Spring 1-1-2013

Balancing Latent Heat Load Between Display Cases and Store Comfort Cooling

Penelope J. Cole
University of Colorado at Boulder, pennytravelling@yahoo.com

Balancing Latent Heat Load Between Display Cases and Store Comfort Cooling

by

Penelope J. Cole

B.E. (Hons), University of Canterbury, 2001

A thesis submitted to the

Faculty of the Graduate School of the

University of Colorado in partial fulfillment

of the requirements for the degree of

Masters of Science

Department of Civil, Environmental and Architectural Engineering

2013

This thesis entitled:
Balancing Latent Heat Load Between Display Cases and Store Comfort Cooling
written by Penelope J. Cole
has been approved for the Department of Civil, Environmental and Architectural Engineering

____
Prof. Michael Brandemuehl, PhD, P.E.

____
Prof. Moncef Krarti, PhD, P.E.

____
Prof. John Zhai, PhD

Date________________

The final copy of this thesis has been examined by the signatories, and we find that both the content and the form meet acceptable presentation standards of scholarly work in the above mentioned discipline.

Cole, Penelope J. (M.S., Department of Civil, Environmental and Architectural Engineering)

Balancing Latent Heat Load Between Display Cases and Store Comfort Cooling

Thesis directed by Prof. Michael Brandemuehl, PhD, P.E.

Supermarkets are the most energy intensive buildings in the commercial sector, and are responsible for approximately 54.5 billion kWh of electricity annually. Refrigeration makes up approximately half of this electricity use, with store temperature and humidity having a significant impact on this. Conditioned store air exchanges both moisture and heat with the refrigerated cases, and increases in store temperature and humidity impose higher loads on the refrigeration and cause sweating and frost.

Reducing the store humidity level has been shown previously to have a positive effect on the refrigerated case energy use, however dehumidification has an energy penalty on the HVAC system.

The project assessed the potential for energy savings due to humidity in supermarkets by optimizing the design and operation of the combined HVAC and refrigeration systems. The assessment included the effects of climate, space temperature and humidity setpoint controls, HVAC system and layout, and the design, operation and layout of the refrigerated cases.

EnergyPlus was used to model a typical store, and a fractional factorial analysis was conducted to analyze the effects and interactions of a selection of refrigeration and HVAC factors. The results showed that, contrary to initial expectations, the HVAC factors had significantly less influence than had been expected. Refrigeration factors dominated all cases, and changing the zoning and humidity setpoints of the supermarket had very small influence on the total electricity use. However moving the refrigerated cases around between store zones had a significant influence. If only a limited budget is available, it would be best spent on improvements to the refrigeration system, as this will have the most significant influence on the energy use of the store.

Acknowledgements

I would like to express my thanks to all those who have supported the completion of this thesis. First, I offer my sincerest gratitude to my advisor, Professor Michael Brandemuehl, for all of his support and assistance. Without his valuable insights, encouragement, and guidance I could not have succeeded in my endeavors. I would also like to thank the other members of my committee, Professor Moncef Krarti, and Professor John Zhai for their participation.

I would also like to acknowledge the American Society of Heating, Refrigeration, and Air conditioning Engineers for providing funding for this research, as well as the members of the guiding committee for their invaluable assistance with the process.

Finally, I would like to express my gratitude to my husband, Tony Hahn, for his support throughout this process, as well as my thanks to my friends and colleagues in the BSP program for both their assistance and the loaning of a friendly ear in a time of need.

Contents

Tables

Figures

Chapter 1

1 Project Overview

1.1 Introduction

Supermarkets are the most energy intensive buildings in the commercial sector, and are responsible for approximately 54.5 billion kWh of electricity annually. Refrigeration makes up approximately 50% of this electricity use. While the outdoor condensing temperature is the dominant effect on the refrigeration energy use, the store temperature and humidity can also have a significant impact [1].

There is a strong interaction between a supermarket's HVAC system and its refrigerated display cases. Conditioned store air exchanges both moisture and heat with the refrigerated cases. Increases in store humidity impose heavier loads on the refrigeration equipment and cause sweating on products and shelving, as well as frost on evaporator coils [2][3].

Modern supermarkets have a high percentage of refrigerated cases with glass doors, which somewhat reduce the problem of both sensible and latent heat exchange with the environment by reducing the air change rates. However the doors require anti-sweat heaters and not all retailers accept these cases for all display types [2].

However, reducing the store humidity level can have a positive effect on the refrigerated case energy use. Reducing the conditions from 55% to 35% relative humidity (RH) has been demonstrated to produce an 18% decrease in compressor power demand for open case refrigeration [4]. However dehumidification has an energy penalty on the HVAC system, and this penalty can be large when considering the volume of air that a typical supermarket HVAC system deals with.

If it could be proven that the dehumidifying effect of the refrigerated cases was localized, then supermarket HVAC design could be modified such that dehumidified air was provided to the case area, while the dry goods area was maintained at a higher humidity ratio. Typically energy savings can be realized within a comfort band of 30-60% RH [1]. This has the potential to save considerable energy in the refrigeration system, while minimizing the energy penalty on the HVAC system.

1.2 Scope

The overall objective of this project was to provide a comprehensive assessment of the potential for energy savings due to humidity in supermarkets by optimized design and operation of the combined HVAC and refrigeration systems. The assessment included the effects of climate, space temperature and humidity setpoint controls, HVAC system type and characteristics, and the design and operation of the refrigerated cases. Furthermore, the project addressed the overall layout of HVAC and refrigeration system components in supermarkets, including HVAC zoning, and the overall air distribution patterns in the supermarket.

1.3 Thesis Organization

The following thesis first presents an analysis of the energy modeling tools that were available for use on the project. Next, model validation is undertaken with real world measured data to prove the selected tool.

A prototype store is developed to mimic a “typical” modern store, drawing from a variety of sources, then a sensitivity analysis is undertaken with some elements of the store design considered less critical to the overall study.

Next in Chapter 6 a parametric analysis is undertaken with a matrix of factors considered to be important to the study.

The factors are then refined to be used in a fractional factorial analysis study, and an explanation of fractional factorial analysis is also included.

Finally, conclusions about the modeling, the effects found through the analysis, use of the results, and application of this study are presented. Options for potential future work are also presented.

Chapter 2

2 Energy Modeling

2.1 Available Modeling Programs

The first objective of the project was to identify the most appropriate modeling tool, able to simulate the overall supermarket building and conditioned space, the refrigerated display cases and associated refrigeration system, and the range of available HVAC systems which were to be considered.

The review of the available simulation models for supermarkets began with a detailed review of the capabilities and methods of EnergyPlus, which was initially determined to be the model most likely to be suitable, with other models being assessed against this as a benchmark. Simulation models based in Excel, including RetScreen, ClimTop and ORNL, were excluded as the capabilities of these were considered not able to match the more detailed modeling programs. The models reviewed were:

- EnergyPlus, developed by the U.S. Department of Energy
- eQuest, developed primary by James J. Hirsch and Associates and the Lawrence Berkeley National Laboratory (LBNL), USA.
- Cybermart, developed by the Royal Institute of Technology, Sweden
- SST, developed by the Electric Power Research Institute (EPRI), USA
- SuperSIM, developed by Brunel University, Uxbridge, UK

2.2 Energy Plus

EnergyPlus is the official energy simulation program of the United States Department of Energy (DOE). Developed from the BLAST and DOE-2 programs, it is an energy analysis and thermal load simulation program. The program models heating, cooling, lighting and ventilation of buildings. It did not have a graphical interface at this stage. While one was available for purchase it focused on the building inputs and not the system inputs which are considerably more important for the purposes of this project. The then current version of EnergyPlus was version 4.0, released in October 2009. The version used for the majority of this project was v6.0. EnergyPlus can model a wide range of HVAC systems, as well as multi-zone models and airflow networks in buildings.

EnergyPlus can model refrigeration equipment including compressor racks, refrigerated cases, walk-in coolers, and heat reclaim air and water heating coils. The refrigeration models undertake the following functions:

- calculate the electric consumption of refrigerated cases and walk-in coolers connected to a compressor rack
- determine the impact of refrigerated cases and walk-in coolers on zone cooling and dehumidification loads
- calculate the electric consumption and COP of the compressor rack, and the consumption related to cooling the compressor rack's condenser
- determine the total amount of heat rejected by the compressor rack's condenser and store this information for use by waste heat recovery models [14]

The performance of the case and walk-in models is based on evaporator load, fan operation, lighting, defrost type, and anti-sweat heater operation. Air and water heating coils can also be modeled to reclaim available waste heat from the compressor rack. Issues were later determined with the calculation of this reclaim, but at the time of program selection it was believed that this was true.

The system can be modeled in two ways. The first, simpler, way is to create an object which combines the compressors and the condenser into a single entity. This then works in conjunction with a modeled refrigerated case or walkin cooler to model a simple supermarket system. The performance of this unit is driven by the total case load and the heat rejection environment. The second modeling option allows more detailed input, and makes use of separate compressor and condenser objects. It also includes modeling capabilities for subcoolers, cascade condensers, and secondary loops.

The modeling of refrigerated cases, and of walkin coolers and freezers, is the same for either simple or detailed models. The models are built up in EnergyPlus using a combination of manufacturers' data and built in assumptions and calculations.

2.2.1 Refrigerated Case Models

Refrigerated cases are modeled using inputted performance information. Inputs allow calculation of the energy use of lights, fans and anti-sweat heaters. The model accounts for sensible and latent heat exchange with the environment in which the case is located.

The total calculated load on the refrigerated case evaporator is the sum of the loads experienced by the case. These load components are typically known for a refrigerated case at rated ambient air conditions (typically 75°F and 55% RH) and the specified case operating temperature.

Several of the load components are provided by the case manufacturer (e.g., fan, lighting, antisweat heater, and defrost loads). The remaining load components must be estimated. A combination of user input curves and fixed EnergyPlus defined relations adjust for case performance at off-rated conditions.

Undefined case loads due to the wall heat conduction, radiation, and infiltration are estimated by the model as a single value by subtracting the known loads at rated conditions from the rated total cooling capacity of the case.

If the total heat load on the case is greater than the available evaporator capacity the load is accumulated to be met during subsequent time steps.

The components of the case load are as follows:

Evaporator Fan: Calculated as a product of the case fan power per unit length of case, the length of the case, and the fraction of time that the case is not being defrosted. The load is assumed to be entirely within case.

Lighting: Calculated as a product of the standard case lighting power per unit length of case, the lighting schedule value, and the length of the refrigerated case. Scheduling can be used to mimic high efficiency lighting. Can define the portion of the load received by the case and the portion seen as a space load.

Anti-sweat Heater: Available control strategies are none, constant, linear variation with ambient relative humidity or dewpoint temperature, and a theoretical model that determines the minimum anti-sweat heater power required to maintain the case surface just above the temperature where condensation would occur.

Restocking: The stocking schedule is entered as a heat gain rate per unit length of the refrigerated case. This load is only sensible, no latent component is included.

Defrost: Defrost strategies that can be simulated are: none, off-cycle, electric, hot-gas, and hot-brine.

Sensible Case Credits: EnergyPlus terminology for sensible energy removed from surrounding environment by the case. The model first calculates the rated sensible case credits by subtracting the known loads at rated conditions from the rated sensible cooling capacity of the case. For every time step, rated credits are adjusted to account for off-rated variations. The adjusted case credits are calculated from the following equation:

$$\dot{Q}cc_{sens} = \dot{Q}cc_{sens,rated}\left(\frac{T_{db,air} - T_{case}}{T_{db,rated} - T_{case}}\right)(SCH_{cc})$$

where:

$\dot{Q}cc_{sens}$ = sensible case credits adjusted for ambient temperature and case credit fraction

$\dot{Q}cc_{sens,rated}$= sensible case credits at rated conditions (W)

$T_{db,air}$ = dry-bulb temperature of the ambient (zone) air (°C)

T_{case} = case operating temperature (°C)

$T_{db,rated}$ = rated ambient (zone) dry-bulb temperature (°C)

SCH_{cc} = case credit fraction (schedule value, 0 to 1)

Latent Case Credits: terminology for latent energy removed from surrounding environment by the case, composed solely of the latent heat transfer by air infiltration. The calculation is a product of the case length, total cooling capacity per unit length, latent heat ratio, and runtime fraction at rated conditions, as well as a factor to account for lower ambient humidity levels. The same schedule is used as for the sensible credits to allow for covers etc. The modified latent credits are calculated by:

$$\dot{Q}_{inf,lat} = -\dot{Q}cc_{lat} = \dot{Q}_{case,rated}(LHR_{rated})(RTF_{rated})(SCH_{cc})(LatentRatio)L_{case}$$

where:

$\dot{Q}_{inf,lat}$ = latent load on the refrigerated case evaporator at current ambient conditions

$\dot{Q}cc_{lat}$= latent case credit impact on zone load, negative for dehumidification (W)

$\dot{Q}_{case,rated}$ = case rated total cooling capacity per unit length (W/m)

LHR_{rated}= latent heat ratio of the refrigerated case at rated conditions

RTF_{rated} = runtime fraction of the refrigerated case at rated conditions

SCH_{CC} = case credit fraction (schedule value, 0 to 1)

LatentRatio = ratio of actual latent load to rated latent load on the case, based on latent case credit curve

L_{case} = case length (m)

Allows the user to specify a latent case credit curve to adjust the load based on ambient humidity, and the user can select from three curve types: Case Temperature Method, Relative Humidity Method, or Dewpoint Method.

Under Case Return Air: Model uses a predetermined relationship to determine the fraction of case credits that directly cool and dehumidify the HVAC system return air if under case return air is used.

2.2.2 WalkIn Cooler and Freezer Models

Walkins differ from cases in that they may have surfaces facing more than one zone, and that they are always equipped with doors. As such their sensible and latent exchange with zones is calculated differently.

Sensible and Latent Heat Exchange: Both sensible and latent energy can be exchanged with multiple zones depending on store layout, so heat transfer calculations are performed separately for each zone with user defined values for area and conductance. Sensible and latent infiltration through doors is modeled based on door type.

Fans, Heaters, Lighting and Restocking: These loads are calculated as per the cases, but are entirely allocated to the walkin, not surrounding zones. A general circulation fan at the cooling coil is added and assumed to run constantly.

2.2.3 Detailed Refrigeration Systems

Simple refrigeration systems are available in EnergyPlus, however the detailed systems were more appropriate for this project. The detailed system model differs from the simple model as follows: a) Performance data inputted for each compressor, b) Performance curves inputted for each condenser, c)

Calculates the amount of superheat available for reclaim, d) Allows suction temperature to rise when case loads are less than design loads, improving compressor efficiency, e) Allows cascade condensers and secondary systems, and mechanical subcoolers, f) Allows the use of liquid suction heat exchangers, g) Models three condenser fan types, h) Does not assume that the compressor and condenser capacity is sufficient to meet the case loads, instead carries unmet load over to the next time step, i) Provides optional suction piping heat gain, j) Can't be used for a compressor rack discharging heat into a conditioned zone.

The loads from the connected refrigerated cases and walkins are summed to give the initial refrigeration load and evaporating temperature. Suction line heat gain can be included if desired. An initial estimate of condensing temperature is used to calculate compressor power use.

From this the heat rejection load on the condenser is calculated, giving a new estimate for the condensing temperature. This process is iterated to get final condensing temperature and compressor power at each time step. This is done for each system, then energy transfers between the systems are solved, twice to ensure systems are balanced.

The loads for this system are:

<u>Compressor Energy Use:</u>

The compressor energy calculations begin with the calculation of the suction and discharge conditions. These temperatures are then used with each compressors performance curve. The rated values for cooling capacity and power consumption from the manufacturer include a specified amount of subcooling before the thermal expansion valve and a certain amount of superheat in the suction gas, and adjustments must be made to reflect the actual subcooling and superheat conditions. Actual subcooling is determined by the condenser's rated subcooling and by the subcooling provided by optional subcoolers. The actual superheat is determined by the refrigerated case superheat and the effect from any optional subcoolers.

The capacity corrections are calculated from the following equations:

$$Cap_{corrected} = \frac{\rho_{1b}}{\rho_{1c}} \times \frac{(h_{1b} - h_4)}{(h_{1c} - h_4)} Cap_{rated}$$

$$\dot{m} = \frac{Cap_{corrected}}{(h_{1b} - h_4)}$$

where:

$\dot{m}$= mass flow rate of refrigerant (kg/s)

ρ = density (kg/m^3)

h = enthalpy (J/kg)

Cap = refrigeration capacity of an individual compressor (W)

Compressor capacities are then applied one at a time to the system load until it is met. The model does not allow for part-load cycling. Unmet load is saved for the next time step.

Condenser Energy Use: The condenser, one per system, can be dry air cooling, wet evaporative cooling, water loop cooling, or cascade cooling, and determines: (a) the condensing temperature and enthalpy of the refrigerant entering the refrigerated cases attached to the suction group, (b) auxiliary power consumption for fans and pumps, and (c) water consumption for evaporative and water-cooled condensers. Can also simulate waste heat being reclaimed for use by refrigerant-to-air and refrigerant-to-water heating coils.

2.2.4 HVAC Systems

EnergyPlus is capable of modeling the typical supermarket HVAC systems, including gas fired desiccant systems, conventional DX systems with airflow and bypass control, heat pipes and run around coils, as it is originally an HVAC system modeling program.

As part of the HVAC modeling capabilities, EnergyPlus is also capable of modeling multi-zone spaces. It was proposed building the supermarket model using multiple zones to separately describe the conditions in the refrigerated cases area, back of house spaces, and the rest of the store. EnergyPlus is capable of calculating interzone airflow. In addition it can calculate some temperature variation within a zone, from effects such as stratification and underfloor or sidewall air distribution, however to fully model airflow within a zone it would be necessary to use CFD modeling.

2.3 eQuest

The eQuest program is a DOE-2.2 based software developed by James J. Hirsch & Associates in collaboration with Lawrence Berkeley National Laboratory, and calculates the hourly energy use of a building based on user inputted data about weather, building construction, and HVAC system. The then current version of eQuest was version 3.63, released in May 2009, with updates released in July.

In addition to the standard eQuest module there is a refrigeration module especially designed to provide hourly analysis of supermarket performance. This module had not been updated since the version 3.61 release of the program in 2006.

The refrigeration module allows the user to build up a refrigeration system out of components, such as display fixtures, compressors, condensers, subcoolers, refrigerants, etc. Multiple circuits, compressors and compressor temperatures, and subcoolers can be modeled. The program models the effect of refrigeration design savings and their resultant impact on the HVAC system [7].

The capabilities of the program are not as extensive as those of EnergyPlus, and as the program had not been updated since 2006 and no new release was imminent it was assumed that it would not be as up to date with technology as EnergyPlus.

2.4 CyberMart

The CyberMart simulation program was developed at the Royal Institute of Technology, Stockholm, Sweden in 2005.

The program can predict the hourly energy usage of the supermarket refrigeration and HVAC systems and can determine the store's environmental impact, characterized by the Total Equivalent Warming Impact (TEWI), a focus likely due to the programs relationship to the IEA Annex 26 work. Several refrigeration system designs can be modeled, including direct systems, indirect systems, cascade systems and district cooling systems, however the range of options available for HVAC systems is somewhat limited, with chillers or district cooling being the cooling options, and oil or district heating as the heating options, and only a single system layout available. The program does not allow for multiple zones.

