options

Economic Assessment Measures		Option 1	Option 2	Option 3	Option 4
Net Present Value (NPV)	$	1,727,169	2,399,552	2,429,407	2,747,398
Internal Rate of Return (IRR)	%	19%	22%	30%	30%
Simple Pay-Back	Years	4.1	3.7	2.7	2.7
Investors Simple Return on Investment (IROR)	%	39%	42%	56%	57%
Time until zero NPV	Years	7.1	6.0	4.1	4.0

8.4 Sensitivity Analysis

The parameters selected for any economic analysis are only as accurate as the data used for the analysis. Therefore, it is easy for any analysis to contain uncertainties, which include: incorrect estimates, unforeseen changes in the future, or inaccuracies in one or more of the basic assumptions. Hence a sensitivity analysis was performed to assess the impact of several of the parameters used in the economic assessment. Option 4 was selected for the sensitivity analysis as it demonstrated the highest savings.

Using the guidance from Baxter (1997) several parameters were selected for the sensitivity analysis. These parameters are categorized according to the following:

- Prime mover (IC Engine) size
- Design parameters of the prime mover
- Electric and thermal loads
- Electric and fuel costs
- Capital and O&M costs
- Financial and economic rates

In this analysis, except for the prime mover size, the selected parameters were varied by ±25% of their original value and the resultant percentage change in NPV was recorded[13]. The prime mover size was varied depending on the engine sizes available and the resultant percent change in IRR and time to zero NPV were recorded. Results of this analysis are presented in Figure 8-2 to Figure 8-7 of this chapter.

8.4.1 Prime Mover (IC Engine) Size

Varying the size of the IC engine is typically investigated to assess the project feasibility. For this analysis different sizes of the prime mover were selected from the same manufacturer[14]. Engine sizes of 300 kW, 350 kW, 400 kW, 440 kW, 500 kW and 600 kW were selected in addition to the 300 kW base case IC engine. Unfortunately, engine sizes smaller than 300 kW could not be assessed because no such sizes are produced by this manufacturer.

On varying the IC engine size corresponding design parameters of the selected prime mover such as the fuel rate, exhaust flow, exhaust temperature and heat rejected to the jacket water were also altered accordingly. The variations in the design parameters of the IC engine

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[13] Smaller values of NPV indicate longer payback periods. Bigger values for NPV indicate shorter payback periods.
[14] The engine sizes and corresponding design characteristics were selected from Cummins. Specifications for these IC engines are provided in Appendix G of this study.

impacted the electricity and thermal energy available to the store as well as surplus energy available to the residential buildings. Accordingly sizes of parameters such as the auxiliary boiler, heat exchangers, thermal storage tank and hot water distribution system were varied accordingly.

Bigger engine sizes provided more electricity and thermal energy for meeting the requirements of the grocery store resulting in a smaller dependence of the grocery store on electricity and natural gas utilities. This resulted in smaller sizes for auxiliary boilers and smaller costs to purchase and install such boilers. On the other hand, a bigger engine size resulted in more surplus energy available to serve a greater number of residential units. This increased the cost of transferring, storing and distributing the surplus thermal energy made available for these residential buildings. Trends for percentage change in time to zero NPV and IRR over the base-case values are presented in Figure 8-2. Greater positive values for percentage difference indicate shorter periods to reach zero NPV and greater negative values for the percentage difference on the secondary y axis indicate a greater value for IRR.

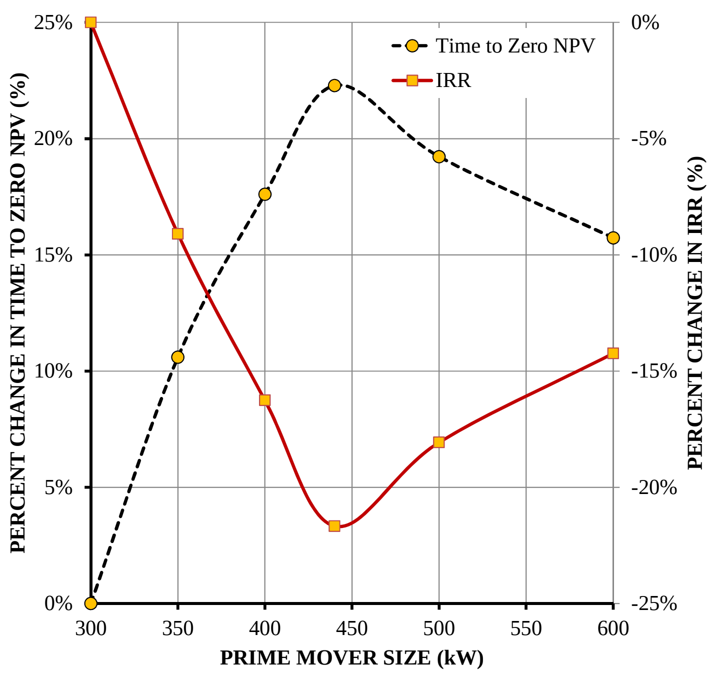

Figure 8-2: Impact on Percent Change in Time to Zero NPV and IRR on Varying the Prime Mover Size for Option 4

8.4.2 Sensitivity to Change in Design and Economic Parameters

Sensitivity to change in design and economic parameters is provided in Figure 8-3. Design parameters of the IC engine selected for the sensitivity analysis included the fuel rate, exhaust rate, exhaust temperature and thermal energy from jacket coolant. The following observations were made:

- A linear pattern was observed in the percent change in NPV on varying the four parameters. The slopes are reported in parenthesis in Figure 8.3.
- The percent change in the NPV was most sensitive to variation in the thermal energy from jacket water. Increasing the amount of thermal energy available from the jacket coolant increased the provision of surplus thermal energy available to be absorbed by the grocery store and the surrounding residential buildings and increased the cost of equipment to transfer, store and distribute this surplus thermal energy. The resultant NPV decreased with the increase in thermal energy from jacket water in the ratio of 10:1.2.
- The percent change in the NPV was similarly sensitive to the variation in fuel rate. Increasing the fuel rate decreases the thermal efficiency of the IC engine, with less electricity generated per unit of fuel input to the engine. Increase in fuel rate also implies that more thermal energy is wasted in the production of electricity, increasing the cost of equipment to transfer, store and distribute this surplus thermal energy. The resultant NPV decreased with the increase in fuel rate in the ratio of 10:1.
- On the other hand, change in values of the rate of exhaust gas and temperature of the exhaust gas from the IC engine provided less than 1% change in the NPV. Increase in both these quantities reduced the dependence of absorption chillers on auxiliary burners which in turn reduced the requirements of natural gas to operate the burners. The resultant NPV increased with the increase in exhaust rate and exhaust temperature in the ratio of 10:-0.3.

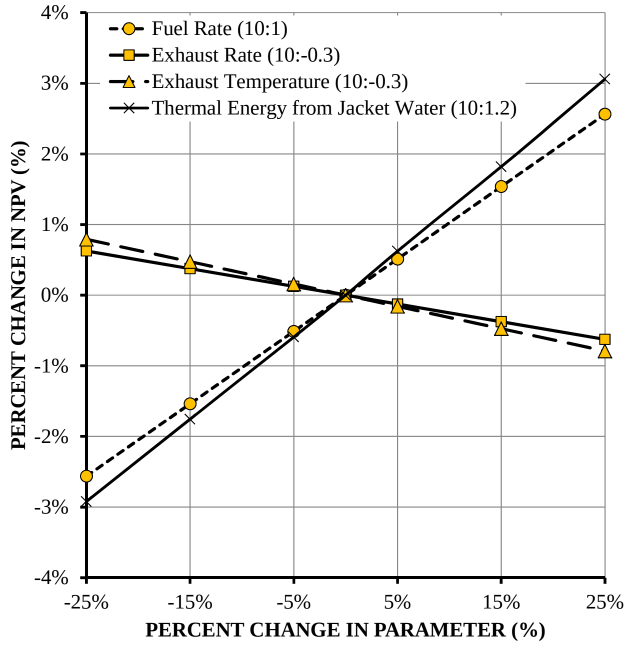

Figure 8-3: Impact on Percent Change in NPV on Varying the Equipment Design Parameters for Option 4

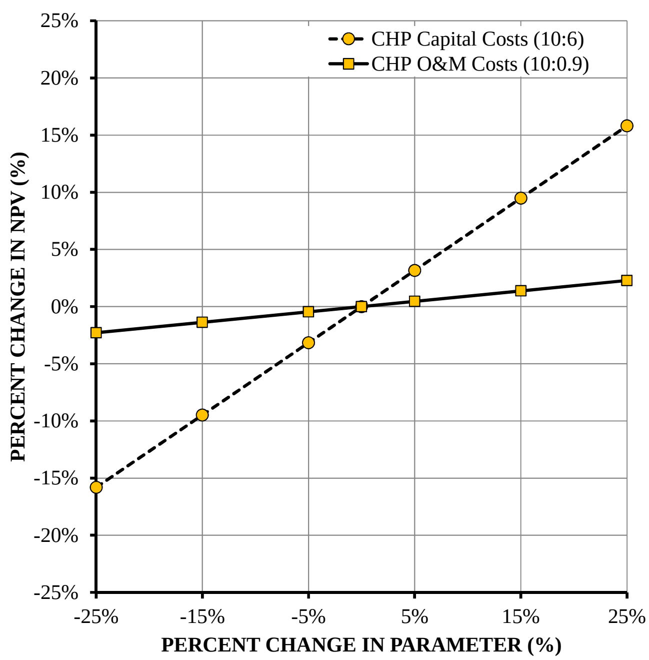

Figure 8-4: Impact on Percent Change in NPV in Varying the Capital and O&M Costs for Option 4

Sensitivity to change on varying the capital costs and O&M costs of the CHP system implemented in the grocery store are provided in Figure 8-4. The following observations were made:

- A linear pattern was observed in the percent change in NPV on varying the two parameters. The slopes are reported in parenthesis in Figure 8.4.
- A 15% variation in the percent change in NPV was observed on varying the capital cost by ±25%. Decreasing the capital costs increased the NPV of the tested case resulting in a negative percent change in the NPV and vice versa in the ratio of 10:6.
- On the other hand, less than ±5% change in NPV was observed on varying the O&M costs by ±25%. Decreasing the O&M costs increased the NPV of the tested case resulting in a negative percent change in the NPV in the ratio of 10:0.9. Varying the O&M costs provide similar trends to the trends projected by varying the capital costs. However, the magnitude of variation is smaller because these costs form only a small portion of the total costs involved in the purchase, installation, operation and maintenance of the CHP system in the grocery store.

In the next set of comparisons, the electricity loads and thermal energy loads were varied in the grocery store. The sensitivity to change on varying the electricity loads and thermal energy loads of the implemented CHP system are provided in Figure 8-5. The following observations were made:

- A linear pattern was observed in the percent change in NPV on varying the two parameters. The slopes are reported in parenthesis in Figure 8.5.
- Varying the electricity loads by ± 25% provided a percent change in NPV in the range of ± 40%. Percent decrease in electricity loads of the tested case provided an increase in the corresponding NPV of the tested case resulting in a negative percent change in NPV and vice-versa in the ratio of 10:16.
- On the other hand, varying the thermal loads in the grocery store by ± 25% resulted in change in the NPV within ±2%. Percent decrease in the thermal loads of the tested cases provided an increase in the corresponding NPV of the tested case resulting in a negative percent change in NPV and vice-versa in the ratio of 10:0.7.

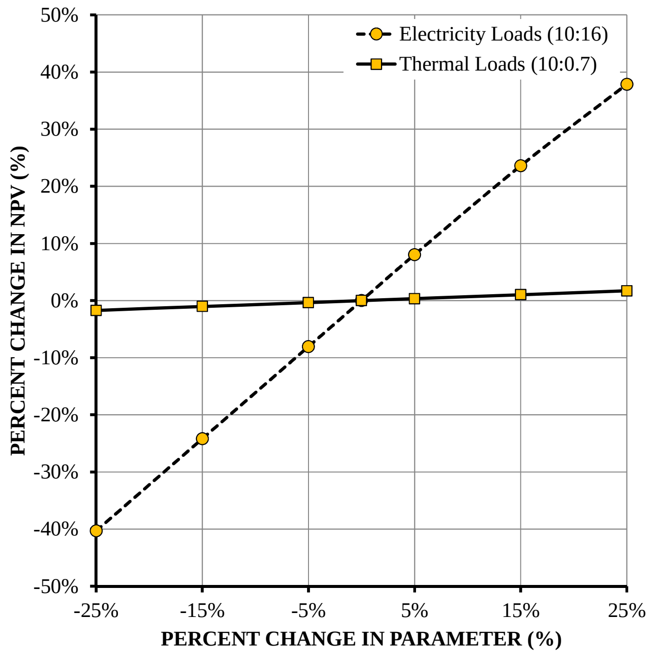

Figure 8-5: Impact on Percent Change in NPV on Variation in Electricity and Thermal Loads for Option 4

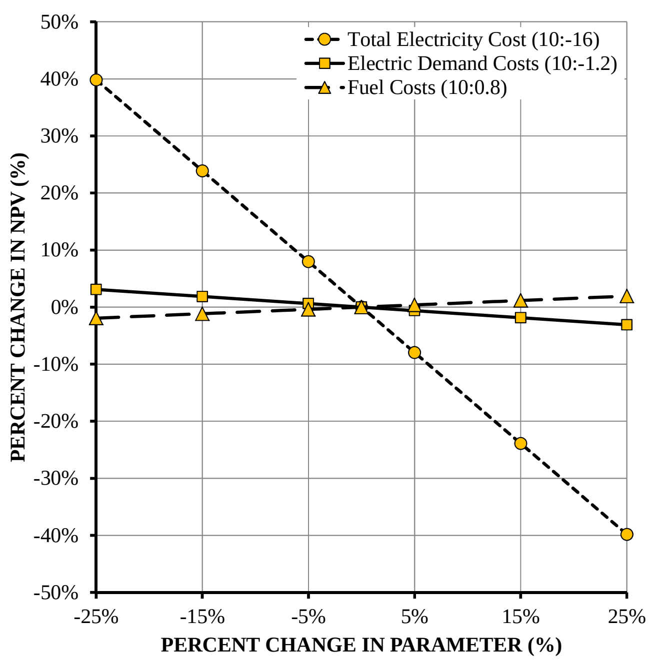

Figure 8-6: Impact on Percent Change in NPV on Variation in Electricity, Electric Demand and Fuel Costs for Option 4

Sensitivity to change on varying the utility costs for electricity, electricity demand and natural gas of the implemented CHP system are provided in Figure 8-6. The following observations were made:

- A linear pattern was observed in the percent change in NPV on varying the two parameters. The slopes are reported in parenthesis in Figure 8.6.
- Varying the cost of electricity had by far the greatest impact on the percent change in the NPV with a percent change up to ±40% corresponding to ±25% variation in the electricity costs. Percent decrease in electricity costs provided a decrease in the corresponding NPV of the tested case resulting in a positive percent change in NPV in the ratio of 10:-16.
- Varying the electricity demand costs by ±25% provides much lower percent change in the NPV, which is within ±3%. However, the trends are similar to those projected by varying the electricity costs. Percent decrease in electricity costs provided a decrease in the corresponding NPV of the tested case resulting in a positive percent change in NPV in the ratio of 10:-1.2.
- In comparison, varying the natural gas costs by ±25% provides a variation in the corresponding NPV within ±2%. In contrast to the trends exhibited by varying the electricity and electric demand costs, percent decrease the natural as costs provided an increase in the corresponding NPV of the tested case resulting in a negative percent change in NPV. Percent decrease in natural gas costs provided an increase in the corresponding NPV of the tested case resulting in a negative percent change in NPV in the ratio of 10:0.8.

Sensitivity to change on varying the income tax rates, discount factor and escalation rates for electricity and natural gas costs for the implemented CHP system are provided in Figure 8-7. The following observations were made:

- A linear pattern was observed in the percent change in NPV on varying the parameters except the discount factor. The slopes are reported in parenthesis in Figure 8.7.
- Varying the income tax rate by ±25% results in percent change in NPV by ±13%. Percent decrease in the income tax rates resulted in an increase in the corresponding NPV of the tested cases resulting in a negative percent change in NPV in the ratio of 10:5.6.
- Varying the discount factor by ±25% of the currently assumed rate results in percent change in NPV by ±23%. Percent decrease in the discount factor rates resulted in an increase in the corresponding NPV of the tested cases resulting in a negative percent change in NPV.

- Varying the electricity cost escalation rate by ±25% within the currently assumed rate results in a percent change in NPV by ±3.8%. The trends are inverse of what was observed for both variation in the income tax rates and discount rates. Percent decrease in the electricity cost escalation rates decreased the corresponding NPV of the tested cases resulting in a positive percent change in the NPV in the ratio of 10:-1.7.
- The trends projected by increasing escalation rates of fuel costs are similar to those projected by varying the income tax rates and the discount rates. However, the resultant change in NPV is much smaller in magnitude. Varying the fuel cost escalation factor by ±25% within the currently assumed rate results in a percent change in NPV by ±0.2%. Percent decrease in the income tax rates resulted in an increase in the corresponding NPV of the tested cases resulting in a negative percent change in NPV in the ratio of 10:0.1.

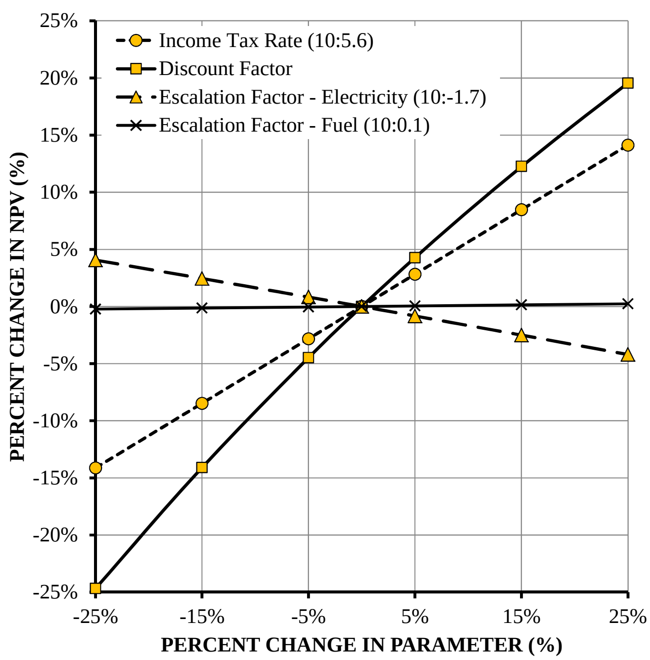

Figure 8-7: Impact on Percent Change in NPV on Variation in Income Tax Rates, Discount Factors, Escalation Rates for Electricity and Natural Gas for Option 4

8.4.3 Summary of the Sensitivity Analysis

On varying the prime mover size, the following was concluded:

- Although installation of smaller prime movers result in smaller first costs for the grocery store as well as lesser costs for transferring, storing and distributing thermal energy to the surrounding residential community, the lower first costs were outweighed by the higher costs of purchasing energy from the utilities to meet the requirements of the grocery store and the surrounding residential buildings.
- On the other hand, the provision of increased energy from larger prime movers both to the grocery store as well as the surrounding residential community was outweighed by the increased first costs and maintenance costs associated with large systems as well as the increased piping network and storage facilities required to distribute this thermal energy to the surrounding residential community.
- The optimal size for the IC engine best suited to operate 440 kW was determined to be the optimal size for the prime mover to be selected for Option 4.

On varying the engine design parameters, costs, energy consumption loads, utility costs and an economic index, the following was concluded:

- The analysis was most sensitive to three economic parameters. The parameters include electricity loads, electricity costs and discount factor rates.
- The analysis was also found to be moderately sensitive to the income tax rates and capital costs.
- The high first costs associated with the transfer, storage and distribution of thermal energy made variation in the design parameters of the prime mover insignificant.

8.5 Summary and Conclusions

This chapter described the economic analysis performed on the four CHP options selected in Chapter 7 to reduce energy consumption of the grocery store. The economic analysis was performed by investigating the lifecycle costs and payback periods associated with each option. Calculations for simple payback, investor's rate of return (IROR), net present value (NPV) and internal rate of return (IRR) were performed to carry out the assessment.

On assessing the four options it was concluded that Option 4 performs best with the greatest net present value (NPV) of $2,747,398 and an IRR of 30%. This option had the second lowest capital cost, which was inclusive of the CHP system installed to meet the energy requirements of the grocery store as well as the equipment required to transfer, store and distribute thermal energy to the surrounding residential community. Most of the thermal energy generated by the CHP system in this option was absorbed all year round within the store itself by medium and low temperature absorption refrigeration systems. This arrangement decreased the amount of surplus thermal energy that was available for consumption by the surrounding residential units, which in turn decreased the capital costs of the equipment required for transporting and storing thermal energy for residential thermal energy usage[15]. On the other hand, this option increased the amount of surplus electricity available for consumption by the surrounding units.

In the next step of this evaluation, a sensitivity test was performed on Option 4. The parameters selected for this test included the prime mover size, design parameters of the prime mover, electric and thermal loads, electricity and fuel costs, capital and O&M costs, and financial and economic rates. On varying the prime mover size, an engine size of 440 kW was determined to be an optimal selection for the CHP system in the grocery store. It was also concluded that the analysis was most sensitive to variation in electricity loads, electricity costs and discount factor rates; the analysis was also found to be moderately sensitive to the income tax rates and capital costs; and the analysis was found to be least sensitive to the design parameters of the CHP system as well as fuel costs.

[15] It should be noted that the cost of transporting and storing thermal energy was found to be prohibitive by a study conducted by Phetteplace (1995) for such construction in the United States.

CHAPTER IX

SUMMARY AND CONCLUSIONS

9.1 Overview

A summary of this dissertation is presented in this chapter. The summary includes a discussion of the process adopted for the analysis and a discussion of the results, limitation and future work. The first six sections summarize the previous chapters of this study. Next, a summary of the results is presented which summarizes the overall results of this research. A discussion is provided which talks about the opportunities, challenges and limitations faced by this study. This section is followed by a conclusion in which the overall outcome of this research is discussed. Finally the limitations of this study are presented. These limitations form the basis of the last section of this report which also presents recommendations for future work.

9.2 Summary of Research Objectives

This study claimed that more efficient use of energy resources could be obtained from a decentralized approach to the generation of electricity. To prove this proposition the study considered a high energy use building such as a grocery store as part of a residential community. The intension was to assess the potential of sharing thermal energy and electricity across building boundaries in order to reduce the overall energy consumption of the building and the community combined, as compared to a traditional electric power plant.

In order assess this proposition the study first considered a conventional grocery store and assessed several energy efficiency measures in order to determine the maximum savings possible. The study then examined the option of installing a CHP system to power the grocery store and a portion of the surrounding community in order to further reduce the total source energy consumption of the grocery store and the community combined.

9.3 Summary of Methodology

In order assess this proposition two building types were considered – a grocery store and a multi-family building. The study was conducted for the hot and humid climate of Texas (Climate Zone 2). The study was divided into two parts. In the first part the study developed a calibrated grocery store model and investigated the potential efficiency measures available to reduce the energy consumption of the grocery store. In the second part, the study investigated further reduction in energy consumption levels in the grocery store by the implementation of an appropriate CHP system.

To evaluate the potential energy efficiency measures in the grocery store, the eQUEST-Refrigeration software (Ver. 3.61) was used along with TMY3 hourly weather data for College Station, Texas. To evaluate the impact of implementing a CHP system in the grocery store, a spreadsheet was created using Excel (Microsoft 2010). The spreadsheet used hourly output reports generated by the eQUEST-Refrigeration simulation program in conjunction with hourly calculations performed for the IC engine to assess the performance of the CHP system. Energy reductions from both the parts of the analysis were monitored at site and source levels. Finally, an economic assessment was performed to determine the economic feasibility of options of CHP selected for this analysis. The economic assessment was performed in terms of assessing the life-cycle costs and payback periods.

9.4 Summary of Calibration Process Implemented for the Grocery Model

In this section of the analysis a base-case building was modeled. This base-case model would subsequently be used to assess the various energy efficiency measures selected. The base-case building was modeled using the eQUEST-Refrigeration software using information provided by the case-study store, which was situated in the hot and humid climate of central Texas. Other assumptions were made, which included default values provided by the eQUEST-Refrigeration program and other reputable sources.

To ensure that the simulation model was performing correctly it was found necessary to calibrate the model using information from the case-study store. The calibration was performed using hourly data for electricity consumption of the store and monthly data for natural gas consumption. The calibration was performed using statistical indices, which include RMSE, CV(RMSE) and MBE. A set of 51 iterations was performed. The initial RMSE, CVRMSE and MBE values were 47.71, 0.10 and -6.41 respectively. The RMSE, CV(RMSE) and MBE values of the final run were established to be 32.15, 0.07 and 0.84 respectively (Table 5-13).

9.5 Summary of Energy Efficiency Measures (EEMs) Implemented in the Grocery Store

The calibrated base-case model was subsequently used in the assessment the energy efficiency measures (EEMs) selected for the grocery store. Several such measures for the grocery store were considered and assessed. These included EEMs for the building envelope, lighting, HVAC and refrigeration systems of the grocery store. The measures were first assessed individually and then combined to provide a cumulative energy savings. Energy savings were assessed in terms of site and source energy consumption.

Effective EEMS (those with greater than 5% source energy reductions) included a reduction of lighting power; implementation of heat reclaim from refrigeration compressors for space heating; installation of glass doors over open-sided display cases; installation of LEDs for display-case lighting; and installation of ECM motors for evaporator fans.

When considering site energy consumption of all the effective envelope measures grouped together provide a savings of 5.6%. In addition to the envelope measures, all the effective lighting measures grouped together provide a combined saving of 15.6%. In addition to the envelope and lighting measures, all the effective EEMs for HVAC system groped together provide a saving of 30.7%. Finally the inclusion of effective refrigeration EEMs in addition to envelope, lighting and HVAC EEMs provide a cumulative savings of 57.99% (Table 6-13 and Figure 6-17).

When considering source energy consumption of the effective envelope EEMs grouped together provide a savings of 3.3%. In addition to the envelope EEMs, all the effective lighting measures grouped together provide a saving of 21.1%. In addition to the envelope and lighting EEMs, all the effective EEMs for HVAC systems grouped together provided a saving of 30.3%. Finally, the inclusion of all the effective refrigeration EEMs to the envelope, lighting and HVAC EEMs provide a cumulative savings of 56.0% (Table 6-13).

9.6 Summary of Performance of the CHP Options Implemented in the Grocery Store

In the second part of the analysis, four options regarding CHP systems were selected for assessment. The selection criteria were based on the utilization of the thermal energy available from the CHP system. The impact of implementing these options on annual and hourly energy consumption was discussed in this chapter.

When assessing the annual energy consumption of the four options at the source, the percentage savings from implementing the CHP scenarios above the corresponding base-case scenarios was within the range of 18% and 33%, assuming that the entire amount of surplus thermal energy available from the grocery store was absorbed by the surrounding residential community (Table 7-3).

On the other hand, when assessing hourly patterns of availability of surplus energy from the grocery store and the hourly patterns of electricity and thermal energy consumption from the multi-family units, an asynchronous pattern was observed. This asynchronous pattern was removed by the installation of thermal storage systems.

9.7 Summary of the Economic Evaluation of the CHP Options

The economic analysis was performed by investigating the life-cycle costs and payback periods associated with each option. The assessment included calculations for simple payback, investor's rate of return (IROR), net present value (NPV) and internal rate of return (IRR) were performed to carry out the assessment.

An assessment of the four options concluded that Option 4 performs best with the greatest net present value (NPV) of $2,747,398 and an IRR of 30% (Table 8-10). This option had the second lowest capital cost, which was inclusive of the CHP system installed to meet the energy requirements of the grocery store as well as the equipment required to transfer, store and distribute thermal energy to the surrounding residential community.

A sensitivity test was then performed on Option 4. The parameters selected for this test included the prime mover size, design parameters of the prime mover, electric and thermal loads, electricity and fuel costs, capital and O&M costs, and financial and economic rates. On varying the prime mover size, an engine size of 440 kW was determined to be an optimal selection for the CHP system in the grocery store. It was also concluded that the analysis was most sensitive to variation in electricity loads, electricity costs and discount factor rates; the analysis was also found to be moderately sensitive to the income tax rates and capital costs. Finally, the analysis was found to be least sensitive to the design parameters of the CHP system as well as fuel costs.

9.8 Summary of Results

Figure 9.1 summarizes the results looking across the two parts of the study. The figure shows source energy consumption for the four CHP options in the grocery store and the corresponding savings associated with the reduced consumption.

The first column of each graph provides the electricity and natural gas consumption of the calibrated base-case grocery store model along with the energy consumption of the residential multi-family units that could potentially be served by the CHP facility implemented in the grocery store. The second column provides the electricity and natural gas consumption of a grocery store in which energy efficiency measures have been being implemented. In this case too, the energy consumption of the multi-family units that could potentially be served by the CHP facility was added to the energy consumption of the grocery store. The third column provides the energy consumption of the CHP option implemented in the grocery store. The CHP option not only provides energy for the grocery store but also provides energy for the residential community.

For Option 1 the following results were recorded:

- The total energy consumption of the base-case scenario is 60,089 MMBtu/yr.
- The total energy consumption for energy efficient scenario is 37,100 MMBtu/yr, which provided savings of 38% over the base-case scenario
- The total energy consumption for the CHP scenario is 27,669 MMBtu/yr, which provided 54% savings over the base-case scenario.
- The number of years taken to reach NPV was 7.1 and the IRR was calculated to be 19%.

For Option 2 the following results were recorded:

- The total energy consumption of the base-case scenario is 56,390 MMBtu/yr.
- The total energy consumption for energy efficient scenario is 41,174 MMBtu/yr, which provided savings of 27% over the base-case scenario
- The total energy consumption for the CHP scenario is 27,635 MMBtu/yr, which provided 51% savings over the base-case scenario.
- The number of years taken to reach NPV was 6.0 and the IRR was calculated to be 22%.

For Option 3 the following results were recorded:

- The total energy consumption of the base-case scenario is 58,009 MMBtu/yr.
- The total energy consumption for energy efficient scenario is 36,364 MMBtu/yr, which provided savings of 37% over the base-case scenario
- The total energy consumption for the CHP scenario is 28,034 MMBtu/yr, which provided 50% savings over the base-case scenario.
- The number of years taken to reach NPV was 4.1 and the IRR was calculated to be 30%.

For Option 4 the following results were recorded:

- The total energy consumption of the base-case scenario is 60,021 MMBtu/yr.
- The total energy consumption for energy efficient scenario is 38,886 MMBtu/yr, which provided savings of 35% over the base-case scenario
- The total energy consumption for the CHP scenario is 32,008 MMBtu/yr, which provided 47% savings over the base-case scenario.
- The number of years taken to reach NPV was 4.0 and the IRR was calculated to be 30%.

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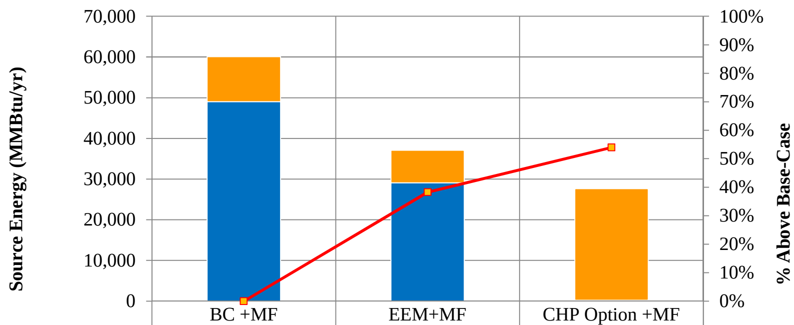
OPTION 1

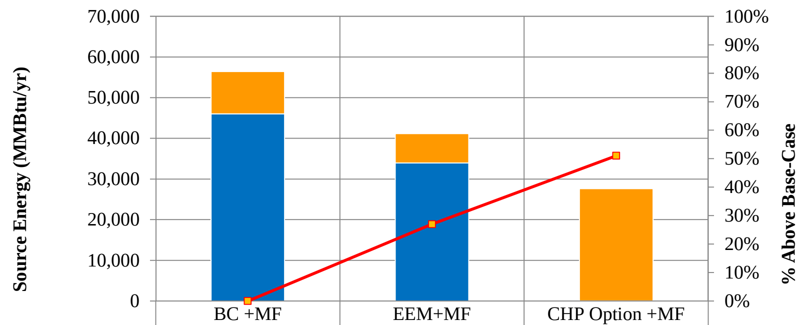
Time until Zero NPV: 7.1 years
IRR: 19%



	BC +MF	EEM+MF	CHP Option +MF
Nat.Gas	11,059	8,006	27,454
Electricity	49,030	29,094	214
% Above BC	0%	38%	54%

OPTION 2

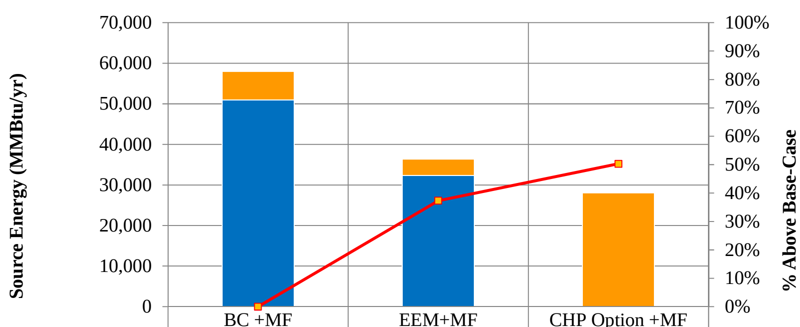
Time until Zero NPV: 6.0 years
IRR: 22%



	BC +MF	EEM+MF	CHP Option +MF
Nat. Gas	10349	7244	27632
Electricity	46040	33930	3
% Above BC	0	27%	51%

OPTION 3

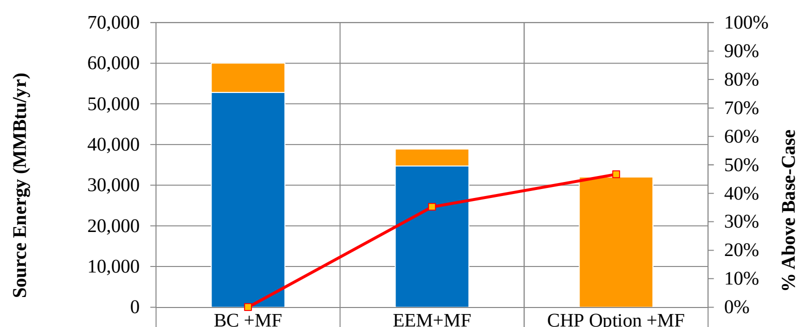
Time until Zero NPV: 4.1 years
IRR: 30%



	BC +MF	EEM+MF	CHP Option +MF
Electricity	7054	4012	27996
Nat. Gas	50955	32353	38
% Above Code	0	37%	50%

OPTION 4

Time until Zero NPV: 4.0 years
IRR: 30%



	BC +MF	EEM+MF	CHP Option +MF
Nat. Gas	7192	4151	32008
Electricity	52829	34735	0
% Above BC	0	35%	47%

Figure 9-1: Summary of Reduction in Source Energy Consumption for the Four Options

In conclusion, 47% - 54% savings in source energy over the base-case scenario consumption were achieved by implementing the options for CHP described in this study. The corresponding payback period calculated in number of years to zero NPV was in the range of 4.0 to 7.1 years and the IRR was in the range of 19% - 30%

9.9 Limitations

The limitations of this study can be categorized under three broad categories which include:

- Limitations due to the design of building systems,
- Limitations due to selection of individual design software, and
- Limitations due to the integration process.

Each of these categories is elaborated in the sub-sections that follow.

9.9.1 Limitations due to Design of Building Systems

Several assumptions had to be made in the sizing and costing of equipment due to level of detail provided in the design of systems. These assumptions apply to Chapter 8, CHP Options for the Grocery Store; and Chapter 9, Economic Evaluation of CHP systems. These include the following:

- The analysis was limited to assessing the performance of the CHP system that is integrated with the electricity grid. A stand-alone CHP system that is completely independent from the electricity grid was not assessed.
- The analysis was limited to evaluating a single configuration for layout of equipment in the grocery store. Other configurations were not assessed.
- No specific layout was proposed for the hot water and chilled water piping of the store. As a result, pumping power required for the operation of circulation pumps had to be approximated. In addition, costs for hot water and chilled water piping had to be approximated.
- No specific layout was investigated or proposed for the surrounding residential buildings. This created a challenge to assess the costs associated with the distribution of surplus thermal energy available from the store to the multifamily units. These include heat losses associated with buried piping, piping and pumping costs.
- Surplus electricity and thermal energy from the grocery was assumed to be fully absorbed by the neighboring residential units.

- This study did not model storage of either electricity or chilled water. Thermal energy storage was modeled using hot water storage tanks.

