CHAPTER 7. ENERGY USE CHARACTERIZATION

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CHAPTER 7. ENERGY USE CHARACTERIZATION

7.1 INTRODUCTION

The engineering analysis provides estimates of annual energy consumption for commercial refrigeration equipment (CRE), which are used in the subsequent life-cycle cost (LCC) and payback period (PBP) analyses (Chapter 8 of the TSD) and national impacts analyses (Chapter 11 of the TSD). The U.S. Department of Energy (DOE) estimated the energy consumption for the 15 equipment classes analyzed in the engineering analysis (Chapter 5 of the TSD). For the NOPR analyses, these energy consumption estimates were then validated with annual simulation modeling of selected equipment classes and efficiency levels. There were no comments on the validation. DOE determined that it would use the energy consumption estimates developed in the engineering analysis. For the final rule, DOE updated the engineering analysis, but did not redo the annual simulation modeling discussed in Chapter 7. As a consequence, the information which follows reflects that developed for the NOPR publication and the engineering and energy analyses provided for that document. The energy use characterization presented in Chapter 7 was not used in the final rule analysis and is presented here for information only.

7.2 OVERVIEW OF ENERGY USE FOR COMMERCIAL REFRIGERATION EQUIPMENT

American National Standards Institute (ANSI)/Air-Conditioning and Refrigeration Institute (ARI) Standard 1200-2006 (ARI 1200-2006),[1] which references ANSI/American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) Standard 72-2005,[2] is an industry-developed test procedure for determining the energy consumption of commercial refrigeration equipment. ARI 1200-2006 provides a method for estimating the daily energy consumption for a display case under steady-state conditions. The use of daily energy consumption is an appropriate approach for commercial refrigeration equipment for several reasons. First, because the useful output of commercial refrigeration equipment is the ability to display a product while maintaining that product at an adequate temperature, there is no easily quantifiable output metric (such as British thermal units (Btu) of cooling provided). Second, commercial refrigeration equipment typically provide both cold storage and lighting of product with, varying, but often on the same order of magnitude, energy consumption. DOE’s need for a single metric to describe both makes defining a useful output metric difficult.

DOE used ARI 1200-2006 as the basis for the engineering analysis calculations. ARI 1200-2006 treats remote condensing and self-contained commercial refrigeration equipment differently. In the case of remote condensing equipment, the test procedure measures the energy use of each component (e.g., fans and lights) as well as the total refrigeration load of the equipment. The total refrigeration load is used to calculate compressor energy consumption based on a standardized relationship of evaporator temperature and compressor energy efficiency ratio. In the case of self-contained equipment, the test procedure measures the total energy use of the equipment as a whole, including both component energy use and compressor energy use. The resulting daily energy consumption estimate is either calculated daily energy consumption (CDEC) for remote condensing equipment or total daily energy consumption (TDEC) for self-contained equipment. Both metrics represent the sum of compressor energy consumption and direct energy consumption used by evaporator fan motors, lighting, anti-sweat heaters, defrost and drain heaters, and condensate evaporator pan heaters. This sum is represented in Eqs. 7.1 and 7.2; however, it is only calculated from individual components for remote condensing equipment. For self-contained equipment, the TDEC is based on the measured total power draw.

$$CDEC = CEC + FEC + LEC + AEC + DEC + PEC \qquad \textbf{Eq. 7.1}$$
$$TDEC = CEC + FEC + LEC + AEC + DEC + PEC \qquad \textbf{Eq. 7.2}$$

where

$CDEC$ = Calculated daily energy consumption for remote condensing equipment in kilowatt hours per foot per day (kWh/ft per day),
$TDEC$ = Total daily energy consumption for self-contained equipment (kWh/ft per day),
CEC = Compressor energy consumption (kWh/ft per day),
FEC = Fan energy consumption (kWh/ft per day),
LEC = Lighting energy consumption (kWh/ft per day),
AEC = Anti-sweat heater energy consumption (kWh/ft per day),
DEC = Defrost and drain heater energy consumption (kWh/ft per day),
PEC = Condensate evaporator pan energy consumption (kWh/ft per day)

The primary difference between the *CDEC* and *TDEC* metrics is the way compressor energy consumption is determined. In the *CDEC* metric, the cooling load (in Btu/hour) is measured, and the compressor energy consumption is calculated using a compressor energy efficiency ratio (EER) table provided in ARI 1200-2006. In the *TDEC* metric, the compressor energy consumption is directly measured as a power draw during testing.

7.2.1 Direct Energy Consumption

Direct electric energy consumption is the total energy consumption of evaporator fan motors, lighting, anti-sweat heaters, defrost and drain heaters, and condensate pan heaters.

7.2.1.1 Lighting Energy Consumption

Commercial refrigeration equipment is generally outfitted with lighting to illuminate the products it refrigerates. For most equipment, lighting represents a significant fraction of the total energy consumption. Lighting configuration can vary significantly for different types of cases. Lamps and ballasts can be located inside the refrigerated space and/or outside the refrigerated space. For vertical open (VOP) type of refrigerated equipment[a], standard lighting consists of 1 or 2 rows of lamps located in the top canopy. Additional lighting is sometimes placed along the lower rail of the case and on the bottom of each shelf to illuminate the shelf below it. Lighting in

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[a] Equipment class designations consist of a combination (in sequential order separated by periods) of a equipment family code (VOP=vertical open, SVO=semivertical open, HZO=horizontal open, VCT=vertical transparent doors, VCS=vertical solid doors, HCT=horizontal transparent doors, HCS=horizontal solid doors, or SOC=service over counter), an operating mode code (RC=remote condensing or SC=self-contained), and a rating temperature code (M=medium temperature (38°F), L=low temperature (0°F), or I=ice-cream temperature (-15°F)). For example, "VOP.RC.M refers to the "vertical open, remote condensing, medium temperature" equipment class. See Chapter 3, market and technology assessment, for a more detailed explanation of the equipment class terminology

frozen food reach-in cases is located in the vertical mullions between the doors. Cases with solid doors usually have no lighting.[3]

As described in the engineering analysis (Chapter 5), there are three types of lighting technologies (T-8, T-8S and LED) for the horizontal configuration in the VOP, semi-vertical open (SVO), horizontal open (HZO), and service over counter (SOC) equipment families, and two types of lighting technologies (T-8 and LED) for the vertical configuration in the vertical closed transparent (VCT) equipment family that were considered by DOE. Each type of technology was implemented in a way to provide similar lighting of products in a display case, but with different levels of electric power consumption.

7.2.1.2 Evaporator Fan Motors

Evaporator fans are used to circulate chilled air through the refrigerated compartments of commercial refrigeration equipment. Most fan motors used in commercial refrigeration applications are inexpensive and inefficient single-phase shaded pole motors (SPM). The efficiency of permanent split capacitor (PSC) or electronically commutated motors (ECM) is significantly better. Estimates of efficiency and power requirements for these three motor types are compared in Chapter 5.

7.2.1.3 Anti-Sweat Heaters and Condensate Pan Heaters

In some equipment, anti-sweat heaters are energized continuously and no anti-sweat control is provided. This results in a steady contribution of heat to the cooling load (and thus the CEC) and of energy to the direct energy consumption of the refrigeration equipment. Retrofit of conventional glass doors with high performance doors can reduce the power requirement of heaters for low-temperature cases and eliminate the anti-sweat heaters for medium-temperature cases. Power estimates for two levels of door types, thermal performance and corresponding anti-sweat heaters used, on the equipment covered in this rulemaking are described in Chapter 5.