Cybermart comes with a database of approximately 180 refrigerated display cases from two manufacturers, however additional display cases can be created by the user by specifying the detailed cabinet data in a text file. Cybermart is limited in the choice for refrigerant, as it assumes uses refrigerant R 404A, and in its choice of compressors, allowing you to choose from three - Bitzer reciprocating, Copeland reciprocating or scroll. In addition, CyberMart can perform a life cycle cost analysis to compare investment and operational costs of alternative system designs [5,15].

Due to the restricted refrigerant options, and more importantly the limited HVAC functionality, this program was not considered suitable for the research being undertaken.

2.5 SST

The program SST was created by EPRI, and is an hourly building simulation program with a detailed model of a supermarket refrigeration system. The program was last updated to version 3.0 in 2000. The program uses hourly weather data to predict building loads and HVAC energy use. Detailed display case and cold room models consider the impact of indoor humidity levels on refrigeration loads, defrost requirements and anti-condensate heater operation. Simultaneously, the cooling and dehumidification provided by the cabinets is taken into consideration in the heat and moisture balance calculation. SST considers the use of heat recovery from the refrigeration system for space heating. The software interface allows the user to quickly assemble the components necessary to define the refrigeration, HVAC and building envelope systems. Components of a system such as compressors or cabinets are represented as icons and can be defined or modified by clicking on the item. The performance of each component is modeled using algorithms consistent with available data from manufacturers [8]. The program includes a library of components, however at this point the library is likely long out of date.

The capabilities of the program are not as extensive as those of EnergyPlus, and the program is no longer being updated, so it was not considered the best option for this project.

2.6 SuperSIM

The program SuperSIM was created by Brunel University in the U.K. It is a TRYNSYS based program, incorporating a large number of component models to create an hourly simulation program allowing the modeling of both primary and secondary refrigeration systems, as well as predicting building loads and HVAC energy use. The major component models include the compressor, air-cooled condenser, thermostatic expansion device, display cabinet and control. The program allows the use of water cooled refrigeration systems, heat rejection to a common water loop, and the use of both integral and remote refrigeration units [9].

The program had not yet been widely adopted, and while it was under development verification evidence had not yet been sighted. As a result this was not considered the most appropriate program for this project.

2.7 Conclusion

EnergyPlus was selected as the primary simulation tool for the evaluation of supermarket performance.

Chapter 3

3 Model Validation

3.1 Test Supermarket Description

In order to confirm the modeling capabilities of EnergyPlus, particularly with regards to the refrigeration components, a supermarket was selected from available reported field measurements of energy use from a previously conducted EPRI test site, for use as a validation store. The data collected spans one year, and was recorded in 15 minute time steps. This store was not used as the basis of the prototypical store, but only for the validation of the modeling program.

The validation store was in the Midwestern US. At 40°N latitude, the climate includes sub-zero temperatures in winter as well as hot and humid conditions in summer. The 99% winter design temperature was -6°F, the 1% summer design dry bulb temperature is 92°F with a mean coincident wet bulb temperature of 75°F. The 1% design wet bulb temperature was 78°F. By comparison, the 1% design wet bulb temperature in Atlanta, GA, is 77°F.

The supermarket is fairly typical of full-service supermarkets in the U.S. A schematic diagram of the supermarket is given in Figure 1 and a summary of building characteristics is given in Table 1.

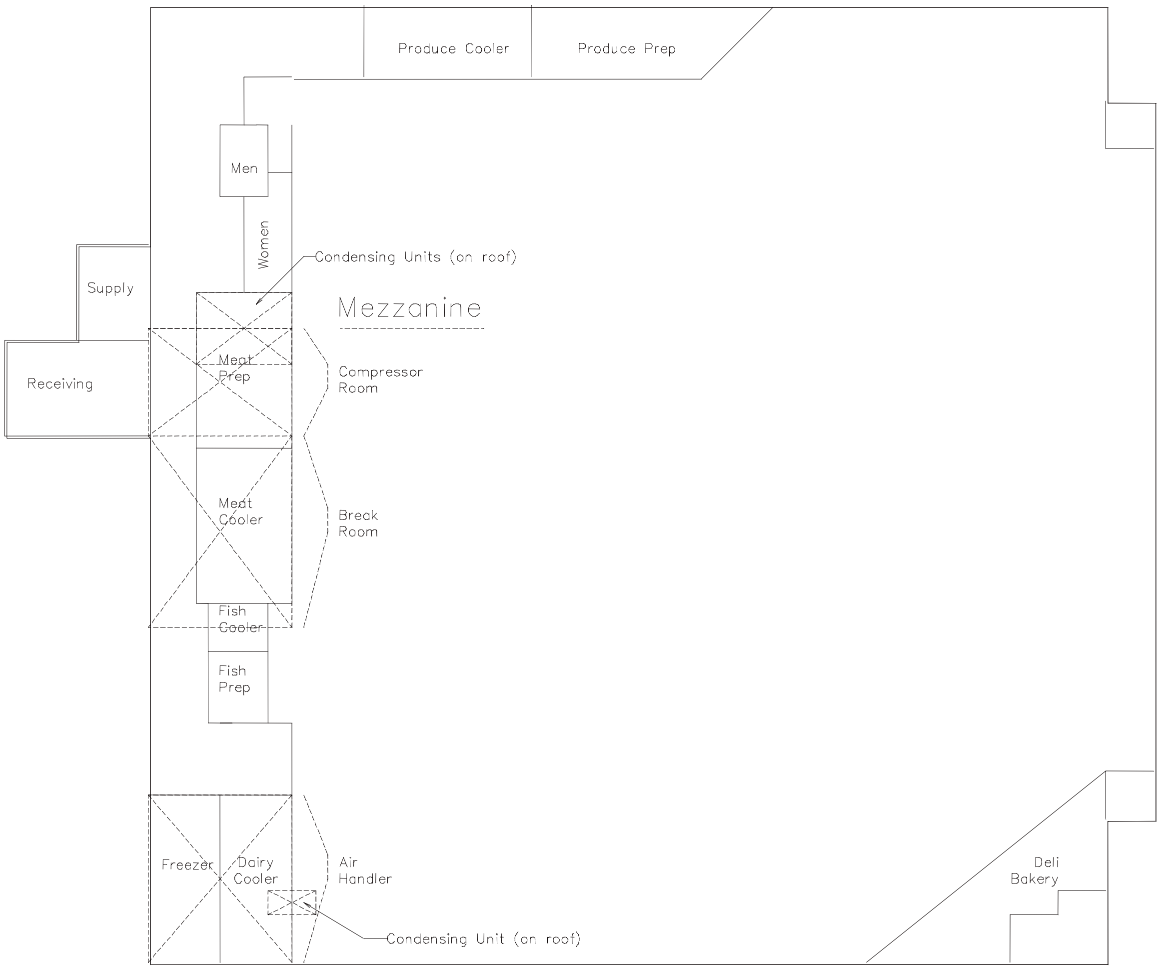


Figure 1: Schematic diagram of Validation Supermarket

The building was approximately rectangular and had a gross floor area of 42,000 ft^2 with a net sales floor area of 32,600 ft^2. The front entrance to the building faced east and delivery and receiving doors were located to the rear of the building. The store was open from 6:00 a.m. to 11:00 p.m. daily.

Table 1: General site information

Characteristic	Value
Floor area	42,000 ft2
Wall construction	12” concrete block
Roof construction	Metal deck, 3.75” Insulation above deck, gravel ballast
Floor	4” slab on grade
Window area	1140 ft2
Ceiling height	20 ft
Infiltration	0.24 ACH
Heating	Gas duct heater, with heat recovery from refrigeration system for reheat
Cooling	Packaged DX
Ventilation	Infiltration only
Occupancy	125 ft^2 per person
Lighting, sales area	2.7 W/ft^2
Internal heat gains, total	2.9 W/ft^2

Wall construction was typically 12-inch concrete block with varying interior finishes. Roof construction consisted of a metal deck covered with 3.75 in. insulation and gravel ballast. The floor was a 4 inch concrete slab-on-grade. The only glazing was located on the east side of the building.

3.1.1 Refrigeration System

The store contained 428 linear feet of open refrigerated cases, including 42 feet of low-temperature coffin cases, and two aisles of low temperature closed multi-deck cases having 88 total doors. In addition, there were 1810 ft^2 of low temperature walk-in freezers and 1510 ft^2 of medium temperature walk-in coolers. The refrigerated cases and walk-in units were served by three parallel compressor racks located on the mezzanine at the rear of the store. The refrigeration system had a total capacity of 125 tons, including a desuperheater water heater and mechanical subcooler.

Table 2 gives a summary of the refrigeration equipment in the store and its breakdown by rack.

A variety of other electrical equipment was present in the store. The deli/bakery area contained heated and non-refrigerated food cases, as well as processing and cooking equipment. The meat and seafood service areas also contained food processing equipment. General store sales area equipment included cash registers and several compressor-bearing refrigeration units for ice machine and beverage coolers.

Table 2: Refrigeration equipment summary

Compressor Rack A	**Split header, low temp / high temp**
Suction Temperature, F	-25 / 28
Total Refrigeration Capacity, MBH	573.6
Refrigerant	R - 22
Compressors	3 - 13.4 kW each
	2 - 6.16 kW each
	1 - 3.98 kW
Fixtures Served	
Walk in Storage, Sq. Ft.	1810
Closed Multi-deck, No. Doors	88
Open Counters, Linear Feet	42
Condenser	Hussman HACVI-8408M (2x4), Remote
Fans	8 - 1/2 hp each
Compressor Rack B	**Medium Temperature**
Suction Temperature, F	9
Total Refrigeration Capacity, MBH	464.7
Refrigerant	R - 22
Compressors	3 - 34.8 kW each
	1 - 5.05 kW
Fixtures Served	
Walk in Storage, Sq. Ft.	660
Open Counters, Linear Feet	196
Condenser	Hussman HACVI8408M (2x4), Remote
Fans	8 - 1/2 hp each
Compressor Rack C	**High Temperature**
Suction Temperature, F	18
Total Refrigeration Capacity, MBH	464.7
Refrigerant	R - 22
Compressors	3 - 34.8 kW each
	1 - 5.05 kW
Fixtures Served	
Walk in Storage, Sq. Ft.	850
Open Counters, Linear Feet	190
Condenser	Hussman HAVCI6408M, (2x3), Remote
Fans	6 - 1/2 hp each

3.1.2 HVAC System

The HVAC system at the site consisted of a condensing unit located on the roof, and a single constant-volume air handler located on a mezzanine above the walk-in coolers. The condensing unit contained two parallel compressor systems, each of which was connected to a direct expansion coil circuit in the air handler. The two coil circuits were "stacked" to form a single face-split coil. Each compressor had one level of unloading, giving four stages of cooling capacity control. (Note that when unloaded, each compressor still delivered its refrigerant through the full evaporator coil.) A summary of the HVAC system is provided in Table 3.

Table 3: HVAC system summary

Characteristic	**Value**
System Type	Single Path
Heat Recovery from Refrigeration System	yes
Refrigerant	R - 22
Cooling Capacity, MBH	798
Condensing Unit	Trane RAUC C804BA002DF
Nominal Tons	80
Compressors	2 @ 29KW Each
Cooling Stages	4
Coil Area, Ft. Sq.	126.4
Fins Per Foot	168
Number of Rows	3
Condenser Fans	8
Air Flow, CFM	49600
Air Handling Unit	Trane CLCH-3A, DX Unit
Supply Air Flow, CFM	31,400
Outdoor Air Flow, CFM	0 (closed by store mgmt)
Blower	30"Forward Curved, 648 rpm
Blower Motor	30 hp, 37 F.L.A.
External Static Pressure, iwc	2.75
Coil Area, Sq. Ft.	60.5
Number of Rows	4
Fins per Foot	104

A schematic representation of the HVAC system air handler is shown in Figure 2. The air handler mixed two return air streams with an outdoor air stream and drew the air through cooling coil. Rejected heat from the refrigeration condensers was available for heating using a heat recovery coil downstream of the supply fan. Supplemental heating was provided by a gas-fired duct heater.

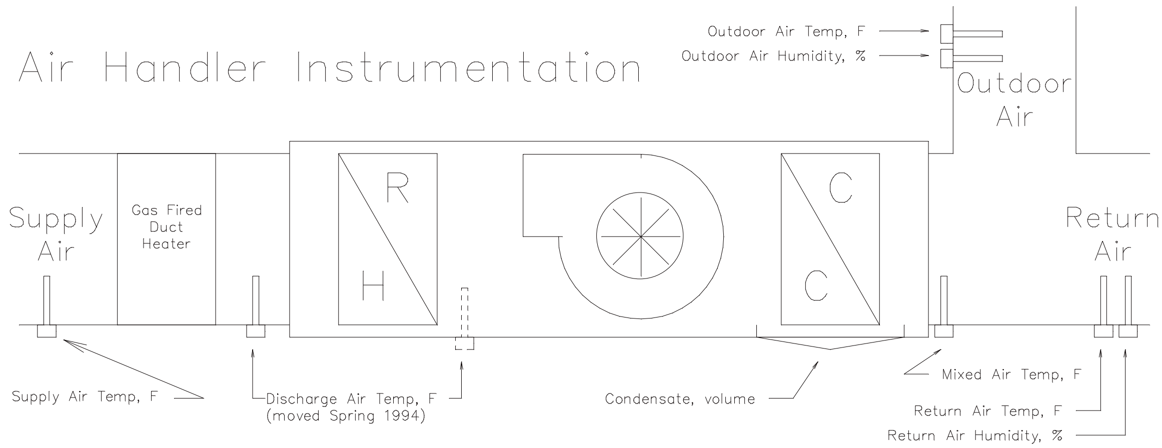


Figure 2: Schematic diagram of HVAC system air handler

The air handler and air distribution system were originally designed to deliver 31,400 cfm, or 0.96 cfm/ft^2 of sales area. A review of the test and balance report indicated that the actual airflow was 28,452 cfm, delivered by a 30 hp blower motor across a differential static pressure of 2.74 iwc. However, it must be noted that the outside air damper was closed when the testing was undertaken, eliminating all ventilation airflow. The store management team believed that there was adequate infiltration to maintain indoor air quality.

3.1.3 Instrumentation and Data Acquisition System

Instrumentation was installed at the site to measure electrical energy consumption and to characterize the HVAC system loads and equipment performance. A total of twenty measurements were recorded and stored at 15-minute intervals throughout the monitoring period.

In addition to total site electrical energy use, end-use power measurements were obtained to isolate the energy use by refrigeration compressors, HVAC equipment, and lighting. Refrigeration compressor energy use was further subdivided to separate low-temperature compressors from medium-temperature compressors.

Heating, cooling, and dehumidification characteristics of the store and the performance of the HVAC equipment were obtained by monitoring of the HVAC system air handler as shown in Figure 2. Temperature and humidity instrumentation was placed in the outside and return air streams. During the winter operation, separate temperature sensors were placed downstream of the heat recovery coil and the gas-fired heater, as shown in the figure. In the spring of 1994, the "discharge air temperature" sensor was moved to upstream of the heat recovery coil to allow separate identification of the cooling coil and heat recovery coil energy transfer to the air stream.

Data from the instrumentation were recorded and stored using a Synergistic Model C180E data recorder, manufactured by Synergistics Control Systems, Inc. The data recorder can record up to 16 current transformer inputs, 16 digital inputs, and 15 analog inputs. Power consumption is measured by detecting instantaneous potential and current at a distribution box, giving true RMS voltage, current, and power, as well as apparent power, power factor, and energy use. Power measurements have an accuracy of ±0.5% of reading. Analog inputs are measured using 4-20 mA transducers to an accuracy of ±0.25% full scale.

3.2 EnergyPlus Simulations

An EnergyPlus model was developed to simulate the performance of the test supermarket. The simulation model was developed through the following process.

1. The EnergyPlus model of the prototypical supermarket described in Section 4 was used as the starting point for model development.

2. Physical characteristics of the test building floor plan and envelope were described.

3. Internal heat gains from lighting and equipment were described using general operating schedules and measured energy from the site. Lighting power was directly measured. Other electrical heat gains were estimated using total building power measurements, less the measured power for refrigeration, HVAC, and lighting. It should be noted that these internal

heat gains were 2.9 W/ft^2 and were considerably higher than the gains initially assumed in the prototypical supermarket.

4. The capacity and length of refrigerated cases in the model were matched to the information from the test store. Note that the details of evaporator fan power, case lighting, and anti-sweat heater power were not available. Instead, the characteristics of the Hussmann cases used in the prototypical store were used. A similar approach was taken for the walkin coolers.

5. The capacity of the refrigeration compressors and condensers in the model were matched to the information from the test store. All other characteristics of the equipment were modeled using the Copeland compressor characteristics modeled in the prototypical store. A minimum condensing temperature of 110°F was used for all racks to provide sufficient heat reclaim. The specific value was calibrated to match the measured energy use profile over the course of the year.

6. The HVAC equipment was modeled using the generic DX system in EnergyPlus with a rated efficiency matching that of the site system. A refrigeration heat reclaim coil from the medium temperature refrigeration rack is integrated into the HVAC system. As noted in the description of the test site, the HVAC system provided no outdoor air for mechanical ventilation as the outside air damper had been closed.

7. While some limited weather data was available at the test site, the data was not complete enough to drive the energy simulations. Rather, simulations were performed using TMY2 weather data for the supermarket location. It must be noted that the actual summer at the site was hotter than the typical year.

When calibrating the simulation model, the single largest calibration factor was the miscellaneous electrical load in the store. Data from the test site lacked detailed information about the various equipment and process loads in the bakery, deli, meat, seafood, produce prep areas, or back of house area, and without detailed information about compressor-bearing coolers, spot merchandize lighting, checkout equipment, or office equipment. For the purposes of this validation, the electrical consumption and heat gains from these miscellaneous loads were obtained from the measured "other" loads by subtracting the calculated refrigerated case loads and HVAC supply fan load, and making simple assumptions for back of house and office loads.

3.3 Comparison of Measured and Simulated Performance

It was important to compare measured and simulated performance in order to validate that EnergyPlus would be capable of the simulations that were planned for this project.

The validation of the EnergyPlus simulation was performed by comparing the results of the simulation to the measurements at the test site. Comparisons were made between monthly energy consumption, indoor temperature and humidity conditions, and HVAC system heating and cooling load profiles.

Figure 3 shows a set of comparisons between simulated and measured energy consumption. The figure shows total building electrical consumption and various energy end-use consumptions, including the refrigeration system, HVAC cooling, HVAC heating, lighting, and other electrical sources. Due to configuration of the electrical system in the test store, it was not convenient to explicitly measure the electrical consumption associated with the refrigerated cases. As a result, the data for the refrigeration system includes only the compressor racks and condensers. The electricity consumption for case evaporator fans, case lighting, and anti-sweat heaters is included in the "other" category. Similarly, heating and cooling energy does not include the supply fan, which is included in the "other" category.

The results showed excellent comparison. The error in total electricity consumption for each month was less than 3.5% and the annual error was 1.1%. For the electrical end-uses of refrigeration, cooling, lighting, and other, all annual differences between measured and simulation consumption were less than 5%. Maximum difference in monthly end-use consumption was typically less than 10%.