9.9.2 Limitations Due to Selection of Individual Design Software

This study was limited to the capabilities of the analysis programs and methods used. All of these assumptions apply to Chapter 7, Energy Efficiency Measures for the Grocery Stores. These include the following:

- This selection of the excluded the analysis of a number of systems, which included exploring the performance of renewable energy systems such as the use of wind and solar power, biomass and gasification of wasted food products; ground source heat pumps; innovative strategies in HVAC and refrigeration equipment such as DOAS, desiccant dehumidification systems, secondary loop refrigeration systems, multiplex refrigeration systems, flooded evaporators; exploring the implementation of improved control devices such as electronic expansion valves; exploring strategies to reduce the quantity of refrigerants in order to reduce the global warming potential (GWP), and evaluating the impact of different refrigerants. In addition, the software selected for the analysis could not evaluate the interaction between the layout of refrigerated display cases in the grocery store and comfort cooling, which is an important issue to consider given the nature of supermarket operation.
- Approximate estimations were made to assess energy savings from several other energy efficiency measures using these tools. The approximations were made using information from reputable sources. For example, the impact of implementing glass doors for open refrigerated cases was modeled by reducing the infiltration loads on these display cases by 80% and increasing the conduction loads by 20% as determined by an experiment conducted by Faramarzi et al. (2002).

9.9.3 Limitations Due to the Integration Process

Several components that were modeled as part of the analysis in Chapter 8 were simplified due to the integration of outputs from the eQUEST-Refrigeration program and the results from the spreadsheet analysis. These include the following:

- Evaluation of CHP systems was performed with a spreadsheet analysis. The use of a spreadsheet analysis restricted the exploration of options for CHP systems and the various potential arrangements in the grocery store to utilize thermal energy from the CHP system.

- In addition, the size of prime movers implemented in the four CHP options was not optimized.
- The modeling of the hot water storage tank was simplified to reflect fully-mixed water storage system. A more practical model of stratified water storage system was not attempted. The viability of other options such as phase-change energy storage and packed bed exchanger storage were not explored.
- The estimating performance of absorption chillers and the auxiliary equipment that are used to operate these chillers at part load conditions; circulation pumps used in hot water and chilled water loops; heat rejection units and performance of heat exchangers. For example, the power requirements of circulation pumps were approximately estimated by the pressure difference across the equipment connected to the circulation loop. In addition, without the availability of a detailed design of piping layout in the grocery store, the length of the loop in which the circulation pump was installed had to be approximated.
- The thermal energy storage tank was sized using demand of the residential units that could not be met over a 24 hour period.

9.10 Conclusions

This study showed how to reduce energy consumption in grocery stores in hot and humid climates. The study was conducted in two steps. In the first step - the grocery store was considered as an individual entity. Efficiency measures were applied and the resultant energy consumption was assessed in terms of site and source energy consumption. In the second step a CHP facility was implemented in the grocery store. However, in this case the store was considered as part of a residential community. Surplus electricity and thermal energy from the CHP facility, when not consumed by the grocery store was absorbed by the surrounding residential community. The study also assessed the CHP facility to further reduce the energy consumption of the grocery store.

Source energy savings were in the range of 47% to 54% depending on the EEMs selected and the CHP configuration determined in the grocery store. Economic payback periods in the range of 5 to 12 years were seen. The selection of appropriate options was narrowed down to two options (Option 3 and Option 4) that utilized more thermal energy within the boundaries of the store and generated more amount of surplus energy to be absorbed by the neighboring residential buildings.

9.11 Recommendations for Future Work

The limitations cited in the previous section of this study present numerous opportunities for future work. A list of these opportunities is presented below:

- For calibrating the grocery store, sub-metered data was available only for the refrigeration compressors. In addition, only monthly data was available for natural gas consumption of the grocery store. Future work should include obtaining hourly sub-metered data for calibration. Sub-metered data should include both hourly electricity and natural gas consumption of various end-uses in the grocery store.
- In order to improve the calibration procedure, future study should include first hand observations and independent measurements such as lighting fixture and equipment counts; hourly lighting, equipment and occupancy schedules; and performing blower door tests.
- Although manufacturer's literature for efficiency curves of the HVAC and refrigeration compressors in the grocery store were available, default curves provided by the eQUEST-Refrigeration software were used. Future work should include incorporating such specifications into the calibration process.
- The selection of energy efficiency measures for the grocery store was constrained by the analysis capabilities of eQUEST-Refrigeration (Version 3.61). Efficiency measures for display cases include advanced refrigeration controls such as electronic controls for evaporator pressure regulators controls and thermal expansion valves, impact of layout of refrigerated display cases on operation of HVAC systems in the store, proper loading of products, flooded evaporators; efficiency measures for condensers include, evaporative cooling of condensers, rejecting condenser heat to the ground; for compressors measures include testing of various alternates to the semi-hermetic reciprocating compressors used in this analysis. In addition, several assumptions had to be made in order to model the impact of certain measures. Future work should include using other software such as TRNSYS for simulating refrigeration systems in the grocery store.
- Measures to reduce refrigerant charge were not considered by this analysis. In addition, the implementation of certain energy efficiency measures may have an adverse impact on the refrigerant charge, which may lead to increased emissions of global warming potential refrigerants. This affect was not considered by this analysis. No attempt was made to

calculate the change in the amount of charge. Future work may include addressing the issue of reducing refrigerant charge in addition to the energy reduction in the grocery store.

- Impact on demand reduction on the implementation of energy efficiency measures has not been considered by this study. Future work is encouraged to include assessment of demand reduction on the implementation of efficiency measures.
- Optimization is an important aspect of selecting an appropriate cogeneration system for the store. However, in this case the four options selected for analysis presented different methods of how waste thermal energy could be absorbed in the store. These options were not a result of an optimization exercise. The CHP options were selected on the basis of varying usage of thermal energy generated by the on-site production of electricity. No attempt was made to optimize the selected CHP options. Future work may include presenting and discussing an optimization method for the CHP facility being considered for the grocery store.
- Emissions reduction (i.e., SO_x, NO_x and CO_2) is an important part of assessing the performance of IC engines. This study did not investigate the emissions reductions as a result of operating IC engines. Future work should include the assessment of emission reductions in addition to energy reductions on implementing the CHP option for energy generation.
- Design of CHP systems was limited to an hourly spreadsheet analysis. Several simplifications were made that included a steady-state analysis of the CHP option selected. Several components of the CHP model such as operation of pumps and heat rejection units, piping heat losses for the various systems in the grocery store were greatly simplified. In addition, efficiency strategies for components proposed in the CHP model such as absorption chillers and heat rejection devices were not considered. Future work may include a more detailed development of a CHP assessment tool, which incorporates assessment of different configurations of CHP and the incorporation of CHP into buildings.
- Although the equations provided in the CHP spreadsheet was tested internally against each other, these equations were not compared with other software. The self-testing equations are presented in Appendix C of this report (Section C – 1). Future work may include a comparative study of results obtained from the CHP spreadsheet and the different software.
- Community-based cogeneration can have many scales of operation. However, this study focused on the grocery store being the prime consumer of energy for which the cogeneration

system operates. Future work should look at other scales for operating the CHP facility. In addition, future work may look at a more detailed design of district heating systems.

- Methods for the storage of electricity were not explored by this study.
- The study addresses grocery store and residential units only. The sharing of energy across boundaries is limited to the absorption of this energy by multiples of 8-unit multi-family apartments. The performance of other potential recipients of surplus thermal energy was not considered by this study. Future work could address the reduced energy needs of a small community, which would include not only a grocery store and residential units as well as other components of a typical community such as schools, laundries, hospitals, and restaurants. An overall idea of the future work is presented in Figure 9-2.
- This study made no attempt to improve the energy efficiency of the residential units that are used to absorb surplus energy from the grocery store. Future work should incorporate net-zero energy residential buildings in the analysis of CHP systems.
- Diversity of residential loads not considered. Future work may include the implementation of diversity factors for residential loads to provide improved residential load profiles that have to be addressed when absorbing surplus electricity and thermal energy from the grocery store.
- This study examined the process in which the energy consumption of grocery stores could be reduced towards net-zero levels. Actual attainment of net-zero energy consumption levels in stores was neither attempted nor discussed. The potential of renewable resources such as wind, biomass and solar refrigeration to produce energy for grocery stores, therefore was not examined. Future work may include investigating methods of achieving net zero levels in grocery stores using renewable energy to power the facility and to provide remaining energy requirements of the grocery store and the surrounding residential community.
- The study focuses on the performance of a grocery store and CHP system in the hot and humid climate of Texas. Reducing energy consumption of the base-case grocery store and the performance of CHP facilities in other climates has not been considered by this study. Future work should include an analysis for different climates.
- Time constraints prevented this study from performing a detailed economic assessment of the energy efficiency strategies implemented in the grocery store. The economic analysis did not consider the costs associated with the installation of energy efficiency measures in the grocery store. The current economic assessment was restricted to the assessment of the CHP

options. The economic analysis in this study did not consider the cost of extra space that would be used for the installation of a CHP system in the grocery store. Future work should incorporate these costs in the economic analysis.

- Financial arrangement, which is linked to the ownership of the CHP project, is critical to its success (Caton 2010). However, in this study the ownership of the CHP project was not discussed. Future work may include a discussion on the ownership of the CHP project and the subsequent implications.
- The study assumes the same rates of purchase from the utilities and sale of electricity to the utilities. In addition, the regulation for sale of electricity between the utilities and the CHP facility that were enforced by PURPA[1] bill passed by the US congress, were not considered. Future work may include using rates from concerned authorities[2] for selling back power to the grid. Future work may also consider the impact of PURPA and subsequent acts passed by the US congress to assess the viability of implementing cogeneration systems.

[1] Public Utility Regulatory Policies Act (PURPA) of 1978, enabled cogeneration to become a viable option for electric power generation for various facilities. The act required electric utilities to purchase electric power from the cogenerators, put a check on the discriminatory practices practiced by the utilities to provide back-up power for cogenerators, exempted certain regulations for cogenerators that pertain to electric utilities (Caton 2010).

[2] Public Utility Commission of Texas is the authority in the State of Texas that regulates the state's electric utilities.

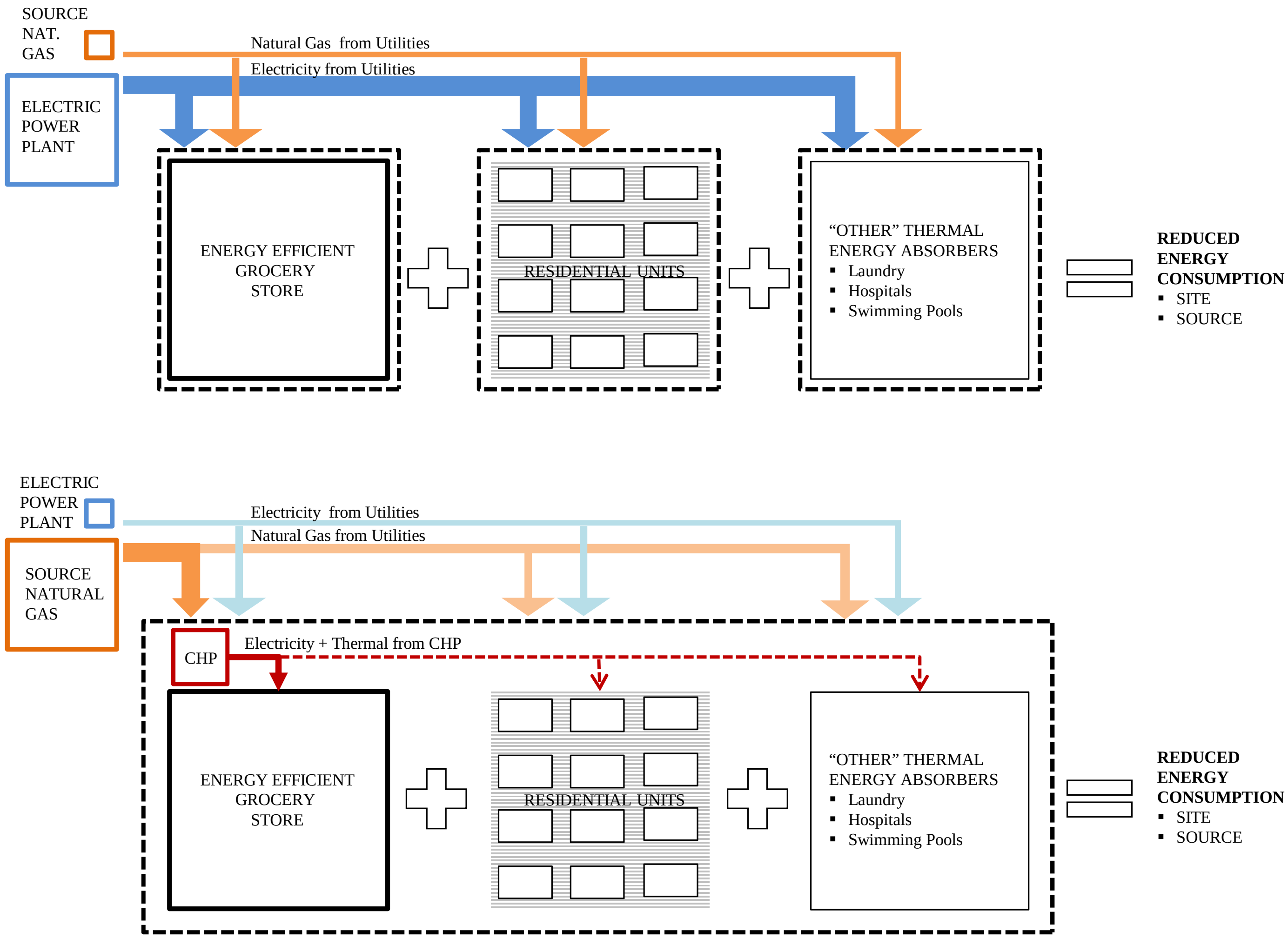


Figure 9-2: Future Research Methodology

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APPENDIX A

GROUND TEMPERATURE AND AMBIENT TEMPERATURE PROFILES FOR COLLEGE STATION, TEXAS

This appendix describes the ground temperature and ambient temperature profiles for College Station, Texas. The temperatures are obtained from the TMY3 weather data for College Station TX.

A – 1: Ambient Temperature Profiles for College Station, TX

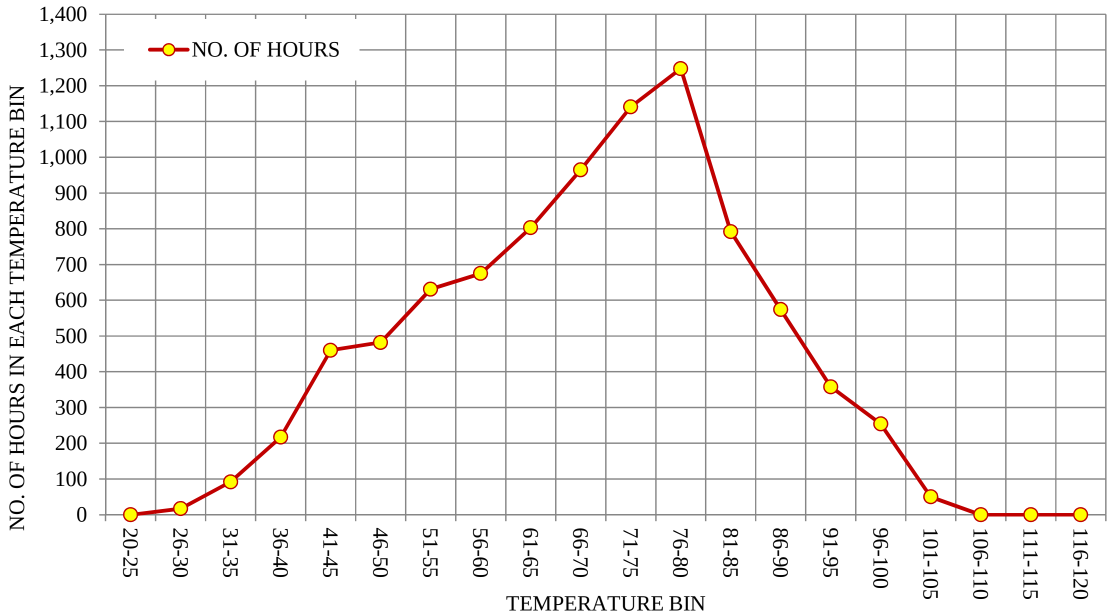


Figure A - 1: Ambient Temperature Distribution for Twelve Months for College Station, TX (Source: TMY3 Weather Data for College Station, TX)

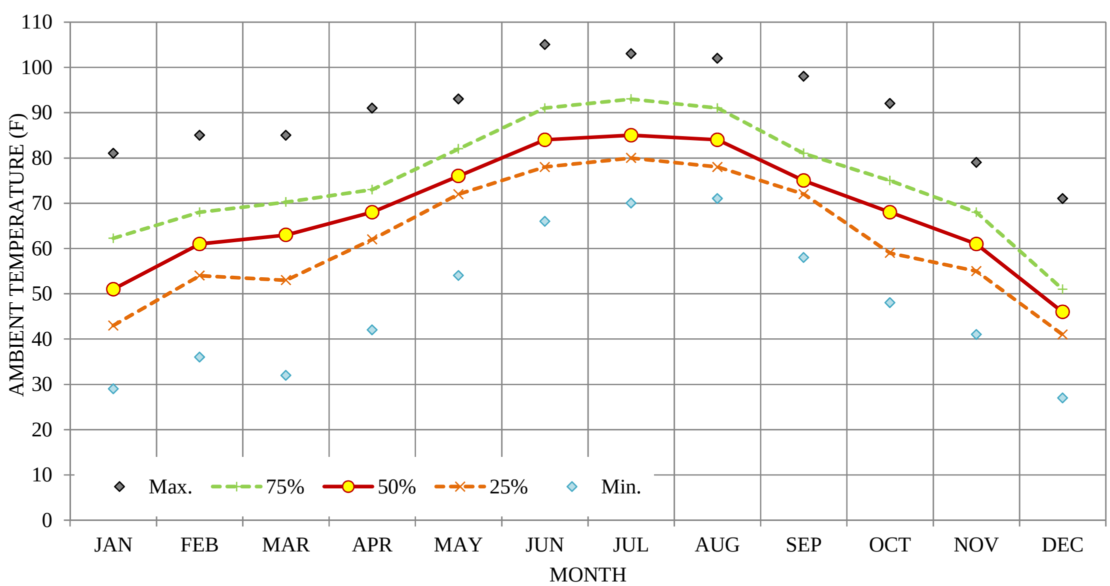


Figure A - 2: Monthly Variation in Ambient Temperature for Twelve Months for College Station, TX (Source: TMY3 Weather Data for College Station, TX)

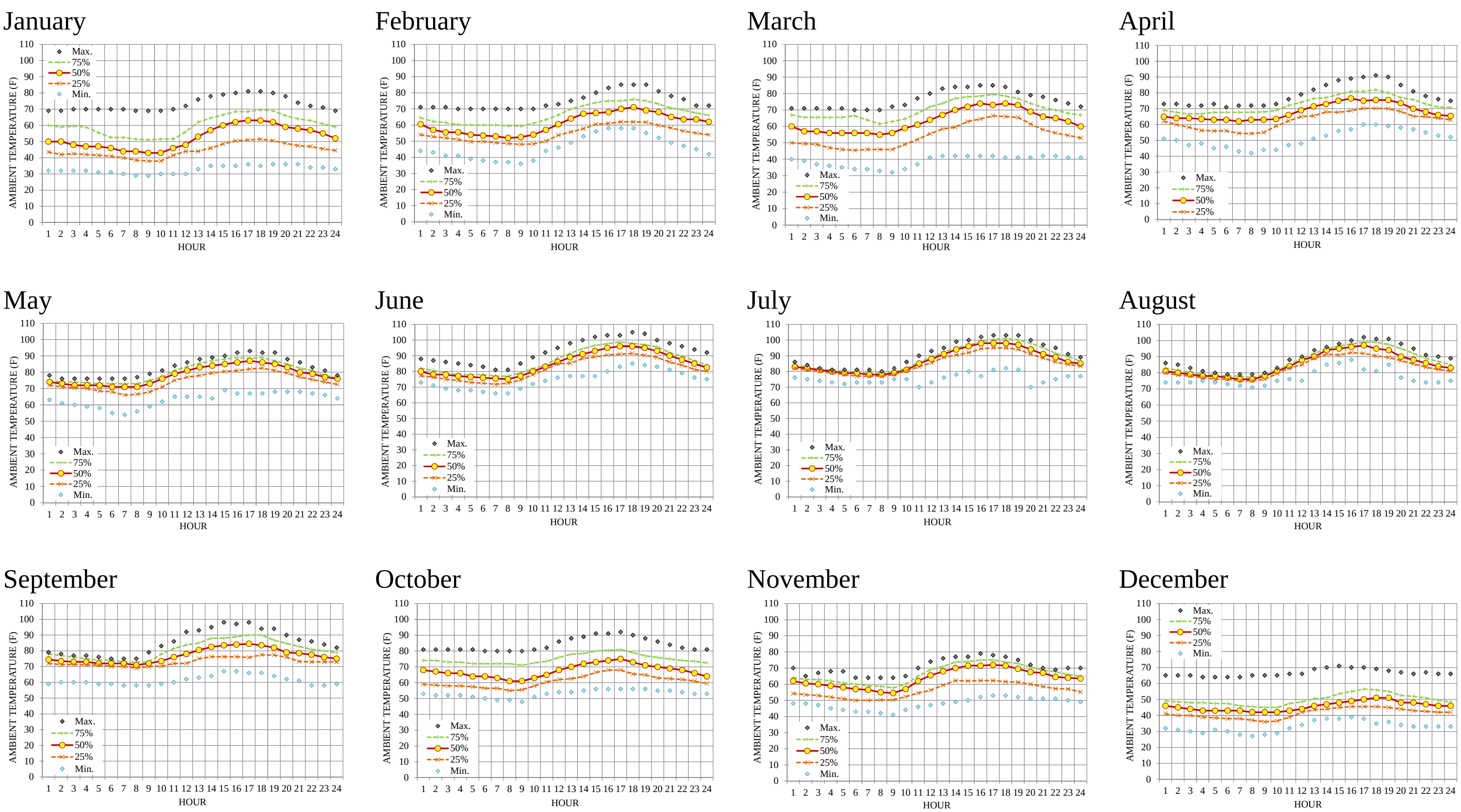


Figure A - 3: Diurnal Variation in Ambient Temperature for Twelve Months for College Station, TX (Source: TMY3 Weather Data for College Station, TX)

A – 2: Ground Temperature Monthly Profiles for College Station, TX

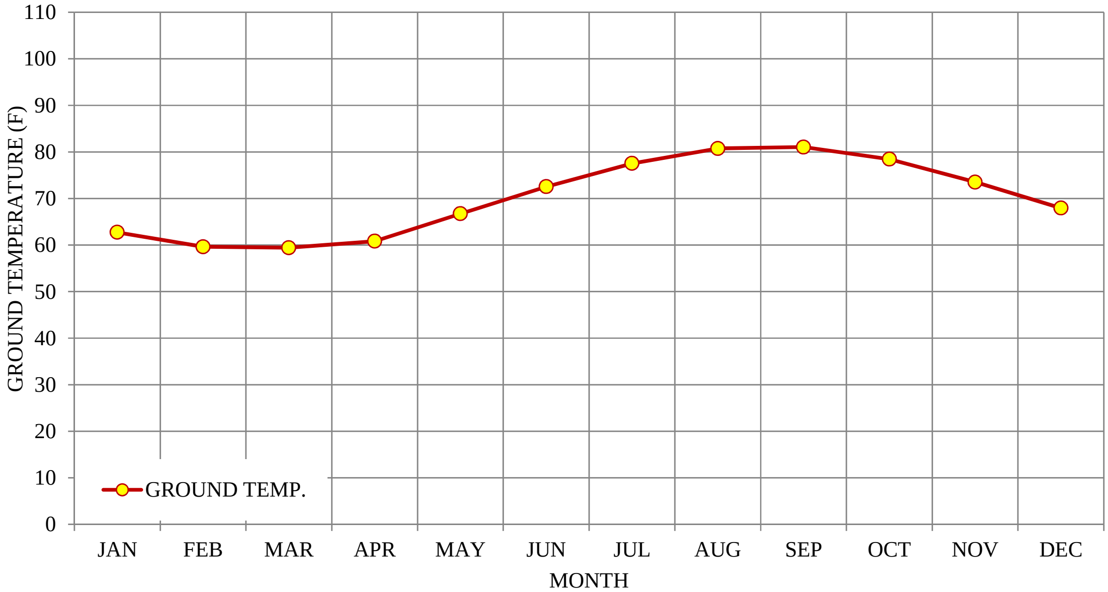


Figure A - 4: Monthly Variation in Ground Temperature for Twelve Months for College Station, TX (Source: TMY3 Weather Data for College Station, TX)

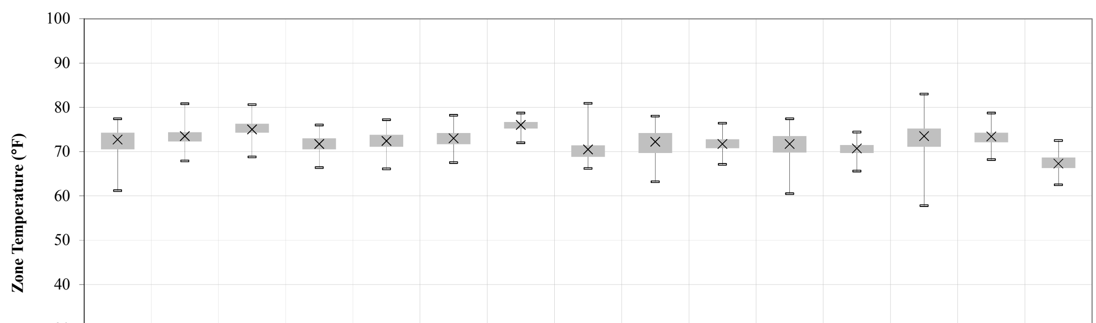
APPENDIX B

SPECIFICATIONS FOR THE BASE-CASE SIMULATION MODEL OF THE GROCERY STORE

This appendix provides the specifications of the base-case simulation model of the grocery store. Specifications include the criteria for combining thermal zones in the case-study store (on which the grocery store simulation model is based) to create thermal zones in the base-case simulation model; specifications for the various building systems in the base-case model, which include the building envelope, HVAC, lighting and refrigeration systems; a list of channels that are recorded by the on-site monitoring system installed in the case-study store; a description of the independent measurement of temperature and relative humidity conducted in the case-study store; and certain schedules in the base-case store that have been modified to accommodate for assessment of energy efficiency measures.

B – 1 Combining Thermal Zones from Case-Study Store to Create Zones in Base-Case Simulation Model

To simplify the model, it was thought best to combine zones which had similar characteristics. In order to do so, hourly zone temperature from each zone in the store were collected for the entire year. Some of these temperatures were verified with independent measurements conducted with calibrated instruments. The calibration procedure of these instruments is presented in Section B-4. These temperatures were then analyzed using box-whisker plots. The box-whisker plots presented the minimum, maximum, 25th percentile, 75th percentile and the median temperature for each zone in the store. The results are presented in Figure B-1 below. Analyzing the box-whisker plots zones with similar temperature profiles were clubbed together. As a result the simulation model was divided into 5 zones served by RTU's. The modified RTU specifications are provided in Table B-1. The grocery store also has storage and preparation spaces that require lower temperatures. These spaces include coolers, freezers and preparations rooms. The temperature of these rooms range between -25 F for freezers and 50 F for preparation rooms. The conditions of these spaces is controlled by the refrigeration system.



	RTU-01	RTU-02	RTU-03	RTU-04	RTU-05	RTU-06	RTU-07	RTU-08	RTU-09	RTU-10	RTU-11	RTU-12	RTU-13	RTU-14	RTU-15
75th Percentile	70.5	72.3	74.3	70.5	71.1	71.7	75.2	68.8	69.7	70.8	69.8	69.7	71.1	72.1	66.3
Maximum	61.2	67.9	68.8	66.4	66.1	67.5	72	66.2	63.2	67.1	60.5	65.6	57.8	68.2	62.5
Minimum	77.4	80.8	80.6	76	77.2	78.2	78.7	80.9	78	76.4	77.4	74.4	83	78.7	72.5
25th Percentile	74.3	74.4	76.3	73	73.8	74.2	76.7	71.4	74.2	72.8	73.5	71.5	75.2	74.3	68.7
Median	72.7	73.5	75	71.7	72.4	73	76	70.5	72.2	71.7	71.7	70.7	73.5	73.4	67.3

Figure B - 1: Box-whisker plot of the fifteen zones in the case-study store

Table B-1: Area per Zone and Space Conditions in the Base-Case Grocery Store Model

Thermal Zones in Base-Case Simulation Model	Roof Top Units in the Case-Study Store
General Merchandise	RTU-02
	RTU-03
	RTU-04
	RTU-05
	RTU-06
	RTU-09
	RTU-10
	RTU-14
Display Cases	RTU-01
	RTU-11
	RTU-12
	RTU-15
Bakery	RTU-07
Produce	RTU-08
Gen Load	RTU-13
Freezer	-
Cooler	-
Preparation Room	-

B – 2 Specifications of the Base-case Simulation Model

The following tables describe the specification for the grocery store simulation model.

Table B-2: Area per Zone and Space Conditions in the Base-Case Grocery Store Model

Zone Name	Floor Area (sq.ft.)	Percent of Total Area (%)	Space Temp. (F)
General Merchandise	52,086	56.0	72
Display Case	23,795	25.6	72
Loading Dock - Produce	2,452	2.6	72
Loading Dock - General	3,594	3.9	72
Bakery	3,894	4.2	72
Coolers	3,565	3.8	33.5
Freezers	2,199	2.4	-15.4
Preparation	1,366	1.5	50
TOTAL AREA	**92,952**	**100.0**	

Table B-3: Area per Person / Number of Persons per Zone in the Base-Case Grocery Store Model

Zone Name	Area per Person (sq.ft./person)	Number of Persons
General Merchandise	100 (80)	-
Display Case Area	68 (80)	-
Loading Dock - Produce	-	9
Loading Dock - General	-	9
Bakery	-	8
Coolers	-	1
Freezers	-	1
Preparation	-	16

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red.

Table B-4: Lighting Power Density per Zone in the Base-Case Grocery Store Model

Zone Name	LPD (W/sq.ft.)
General Merchandise	1.8 (1.6)
Display Case Area	1.8 (1.6)
Loading Dock - Produce	1.8
Loading Dock - General	1.8
Bakery	1.8
Coolers	0.8
Freezers	0.8
Preparation	1.8

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red.

Table B-5: Equipment Power Density per Zone in the Base-Case Grocery Store Model

Zone Name	Plug Electric (W/sq.ft.)	Plug Gas (W/sq.ft.)	Plug Electric (W)	Plug Process / Gas (Btu/hr)
General Merchandise	0.5	0	-	-
Display Case Area	0.5	0	-	-
Loading Dock - Produce	0.8 (0.5)	0	-	-
Loading Dock - General	0.8 (0.5)	0	-	-
Bakery	-	-	8,000 (3,000)	58,000
Coolers	0	0	-	28,639 (0)
Freezers	0	0	-	17,127 (0)
Preparation	0.5	0	-	36,156 (0)

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red.

Table B-6: HVAC Equipment Specifications

Zone Name	Total Supply Air	OA	Furnace HIR	Cooling EER
General Merchandise	0.5 cfm/sqft (58,100 cfm)	14.6 cfm/person (9,260 cfm)	Default (1.19)	Default (10.74)
Display Case	0.5 cfm/sqft (17,200 cfm)	14.6 cfm/person (2,320 cfm)	Default (1.19)	Default (11.0)
Loading Dock - Produce	0.5 cfm/sqft (3,500 cfm)	14.6 cfm/person (175 cfm)	Default (1.25)	Default (11.0)
Loading Dock - General	0.5 cfm/sqft (3,200 cfm)	14.6 cfm/person (320 cfm)	Default (1.25)	Default (10.8)
Bakery	0.5 cfm/sqft (7,200 cfm)	14.6 cfm/person (1,000 cfm)	-	Default (11.2)

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red.

Table B-7: Exhaust Fan Specifications

Thermal Zone	Exhaust Air Flow (CFM)	Total Static Pressure (in. WC)	Fan Power (kW/cfm)
Bakery	5,800 (5,561)	0.3 (1.2)	(0.000246)
Main	0 (1,200)	- (0.50)	(0.000155)

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red.

Table B-8: Service Hot Water Heater Specifications

Heater Type	Water Temp. (F)	Tank Volume (Gallons)	Capacity (Btu/hr)	HIR / EIR	Flow Rate (gpm)
Gas	140	119	750,000	1.28 (HIR)	2
Gas	125	40	40,000	1.28 (HIR)	0.15
(Electric)	(125)	(6)	(20,473)	(1.03 (EIR))	(0.15)
(Electric)	(125)	(6)	(20,473)	(1.03 (EIR))	(0.15)

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red.

Table B-9: Refrigeration Compressor Specifications

Comp. Rack	Compressor	Suction Temp. (F)	Power (HP)	Capacity (Btu/hr)
RACK A	3DS3-1500-TFD	22	15	125,180
	3DS3-1500-TFD	22	15	125,180
	3DA3-0750-TFD	22	7.5	79,039
	TOTAL CAPACITY @ +22 SST			**329,399**
	4DL3-1500-TSK W/ DEMAND COOLING	-22	15	51,633
	4DL3-1500-TSK W/ DEMAND COOLING	-22	15	51,633
	4DL3-1500-TSK W/ DEMAND COOLING	-22	15	51,633
	3DS3-1000-TFD W/ DEMAND COOLING	-22	10	37,402
	3DA3-0600-TFD W/ DEMAND COOLING	-22	6	23,015
	TOTAL CAPACITY @ -22 SST			**215,316**
RACK B	3DS3-1500-TFD	16	15	109,792
	3DB3-1000-TFD	16	10	82,676
	2DL3-0750-TFD	16	7.5	49,429
	TOTAL CAPACITY @ +16 SST			**241,897**
	4DL3-1500-TSK W/ DEMAND COOLING	-31	15	37,003
	4DL3-1500-TSK W/ DEMAND COOLING	-31	15	37,003
	3DS3-1000-TFD W/ DEMAND COOLING	-31	10	27,141
	2DB3-0600-TFD W/ DEMAND COOLING	-31	6	14,525
	TOTAL CAPACITY @ -31 SST			**115,672**
RACK C	3DF3-0900-TFD W/ DEMAND COOLING	-23	9	32942
	3DA3-0600-TFD W/ DEMAND COOLING	-23	6	22218
	2DF3-0300-TFD W/ DEMAND COOLING	-23	3	12679
	TOTAL CAPACITY @ -23 SST			**67839**
	3DS3-1500-TFD	18	15	114757
	3DS3-1500-TFD	18	15	114757
	3DS3-1500-TFD	18	15	114757
	3DS3-1500-TFD	18	15	114757
	3DB3-1000-TFD	18	10	86462
	TOTAL CAPACITY @ 18 SST			
RACK D	3DS3-1500-TFD	16	15	106336
	3DS3-1500-TFD	16	15	106336
	3DS3-1500-TFD	16	15	106336
	2DL3-0750-TFD	16	10	47414
	TOTAL CAPACITY @ 16 SST			**366422**
	3DB3-1000-TFD	33	10	115040
	2DD3-0500-TFD	33	5	56104
	2DC3-0500-TFD	33	5	48198
	TOTAL CAPACITY @ 33 SST			**219342**

Table B-10: Refrigeration Display-Case Specifications: Summary of Type and Size of Display-Cases

Display-Case Category	Area / Length / No. of Doors
Med. Temp. Open Vertical	632 ft.
Low Temp. w/ Doors	169 doors
Coffin	118 ft.
Service Cases	80 ft.
Walk-in Freezers	31,239 sq.ft.
Walk-in Coolers	33,805 sq.ft.
Preparation Rooms	9,175 sq.ft.