Condensate pan heaters are installed in commercial refrigeration equipment to evaporate defrost-melt water. These heaters are usually used in self-contained equipment because self-contained units are designed to be mobile and used where there is no drain available.

7.2.1.4 Defrost and Drain Heaters

Off-cycle defrost is commonly used for medium temperature equipment, while electric defrost or hot gas defrost is used for low temperature equipment. The number of defrost cycles required for a display case is dependent upon its type and design. For remote condensing equipment, a drain heater is usually installed to heat the drain pipe during defrost to prevent freezing. As discussed in Chapter 5, hot-gas defrost, defrost cycle control, and defrost mechanism were not considered as design options. Power requirements and run time of defrost and drain heaters are provided in Table B.2.2 of Appendix B, and these values are used in the energy modeling.

7.2.2 Refrigeration Load and Compressor Energy Consumption

Lighting, evaporator fan motors, anti-sweat heaters, and defrost and drain heaters contribute to the refrigeration load of the display case as well as directly consuming electricity. To maintain the temperature of refrigerated products, the compressor and condenser (collectively, the condensing unit) of the refrigeration system needs to remove heat from display cases. The amount of heat that must be removed from a given display case is called the total refrigeration load. In addition to the load contributed by direct electric use equipment, the display case refrigeration load also consists of heat conduction through case panels and walls, thermal radiation from the ambient to the refrigerated product and display case interior, and air infiltration through the open front of the case.

The characteristics of refrigeration load components are different for each equipment class. The VOP family has a large portion (roughly 80 percent) of the refrigeration load attributable to ambient air entrainment through the display case air curtain.[3] The SVO and HZO families have a significant load component that is due to radiation exchange with the surrounding environment. The refrigeration load of the VCT and HCT families is significantly influenced by the use of anti-sweat heaters. DOE made necessary assumptions to estimate the refrigeration load in terms of radiation, conduction and infiltration loads as well as refrigeration load due to direct energy consumption based on the rating conditions. Refer to Chapter 5, Engineering Analysis, for detailed models and assumptions.

The electricity consumed by the condensing unit to remove the heat from the case can be determined by the total refrigeration load of the case and compressor EER. However, there are five issues regarding the accuracy of using ARI 1200-2006 to estimate case refrigeration load and compressor energy consumption that should be noted.

(1) In the engineering analysis, lighting is assumed to be on for 24 hours for all cases that are equipped with lighting. In practice, the lighting may be running for less time depending on store operation hours and lighting control practices, which may cause the case load to be less than the refrigeration load determined from the test procedure.

(2) The relative humidity in the test chamber is maintained at 55 percent and this value influences the case infiltration load for the test. In reality, the relative humidity of store space will fluctuate, which results in the actual cooling load deviating from the estimated load. This is particularly important for open display cases and VCT cases with anti-sweat heaters.

(3) For remote condensing equipment, the condensing unit must be capable of removing the total refrigeration load from the cases and rejecting it to the outdoor environment via the condensers. ARI 1200-2006 uses a refrigeration performance map (Table 1 in ARI 1200-2006 provides EER as a function of suction temperature) to calculate the compressor energy consumption. This map assumes a constant EER for the condensing unit for a given suction temperature. A single condensing temperature rating point of 90°F was assumed in ARI 1200-2006 for the condensing unit.[4] In

practice, the EER of the condensing unit will vary with each installation and system design as well as with climate.

(4) For remote condensing systems, there are additional system heat gains and other system effects that may result in additional "superheat load" on the compressor. These result from the difference in temperature between the vapor leaving the evaporator and the vapor entering the compressor. This additional load is due to heat gain in the long refrigerant piping runs typically found in a supermarket or grocery store and is not accounted for in ARI 1200-2006. However, a 65°F return-gas temperature is implied as the basis of ARI 1200-2006 Table 1 values, which are derived from ARI ratings of specific compressors according to ARI Standard 540-2004.[5] ARI Standard 540-2004 assumes a 65°F return-gas temperature for all external drive and accessible-hermetic type compressors. This return-gas temperature is higher than the temperature of refrigerant vapor leaving a typical display case (which is typically only 5°F to 10°F higher than the saturated suction temperature – as controlled by the expansion device). The ARI 1200-2006 test procedure was intended to provide relative comparisons of energy use for different cases. It was not intended to define the actual energy consumption for any case. Implicit in the ARI assumption of using the compressor rated coefficient of performance (COP) is that this additional superheat load from the piping is not attributed to the case. For instance, this might be true if the additional superheat load would be unaffected by improvements to the case, treating it effectively as a constant load (for cases of the same suction temperature), and therefore the energy needed to satisfy this load (additional superheat load divided by total COP) would also be constant. In comparing any two cases, the tacit assumption is that the energy savings from a reduction in case load does not need to take into account the "additional superheat load." Practically, additional superheat load needed to get to the 65°F return-gas temperature implicit in the compressor efficiencies is not a constant. If one assumes the same temperatures for refrigerant leaving the case and entering the compressor, but reduced refrigerant mass flow rate resulting from reduced case load, the additional superheat load must vary in proportion to the mass flow rate change. This reduced additional superheat load for more efficient cases apparently reduces the compressor energy consumption as well, which is not accounted for in ARI Standard 1200-2006. Annual energy simulation accounting for this "additional superheat load" would give some insights into this issue.

(5) Heat-recovery systems are commonly used for water heating and supplemental space-heating in supermarkets. These systems transfer heat from the condenser to the building heating, ventilating and air-conditioning (HVAC) and/or water-heating system. Heat recovered in this way is "free" in the sense that it would otherwise be rejected to the outdoor environment. The implementations of lower energy consumption display cases in a supermarket refrigeration system can reduce the amount of heat rejected to the condensing unit, and would, therefore, reduce the amount of heat available for recovery.

7.3 ENERGY ANALYSIS APPROACHES

Several options were considered for providing estimates of CRE energy consumption. These options included: using the ARI 1200-2006 test procedure; using an existing simulation program that would analyze display case and compressor energy use on an annual basis; and using a whole-building simulation which would analyze case, compressor, and HVAC impacts. DOE's approach for the advance notice of proposed rulemaking (ANOPR) analysis was to use the engineering analysis estimates of energy consumption directly in the LCC analysis, but to also run selected design options for the equipment through whole-building simulation software, which included the simulation of refrigeration systems, to provide a check on the engineering analysis estimates of the annual energy consumption of the equipment.

Whole-building simulation with whole-building energy impact analyzed was the option first considered by DOE and discussed during DOE's Framework public meeting. During that meeting, several stakeholders commented that a whole-building analysis is the desired approach,[b] others were concerned about the additional difficulty and complexity of the analysis.[c] DOE decided to use energy consumption estimates developed from the engineering analysis and to validate the estimates with whole-building simulation of supermarkets which included simulation of the refrigeration system. Reasons for adopting this approach included:

1. The energy consumption ratings provided by ARI 1200-2006 do not distinguish between energy consumption by the compressor (which may vary as a function of environmental conditions) and energy consumption by other components in the case (e.g., lighting) which do not vary as a function of environmental conditions. These two types of energy consumption are roughly similar in magnitude, and it is difficult to assess where the energy savings are coming from or what the impact on a building HVAC load might be.

2. The initial engineering analysis (see Chapter 5 of the TSD) did not suggest design options that would provide significant changes to the building load relative to the commercial refrigeration system energy consumption.