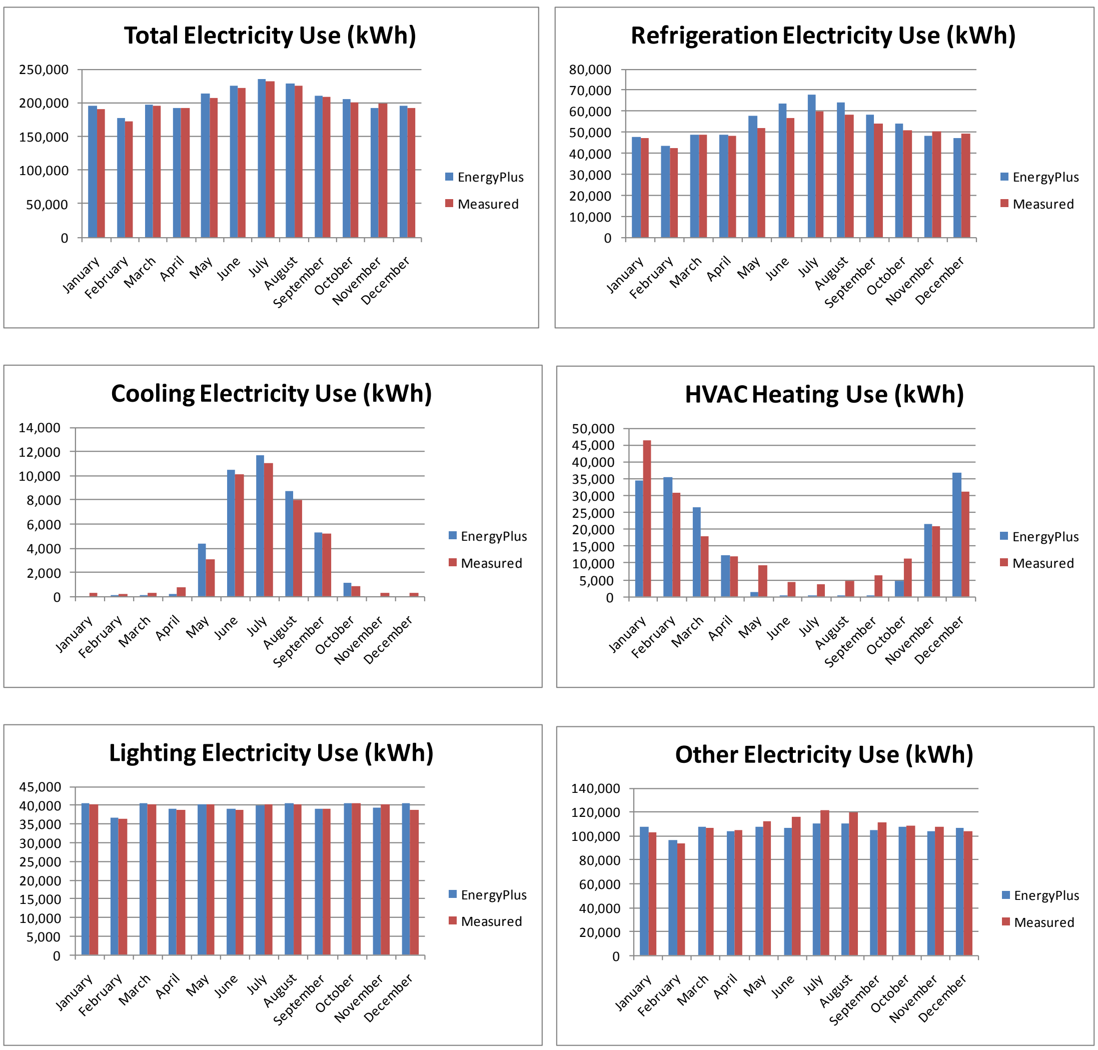


Figure 3 Energy use comparisons between measurement and simulation

The largest differences in end-use consumption were associated with heating and cooling. These larger differences reflect the inherent sensitivity of these energy end-uses to other store characteristics. Cooling electricity use is particularly sensitive to outdoor conditions, miscellaneous heat gains, and the cooling produced by the refrigerated cases, (i.e., case credits). The cooling energy use is also relatively small compared to lighting, refrigeration, and other. To the extent the cooling loads are largely the difference between the large heat gains of lighting and miscellaneous heat gains and the large cooling effect of the refrigerated cases, it is not surprising that cooling electricity use is difficult to calibrate. Heating energy is also very sensitive to the large cooling effect of the refrigerated space and the large heating effect of the internal heat gains.

Small adjustments to either the internal heat gains or the refrigeration load can have a large impact on heating and cooling energy use. For example, lighting energy use in May is approximately 40,000 kWh. A 2% change in lighting energy use is 800 kWh, which directly increases the cooling load by the same amount. If the cooling system has an EER of 11, the cooling electrical consumption would increase by 250 kW. Since the measured cooling energy use in May is 3000 kWh, the 2% change in lighting produces an 8.3% change in cooling energy use.

The HVAC comparisons were confounded by the fact that the simulations are performed with weather data from typical meteorological year which does not necessarily represent the year of measured data. One approach to account for the differences in weather conditions is to plot the daily average sensible heating and cooling loads as a function of the outdoor temperature. These plots for both measured and simulated sensible loads are shown in Figure 6. The plots show the same basic trend, with similar rate of change of sensible load with temperature. The slope of this line is often called the Building Load Coefficient (BLC) or overall building loss coefficient. Despite the scatter in the data, the measured and simulated values are within 13%.

The comparisons were further compounded by the recognition that the outdoor temperature measurement at the site was located in the outdoor air intake to the HVAC system. Unfortunately, since the store operators closed the outdoor air dampers and relied on infiltration to maintain indoor air quality, the outdoor air sensor was positioned in stagnant air on the sunlit roof. Anecdotal comparisons with historical weather data from the nearby airport suggest that the measured temperatures at the site were approximately 6°F warmer than actual outdoor air temperatures. Comparisons with annual measured data for the location obtained for the same year from Weather Underground [10] show a much better comparison of outdoor temperatures with the TMY data as shown in the following plots of mean temperatures, CDD and HDD.

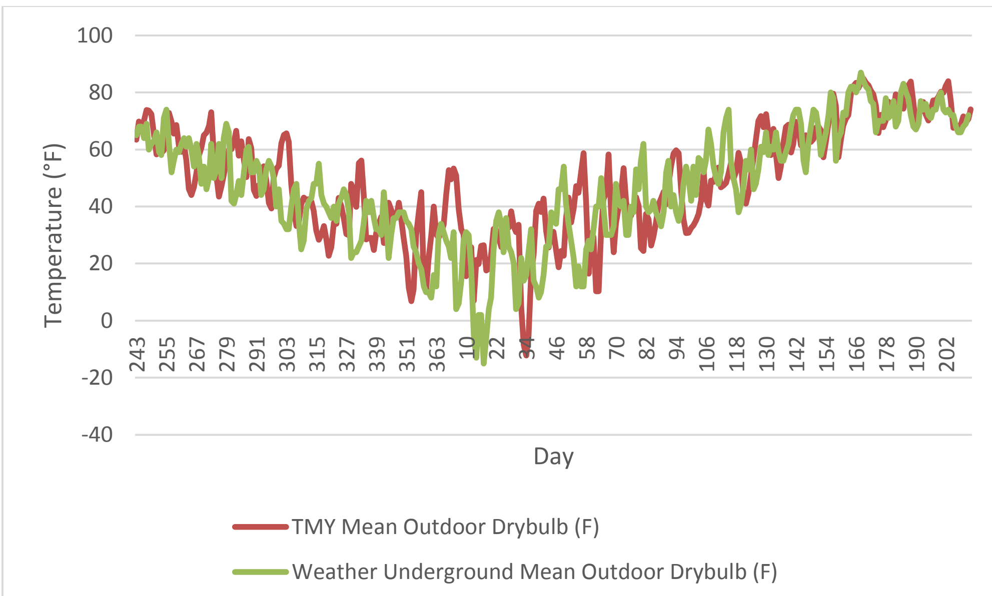


Figure 4 Daily mean temperature, Weather Underground measured vs TMY data

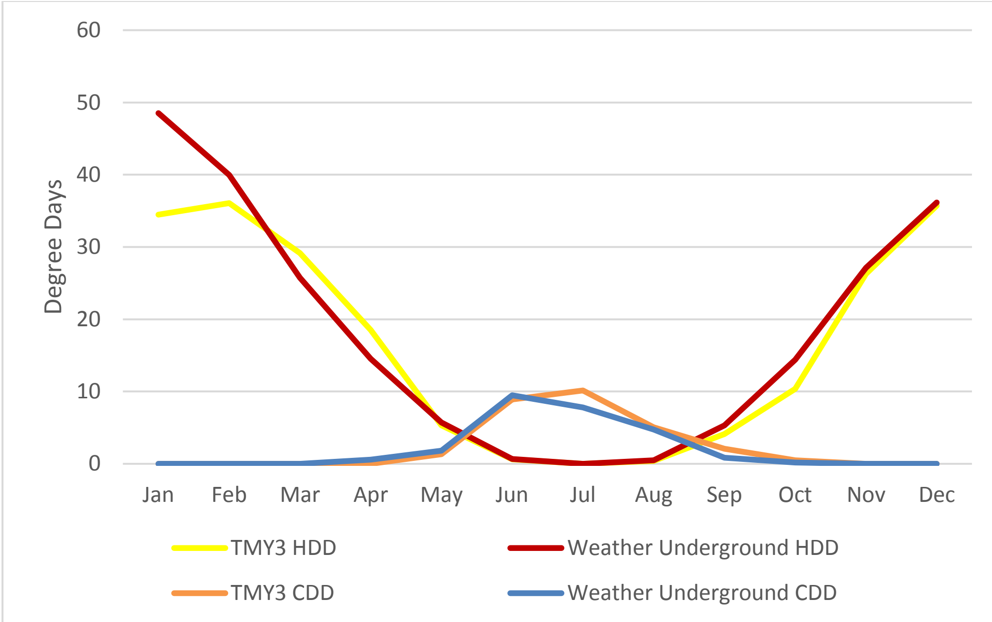
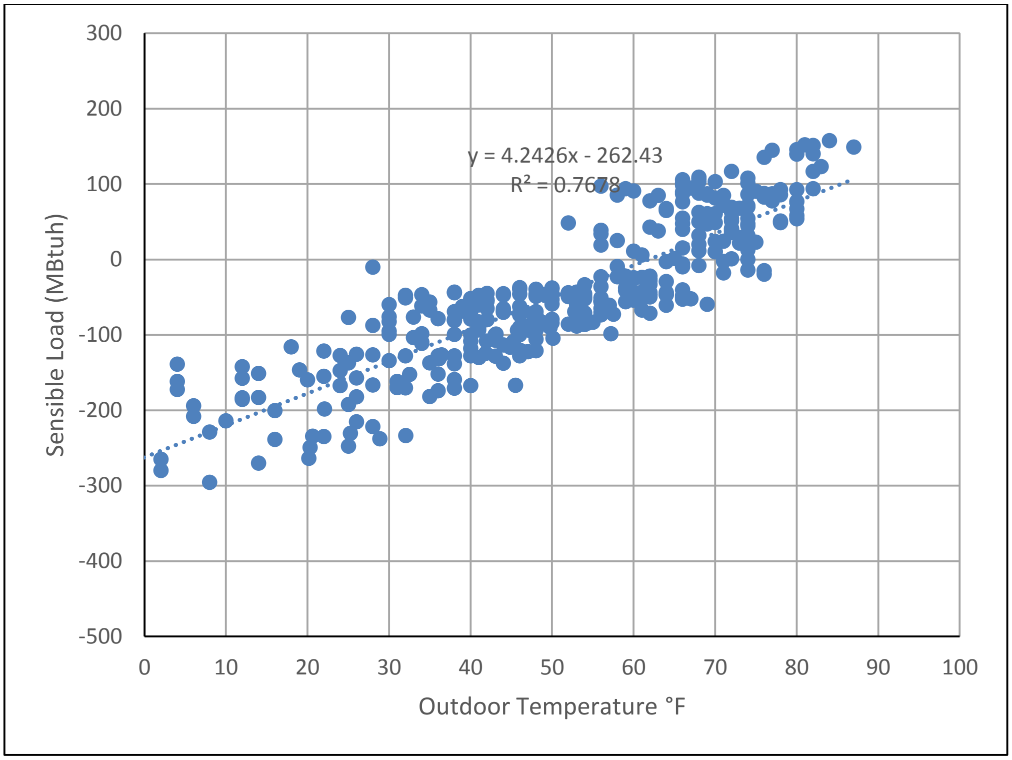


Figure 5 HDD and CDD for validation year, Weather Underground measured vs TMY data

When this Weather Underground data is used as the weather data against which to plot the measured data, the formulas for BLC slopes are extremely close. Some more scatter is seen in the measured data, as discussed above cooling energy use is difficult to calibrate, and heating energy use is very sensitive to assumptions made about the refrigerated space, so it is likely the additional scatter in the measured data is due to assumptions made about the makeup and operation of the store and refrigeration rather than the weather.



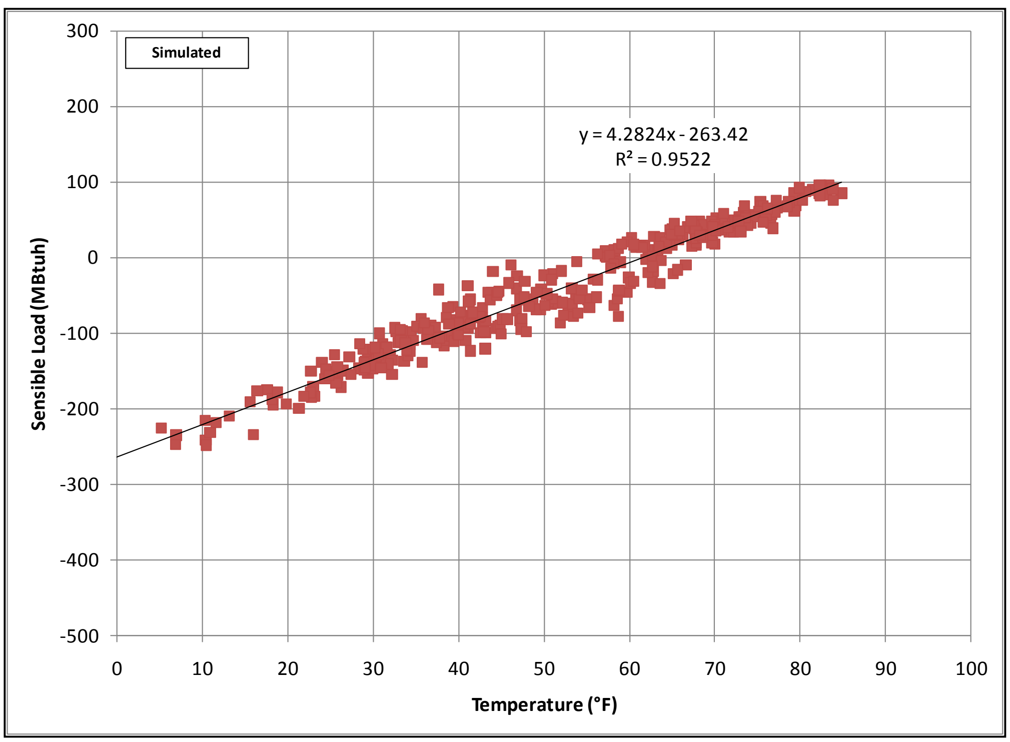
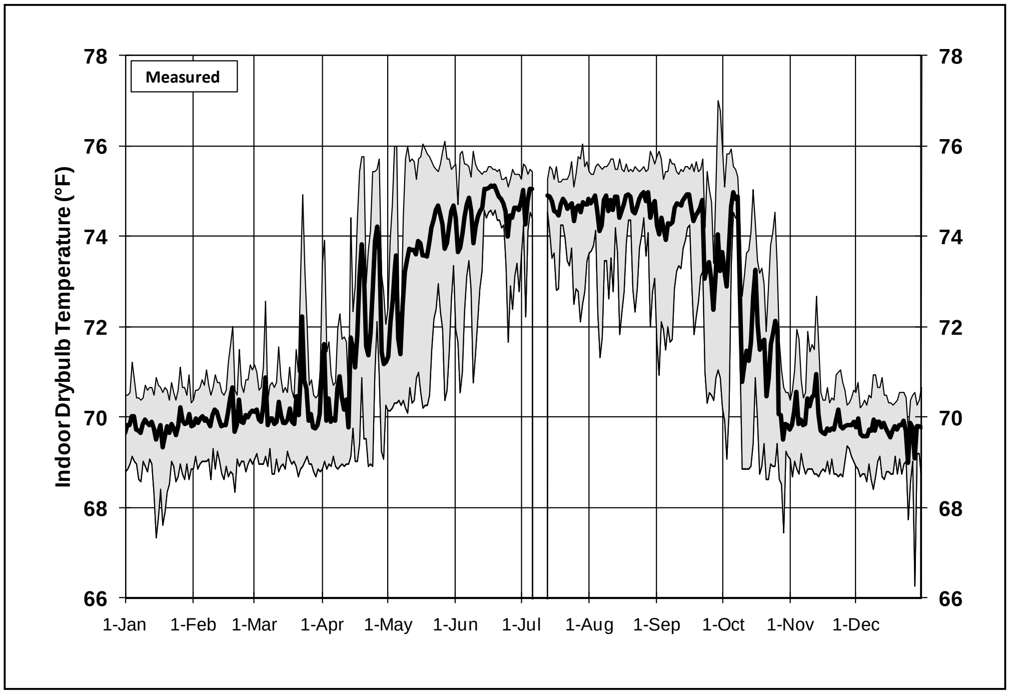


Figure 6 Sensible load vs. outdoor temperature, measured and simulated

As a final comparison of the measured and simulated results, Figure 7 and Figure 8 show the comparison between indoor drybulb temperatures and indoor dewpoint temperatures, respectively. Each figure shows the average daily temperature and the daily range. While the results show similar trends, the simulation results suggest that the simulated HVAC system often overcools the space in the summer in an effort to maintain the humidity setpoint. This was helpful to us in the development of the project as it showed that EnergyPlus was capable of modeling the humidity variances that were caused in the store by the HVAC system and refrigeration system working in tandem, and also gave an important point of focus for result checking as simulations continued, to determine if items in the parametric analysis negatively affected store comfort conditions.