Table B-11: Refrigeration Display-Case Specifications: Details of Type and Size of Cases

Comp. Rack	Display-Case No.	Type of Rack	Length / Area
RACK A	RADA01	Multi Deck Rear Load Dairy #710R	20 ft
	RADA02	Multi Deck Dairy #710	20 ft
	RADA03	Dairy Cooler w/ Reach-in Doors	16'x66'x10'
	RADA04	Multi Deck Rear Load Dairy #710R	16 ft
	RADA05	Multi Deck Dairy #710	12 ft
	RADA06	Multi Deck Dairy #711	32 ft
	RADA07	Multi Deck Dairy #711	32 ft
	RAFF09	Reach In Glass Door Frozen Food #970	19 Doors
	RAIC10	Reach In Glass Door Ice Cream #970	22 Doors
	RAFF11	Reach In Glass Door Frozen Food #970	19 Doors
	RAIC12	Reach In Glass Door Ice Cream #970	25 Doors
	RAFF13	Reach In Glass Door Frozen Food #970	19 Doors
	RAFF14	Reach In Glass Door Frozen Food #970	22 Doors
	RAFF15	Reach In Glass Door Frozen Food #970	22 Doors
RACK B	RBDE01	4-deck deli meat	12 ft
	RBDE02	Service deli meat	28 ft
	RBFL03	Floral Cooler w/(6) reach-in doors	8x16x10
	RBDE04	Chicken Cooler	10X11x10
	RBBK05	Shared Cooler	16x18x10
	RBDE06	6-Deck Deli Meat #640	40 ft
	RBBK07 a,b	Full Service Cake Cases	13 ft
	c,d	Self Service Cake Cases	15 ft
	RBPR08	Multi Deck Produce #941	36 ft
	RBBK10	Shared Freezer	20x23x10
	RBFF11	1/2 Frozen Food Freezer	22x48x11
	RBFF12	1/2 Frozen Food Freezer	22x48x11
	RBIC13	Ice Cream Freezer	13'-6"x20'-6"x10'
RACK C	RCMK01	Dual Temperature Meat End Cap # 250	6 ft
	RCMK02	Island Frozen Meat (+) Dual Temp End Cap #250	56 ft
	RCMK03	Reach In Door Frozen Market #970	5 DOORS
	RCMK04	Reach In Door Frozen Market #970	3 DOORS
	RCMK05	Reach In Door Frozen Market #970	3 DOORS
	RCMK06	Seafood Freezer	8'X8'X10'
	RCMK07	Seafood Flaker	
	RCMK08	Reach In Door Frozen Market #970	9 DOORS
	RCMK10	Multi Deck Fresh Meat #212	36 ft
	RCDE11	Multi Deck Deli Meat #220	36 ft
	RCBW12	Multi Deck Beer & Wine #730	36 ft
	RCMK13	Multi Deck Fresh Meat #212	28 ft
	RCMK14	Island Fresh Meat #250	56 ft
	RCMK16	Multi Deck Fresh Meat #212	40 ft
	RCDE17	Multi Deck Deli Meat #220	36 ft
	RCBW18	Multi Deck Beer & Wine #730	36 ft
	RCDE19	Multi Deck Deli Meat #220	36 ft
RACK D	RDMK01	Meat Cooler	23'-6"X37'X10'
	RDMK02	Meat Holding Cooler	11'X15'X10'
	RDMK03	Seafood Cooler	7'X8'10'
	RDMK04	Service Meat Case	12 ft
	RDMK05	Multi Deck Seafood #213	8 ft
	RDMK06	Curved Glass Service Fish	20 ft
	RDPR07	Multi Deck Rear Load Produce #941	24 ft
	RDPR08	34 PHF Produce Cooler	12'X23'X10'
	RDPR09	Multi Deck Produce #941	36 ft
	RDPR10	Multi Deck Produce #941	24 ft
	RDPR11	Multi Deck Produce #941	36 ft
	RDMK13	Meat Preparation	24'X35'X10'
	RDMK14	Seafood Preparation	775 sq.ft
	RDPR15	45 Produce Cooler	18'X24'X10'

Note: Rows marked in red indicate low temperature spaces

Table B-12: Refrigeration Display-Case Specifications: Case Temperatures

Comp. Rack	Display-Case No.	Fixture Load (Btu/hr)	Suction T (F)	Evap. T (F)	Discharge T (F)
RACK A	RADA01	31,400	22	24	32
RACK A	RADA02	28,500	22	24	32
RACK A	RADA03	74,000	22	24	34
RACK A	RADA04	25,120	22	24	32
RACK A	RADA05	17,100	22	24	32
RACK A	RADA06	60,480	22	24	31
RACK A	RADA07	60,480	22	24	31
RACK A	RAFF09	24,700	-14	-11	-5
RACK A	RAIC10	30,140	-22	-19	-12
RACK A	RAFF11	24,700	-14	-11	-5
RACK A	RAIC12	34,250	-22	-19	-12
RACK A	RAFF13	24,700	-14	-11	-5
RACK A	RAFF14	28,600	-14	-11	-5
RACK A	RAFF15	28,600	-14	-11	-5
RACK B	RBDE01	14,124	19	21	27
RACK B	RBDE02	11,760	16	18	24
RACK B	RBFL03	14,000	32	34	40
RACK B	RBDE04	9,500	18	20	28
RACK B	RBBK05	22,600	22	24	34
RACK B	RBDE06	67,400	22	24	32
RACK B	RBBK07 a,b	5,850	18	20	28
RACK B	c,d	9,750	18	20	28
RACK B	RBPR08	62,280	22	24	31
RACK B	RBBK10	28,000	-28	-25	-15
RACK B	RBFF11	27,900	-28	-25	-15
RACK B	RBFF12	27,900	-28	-25	-15
RACK B	RBIC13	21,500	-31	-28	-20
RACK C	RCMK01	1,960	-23	-20	-12
RACK C	RCMK02	20,300	-23	-20	-12
RACK C	RCMK03	6,500	-14	-11	-5
RACK C	RCMK04	3,900	-14	-11	-5
RACK C	RCMK05	3,900	-14	-11	-5
RACK C	RCMK06	7,400	-23	-20	-11
RACK C	RCMK07	9,500	0	-	-
RACK C	RCMK08	11,700	-14	-11	-5
RACK C	RCMK10	64,620	19	21	29
RACK C	RCDE11	64,620	19	21	29
RACK C	RCBW12	51,300	22	24	32
RACK C	RCMK13	50,260	19	21	29
RACK C	RCMK14	18,340	19	21	26
RACK C	RCMK16	71,800	19	20	29
RACK C	RCDE17	64,620	19	21	29
RACK C	RCBW18	51,300	22	24	32
RACK C	RCDE19	64,620	19	21	29
RACK D	RDMK01	48,000	18	20	28
RACK D	RDMK02	11,500	18	20	28
RACK D	RDMK03	6,800	18	20	28
RACK D	RDMK04	10,800	16	18	28
RACK D	RDMK05	14,360	19	21	29
RACK D	RDMK06	13,000	18	20	26
RACK D	RDPR07	37,680	22	24	32
RACK D	RDPR08	15,768	22	24	34
RACK D	RDPR09	62,280	22	24	31
RACK D	RDPR10	41,520	22	24	31
RACK D	RDPR11	62,280	22	24	31
RACK D	RDMK13	65,000	33	35	50
RACK D	RDMK14	110,000	33	35	50
RACK D	RDPR15	24,700	35	37	45

Note: Rows marked in red indicate low temperature spaces

Table B-13: Refrigeration Display-Case Specifications: Defrost Specifications

Comp. Rack	Display-Case No.	Amps (208V 60Hz)	Per Day	Termination (Temp. / Time)
RACK A	RADA01	OFF-CYCLE	4	48
	RADA02	OFF-CYCLE	4	48
	RADA03	OFF-CYCLE	4	Time
	RADA04	OFF-CYCLE	4	48
	RADA05	OFF-CYCLE	4	48
	RADA06	OFF-CYCLE	6	48
	RADA07	OFF-CYCLE	6	48
	RAFF09	14@39.5/5@22.8	1	Klixon
	RAIC10	7@27.0/15@39.5	1	Klixon
	RAFF11	14@39.5/5@22.8	1	Klixon
	RAIC12	12@35.4/13@39.5	1	Klixon
	RAFF13	15@39.5/4@18	1	Klixon
	RAFF14	13@39.5/9@35.4	1	Klixon
	RAFF15	12@35.4/10@39.5(X)	1	Klixon
RACK B	RBDE01	OFF-CYCLE	4	48
	RBDE02	OFF-CYCLE	2	43
	RBFL03	OFF-CYCLE	4	Time
	RBDE04	8.7	4	Klixon
	RBBK05	OFF-CYCLE	4	Time
	RBDE06	OFF-CYCLE	4	48
	RBBK07 a,b	OFF-CYCLE	4	48
	c,d		4	48
	RBPR08	OFF-CYCLE	4	48
	RBBK10	23.2 (X)	46	Time
	RBFF11	23.2	46	Klixon
	RBFF12	23.2	46	Klixon
	RBIC13	17.4	46	Klixon
RACK C	RCMK01	5	1	48
	RCMK02	38.2	1	48
	RCMK03	22.8	1	Klixon
	RCMK04	13	1	Klixon
	RCMK05	13	1	Klixon
	RCMK06	5.8	4	Klixon
	RCMK07	OFF-TIME		
	RCMK08	35.4 (X)	1	Klixon
	RCMK10	OFF-CYCLE	4	48
	RCDE11	OFF-CYCLE	4	48
	RCBW12	OFF-CYCLE	4	48
	RCMK13	OFF-CYCLE	4	48
	RCMK14	OFF-CYCLE	1	43
	RCMK16	OFF-CYCLE	4	48
	RCDE17	OFF-CYCLE	4	48
	RCBW18	OFF-CYCLE	4	48
	RCDE19	OFF-CYCLE	4	48
RACK D	RDMK01	34.8	4	Klixon
	RDMK02	8.7	4	Klixon
	RDMK03	5.8	4	Klixon
	RDMK04	OFF-CYCLE	4	48
	RDMK05	OFF-CYCLE	4	48
	RDMK06	OFF-CYCLE	1	48
	RDPR07	OFF-CYCLE	4	48
	RDPR08	OFF-CYCLE	4	48
	RDPR09	OFF-CYCLE	4	48
	RDPR10	OFF-CYCLE	4	48
	RDPR11	OFF-CYCLE	4	48
	RDMK13	OFF-CYCLE	2	Time
	RDMK14	OFF-CYCLE	2	Time
	RDPR15	OFF-CYCLE	4	Time

Note: Rows marked in red indicate low temperature spaces

Table B-14: Refrigeration Display-Case Specifications: Lights, Fans and Anti-Sweat Heaters

Comp. Rack	Display-Case No.	Light Amp 115V	Fans Amp 115V	Fan Amp 208V	Anti-Sweat 115V
RACK A	RADA01	3.9	7		
	RADA02	3.9	3.5		
	RADA03	5.5	29.4		4.1
	RADA04	3.1	5.6		
	RADA05	2.3	2.1		
	RADA06	6.2	11.2		
	RADA07	6.2	11.2		
	RAFF09	12.7	13.3		22.2
	RAIC10	14.9	15.4		25.7
	RAFF11	12.7	13.3		22.2
	RAIC12	17.1	17.5		29.2
	RAFF13	12.7	13.3		22.2
	RAFF14	14.9	15.4		25.7
	RAFF15	14.9	15.4		25.7
RACK B	RBDE01	4.6	1.8		
	RBDE02	10.3	4.5		
	RBFL03	4.6	8		2.94
	RBDE04			3.3	
	RBBK05		8.4		
	RBDE06	17.92	14		
	RBBK07 a,b	3.6	1.4		
	c,d	2.6	1.1		
	RBPR08	6.93	12.6		
	RBBK10			8.8	
	RBFF11			8.8	
	RBFF12			8.8	
	RBIC13			6.6	
RACK C	RCMK01		3		1
	RCMK02		3.9		7.5
	RCMK03	3.3	3.5		5
	RCMK04	2.2	2.1		3.5
	RCMK05	2.2	2.1		3.5
	RCMK06			2.2	
	RCMK07			5	
	RCMK08	6.1	6.3		10.5
	RCMK10	16.2	12.6		
	RCDE11	6.93	12.6		
	RCBW12	6.93	6.3		
	RCMK13	12.6	9.8		
	RCMK14		3.6		6.5
	RCMK16	17.9	14		
	RCDE17	6.93	12.6		
	RCBW18	6.93	6.3		
	RCDE19	6.9	12.6		
RACK D	RDMK01			13.2	
	RDMK02			3.3	
	RDMK03			2.2	
	RDMK04	5.6	9.5		
	RDMK05	3.6	2.8		
	RDMK06	5.9	5.1		6.7
	RDPR07	4.6	8.4		
	RDPR08		8.4		
	RDPR09	6.93	12.6		
	RDPR10	4.62	8.4		
	RDPR11	6.93	12.6		
	RDMK13		16		
	RDMK14		17.6		
	RDPR15		8.4		

Note: Rows marked in red indicate low temperature spaces

Table B-15: Refrigeration Condenser Specifications

System No.	Des. Cond. Temp. (F)	Total Heat of Rejection (Btu/hr)	Condenser Capacity @ 1F ΔT	Actual Cond. (ΔT)	No. of Compressors	Total HP
Rack A	115	Default (828,591)	Default (96,400)	10 (8.6)	12	Default (98.5)
Rack B	115	Default (557,626)	Default (57,270)	10 (9.7)	8	Default (78.5)
Rack C	115	Default (875,589)	Default (96,400)	10 (9.1)	12	Default (88)
Rack D	120	Default (811,666)	Default (62,470)	10 (13)	8	Default (75)

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red.

Table B-16: Schedules for Occupancy in the Base-Case Grocery Store Model

Hour	Main Areas				Bakery and Deli		Preparation Rooms	
	Mon - Fri	Sat	Sun	Hol.	Mon - Sun	Hol.	Mon - Thur	Hol
1	0.01	0.01	(0.01)	0.01	0.0	0.0	0.0	0.0
2	0.01	0.01	(0.01)	0.01	0.25	0.0	0.0	0.0
3	0.01	0.01	(0.01)	0.01	0.25	0.0	0.0	0.0
4	0.01	0.01	(0.01)	0.01	0.25	0.0	0.125	0.0
5	0.01	0.01	(0.01)	0.01	0.75	0.0	0.125	0.0
6	0.05 (0.01)	0.05 (0.01)	0.05 (0.01)	0.01	0.75	0.0	0.125	0.0
7	0.1	0.1	0.1	0.01	0.75	0.0	0.25	0.0
8	0.1 (0.1)	0.1	0.1	0.01	0.75	0.0	0.25	0.0
9	0.1 (0.2)	0.1 (0.2)	0.1	0.01	0.75	0.0	0.25	0.0
10	0.2 (0.5)	0.2 (0.5)	0.1	0.01	0.75	0.0	0.375	0.0
11	0.2 (0.5)	0.3 (0.6)	0.1 (0.2)	0.01	0.75	0.0	0.375	0.0
12	0.4 (0.7)	0.4 (0.8)	0.2	0.01	0.75	0.0	0.375	0.0
13	0.4 (0.7)	0.6 (0.8)	0.5 (0.2)	0.01	0.75	0.0	0.375	0.0
14	0.25 (0.7)	0.7 (0.8)	0.5 (0.4)	0.01	0.75	0.0	0.375	0.0
15	0.25 (0.7)	0.7 (0.8)	0.5 (0.4)	0.01	0.75	0.0	0.375	0.0
16	0.5 (0.8)	0.7 (0.8)	0.5 (0.4)	0.01	0.75	0.0	0.375	0.0
17	0.5 (0.7)	0.7 (0.8)	0.5 (0.4)	0.01	0.75	0.0	0.375	0.0
18	0.5	0.7 (0.6)	0.3 (0.4)	0.01	0.75	0.0	0.375	0.0
19	0.3 (0.5)	0.6 (0.2)	0.3 (0.2)	0.01	0.75	0.0	0.375	0.0
20	0.3 (0.3)	0.4 (0.2)	0.2 (0.1)	0.01	0.75	0.0	0.25	0.0
21	0.2 (0.3)	0.4 (0.2)	0.1	0.01	0.25	0.0	0.25	0.0
22	0.1 (0.3)	0.2 (0.1)	0.1	0.01	0.25	0.0	0.25	0.0
23	0.05 (0.1)	0.1	0.05 (0.1)	0.01	0.0	0.0	0.0	0.0
24	0.05 (0.1)	0.1	0.05 (0.01)	0.01	0.0	0.0	0.0	0.0

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red. Shaded cells indicate the hours when the grocery store is open to public.

Table B-16: Continued

Hour	Produce Loading Docks			General Loading Docks			Freezers and Coolers		
	Mon - Thur	Fri - Sun	Hol	Mon - Thur	Fri - Sun	Hol	Mon - Thur	Fri - Sun	Hol
1	0.0	0.0	0.0	0.22	0.22	0.0	0	0	0.0
2	0.0	0.0	0.0	0.22	0.22	0.0	0	0	0.0
3	0.0	0.0	0.0	0.22	0.22	0.0	0	0	0.0
4	0.11	0.11	0.0	0.22	0.22	0.0	0.01	0.01	0.0
5	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
6	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
7	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
8	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
9	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
10	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
11	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
12	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
13	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
14	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
15	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
16	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
17	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
18	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
19	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
20	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
21	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
22	0.44	0.44	0.0	0.44	0.44	0.0	0.01	0.01	0.0
23	0.0	0.0	0.0	0.22	0.22	0.0	0	0	0.0
24	0.0	0.0	0.0	0.22	0.22	0.0	0	0	0.0

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red. Shaded cells indicate the hours when the grocery store is open to public.

Table B-17: Schedules for Lighting in the Base-Case Grocery Store Model

Hour	Main Areas			Bakery and Deli			Preparation Rooms		
	Mon - Thur	Fri - Sun	Hol	Mon - Thur	Fri - Sun	Hol	Mon - Thur	Fri - Sun	Hol
1	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)
2	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)
3	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.5 (0.5)	0.5 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)
4	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.5 (0.5)	0.5 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)
5	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
6	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
7	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
8	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
9	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
10	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
11	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
12	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
13	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
14	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
15	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
16	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
17	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
18	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
19	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
20	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
21	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
22	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)
23	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)
24	0.95 (1.0)	0.95 (1.0)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red. Shaded cells indicate the hours when the grocery store is open to public.

Table B-17: Continued

Hour	Produce Loading Docks			General Loading Docks			Freezers and Coolers		
	Mon - Thur	Fri - Sun	Hol	Mon - Thur	Fri - Sun	Hol	Mon - Thur	Fri - Sun	Hol
1	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
2	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
3	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
4	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
5	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
6	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
7	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
8	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
9	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
10	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
11	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
12	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
13	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
14	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
15	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
16	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
17	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
18	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
19	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
20	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
21	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
22	0.95	0.95	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
23	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95
24	0.25 (0.5)	0.25 (0.5)	0.25 (0.5)	0.95	0.95	0.25 (0.5)	0.95	0.95	0.95

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red. Shaded cells indicate the hours when the grocery store is open to public.

Table B-18: Schedules for Plug and Process Loads in the Base-Case Grocery Store Model

Hour	Main Areas			Bakery and Deli			Preparation Rooms		
	Mon - Thur	Fri - Sun	Hol	Mon - Thur	Fri - Sun	Hol	Mon - Thur	Fri - Sun	Hol
1	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15
2	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15
3	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15
4	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15
5	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15
6	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15
7	0.4	0.3 (0.4)	0.3 (0.15)	0.4	0.3 (0.4)	0.3 (0.15)	0.4	0.3 (0.4)	0.3 (0.15)
8	0.4	0.3 (0.4)	0.3 (0.15)	0.4	0.3 (0.4)	0.3 (0.15)	0.4	0.3 (0.4)	0.3 (0.15)
9	0.7	0.5 (0.7)	0.3 (0.15)	0.7	0.5 (0.7)	0.3 (0.15)	0.7	0.5 (0.7)	0.3 (0.15)
10	0.9	0.8 (0.9)	0.3 (0.15)	0.9	0.8 (0.9)	0.3 (0.15)	0.9	0.8 (0.9)	0.3 (0.15)
11	0.9	0.9	0.6 (0.15)	0.9	0.9	0.6 (0.15)	0.9	0.9	0.6 (0.15)
12	0.9	0.9	0.6 (0.15)	0.9	0.9	0.6 (0.15)	0.9	0.9	0.6 (0.15)
13	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)
14	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)
15	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)
16	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)
17	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)
18	0.9	0.9	0.6 (0.15)	0.9	0.9	0.6 (0.15)	0.9	0.9	0.6 (0.15)
19	0.8	0.7 (0.8)	0.4 (0.15)	0.8	0.7 (0.8)	0.4 (0.15)	0.8	0.7 (0.8)	0.4 (0.15)
20	0.8	0.5 (0.8)	0.4 (0.15)	0.8	0.5 (0.8)	0.4 (0.15)	0.8	0.5 (0.8)	0.4 (0.15)
21	0.7	0.5 (0.7)	0.4 (0.15)	0.7	0.5 (0.7)	0.4 (0.15)	0.7	0.5 (0.7)	0.4 (0.15)
22	0.4	0.3 (0.4)	0.4 (0.15)	0.4	0.3 (0.4)	0.4 (0.15)	0.4	0.3 (0.4)	0.4 (0.15)
23	0.2	0.3 (0.2)	0.15 (0.15)	0.2	0.3 (0.2)	0.15 (0.15)	0.2	0.3 (0.2)	0.15 (0.15)
24	0.2	0.3 (0.2)	0.15 (0.15)	0.2	0.3 (0.2)	0.15 (0.15)	0.2	0.3 (0.2)	0.15 (0.15)

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red. Shaded cells indicate the hours when the grocery store is open to public.

Table B-18: Continued

Hour	Produce Loading Docks			General Loading Docks			Freezers and Coolers		
	Mon - Thur	Fri - Sun	Hol	Mon - Thur	Fri - Sun	Hol	Mon - Thur	Fri - Sun	Hol
1	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15	0.21 (n.a)	0.15 (n.a)	0.15 (n.a)
2	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15	0.21 (n.a)	0.15 (n.a)	0.15 (n.a)
3	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15	0.21 (n.a)	0.15 (n.a)	0.15 (n.a)
4	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15	0.21 (n.a)	0.15 (n.a)	0.15 (n.a)
5	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15	0.21 (n.a)	0.15 (n.a)	0.15 (n.a)
6	0.21	0.15 (0.21)	0.15	0.21	0.15 (0.21)	0.15	0.21 (n.a)	0.15 (n.a)	0.15 (n.a)
7	0.4	0.3 (0.4)	0.3 (0.15)	0.4	0.3 (0.4)	0.3 (0.15)	0.4 (n.a)	0.3 (n.a)	0.3 (n.a)
8	0.4	0.3 (0.4)	0.3 (0.15)	0.4	0.3 (0.4)	0.3 (0.15)	0.4 (n.a)	0.3 (n.a)	0.3 (n.a)
9	0.7	0.5 (0.7)	0.3 (0.15)	0.7	0.5 (0.7)	0.3 (0.15)	0.7 (n.a)	0.5 (n.a)	0.3 (n.a)
10	0.9	0.8 (0.9)	0.3 (0.15)	0.9	0.8 (0.9)	0.3 (0.15)	0.9 (n.a)	0.8 (n.a)	0.3 (n.a)
11	0.9	0.9	0.6 (0.15)	0.9	0.9	0.6 (0.15)	0.9 (n.a)	0.9 (n.a)	0.6 (n.a)
12	0.9	0.9	0.6 (0.15)	0.9	0.9	0.6 (0.15)	0.9 (n.a)	0.9 (n.a)	0.6 (n.a)
13	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)	0.9 (n.a)	0.9 (n.a)	0.8 (n.a)
14	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)	0.9 (n.a)	0.9 (n.a)	0.8 (n.a)
15	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)	0.9 (n.a)	0.9 (n.a)	0.8 (n.a)
16	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)	0.9 (n.a)	0.9 (n.a)	0.8 (n.a)
17	0.9	0.9	0.8 (0.15)	0.9	0.9	0.8 (0.15)	0.9 (n.a)	0.9 (n.a)	0.8 (n.a)
18	0.9	0.9	0.6 (0.15)	0.9	0.9	0.6 (0.15)	0.9 (n.a)	0.9 (n.a)	0.6 (n.a)
19	0.8	0.7 (0.8)	0.4 (0.15)	0.8	0.7 (0.8)	0.4 (0.15)	0.8 (n.a)	0.7 (n.a)	0.4 (n.a)
20	0.8	0.5 (0.8)	0.4 (0.15)	0.8	0.5 (0.8)	0.4 (0.15)	0.8 (n.a)	0.5 (n.a)	0.4 (n.a)
21	0.7	0.5 (0.7)	0.4 (0.15)	0.7	0.5 (0.7)	0.4 (0.15)	0.7 (n.a)	0.5 (n.a)	0.4 (n.a)
22	0.4	0.3 (0.4)	0.4 (0.15)	0.4	0.3 (0.4)	0.4 (0.15)	0.4 (n.a)	0.3 (n.a)	0.4 (n.a)
23	0.2	0.3 (0.2)	0.15 (0.15)	0.2	0.3 (0.2)	0.15 (0.15)	0.2 (n.a)	0.3 (n.a)	0.15 (n.a)
24	0.2	0.3 (0.2)	0.15 (0.15)	0.2	0.3 (0.2)	0.15 (0.15)	0.2 (n.a)	0.3 (n.a)	0.15 (n.a)

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red. Shaded cells indicate the hours when the grocery store is open to public.

Table B-19: Schedules for Infiltration in the Base-Case Grocery Store Model

Hour	General Loading Docks			Produce Loading Docks			All Other Areas		
	Mon - Thurs	Fri - Sun	Hol	Mon - Thurs	Fri - Sun	Hol	Mon - Thurs	Fri - Sun	Hol
1	1.00 (1.25)	1.00 (1.25)	1.00 (1.25)	1.00	1.00	1.00	1.00	1.00	1.00
2	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
3	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
4	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
5	1.00 (1.25)	1.00 (1.25)	1.00 (1.25)	1.00 (1.25)	1.00 (1.25)	1.00 (1.25)	1.00	1.00	1.00
6	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
7	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
8	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
9	1.00 (1.25)	1.00 (1.25)	1.00 (1.25)	1.00	1.00	1.00	1.00	1.00	1.00
10	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
11	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
12	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
13	1.00 (1.25)	1.00 (1.25)	1.00 (1.25)	1.00	1.00	1.00	1.00	1.00	1.00
14	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
15	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
16	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
17	1.00 (1.25)	1.00 (1.25)	1.00 (1.25)	1.00	1.00	1.00	1.00	1.00	1.00
18	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
19	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
20	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
21	1.00 (1.25)	1.00 (1.25)	1.00 (1.25)	1.00	1.00	1.00	1.00	1.00	1.00
22	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
23	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
24	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00

Note: Assumptions made in the final calibrated base-case grocery store model are presented in parenthesis and are marked in red.
Shaded cells indicate the hours when the grocery store is open to public.

Table B-20: Schedules for Service Hot Water Usage in the Base-Case Grocery Store Model

Hour	Mon - Thurs	Fri - Sun	Hol
1	0.04	0.11	0.07
2	0.05	0.1	0.07
3	0.05	0.08	0.07
4	0.04	0.06	0.06
5	0.04	0.06	0.06
6	0.04	0.06	0.06
7	0.04	0.07	0.07
8	0.15	0.2 0	0.1
9	0.23	0.24	0.12
10	0.32	0.27	0.14
11	0.41	0.42	0.29
12	0.57	0.54	0.31
13	0.62	0.59	0.36
14	0.61	0.6	0.36
15	0.5	0.49	0.34
16	0.45	0.48	0.35
17	0.46	0.47	0.37
18	0.47	0.46	0.34
19	0.42	0.44	0.25
20	0.34	0.36	0.27
21	0.33	0.29	0.21
22	0.23	0.22	0.16
23	0.13	0.16	0.1
24	0.08	0.13	0.06

Table B-21: Holiday Schedule for the Case-Study Store

List of Holidays in Initial Base-Case Simulation Model	List of Holidays in Final Base-Case Simulation Model (Case-Study Store)
New Year's Day M.L. King Birthday: Third Monday in January Washington's Birthday: Third Monday in February Memorial Day: Last Monday in May Fourth of July Labor Day: First Monday in September Columbus Day: Second Monday in October Veterans Day Thanksgiving: Fourth Thursday in November Christmas: 25th December	Easter: 12th April Christmas: 25th December

Table B-22: Specifications for Low Temperature Spaces

System No.	Design Temp. (F)	Fan Power (CFM/kW)	Fan Control	Supply Flow	Capacity (Btu/hr)	Infiltration (CFM/ft^2)
Freezer	-15 (-16)	0.000169 (0.000524)	Var.Speed (Const. Vol.)	0.5 (Default)	112,700	0.07 (0.09)
Cooler	33	0.000169 (0.000538)	Var.Speed (Const. Vol.)	5,938 (Default)	226,868	0.07 (0.05)
Prep. Room	50	0.000133 (0.000382)	Var. Speed (Const. Vol.)	0.5 (Default)	175,000	0.07 (0.05)

B – 3 Channels Recorded by the On-Site Monitoring System in the Case-Study Store

The following table describes the different channels in the on-site monitoring system in the case-study store.

Table B-23: List of Channels Recorded by the On-Site Monitoring System in the Case-Study Store (HVAC Systems)

Building System	Measurement Channel
HVAC Systems RTU-01 (Washdown) RTU-02 (Photo Area) RTU-03 (Checkouts) RTU-04 (Business Center) RTU-05 (Produce) RTU-06(General Merchandise) RTU-07(Bakery/Deli) RTU-08 (Produce Dock) RTU-09 (Tortilleria) RTU-10 (Grocery) RTU-11 (Merchandise Support) RTU-12 (Merchandise Support 2) RTU-13 (Receiving) RTU-14(Frozen) RTU-15 (Market)	Dew Point (°F)
	Humidity (% rh)
	Outside AirTemperature (°F)
	Cooling Number of Stages On
	Cooling Stage 1 (0 = Off)
	Cooling Stage 2 (0 = Off)
	Fan (0 = Off)
	Fan Status Switch (On/Off)
	Mixed Air Temperature (°F)
	Outside Air Damper (Fraction)
	Heating Number of Stages On
	Heating Stage 1 (0 = Off)
	Heating Stage 2 (0 = Off)
	Pre-heating Number of Stages On
	Pre-heat Stage 1 (0 = Off)
	Return Air Temperature (°F)
	Space Dew Point (°F)
	Space Humidity (%rh)
	Space Temperature (°F)
	Supply Air Temperature (°F)

Table B-24: List of Channels Recorded by the On-Site Monitoring System in the Case-Study Store (Lighting)

Building System	Measurement Channel
Lighting Systems	Outside Light Level (Fc)
	Zone 2 North Light Level (Fc)
	Zone 3 West Light Level (Fc)
	Zone 3 South Light Level (Fc)
	Zone 3 East Light Level (Fc)
	Average Inside Light Level (Fc)
	Cases (0=Off)
	Egress (0=Off)
	Pharmacy (0=Off)
	Perimeter (0=Off)
	Sign/Canopy (0=Off)
	Security (0=Off)
	Sales Lights A (0=Off)
	Sales Lights B (0=Off)
	Outside Light Level (Fc)
	Zone 2 North Light Level (Fc)
	Zone 3 West Light Level (Fc)
	Zone 3 South Light Level (Fc)
	Zone 3 East Light Level (Fc)
	Average Inside Light Level (Fc)

Table B-25: List of Channels Recorded by the On-Site Monitoring System in the Case-Study Store (Refrigeration System)

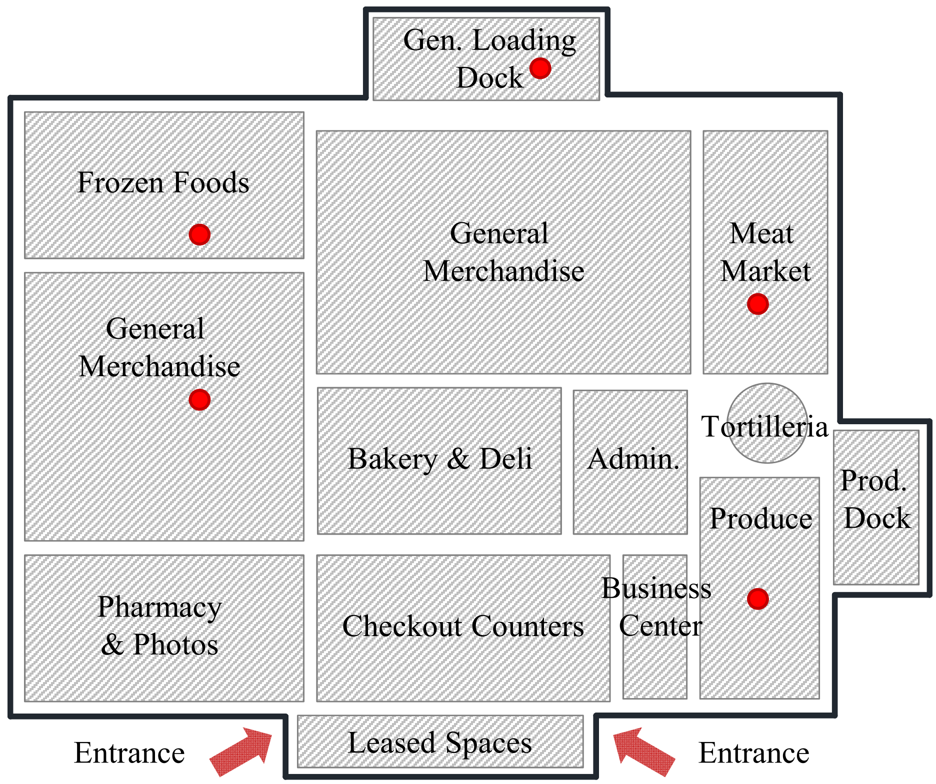
Building System	Measurement Channel
Compressor Racks : Inclusive of specifications for compressors, condensers, sub-coolers and refrigerant properties and power consumption Rack A Rack B Rack C Rack D	Outside Air Temperature (°F)
	Suction Temperature (°F)
	Setpoint for Suction Pressure (psig)
	Suction Pressure (psig)
	Discharge Temperature (°F)
	Discharge Pressure (psig)
	Setpoint for Discharge Pressure (psig)
	Compressor Overload (On/Off)
	Compressor SLA
	Refrigerant Level (%)
	Requested Power (%)
	Actual Power (kW)
	Sub-cooler Liquid Temperature (°F)
	Sub-cooler Dew Protection
	Sub-cooler Stage
	Sub-cooler Liquid Temperature Setpoint (°F)
	Condensing Temperature (°F)
	Condenser Temperature Difference (Δ°F)
	Condenser Temperature Difference Setpoint (Δ°F)
	Condenser Fan Schedule (On/Off) (For multiple fans)
Display Cases Several display cases connected to each compressor rack	Display Case Lighting Level (fc)
	Display Case Refrigeration Temperature (°F)
	Display Case EEPR Valve Opening Percentage (%)

Table B-26: List of Channels Recorded by the On-Site Monitoring System in the Case-Study Store (Whole-Building Power Consumption)

Building System	Measurement Channel
Whole-Building Power	Current Level
	Current Mode
	Current Peak (kW)
	PM Amps Phase A
	PM Amps Phase B
	PM Amps Phase C
	PM Demand (kW)
	PM Energy Phase A (kWhr)
	PM Energy Phase B (kWhr)
	PM Energy Phase C (kWhr)
	PM Energy Sum (kWhr)
	PM Frequency (hz)
	PM Frequency Phase A (hz)
	PM Frequency Phase B (hz)
	PM Frequency Phase C (hz)
	PM Peak Demand (kW)
	PM Peak Demand Day (kW)
	PM Peak Demand Hour (kW)
	PM Peak Demand Minute (kW)
	PM Peak Demand Month (kW)
	PK Peak Demand Second (kW)
	PK Peak Demand Year (kW)
	PM Power Average (kW)
	PM Power Factor Phase A
	PM Power Factor Phase B
	PM Power Factor Phase C
	PM Power Phase A (kW)
	PM Power Phase B (kW)
	PM Power Phase C (kW)
	PM Power Sum (kW)
	PM Reactive Energy Sum (kWhr)
	PM Reactive Power Phase A (kW)
	PM Reactive Power Phase B (kW)
	PM Reactive Power Sum (kW)
	PM Volts Phase A (V)
	PM Volts Phase B (V)
	PM Volts Phase C (V)
	Peak Reset ()
	kW Load Shed (kW)
	kW Reading (kW)
	~Load Shed\Count Stage ()
	Load Shed\kW (kW)
	Load Shed\LoadShed (kW)
	Load Shed\Mode ()
	Load Shed\Total kW (kW)

B – 4 Independent Measurement of Temperature and Relative Humidity in the Case-Study Store

B – 4.1 Measurement Plan



Note: Red dots indicate the position of data loggers in the case-study store.

Figure B - 2: Plan of Case-Study Store Showing Layout of Measurement Sensors

B – 4.2 Calibration Procedure

The calibration process for the HOBO data loggers has been outlined in the dissertation by S. Kim (2006). The calibrated HOBO data loggers were then used to calibrate the temperature measurements provided by the on-site monitoring system.