3. The net interaction between the refrigeration system and HVAC energy consumption is a function of the variation in HVAC designs. HVAC system designs for food sales buildings, like supermarkets, may incorporate such features as separate dehumidification and refrigerant condenser reheat systems, which make assessing overall HVAC impact complicated. Also, detailed data on the relative prevalence of different HVAC system designs incorporating these features is not readily available.

4. The interaction between the refrigeration and overall HVAC energy consumption is a function of the ratio of the total heat removed from the space by the display cases relative to the other internal loads (lighting, occupancy, and plug load) and external loads

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[b] Southern Company, Commercial Refrigeration Equipment Framework Public Meeting Transcript, No. 9 at pp. 151; Air-Conditioning and Refrigeration Institute, Commercial Refrigeration Equipment Framework Public Meeting Transcript, No. 16 at pp. 151.

[c] Northwest Power Planning Council, Commercial Refrigeration Equipment Framework Public Meeting Transcript, No. 2 at pp. 161; Appliance Standards Awareness Project, Commercial Refrigeration Equipment Framework Public Meeting Transcript, No. 14 at pp. 155.

(building envelope and ventilation driven) in the building. This ratio determines the fraction of the year that the building is either in heating or cooling mode. However, the balance of refrigeration-driven space loads to the other space loads is impacted by the efficiency levels for all CRE equipment classes, complicating the analysis of each equipment class individually.

A second option was to simulate the performance of commercial refrigeration equipment using available software and to report the simulation program's results. However, unlike most other equipment regulated by DOE, the majority of commercial refrigeration equipment covered under this rulemaking is of the remote condensing type where the actual efficiency of the condensing unit is not regulated. Finally, commercial refrigeration equipment operating over a range of refrigerant suction temperatures is commonly assigned to operate with compressor racks that operate at a single temperature. This is accomplished through the use of evaporator pressure regulator (EPR) valves which provide for a drop in line suction pressure from that used to maintain temperature for a specific group of CRE cases down to that of the compressor rack. This system allows all cases in a large supermarket refrigerant system, with all its varied operating temperatures, to be served by two refrigerant racks (a medium-temperature and a low-temperature rack). In addition, performance of the refrigeration system can be impacted by decisions that provide heat exchange (primarily for subcooling) between the medium- and low-temperature racks.

The third option considered was to rely on estimates of the CRE energy consumption for various categories of equipment as were developed during the engineering analysis. This would provide energy consumption estimates for the wide range of equipment types covered under the rulemaking and was the most expeditious option. However, there were concerns over the reliance on the test procedure and its accuracy. DOE's primary concern was that the test procedure did not take into account the building space conditions that might impact the load on the commercial refrigeration equipment and, in particular, space-humidity loads in open cases. Nor was it certain that the annual energy efficiency of a typical compressor rack would be the same as the single point efficiencies used in the test procedure.

As discussed, DOE's approach for the ANOPR analysis was to use the engineering analysis estimates of energy consumption directly in the LCC analysis, but to also run selected design options for the equipment through whole-building simulation software, which included the simulation of refrigeration systems, to provide a check on the engineering analysis estimates of the annual energy consumption of the equipment. Refrigeration load and relative energy consumption for different types of commercial refrigeration equipment as a function of climate and energy consumption level were examined and simulation results were compared with engineering analysis estimates.

7.4 WHOLE-BUILDING ANNUAL ENERGY SIMULATION

The annual energy simulation of refrigeration equipment involved building a typical supermarket energy model with a realistic refrigeration system and selection of representative climate locations.

7.4.1 Modeling Tool

DOE used a whole-building simulation tool, DOE-2.2 refrigeration version (DOE-2.2R), to model a typical supermarket.[6] DOE-2.2R uses an extension of the circulation loop concept of DOE-2.2 for refrigeration systems. This tool is capable of simulating supermarket and industrial refrigeration systems in detail, as well as their interactions with the building and HVAC systems on an hourly basis. The refrigeration simulation model in DOE-2.2R is component-based, which allows users to build up a refrigeration system model out of individual components such as display cases, compressors, condensers and refrigerant circuits.

7.4.2 Climate Locations for Simulation

DOE selected five representative climate locations to simulate annual energy consumption of commercial refrigeration equipment: Baltimore, Chicago, Houston, Los Angeles and Memphis. DOE considered these climates as representative of the five large climates zones it uses in its building energy codes development work,[7] and together the five climate zones encompass about three-fourths of the U.S. population. These climate zones capture the significant variability of both outside air temperature and humidity in these climates. Chicago has cold and dry winters and represents the single most populous climate zone. Los Angeles is warm and dry during most of the year and represents the southwestern U.S. maritime region. Memphis and Baltimore have mild weather and represent the middle latitudes of the U. S. Houston represents the hot and humid climates found along the U.S. Gulf Coast.

7.4.3 Typical Supermarket Model

A whole-building energy model of supermarkets requires modeling the building envelope, internal loads and schedules, HVAC system and refrigeration system.

7.4.3.1 Building Description

Data from the Food Marketing Industry shows median average supermarket size in 2005 is 48,058 ft^2.[8] Based on this data, DOE modeled a typical 45,000 ft^2 supermarket with an aspect ratio of 1.0. The exterior walls of the building were modeled as 8-inch solid concrete block with R-5.7 insulation. The roof material was modeled as steel deck with rigid insulation and the floor was modeled as 12-inch soil and 6 inches of 140-pound heavyweight concrete.

DOE assumed that the supermarket was operated from 8 am to 12 am (16 hours/day). The lighting density and plug load density in sales area were assumed to be 1.5 W/ft^2 and 0.4 W/ft^2, respectively. The lighting power density was based on ASHRAE Standard 90.1-2004 Table 9.5.1.[9] Sensible load and latent load from occupants are 250 Btu/hr-person and 200 Btu/hr-person.[10] A constant volume packaged rooftop unit with electric direct expansion (DX) cooling and gas-fired furnace provided space-cooling and space-heating needs, respectively, in the sales area. The space thermostat set point was set at 70°F for heating and 75°F for cooling. Set-back controls and economizers were not used in the HVAC system. A summary of model assumptions for the building envelope, internal loads and HVAC system is provided in Table 7.4.1 and Figure 7.4.1.

7.4.3.2 Refrigeration System Model

A typical supermarket contains several compressor racks, each with multiple compressors, which are piped either in a loop or direct circuit to display cases, and walk-in coolers and freezers. The racks accommodate the evaporators by maintaining the lowest suction temperatures for the group. Typically, three to five compressor racks will be employed to provide all refrigeration in the supermarket. Each compressor rack may have from 3 to 5 compressors serving a series of loads with nearly identical evaporator temperature. A typical supermarket will have about 60 to 80 display cases.[11] About 60 percent of these are medium-temperature cases and 40 percent are low-temperature cases. Most of the medium-temperature cases are open display cases.