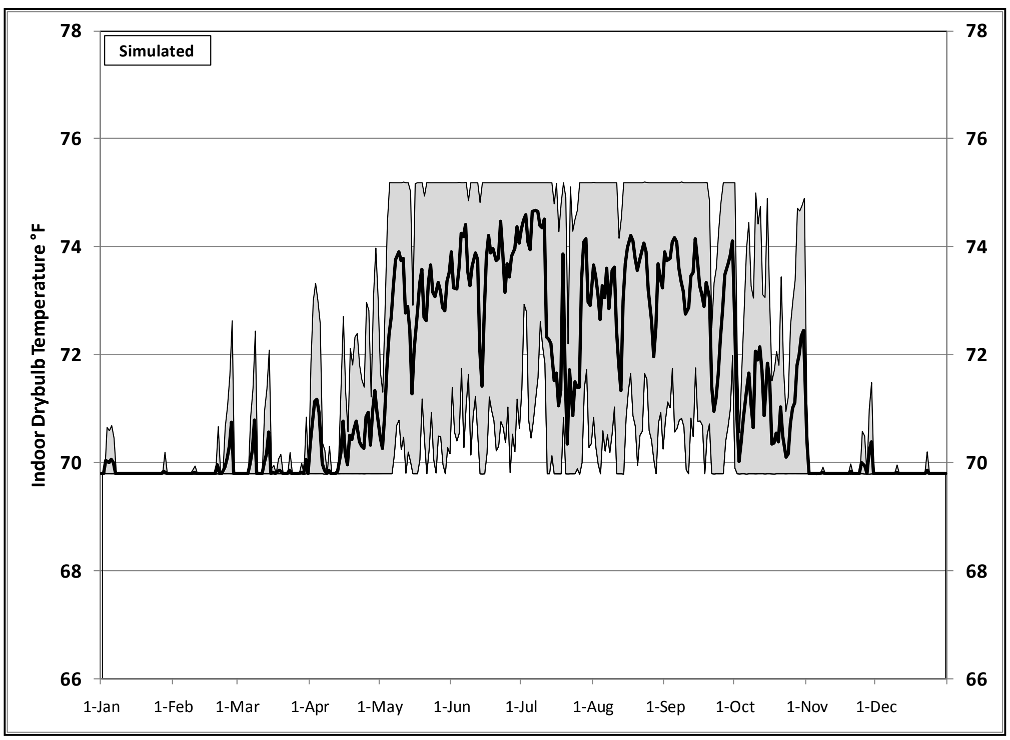


Figure 7 Indoor drybulb temperature, daily average and range, measured and simulated

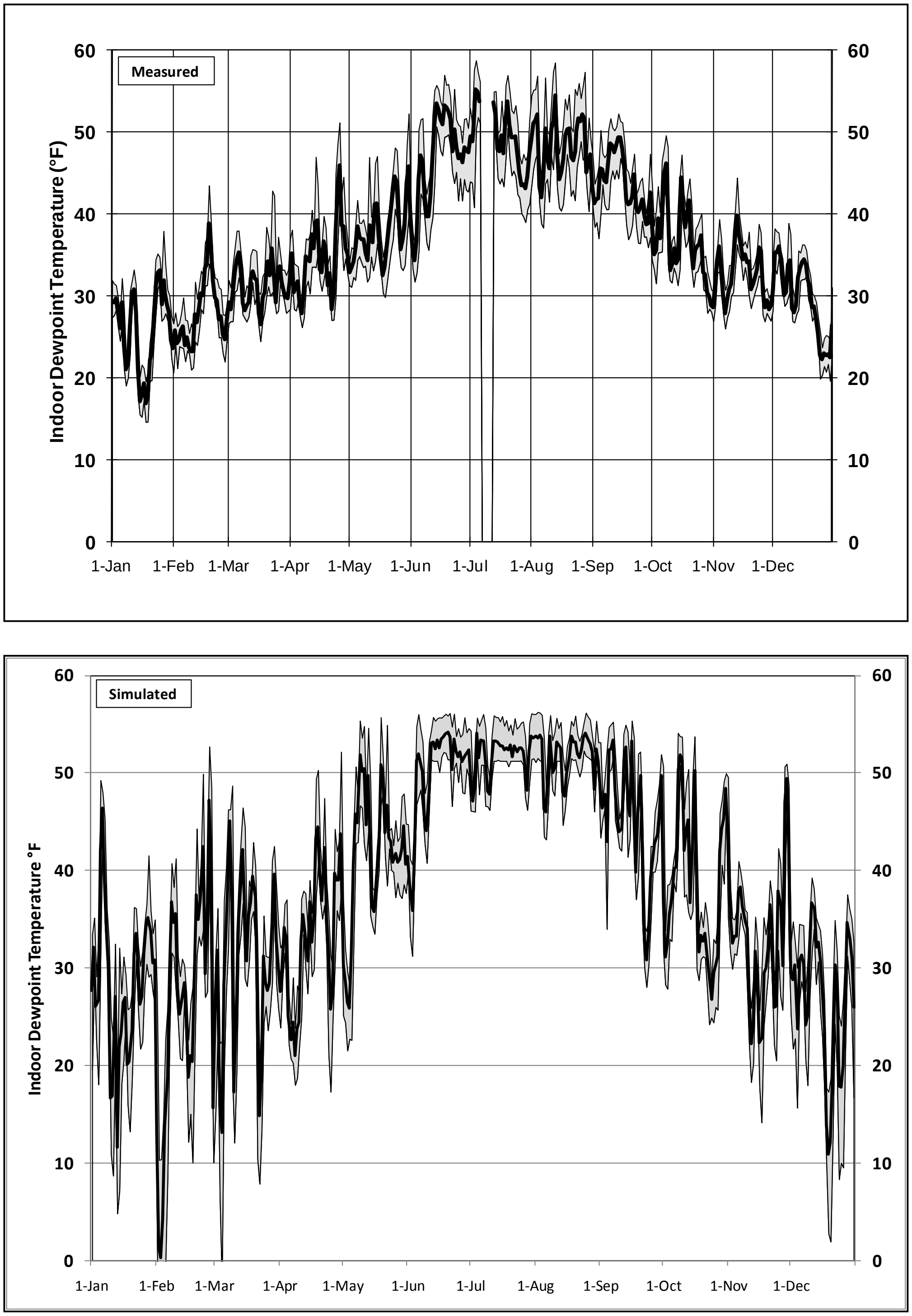


Figure 8 Indoor dewpoint temperature, daily average and range, measured and simulated

Chapter 4

4 Prototype Store

4.1 Prototypical Supermarket Characteristics

The assessment of energy saving opportunities from optimal design and operation of supermarket HVAC and refrigeration systems required the identification and specification of a “typical” supermarket as a context for the assessment. As such, a typical store was developed which could be used as a baseline to represent current design practice for new construction of supermarkets. The description of the prototype includes such characteristics as floor area, floor plan, ceiling height, wall construction, window areas, infiltration rates, occupancy schedules, lighting levels, and internal heat and moisture gains.

A three-step approach was taken to defining the supermarket prototype model. First, the work at the DOE national laboratories to develop a set of “building benchmark models,” which describe typical commercial building characteristics in the US was drawn upon. The DOE developed the DOE Commercial Buildings Benchmarks V1.2 [13], and the supermarket model was made use of as a first step. Second, practitioners in the supermarket industry were surveyed to obtain information about supermarket design. Finally, the results of the industry survey were integrated with the DOE benchmark supermarket model to develop the prototypical models.

4.1.1 DOE Benchmark Model

The DOE Commercial Building Benchmark Models (Version 1.2) are complete descriptions of buildings and their systems based on data from the 1999 and 2003 Commercial Buildings Energy Consumption Survey (CBECS). There are benchmark models of three vintages of sixteen different building types that represent over 70% of the commercial buildings in the US.

The DOE benchmark models include a supermarket with a floor area of 45,000 ft^2 schematically shown in Figure 9. This floor plan was retained for the prototypical store, although the deli and bakery zones were joined.

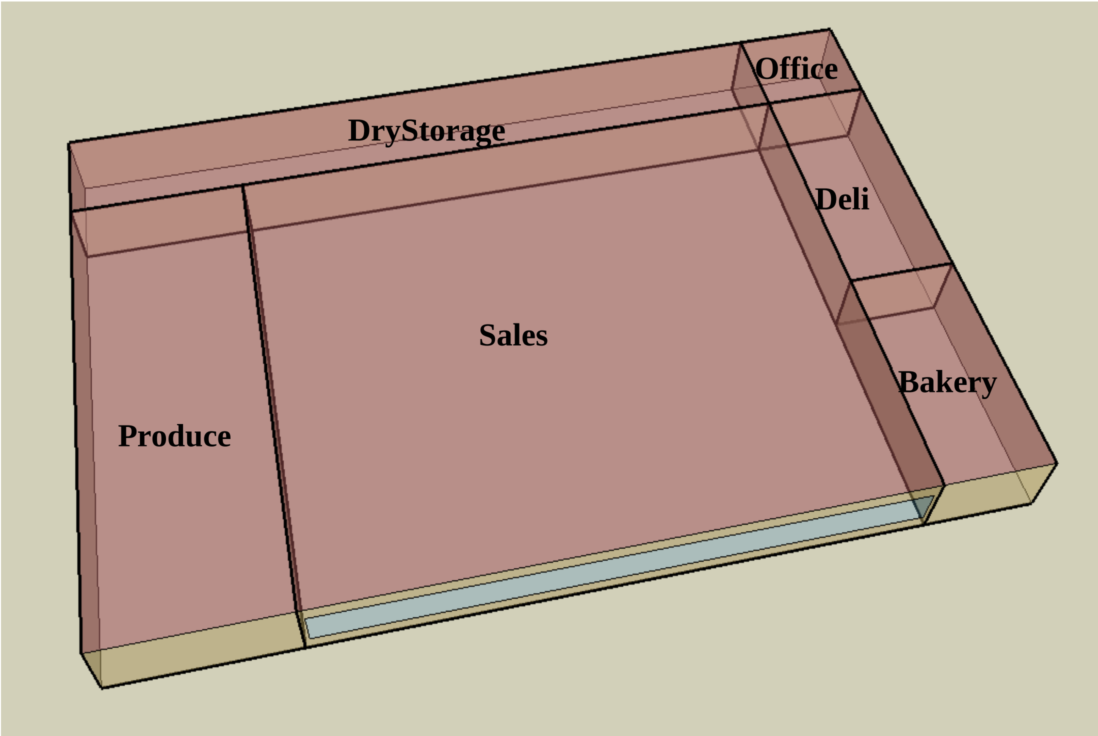


Figure 9: Prototypical supermarket layout

The models include separate descriptions for new construction, construction since 1980, and construction prior to 1980. There are separate models for each of sixteen climate zones, representing the energy code requirements for envelope insulation and window performance, these were rechecked against the latest code requirements. The models also include detailed descriptions of hourly profiles for occupancy, lighting, and internal heat and moisture gains, and descriptions of HVAC and refrigeration equipment in the store. The basic characteristics of the DOE building model are shown in Table 4. The benchmark models are ultimately expressed as complete EnergyPlus input files for each location.

Table 4: DOE benchmark model characteristics

Characteristic	**Value**
Floor area	45,000 ft²
Wall construction	8" CMU
Roof construction	Insulation entirely above deck
Floor	4" slab on grade
Window area	1880 ft²
Ceiling height	20 ft
Infiltration	0.25 ACH
Heating	Gas furnace
Cooling	Packaged DX
Ventilation	Constant air volume
Occupancy	130 ft² per person
Lighting, sales area	1.7 W/ft²
Internal heat gains, total	1.27 W/ft²

4.1.2 Survey of Supermarket Retailers

To supplement the DOE benchmark models, a limited survey of supermarket characteristics was undertaken with operators. The aim of the survey was to determine the type of refrigeration and HVAC systems, the typical layout of refrigeration fixtures within a store, as well as the typical lineup of medium- and low-temperature refrigerated cases. At the same time, the survey presented the opportunity to compare other building features that were represented by the DOE benchmark models.

Survey information was obtained from twelve different supermarket retailers. Some of the information was from older stores and other information was incomplete. The discussion here describes the characteristics of seven different stores, and illustrates the range of characteristics.

Table 5 shows the range of installed refrigeration system capacities for the seven stores, which range in size from 40,000 ft^2 to a "superstore" with 180,000 ft^2. Interestingly, the superstore (Store D) is in the middle of the pack for installed refrigeration capacity.

Table 5: Refrigerated case capacities from survey

	Medium Temp (MBH)		**Low Temp (MBH)**	
Store	**Sales Floor**	**Back of House**	**Sales Floor**	**Back of House**
Store A	571	207	155	44
Store B	906	80	114	54
Store C	422	104	145	40
Store D	746	96	224	60
Store E	662	215	424	83
Store F	791	166	235	44
Store G	955	168	322	88

Each of the following paragraphs describes the layout of the refrigerated cases in the store floor plan, the general type of cases, and the type of HVAC system. In the layouts shown below, the red indicates low temperature cases, while the yellow indicates medium temperature cases.

Store A: Refrigeration system: 778 MBH of total medium temperature cases, with 207 MBH of this located in back of house (BOH) area; 199 MBH of low temperature cases, 44 MBH located in BOH area. Layout shows walkins located in BOH area. Refrigeration covers three sides of the sales area perimeter. Freezers with doors are located at one end of the store. Most refrigerated cases are open, with some coffin cases. The HVAC system consists of a desiccant system serving the freezer aisles, and a conventional DX system serving the rest of the store.

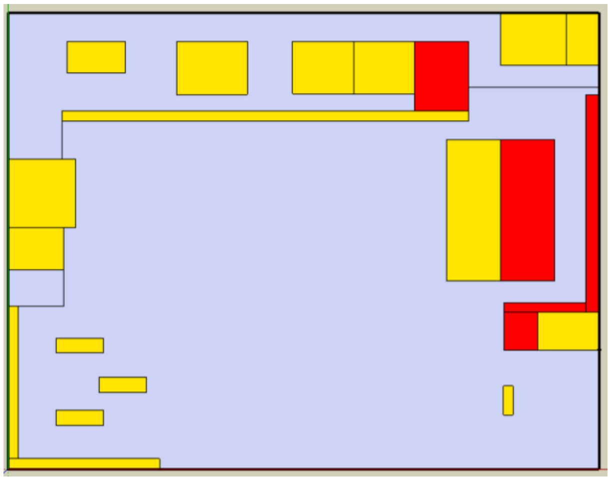


Figure 10: Store A layout

Store B: Refrigeration system: 986 MBH of total medium temperature cases, with 80 MBH of this located in BOH area; 168 MBH of low temperature cases, with 54 MBH located in BOH area. The store includes 70 MBH of dual temperature cases, all located in front of house area. Layout shows walkins located in BOH area. Dairy cooler is walkin in rear, with doors on front side for public use. Refrigeration covers three sides of the store perimeter. Freezers with doors are located towards the middle of the store. Most cases are open, with some coffin cases. HVAC system uses packaged DX cooling with face split coils.

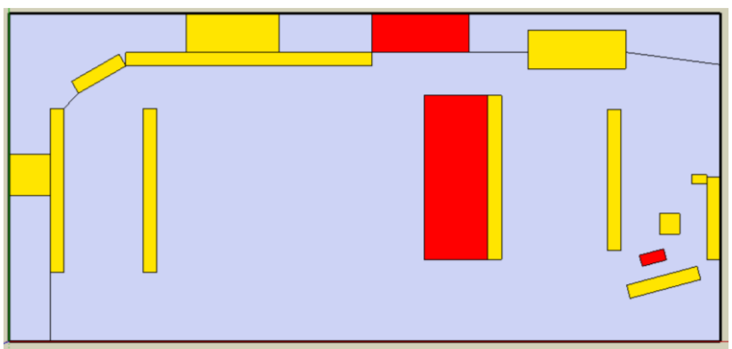


Figure 11: Store B layout

Store C: Refrigeration system: 526 MBH of medium temperature cases, with 104 MBH of this located in BOH area; 185 MBH of low temperature cases, with 40 MBH located in BOH area. Layout shows walkins located in BOH area. Refrigeration covers three sides of the store perimeter. Freezers with doors are located towards middle of the store. Most cases are open, with some coffin cases. The HVAC system consists of an air cooled condensing unit, a single constant volume AHU, reheat from the refrigeration system for heating and supplemental gas fired duct heating.

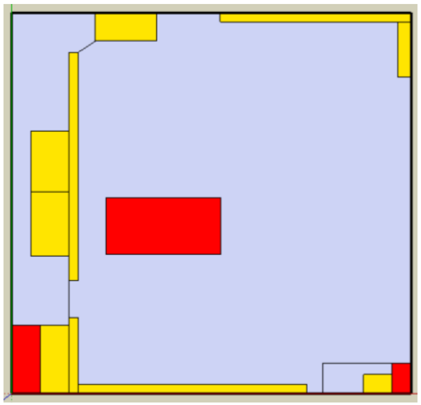


Figure 12: Store C layout

Store D: Superstore with 180,000 ft2 floor area. Refrigeration system: 842 MBH of medium temperature cases, with 96 MBH of this located in BOH area; 284 MBH of low temperature cases, with 60 MBH located in BOH area. Layout shows walkins located in BOH area. Refrigeration covers three sides of the grocery area of the store. Freezers with doors are located at one end of the store. Cases are typically open. Grocery area comprises approximately 22% of superstore, or 40,000 ft2, which is similar to the entire size of most of these supermarkets.

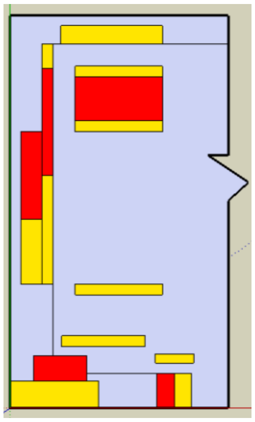


Figure 13: Store D layout, grocery section

<u>Store E:</u> Refrigeration system: 877 MBH of medium temperature cases, with 215 MBH of this located in BOH area; 507 MBH of low temperature cases, with 83 MBH located in BOH area. Layout shows walkins located in BOH area. Refrigeration covers three sides of the store. Freezers with doors are located towards middle of the store. Most cases are open, with some coffin cases. The HVAC system included one AHU serving the major portion of the store, this AHU included under case air return. Small units served peripheral areas of the store.

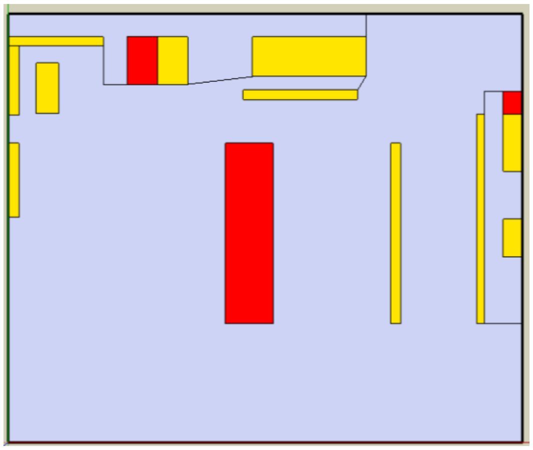


Figure 14: Store E layout

Store F: Refrigeration system: 957 MBH of medium temperature cases, with 166 MBH of this located in BOH area; 279 MBH of low temperature cases, with 44 MBH located in BOH area. Dairy cooler is walkin in rear, with doors on front side for public use. Refrigeration covers three sides of the store perimeter. Freezers with doors are located towards the middle of the store. Most cases are open, with some coffin cases. HVAC system uses packaged DX cooling with face split coils.

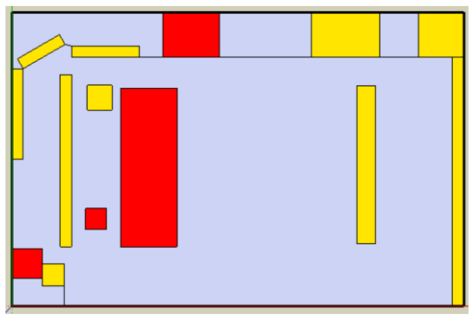


Figure 15: Store F layout

Store G: Large supermarket with 100,000 ft2 floor area. Refrigeration system: 1123 MBH of medium temperature cases, with 168 MBH of this located in BOH area; 406 MBH of low temperature cases, with 88 MBH located in BOH area. Layout shows walkins located in BOH area. Dairy cooler is walkin in rear, with doors on front side for public use. Refrigeration covers two sides of the store. Freezers with doors were located towards the middle of the store. Most cases are open multideck, with some coffin cases. Air conditioning is provided by fifteen rooftop units. A single unit is used to serve the freezer aisles, however no specific dehumidification was added to this unit. The units are air source heat pumps with heat reclaim from the refrigeration system to provide heating and additional gas fired heating as required.