B – 5 Schedules for Efficiency Measures for the Grocery Store

Table B-27: Schedules for Time Clocks and Occupancy Sensors for Lighting

Hour	Main Areas		Bakery and Deli		Preparation Rooms	
	Mon - Sun	Hol	Mon - Sun	Hol	Mon - Sun	Hol
1	0.25	0.05	0.05	0.05	0.05	0.05
2	0.25	0.05	0.05	0.05	0.05	0.05
3	0.05	0.05	0.5	0.05	0.05	0.05
4	0.05	0.05	0.5	0.05	0.05	0.05
5	0.05	0.05	1.0	0.05	1.0	0.05
6	0.05	0.05	1.0	0.05	1.0	0.05
7	0.99	0.05	1.0	0.05	1.0	0.05
8	0.99	0.05	1.0	0.05	1.0	0.05
9	0.99	0.05	1.0	0.05	1.0	0.05
10	0.99	0.05	1.0	0.05	1.0	0.05
11	0.99	0.05	1.0	0.05	1.0	0.05
12	0.99	0.05	1.0	0.05	1.0	0.05
13	0.99	0.05	1.0	0.05	1.0	0.05
14	0.99	0.05	1.0	0.05	1.0	0.05
15	0.99	0.05	1.0	0.05	1.0	0.05
16	0.99	0.05	1.0	0.05	1.0	0.05
17	0.99	0.05	1.0	0.05	1.0	0.05
18	0.99	0.05	1.0	0.05	1.0	0.05
19	0.99	0.05	1.0	0.05	1.0	0.05
20	0.99	0.05	1.0	0.05	1.0	0.05
21	0.99	0.05	1.0	0.05	1.0	0.05
22	0.99	0.05	1.0	0.05	1.0	0.05
23	0.99	0.05	0.05	0.05	0.05	0.05
24	0.99	0.05	0.05	0.05	0.05	0.05

Note: Shaded cells indicate the hours when the grocery store is open to public.

Table B-27: Continued

Hour	Produce Loading Docks		General Loading Docks		Freezers and Coolers	
	Mon - Sun	Hol	Mon - Sun	Hol	Mon - Sun	Hol
1	0.05	0.05	0.9	0.05	0.25	0.05
2	0.05	0.05	0.9	0.05	0.25	0.05
3	0.05	0.05	0.9	0.05	0.05	0.05
4	0.05	0.05	0.9	0.05	0.05	0.05
5	0.9	0.05	0.9	0.05	0.05	0.05
6	0.9	0.05	0.9	0.05	0.05	0.05
7	0.9	0.05	0.9	0.05	0.5	0.05
8	0.9	0.05	0.9	0.05	0.5	0.05
9	0.9	0.05	0.9	0.05	0.5	0.05
10	0.9	0.05	0.9	0.05	0.5	0.05
11	0.9	0.05	0.9	0.05	0.5	0.05
12	0.9	0.05	0.9	0.05	0.5	0.05
13	0.9	0.05	0.9	0.05	0.5	0.05
14	0.9	0.05	0.9	0.05	0.5	0.05
15	0.9	0.05	0.9	0.05	0.5	0.05
16	0.9	0.05	0.9	0.05	0.5	0.05
17	0.9	0.05	0.9	0.05	0.5	0.05
18	0.9	0.05	0.9	0.05	0.5	0.05
19	0.9	0.05	0.9	0.05	0.5	0.05
20	0.9	0.05	0.9	0.05	0.5	0.05
21	0.9	0.05	0.9	0.05	0.5	0.05
22	0.9	0.05	0.9	0.05	0.5	0.05
23	0.05	0.05	0.9	0.05	0.5	0.05
24	0.05	0.05	0.9	0.05	0.5	0.05

Note: Shaded cells indicate the hours when the grocery store is open to public.

Table B-28: Schedules for Exterior Parking Lighting

Hour		
	Mon - Sun	Hol
1	0.25	0.25
2	0.25	0.25
3	0.25	0.25
4	0.25	0.25
5	0.25	0.25
6	0.25	0.25
7	0.0	0.0
8	0.0	0.0
9	0.0	0.0
10	0.0	0.0
11	0.0	0.0
12	0.0	0.0
13	0.0	0.0
14	0.0	0.0
15	0.0	0.0
16	0.0	0.0
17	0.0	0.0
18	0.0	0.0
19	0.0	0.0
20	1.0	1.0
21	1.0	1.0
22	1.0	1.0
23	1.0	1.0
24	1.0	1.0

Note Shaded cells indicate the hours when the grocery store is open to public.

Table B-29: Schedules for Equipment for All Zones

Hour	Main Areas	
	Mon - Sun	Hol
1	0.11	0.10
2	0.11	0.10
3	0.11	0.10
4	0.11	0.10
5	0.11	0.10
6	0.11	0.10
7	0.3	0.10
8	0.3	0.10
9	0.6	0.10
10	0.8	0.10
11	0.8	0.10
12	0.8	0.10
13	0.8	0.10
14	0.8	0.10
15	0.8	0.10
16	0.8	0.10
17	0.8	0.10
18	0.8	0.10
19	0.8	0.10
20	0.8	0.10
21	0.6	0.10
22	0.3	0.10
23	0.11	0.10
24	0.11	0.10

Note: Shaded cells indicate the hours when the grocery store is open to public.

Table B-30: Schedules for Exhaust Fans

Hour	Main Areas	
	Mon - Sun	Hol
1	0	0
2	0	0
3	0	0
4	0	0
5	1	0
6	1	0
7	1	0
8	1	0
9	1	0
10	1	0
11	1	0
12	1	0
13	1	0
14	1	0
15	1	0
16	1	0
17	1	0
18	1	0
19	1	0
20	1	0
21	0	0
22	0	0
23	0	0
24	0	0

Note: Shaded cells indicate the hours when the grocery store is open to public.

Table B-31: Schedules for Demand Controlled Ventilation in Each Zone

Hr	'General Merchandise' Zone				'Display Case' Zone				'Bakery' Zone	
	Mon - Fri	Sat	Sun	Hol.	Mon - Fri	Sat	Sun	Hol.	Mon - Sun	Hol.
1	0.0016	0.0016	0.0016	0.0016	0.0014	0.0014	0.0014	0.0014	0.0	0.0
2	0.0016	0.0016	0.0016	0.0016	0.0014	0.0014	0.0014	0.0014	0.0345	0.0
3	0.0016	0.0016	0.0016	0.0016	0.0014	0.0014	0.0014	0.0014	0.0345	0.0
4	0.0016	0.0016	0.0016	0.0016	0.0014	0.0014	0.0014	0.0014	0.0345	0.0
5	0.0016	0.0016	0.0016	0.0016	0.0014	0.0014	0.0014	0.0014	0.1035	0.0
6	0.0080	0.0080	0.0080	0.0016	0.0068	0.0068	0.0068	0.0014	0.1035	0.0
7	0.0159	0.0159	0.0159	0.0016	0.0135	0.0135	0.0135	0.0014	0.1035	0.0
8	0.0159	0.0159	0.0159	0.0016	0.0135	0.0135	0.0135	0.0014	0.1035	0.0
9	0.0159	0.0159	0.0159	0.0016	0.0135	0.0135	0.0135	0.0014	0.1035	0.0
10	0.0318	0.0318	0.0159	0.0016	0.0270	0.0270	0.0135	0.0014	0.1035	0.0
11	0.0318	0.0477	0.0159	0.0016	0.0270	0.0270	0.0135	0.0014	0.1035	0.0
12	0.0636	0.0637	0.0318	0.0016	0.0540	0.0540	0.0270	0.0014	0.1035	0.0
13	0.0636	0.0954	0.0795	0.0016	0.0540	0.0540	0.0675	0.0014	0.1035	0.0
14	0.0398	0.1113	0.0795	0.0016	0.0338	0.0338	0.0675	0.0014	0.1035	0.0
15	0.0398	0.1113	0.0795	0.0016	0.0338	0.0338	0.0675	0.0014	0.1035	0.0
16	0.0795	0.1113	0.0795	0.0016	0.0675	0.0675	0.0675	0.0014	0.1035	0.0
17	0.0795	0.1113	0.0795	0.0016	0.0675	0.0675	0.0675	0.0014	0.1035	0.0
18	0.0795	0.1113	0.0477	0.0016	0.0675	0.0675	0.0405	0.0014	0.1035	0.0
19	0.0477	0.0954	0.0477	0.0016	0.0405	0.0405	0.0405	0.0014	0.1035	0.0
20	0.0477	0.0636	0.0318	0.0016	0.0405	0.0405	0.0270	0.0014	0.1035	0.0
21	0.0318	0.0636	0.0159	0.0016	0.0270	0.0270	0.0135	0.0014	0.0345	0.0
22	0.0159	0.0318	0.0159	0.0016	0.0135	0.0135	0.0135	0.0014	0.0345	0.0
23	0.0080	0.0159	0.0080	0.0016	0.0068	0.0068	0.0068	0.0014	0.0	0.0
24	0.0080	0.0159	0.0080	0.0016	0.0068	0.0068	0.0068	0.0014	0.0	0.0

Note: Shaded cells indicate the hours when the grocery store is open to public.

Table B-31: Continued

Hr	'General Loading Dock' Zone		'Produce Loading Dock' Zone	
	Mon - Sun	Hol.	Mon- Sun	Hol.
1	0.0022	0.0	0.0	0.0
2	0.0022	0.0	0.0	0.0
3	0.0022	0.0	0.0	0.0
4	0.0022	0.0	0.0216	0.0
5	0.0440	0.0	0.0216	0.0
6	0.0440	0.0	0.0216	0.0
7	0.0440	0.0	0.0216	0.0
8	0.0440	0.0	0.0216	0.0
9	0.0440	0.0	0.0216	0.0
10	0.0440	0.0	0.0216	0.0
11	0.0440	0.0	0.0216	0.0
12	0.0440	0.0	0.0216	0.0
13	0.0440	0.0	0.0216	0.0
14	0.0440	0.0	0.0216	0.0
15	0.0440	0.0	0.0216	0.0
16	0.0440	0.0	0.0216	0.0
17	0.0440	0.0	0.0216	0.0
18	0.0440	0.0	0.0216	0.0
19	0.0440	0.0	0.0216	0.0
20	0.0440	0.0	0.0216	0.0
21	0.0440	0.0	0.0216	0.0
22	0.0440	0.0	0.0216	0.0
23	0.0220	0.0	0.0	0.0
24	0.0220	0.0	0.0	0.0

Note: Shaded cells indicate the hours when the grocery store is open to public.

APPENDIX C

EQUATIONS FOR THE

CHP MODEL, THE THERMAL STORAGE MODEL, HOURLY CALIBRATION PROCEDURES AND THE ECONOMICS MODEL

This appendix provides equations for the hourly calibration procedures, the CHP model, the thermal storage model and the economics model implemented in the study.

C – 1 The CHP Model

C – 1.1 List of Abbreviations

$\dot{m}_{load\_abs}$ = Hot water flow rate for absorption cooling (lbm/hr)

$\dot{m}_{load\_sh}$ = Hot water flow rate for space heating (lbm/hr)

$\dot{m}_{load\_shw}$ = Hot water flow rate for SHW heating (lbm/hr)

$\dot{m}_{g}$ = Mass flow rate of exhaust gas (lbm/s)

$\dot{m}_{max\_abs\_a}$ = Maximum hot water flow rate through HRSG calculated with return water temperature conditions (lbm/hr)

$\dot{m}_{max\_abs\_a}$ = Maximum hot water flow rate through HRSG calculated with feed water temperature conditions (lbm/hr)

$\dot{m}_{max\_abs}$ = Maximum hot water flow rate through the HRSG calculated as a weighted average (lbm/hr)

$\dot{m}_{extra\_sh}$ = Extra flow rate required from HRSG for space heating (lbm/hr)

$\dot{m}_{surp\_abs}$ = Surplus hot water available for space heating (lbm/hr)

$\dot{m}_{HRSG\_sh}$ = Maximum hot water flow rate through the HRSG for space heating (lbm/hr)

$\dot{m}_{max\_shw}$ = Maximum mass flow rate available for service hot water heating (lbm/hr)

$\dot{m}_{sup\_abs}$ = Hot water flow rate designated for absorption chillers (lbm/hr)

$\dot{m}_{sup\_sh}$ = Flow rate provided by the HRSG for space heating (lbm/hr)

$\dot{m}_{surp\_sh}$ = Surplus hot water flow rate available from both jacket water as well as HRSG (lbm/hr)

$\dot{m}_{sup\_shw}$ = Hot water flow rate designated for service hot water heating (lbm/hr)

$\dot{m}_{surp\_shw}$ = Surplus hot water available after SHW heating (lbm/hr)

$\dot{m}_{boiler\_abs}$ = Supplementary hot water flow rate from boiler for absorption chiller (lbm/hr)

$\dot{m}_{boiler\_sh}$ = Supplementary hot water flow rate from boiler for space heating (lbm/hr)

$\dot{m}_{boiler\_shw}$ = Hot water flow rate from boiler for SHW heating (lbm/hr)

$\dot{Q}_{load\_abs}$ = Hourly absorption cooling loads as reported from hourly report of eQUEST (MMBtu/hr)

$\dot{Q}_{load\_sh}$ = Hourly space heating load as reported from hourly report of eQUEST (MMBtu/hr)

$\dot{Q}_{load\_shw}$ = Hourly service hot water load as reported from hourly report of eQUEST (MMBtu/hr)

$\dot{Q}_{max\_abs}$ = Maximum thermal energy available for absorption chiller (MMBtu/hr)

$\dot{Q}_{cool\_abs}$ = Maximum cooling energy available from absorption chiller (MMBtu/hr)

$\dot{Q}_{sup\_abs}$ = Recoverable heat for absorption refrigeration (MMBtu/hr)

$\dot{Q}_{surp\_abs}$ = = Surplus thermal energy available from exhaust gases (MMBtu/hr)

$\dot{Q}_{bur\_abs}$ = = Supplementary thermal energy from natural gas burner (MMBtu/hr)

$\dot{Q}_{abs\_boiler}$ = Total thermal energy available for absorption chiller from hot water boiler (Btu/hr)

$\dot{Q}_{jwc}$ = Heat rejection to jacket water and oil cooler from the IC engine (MMBtu/hr)

$\dot{Q}_{max\_jw}$ = Maximum thermal energy available from the jacket water (MMBtu/hr)

$\dot{Q}_{sup\_jw}$ = Thermal energy available from the jacket water coolant (MMBtu/hr)

$\dot{Q}_{surp\_jw}$ = Surplus thermal energy available from jacket water coolant (MMBtu/hr)

$\dot{Q}_{extra\_sh}$ = Supplemental hot water flow required for space heating (MMBtu/hr)

$\dot{Q}_{HRSG\_sh}$ = Thermal energy available from the HRSG for space heating (MMBtu/hr)

$\dot{Q}_{sup\_sh}$ = Total thermal energy available for space heating from HRSG and JW (MMBtu/hr)

$\dot{Q}_{surp\_sh}$ = Surplus thermal energy available from HRSG and jacket water coolant (MMBtu/hr)

$\dot{Q}_{sh\_boiler}$ = Total thermal energy available for space heating from hot water boiler (Btu/hr)

$\dot{Q}_{sup\_shw}$ = Total thermal energy available for service hot water heating (Btu/hr)

$\dot{Q}_{shw\_boiler}$ = Total thermal energy available for SHW heating from hot water boiler (Btu/hr)

h_{sup_hw} = Enthalpy of supply hot water at 30 psia and 180°F (Btu/lbm)

h_{ret_hw} = Enthalpy of return hot water at 30 psia and 140°F (Btu/lbm)

h_{sup_shw} = Enthalpy of supply service hot water at 30 psia and 120°F (Btu/lbm)

h_{fw} = Enthalpy of feed water[1] from water mains at 30 psia (Btu/lbm)

COP = Coefficient of Performance of the absorption chiller[2]

C_{pg} = Specific heat of exhaust gas (Btu/lb°F)

ϵ = Effectiveness of the HRSG

T_{gin} = Entering temperature of exhaust gas (°F)

T_{gout} = Leaving temperature of exhaust gas (°F)

T_{PP}= Pinch Point temperature of the exhaust gases (°F)

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[1] Temperature of the water mains is assumed to be same as the ground temperature and is provided in Appendix A of this study.

[2] The COP of both the absorption refrigeration systems is initially established at rated conditions. The COP is subsequently modified on an hourly basis as a function of chiller capacity. Calculating the COP of the LiBr/Water and Water/NH_3 absorption chillers are is provided in Appendix F.

T_{abs_out} = Temperature of the hot water at exit conditions of the absorption chiller (°F)

ΔT_{PP} = Temperature difference between the pinch-point temperature and the temperature of hot water at exit conditions

$\dot{V}_{pump}$ = Volumetric flow rate of fluid through pump (GPM)

$\dot{W}_{pump}$ = Pump power (kW)

η_{pump} = Pump efficiency

C – 1.2 Calculating Loads for the CHP Model

C – 1.2.1 Calculating Loads for Hot Water-Fired Absorption Chillers

- Hourly loads for the absorption chillers are obtained from the eQUEST simulation model.
- Corresponding hot water flow rate is calculated using these hourly loads. Resultant hot water flow demand of the absorption chiller is provided by the equation:

$$\dot{m}_{load\_abs} = \frac{\dot{Q}_{load\_abs}}{COP \times \left[0.9(h_{sh\_sup} - h_{sh\_ret}) + 0.1(h_{sh\_sup} - h_{fw})\right]}$$

- The operation of absorption chillers requires some amount of electricity consumption. Calculations for electricity consumption of absorption chillers are provided in Appendix F of this study.

C – 1.2.2 Calculating Loads and Energy Consumption for Space Heating

- Hot water flow rate for space heating

$$\dot{m}_{load\_sh} = \frac{\dot{Q}_{load\_sh}}{\left[0.9(h_{sh\_sup} - h_{sh\_ret}) + 0.1(h_{sh\_sup} - h_{fw})\right]}$$

C – 1.2.3 Calculating Loads and Energy Consumption for Service Hot Water Heating

- Hot Water Flow Rate for Service Hot Water Heating

$$\dot{m}_{loads\_shw} = \frac{\dot{Q}_{loads\_shw}}{(h_{shw\_sup} - h_{fw})}$$

C – 1.3 Energy Calculations for Absorption Chiller, Space Heating, Service Water Heating and Boiler Requirements

C – 1.3.1 Calculations for Hot Water-Fired Absorption Chiller

- The thermal energy required to run the hot water-fired absorption chiller is provided primarily by the HRSG and supplemented by the hot water boiler.
- Maximum hot water flow rate from the HRSG at conditions to operate the absorption chiller:

$$3600\, \dot{m}_g C_{pg}\left(T_{gin} - T_{gout}\right) = \dot{m}_{max\_abs}\,(h_{sup\_hw} - h_{ret\_hw})$$

$$\dot{m}_{max\_abs\_a} = \frac{3600\, \dot{m}_g C_{pg}\left(T_{gin} - T_{gout}\right)}{(h_{sup\_hw} - h_{ret\_hw})}$$

Also,

$$3600\, \dot{m}_g C_{pg}\left(T_{gin} - T_{gout}\right) = \dot{m}_{max\_abs}\,(h_{sup\_hw} - h_{fw})$$

$$\dot{m}_{max\_abs\_b} = \frac{3600\, \dot{m}_g C_{pg}\left(T_{gin} - T_{gout}\right)}{(h_{abs\_out} - h_{fw})}$$

Hence

$$\dot{m}_{max\_abs} = 0.9\, \dot{m}_{max\_abs\_a} + 0.1\, \dot{m}_{max\_abs\_b}$$

- Checking for pinch-point conditions:

$$\dot{m}_g C_{pg}\left(T_{PP} - T_{gout}\right) = \dot{m}_{max\_abs}\,(h_{abs\_out} - h_{abs\_in})$$

$$\Delta T_{PP} = T_{PP} - T_{abs\_out}$$

The delta pinch point temperature should satisfy a minimum temperature difference of 50F for the steam flow rate through the HRSG to be at maximum value.

- Effectiveness of the HRSG:

$$\epsilon = \frac{T_{gin} - T_{PP}}{T_{gin} - T_{abs\_out}}$$

- Actual hot water flow rate obtained from the HRSG:

IF	$\dot{m}_{load\_abs} > \dot{m}_{max\_abs}$
THEN	$\dot{m}_{sup\_abs} = \dot{m}_{\max\_abs}$
ELSE	$\dot{m}_{sup\_abs} = \dot{m}_{abs\_load}$
ENDIF	

- Surplus flow rates from the HRSG:

IF $\dot{m}_{max\_abs} > \dot{m}_{loads\_abs}$

THEN $\dot{m}_{surp\_abs} = \dot{m}_{max\_abs} - \dot{m}_{loads\_abs}$

ELSE $\dot{m}_{surp\_abs} = 0$

ENDIF

- Recoverable thermal energy:

$$\dot{Q}_{sup\_abs} = \dot{m}_{sup\_abs}\,(h_{sup\_hw} - h_{ret\_hw})$$

- Supplementary flow rates through the hot water boiler are calculated by:

IF $\dot{m}_{max\_abs} < \dot{m}_{loads\_abs}$

THEN $\dot{m}_{boiler\_abs} = \dot{m}_{loads\_abs} - \dot{m}_{max\_abs}$

ELSE $\dot{m}_{boiler\_abs} = 0$

ENDIF

C – 1.3.2 Calculations for Direct-Fired Absorption Chiller

Maximum thermal energy available for direct-fired absorption chillers:

$$\dot{Q}_{max\_abs} = 3600\,\dot{m}_g C_{pg}\left(T_{gin} - T_{gout}\right)$$

Maximum cooling energy available for direct-fired absorption chillers:

$$\dot{Q}_{cool\_abs} = \dot{Q}_{max\_abs} \times \mathrm{COP}$$

Actual thermal energy available for direct-fired absorption chillers:

IF $\dot{Q}_{cool\_abs} \geq \dot{Q}_{load\_abs}$

THEN $\dot{Q}_{sup\_abs} = \dfrac{\dot{Q}_{load\_abs}}{COP}$

ELSE $\dot{Q}_{sup\_abs} = \dfrac{\dot{Q}_{cool\_abs}}{COP}$

ENDIF

Actual cooling energy available for direct-fired absorption chillers:

$$\dot{Q}_{cool\_sup\_abs} = \dot{Q}_{sup\_abs} \times \mathrm{COP}$$

Surplus thermal energy available from exhaust gas:

$$\dot{Q}_{surp\_abs} = \dot{Q}_{max\_abs} - \dot{Q}_{sup\_abs}$$

Supplementary thermal energy from natural gas burner:

IF $\dot{Q}_{cool\_abs} < \dot{Q}_{load\_abs}$

THEN $\dot{Q}_{bur\_abs} = \frac{\dot{Q}_{load\_abs}}{COP} - \dot{Q}_{max\_abs}$

ELSE $\dot{Q}_{bur\_abs} = 0$

ENDIF

C – 1.3.3 Calculations for Space Heating

- Primary source for space heating equipment is provided by thermal energy available from jacket water coolant and oil cooler of the IC engine. Space heating requirements are supplemented by thermal energy available from HRSG.
- Maximum thermal energy for space heating available from jacket water coolant:

$$\dot{Q}_{max\_jw} = \dot{Q}_{jwc}$$

- Thermal energy available from jacket water coolant:

IF $\dot{Q}_{max\_jw} > \dot{Q}_{load\_sh}$

THEN $\dot{Q}_{sup\_jw} = \dot{Q}_{load\_sh}$

ELSE $\dot{Q}_{sup\_jw} = \dot{Q}_{max\_jw}$

ENDIF

- Surplus hot water flow rate from jacket water coolant:

IF $\dot{Q}_{max\_jw} > \dot{Q}_{load\_sh}$

THEN $\dot{Q}_{surp\_jw} = \dot{Q}_{max\_jw} - \dot{Q}_{load\_sh}$

ELSE $\dot{Q}_{surp\_jw} = 0$

ENDIF

- Surplus hot water flow rate required for space heating:

IF $\dot{Q}_{sup\_jw} < \dot{Q}_{load\_sh}$

THEN $\dot{Q}_{extra\_sh} = \dot{Q}_{load\_sh} - \dot{Q}_{sup\_jw}$

ELSE $\dot{Q}_{extra\_sh} = 0$

ENDIF

- Actual hot water flow rate from HRSG to meet space heating requirements:

IF $\dot{m}_{extra\_sh} > \dot{m}_{surp\_abs}$

THEN $\dot{m}_{HRSG\_sh} = \dot{m}_{surp\_abs}$

ELSE $\dot{m}_{HRSG\_sh} = \dot{m}_{extra\_sh}$

ENDIF

- Recoverable thermal energy from HRSG:

$$\dot{Q}_{HRSG\_sh} = \dot{m}_{HRSG\_sh} \left(0.9(h_{sup\_hw} - h_{ret\_hw}) + 0.1(h_{sup\_hw} - h_{fw})\right)$$

- Total thermal energy available for space heating from HRSG and jacket water coolant:

$$\dot{Q}_{sup\_sh} = \dot{Q}_{supply\_jw} + \dot{Q}_{sh\_HRSG}$$

- Total hot water flow rate for space heating from HRSG and jacket water coolant:

$$\dot{m}_{sup\_sh} = \frac{\dot{Q}_{sup\_sh}}{\left(0.9(h_{sup\_hw} - h_{ret\_hw}) + 0.1(h_{sup\_hw} - h_{fw})\right)}$$

- Surplus thermal energy from HRSG and jacket water coolant:

$$\dot{Q}_{surp\_sh} = \dot{Q}_{surp\_jw} + \dot{Q}_{surp\_HRSG}$$

- Surplus hot water flow rate available from both jacket water as well as HRSG:

$$\dot{m}_{surp\_sh} = \frac{\dot{Q}_{surp\_sh}}{\left(0.9(h_{sup\_hw} - h_{ret\_hw}) + 0.1(h_{sup\_hw} - h_{fw})\right)}$$

- Extra thermal energy for space heating from boiler is calculated by:

IF $\dot{Q}_{sup\_sh} < \dot{Q}_{load\_sh}$

THEN $\dot{Q}_{boiler\_sh} = \dot{Q}_{load\_sh} - \dot{Q}_{sup\_sh}$

ELSE $\dot{Q}_{boiler\_sh} = 0$

ENDIF

- Hot water flow rate for space heating from boiler:

$$\dot{m}_{boiler\_sh} = \frac{\dot{Q}_{sh\_boiler}}{\left(0.9(h_{sup\_hw} - h_{ret\_hw}) + 0.1(h_{sup\_hw} - h_{fw})\right)}$$

C – 1.3.4 Calculations for SHW Heating

- Maximum thermal energy available for SHW heating from surplus hot water flow rate available from space heating:

$$\dot{m}_{max\_shw} = \frac{\dot{m}_{surp\_\_HRSG\_sh} \left(0.9(h_{sup\_hw} - h_{ret\_hw}) + 0.1(h_{sup\_hw} - h_{fw})\right)}{(h_{sup\_shw} - h_{fw})}$$

- Actual mass flow rate required for SHW heating:

IF $\dot{m}_{load\_shw} > \dot{m}_{max\_shw}$

THEN $\dot{m}_{sup\_shw} = \dot{m}_{max\_shw}$

ELSE $\dot{m}_{sup\_shw} = \dot{m}_{load\_shw}$

ENDIF

- Surplus mass flow rate after SHW heating is satisfied:

IF $\dot{m}_{max\_shw} > \dot{m}_{load\_shw}$

THEN $\dot{m}_{surp\_shw} = \dot{m}_{max\_shw} - \dot{m}_{load\_shw}$

ELSE $\dot{m}_{surp\_shw} = 0$

ENDIF

- Thermal energy for SHW heating provided by surplus generated from space heating:

$$\dot{Q}_{sup\_shw} = \dot{m}_{sup\_shw}\left(h_{sup\_shw} - h_{fw}\right)$$

- Supplemental thermal energy for SHW heating from boiler:

IF $\dot{Q}_{sup\_shw} < \dot{Q}_{load\_shw}$

THEN $\dot{Q}_{shw\_boiler} = \dot{Q}_{load\_shw} - \dot{Q}_{sup\_shw}$

ELSE $\dot{Q}_{shw\_boiler} = 0$

ENDIF

- Hot water flow rate for space heating from boiler:

$$\dot{m}_{shw\_boiler} = \frac{\dot{Q}_{shw\_boiler}}{\left(h_{sup\_shw} - h_{fw}\right)}$$

- Surplus hot water flow rate available after meeting SHW loads:

IF $\dot{m}_{surp\_\_sh\_HRSG} > 0$

THEN $$\dot{m}_{surp\_shw} = \dot{m}_{surp\_\_sh\_HRSG} - \frac{\dot{m}_{sup\_shw}\left(h_{sup\_shw} - h_{fw}\right)}{\left(0.9\left(h_{sup\_hw} - h_{ret\_hw}\right) + 0.1\left(h_{sup\_hw} - h_{fw}\right)\right)}$$

ELSE $\dot{m}_{surp\_shw} = 0$

ENDIF

C – 1.3.5 Calculations for Auxiliary Boilers

- 95% Efficiency.
- Part-load curve utilized for the auxiliary boiler (Winkelmann et al. 1993) is provided by a quadratic curve in the form of:

$$F = a + bx + cx^2$$

Where:

F = Part load ratio for the heat input ratio (HIR) of the hot water boiler,

a = 0.082597,

b = 0.996764, and

c = – 0.079361, and

$x =$ Part load on boiler.

The curve is normalized to 1 at full load.

C – 1.3.6 Calculations for Hot Water, Chilled Water and Circulation Pumps

$$\dot{W}_{pump} = \frac{\dot{V}_{pump}\, H}{3960\, \eta_{pump}} \times 0.746$$

C – 2 Thermal Storage Model

C – 2.1 List of Abbreviations

Calculations for thermal storage model:

Parameters used in the code for thermal storage model:

- ✓ Hourly Demand Satisfied: '**HDS**'
- ✓ Amount To Storage Tank: '**TST**'
- ✓ Current Storage Tank Capacity: '**CSTC**'
- ✓ Storage Tank Capacity for Previous Hour: '**CSTC'**'
- ✓ Storage Tank Capacity for Previous 24th Hour: '**CSTC''**'
- ✓ From Natural Gas Burner '**NG**'
- ✓ Amount of Heat Rejected '**HR**'
- ✓ Maximum Storage Capacity '**MXST**'
- ✓ Losses to Ambient '**LA**'

Hourly inputs:

- ✓ Hourly Surplus Thermal: '**HST**'
- ✓ Hourly Demand: '**HD**'

Calculations for losses to the ambient:

- ✓ Q = Loss to the ambient (Btu/hr),
- ✓ T_i = Hot water temperature in the tank (°F),
- ✓ T_o = Ambient temperature (°F),
- ✓ UA = Heat transfer conductance (Btu/hr °F),
- ✓ r_1 = Radius of the internal surface of the tank (ft),
- ✓ r_2= Radius of the external surface of the tank (ft),
- ✓ r_3= Radium of the external surface of the insulation (ft),
- ✓ L = Length / height of the tank (ft),
- ✓ h_{co}= Inside heat transfer coefficient (Btu/hr ft °F),
- ✓ h_{ro} = Outside radiation coefficient (Btu/hr ft °F),
- ✓ k_A = Conductivity of tank wall (Btu/hr ft °F),
- ✓ k_B = Conductivity of tank insulation (Btu/hr ft °F).

C – 2.1 Excel 2010 Code for the Thermal Storage Model

Hourly Demand Satisfied ‘**HDS**’:

IF ‘**HST**’ ≤ ‘**HD**’

THEN ‘**HDS**’ *EQS* MIN[(‘**HST**’ + ‘**CSTC**`’), ‘**HD**’]

ELSE ‘**HDS**’*EQS* ‘**HD**’

ENDIF

Amount To Storage Tank ‘**TST**’:

IF ‘**HST**’ ≤ ‘**HD**’

THEN ‘**TST**’ *EQS* 0

ELSE ‘**TST**’ *EQS* ‘**HST**’ – ‘**HD**’

ENDIF

Current Storage Tank Capacity ‘**CSTC**’:

IF ‘**TST**’ > 0

THEN ‘**CSTC**’ *EQS* MIN{[‘**TST**’+ SUM(‘**CSTC** `’: ‘**CSTC** ``’)], ‘**MXST’**} – ‘**LA**’

ELSE ‘**CSTC**’ *EQS* MIN {[SUM(‘**CSTC** `’: ‘**CSTC** ``’) – (‘HD’ – ‘HST’)], ‘**MXST’**}

– ‘**LA**’

ENDIF

From Natural Gas Burner ‘**NG**’:

‘**NG**’ EQS ‘**HD**’ – ‘**HST**’

Amount of Heat Rejected ‘**HR**’:

IF ‘**CSTC**’ *EQS* ‘**MXST’**

THEN

IF ‘**HST**’ – ‘**HD**’ >0

THEN ‘**HR**’ *EQS* ‘**HST**’ – ‘**HD**’

ELSE ‘**HR**’ *EQS* 0

ELSE ‘**HR**’ *EQS* 0

ENDIF

C – 2.2 Calculating Losses to the Ambient [3]

C – 2.2.1 Equation for Calculating Losses to the Ambient

- Calculating losses to the ambient:

$$Q = UA\,(T_i - T_o)$$

- Calculating the 'UA':

$$\frac{1}{UA} = \frac{1}{2\pi r_1 L h_{ci}} + \frac{\ln\left(r_2/r_1\right)}{2\pi k_A L} + \frac{\ln\left(r_3/r_2\right)}{2\pi k_B L} + \frac{1}{2\pi r_3 L (h_{co} - h_{ro})}$$

[3] This section describes the procedure adopted by the thermal storage model to calculate the losses to the ambient which is referenced as 'LA' in the Section 2.1.

C – 3 The Economics Model

C – 3.1 List of Abbreviations

$Net\ Initial\ Investment$ = Annual capital and operation costs ($)

$Net\ Average\ Annual\ Revenue$ =Difference between annual benefits and annual costs ($)

$Net\ Revenue\ (t)$ = Revenue for the year under consideration ($)

$Investment$ = Initial investment ($)

t = Year under consideration

N = Total number of years

r = Net revenue for the year t ($)

r^* = Internal rate of return

C – 3.2 Calculations for the Financial Parameters

- Simple Payback :

$$Simple\ Payback\ = \frac{Net\ Initial\ Investment}{Net\ Average\ Annual\ Revenue}$$

- Investors Rate of Return (IRR):

$$Investors\ Rate\ of\ Return = \frac{Net\ Average\ Annual\ Revenue}{Net\ Initial\ Investment}$$

- Net Present Value (NPV):

$$Net\ Present\ Value = \sum_{t=1}^{t=N} \frac{Net\ Revenue\ (t)}{(1+r)^t} - Investment$$

- Internal Rate of Return (IRR):

$$Investment\ = \sum_{t=1}^{t=N} \frac{Net\ Revenue\ (t)}{(1+r^*)^t}$$

Computation of the IRR is available as a function in Excel 2010 and has been used for this analysis.

C – 4 Statistical Calibration Indices

C – 4.1 List of Abbreviations for the Statistical Calibration Indices

$Y_{pred,i}$ = Predicted monthly value for the building energy use

$Y_{data,i}$ = Measured monthly value for the building energy use

$y_{pred,i}$ = Predicted hourly value for the building energy use

$y_{data,i}$ = Measured hourly value for the building energy use

N = Number of months used in the simulation

n = Number of hours being considered for the analysis

Residual = Deviation between the simulated results and measurements

C – 4.2 Equation for Calculating the Statistical Calibration Indices

C-4.2.1 Equation for Calculating the Percent Difference

$$Percent\ Difference = \frac{\sum_{i=1}^{N} Y_{pred,i} - \sum_{i=1}^{N} Y_{data,i}}{\sum_{i=1}^{N} Y_{data,i}}$$

C-4.2.2 Equation for Calculating the Root Mean Square Error (RMSE)

$$RMSE = \sqrt{\sum_{i=1}^{n} \frac{Residual_i^2}{n-2}}$$

C-4.2.3 Equation for Calculating the Mean Bias Error (MBE)

$$MBE = \frac{\sum_{i=1}^{n} (y_{pred,i} - y_{data,i})}{n}$$

C-4.2.4 Equation for Calculating the Coefficient of Variation Root Mean Square Error CV(RMSE)

$$CVRMSE = \frac{\sqrt{\sum_{i=1}^{n} \frac{Residual_i^2}{n-2}}}{\bar{y}_{data,i}}$$

Where:

$$\bar{y}_{data,i} = \sum_{i=1}^{n} \frac{y_i}{n}$$

C – 5 Graphical Calibration Indices

<u>C – 5.1 Equation for Calculating the Graphical Calibration Indices</u>

C-5.1.1 Equation for Calculating the Characteristic Signature

$$Characteristic\ Signature = \frac{Change\ in\ Energy\ Consumption}{Maximum\ Energy\ Consumption} \times 100\%$$

C-5.1.2 Equation for Calculating the Calibration Signature

$$Calibration\ Signature = \frac{-Residual}{Maximum\ Energy\ Consumption} \times 100\%$$

Where:

$$Residual = Simulated\ Consumption - Measured\ Consumption$$

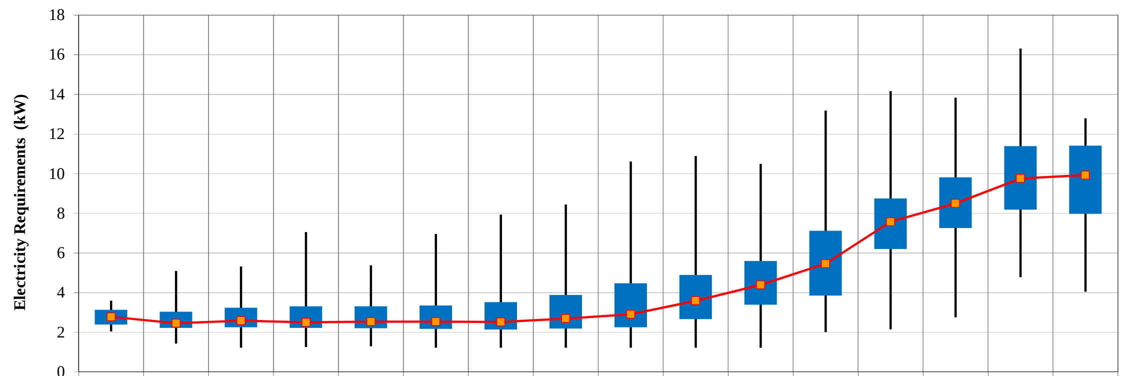
APPENDIX D

TEMPERATURE-BIN AND HOURLY LOAD PROFILES FOR THE MULTI-FAMILY MODEL

This appendix provides the temperature-bin and hourly load profiles for the multi-family buildings using the multi-family model (MF model). Surplus energy available from the grocery store on implementing the four CHP options is also presented. The electricity and thermal energy consumption of a single 8-unit multifamily building is presented in the Section D-1 of this appendix. In the next four sections of this appendix, the electricity and thermal energy requirements of multiple multi-family buildings is presented, which potentially can absorb the surplus energy available from the grocery store. The surplus energy available from the grocery store is superimposed in the temperature-bin and hourly load profiles of the multiple multi-family buildings.