Table 7.4.1 Summary of Energy Model Assumptions for Building Envelope, Internal Loads and HVAC System

Characteristic	Model Assumptions
Building Envelope	
Gross Area, ft^2	45,000
Aspect Ratio	1.0
Number of Floors	1
Window to Wall Ratio	13%
Floor-to-Ceiling Height, ft	15
Floor-to-Floor Height, ft	18
Exterior Wall	
Structure	8 in. solid concrete block
Insulation	R-5.7
Roof	
Structure	Steel deck with rigid insulation
Insulation	R-15 ci
Floor	
Structure	Slab-on-grade
Floor Insulation	None
Floor F-factor	0.73
Window/Door	
Window Type	Double-pane clear glass w/alum. frame
Low-e	No
Actual DOE-2 Glazing Input	Glazing Code = 2000, U=0.57, SHGC=0.76
Window Overhangs/Interior Shading	None
Internal Loads	
Number of Occupancy	675
Occupancy Schedule	See Figure 7.4.1
People Sensible Heat Gain, Btu/h-person	250
People Latent Heat Gain, Btu/h-person	200
Peak Lighting Power, w/ ft^2	1.5
Lighting Schedule	See Figure 7.4.1
Peak Load, w/ ft^2	0.4
Equipment Schedule	See Figure 7.4.1
HVAC System	
HVAC System Type	Single-package rooftop unit, constant air volume system, electric DX cooling with gas-fired furnace
Number of Thermal Zones	1
Number of HVAC Units	1
Space T-stat Set Point	75°F cooling / 70°F heating
Ventilation Control Mode	Outside air damper remains open at minimum position
Fan Schedule	Always on
Outside Air Supply, cfm	6000
Return Air Path	Plenum return
Economizer	No

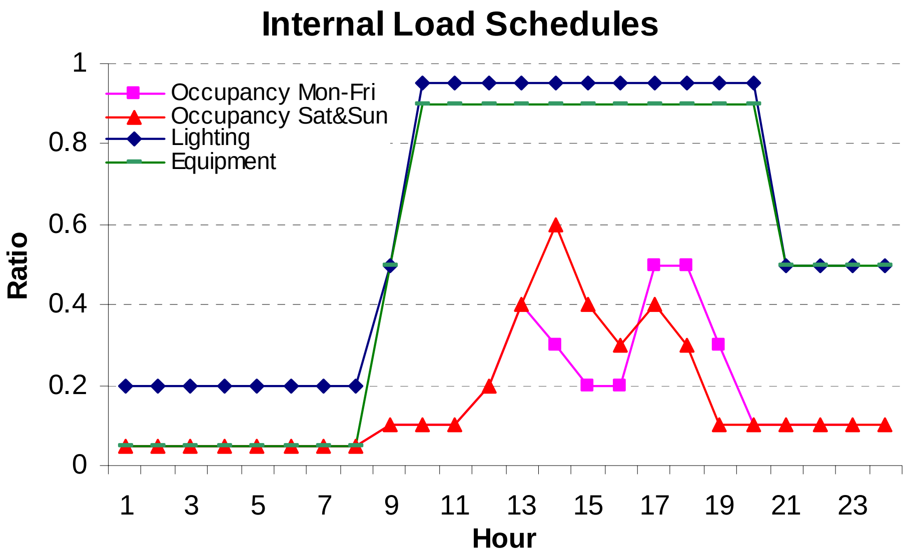

Figure 7.4.1 Internal Load Schedule Assumptions in Energy Model

Table 7.4.2 Line-up Length of Modeled Display Cases

Equipment Class	Unit Length (ft)	Line-Up Length (ft)
VOP.RC.M	12	384
SVO.RC.M	12	84
VCT.RC.M	12.7	25.4
HZO.RC.M	12	24
SOC.RC.M	12	24
VOP.RC.L	12	24
VCT.RC.L	12.7	266.7

In total, seven equipment classes of remote condensing equipment were modeled in this study, representing most ARI shipments (shipments data are given in Chapter 3, Market and Technology Assessment) and different characteristics. Figure 7.4.2 summarizes the main features of these display cases. Number of cases for each modeled equipment class was chosen to represent the distribution of display cases in a typical supermarket. Self-contained equipment were not included in this energy analysis because the compressor energy use of self-contained unit is only a function of refrigeration load and not dependent on climate conditions. Therefore, the energy consumption behavior of the self-contained equipment has similar features to remote condensing equipment with the same configuration at a specified climate location.

For each equipment class, DOE selected four representative efficiency levels from the engineering analysis and modeled them to compare simulated annual energy consumption with energy consumption derived from the engineering analysis. DOE simulated the four sets of efficiency levels for display cases one by one. Therefore, twenty runs (for five climate locations) were performed in the simulation. Table 7.4.3 lists the selected efficiency levels and design options and Tables D.2.1 through D.2.8 in appendix D.2 provide more detailed information on these design options, as well as corresponding input parameters in the DOE-2.2R model. The chosen design options were changes in lighting power, fan power, saturated evaporator temperature, and case door-type.

In an actual supermarket, different types of cases can be attached to a common compressor rack. However, the model assumes only one rack for each equipment class to facilitate data extraction by equipment class and limit interactions that might occur in real life due to cases with different suction temperatures used on the same racks. For example, an EPR valve is usually used to maintain evaporator/temperature control in different cases on the same refrigeration circuit. DOE chose compressors in the model to match those used in the ARI 1200-2006.[5] Both compressors for medium-temperature application and low-temperature application are Copeland discus compressors, which are typical in commercial refrigeration systems. Furthermore, DOE did not examine the impacts of compressor choices for modeled remote condensing equipment in this energy analysis, because DOE believes that using the same compressors as those in ARI 1200-2006 will give a fair comparison between the simulated energy use and the energy use based on the engineering analysis. The compressor model

Table 7.4.3 Modeled Design Options and Efficiency Levels

Equipment Class	**Efficiency Levels from Engineering Analysis**	**Design Options**			
		Lighting	**Saturated Evaporator Temperature (°F)**	**Evaporator Fan Motor-Type**	**Door-Type**
VOP.RC.M	AD1	T-8	15.0	SPM	—
	AD3	T-8	15.5	ECM	—
	AD5	T-8S	21.0	ECM	—
	AD7	LED	21.3	ECM	—
SVO.RC.M	AD1	T-8	15.0	SPM	—
	AD3	T-8	15.4	ECM	—
	AD5	T-8S	21.0	ECM	—
	AD6	LED	21.2	ECM	—
VCT.RC.M	AD1	T-8	27.0	SPM	Standard
	AD3	T-8	27.5	ECM	Standard
	AD4	T-8	28.5	ECM	high-performance
	AD6	LED	29.1	ECM	high-performance
HZO.RC.M	AD1	—	15.0	SPM	—
	AD2	—	15.4	PSC	—
	AD3	—	15.9	ECM	—
	AD4	—	21.2	ECM	—
SOC.RC.M	AD1	T-8	20.0	SPM	Standard
	AD3	T-8	21.0	ECM	Standard
	AD4	T-8S	21.3	ECM	Standard
	AD8	LED	27.2	ECM	high-performance
VOP.RC.L	AD1	T-8	-20.0	SPM	—
	AD3	T-8	-19.2	ECM	—
	AD5	T-8S	-14.2	ECM	—
	AD7	LED	-14.0	ECM	—
VCT.RC.L	AD1	T-8	-11.0	SPM	Standard
	AD3	T-8	-10.7	ECM	Standard
	AD4	T-8	-8.6	ECM	high-performance
	AD6	LED	-8.2	ECM	high-performance

numbers for medium-temperature and low-temperature applications are 3DB3A100E and 3DB3A075E, respectively.

DOE modeled air-cooled condensers located outside of the store because they are the most common type.[12] Evaporative condensers are also used in some supermarkets, primarily in hot, dry climates where a substantial difference in dry-bulb and wet-bulb temperatures exist. Water-cooled condensers are used primarily in urban stores where easy access to the outside for condensers does not exist. The model captured the difference in design condensing temperatures between medium- and low-temperature racks. For air-cooled condenser systems, the medium-temperature rack often operates with a 15°F temperature difference between saturated condensing temperature and design dry-bulb temperature. For the low-temperature rack, a 10°F temperature difference is common.[9]

DOE used floating-head pressure control in the model, which is standard on many refrigeration systems. In outdoor air-cooled condensers, floating-head pressure controls take advantage of low air temperatures to reduce the amount of compressor work by allowing the head pressure to vary with outdoor conditions. This reduces compressor load and energy consumption and can extend compressor life. DOE did not model condenser heat recovery.