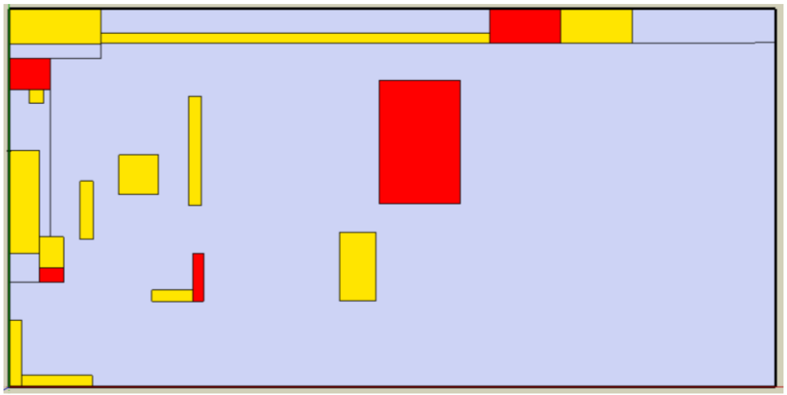


Figure 16: Store G layout

4.1.3 Supermarket Prototype Definition

The supermarket prototype, which was used as the base case for the analyses of this project, was based on the building information from the DOE Benchmark Model, code compliance, and characteristics derived from the survey of supermarket retailers.

The schematic floor plan of the prototypical model conformed generally to the DOE benchmark supermarket model, and was as shown in Figure 17 with the addition of the refrigeration fixtures. The general characteristics of the building are given in Table 6. It was assumed that the supermarket was new construction and conformed to ASHRAE Standard 90.1-2010. As a result, several of the specific characteristics were dependent on the location of the supermarket, including wall insulation, ceiling insulation, and window U-value and SHGC. HVAC equipment performance will also be specified to conform to the Standard.

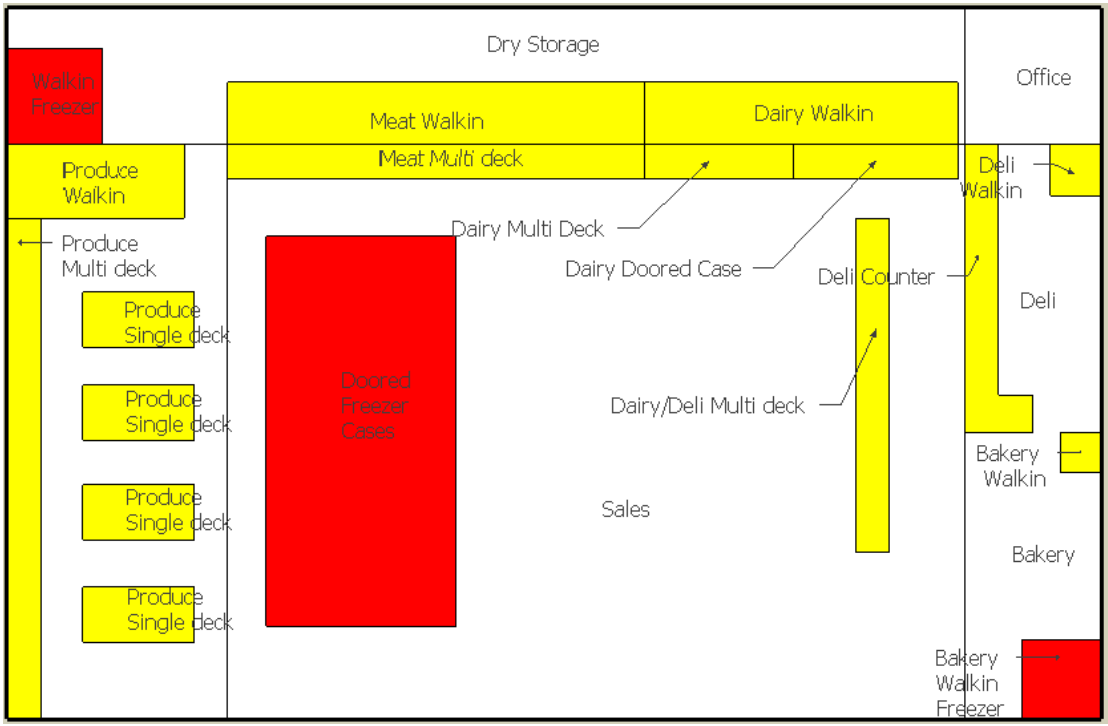


Figure 17 Prototypical supermarket layout

Zones were served by separate units with local supply and return. However there is air mixing between the zones, and this was a factor that was determined to be important, particularly as only the sales zone was initially dehumidified.

Table 6: Prototypical supermarket building characteristics

Characteristic	**Value**
Total floor area	*45,000 ft²*
Zones	
Sales area	*25,026 ft²*
DryStorage area	*6,695 ft²*
Produce area	*7,650 ft²*
Deli & Bakery area	*4,491 ft²*
Office area	*958 ft²*
Wall construction	*8" CMU*
Roof construction	*Insulation entirely above deck*
Floor	*4" slab on grade*

Characteristic	Value
Window area	*1880 ft²*
Ceiling height	*20 ft*
Infiltration	*0.25 ACH at heating design*
Heating	*Gas furnace, with heat recovery from refrigeration system for reheat*
Cooling	*Packaged DX,*
Ventilation	*Constant air volume*
Occupancy	*130 ft² per person*
Lighting, sales area	*1.7 W/ft²*
Internal heat gains, total	*1.27 W/ft²*

Occupancy, lighting, and internal electrical and gas loads are given in greater detail in

Table 7. The characteristics are separated by zone. Schedules for occupancy, lighting, and equipment are assumed to be the same in all zones and are given in Figure 18.

Process loads in the deli area were assumed to be on the sales floor and open to the main air mass. However, other process loads, including the bakery, were assumed to be in the back of house area and not directly affecting the main store latent heat balance.

Table 7: Prototypical supermarket model characteristics, miscellaneous loads

Zone	Occupancy	Lighting	Electrical Loads	Gas Loads
Bakery	125 ft²/pers	1.7 W/ft²	11244 W	5622 W
Deli	125 ft²/pers	1.7 W/ft²	12105 W	6053 W
Dry Storage	300 ft²/pers	0.9 W/ft²	0.75 W/ft²	
Office	200 ft²/pers	1.2 W/ft²	0.75 W/ft²	
Produce	125 ft²/pers	1.7 W/ft²	0.5 W/ft²	
Sales	125 ft²/pers	1.7 W/ft²	0.5 W/ft²	

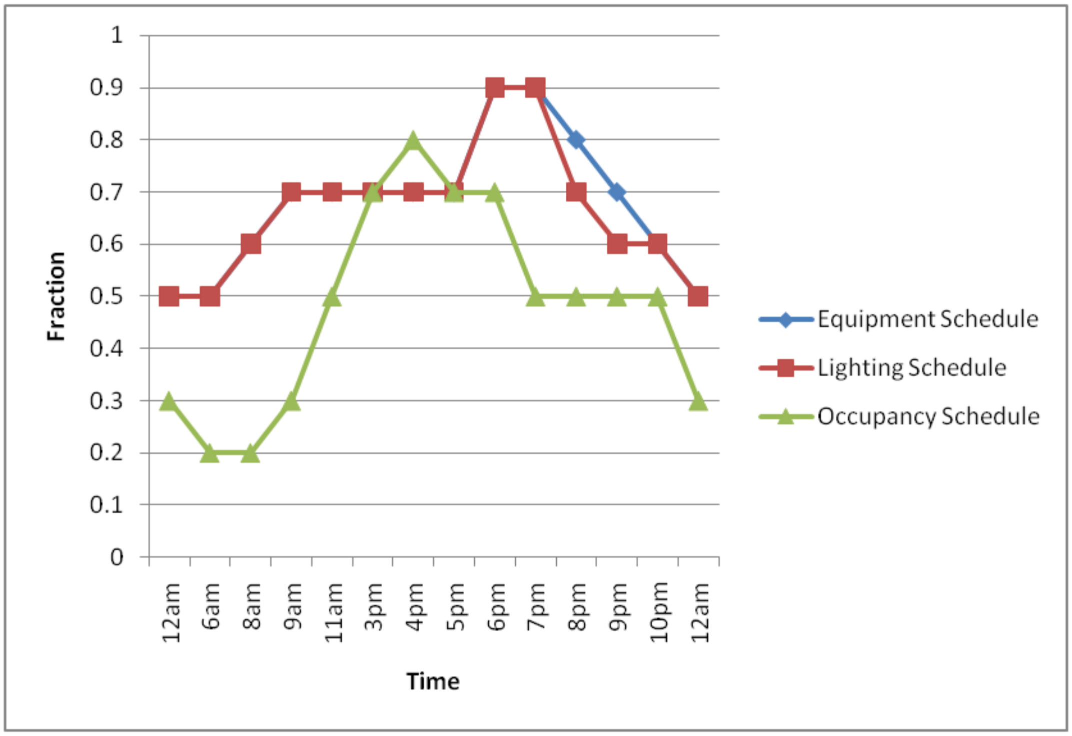


Figure 18 Prototypical supermarket schedules

The layout, case lineup, and sizing of the refrigeration system are based on the information from the industry survey. The refrigeration system model in EnergyPlus makes use of the range of capabilities of the program, including heat reclaim, subcooling, and head pressure control.

The refrigeration system had installed capacities of 900 MBH of medium temperature cases and 285MBH of low temperature cases. Note that these values were slightly higher than the DOE reference model values of 750 MBH of medium temperature and 300 MBH of low temperature cases, with considerably more medium temperature cases, and more closely reflects the recent trends observed in the industry. The refrigerated cases are distributed along three of the four perimeter walls of the sales area. Refrigerated cases were not uniformly distributed throughout the store, but were somewhat clustered towards one end.

The refrigeration system comprised parallel compressor racks for low and medium temperature loads. Each rack had a dedicated air cooled condenser. The components of the refrigeration system were described by design parameters and performance curves modeled after specific, widely-used products in typical new supermarkets. Each rack had four Copeland Discus compressors. Krack Levitor condensers were used to approximate the air cooled condenser curves. The compressors used on the medium temperature rack were Copeland-DISCUS-60HZ_R-22_MED_6DG3-3500-TSN. TSK, with a nominal 254,000 Btu/hr refrigeration capacity and 30.2 kW electrical power each. The compressors used on the low temperature rack were Copeland-DISCUS-60HZ_R-22_LOW_6DL3-2700-TSK, with a nominal 78,300 Btu/hr refrigeration capacity, and 16.1 kW electrical power each.

Heat recovery from a medium temperature rack provides heat to the sales area reheat coil. Heat recovery from a low temperature rack provides heat to the produce area reheat coil.

The low temperature and medium temperature refrigerated cases for the prototype were selected based on the industry survey described above. The refrigeration system serves a load of 98.7 tons (347 kW). Of the total refrigeration load, 25% was low temperature and 78% was installed in the sales area. The basic case line-up is given in Table 8 and the walk-in coolers are given in Table 9. The specific cases were modeled using the characteristics of the Hussmann Impact and Excel lines. Unit capacities of the walk-in coolers were based on the industry survey and modeled after Krack cooler characteristics.

Table 8: Prototypical supermarket refrigerated case line-up

Case	**Location**	**Type**	**Size**	**Rated Cooling Capacity (Btu/hr)**
Doored Freezer Cases	Sales Area	Impact RL	455 ft	229,264
Doored Dairy Case	Sales Area	DD6X-ULP	39 ft	18,766
Meat Case	Sales Area	M5X-GE	100 ft	154,161
Dairy Case	Sales Area	D5X-LRE	65 ft	114,265
Deli/Dairy Case	Sales Area	D5X-ULEP	80 ft	135,412
Deli Counter Case	Deli Area	C2X-E	75 ft	85,715
Produce Single Deck	Produce Area	P1X-E	108 ft	60,000
Produce Multi Deck	Produce Area	P2X-E	120 ft	130,995
			Total:	**928,578**

Table 9: Prototypical supermarket refrigerated walk-in line-up

Walk-in	**Location**	**Size**	**Rated Cooling Capacity (Btu/hr)**
Walkin Freezer	Dry Storage	360 ft²	32,020
Bakery Freezer	Bakery Area	360 ft²	23,615
Meat Walkin	Dry Storage	1345 ft²	72,046
Dairy Walkin	Dry Storage	1124 ft²	54,035
Deli Walkin	Deli Area	150 ft²	14,808
Bakery Walkin	Bakery Area	97 ft²	22,014
Produce Walkin	Produce Area	753 ft²	37,023
		Total:	**255,561**

The HVAC system for the prototypical store was a conventional packaged roof-top unit with gas furnace and air-cooled DX system with constant airflow and fixed ventilation. Ventilation rates for each system conformed to ASHRAE Standard 62.1-2010, with 7.5 cfm/person and 0.06 cfm/ft^2. Each zone was served by a separate unit. Temperature setpoints for heating and cooling in each zone were 70°F and 75°F, respectively. The HVAC units included a heating coil downstream of the cooling coil, and the system serving the sales area was controlled to maintain both temperature and humidity in the space. The primary heating coil in both the sales and produce zones used reclaimed heat from the refrigeration system condensers as a heat source, with gas backup.

The DX system in the sales zone was engaged when either the zone temperature or zone humidity is above setpoint. The nominal dehumidification setpoint was 55% relative humidity. In some cases, the zone may be overcooled to meet dehumidification requirements. If the zone temperature dropped to the heating setpoint, the heating coil will be engaged to maintain minimum zone temperature.

Chapter 5

5 Sensitivity Analysis

5.1 Initial Sensitivity Factors

The prototypical store that was defined in Chapter 4 served as a base case initially for a sensitivity analysis. This analysis was designed to both reflect the variations in common supermarket characteristics as well as recognize the sensitivity of performance results to modeling assumptions and system details. These items were intended to be variables that were expected to be of secondary importance to the latent heat balance of the supermarket and its systems.

The factors that were studied are described in the following table:

Table 10: Sensitivity Analysis Factors

Variable	**Description**
Operation Hours	Store open 24 hours per day
Night Curtains	Cover open cases with curtains at night
Occupancy	Increase base occupancy by 25%
Lighting	Increase lighting power density by 25%
Heating Setpoint	Increase heating setpoint by 2.7°F
Cooling Setpoint	Reduce cooling setpoint by 2.7°F
Heating Efficiency	Increase heating efficiency by 10%
Cooling Efficiency	Increase cooling efficiency by 10%
Fan Static Pressure	Increase fan static pressure by 10%

Additional analysis was conducted around the internal process loads, namely kitchen loads and produce mister loads, and the mixing between zones in the store.

5.2 Sensitivity Results

Exploratory simulations were performed on the prototype supermarket in Atlanta to examine the sensitivity of performance to a variety of characteristics.

Table 11 shows a summary of the results including changes in HVAC and refrigeration energy consumption, and changes in average temperature and humidity in the sales area compared to the base case. The average temperature and humidity are expressed in terms of the monthly conditions in January and July.

Table 11: Sensitivity Analysis Results

Variable	**Description**	**HVAC Energy Change (%)**	**Refrig. Energy Change (%)**	**Sales Temp Change, °F (Jan/Jul)**	**Sales Humidity Change, (%RH) (Jan/Jul)**
Operation	Store open 24 hours per day	-5.8	5.0	0.0/0.0	0.7/0.1
Night Curtains	Cover open cases with curtains at night	-10.1	-3.2	0.0/0.0	0.1/0.0
Occupancy	Increase base occupancy by 25%	0.8	0.2	0.0/0.0	0.6/0.3
Lighting	Increase lighting power density by 25%	-2.5	0.1	0.0/0.0	0.0/0.0
Heating Setpoint	Increase heating setpoint by 2.7°F	21.7	1.4	2.7/2.3	-1.7/-0.4
Cooling Setpoint	Reduce cooling setpoint by 2.7°F	1.6	-0.1	0.0/-0.2	0.0/0.0
Heating Efficiency	Increase heating efficiency by 10%	-8.3	0.0	0.0/0.0	0.0/0.0
Cooling Efficiency	Increase cooling efficiency by 10%	-0.7	0.0	0.0/0.0	0.0/0.0

Variable	Description	HVAC Energy Change (%)	Refrig. Energy Change (%)	Sales Temp Change, °F (Jan/Jul)	Sales Humidity Change, (%RH) (Jan/Jul)
Fan Static Pressure	Increase fan static pressure by 10%	1.5	0.0	0.0/0.0	0.0/0.0

5.2.1 Operating Hours & Night Curtains

Discussions with different retailers suggested that there is considerable variation in store operating hours. Most large supermarkets operate 24 hours per day, though many are closed at night. The base supermarket model initially assumed that the store was closed for eight hours each night, with reduced loads for lighting, equipment, and occupancy. Given that stores with 24 hour operation seemed to be the most common approach among newer supermarkets, the effect of continuous operation was examined with more modest reduction in internal gains. With 24 hour operation, HVAC energy use was reduced by 5.8% and refrigeration energy use was increased by 5.0% The HVAC reduction was driven by the lower heating loads due to the higher internal gains at night.

Interestingly, even in the stores that were not open 24 hours, none that were surveyed used night curtains on their refrigerated cases. Some said that they did not meet payback criteria; others note that they are not used if when they were installed. The impact of the curtains on open cases was examined. Results indicated significant reductions in both HVAC and refrigeration energy consumption. HVAC heating energy use was especially affected due to the reduction in the cooling effect of the cases. Refrigeration energy was also reduced due to the lower losses from the cases.

As a result of discussions with the retailers it was decided to implement 24 hour operation as this seemed to be dominant, and not to implement night curtains as these became irrelevant in a 24 hour store.

5.2.2 Occupancy and Lighting

Occupant density and lighting power density both affect the loads on the HVAC system. For the supermarket sales area, the base model assumed a general lighting power density of 1.7 W/ft^2 and an occupant density of 130 ft^2/person. Note that the lighting power does not include lighting within the refrigerated cases. Increasing lighting and occupancy by 25% had relatively small effects on overall HVAC and refrigeration energy consumption. Additional lighting reduced heating energy requirements and reduced overall HVAC energy by 2.5%. The additional occupancy had a slightly opposite effect – while it reduced heating requirements, the additional latent load increased dehumidification requirements and produced a slight net increase in HVAC and refrigeration energy use. Note also that the average relative humidity level in the sales area increased by a fraction of a percent relative humidity in both January and July.

5.2.3 Setpoints

Changes in heating and cooling setpoints had very predictable effects. Increasing the heating setpoint by 2.7°F (1.5°C) caused a 22% increase in HVAC energy use due to the greater heating energy consumption. It should be noted that the setpoint changes were applied throughout the supermarket. While the sales area captured condenser heat from the refrigeration system through a reheat coil in the air handler, other areas of the supermarket relied on natural gas furnaces for heating, including the produce zone in this earlier model.

Changes in the cooling setpoint had very little impact due to the humidity control requirements of the store. Through most of the summer, the HVAC system is operated to maintain the required indoor relative humidity. The relatively poor dehumidification performance of the conventional DX HVAC system used in the base store produced over-cooling of the space, requiring reheat that was engaged when the zone temperature dropped to the heating setpoint.

5.2.4 HVAC and Fan Energy

Changes to the efficiency and base EER of the heating and cooling equipment had predictable effects. Increasing efficiency reduced HVAC energy consumption with no impact on refrigeration energy consumption or indoor temperature and humidity. Note that the changes in efficiency did not affect the dehumidification performance of the HVAC equipment.

Changes in duct system design, requiring higher supply air pressure, also had predictable effects. HVAC energy was increased by 1.5% when fan static pressure requirements were increased by 10%.