D – 1 Load Profiles for a Base-Case 8-Unit Multi-Family Building

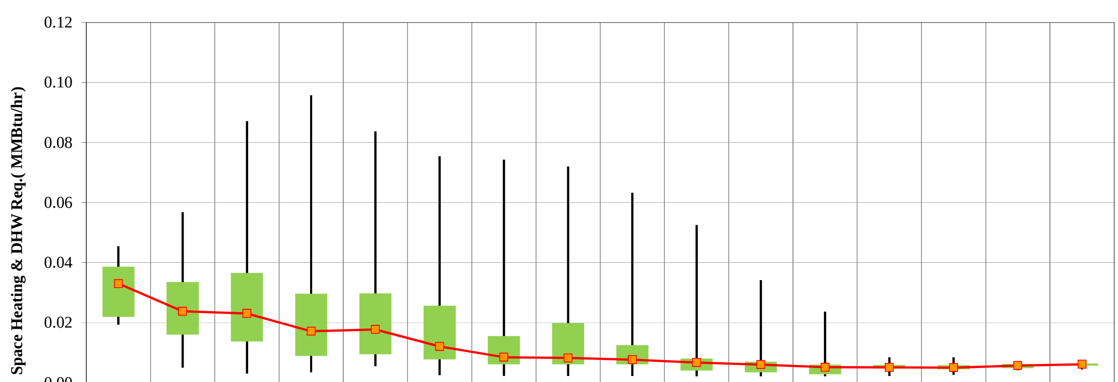
a) ELECTRICITY REQUIREMENTS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
25%	2.4	2.2	2.3	2.2	2.2	2.2	2.1	2.2	2.3	2.7	3.4	3.9	6.2	7.3	8.2	8.0
Min	2.0	1.4	1.2	1.3	1.3	1.2	1.2	1.2	1.2	1.2	1.2	2.0	2.2	2.8	4.8	4.1
Max	3.6	5.1	5.3	7.1	5.4	7.0	7.9	8.4	10.6	10.9	10.5	13.2	14.2	13.8	16.3	12.8
75%	3.1	3.0	3.2	3.3	3.3	3.4	3.5	3.9	4.5	4.9	5.6	7.1	8.8	9.8	11.4	11.4
Median	2.8	2.5	2.6	2.5	2.5	2.5	2.5	2.7	2.9	3.6	4.4	5.5	7.6	8.5	9.8	9.9

Dry Bulb Temperature Bins (F)

b) SPACE HEATING AND DHW LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
25%	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Min	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Max	0.0	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.0	0.0	0.0	0.0	0.0	0.0
75%	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Median	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Dry Bulb Temperature Bins (F)

Figure D - 1: Temperature Bin Distribution of Electricity and Thermal Energy Requirements for the Base-Case 8-Unit Multi-Family Building

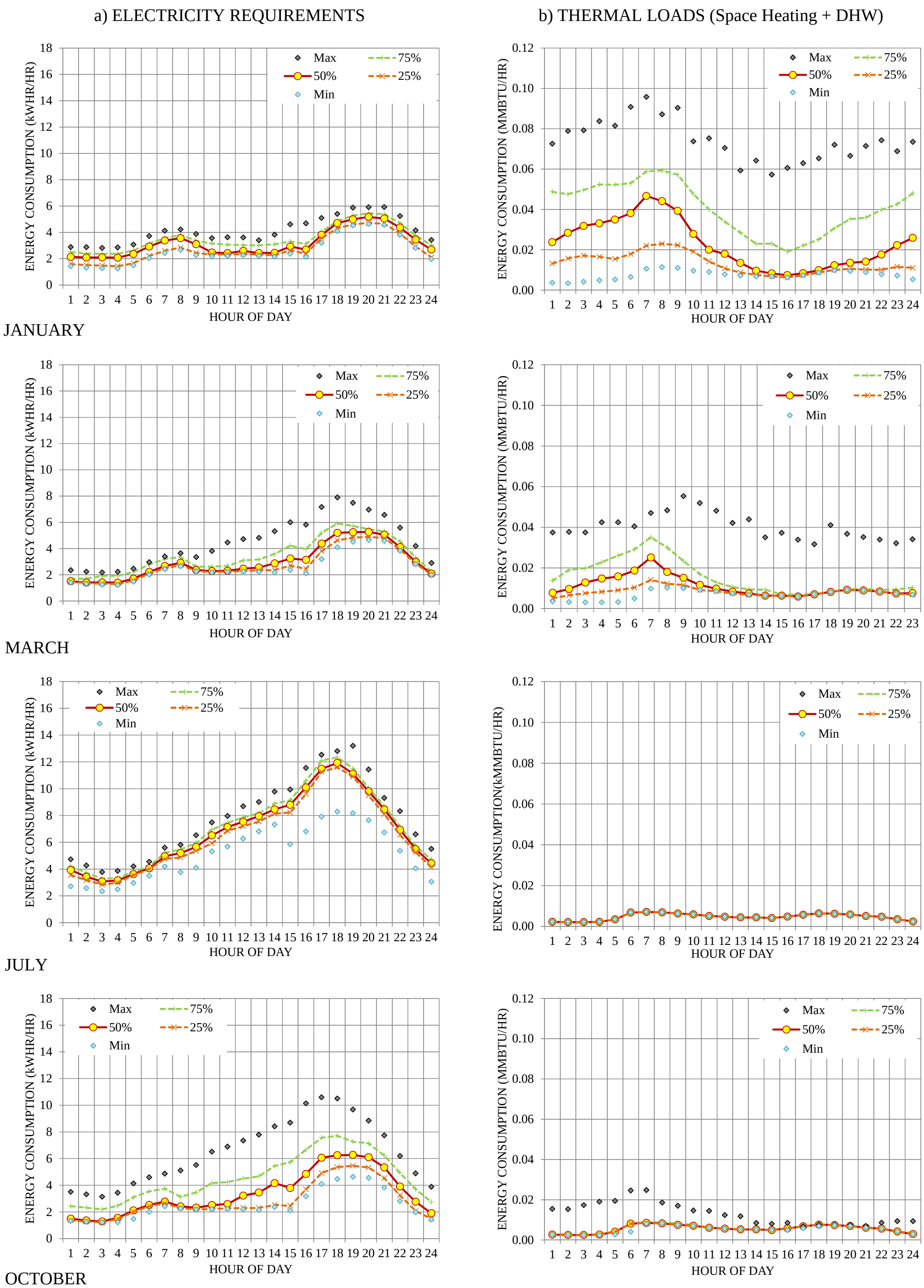
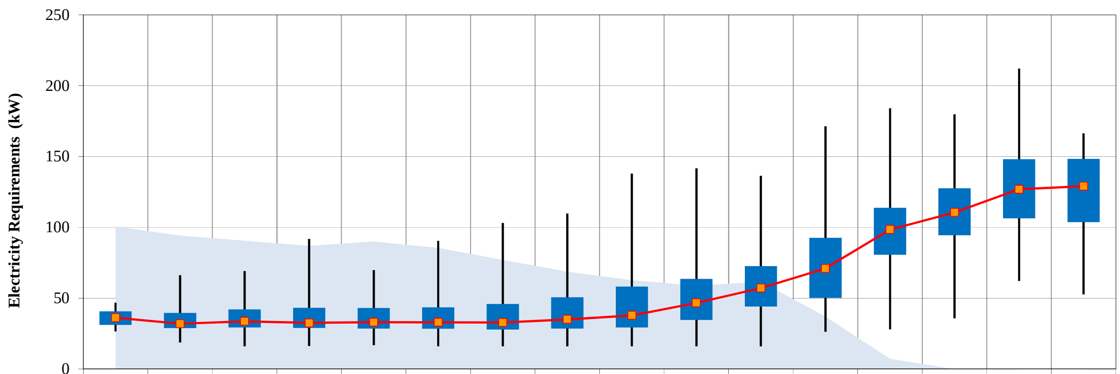


Figure D - 2: Typical Hourly Profiles for the Base-Case 8-Unit Multi-Family Building

D – 2 Load Profiles for Option 1

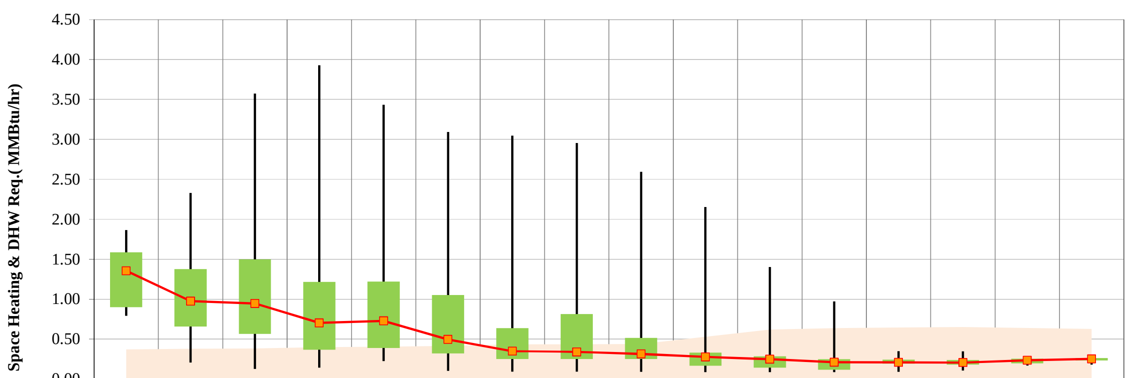
a) ELECTRICITY REQUIREMENTS (13 MF Buildings)



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
Met by CHP	100.7	94.1	90.6	86.9	90.0	85.5	77.1	68.7	62.7	58.9	61.6	37.0	7.2	0.0	0.0	0.0
25%	31.1	28.9	29.3	29.0	28.6	28.3	27.9	28.5	29.4	34.6	44.1	50.1	80.6	94.4	106.4	103.7
Min	26.6	18.7	15.9	16.3	16.8	15.9	15.9	15.9	15.9	15.9	15.9	26.2	28.0	35.8	62.2	52.7
Max	46.7	66.2	69.2	91.8	69.9	90.5	103.1	109.8	138.0	141.6	136.5	171.4	184.2	179.8	212.1	166.3
75%	40.7	39.5	42.1	43.1	43.1	43.6	45.9	50.6	58.2	63.7	72.7	92.6	113.8	127.6	148.1	148.3
Median	36.2	31.9	33.6	32.6	33.0	33.0	32.8	35.0	37.8	46.7	57.2	71.0	98.5	110.5	126.9	129.1

Dry Bulb Temperature Bins (F)

b) SPACE HEATING AND DHW LOADS (41 MF Buildings)



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
Met by CHP	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.5	0.6	0.6	0.6	0.7	0.6	0.6
25%	0.9	0.7	0.6	0.4	0.4	0.3	0.3	0.3	0.3	0.2	0.1	0.1	0.2	0.2	0.2	0.2
Min	0.8	0.2	0.1	0.1	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.2	0.2
Max	1.9	2.3	3.6	3.9	3.4	3.1	3.0	3.0	2.6	2.2	1.4	1.0	0.3	0.3	0.3	0.3
75%	1.6	1.4	1.5	1.2	1.2	1.1	0.6	0.8	0.5	0.3	0.3	0.2	0.2	0.2	0.3	0.3
Median	1.4	1.0	0.9	0.7	0.7	0.5	0.3	0.3	0.3	0.3	0.2	0.2	0.2	0.2	0.2	0.3

Dry Bulb Temperature Bins (F)

Figure D - 3: Temperature Bin Distribution of Electricity and Thermal Energy Requirements for Option 1

Note: 'Max Surp.' indicates the maximum surplus electrcity or thermal energy obtained from the CHP facility installed in the grocery store.

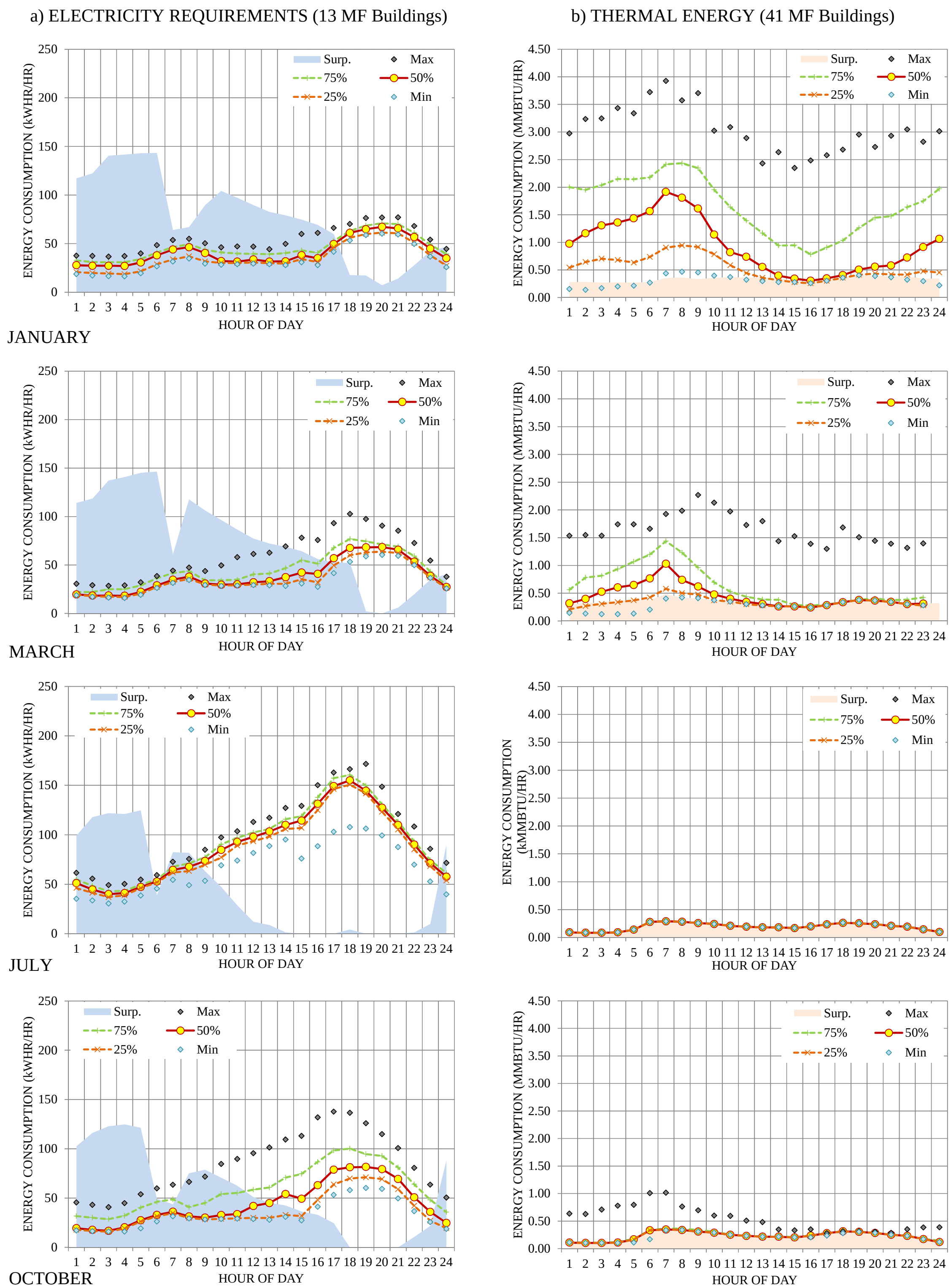
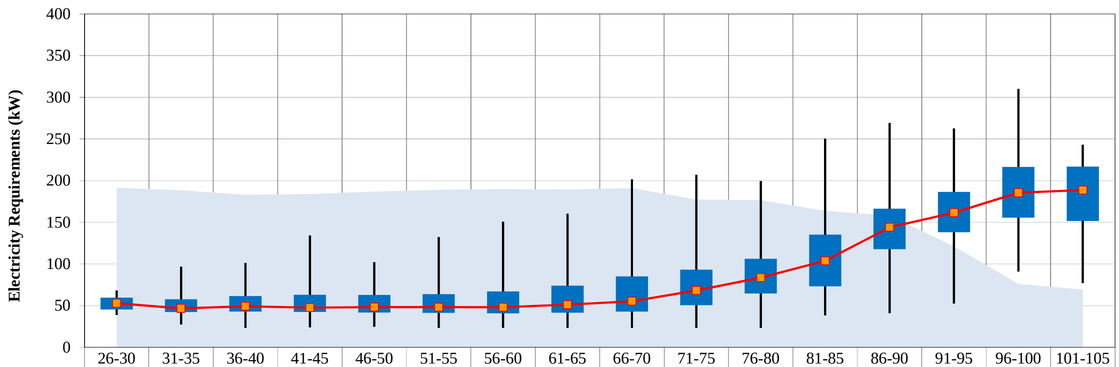


Figure D - 4: Typical Hourly Profiles for Electricity and Absorption Chiller Loads for Option 1

D – 3 Load Profiles for Option 2

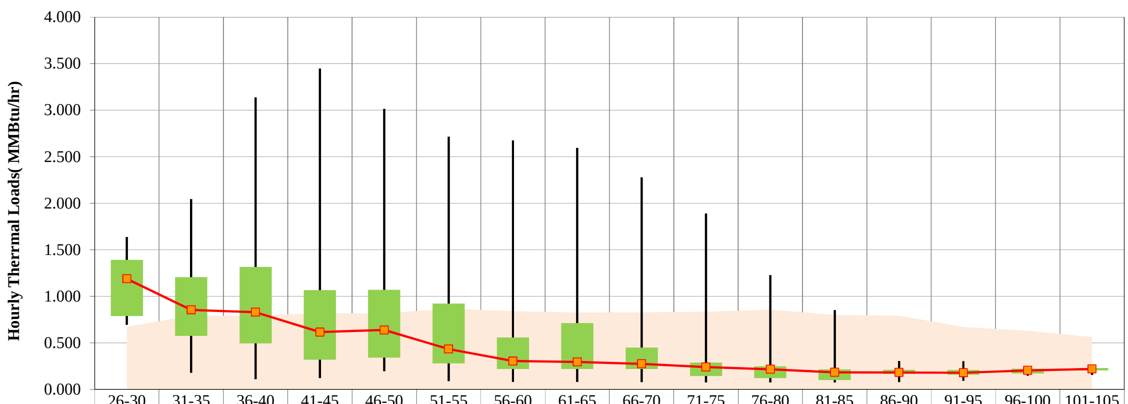
a) ELECTRICITY REQUIREMENTS (19 MF Buildings)



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
Met by CHP	191.5	188.4	183.1	183.9	186.9	189.0	189.9	189.4	191.1	177.3	176.6	163.6	158.1	121.0	76.1	69.4
25%	45.4	42.2	42.8	42.3	41.8	41.4	40.7	41.6	42.9	50.6	64.5	73.2	117.9	138.0	155.5	151.6
Min	38.8	27.3	23.3	23.8	24.6	23.3	23.3	23.3	23.3	23.3	23.3	38.3	40.9	52.3	90.9	77.0
Max	68.3	96.7	101.1	134.2	102.2	132.3	150.7	160.4	201.6	207.0	199.4	250.4	269.2	262.8	310.1	243.1
75%	59.5	57.8	61.6	63.0	63.0	63.7	67.1	74.0	85.1	93.1	106.3	135.3	166.4	186.5	216.4	216.8
Median	52.9	46.7	49.2	47.6	48.3	48.2	48.0	51.2	55.3	68.2	83.5	103.8	143.9	161.5	185.5	188.7

Dry Bulb Temperature Bins (F)

b) SPACE HEATING AND DHW LOADS (36 MF Buildings)



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
Met by CHP	0.7	0.8	0.8	0.8	0.8	0.9	0.8	0.8	0.8	0.8	0.9	0.8	0.8	0.7	0.6	0.6
25%	0.8	0.6	0.5	0.3	0.3	0.3	0.2	0.2	0.2	0.1	0.1	0.1	0.2	0.2	0.2	0.2
Min	0.7	0.2	0.1	0.1	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.2
Max	1.6	2.0	3.1	3.4	3.0	2.7	2.7	2.6	2.3	1.9	1.2	0.9	0.3	0.3	0.2	0.2
75%	1.4	1.2	1.3	1.1	1.1	0.9	0.6	0.7	0.5	0.3	0.3	0.2	0.2	0.2	0.2	0.2
Median	1.2	0.9	0.8	0.6	0.6	0.4	0.3	0.3	0.3	0.2	0.2	0.2	0.2	0.2	0.2	0.2

Dry Bulb Temperature Bins (F)

Figure D - 5: Temperature Bin Distribution of Electricity and Thermal Energy Requirements for Option 2

Note: 'Max Surp.' indicates the maximum surplus electrcity or thermal energy obtained from the CHP facility installed in the grocery store.

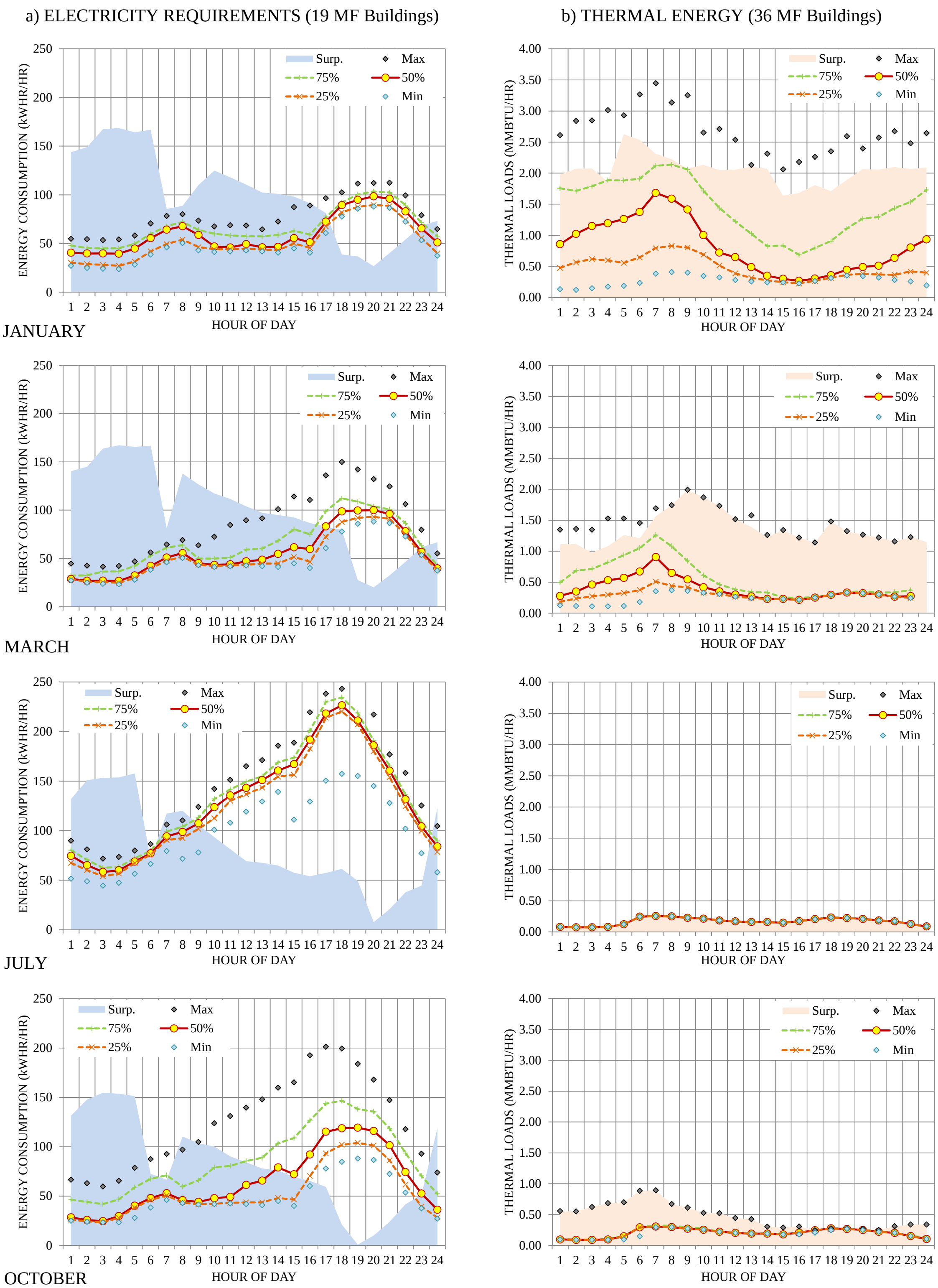
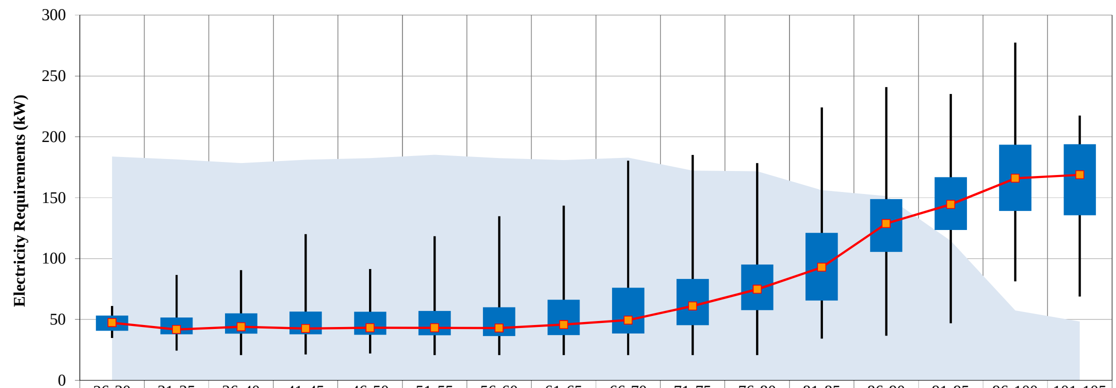


Figure D - 6: Typical Hourly Profiles for Electricity and Absorption Chiller Loads for Option 2

D – 4 Load Profiles for Option 3

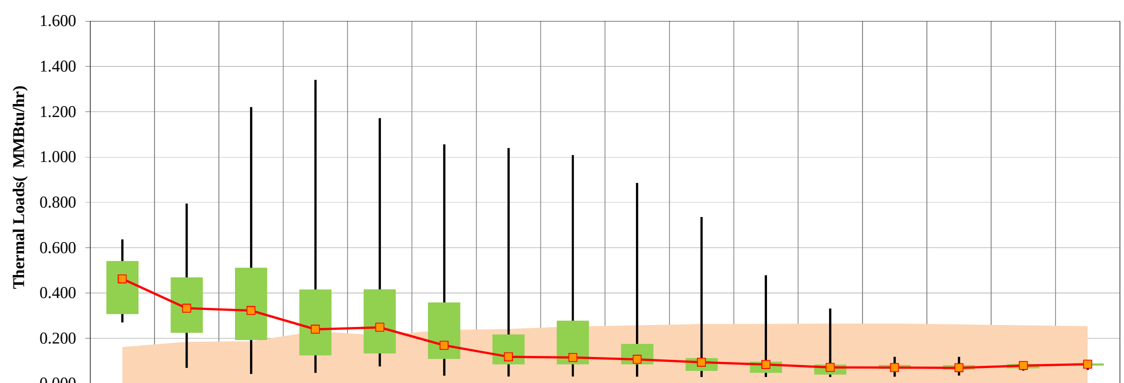
a) ELECTRICITY REQUIREMENTS (17 MF Buildings)



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
Max. Surp.	183.8	181.4	178.5	181.1	182.5	185.3	182.5	180.9	182.9	172.2	171.7	156.3	151.4	114.6	57.4	48.4
25%	40.7	37.8	38.3	37.9	37.4	37.1	36.4	37.2	38.4	45.3	57.7	65.5	105.4	123.5	139.1	135.6
Min	34.7	24.4	20.8	21.3	22.0	20.8	20.8	20.8	20.8	20.8	20.8	34.3	36.6	46.8	81.3	68.9
Max	61.1	86.6	90.5	120.1	91.4	118.4	134.8	143.5	180.4	185.2	178.4	224.1	240.8	235.1	277.4	217.5
75%	53.3	51.7	55.1	56.4	56.3	57.0	60.0	66.2	76.2	83.3	95.1	121.0	148.8	166.8	193.6	194.0
Median	47.3	41.8	44.0	42.6	43.2	43.1	42.9	45.8	49.5	61.0	74.7	92.9	128.8	144.5	166.0	168.8

Dry Bulb Temperature Bins (F)

b) SPACE HEATING AND DHW LOADS (14 MF Buildings)



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
Max. Surp.	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
25%	0.3	0.2	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.0	0.0	0.1	0.1	0.1	0.1
Min	0.3	0.1	0.0	0.0	0.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.1	0.1
Max	0.6	0.8	1.2	1.3	1.2	1.1	1.0	1.0	0.9	0.7	0.5	0.3	0.1	0.1	0.1	0.1
75%	0.5	0.5	0.5	0.4	0.4	0.4	0.2	0.3	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1
Median	0.5	0.3	0.3	0.2	0.2	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1

Dry Bulb Temperature Bins (F)

Figure D - 7: Temperature Bin Distribution of Electricity and Thermal Energy Requirements for Option 3

Note: 'Max Surp.' indicates the maximum surplus electrcity or thermal energy obtained from the CHP facility installed in the grocery store.

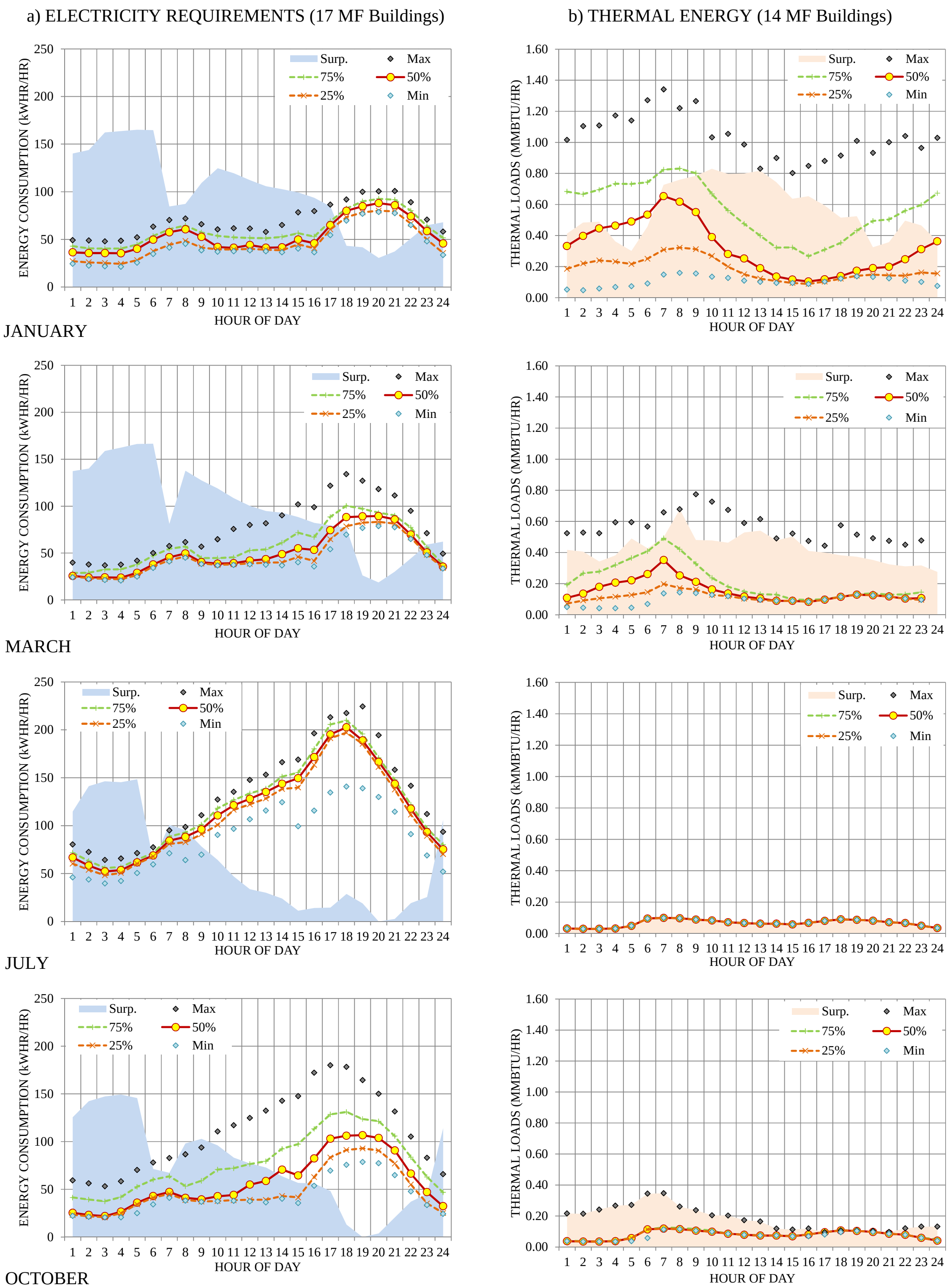
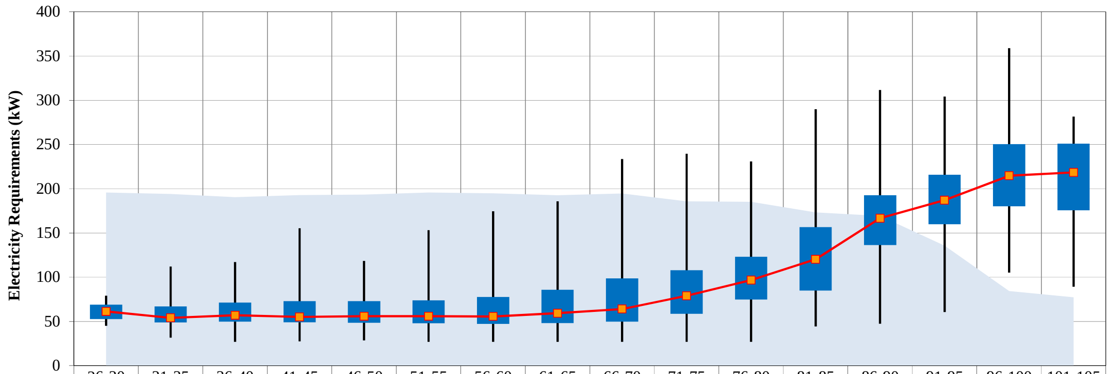


Figure D - 8: Typical Hourly Profiles for Electricity and Absorption Chiller Loads for Option 3

D – 4 Load Profiles for Option 4

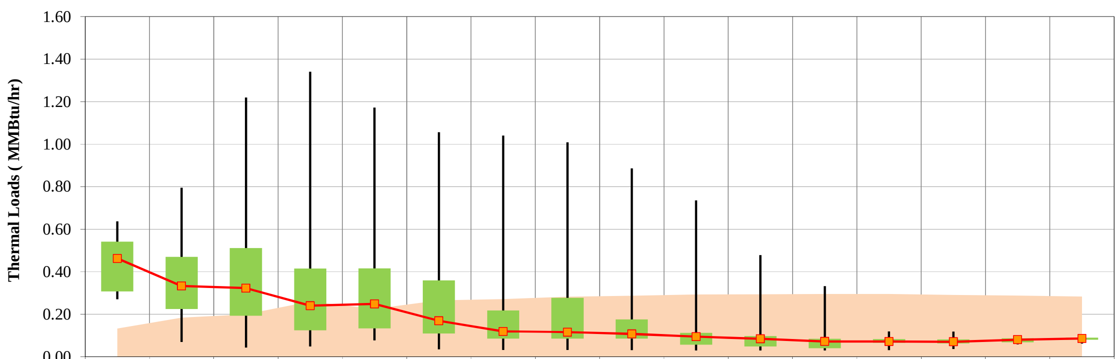
a) ELECTRICITY REQUIREMENTS (22 MF Buildings)



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
Max. Surp.	195.7	194.2	190.5	192.8	193.3	195.8	194.8	192.7	194.7	185.9	185.3	173.3	169.2	135.6	84.4	77.2
25%	52.6	48.9	49.6	49.0	48.4	47.9	47.1	48.2	49.7	58.6	74.6	84.8	136.5	159.8	180.1	175.5
Min	45.0	31.6	26.9	27.5	28.5	26.9	26.9	26.9	26.9	26.9	26.9	44.4	47.3	60.6	105.2	89.1
Max	79.1	112.0	117.1	155.4	118.3	153.2	174.5	185.8	233.5	239.7	230.9	290.0	311.7	304.3	359.0	281.5
75%	68.9	66.9	71.3	73.0	72.9	73.7	77.6	85.7	98.6	107.8	123.1	156.7	192.6	215.9	250.5	251.0
Median	61.2	54.0	56.9	55.1	55.9	55.8	55.6	59.3	64.0	79.0	96.7	120.2	166.7	187.0	214.8	218.4

Dry Bulb Temperature Bins (F)

b) SPACE HEATING AND DHW LOADS (14 MF Buildings)



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
Max. Surp.	0.1	0.2	0.2	0.3	0.2	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
25%	0.3	0.2	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.0	0.0	0.1	0.1	0.1	0.1
Min	0.3	0.1	0.0	0.0	0.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.1	0.1
Max	0.6	0.8	1.2	1.3	1.2	1.1	1.0	1.0	0.9	0.7	0.5	0.3	0.1	0.1	0.1	0.1
75%	0.5	0.5	0.5	0.4	0.4	0.4	0.2	0.3	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1
Median	0.5	0.3	0.3	0.2	0.2	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1

Dry Bulb Temperature Bins (F)

Figure D - 9: Temperature Bin Distribution of Electricity and Thermal Energy Requirements for Option 4

Note: ‘Max Surp.’ indicates the maximum surplus electrcity or thermal energy obtained from the CHP facility installed in the grocery store.