7.4.4 Summary of Simulation Results

DOE simulated annual energy use for seven equipment classes was simulated at four representative efficiency levels. To examine the impacts of space relative humidity, refrigeration piping system thermal loss and climate locations on energy consumption of commercial refrigeration equipment, annual simulation data was compared with the consumption values calculated in the engineering analysis. The annual energy use and refrigeration load data extracted from the model were divided by 365 to provide average daily energy consumption and refrigeration load. Also, DOE assessed the magnitude of interactions between the refrigeration system and the building HVAC system.

7.4.4.1 Comparison of Case Refrigeration Load between Simulation Results and Engineering Analysis

Appendix D.4 summarizes the comparison of case refrigeration load between the simulation and engineering analyses at different climates and efficiency levels. Figure 7.4.2 gives an example of the comparison for VOP.RC.M equipment class.

From appendix D.4, DOE notes the following:

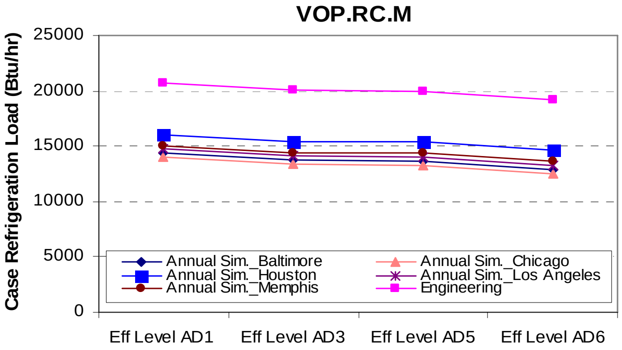

Figure 7.4.2 Case Refrigeration Load for Vertical Open, Remote Condensing, Medium Temperature Equipment Class

(1) In general, simulated case refrigeration load is lower than the load determined from the engineering analysis, because during most of the year, space relative humidity in the model is lower than that assumed in the test procedure. Space relative humidity is determined by HVAC-system humidity control and climate. In the building model, as in most supermarkets, there is no dedicated humidity control in the HVAC system. Instead, space dehumidification is solely driven by the space-cooling load.

(2) The difference between the simulation results and engineering analysis is greater for open equipment than closed equipment. Open equipment have much greater infiltration loads than closed equipment. Consequently, space humidity has a larger impact on open equipment.

(3) Case refrigeration load decreases with increased levels of efficiency. The percentage of load reduction in closed equipment is greater than for open equipment, and is depicted by the relatively flat curves in appendix D.4. The refrigeration load in open equipment is largely attributable to infiltration. However, reduction in infiltration load was not a design option and DOE did not analyze it. Instead, DOE assumed that the infiltration load was constant at all efficiency levels, which caused the load reduction in open equipment to be less significant in comparison to that in closed equipment.

(4) For closed equipment, SOC.RC.M, VCT.RC.M and VCT.RC.L, climate location has only minimal impact on case load. This is because infiltration load, which is driven by space relative humidity, which is in turn driven by climate, is relatively small at all efficiency levels.

7.4.4.2 Impact of Climate on Daily Energy Consumption

To assess the impact of climate conditions on display case energy consumption, DOE compared case daily energy consumption data (at the baseline efficiency level) for the five climate locations, ranging from cold and dry climate to hot and humid climate. Figures D.5.1 to D.5.7 in appendix D summarize the results, and Figure 7.4.3 provides an example for the VOP.RC.M equipment class.

In view of the results, DOE notes that climate locations have no influence on case energy consumption for all equipment classes and have only little influence on compressor energy consumption for equipment classes with doors. As expected, climate conditions have significant impact on compressor energy consumption for open equipment. Compressor energy consumption is determined by total refrigeration load and compressor efficiency, and both are affected by climate conditions for remote condensing equipment. An outdoor air-cooled condenser with floating-head pressure control operates more efficiently than at rated conditions, which reduces compressor energy consumption.

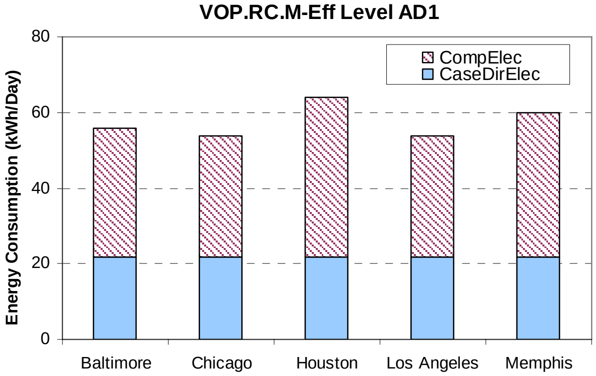

Figure 7.4.3 Baseline Daily Energy Consumption for Vertical Open, Remote Condensing, Medium Temperature Equipment Class for Five Climate Locations

7.4.4.3 Comparison of Daily Energy Consumption between Simulation and Engineering Analysis

Case daily energy consumption is comprised of case direct electric use and compressor electric use. Appendix D.6 shows comparisons of daily energy consumption between simulation and engineering analysis at different efficiency levels. Figure 7.4.4 gives an example for VOP.RC.M equipment class.

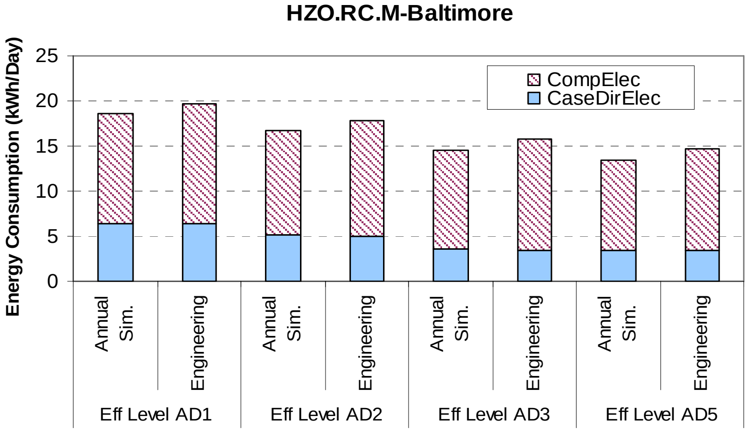

Figure 7.4.4 Daily Energy Consumption for Vertical Open, Remote Condensing, Medium Temperature Equipment Class at Baltimore

For a given efficiency level, case direct energy consumption is essentially the same for the simulation and engineering analysis. This is because case direct energy consumption is not dependent on space humidity and climate condition, and it does not change over the year or from location to location and because time based defrost termination was used. For HZO.RC.M, the case direct energy consumption was from 0-9 percent points higher in the simulation as additional defrost energy was required during some periods of the year in some climates.

(1) In Houston, compressor energy consumption is greater for the simulation for all equipment classes although simulated refrigeration load is lower than load derived in the engineering analysis, and the simulated compressors are on average more efficient over the year. This is because additional piping heat gains are dealt with in the simulation, but not in ARI Standard 1200-2006.