5.2.5 Internal Process Loads

The load on the HVAC system, especially the cooling load, is produced by the balance between the steady heat removal by the refrigerated cases, the steady heat gains from internal loads, and the variable loads due to outdoor environmental conditions. In general, the internal heat and moisture gains due to occupants, lighting, and miscellaneous processes in the supermarket are a large component of the overall load. The impact of lighting and occupancy loads were discussed earlier. The following sections address two categories of sensible and latent process loads: the heat gain of kitchen equipment in the bakery and deli areas, and the moisture gains due to spraying of vegetables in the produce area.

<u>Kitchen Loads</u>

A survey was taken of the miscellaneous internal loads in several supermarkets to determine their agreement with the levels in the Benchmark model, after the internal loads in the Validation model were found to be significantly different.

The load profile that was determined was based on a combination of information from retail stores and analysis from the RP 1362 "Revised Heat Gain and Capture and Containment Exhaust Rates from Typical Commercial Cooking Appliances" Project. Based on the information from that project, which has been incorporated into the ASHRAE Handbook, and from previous research, convective and latent heat from cooking is exhausted via hoods and do not typically enter the kitchen. Hence the entire load from the kitchen equipment on the space is considered to be sensible.

The loads determined by RP 1362 for cooking appliances were based on standby heat gains, since the appliances may only be operating at rated power for a very small proportion of the time, or in some cases not ever. The heat gain to the space from an appliance does not necessarily correlate to either the rated input energy rate or the energy consumption rate under operating conditions and in most cases is considerably less. Because this standby rate has been used to determine the total load, which already takes account of the fact that the appliance is not operating at full load, the schedule is then able to be at 100% for the full day.

Produce Misters

Produce misters add humidity to the zone air, affecting moisture removal performance of the refrigeration and HVAC systems, and were identified by the PMSC as an issue that should be further addressed. Information received from manufacturers indicated that the required load was approximately 0.5 gallon of water delivered per foot of wet side produce case per day. With approximately 60 linear feet of wet produce case assumed in the prototypical store, the average increase in the continuous latent load was approximately 3000W.

5.2.6 Interactions Among Zones in Supermarkets

One challenging feature of supermarkets is that many diverse loads are experienced in a single, large, open area. Analysis using computational fluid dynamics (CFD), described more fully in the additional report included in the Appendix, indicated that significant humidity gradients could be maintained in supermarkets due to the variations in sensible and latent loads and the impact of sensible and latent removal rates of refrigerated cases. The results suggested that there could be opportunities for energy savings by designing HVAC systems with different characteristics and setpoints for the different zones in the supermarket. Our simulation analysis included analysis of these opportunities at a later stage of the project, including an exploration of supermarket designs that isolate low temperature refrigerated cases.

Nevertheless, it was recognized that there is some level of mixing within the open areas of a supermarket. Not all of the moisture generated in the produce section and not all of the heat generated in the bakery section remain exclusively in their respective zones. Some of the interzonal communication occurs due to the flow patterns of supply and return ductwork, pressure imbalances between air handlers, make-up air imbalances, infiltration air movement, and the natural circulation of people within the store. In some cases, the mixing occurs by design, for example, by exhausting excess air from the seafood area and encouraging makeup air to be drawn from the sales area to reduce unpleasant odors in the sales area.

Within EnergyPlus, there are several approaches to modeling multiple HVAC systems within a single space. In the simplest case, each zone is modeled as though there is no communication between zones, treating the boundary as an imaginary wall. A more complex analysis uses an airflow network, which treats each zone as fully mixed, but allows airflow exchange between zones and the outdoors based on driving forces that influence the pressure difference across openings. Such an analysis requires considerable knowledge about these driving potentials and the pressure balance among the systems. Alternatively, EnergyPlus also allows the user to specify an airflow rate between the fully mixed zones.

Given the uncertainty about the pressure balances within supermarkets, we analyzed the impact of interzonal communication by specifying exchange airflow rates. The analysis had two components: a review of the CFD analysis of air communication between zones which had been conducted, and a sensitivity analysis of the impact of the communication on the latent heat balance and energy consumption in supermarkets.

The sensitivity analysis was performed by exploring the impact of interzonal exchange on the HVAC energy consumption, refrigeration energy consumption, and zone relative humidity levels. For the purpose of this analysis two zoning configurations were considered. In the first case, the supermarket was zoned as per the prototypical store, with a single large sales area including both dry goods and freezer cases. Interzonal mixing occurred within the three main sales zones of the open store, i.e., the sales, produce, bakery/deli zones. Specifically, the produce and bakery/deli zones each exchanged airflow with the sales area. While further detailed CFD analysis could provide greater insight into the exact amount of air exchange, our analysis assumed that the amount of air exchanged is 20% of the air handler airflow of the smaller zone. For example, for exchange between the sales and produce zones, it was assumed that the mixing airflow between the two zones is in the amount of 20% of the produce zone air handler supply airflow.

A second mixing analysis was performed with the large sales area further subdivided to isolate the freezer aisles with a separate air handler. With six total zones, such a zoning configuration represented the largest number of air handlers expected in the analysis of this project.

The results of the analysis showed that the supermarket performance was influenced by the interzonal mixing, but that the impact was relatively small. Results are shown in Table 12. The table shows the change in energy and refrigeration energy consumption, and changes in the monthly average sales area temperature and humidity in January and July, with 20% interzonal air mixing for both the five zone and six zone cases. As expected, the zone temperatures and summer humidity were directly controlled by the HVAC system and show no significant changes. The humidity level in the sales area in January increased slightly with mixing as more humid air from the produce zone mixes with the sales zone. Note that the relative humidity in the produce area is typically about 5-15% RH more humid in the produce area due to produce misting.

With the relatively small changes in zone temperature and humidity, there is only a very small change in refrigeration energy consumption. However, HVAC energy consumption was more directly influenced by the mixing. With active humidity control in the sales area, the additional latent loads from adjacent zones increased HVAC cooling energy consumption by 1-3%. The six zone case showed higher energy consumption because the refrigerated cases were more isolated from other sales area loads and did not provide the same level of cooling of the sales area as in the five zone case. That is, there were more times during the year when the freezer aisles needed heating while the rest of the store needed cooling.

The results indicated that the indoor conditions and the energy consumption were only slightly influenced by interzonal mixing. While the number of zones can have an impact, especially within the sales area, the mixing among zones is less significant.

Table 12: Interzonal Air Mixing Analysis Results

Variable	Description	HVAC Energy Change (%)	Refrig. Energy Change (%)	Sales Temp Change, °F (Jan/Jul)	Sales Humidity Change, (%RH) (Jan/Jul)
Five Zones	Mix 20% of airflow with adjacent zones	1.8	0.01	0.0/0.0	0.3/-0.1
Six Zones	Mix 20% of airflow with adjacent zones	3.1	-0.3	0.0/0.0	0.1/-0.1

Chapter 6

6 Parametric Analysis

6.1 Parameters

Following the development of the initial prototypical store in Chapter 4, and the preliminary analyses in Chapter 5, a base case supermarket model was formed that modified the original supermarket prototype. The basic supermarket characteristics remain unchanged, but control parameters were adjusted to include 24-hour operation, 20% interzonal air mixing, and no night curtains on cases.

Given the base case model of the supermarket prototype, a set of 25 parameters were identified to explore the sensitivity of energy consumption and maintained indoor conditions to parameter values. The 25 parameters were identified as the set of potentially most significant parameters to affect the latent heat balance and resulting energy consumption. The parametric simulation analysis was performed by independently changing the value of each parameter from its “base” value to its “test” value. The twenty-five parameters, with their base and test values, are given in Table 13.

The set of simulations was performed in nine locations that are representative of the US climate zones - Miami, Phoenix, Atlanta, Los Angeles, St. Louis, Seattle, New York, Denver, and Minneapolis.

The results of these 234 runs (26 cases in 9 locations) are summarized here. The main discussion will focus on the results for Atlanta, with a brief discussion of significant changes in other locations. For context, we begin with a discussion of the base case across the nine locations.

6.2 Base Case Comparisons

Figure 19 shows the energy breakdown of the base case for each of the nine locations. The most noticeable result from Figure 19 is that refrigeration dominates supermarket energy consumption. Refrigeration energy use is far more significant than HVAC electricity use in all cases. For the Atlanta base case, refrigeration accounts for 83% of building electrical energy consumption, which is a far greater fraction of total consumption and a greater energy intensity than revealed in the measured data from the test store in Chapter 3. For comparison, the refrigeration energy consumption comprised about 67% of the electricity use of the validation supermarket. The overall site energy use intensity (EUI), defined as the end-use energy consumption per gross floor area, was 2250 GJ/m^2 for the validation store and 4370 GJ/m^2 for the Atlanta base prototype store.

Table 13: Preliminary Parametric Analysis Factors

Factor	Base	Test
Store floor area	45,000 ft^2	Increase 30%, uniformly through all zones
Building Envelope	Standard 90.1-2010	Increase insulation 30%
Miscellaneous sensible and latent gains	1.3 W/ft^2	Increase 30%
Infiltration	0.25 ACH at heating design	Increase 30%
Refrigeration capacity	98.7 tons (347 kW)	Increase 30%
Refrigerated case distribution	78% in sales area (Table 8)	Increase 30% in sales space, maintain cap
Display case line-up	25% low-temp (Table 8)	Increase 30% low temp (coffins), maintain cap
Defrost strategy	Electric (freezers only)	Hot gas with temp termination
Antisweat heater control	Linear	Constantly on
Case lighting	Base (Hussmann Impact and Excel)	Decrease 30%
Case sensible and latent cooling credit	Base (Hussmann Impact and Excel)	Increase latent credit fraction 50%
Air vs. evap cooled condensers	Air cooled	Evaporatively cooled
Mechanical subcooling	None	Add mechanical subcooling
Head pressure controls	122 F minimum	104 F minimum
Humidity setpoint	55%RH	40%RH
Zoning	Open sales area	Separate zone in sales area for freezers
Return air location	High level	Under case return
Ventilation control	Fixed	Demand controlled ventilation
HVAC airflow	Base	Decrease 30%
DX refrigerant coil control	Interlaced	Face split coils
Runaround coil (wrap-around heat pipe)	None	Heat pipe
Dedicated outdoor air systems	None	Dedicated outdoor air system
Desiccant dehumidification	None	Desiccant system
High performance model	Base	Increase insulation 30%, reduce sales floor lighting 30%

Factor	Base	Test
Case credit model	Base	Add 30% med temp refrigeration to sales floor, add 30% low temp as coffin cases, reduce case lighting 30%

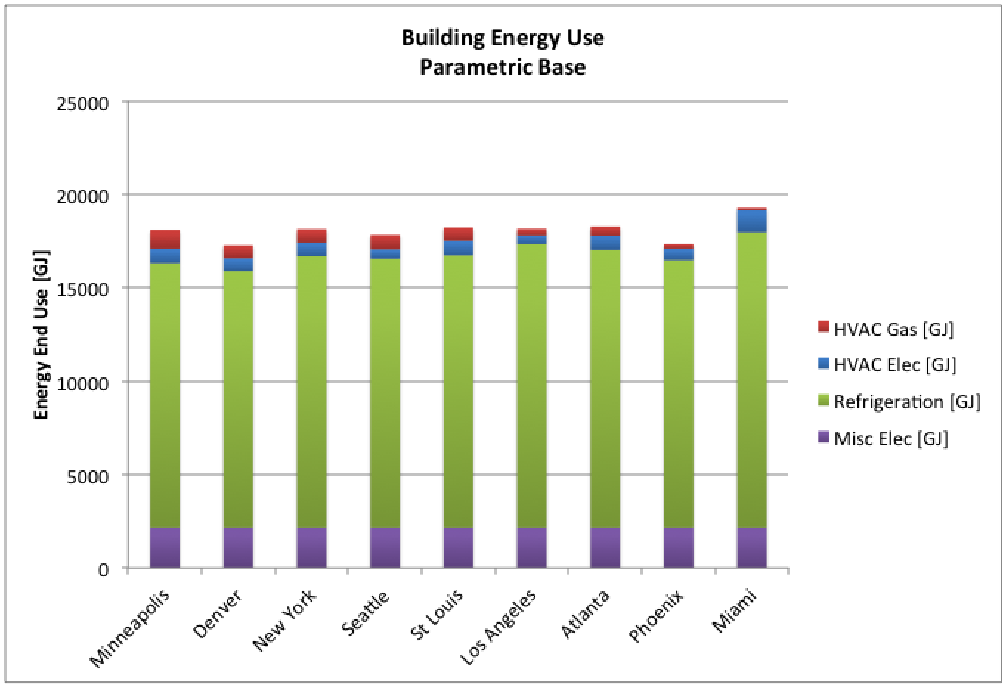


Figure 19 Base case energy performance in US locations

The main reasons for the difference in refrigeration energy use is that the base prototype store developed for this parametric analysis assumes that there is no mechanical subcooling and that the head pressure is maintained at 55°C. As discussed later in this chapter, the use of mechanical subcooling and floating head pressure can dramatically reduce refrigeration system energy use and produce energy consumption numbers that are more consistent with typical values observed in practice. It should be noted that these two factors affect refrigeration energy consumption, but do not have strong influence over HVAC or other energy use in the store. Nevertheless, a subsequent analysis of factors that most influence the latent heat balance, discussed in Chapter 7, establishes a new base prototype store with revised values.

In general, HVAC energy consumption is a small fraction of the total. The dominant element is typically fan energy use. A comparison of the base cases showed that very little cooling was required in Denver and Seattle due to the minimal dehumidification requirements. The cooling was lowest in Seattle due to the low outdoor temperatures year round. Cooling requirements were by far the largest in Miami, due to both the high outdoor temperature and humidity. While in most climates the HVAC electricity loads, a combination of the cooling loads and the fan loads, were significantly dominated by the fans, in Miami the cooling made up 61% of the HVAC loads. Despite having higher cooling loads, the lowest HVAC electricity loads were actually seen in Los Angeles due to lower fan energy.

Conversely in the heating gas usage Miami used the least, while Minneapolis used nearly 8 times as much gas. Denver used the least refrigeration electricity, while Miami used the most, a 15% increase over Denver. Denver also had the lowest total electricity usage.

Figure 20 shows monthly energy use for the base case in Atlanta. Refrigeration energy use varies with changes in outdoor conditions. While gas usage is significantly reduced in the summer, there is still some heating in the summer months, even in those zones without free heat; however it was less than 0.1% of that used in the peak months. Cooling in the winter months was off in all but the office space, which was again less than 0.1% of the peak cooling month. The lowest total electricity use was seen in February, since this is when refrigeration use is lowest, as well as cooling.

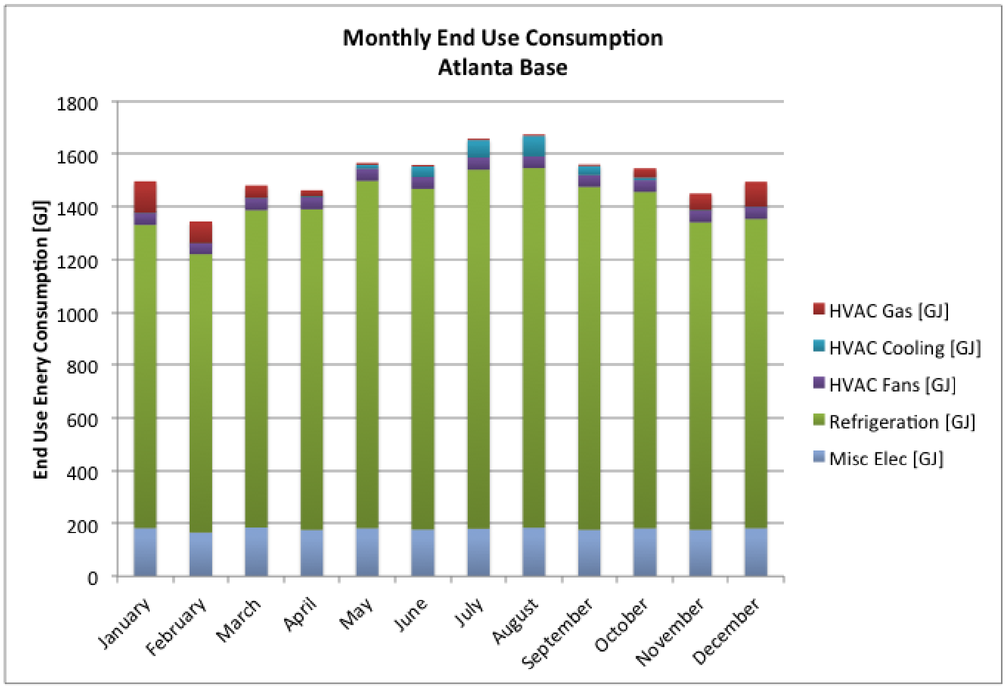


Figure 20 Monthly energy end use consumption, base case, Atlanta

Figure 21 shows that the temperatures in the sales zone almost always hovered near the heating setpoint, despite there being a 4°C deadband between the heating and cooling setpoints. Figure 22 shows that humidities were generally low in winter and rise toward the setpoint of 55% in summer, except in dryer climates such as Denver and Phoenix where they are below set point year round. In the summer, zone temperatures typically remain near the heating setpoint because the cooling coil is controlled to maintain both temperature and humidity setpoints. In all but the driest climates, the humidity control requirements dictate extended cooling coil operation, which drives indoor temperatures lower until reheat is needed to maintain the heating setpoint.