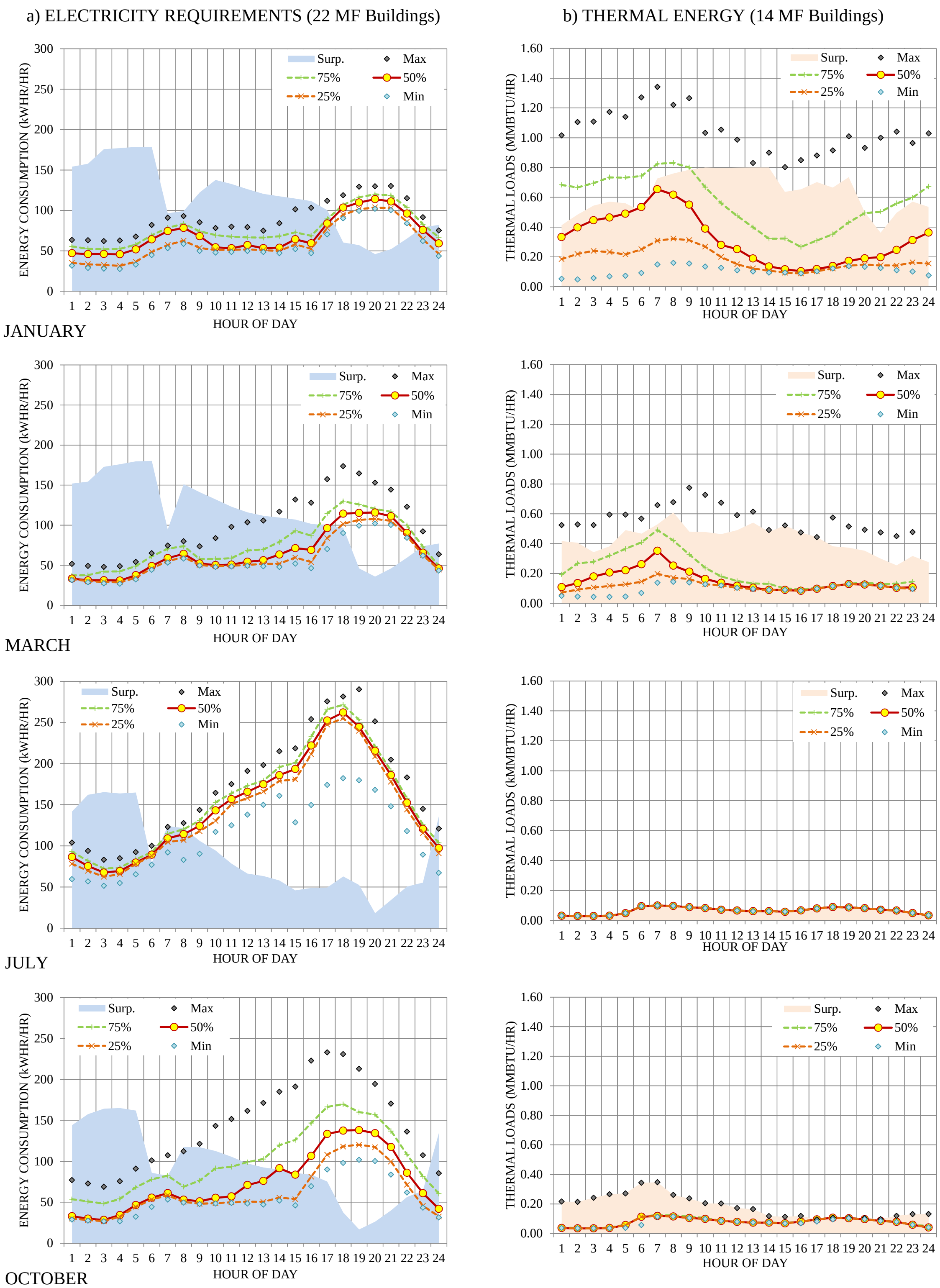


Figure D - 10: Typical Hourly Profiles for Electricity and Absorption Chiller Loads for Option 4

APPENDIX E

GROCERY STORE LOAD PROFILES AND CHP PERFORMANCE SUMMARY PLOTS

This appendix provides the breakdown of the annual energy consumption of the grocery store upon installing the CHP option. A breakdown of the annual energy consumption of the corresponding base-case scenarios is also presented. The appendix also provides the electricity consumption and thermal load profiles for the grocery store for the four CHP scenarios. The load profiles are presented using temperature-bin and hourly plots. The electricity and thermal energy provided by the installed CHP system in the four options is overlaid on the temperature-bin and hourly plots of the grocery store.

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E – 1.1 Annual Energy Consumption for Option 1

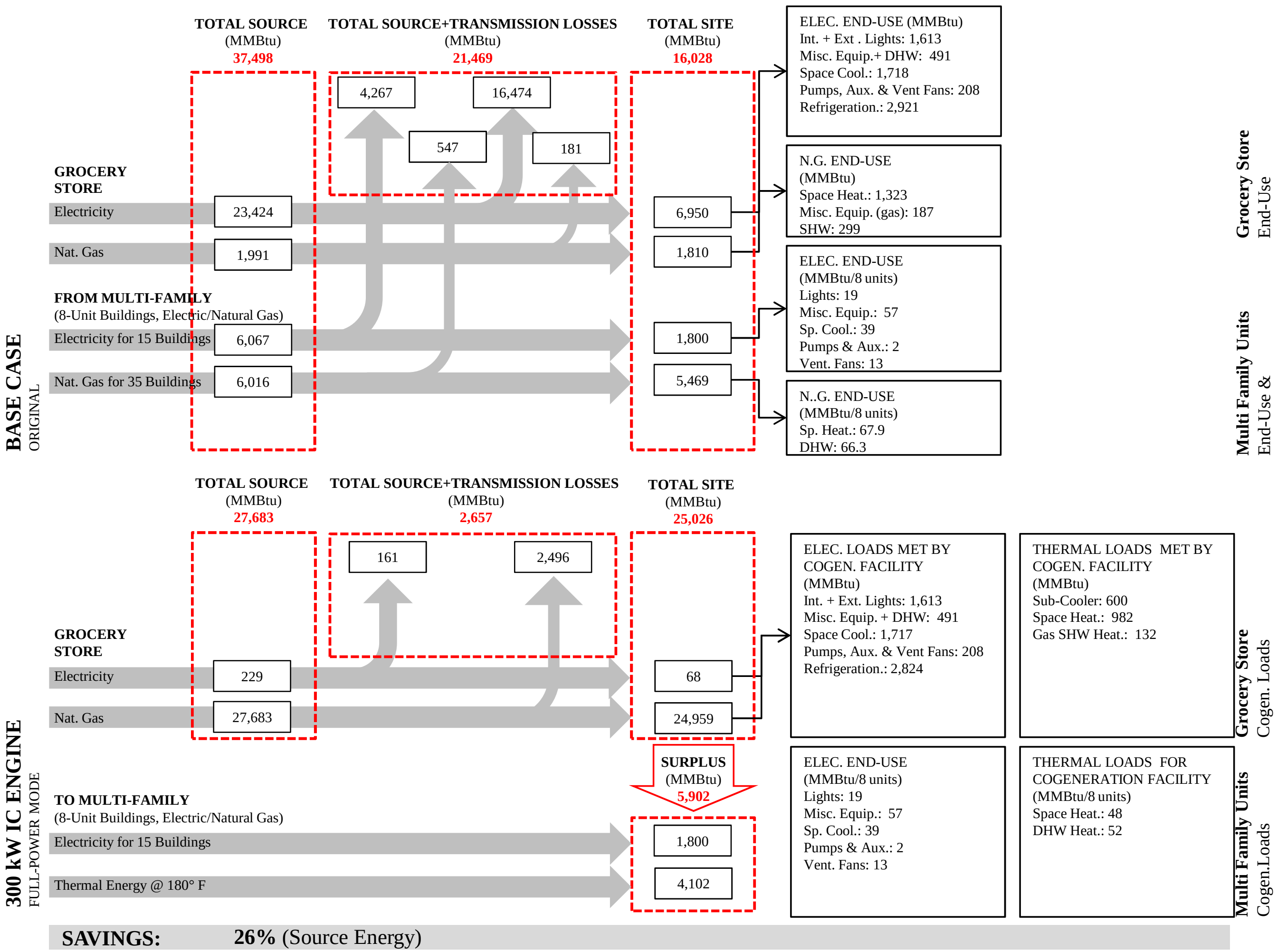
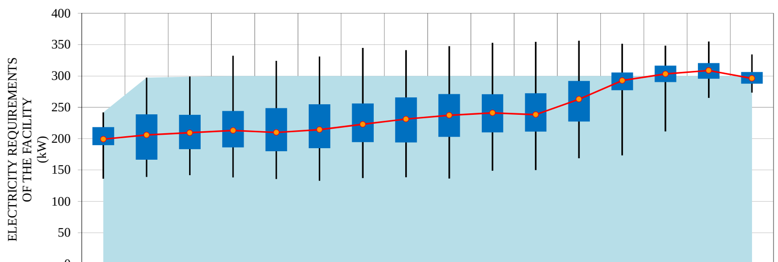


Figure E - 1: Annual Energy Consumption for Option

E – 1.2 Hourly Load Profiles for Option 1

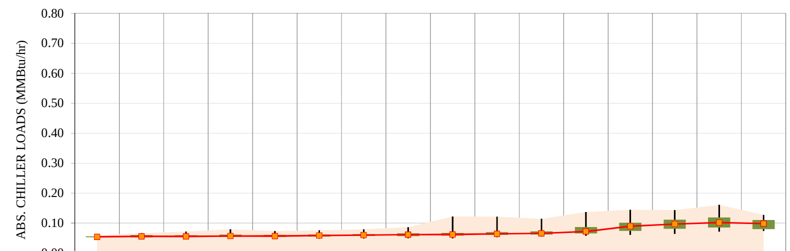
a) ELECTRICITY REQUIREMENTS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Elec.	241.9	297.5	299.1	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0
25%	189.7	166.4	183.2	186.2	180.0	185.0	194.5	194.1	203.0	210.1	211.3	227.3	277.4	290.2	295.7	287.6
Min	136.2	138.9	141.8	138.2	135.8	132.8	137.3	138.5	136.5	148.9	150.0	168.7	173.3	211.5	265.1	273.5
Max	241.9	297.5	299.1	332.3	324.1	331.1	344.9	341.4	347.5	353.1	354.5	356.2	351.4	348.2	354.9	334.5
75%	218.5	238.8	238.0	244.2	248.6	254.8	256.2	265.8	271.1	270.8	272.3	292.2	305.6	316.4	320.6	306.3
Median	199.3	205.9	209.4	213.1	210.0	214.5	222.9	231.3	237.3	241.1	238.4	263.0	292.8	303.3	308.8	296.2

DRY BULB TEMPERATURE BINS (F)

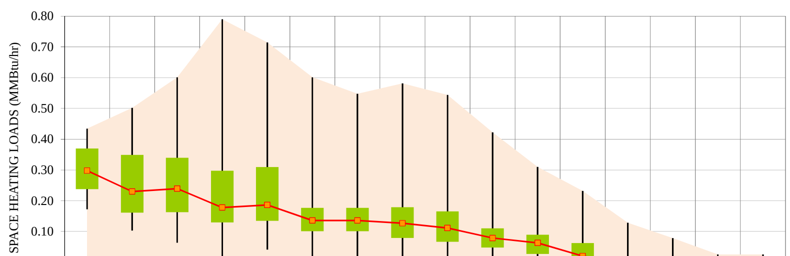
b) ABS. CHILLER LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Th.	0.059	0.067	0.072	0.079	0.073	0.076	0.079	0.086	0.122	0.121	0.114	0.137	0.144	0.143	0.161	0.127
25%	0.052	0.052	0.053	0.053	0.054	0.055	0.057	0.056	0.058	0.060	0.061	0.065	0.074	0.081	0.085	0.080
Min	0.050	0.049	0.048	0.048	0.048	0.047	0.048	0.049	0.049	0.053	0.055	0.058	0.061	0.064	0.071	0.073
Max	0.059	0.067	0.072	0.079	0.073	0.076	0.079	0.086	0.122	0.121	0.114	0.137	0.144	0.143	0.161	0.127
75%	0.056	0.060	0.059	0.062	0.061	0.062	0.063	0.066	0.067	0.070	0.072	0.087	0.101	0.111	0.119	0.110
Median	0.054	0.056	0.055	0.057	0.056	0.058	0.060	0.061	0.062	0.064	0.066	0.072	0.089	0.097	0.102	0.098

DRY BULB TEMPERATURE (F)

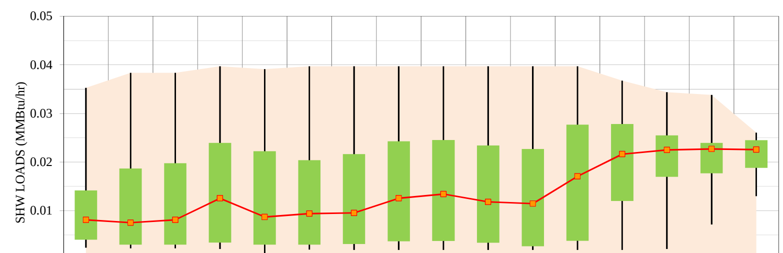
c) SPACE HEATING LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Th.	0.434	0.502	0.601	0.790	0.715	0.602	0.548	0.582	0.544	0.422	0.310	0.232	0.129	0.078	0.026	0.026
25%	0.238	0.161	0.163	0.130	0.134	0.101	0.101	0.079	0.066	0.048	0.027	0.000	0.000	0.000	0.000	0.000
Min	0.172	0.103	0.064	0.000	0.041	0.007	0.007	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
Max	0.434	0.502	0.601	0.790	0.715	0.602	0.548	0.582	0.544	0.422	0.310	0.232	0.129	0.078	0.026	0.026
75%	0.370	0.349	0.340	0.297	0.310	0.177	0.177	0.179	0.165	0.110	0.089	0.062	0.000	0.000	0.000	0.000
Median	0.298	0.230	0.239	0.178	0.186	0.136	0.136	0.127	0.111	0.079	0.063	0.020	0.000	0.000	0.000	0.000

DRY BULB TEMPERATURE BINS (F)

d) SHW LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Th.	0.035	0.038	0.038	0.040	0.039	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.037	0.034	0.034	0.026
25%	0.004	0.003	0.003	0.003	0.003	0.003	0.003	0.004	0.004	0.003	0.003	0.004	0.012	0.017	0.018	0.019
Min	0.002	0.002	0.002	0.002	0.001	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.007	0.013
Max	0.035	0.038	0.038	0.040	0.039	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.037	0.034	0.034	0.026
75%	0.014	0.019	0.020	0.024	0.022	0.020	0.022	0.024	0.025	0.023	0.023	0.028	0.028	0.025	0.024	0.024
Median	0.008	0.008	0.008	0.013	0.009	0.009	0.010	0.013	0.013	0.012	0.011	0.017	0.022	0.022	0.023	0.023

DRY BULB TEMPERATURE BINS (F)

Figure E - 2: Temperature Bin Distribution of Electricity, Absorption Chiller, Space Heating and SHW Loads for Option 1

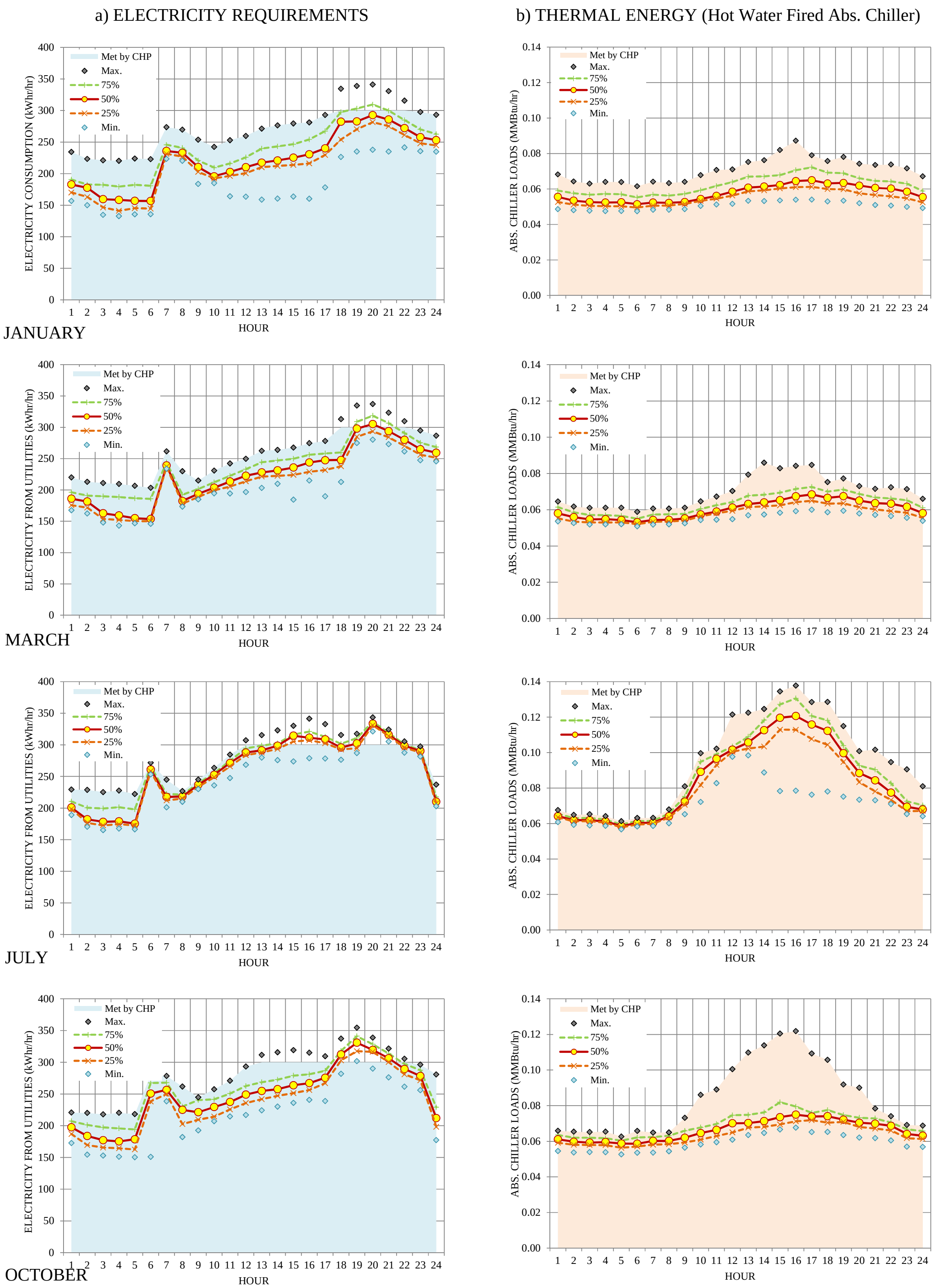


Figure E - 3: Typical Hourly Profiles for Electricity and Absorption Chiller Loads for Option 1

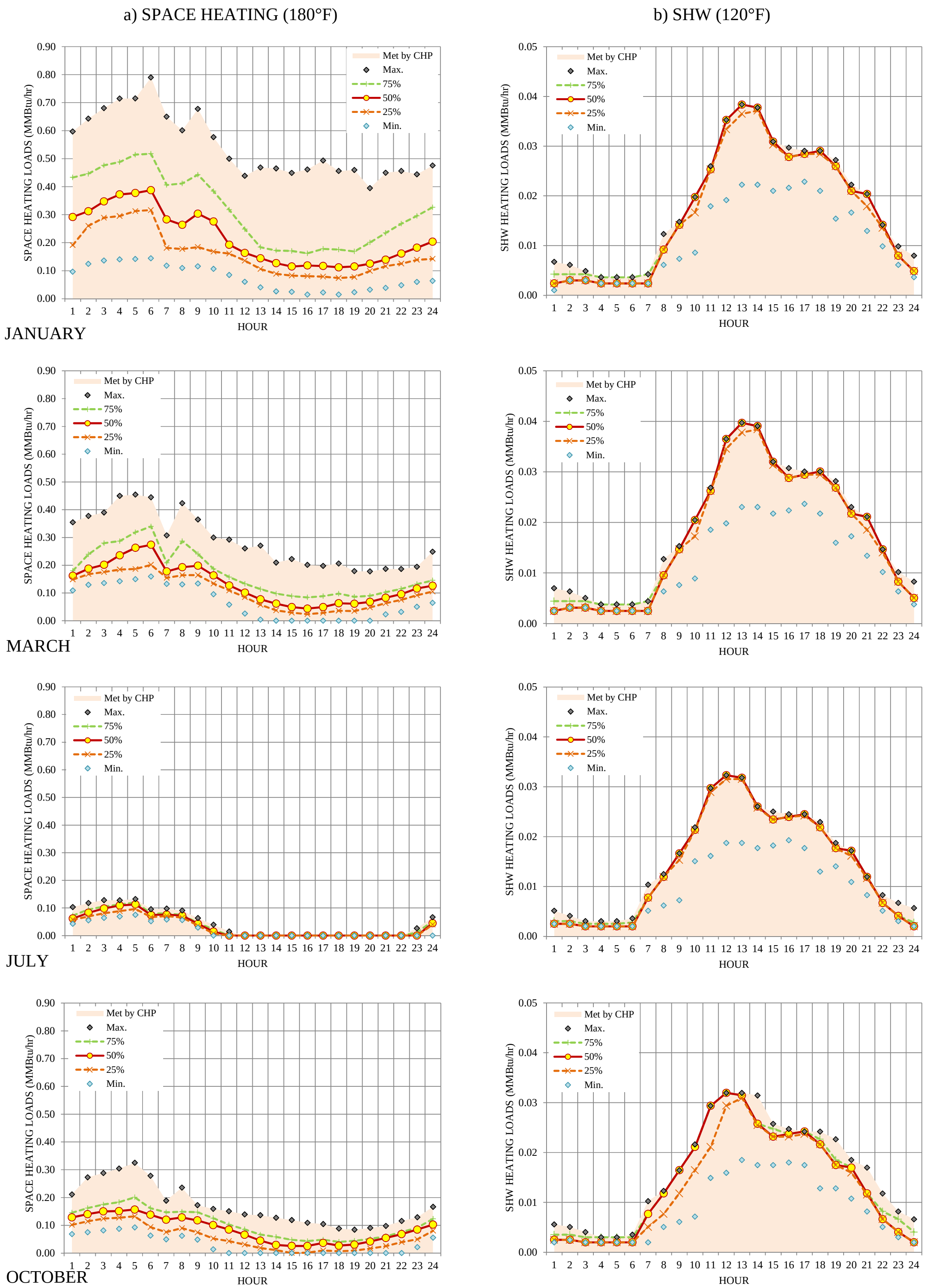


Figure E - 4: Typical Hourly Profiles for Space Heating and SHW Loads for Option 1

Figure E - 5: Typical Hourly Profiles for Surplus Electricity and Thermal Energy from Option 1

E – 2.1 Annual Energy Consumption for Option 2

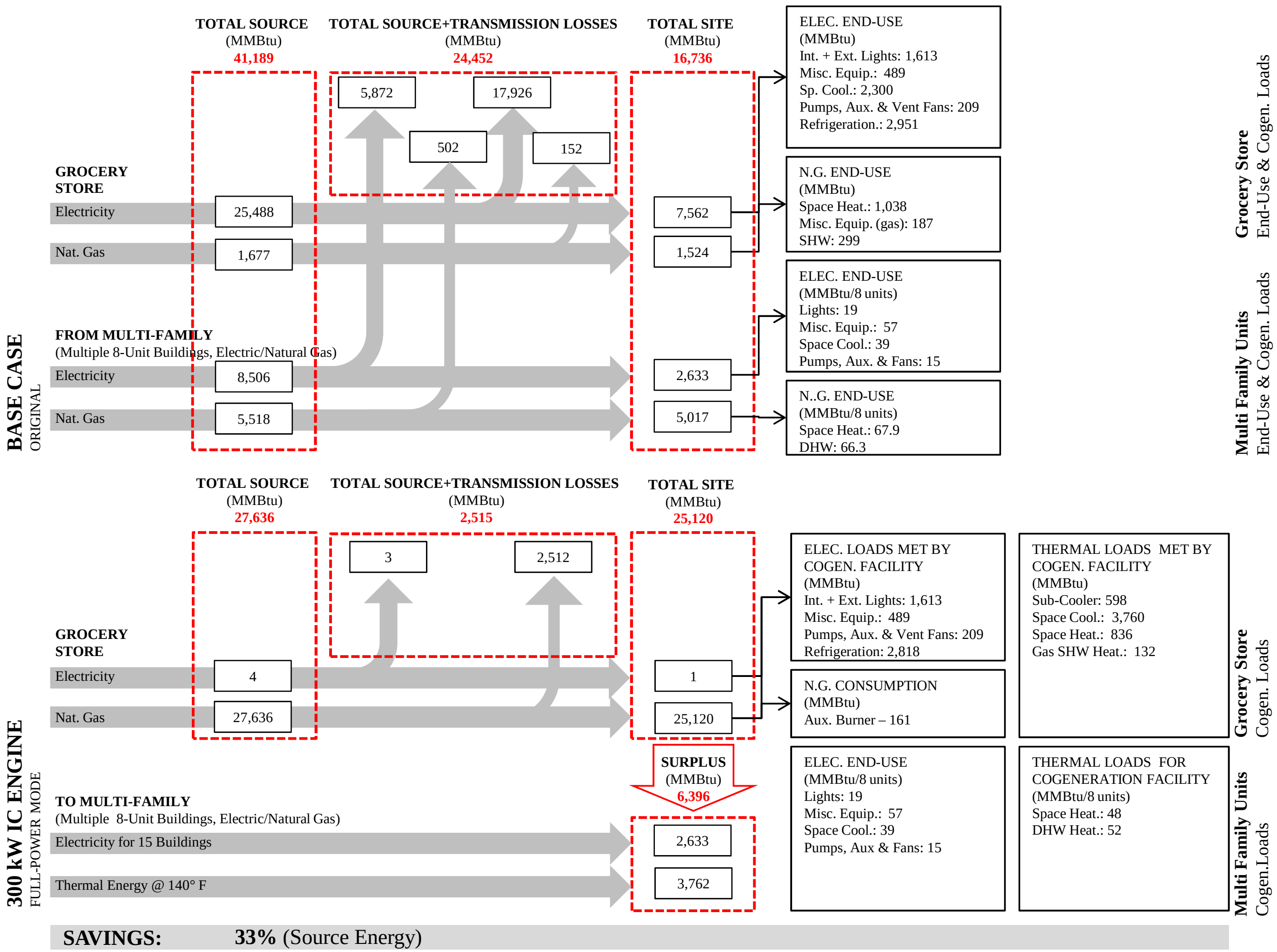
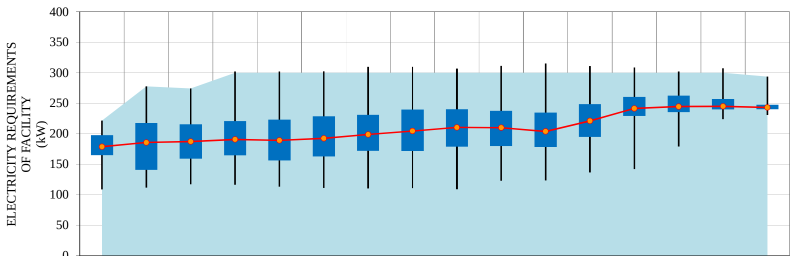


Figure E - 6: Annual Energy Consumption for Option 2

E – 2.2 Load Profiles for Option 2

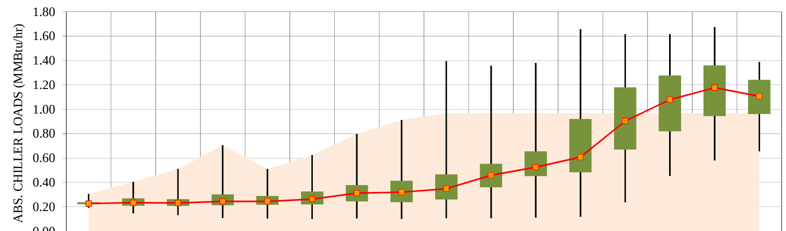
a) ELECTRICITY REQUIREMENTS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Elec.	221.5	277.8	274.3	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	293.8
25%	164.9	140.5	159.1	164.5	156.2	162.7	171.9	171.7	178.7	179.8	178.2	194.7	229.1	235.3	239.8	240.4
Min	108.5	111.6	116.9	116.1	113.1	111.0	110.1	110.6	108.9	122.7	123.4	136.4	141.9	179.0	223.9	230.6
Max	221.5	277.8	274.3	302.2	302.2	302.4	309.6	309.7	306.8	311.2	315.4	311.0	308.7	302.2	307.4	293.8
75%	197.7	217.5	215.5	220.7	223.0	228.7	230.9	239.7	240.1	237.4	234.8	248.6	260.4	262.7	256.9	247.6
Median	178.7	185.6	187.1	190.6	189.1	192.3	198.8	204.4	210.3	210.0	203.8	221.0	241.5	244.5	244.9	243.1

DRY BULB TEMPERATURE (F)

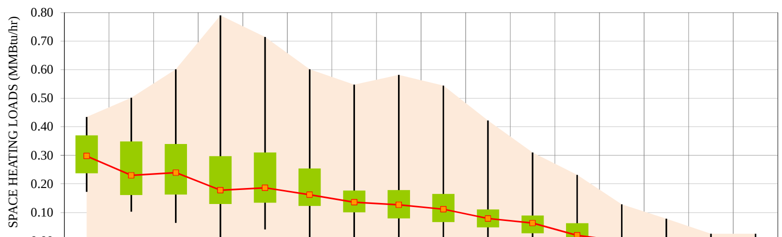
b) ABS. CHILLER LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Thermal	0.306	0.406	0.511	0.706	0.511	0.625	0.796	0.913	0.967	0.967	0.967	0.967	0.967	0.967	0.967	0.967
25%	0.220	0.209	0.208	0.213	0.217	0.219	0.245	0.238	0.260	0.360	0.452	0.484	0.669	0.819	0.945	0.960
Min	0.192	0.146	0.132	0.108	0.105	0.100	0.104	0.100	0.106	0.107	0.112	0.119	0.236	0.453	0.580	0.656
Max	0.306	0.406	0.511	0.706	0.511	0.625	0.796	0.913	1.395	1.358	1.382	1.658	1.618	1.617	1.676	1.389
75%	0.239	0.270	0.263	0.303	0.289	0.327	0.378	0.414	0.466	0.554	0.655	0.921	1.181	1.277	1.360	1.242
Median	0.227	0.234	0.232	0.245	0.245	0.263	0.313	0.321	0.349	0.459	0.525	0.608	0.904	1.079	1.177	1.107

DRY BULB TEMPERATURE BINS (F)

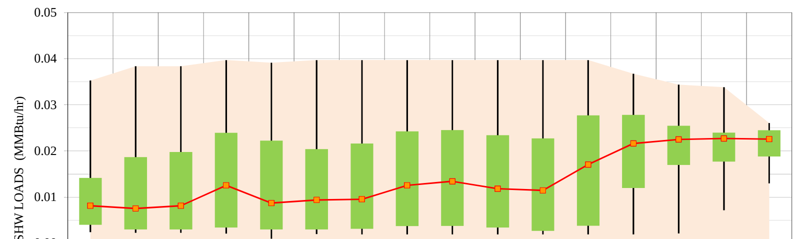
c) SPACE HEATING LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Thermal	0.434	0.502	0.601	0.790	0.715	0.602	0.548	0.582	0.544	0.422	0.310	0.232	0.129	0.078	0.026	0.026
25%	0.238	0.161	0.163	0.130	0.134	0.123	0.101	0.079	0.066	0.048	0.027	0.000	0.000	0.000	0.000	0.000
Min	0.172	0.103	0.064	0.000	0.041	0.000	0.007	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
Max	0.434	0.502	0.601	0.790	0.715	0.602	0.548	0.582	0.544	0.422	0.310	0.232	0.129	0.078	0.026	0.026
75%	0.370	0.349	0.340	0.297	0.310	0.254	0.177	0.179	0.165	0.110	0.089	0.062	0.000	0.000	0.000	0.000
Median	0.298	0.230	0.239	0.178	0.186	0.162	0.136	0.127	0.111	0.079	0.063	0.020	0.000	0.000	0.000	0.000

DRY BULB TEMPERATURE BINS (F)

d) SHW LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Thermal	0.035	0.038	0.038	0.040	0.039	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.037	0.034	0.034	0.026
25%	0.004	0.003	0.003	0.003	0.003	0.003	0.003	0.004	0.004	0.003	0.003	0.004	0.012	0.017	0.018	0.019
Min	0.002	0.002	0.002	0.002	0.001	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.007	0.013
Max	0.035	0.038	0.038	0.040	0.039	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.037	0.034	0.034	0.026
75%	0.014	0.019	0.020	0.024	0.022	0.020	0.022	0.024	0.025	0.023	0.023	0.028	0.028	0.025	0.024	0.024
Median	0.008	0.008	0.008	0.013	0.009	0.009	0.010	0.013	0.013	0.012	0.011	0.017	0.022	0.022	0.023	0.023

DRY BULB TEMPERATURE BINS (F)

Figure E - 7: Temperature Bin Distribution of Electricity, Absorption Chiller, Space Heating and SHW Loads for Option 2