(2) For VOP.RC.M and SVO.RC.M and HZO.RC.M equipment classes at all efficiency levels, simulated compressor energy consumption is lower than engineering analysis data in Baltimore, Chicago and Los Angeles. For VCT.RC.M and SOC.RC.M equipment classes at higher efficiency levels, simulated compressor energy consumption is lower than engineering analysis data at Chicago and Los Angeles. This is because the simulated refrigeration load for these equipment classes is much lower than the engineering analysis data and the average compressor efficiency is improved compared to warmer climates. Therefore, even the inclusion of “additional superheat load” in the simulation still requires less compressor energy use.

(3) For VOP.RC.L and VCT.RC.L equipment classes (Figures D.6.26 through D.6.35), simulated compressor energy consumption is greater than the engineering analysis data (except for VCT.RC.L at higher efficiency levels in Los Angeles) because the lower refrigeration load (compared to the engineering analysis) seen with the simulation results does not offset the “additional superheat load” calculated in the simulation.

(4) The difference in compressor energy consumption between the simulation and engineering analysis data can be as much as 11 percent to 25 percent in the hot and humid climate represented by Houston.

7.4.4.4 Energy Savings Comparison at Different Efficiency Levels and Climates

Appendix D.7 compares case energy savings (relative to a baseline efficiency level) for the efficiency levels at five climate locations. Figure 7.4.5 is one example of appendix D.7 for the VOP.RC.M equipment class.

(1) As expected, energy savings increases with higher efficiency levels.

(2) Generally, simulated average daily energy savings are somewhat greater than the savings calculated by the engineering analysis. This is caused by underestimation of energy savings in ARI Standard 1200-2006 due to “additional superheat load” issue.

(3) Warm and humid climates (Houston and Memphis) have greater energy savings than relatively cooler and drier climates (Baltimore, Chicago and Los Angeles) at efficiency level AD4 or higher, though the difference is not significant for closed cases

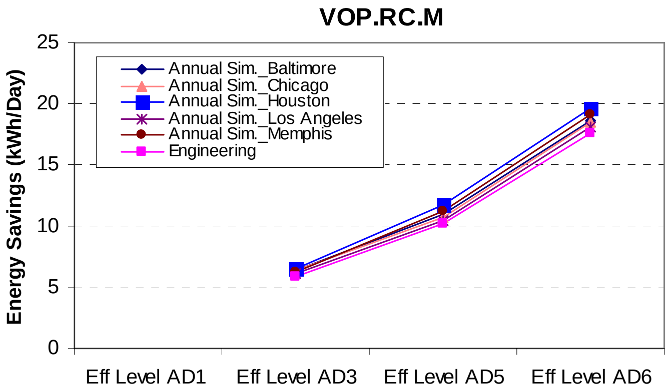

Figure 7.4.5 Daily Energy Savings for Vertical Open, Remote Condensing, Medium Temperature Equipment Class

7.4.4.5 Impacts of Display Case Improvements on Whole-Building Energy Consumption

DOE examined whole-building energy consumption and refrigeration-system energy consumption for selected efficiency levels and climate locations to determine if the display case improvements considered in the engineering analysis would have a significant effect on building HVAC-system energy use. The influence of reduced display case energy consumption on HVAC-system energy use varies depending on characteristics of design options. For example, improved lighting efficiency would reduce energy consumptions of both refrigeration system and potentially the space cooling load depending on lighting placement. By contrast, reduction in conduction and radiation loads on refrigeration equipment would increase the space-cooling load on the HVAC system while decreasing the space-heating load.

As discussed in section 7.4.3.2, DOE selected four efficiency levels for each type of display case for its annual-energy simulation. To calculate whole building energy savings at different sets of efficiency levels, DOE denoted the first set of efficiency levels (Level AD1 for all equipment classes) in combination as Level A. The second, third and fourth sets of efficiency levels selected for all products were combined as Levels B, C, and D respectively. Table 7.4.4 shows the mapping of the efficiency levels from the engineering analysis to the efficiency levels used to calculate whole building energy savings.

Table 7.4.4 Mapping of the Efficiency Levels for the Whole Building Analysis

Equipment Class	Level A	Level B	Level C	Level D
VOP.RC.M	AD1	AD3	AD5	AD7
SVO.RC.M	AD1	AD3	AD5	AD6
VCT.RC.M	AD1	AD3	AD4	AD6
HZO.RC.M	AD1	AD2	AD3	AD4
SOC.RC.M	AD1	AD3	AD4	AD8
VOP.RC.L	AD1	AD3	AD5	AD7
VCT.RC.L	AD1	AD3	AD4	AD6

The net energy savings at a specified level are shown as the percent of the difference in energy consumption between a specified level and Level A (baseline). Table 7.4.5 summarizes energy savings results of refrigeration systems and whole buildings for the five climate locations. Also, appendix D.8 graphically compares energy savings between the refrigeration system and the whole building. Figure 7.4.6 provides an example of these graphs.

Whole-building energy simulation results show that refrigeration system energy consumption ranges from 42 percent to 57 percent of the whole-building energy consumption at baseline (Level A) and 32 percent to 47 percent at the highest efficiency level (Level D). Results also showed that the simulated supermarket is heating-load dominated with a small amount of space-cooling energy consumption.

Table 7.4.5 Summary of Energy Savings for Refrigeration System and Whole Building

Location	Level	Refrigeration System Savings (MBtu)	Whole Building Savings (MBtu)			Difference (%)
			Natural Gas	Electricity	Total	
Baltimore	Level B	460.9	-9.6	461.2	451.6	-2%
	Level C	1250.5	132.6	1246.4	1379.0	10%
	Level D	1764.4	134.4	1760.1	1894.5	7%
Chicago	Level B	456.0	-10.0	456.0	446.0	-2%
	Level C	1223.0	134.3	1219.9	1354.2	11%
	Level D	1736.6	135.1	1733.2	1868.3	8%
Houston	Level B	479.8	-10.4	480.6	470.2	-2%
	Level C	1352.4	121.9	1341.9	1463.8	8%
	Level D	1896.5	124.4	1885.2	2009.6	6%
Los Angeles	Level B	447.6	-11.7	447.7	436.0	-3%
	Level C	1223.5	136.7	1222.5	1359.2	11%
	Level D	1733.0	137.8	1731.9	1869.7	8%
Memphis	Level B	467.4	-8.4	467.9	459.5	-2%
	Level C	1298.9	124.6	1291.1	1415.7	9%
	Level D	1827.0	127.0	1818.5	1945.5	6%

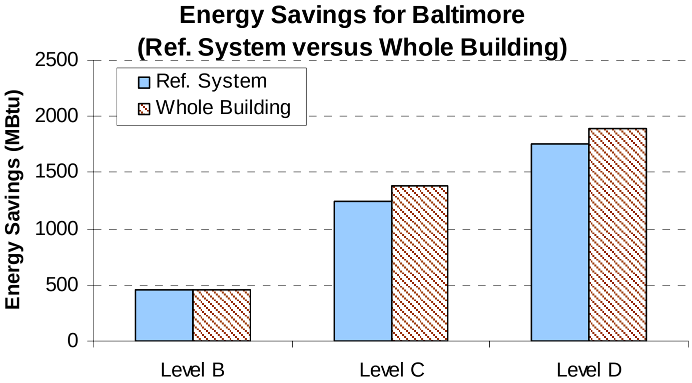

Figure 7.4.6 Energy Savings for Baltimore (Refrigeration System versus Whole Building)

Some important observations of Table 7.4.5 and appendix D.8 are summarized as follows:

(1) The difference in energy savings, between the refrigeration system and the whole building, is less than 11 percent at all levels (with the whole building energy savings averaging 7% at the highest level). This means that the whole-building energy savings are mostly attributable to the energy savings of the refrigeration system and that the interaction between the display cases changes and the HVAC system is small.