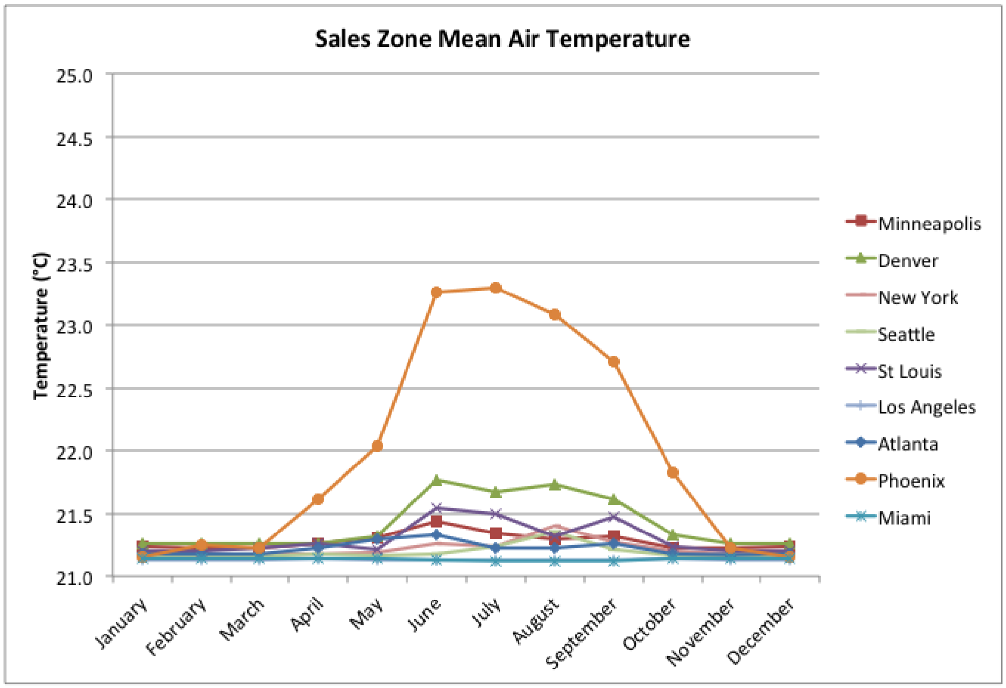


Figure 21 Mean air temperature in sales zone, base case, US locations

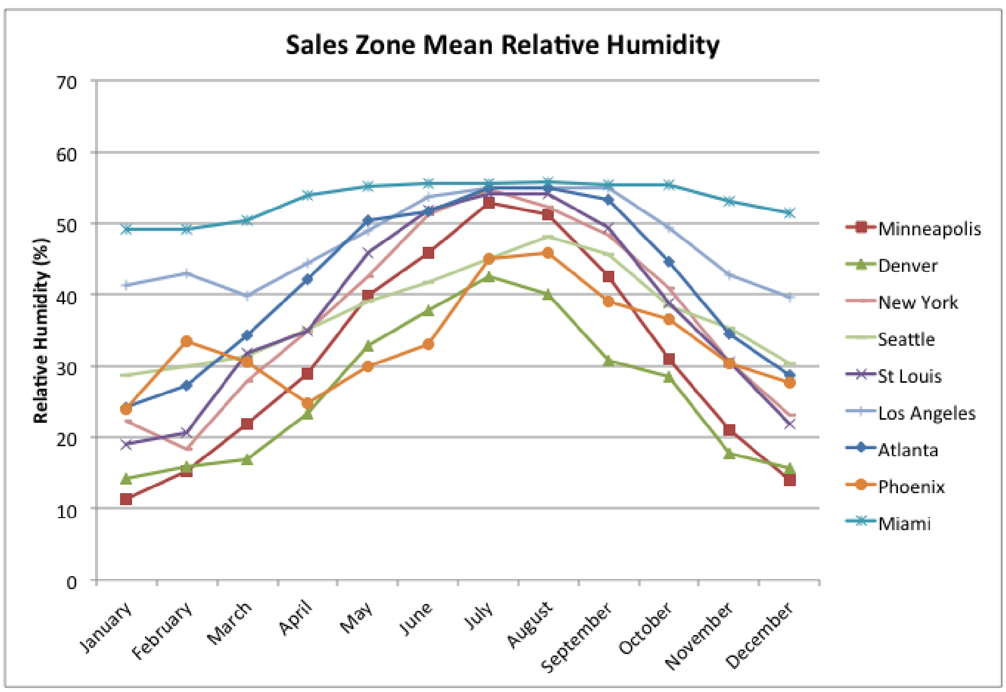


Figure 22 Mean relative humidity in sales zone, base case, US locations

6.3 Parametric Results

Table 14 shows energy performance for the base case and the 25 parametric variations for Atlanta. The results are shown graphically in Figure Figure 23 through Figure 27.

Table 14: Annual Energy Use Results – Atlanta

Case	Total Building Elec [GJ]	Refrigeration [GJ]	HVAC Fans [GJ]	HVAC Cooling [GJ]	HVAC Gas [GJ]
Base	17793	14850	535	246	482
Area	18557	14903	588	346	466
Envelope	17768	14846	517	243	458
Gains	18047	14875	536	257	394
Infiltration	17867	14859	569	276	515
Capacity	21698	18737	597	200	594
Case Distribution	18306	15305	546	292	267
Lineup	18706	15759	540	245	482
Defrost	17924	14981	535	246	482
Antisweat	18413	15472	533	246	482
Case Lighting	17603	14648	545	248	491
Credit Fraction	18127	15283	517	164	447
Evaporative	17799	14856	535	246	482
Subcooling	15151	12208	535	246	482
Head Pressure	12017	9073	535	246	482
Humidity	17597	14341	535	559	482
Zoning	17839	14849	526	302	496
Return	17759	14851	502	243	486
DCV	17743	14841	535	204	482
Airflow	17648	14838	412	235	502
Facesplit	17779	14850	535	232	482
Heatpipe	17741	14831	535	212	482
DOAS	-	-	-	-	-
Desiccant	17653	14917	545	28	1477
High Performance	17369	14840	515	241	501
Case Credit	18509	15511	546	290	267

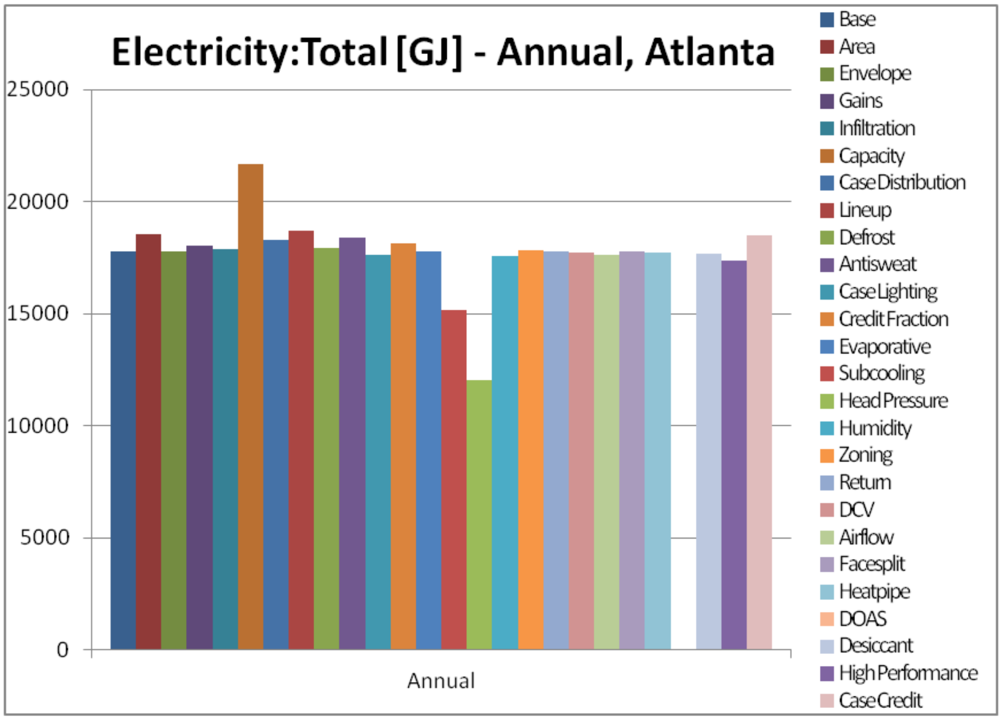


Figure 23. Electricity: Total [GJ] - Annual, Atlanta

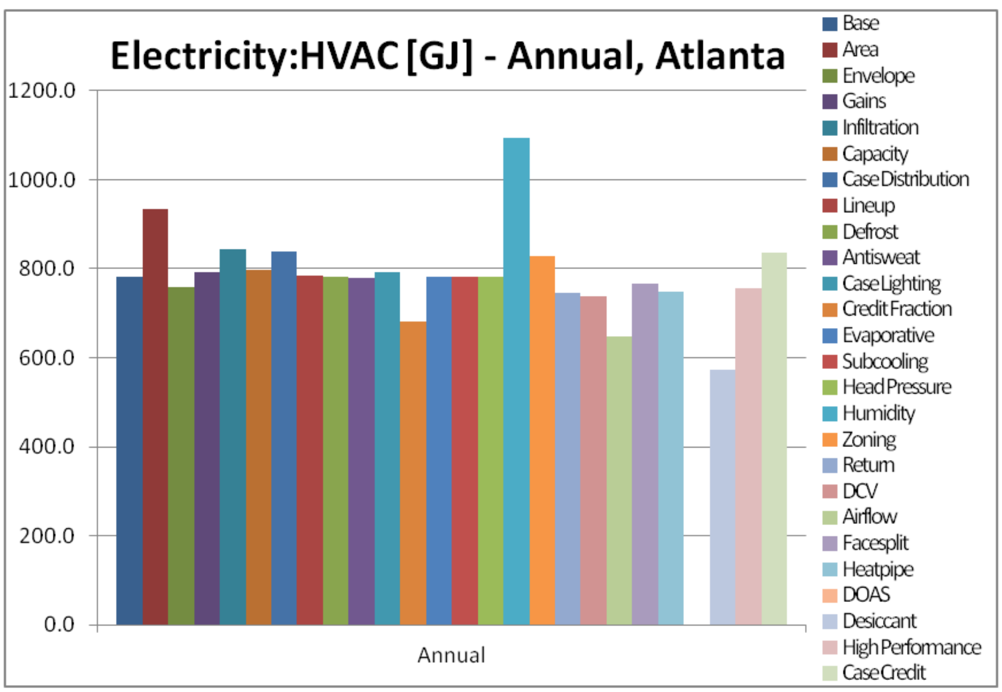


Figure 24. Electricity: HVAC [GJ] - Annual, Atlanta

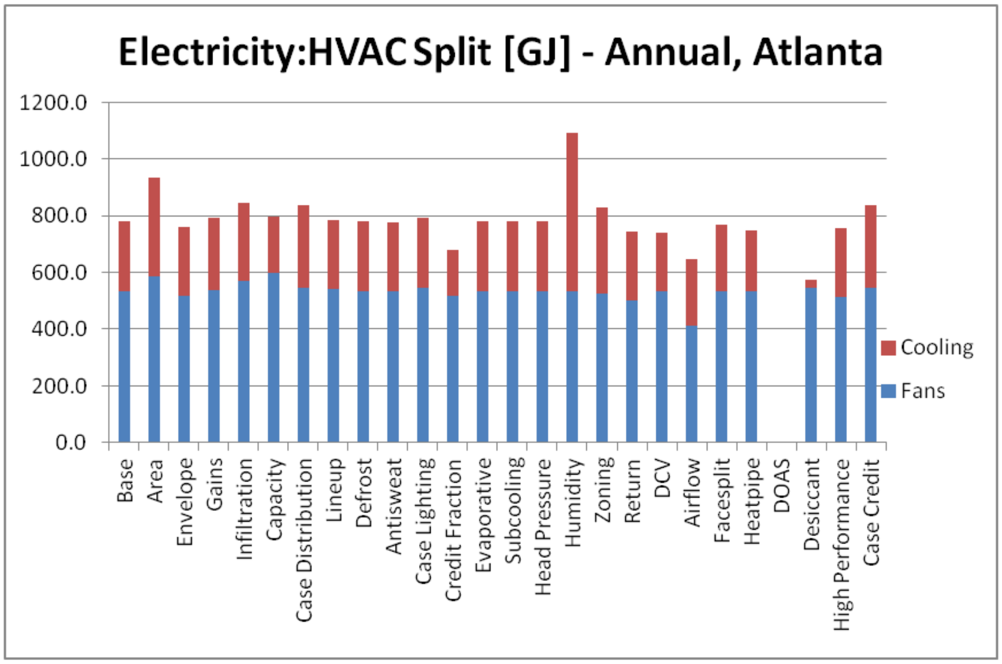


Figure 25. Electricity: HVAC Split [GJ] - Annual, Atlanta

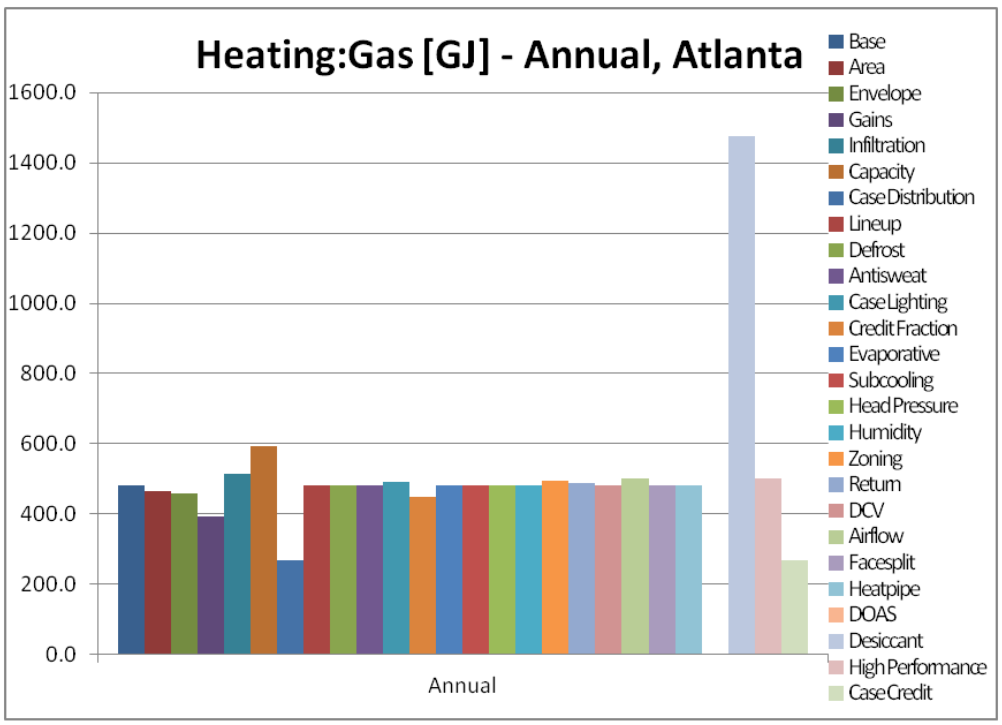


Figure 26. Heating: Gas [GJ] - Annual, Atlanta

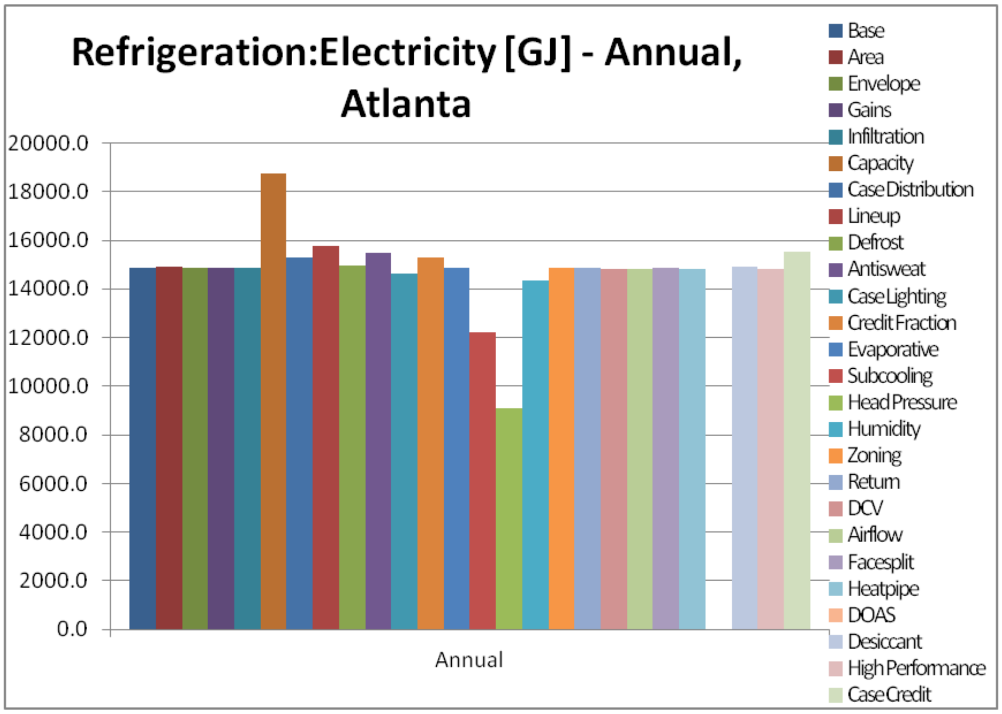


Figure 27. Refrigeration: Electricity [GJ] - Annual, Atlanta

One general observation of the parametric results is that HVAC energy use is dominated by fan electricity consumption. Fans operate continuously throughout the year and are sized based on heating, requirements, cooling requirements, and an imposed minimum airflow rate per floor area. In almost all cases, the fan sizing for the Sales and Dry Storage zones are driven by the minimum allowable airflow rate of 0.7 cfm/ft^2. This requirement was reduced to 0.5 cfm/ft^2 for the Airflow case discussed below. In other zones, the sizing of the HVAC fans is typically for heating requirements except in Miami.

6.3.1 Floor Area

<u>Increase the floor area in all zones by 30%</u>

The results of the increase in floor area were a predictable increase in general electricity usage, roughly in proportion with the area increase, as well as a larger increase in cooling electricity usage, presumably because the refrigeration capacity did not increase at the same time as the area so the HVAC system was taking a larger portion of the cooling load. For example the cooling coil in the Sales zone was approximately 10% larger to take the additional cooling and dehumidification load, while the fan was increased in size by 30% in order to maintain 0.7cfm/ft^2. The resulting increase in fan energy use was 10%, and cooling energy use was 41%. The gas energy use actually decreased 3% due to a decreased heating load in the Bakery and Deli zone, likely as a result of mixing from the Sales zone. There was minimal effect on the refrigeration electricity.

The increase in cooling electricity was mirrored in Minneapolis as a colder climate, however it was not seen in Miami, a hotter more humid climate. This is likely because the cooling loads even in the base case were so heavily dominated by the dehumidification loads, and the building was overcooled to the extent that the additional internal sensible loads simply allowed the space temperature to float up slightly within the deadband.

6.3.2 Envelope

<u>Increase insulation from 90.1 levels by 30%</u>

This case resulted in a 3% decrease in fan energy use, as when the fans were already meeting the flow per area requirements the increase in insulation allowed downsizing when they were sizing for heating requirements. Gas use was reduced by 5% in the three zones that were using gas for heating. Cooling energy was decreased as well, but this was not as significant, at 1%, because the cooling load was dominated by dehumidification rather than envelope loads.

6.3.3 Gains

<u>Increase sensible and latent gains by 30%</u>

The increased internal gains caused a 4.5% increase in cooling energy. An 18% decrease in gas use was shown due to the lowered heating requirements.

6.3.4 Infiltration

<u>Increase infiltration by 30% based on flow per exterior surface area</u>

Fan energy use increased 6% in this case due to fans increasing in size for larger heating requirements, while gas use increased 7%. Cooling energy increased 12%, a larger proportion because of the dehumidification load in the Sales zone.

The free heating use in the Sales and Produce zones increased by 4% to take up the extra reheat load.

6.3.5 Refrigeration Capacity

<u>Increase the length of cases 30%, increase the wattage of walk-ins 30%</u>

Fan energy increased 12%, while cooling energy decreased 19% as a result of the extra cooling from the refrigeration. This totaled an overall increase in HVAC electricity use of only 2%. Refrigeration energy increased 26%. All of this came to an increase of 22% in total electricity use. Gas use increased 23% to offset the cooling from the cases.

The free heating use in the Sales and Produce zones increased by 36% to offset the cases.

In Minneapolis gas use only increased 13%, likely because the base use was proportionally higher due to the colder climate. On the other hand in Miami gas use increased 111% because the initial usage level was so low being a warm climate.

6.3.6 Refrigerated Case Distribution

<u>Increase case load in Sales zone 30% by relocating cases from other zones</u>

Fan energy increased 2%, and cooling energy increased 19% as cooling was actually required finally in the Produce zone due to some of the large concentration of cases being relocated out of the zone.

Gas use decreased 55% due to reduced heating loads from cases in the Bakery and Deli. There was a 3% increase in refrigeration loads, this was due to the increased temperatures in the Produce zone and the cases having to work harder as a result. Temperatures in the Produce zone averaged up to 24C in the summer due to reduced case loads where previously they had been down around 21.2C, and the relative humidity in the zone dropped to 59% when it had been 69% in the base case due to the temperature difference in the zone, the absolute humidity was the same.