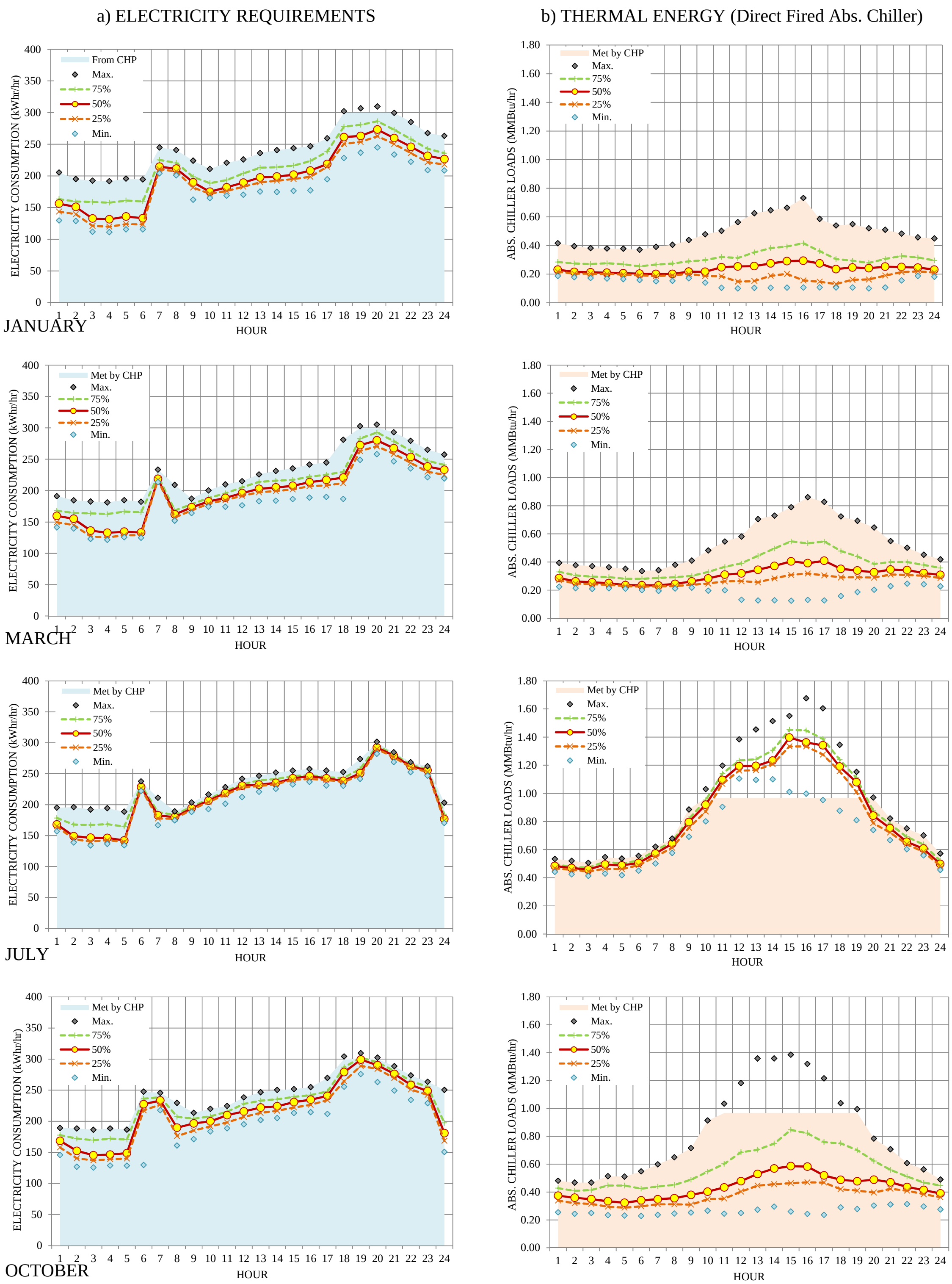


Figure E - 8: Typical Hourly Profiles for Electricity and Absorption Chiller Loads for Option 2

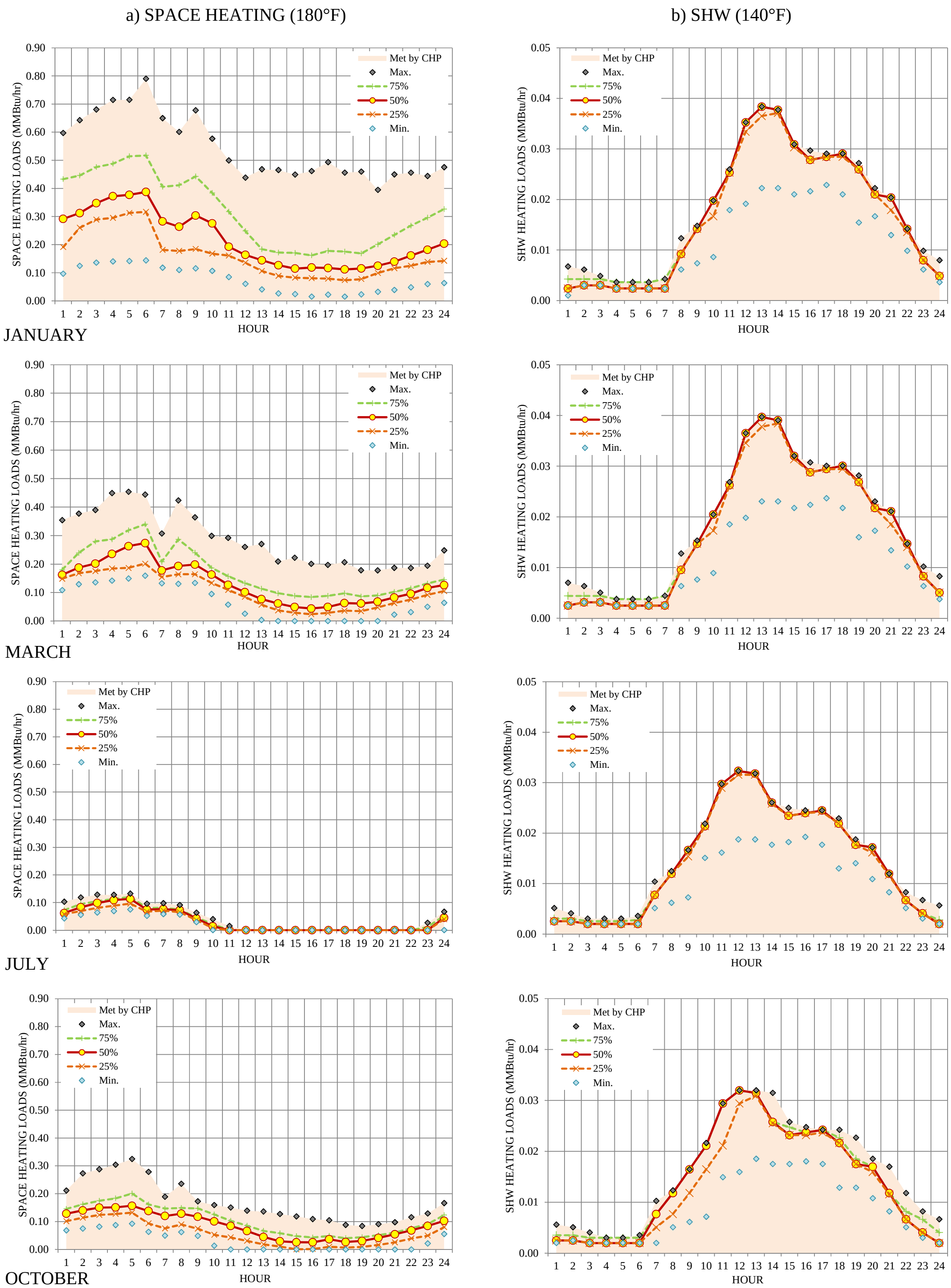


Figure E - 9: Typical Hourly Profiles for Space Heating and SHW Loads for Option 2

Figure E - 10: Typical Hourly Profiles for Surplus Electricity and Thermal Energy from Option 2

E – 3.1 Annual Energy Consumption for Option 3

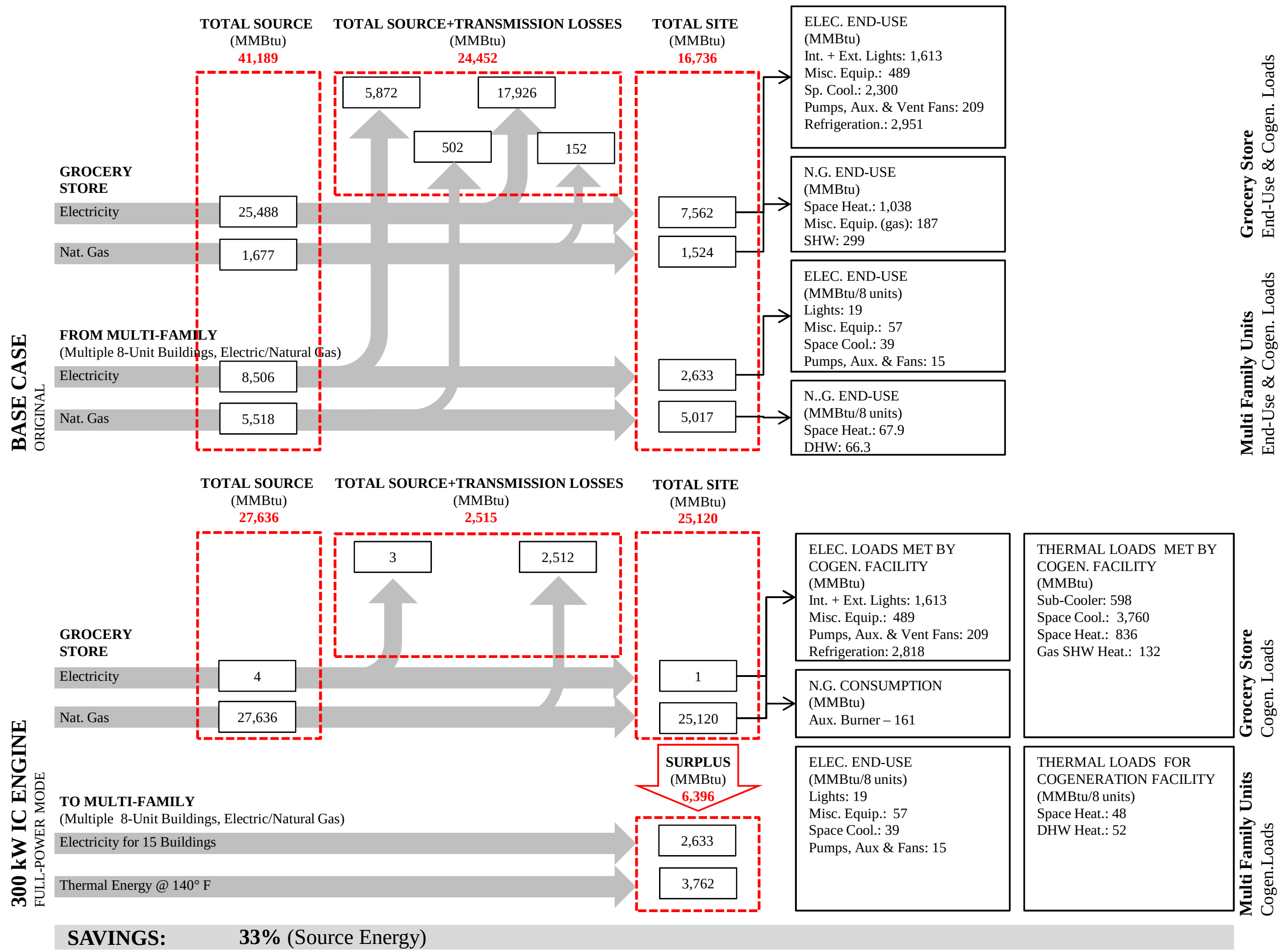
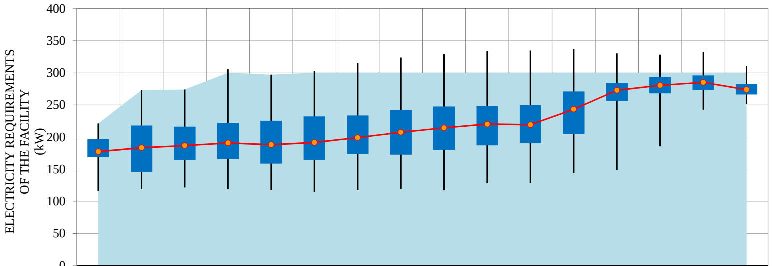


Figure E - 11: Annual energy Consumption for Option 3

E – 2.2 Load Profiles for Option 3

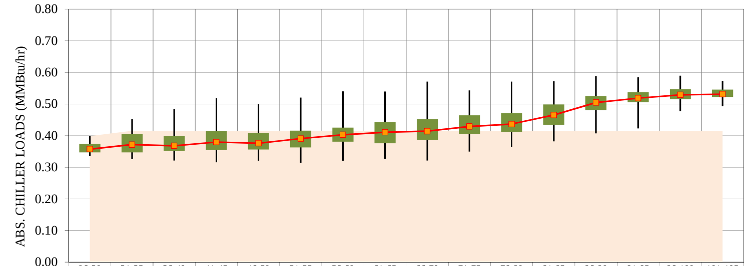
a) ELECTRICITY REQUIREMENTS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Elec.	221.0	272.9	273.8	300.0	297.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0	300.0
25%	168.5	145.4	163.9	165.8	158.5	164.0	173.2	172.6	180.1	186.8	190.1	204.9	256.1	267.9	273.3	266.2
Min	116.2	118.6	121.5	118.9	117.5	114.7	117.5	119.1	117.1	127.8	128.3	143.7	148.6	185.4	242.6	251.6
Max	221.0	272.9	273.8	305.5	297.0	302.5	315.4	323.7	328.8	334.3	334.7	336.9	330.3	328.1	332.7	310.7
75%	196.8	217.8	216.1	222.0	225.4	232.1	233.4	241.9	247.4	247.9	249.7	271.0	283.6	293.1	295.8	282.9
Median	177.2	183.3	186.6	190.7	187.9	191.4	199.0	207.3	214.2	220.0	219.1	243.2	272.7	280.5	285.0	273.7

DRY BULB TEMPERATURE BINS(F)

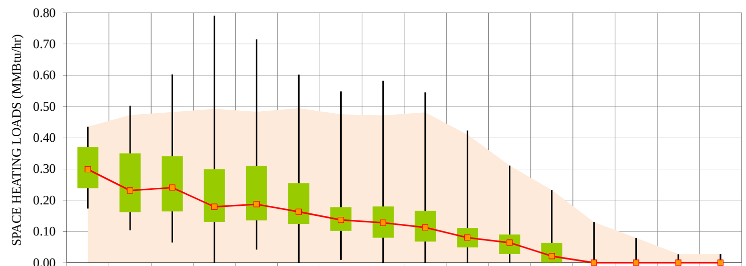
b) ABS. CHILLER LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Thermal	0.398	0.415	0.415	0.415	0.415	0.415	0.415	0.415	0.415	0.415	0.415	0.415	0.415	0.415	0.415	0.415
25%	0.347	0.347	0.352	0.355	0.356	0.363	0.381	0.376	0.386	0.405	0.412	0.434	0.481	0.505	0.515	0.522
Min	0.335	0.326	0.321	0.315	0.320	0.314	0.321	0.327	0.322	0.350	0.364	0.382	0.407	0.423	0.477	0.493
Max	0.398	0.452	0.485	0.518	0.499	0.520	0.540	0.539	0.571	0.543	0.571	0.572	0.589	0.584	0.589	0.573
75%	0.374	0.405	0.398	0.414	0.409	0.416	0.425	0.443	0.452	0.464	0.471	0.499	0.525	0.537	0.547	0.545
Median	0.357	0.372	0.368	0.379	0.376	0.391	0.403	0.411	0.414	0.429	0.436	0.465	0.505	0.519	0.529	0.531

DRY BULB TEMPERATURE BINS (F)

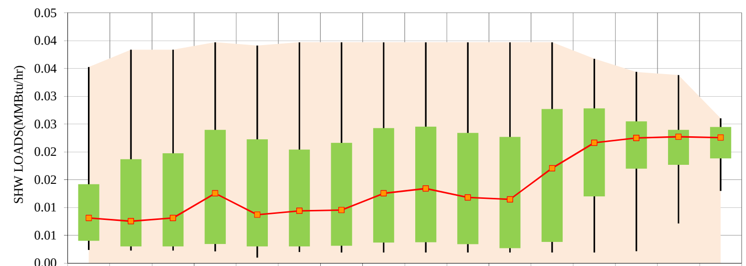
c) SPACE HEATING LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Thermal	0.435	0.473	0.482	0.493	0.483	0.495	0.475	0.471	0.481	0.410	0.311	0.233	0.130	0.079	0.027	0.027
25%	0.239	0.162	0.164	0.131	0.135	0.124	0.102	0.080	0.067	0.049	0.028	0.000	0.000	0.000	0.000	0.000
Min	0.173	0.104	0.065	0.000	0.042	0.000	0.009	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
Max	0.435	0.503	0.603	0.791	0.715	0.602	0.549	0.583	0.545	0.423	0.311	0.233	0.130	0.079	0.027	0.027
75%	0.371	0.350	0.341	0.299	0.311	0.255	0.178	0.180	0.166	0.112	0.090	0.063	0.002	0.000	0.000	0.000
Median	0.299	0.231	0.240	0.179	0.187	0.163	0.137	0.128	0.113	0.080	0.064	0.021	0.000	0.000	0.000	0.000

DRY BULB TEMPERATURE (F)

d) SHW LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Thermal	0.035	0.038	0.038	0.040	0.039	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.037	0.034	0.034	0.026
25%	0.004	0.003	0.003	0.003	0.003	0.003	0.003	0.004	0.004	0.003	0.003	0.004	0.012	0.017	0.018	0.019
Min	0.002	0.002	0.002	0.002	0.001	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.007	0.013
Max	0.035	0.038	0.038	0.040	0.039	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.037	0.034	0.034	0.026
75%	0.014	0.019	0.020	0.024	0.022	0.020	0.022	0.024	0.025	0.023	0.023	0.028	0.028	0.025	0.024	0.024
Median	0.008	0.008	0.008	0.013	0.009	0.009	0.010	0.013	0.013	0.012	0.011	0.017	0.022	0.022	0.023	0.023

DRY BULB TEMPERATURE BINS (F)

Figure E - 12: Temperature Bin Distribution of Electricity, Absorption Chiller, Space Heating and SHW Loads for Option 3

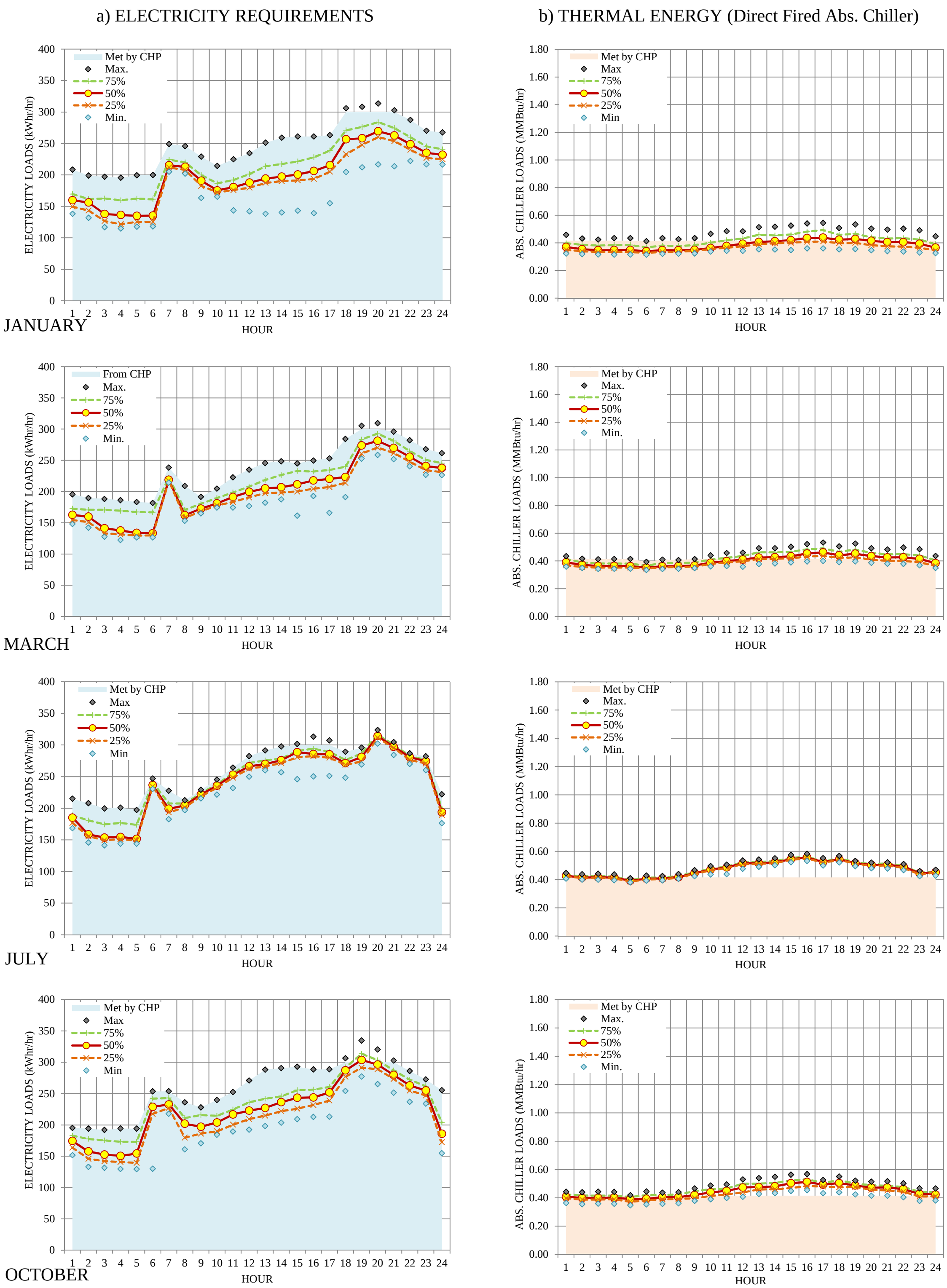


Figure E - 13: Typical Hourly Profiles for Electricity and Absorption Chiller Loads for Option 3

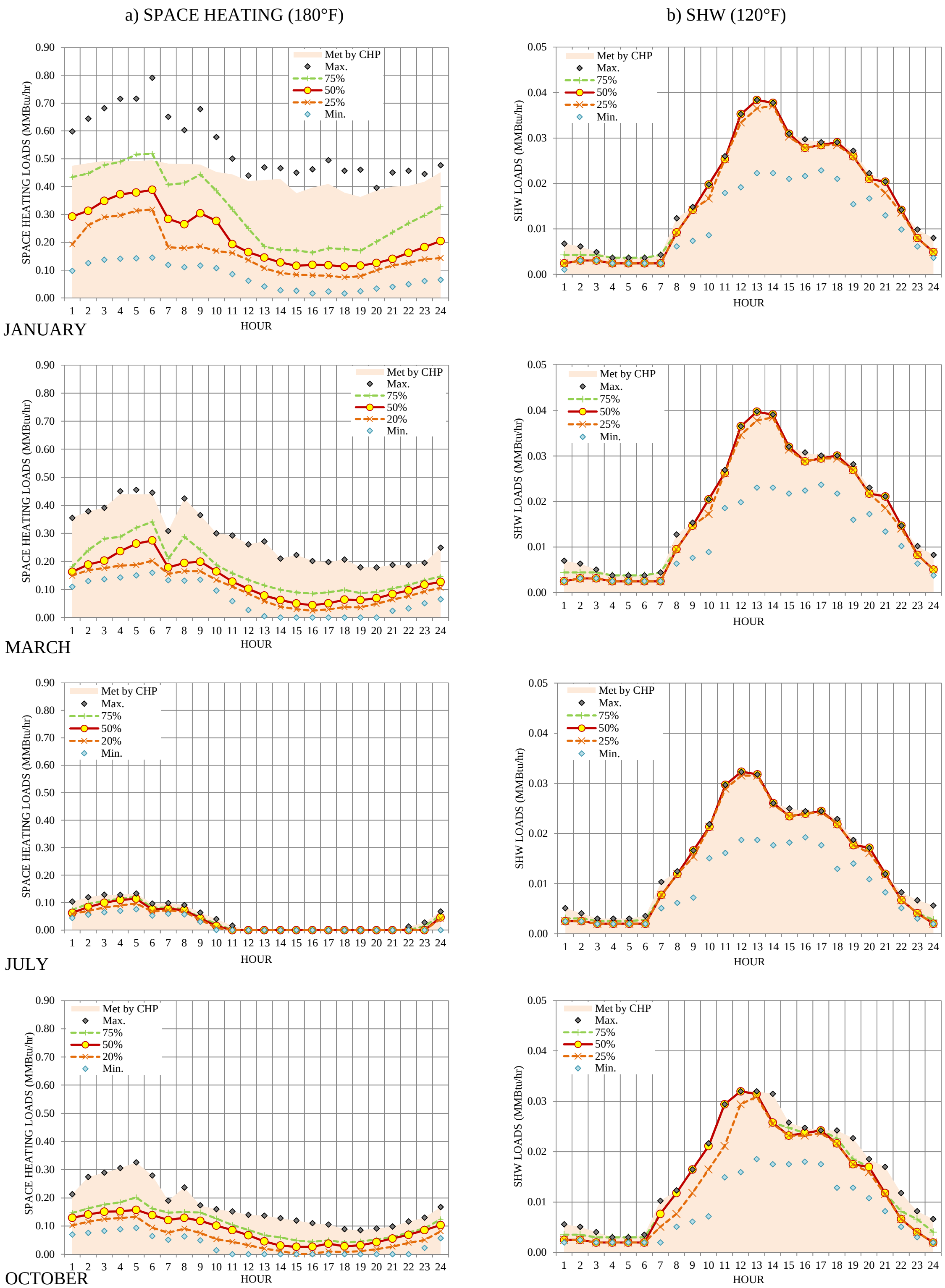


Figure E - 14: Typical Hourly Profiles for Space Heating and SHW Loads for Option 3

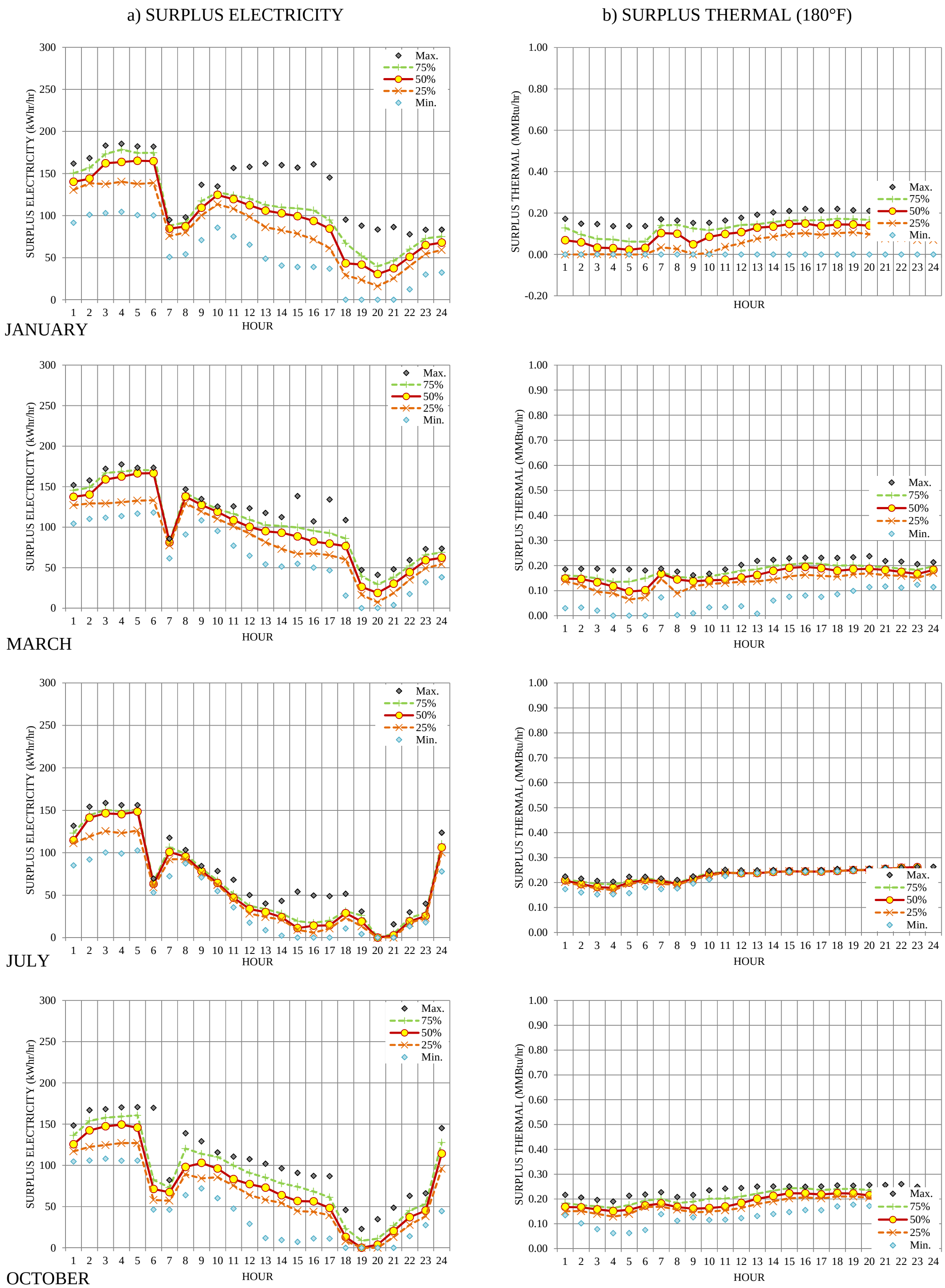


Figure E - 15: Typical Hourly Profiles for Surplus Electricity and Thermal Energy from Option 3

E – 4.1 Annual Energy Consumption for Option 4

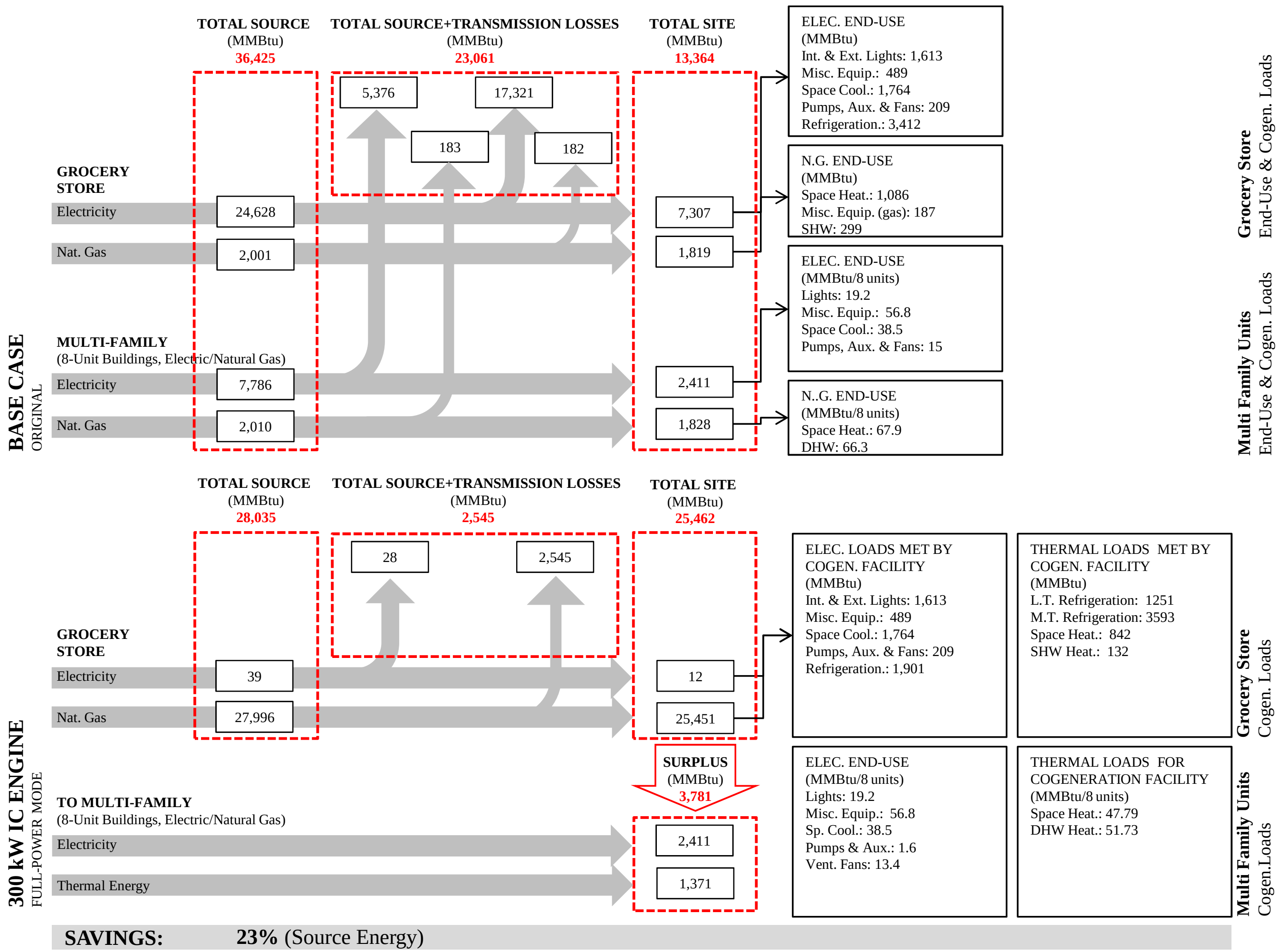
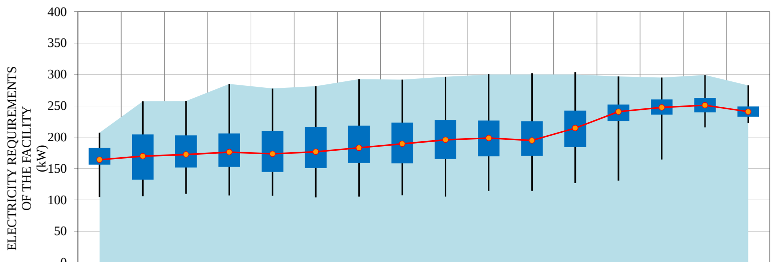


Figure E - 16: Annual Energy Consumption for Option 4

E – 4.2 Load Profiles for Option 4

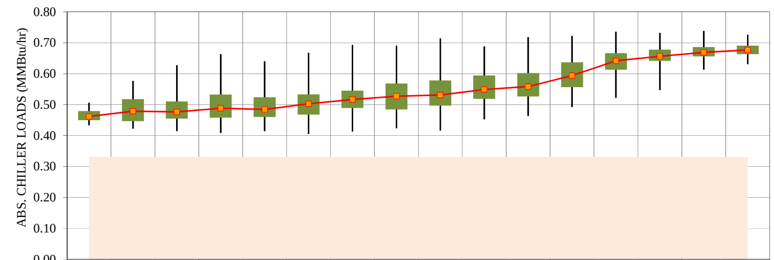
a) ELECTRICITY REQUIREMENTS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Elec.	207.217	257.136	257.772	284.997	277.691	281.306	292.582	291.826	296.323	300.000	300.000	300.000	296.930	295.170	299.281	282.606
25%	156.5	132.4	151.8	152.8	144.5	150.8	158.8	158.3	165.2	169.6	170.3	184.1	225.8	236.0	239.7	232.7
Min	104.3	105.8	109.5	107.2	106.7	104.2	105.2	107.3	105.3	114.1	114.7	126.7	130.8	164.4	215.6	222.8
Max	207.2	257.1	257.8	285.0	277.7	281.3	292.6	291.8	296.3	300.9	301.9	303.8	296.9	295.2	299.3	282.6
75%	183.0	204.5	203.0	206.1	210.2	216.8	218.5	223.4	227.2	226.6	225.2	242.3	252.1	260.1	262.9	248.9
Median	164.1	169.8	172.5	176.3	173.5	176.7	183.1	189.5	195.8	198.6	194.8	214.5	240.8	247.5	251.0	240.7

DRY BULB TEMPERATURE BINS (F)

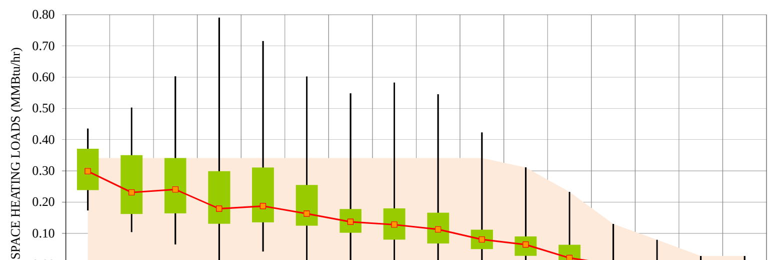
b) ABS. CHILLER LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Therm.	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
25%	0.4	0.4	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.6	0.6	0.6	0.7	0.7
Min	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.5	0.5	0.5	0.5	0.5	0.6	0.6
Max	0.5	0.6	0.6	0.7	0.6	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
75%	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.6	0.6	0.6	0.6	0.6	0.7	0.7	0.7	0.7
Median	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.6	0.6	0.6	0.7	0.7	0.7

DRY BULB TEMPERATURE BINS (F)

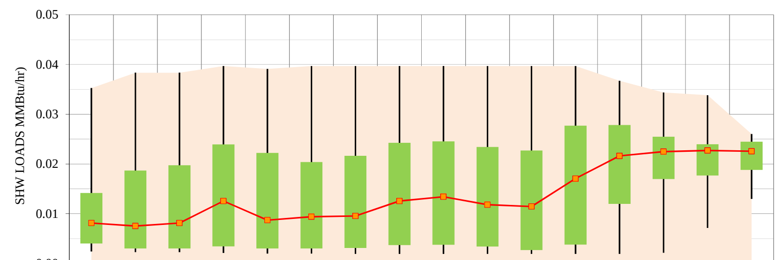
c) SPACE HEATING LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Therm.	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.2	0.1	0.1	0.0	0.0
25%	0.2	0.2	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Min	0.2	0.1	0.1	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Max	0.4	0.5	0.6	0.8	0.7	0.6	0.5	0.6	0.5	0.4	0.3	0.2	0.1	0.1	0.0	0.0
75%	0.4	0.4	0.3	0.3	0.3	0.3	0.2	0.2	0.2	0.1	0.1	0.1	0.0	0.0	0.0	0.0
Median	0.3	0.2	0.2	0.2	0.2	0.2	0.1	0.1	0.1	0.1	0.1	0.0	0.0	0.0	0.0	0.0