(2) The improvements on equipment have little to no significant impact on space-cooling and space-heating energy use, partially because case infiltration, which is the most important factor for HVAC system loads, was not considered as a design option in the engineering analysis.

(3) At Level B, the refrigeration system energy savings is slightly higher than the whole-building energy savings due to a slight change in the natural gas heating usage.

(4) The energy savings differences at Level C are relatively higher than at the other levels. As shown in appendix D.2, saturated evaporator temperatures for equipment are increased at this level. Thus, the energy savings are almost entirely from the refrigeration system and without any interaction with the building HVAC system.

7.5 SENSITIVITY TO DISPLAY CASE LIGHTING OPERATING HOURS

In addition to its examination of the energy consumption calculations using whole-building simulation tool, DOE also undertook a sensitivity study of the energy savings for higher efficiency levels as a function of the display case lighting operating hours.

In commercial refrigeration equipment with installed lighting, the operating hours of the lighting system impact the total daily or annual energy consumption of the equipment. DOE presumed a 24-hour lighting energy use for this equipment in the engineering analysis and in the comparisons to whole-building simulation estimates discussed previously in this Chapter. In the field, however, lighting operating hours can vary. DOE believes that a relatively small fraction of supermarkets are open for business on a 24-hour basis.

A higher fraction of convenience stores do operate on a 24-hour basis. However, most stores in both categories are open to customers fewer hours. Building operating-hours data from the Energy Information Administration’s (EIA) 2003 Commercial Building Energy Consumption Survey (CBECS)[13] are shown in Table 7.5.1 as square-foot-averaged number of hours open per week for each of the four building types which make up the CBECS “Food Sales Principal Building Activity” category.

Table 7.5.1 Average Reported Hours Open – Commercial Buildings Energy Consumption Survey 2003

	Convenience store	Convenience store with gas station	Grocery store/food market	Other food sales	Average All Food Sales Bldgs
Hours per week (Max 168)	121	135	107	64	112
Hours per day (Max 24)	17.3	19.3	15.3	9.2	16.0

While a supermarket may be open a specified number of hours for customers, commonly night hours are used for stocking purposes. During stocking, it is common to have some lighting on in the display cases.

Some supermarkets and convenience stores operate on a 24-hour basis for customers. In these stores, stocking of cases is done often at night when shoppers are less frequent and stocking has less impact on these shoppers. DOE presumed that display case lighting is on 24 hours in these stores.

In other stores, display case lighting is on 24 hours per day even when customers are not present. A common reason cited is potential or perceived difficulty of restarting fluorescent lighting in low-temperature cases.[14] Analysis of energy savings for display-case lighting controls often presumes this mode of operation.

A second common operating mode is to reduce the display case lighting during hours the store is not open. Case lighting may be operated at a 50 percent level during stocking, and then shut off for the remainder of the evening.[15] In yet another strategy, display case lighting may be shut down to a 50 percent level, but not turned completely off during unoccupied periods. The rationale here is that lighting may be left on for security reasons or to make sure that some lights will be available when customers arrive the next morning.[16]

DOE has no definitive data on the number or square footage of stores falling into any of these categories. A survey undertaken by the Food Industry Center at the University of Minnesota suggests that an automated energy management system either for lighting, HVAC, or both exist in 44 percent of supermarkets.[17] The survey found that these systems were more prevalent in large chain stores than smaller independent stores. The survey neither contained data on whether these systems actually controlled the display-case lighting, nor any after-hours lighting strategy.

For the ANOPR, DOE based the energy consumption analysis on the assumption of 24-hour lighting. However, DOE recognized that this may overstate the energy savings for some equipment classes and design options. DOE conducted a sensitivity study of the relative energy consumption as a function of display-case lighting operating hours per day, and used the spreadsheet-based engineering analysis tool and the analytically derived energy consumption levels for the 15 equipment classes. (See Chapter 5, Engineering Analysis.) The display-case lighting operating hours per day was varied from 16 hours to 24 hours to assess the daily energy consumption savings calculated for each efficiency level (savings referenced to the Level 1 baseline) relative to the daily energy consumption savings calculated assuming 24-hour display case lighting. The results are shown in Figure 7.5.1 for each equipment class assuming an intermediate efficiency level (Analytical Design Level 3), and in Figure 7.5.2 assuming the highest (max tech) efficiency level developed for each equipment class. For the equipment class with no case lighting, there is no impact on the energy consumption. For all other equipment classes, a reduction in case lighting hours results in a reduction in daily energy savings. For equipment classes with lighting but where the efficiency level uses the same lighting technology as the baseline level, the relative reduction in energy savings can be very slight (Figure 7.5.1, VCT.RC.L equipment class). However where a change in lighting technology occurs, the difference in energy savings can be significant (Figure 7.5.1, SVO.RC.M equipment class).

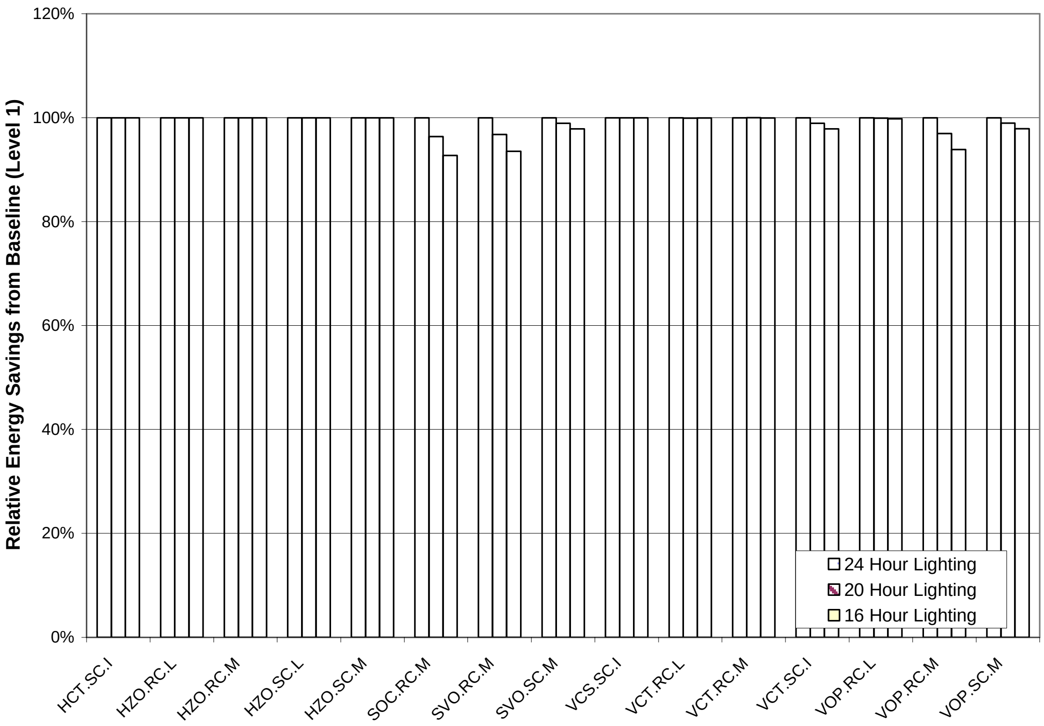

Figure 7.5.1 Energy Savings versus Case Lighting Operating Hours for Efficiency Level AD4

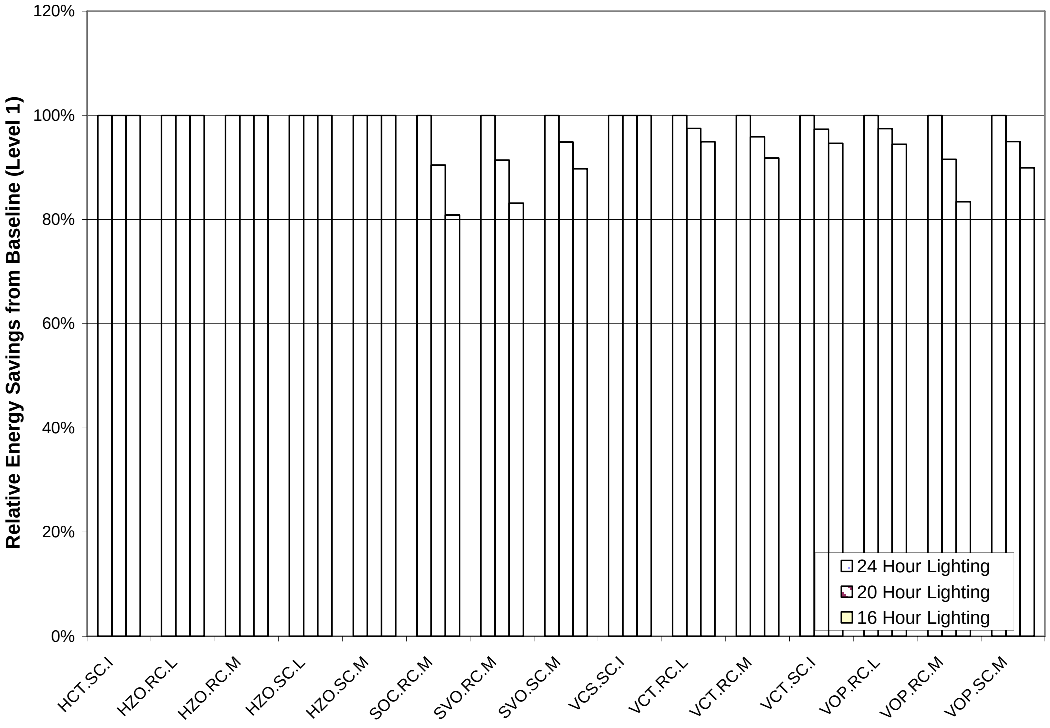

Figure 7.5.2 Energy Savings versus Case Lighting Operating Hours for Highest Efficiency Level Analyzed

The reduction in calculated energy saving across all levels considered in the engineering analysis for the products with display-case lighting was between 0 percent and 4 percent at AD4 and at a 20 hours-per-day lighting use schedule (averaging 1.5 percent across all equipment classes with installed lighting). It was between 0 and 7 percent at a 16 hours-per-day lighting use schedule (average 3 percent reduction across all equipment with installed lighting). At the max-tech efficiency level, the reduction in energy savings for equipment with display-case lighting was between 3 and 10 percent at 20 hours-per-day, averaging 5 percent. At 16 hours-per-day lighting, the reduction in energy savings was between 5 and 19 percent, averaging 11 percent.

At the ANOPR public meeting, DOE presented its analysis of the energy impact of differing lighting operating hour assumptions. Manufacturers at the meeting and industry trade groups in follow-up comments stated that 24 hour cases lighting is a valid assumption for DOE's energy analysis, citing concerns by equipment owners with lighting restarts in low-temperature display cases and moisture-related lighting maintenance issues. It was noted that these problems didn't exist with LED lighting.[d] Based on these comments, DOE continued to use the 24-hour assumption of display case lighting operating hours for the NOPR analysis. Additional detail on the energy use characterization can be found in Chapter 7 of the TSD.

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[d] Hussman; Public Meeting Transcript, No. 13.5 at p. 118; Hill Phoenix Public Meeting Transcript, No. 13.5 at p. 118, Zero Zone, No. 17 at p.4, and ARI, No. 18 at p.4.

7.6 CONCLUSIONS

DOE conducted annual simulations of the refrigeration system energy consumption (including both direct case-energy use as well as compressor energy consumption) over four efficiency levels selected to span the range of efficiency levels analyzed by DOE in its engineering analysis. The inputs to the annual simulations reflected the engineering analysis assumptions for these equipment classes. Simulations were done in five different climates to ascertain the impact of climate on the case load, energy consumption, and energy savings.

In general, the refrigeration load was smaller than that predicted by the engineering analysis, due to differences between the building-space conditions through the year and those used for the steady-state test procedure. The simulated energy consumption of the compressors was higher than that predicted by the engineering analysis for medium temperature cases in warm and humid climates (Memphis, Houston) with all equipment classes and for low temperature cases in general. The simulated energy consumption of the compressors was lower than in the engineering analysis for medium temperature cases in colder/drier climates (Baltimore, Chicago, Los Angeles). The difference in energy consumption between the simulation and engineering analysis is due to the differences in refrigeration loads from the case, the fact that the simulation accounts for changes in condensing temperatures over the year for each climate location, and partly due to the additional superheat loads calculated by the simulation software to bring the return refrigerant vapor up to the compressor inlet temperature conditions (assumed to be 65°F).

Analysis of the annual refrigeration system energy savings for each of the three efficiency levels above the baseline efficiency level (Level AD1) were within 14 percent of that predicted by the engineering analysis for the 6 equipment classes examined, but averaged 4.2 percent higher for the highest efficiency level examined. For the VOP.RC.L equipment class as discussed above, the annual energy savings deviated by as much as 21 percent. Shipments for this equipment class are unknown, but are expected to be small. This suggests that for the majority of the equipment classes, the energy savings predicted by the test procedure agreed reasonably well with the annual simulation results, although the impact of individual design options may differ.

The same simulations allowed DOE to examine the energy savings calculated for the refrigeration system alone compared with energy savings for the whole building. For all efficiency levels analyzed, the annual whole-building energy savings was within 11 percent of that calculated for the refrigeration system alone. The widest deviations between whole building and refrigeration-only energy savings appears at a mid-range efficiency level (Level C). At the max-tech level the energy savings varied between 6 and 8 percent across the five climates. The simulation results suggest that the overall impact of the design options when taken together had only a minor effect on the overall HVAC energy consumption.

Based on the general agreement between the energy savings results from the annual simulation modeling and the energy savings estimates from the spreadsheet-based engineering estimates of energy use, DOE used the energy use estimates from the engineering analysis for the LCC and subsequent analysis.

A sensitivity analysis of the impact of reduced display case lighting hours from the 24 hours presumed in the engineering analysis showed that such reduction would also reduce the energy savings calculated for all equipment with display-case lighting. The actual reduction in energy savings is dependent on the equipment class and on design options incorporated at each efficiency level but for equipment which utilize lighting the reduction in energy savings at the max-tech level would typically vary between 3 percent and 10 percent at a 20-hour-per-day lighting-use schedule (averaging 5 percent) and between 5 percent and 19 percent at a 16-hour-per-day lighting-use schedule (averaging 11 percent).

DOE presents more detailed results and supporting data for the energy use characterization in appendix D.

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[15] Personal conversation with energy management personnel at Phoenix, Arizona Safeway during Food Marketing institute tours. Cited typical hour stocking periods (10 pm-4 am) with case lighting at 50%, with case lighting turned off for additional 4 hour period before store reopens. Sept 12, 2006

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