DRY BULB TEMPERATURE BINS (F)

d) SHW LOADS



	26-30	31-35	36-40	41-45	46-50	51-55	56-60	61-65	66-70	71-75	76-80	81-85	86-90	91-95	96-100	101-105
CHP Therm.	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
25%	0.004	0.003	0.003	0.003	0.003	0.003	0.003	0.004	0.004	0.003	0.003	0.004	0.012	0.017	0.018	0.019
Min	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.007	0.013
Max	0.035	0.038	0.038	0.040	0.039	0.040	0.040	0.040	0.040	0.040	0.040	0.040	0.037	0.034	0.034	0.026
75%	0.014	0.019	0.020	0.024	0.022	0.020	0.022	0.024	0.025	0.023	0.023	0.028	0.028	0.025	0.024	0.024
Median	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

DRY BULB TEMPERATURE BINS (F)

Figure E - 17: Temperature Bin Distribution of Electricity, Absorption Chiller, Space Heating and SHW Loads for Option 4

Figure E - 18: Typical Hourly Profiles for Electricity and Absorption Chiller Loads for Option 4

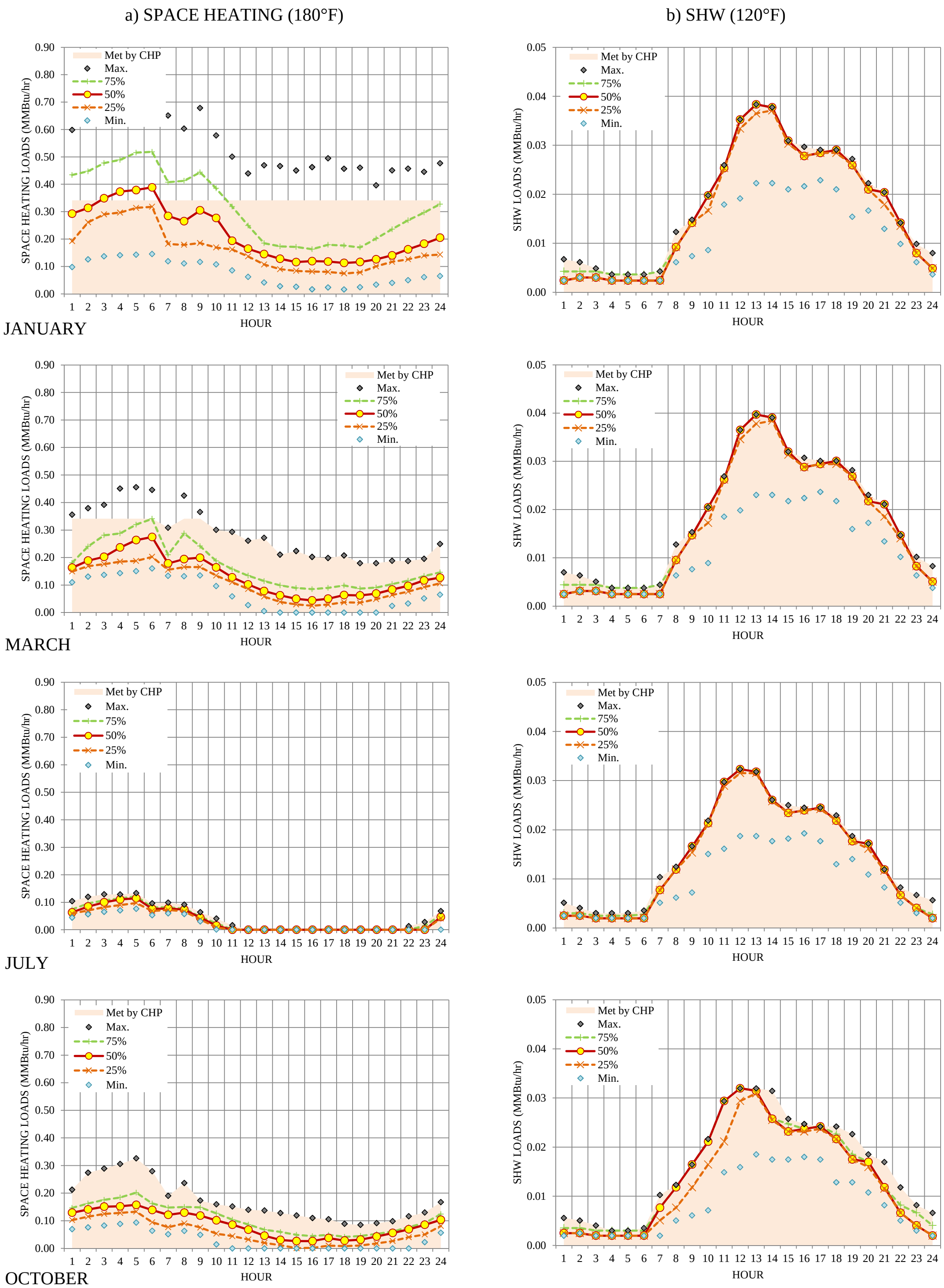


Figure E - 19: Typical Hourly Profiles for Space Heating and SHW Loads for Option 4

Figure E - 20: Typical Hourly Profiles for Surplus Electricity and Thermal Energy from Option 4

APPENDIX F

MANUFACTURERS SPECIFICATIONS FOR ABSORPTION REFRIGERATION SYSTEMS

This appendix provides manufacturers specifications for for LiBr/Water chiller and Water/NH_3 absorption chillers. Specifications for these chillers are obtained from manufacturers such as Broad, Carrier, Thermax, Yazaki, York, Trane, McQuay and Robur. The information includes the cooling capacity, inlet and outlet chilled water temperatures, rated COP and the amount of thermal energy required to operate the chillers. In addition, calculations that estimate auxiliary energy consumed by these chillers and part load performance curves used to model these chillers are also provided.

F – 1 Manufacturers Specifications for LiBr/Water Absorption Chillers

Table F-1: Manufacturers Specifications for Steam / Hot Water Fired LiBr/Water Absorption Chillers

Company	Product Name	Energy Source	Rated COP	CWT	Cooling Capacity	Link / Reference
Single-Effect Hot Water / Steam Fired						
Broad	Single Stage Steam Chiller	Steam < 30 psig	0.79	> 41°F	66 – 1,984 RT	http://www.broadusa.com/P_view.asp?pid=23
Carrier	16TJ Single-Effect Steam-Fired Absorption Chiller	Sat. Steam 15 psig	0.74	> 41°F	100 – 700 RT	http://www.docs.hvacpartners.com/idc/groups/public/documents/techlit/16tj-1pd.pdf
Thermax	SS 20A C - 80D C Single-Effect Steam-Fired	Sat. Steam 56.9 psig	0.75	44 – 54°F[1]	132 – 1,685 RT	http://www.trane.com/CPS/uploads/userfiles/chillers/absorption/steam_drivenabsorptionchillers.pdf
Yazaki	WFC-S Series Water Fired	Hot Water 158 – 203°F (Rated at 190.4)	0.7	44 – 54°F	10, 20, 30 RT	http://www.yazakienergy.com/docs/SB-WFCS-1009.pdf
York	1A1 – 14F3 IsoFlow	Sat. Steam 15 psig	0.71	44 – 54°F	120 – 1377 RT	http://m.master.ca/documents/Engineering_Guide_Absorption_Single_Stage_Engineering_Guide_PDF.pdf
Double-Effect Hot Water / Steam Fired						
Broad	Two-Stage Steam Chiller	Steam 45 – 130 psig	1.41	> 41°F	66 – 3,307 RT	http://www.broadusa.com/P_view.asp?pid=24
Carrier	16NK Double-Effect Steam-Fired Absorption Chiller	Sat. Steam 115 psig	1.5	44 °F	98 – 1,323 RT	http://83.138.143.21/images/uploads/summary_data/16NK_11-81_SummaryData.pdf
McQuay	TSA-NC Double-Effect Steam Fired	Sat. Steam 114 psig	1.2	44 – 54°F	100 – 1,500 RT	http://www.daikinmcquay.com/mcquaybiz/literature/lit_ch_wc/Catalogs/Pmabsorb.pdf
Thermax	Double-Effect Steam Fired SD 20A CX SD 80D CX	Sat. Steam 113.78 psig	1.6	44 – 54°F	111 – 1,685 RT	http://www.trane.com/CPS/uploads/userfiles/chillers/absorption/steam_drivenabsorptionchillers.pdf
Trane	Two-Stage Steam-Fired	Sat. Steam 120 psig	1.19 – 1,24	44 – 54°F	360 – 1,721 RT	http://www.trane.com/download/equipmentpdfs/absds4.pdf

[1] Entering – exiting chilled water temperature.

Table F-2: Manufacturers Specifications for Direct-Fired LiBr/Water Absorption Chillers

Company	Product	Energy Source	Rated COP	CWT	Cooling Capacity	Link / Reference
Double-Effect Direct-Fired Chillers						
Broad[2]	Two Stage Exhaust Chiller	Exhaust Inlet Temp. > 752 F	1.41	> 41°F	66 – 3,307 RT	http://www.broadusa.com/P_view.asp?pid=26
McQuay	TSA-DC	12,100 Btu/TR	1.00	44 – 54°F	100 – 1,500 RT	http://www.daikinmcquay.com/mcquaybiz/literature/lit_ch_wc/Catalogs/Pmabsorb.pdf
Thermax	GD 20A CX – GD70B CX	10,251 – 10,101 Btu/TR	1.20	44 – 54°F	111 – 1,107 RT	http://www.trane.com/CPS/uploads/userfiles/chillers/absorption/fuel_drivenabsorptionchillers.pdf
Trane	ABDL 100 - 1056	13,395 – 11,448 Btu/TR	1.07 – 1.14[3]	44 – 54°F	96 – 1,056 RT	http://www.trane.com/download/equipmentpdfs/absprc007en_r1.pdf
Yazaki	CH-K 30 - 100	11,760 Btu/TR	1.01 1.02	44.5 – 54.5°F	30 – 100 RT	http://www.yazakienergy.com/gasfiredspecifications.htm
York	YPC ParaFlow	2,400 – 8,100 Btu/hr x 10^3	1.0	44 – 54°F	200 – 675 RT	http://www.usair-eng.com/chillers/ypc-df.pdf

[3] COPs are reported for standard efficiency and high efficiency units using lower heating value (LHV) of the fuel.

F – 2 Manufacturers Specifications for Water/NH_3 Absorption Chillers

Table F-3: Manufacturers Specifications for Water/NH_3 Absorption Chillers

Company	Product	Energy Source	Rated COP	CWT	Cooling Capacity	Link / Reference
Single Effect, Direct Fired						
Robur	AYF	94,900 Btu/hr	0.64	Outlet: 45 F Inlet: 55 F	5 RT	http://www.roburcorp.com/documenti_prodotto/ROBUR_Pocket-Products-Guide_02-2010-20100216144013.pdf
	RTYF[4]	284,700 Btu/hr	0.64		15 RT	
	LB	94,900 Btu/hr	0.47	14F	4 RT	
	HT	94,900 Btu/hr	0.62	41F	5 RT	
	AYF	94,900 Btu/hr	0.64	37.4 F 113 F	5 RT	
	HR	94,900 Btu/hr	0.65	37.4 F 113 F	5 RT	
	ACF	94,900 Btu/hr	0.64	37.4 F 113 F	5 RT	
	RTCF	284,700 Btu/hr	0.64		15 RT	
	W LB	95,500 Btu/hr		23 F 113 F	4 RT	http://www.trane.com/CPS/uploads/userfiles/chillers/absorption/fuel_drivenabsorptionchillers.pdf

[4] Modular Series

F – 3 Calculating Electric Power for LiBr/Water and Water / NH_3 Absorption Refrigeration Systems

The following tables and equations presented in the following subsections of this appendix have been adopted from Dorgan et al. 1995. The values provided by Dorgan et al., are for indirect fired LiBr/Water absorption chillers. However, these values are assumed for direct fired as well as Water/NH3 absorption chillers. In addition, the electricity consumption for absorption chillers was calculated at AHRI conditions, which are different than the range of conditions under which absorption chillers operate. However, these numbers were considered by this study for assessing the performance of absorption chillers.

F – 3.1 Electric Energy Usage from the Absorption Machine

The electrical usage of the absorption machine include:

- Absorbent pump,
- Refrigerant pump,
- Purge pump, and
- Burner blower.

The absorption machine electrical energy use can be determined from the following equation:

$$AM_{kW} = K_1 \times AM_{cap}$$

Where:
AM_{kW} = Absorption machine electrical demand (kW)
K_1 = Value from Table (kW/ton)
AM_{cap} = Absorption machine nominal size (tons)

Table F-4: Absorption Machine Electrical Energy Usage (Source: Dorgan et al. 1995)

Machine Type	Machine Size (Tons)	Recommended Value (kW/ton)
Single-Effect	100 – 200	0.030
	200 – 400	0.022
	400 – 600	0.015
	600 – 1000	0.012
	1000 – 1600	0.012
Double-Effect	100 – 250	0.021
	250 – 450	0.020
	450 – 800	0.018
	800 – 1500	0.015

Notes:
1. The tons listed are for a nominal-sized machine at ARI conditions (44°F) chilled water, 85F cooling water, 12 psig steam single effect, and 115 psig steam double effect.
2. The kW/ton ranges are a compilation of all major manufacturers. The recommended kW/ton is a weighted average and a good approximation of average use.
3. The same numbers are assumed for both indirect fired and direct fired absorption chillers used in the analysis.

F – 3.2 Energy Usage from the Cooling Water Pumps

The condenser water pumps electrical energy use can be determined from the following equation:

$$CWP_{kW} = \frac{K_2 \times AM_{cap} \times C_2}{\eta_{CWP}}$$

Where:
CWP_{kW} = Cooling water pump electrical demand
K_2 = Value from Table bhp/ton
AM_{cap} = Absorption machine nominal size
C_2 = Conversion factor, 0.7457kW/hp
η_{CWP} = Cooling water pump motor efficiency 80%

Table F-5: Cooling Water Pump Sizing (Source: Dorgan et al. 1995)

Machine Type	Water Flow Rate (gpm/ton)	Cooling Water T. (°F)	Pump Sizing (bhp/ton)
Single-Stage	3.6 gpm/ton	10°F	0.12
		15°F	0.09
		20°F	0.06
		25°F	0.05
		30°F	0.04
		35°F	0.04
Two-Staged	4.0 gpm/ton	10°F	0.10
		15°F	0.06
		20°F	0.05
		25°F	0.04

Notes:
1. Water flow rate is calculated at ARI Standard 560 (AHRI 2000) Rating conditions.
2. bhp/ton ratings of the cooling water pumps are based on a system with a 60 foot head and a motor that is approximately 80% efficient.
3. The same numbers are assumed for both indirect fired and direct fired absorption chillers used in the analysis.

F – 3.3 Energy Usage from the Cooling Tower Fans

The cooling tower fan electrical energy use can be determined from the following equation:

$$CTF_{kW} = \frac{K_3 \times AM_{cap} \times C_2 \times K_4}{\eta_{CTF}}$$

Where:
CTF_{kW} = Cooling tower fan electrical demand
K_3 = Value from Table F-6 (bhp/ton)
AM_{cap} = Absorption machine nominal size (ton)
C_2 = Conversion factor, 0.7457 kW/hp
K_3 = Cooling tower fan partial use, (0.4)
η_{CWP} = Cooling tower fan efficiency 80%

Table F-6: Cooling Tower Fan Sizing (Source: Dorgan et al. 1995)

Cond. Water Temp. Diff.	Type of Fan	Single Effect Indirect Fired bhp/ton	Double Effect Indirect Fired bhp/ton
10 F	Propeller Centrifugal	0.13 0.27	0.11 0.23
15 F	Propeller Centrifugal	0.11 0.23	0.09 0.19
20 F	Propeller Centrifugal	0.09 0.19	0.08 0.17

Notes:
1. The bhp/ton are per design ton of cooling output of the absorption machine and are based on ARI conditions.
2. Numbers in this table are based on the average of manufacturers published ratings.

F – 4 Part-Load Performance Curves for Absorption Refrigeration Systems (Source: Winkelmann et al. 1993)

F – 4.1 For Single-staged Indirect Fired LiBr / Water Absorption Chiller

Coefficients for HIR as function of part load ratio:

a = 0.098585

b = 0.583850

c = 0.560658

d = -0.243093

F – 4.2 For Two-staged Direct Fired LiBr / Water Absorption Chiller

Coefficients for HIR as function of part load ratio:

a = 0.13551150

b = 0.61798084

c = 0.24651277

APPENDIX G

MANUFACTURERS SPECIFICATIONS FOR IC ENGINES

This appendix provides manufacturers specifications for IC engines. Specifications are obtained from manufacturers such as Man, Cummins, Caterpillar and Waukesa. The information includes specifications for electrical power generated by the IC engine; fuel rates; exhaust flow rates and temperatures; jacket coolant flow rate and temperature range; heat rejected to the jacket water and oil cooler; and the compression ratio of the IC engine.

Table G-1: Manufacturer’s Specifications for IC Engines Operating on Natural Gas

Manf.	**Model**	**Power** (kW)	**Fuel Rate** (Btu/kW-hr)	**Exhaust Flow** (lb/s)	**Exhaust Temp.** (F)	**Est. Coolant Flow** (Jacket Only) (lbs/s)	**Est. Coolant Ext. Temp. Change** (F)	**Heat Rejection to Jacket Water + Oil Cooler** (Btu/hr)	**Comp. Ratio**	**Notes / References**
MAN	-	100	11,147	0.28	1060	6.85	210 - 195			Note 1
CUMMINS	CUM SCG300	300	9,440	0.84	1202	6.30	203-188	341,214	9.5	Note 2,3,4,8
	CUM SCG350	350	9,427	0.84	1202	7.40	203-188	399,220	9.5	Note 2,3,5,6,8
	CUM SCG400	400	9,426	0.94	1238	8.50	203-188	457,227	9.5	Note 2,3,5,6,8
	CUM SCG440	440	9,430	1.06	1247	9.20	203-188	498,172	9.5	Note 2,3,4,8
	CUM SCG599	500	9,424	1.43	1202	10.60	203-188	569,827	9.5	Note 2,3,5,6,8
	CUM SCG600	600	9,423	1.68	1220	12.60	203-188	682,428	9.5	-
CATERPILLAR	CAT G3516 LE	300	10,967	0.92	1067	12.80	210 - 195			-
	CAT G3412 TA	350	11,563	1.18	892	27.60	210 - 195	1,492,140	9.7	Note 2,5,7,9
	CAT G3508 LE 516GE01	360	10,948	1.38	804	23.00	210 - 195	1,241,340	8.0	Note 2,5,7,9
	CAT G3508 LE 516GEX2	375	9,979	1.23	738	21.00	201	1,134,840	11.0	Note 2,5,7,9
	CAT G3508 516GEX3	370	10,256	1.10	862	26.00	210 - 195	1,402,380	9.0	Note 2,5,7,9
	CAT 516GE01	360	10,948	1.38	804	23.0	210 - 195	1,241,340	8.0	-
	CAT 516GEX2	375	9,979	1.23	738	21.0	201	1,134,840	11.0	-
	CAT 516GEX3	370	10,256	1.10	862	26.0	210 - 195	1,402,380	9.0	-

Notes:
1. http://www.man-mec.com/en/industrial_engines/Diesel_Power_Generation.jsp
2. For heat rejection, numbers for radiator and oil cooler heat are considered.
3. The exhaust temperature has a variation of ± 60 F.
4. Naturally aspirated.
5. With turbocharger
6. With intercooler
7. Aftercooled
8. http://www.sino-cummins.com/catalogue/cummins-natural-gas-generator-set.htm
9. http://www.cat.com/power-generation/generator-sets/gas-generator-sets
10. http://www.dresserwaukesha.com/index.cfm/go/list-prodsubline/productline/power-generation-category/

Table G-1: Manufacturer's Specifications for IC Engines Operating on Natural Gas (Continued)

Manf.	Model	Power (kW)	Fuel Rate (Btu/kW-hr)	Exhaust Flow (lb/s)	Exhaust Temp. (F)	Est. Coolant Flow (Jacket Only) (lbs/s)	Est. Coolant Ext. Temp. Change (F)	Heat Rejection to Jacket Water + Oil Cooler (Btu/hr)	Comp. Ratio	Notes / References
CATERPILLAR	CAT G3512	570	10,376	2.08	802	39.5	210-195	2,131,980	8.0	-
		570	10,312	2.01	730	37.3	210-195	2,013,300	11.0	-
	CAT 516GE01	555	10,516	1.52	869	45.9	210-195	2,479,920	9.0	-
	CAT 516GEX2	375	9,979	1.23	738	21.0	201	1,134,840	11	-
	CAT 516GEX3	370	10,256	1.10	862	26.0	210 - 195	1,402,380	9	-
	CAT G3512	570	10,376	2.08	802	39.5	210-195	2,131,980	8	-
		570	10,312	2.01	730	37.3	210-195	2,013,300	11	-
		555	10,516	1.52	869	45.9	210-195	2,479,920	9	-
	CAT G3516 LE	770	10,168	2.84	842	42.6	210-195	2,301,720	8	-
		770	9,807	2.64	781	38.0	210-195	2,050,680	11	-
		750	10,377	2.03	864	56.5	210-195	3,050,820	9	-
		800	10,246	3	869	20.2	210-195			-
WAUKESHA	VGF18GL/GLD	310	9,796	1.13	839	16.3	190-180	880,332	11	Note 2,6,10
	VGF24GSID	375	10,409	0.96	1114	26.3	190-180	1,422,862	8.6	Note 2,6,10
	VGF24GL/GLD	415	9,718	1.50	842	21.7	190-180	1,173,776	11	Note 2,6,10
	VGF24GSID	560	10,370	1.43	1116	39.6	190-180	2,136,000	8.6	Note 2,6,10
	VHP5900G	595	10,391	1.53	1044	44.5	180-175	2,402,147	0:00	Note 2,6,10
	VGF36GL/GLD	620	9,780	2.25	841	32.7	190-180	1,764,076	0:00	Note 2,6,10
	VGF48GSID	750	10,286	1.91	1113	52.3	190-180	2,825,252	8.6	Note 2,6,10

Notes:
1. http://www.man-mec.com/en/industrial_engines/Diesel_Power_Generation.jsp
2. For heat rejection, numbers for radiator and oil cooler heat are considered.
3. The exhaust temperature has a variation of ± 60 F.
4. Naturally aspirated.
5. With turbocharger
6. With intercooler
7. Aftercooled
8. http://www.sino-cummins.com/catalogue/cummins-natural-gas-generator-set.htm
9. http://www.cat.com/power-generation/generator-sets/gas-generator-sets
10. http://www.dresserwaukesha.com/index.cfm/go/list-prodsubline/productline/power-generation-category/

APPENDIX H

SUMMARY OF COSTS FOR THE FOUR CHP OPTIONS

This appendix provides the summary of costs for the four CHP options installed in the grocery store and the corresponding base-case scenarios. The costs of the base-case scenarios include the capital and maintenance costs for the equipment that is replaced and modified upon the installation the CHP option in the grocery store. The costs for CHP options include capital and maintenance costs of the IC engine and the thermal energy recovery and distribution equipment such as absorption chillers, heat exchangers, piping and pumping. The costs also include capital and maintenance costs for thermal energy recovery and distribution equipment used for residential buildings such as water-to-water heat exchangers, thermal storage tanks as well as equipment required for hot water loop piping and pumping.

H – 1 Summary of Costs for Option 1

Table H-1: Base-Case Maintenance Costs of Replaced / Modified Equipment for Option 1

Base-Case Maintenance Costs of Replaced / Modified Equipment			
Item	**Unit Cost**	**Unit**	**Ref. /Notes**
VC Sub-Cooler	$133	$/Ref. ton/year	RSMeans 2009
Air-cooled Heat Rejection	$4	$/ton/year	RSMeans 2009

Table H-2: Base-Case Capital Costs of Replaced / Modified Equipment for Option 1

Base-Case Capital Cost of Replaced / Modified Equipment						
Item		**Specs.**	**Unit Cost**	**Unit**	**Total Cost**	**Ref. /Notes**
For Grocery Store						
Vapor Compression Refrigeration System	Equipment + Installation	15 Tons	$350	$/ton	$5,250	MCHPAC 2004
Air-cooled Heat Rejection		19 Tons (Total Heat Rejected)[1]	$6,250	$./unit	$6,250	RSMeans 2012
Total					$11,500	

Note:
1. Heat rejection factor of 1.25 was assumed to calculate the total heat rejected for the air-cooled condensers (Carrier 2005).

Table H-3: Capital Costs for Option 1

Capital Cost of Equipment for Option 1							
Item		**Specs.**	**Unit Cost**	**Unit**	**First Cost**	**Total Cost**	**Ref. /Notes**
For Grocery Store							
Reciprocating Engine	Equipment + Installation	300 kW	$1,400	$/kW	$420,000	$420,000	MCHPAC 2004 Costs include appropriate controls, average sized HEX
Refrigeration Equipment							
Abs. Chiller (LiBr/Water) Single-Effect, Hot Water	Equipment + Installation	15 Tons	$1,000	$/ton	$25,000		RSMeans 2012
Heat Rejection Cooling Tower	Equipment + Installation	38 Tons[1]	$230	$/ton	$8,740		RSMeans 2012 Estimate based on 50 ton, axial fan induced draft cooling tower; Pumps & pipes excluded.
Cooling Tower Pumping	Equipment + Installation	38 Tons	$100	$/ton	$3,800		Trane 2007
Sub-Total						$37,540	
Space Heating & SHW Equipment							
Water-to-water HEX	Equipment + Installation	20 GPM	$155	$/GPM	$5,115		RSMeans 2012
Hot Water Loop Piping & Pumping	Equipment + Installation				$50,000		Assumption
Sub-Total						$55,115	
Eng. & Const.	10% of the total cost of grocery store equipment					$51,266	Assumption
Other	5% of the total cost of grocery store equipment					$25,633	Assumption
For Residential Buildings							
Water-to-water HEX	Equipment + Installation	40 GPM	$155	$/GPM	$6,200		RSMeans 2012
Thermal Storage Tank	Equipment + Installation	8,500 Gallons[2] (For 41 MF Buildings)	$1	$/Gallon	$6,375		De Wit 2007
Heat Rejection Equipment	Equipment + Installation	55 Tons	$294	$/Ton	$16,170		RSMeans 2012
Hot Water Loop Piping & Pumping	Equipment + Installation	328 MF Units[4] (For 41 MF Buildings)	$900	$/dwelling unit	$295,200		Boulter 2012 RSMeans 2012
Sub-Total						$323,945	
Total						**$913,498**	

Notes:
1. Tonnage for heat rejection units for absorption chillers are calculated by assuming 30,600 Btu/hr of heat rejected per refrigeration ton of a single-effect absorption chiller (Dorgan et al. 1995).
2. Thermal storage tank sized to store 25% of the thermal energy that is available throughout the year. This is done to avoid oversizing the tank during summer months when thermal energy available from the grocery store is not fully utilized by the residential units.
4. Number of multi-family units calculated using annual energy consumption.

H – 2 Summary of Costs for Option 2

Table H-4: Base-Case Maintenance Costs of Replaced / Modified Equipment for Option 2

Base-Case Maintenance Costs of Replaced / Modified Equipment			
Item	**Unit Cost**	**Unit**	**Ref. /Notes**
VC Sub-Cooler	$133	$/Ref. ton/year	RSMeans 2009
Air-cooled Heat Rejection	$4	$/ton/year	RSMeans 2009
Packaged space Cooling and Heating Units	$148	$/ton/year	RSMeans 2009

Table H-5: Base-Case Capital Costs of Replaced / Modified Equipment for Option 2

Base-Case Capital Cost of Replaced / Modified Equipment						
Item		**Specs.**	**Unit Cost**	**Unit**	**First Cost**	**Ref. /Notes**
For Grocery Store						
Packaged Space Cooling Units w/ Gas Furnace	Equipment + Installation	129 Tons	$788	$/ton	$101,588	RSMeans 2012
Sub-Cooler	Equipment + Installation	15 Tons	$350	$/ton	$5,250	MCHPAC 2004
Air-cooled Heat Rejection	Equipment + Installation	19 Tons [1]	$6,250	$./unit	$6,250	RSMeans 2012
Total					**$113,088**	

Table H-6: Capital Costs for Option 2

Capital Cost of Equipment							
Item		**Specs.**	**Unit Cost**	**Unit**	**First Cost**	**Total Cost**	**Ref. /Notes**
For Grocery Store							
Reciprocating Engine	Equipment + Installation	300kW	$1,400	$/kW	$420,000	$420,000	MCHPAC 2004
Refrigeration Equipment							
Abs. Chiller (LiBr/Water) Double-Effect, Dir. Fired	Equipment + Installation	140 Tons	$1,200	$/ton	$170,400		RSMeans 2012
Aux. Burner	Equipment + Installation	0.506 MBH			$2,245		RSMeans 2012
Heat Rejection Cooling Tower	Equipment + Installation	357 Tons	$139	$/ton	$49,623		RSMeans 2012
Cooling Tower Pumping	Equipment + Installation		$100	$/ton	$35,700		TRANE 2007
Sub-Total						$257,968	
Space Heating and SHW Equipment							
Water-to-water HEX	Equipment + Installation	20 GPM	$155	$/GPM	$3,100		RSMeans 2012
Hot Water & Chilled Water Piping + Pumps	Equipment + Installation				$75,000		
Sub-Total						$78,100	
Eng. & Const.	10% of the total cost of grocery store equipment					$75,607	Assumption
Other	5% of the total cost of grocery store equipment					$37,803	Assumption
For Residential Buildings							
Water-to-water HEX	Equipment + Installation	34 GPM	$155	$/GPM	$5,270		RSMeans 2012
Thermal Storage Tank	Equipment + Installation	33,900 Gallons (36 MF Buildings)	$1	$/Gallon	$25,425		De Wit 2007
Heat Rejection Equipment	Equipment + Installation	55 Tons	$294	$/Ton	$16,170		
Hot Water Loop Piping & Pumping	Equipment + Installation	288 MF Units (36 MF Buildings)	$900	$/Dwelling Unit	$259,200		Boulter 2012
Sub-Total						$306,065	
Total						**$1,175,543**	

H – 3 Summary of Costs for Option 3

Table H-7: Base-Case Maintenance Costs of Replaced / Modified Equipment for Option 3

Base-Case Maintenance Costs of Replaced / Modified Equipment			
Item	**Unit Cost**	**Unit**	**Ref. /Notes**
VC Refrigeration System	$133	$/Ref. ton/year	RSMeans 2009
Air-cooled Heat Rejection	$4	$/ton/year	RSMeans 2009

Table H-8: Base-Case Capital Costs of Replaced / Modified Equipment for Option 3

Base-Case Capital Cost of Replaced / Modified Equipment						
Item		**Specs.**	**Unit Cost**	**Unit**	**First Cost**	**Ref. /Notes**
For Grocery Store						
Vapor Compression Refrigeration System	Equipment + Installation	45 Tons	$800	$/ton	$36,000	Extrapolated from Goetzler et al. 2009
Sub-Cooler	Equipment + Installation	5 Tons	$350	$/ton	$1,750	MCHPAC 2004
Air-cooled Heat Rejection		63 Tons	$294	$./ ton	$18,499	RSMeans 2009
Total					**$56,249**	

Table H-9: Capital Costs for Option 3

Capital Cost of Equipment							
Item		**Specs.**	**Unit Cost**	**Unit**	**First Cost**	**Total Cost**	**Ref. /Notes**
For Grocery Store							
Reciprocating Engine	Equipment + Installation	300 kW	$1,400	$/kW	$420,000	$420,000	MCHPAC 2004.
Refrigeration Equipment							
Abs.Chillers (Water/NH_3) Single-Effect, Dir. Fired	Equipment + Installation	50 Tons	$1,600	$/ton	$80,000		MCHPAC 2004.
Aux. Burner	Equipment + Installation	290 MBH	$4,640	$/Unit	$4,640		RSMeans 2011
Heat Rejection Cooling Tower	Equipment + Installation	128 Tons	$181	$/ton	$23,168		RSMeans 2012
Cooling Tower Piping & Pumping	Equipment + Installation	128 Tons	$250	$/ton	$32,000		TRANE 2007
Sub-Total						$139,808	
Space Heating and SHW Equipment							
Water-to-water HEX	Equipment + Installation	20 GPM	$155	$/GPM	$3,100		RSMeans 2012
Hot Water Piping + Pumps	Equipment + Installation				$50,000		
Aux. Boiler	Equipment + Installation	350 MBH	$14	$/MBH	$4,953		RSMeans 2012
Sub-Total						$58,053	
Eng. & Const.	10% of the total cost of grocery store equipment					$19,786	Caton (2010)
Other	5% of the total cost of grocery store equipment					$9,893	Caton (2010)
For Residential Buildings							
Water-to-water HEX	Equipment + Installation	13 GPM	$155	$/GPM	$2,015		
Thermal Storage Tank	Equipment + Installation	3,800 Gallons (For 14 MF Units)	$1	$/Gallon	$2,888		De Wit 2007
Heat Rejection Equipment	Equipment + Installation	22 Tons	$294	$/Ton	$6,468		
Hot Water Loop Piping & Pumping	Equipment + Installation	112 MF Units (14 MF Buildings)	$900	$/Dwelling Unit	$100,800		Boulter 2012
Sub-Total						$112,171	
Total						**$759,711**	

H – 4 Summary of Costs for Option 4

Table H-10: Base-Case Maintenance Costs of Replaced / Modified Equipment for Option 4

Base-Case Maintenance Costs of Replaced / Modified Equipment			
Item	**Unit Cost**	**Unit**	**Ref. /Notes**
VC Refrigeration System	$133	$/Ref. ton/year	RSMeans 2009
Air-cooled Heat Rejection	$4	$/ton/year	RSMeans 2009

Table H-11: Base-Case Capital Costs of Replaced / Modified Equipment for Option 4

Base-Case Capital Cost of Replaced / Modified Equipment						
Item		**Specs.**	**Unit Cost**	**Unit**	**First Cost**	**Ref. /Notes**
For Grocery Store						
Vapor Compression Refrigeration System	Equipment + Installation	62 Tons	$1,000	$/ton	$62,000	Extrapolated from Goetzler et al. 2009
Air-cooled Heat Rejection	Equipment + Installation	77.5 Tons	$294	$/unit	$22,757	RSMeans 2012
Total					**$84,757**	

Table H-12: Capital Costs for Option 4

Capital Cost of Equipment							
Item		**Specs.**	**Unit Cost**	**Unit**	**First Cost**	**Total Cost**	**Ref. /Notes**
For Grocery Store							
Reciprocating Engine	Equipment + Installation	300 kW	1,400	$/kW	$420,000	$420,000	MCHPAC 2004
Refrigeration Equipment							
Abs.Chillers (Water/NH_3) Single-Effect, Dir. Fired	Equipment + Installation	62 Tons	2,000	$/ton	$124,000		MCHPAC 2004
Aux. Burner	Equipment + Installation	860 MBH	$4,640	$/Unit	$4,640		RSMeans 2012
Heat Rejection Cooling Tower	Equipment + Installation	158 Tons	144	$/ton	$22,752		RSMeans 2012
Cooling Tower Pumping	Equipment + Installation		100	$/ton	$0		TRANE 2007
Sub-Total						$151,392	
Space Heating & SHW Equipment							
Water-to-water HEX	Equipment + Installation	20 GPM	$155	$/GPM	$3,100		RSMeans 2012
Hot Water Piping + Pumps	Equipment + Installation				$50,000		
Boiler	Equipment + Installation	500 MBH	$14	$/MBH	$7,075		RSMeans 2012
Sub-Total						$60,175	
Eng. & Const.	10% of the total cost of grocery store equipment					$63,157	Assumption
Other	5% of the total cost of grocery store equipment					$31,578	Assumption
For Residential Buildings							
Water-to-water HEX	Equipment + Installation	15 GPM	$155	$/Unit	$2,325		
Thermal Storage Tank	Equipment + Installation	5,500 Gallons (For 14 MF Units)	$1	$/Gallon	$4,180		De Wit 2007
Heat Rejection Equipment	Equipment + Installation	22 Tons	$294	$/Ton	$6,468		
Hot Water Loop Piping & Pumping	Equipment	112 MF Units (14 MF Buildings)	$900	$/Dwelling Unit	$100,800		Boulter 2012
Sub-Total						$113,773	
Total						**$840,075**	