The free heating use in the Sales and Produce zones increased by 19% due to the larger capacity of cases in the Sales zone.

6.3.7 Display Case Lineup

<u>Shift 30% of Sales zone medium temperature capacity to low temp coffin cases while maintaining capacity</u>

Fan energy increased 1%, and cooling energy reduced 0.6%, with no change to the gas use. Refrigeration energy use increased 6% for cooling to lower temps since lower evaporator temperatures were required.

6.3.8 Defrost Strategy

<u>Shift from electric to hot gas on freezers, and add temperature termination</u>

The change in defrost strategy resulted in no change to the gas use, fan energy, and cooling energy, and minimal change in refrigeration energy use. Adding temperature termination rather than time termination to the off cycling of the medium temperature fixtures may have made more of a difference.

6.3.9 Antisweat Heater Control

<u>Leave antisweats on full power all the time</u>

The fan energy reduced by only 0.4%, and cooling energy by only 0.1%, with no difference at all in the gas use, however the free heating use in the Sales and Produce zones decreased by 1%, where the antisweats were actually located. The refrigeration energy use increased 4% due to additional load from the antisweat heaters.

In Miami refrigeration energy use increased only 0.6%, because the winter relative humidity in the Sales zone was higher, so the antisweats were turning down less in Miami in the base case.

6.3.10 Case Lighting

<u>Reduce case lighting by 30%</u>

All of the heat from lights in the case is assumed to go into the case. Refrigeration energy reduced 1.4 %. Fan energy increased 2%, and cooling increased by 0.6%, while gas use increased 2%.

6.3.11 Credit Fraction

<u>Increase LHR of cases by 50%</u>

The latent heat ratio affects the proportion of the case capacity which is latent, as a result increasing it increased the amount of dehumidification done by the cases instead of the HVAC system so the cooling energy reduces 33% for this model, while the refrigeration energy increased 3%. Gas use decreased 7%, and fan energy decreased 3%. The free heating use in the Sales and Produce zones decreased by 16%.

Credit fraction had less effect on the cooling energy in Miami, 20%, and the gas use decreased more, at 14%. In Minneapolis the cooling energy reduced 36%, and the gas use reduced 4%.

6.3.12 Evaporative Condenser

<u>Switch refrigerated case air cooled condensers for evaporative models</u>

No noticeable difference was seen in refrigeration use in Atlanta, usage increased 0.5% in Minneapolis, and decreased 0.1% in Atlanta.

6.3.13 Mechanical Subcooling

<u>Add mechanical subcooling to the refrigeration system</u>

Mechanical subcooling resulted in 18% savings in the refrigeration energy use, and no change to the other individual results, but a 15% savings in total electricity use.

6.3.14 Head Pressure

<u>Reduce condensing temperature to 40C from 55C to simulate floating head pressure control</u>

Head pressure controls resulted in 39% savings in the refrigeration energy use, and no change to the other individual results, but a 33% savings in total electricity use.

6.3.15 Humidity Setpoint

<u>Reduce humidity setpoint from 55% to 40%</u>

Reducing the humidity setpoint increased the cooling loads by 127% as a result of the extra dehumidification requirements, and decreased the refrigeration energy by 3.4%. Because of the increase in cooling, but the decrease in refrigeration, which was a smaller percentage but actually a much larger absolute number, the resulting saving in total electricity was 1.1%. The free heating use increased by 37% to undertake the extra reheat.

Cooling load differences were greater and refrigeration electricity load differences were less in Minneapolis, the opposite was true in Miami. The refrigeration load differences varied from 1 to 6% across the 9 climates.

When comparing these refrigeration savings to those seen in literature, a study from Henderson and Khattar [8] showed an energy use drop of approximately 0.4% for each 1% drop in RH, which works out to a 6% energy use saving, while a graph in the ASHRAE Applications Handbook seemed to show similar results of approximately 5% energy savings in refrigeration.

6.3.16 Zoning

Separate zone for freezers in Sales area

The second zone in the Sales area was served again by free heating from the reclaim system, and was assumed to have air mixing only with the main Sales zone.

The separate zone resulted in a 2% decrease in fan energy, and a 23% increase in cooling energy due to a reduced concentration of cases in the Sales zone, and a 3% increase in gas use due to mixing between zones. There was no change in refrigeration energy. The free heating use in the Sales and Produce zones decreased by 84%.

6.3.17 Return Air Location

Locate return air grilles under the cases

The change in return air location resulted in a 6% decrease in fan energy, however at the same time gas use increased 1%, since most of the decrease in heating requirements was in the Produce and Sales zones. A 1% decrease in cooling was also seen. The free heating use in the Sales and Produce zones decreased by 2%, larger than the gas use increase.

6.3.18 Demand Controlled Ventilation (DCV)

Include a DCV system in the HVAC system of the Sales zone

A DCV system resulted in a 17% decrease in cooling energy use by decreasing the outdoor air requirements based on CO_2 controls. The free heating use in the Sales and Produce zones decreased by 7%.

6.3.19 Airflow

Decrease airflow 30% in every zone

Decreasing the airflow in the zones resulted in a 23% decrease in fan energy, a 4% decrease in cooling, and a 4% increase in gas use.

6.3.20 Face Split Coils

Include DX Refrigerant Coil Control in the HVAC system of the Sales zone

Coil control results in a 6% decrease in cooling energy, with no change in fan, gas, reheat or refrigeration energy.

6.3.21 Heat Pipe

Include heat pipe dehumidification system in HVAC system of the Sales zone

The heat pipe system resulted in a 14% decrease in cooling energy, with no change in fan, gas or refrigeration energy. The free heating use in the Sales and Produce zones decreased by 2%.

6.3.22 Dedicated Outdoor Air System (DOAS)

Include DOAS system in HVAC system of the Sales zone

This system was not completed due to issues in the inclusion of this system with the correct controls to properly maintain humidity in the sales zone.

6.3.23 Desiccant

Include desiccant dehumidification system in HVAC system of the Sales zone

The desiccant system resulted in a 2% increase in fan energy, but an 88% decrease in cooling, adding up to a 27% decrease in total HVAC electricity use. A 207% increase in gas use was seen as a result of the gas coil used to regenerate the desiccant coil.

The free heating use in the Sales and Produce zones decreased by 23%, as the dehumidification was now being generally done by the desiccant wheel, not by cooling and reheating, and so the free heat was actually only being used for actual heating requirements.

Based on an assumption that the cost of gas is 1/3 of electricity, the desiccant case resulted in a 1% increase in utility cost overall, and a 13% increase in utility cost looking only at the utilities serving the HVAC system.

In Minneapolis the gas use increase was only 31% from the base, while in Miami it was 2550% since there was almost no base case gas use. However the actual gas use difference between locations for the regeneration coils was a 69% reduction from Atlanta to Minneapolis, and a 227% increase from Atlanta to Miami, due to the different climates in each location. In Minneapolis the desiccant system was not even required in the winter months, while in both Atlanta and Miami it operated year round.

6.3.24 High Performance Building

A combination of the Envelope case, and a 30% reduction in sales floor lighting

The reduction in internal loads resulted in a 4% increase in gas use, while the improved insulation decreased fan energy 4% and cooling energy 2%. Gas use increased 10% in Miami.

6.3.25 Case Credit

Relocate 30% of capacity to Sales zone, shift 30% of Sales zone capacity to low temp coffin cases, and reduce case lighting 30%

As this model was the sum of three other models, the results were effectively similar, with a 2% increase in fan energy, an 18% increase in cooling energy, a 5% increase in refrigeration energy, and a 45% reduction in gas use. The free heating use in the Sales and Produce zones increased by 28%.

An 18% fan energy use increase was seen in Miami.

6.4 Comments on Parametric Analysis

During the parametric analysis described here, several observations should be highlighted:

1. Fan energy made up a significant amount of the HVAC electricity use – 68% in the base case in Atlanta – and pressure drop assumptions made a big difference to the fan energy use.
2. Refrigeration electricity was overall far more significant than HVAC electricity, condensing temperature had a big effect on how much more significant however. When the condensing temperature was set to 55°C the refrigeration electricity was 19 times the total HVAC electricity, and 60 times the cooling electricity of the base case. If the condensing temperature was reduced to 40°C the refrigeration electricity was 12 times the total HVAC electricity, and 37 times the cooling electricity. After further discussions with industry professionals it was determined that a more appropriate condensing temperature for the base case would be 40°C, with the floating head

pressure case being set to 21°C. It was also determined that mechanical subcooling should be included in the base case.

3. A typical supermarket HVAC system uses desuperheater coils, which capture heat rejected from the refrigeration condensers to provide space heating and reheat during times of high dehumidification requirements. In our analysis, we sought to evaluate a low condensing temperature, floating head pressure type scenario, and the tradeoff between refrigeration savings and available reclaimed heat. In theory, the detailed refrigeration system model of EnergyPlus should be able to make specific calculations about available superheat, condensing temperatures, etc. However closer examination showed that we seemed to be able to always have sufficient heating available to heat the zones, no matter what condensing temperature we entered, which seemed to be an unlikely scenario.

In discussions with the developers of refrigeration module at Oak Ridge National Laboratory [17], the EnergyPlus model does not perform detailed heat exchanger calculations and does not limit the amount of heating based on a comparison of condensing temperature and supply air temperature. However, engineering calculations suggest that a minimum condensing temperature of 40°C will provide sufficient heating capacity in all locations. Since the reclaimed heat does not actually affect the latent heat balance in the zone, which was the focus of the project, the decision was made to proceed with the model as it stood, and Oak Ridge was to work on correcting the calculations. EnergyPlus V7 was issued immediately before this report was completed, and is supposed to contain a fix for this problem, however we did not have a chance to test the solution.

Chapter 7

7 Fractional Factorial Analysis

7.1 Fractional Factorial Introduction

The parametric analysis of the previous chapter explored the impact of changing twenty-five different supermarket parameters in nine locations. One of the limitations of the analysis is that each parameter is individually varied from a base value. As a result, the analysis does not provide any insight about the interactions among parameters on supermarket performance. Unfortunately, a full set of results to explore all the combinations of twenty-five parameters at two different values in nine locations would require $9*2^{25}$ = 301,989,888 EnergyPlus simulations.

The fractional factorial analyses of this chapter are performed to explore the more subtle impacts of key variables and to assess the interactions among this set of variables. The factors explored have been informed by the previous parametric analysis and selected as the most influential variables on the latent heat balance in supermarkets. In fact, two separate analyses are performed. The first analysis focuses on the key interactions among the most important refrigeration and HVAC system variables. The second analysis focuses on the interactions between key system and supermarket zone variables. However, first it is important to introduce the theory behind fractional factorial analysis and describe a revised supermarket prototype that will serve as the base case for the analyses.

7.2 Fractional Factorial Theory

Box, Hunter and Hunter [10] describe the methods for conducting a fractional factorial analysis. A full factorial design at two levels investigates all combinations of a set of factors (k) at both a base and test level, requiring 2^k runs. Results from this type of analysis provide information regarding the effects of each factor and two-factor interactions on the output of an experiment. In this case, the factors are parameters within the supermarket model, and the experiments are the energy simulations. The effects of each parameter on energy consumption are quantified by examining output data from each simulation.

A simple example of a full factorial design with three factors at two levels can be seen below. This example assesses three factors at both a base and test level, which are shown by a '-' for the base level and '+' for the test level. The yield (Y) is the calculated energy consumption output by the EnergyPlus model for each case, when run with that particular combination of values for each factor.

Table 15: Example of a 2^3 full factorial design

	Factor			
Case	1	2	3	Yield
1	-	-	-	Y_1
2	+	-	-	Y_2
3	-	+	-	Y_3
4	+	+	-	Y_4
5	-	-	+	Y_5
6	+	-	+	Y_6
7	-	+	+	Y_7
8	+	+	+	Y_8

The main effect for each factor can then be calculated by subtracting the average yields for all cases where the factor is at the base level ($\overline{Y_-}$), from the average yields for all cases where the factor is at the test level ($\overline{Y_+}$) where M_n is the main effect of each factor n, as shown below:

$$M_n = \overline{Y_+} - \overline{Y_-}$$

The relative impact (R_n) of each factor is similar to the main effect, but is normalized to the average of all yields ($\bar{Y}$). By normalizing the effects, we can better compare the effects of multiple factors. The relative impact might be calculated to quantify the effects of each factor on energy consumption, and to rank the factors by level of significance. The relative impact can be calculated using the equation below:

$$R_n = \frac{\overline{Y_+} - \overline{Y_-}}{\bar{Y}}$$

To illustrate the concept and calculations, consider the analysis of supermarket energy use as influenced by three factors: refrigeration capacity, relative humidity setpoint, and supply airflow rate. The analysis will involve two different values of each of the three factors – a base value and a test value. However, instead of just varying one factor at a time, we will explore all combinations of the three factors at each of two values. The "full factorial analysis" is represented by the example of Table 15, where each case represents an annual simulation, the three factors represent the three influencing variables, the "-" and "+" values represent the base and test values of each factor, respectively, and the Y results are the supermarket energy consumption for each case. Notice that the eight cases comprise four simulations at the base value and four simulations at the test value for each factor.

To evaluate the main effect of the refrigeration capacity, Factor 1, on supermarket energy consumption, we calculate the average energy consumption for the four runs at the test value (say, the larger value of the refrigeration capacity) compared to the average energy consumption of the four runs at the base value (say, the smaller value of the refrigeration capacity). That is, M_1 is calculated as

$$M_1 = \overline{Y_+} - \overline{Y_-} = \frac{Y_2 + Y_4 + Y_6 + Y_8}{4} - \frac{Y_1 + Y_3 + Y_5 + Y_7}{4}$$

The relative impact of Factor 1, R_1, is then calculated by dividing the main effect by the average energy consumption of all eight simulations. Algebraically,

$$R_1 = \frac{\overline{Y_+} - \overline{Y_-}}{\bar{Y}} = \frac{2(Y_2 + Y_4 + Y_6 + Y_8 - Y_1 - Y_3 - Y_5 - Y_7)}{Y_1 + Y_2 + Y_3 + Y_4 + Y_5 + Y_6 + Y_7 + Y_8}$$

Similar calculations can be performed to calculate the main effect for each factor and the relative impact of each factor. The most influential variables can be identified by large values of the relative impact.

The effects of two-factor interactions can also be calculated, which show how the interdependency of any two factors affects the results. These interdependencies are often lost when only one-off parametric studies are examined. A well-known example of such interdependencies in building energy analysis is the combined effect of window area and thermal mass on passive solar heating performance. Some improvements in passive solar heating can be obtained by simply increasing window area alone. However, at benefits are limited if the thermal mass is insufficient to store the additional energy admitted through the larger windows. Similarly, some benefit can be obtained by simple increasing the level of thermal mass alone, though the benefit is limited without also increasing the solar energy flow. The combined effect of increasing both the window area and thermal mass together provides a much greater benefit than simply adding the benefits of the individual factors. Note that if the two factors were completely independent of each other, then the combined effect would be exactly equal to the sum of the individual effects.

Mathematically, the equation used to calculate the relative impact of the interactions of any two factors n and m (R_{nm}) is described below:

$$R_{nm} = \frac{(\overline{Y_{nm}} + \overline{Y_{NM}}) - (\overline{Y_{Nm}} + \overline{Y_{nM}})}{\bar{Y}}$$

where $\overline{Y_{nm}}$ is the average yield of the experiment where the factors n and m are at the base level, $\overline{Y_{Nm}}$ and $\overline{Y_{nM}}$ is the average yield of the experiment where one of the factors n and m are at the base and the other is at the test level, and $\overline{Y_{NM}}$ is the average yield of the experiment where the factors N and M are at the test level. Using the example of Table 15, the relative impact of the interactions of Factor n=1 and Factor m=3 can be calculated as

$$R_{13} = \frac{\left(\frac{Y_1 + Y_3}{2} + \frac{Y_6 + Y_8}{2}\right) - \left(\frac{Y_2 + Y_4}{2} + \frac{Y_5 + Y_7}{2}\right)}{\frac{Y_1 + Y_2 + Y_3 + Y_4 + Y_5 + Y_6 + Y_7 + Y_8}{8}}$$

The examples above involve "full" factorial analyses, where the full set of all combinations of variable values are used. However, as the number of factors, *k*, grows, the total number of cases (i.e., EnergyPlus simulations) increase to 2^k, quickly producing an unmanageable set of cases. To reduce the time associated with a full factorial analysis, a fractional factorial analysis was implemented. The fractional factorial analysis systematically reduces the number of simulations needed to calculate the effects of each factor. However, to reduce the number of simulations, the impact of some interactions are confounded with those of other factors or factor combinations, reducing the precision of the analysis. Still, we can assume that higher order interactions between three or more factors are insignificant to the experiment, and in a properly designed fractional factorial experiment, the main effects are confounded only with higher level interactions. Therefore, the ability to identify the effects of individual factors and major interactions is preserved.

The level of confounding in a fractional factorial analysis is characterized by the resolution of the fractional factorial design. Lower resolution designs require fewer simulations or runs but have a higher level of confounding. On the other hand, higher resolution designs require more simulations and less confounding. The most commonly used designs are described below:

Resolution III Designs: Main effects are confounded with two-factor interactions. Such designs would be unacceptable if we seek to clearly identify both main effects and two factor interactions.

Resolution IV Designs: No main effects are confounded with two-factor interactions, but two-factor interactions are aliased with each other

Resolution V Designs: No main effect or two-factor interaction is confounded with any other main effect or two-factor interaction, but two-factor interactions are confounded with three-factor interactions.

7.3 Revised Factorial Analysis Base Case

The parametric analysis of the previous chapter was based on a prototypical supermarket design developed, with significant input from industry professionals, to be representative of new supermarkets, their systems, and their operations. The parametric analysis of twenty-five factors, individually varied from this base case supermarket, allowed assessment of the impacts of the factors on supermarket energy end use.

Upon completion of the parametric analysis, it was observed that several of the base (default) system parameters deserved reconsideration and adjustment before the more critical assessment of the variables affecting the balance between refrigeration and comfort cooling systems. Specifically, the following factors were revised:

1. **Mechanical Subcooling**. The original base case refrigeration system did not utilize mechanical subcooling to improve refrigeration system performance. The parametric analysis showed significant energy savings. Given that almost all modern supermarket refrigeration system use mechanical subcooling, the base was revised to include this feature.
2. **Floating Head Pressure**. The original base case refrigeration was operated with conservative limitations on the head pressure leaving the compressors. Recognizing the opportunity to provide space heating with recovered heat from the refrigeration condensers, it has been common to "prop up" the head pressure to ensure adequate HVAC heating capabilities. After review of the parametric results and further consideration with industry practitioners, the base case was revised to lower the minimum saturated condensing temperature from 55°C to 40°C.
3. **Latent Case Credits**. Refrigerated cases remove both sensible and latent energy from the building zones in which they are located due to convective and radiative heat and mass exchange

between the cases and the environment. In the original base case, the fraction of these case credits that that comprised latent heat removal from the zone was based on research in the engineering literature. However, there is considerable variation in the values reported in the literature. Rather than rely on these general, but uncertain, values, the base case was revised to directly use the manufacturer data for condensate removal at design environmental operating conditions.

4. **Humidity Control in Produce Zone**. In the original analysis, the space humidity was directly controlled only in the main sales zone of the supermarket. Specifically, indoor humidity was not controlled in the office, bakery, deli, dry storage, or produce zones. The produce zone was specifically omitted because many produce products have improved shelf life at higher humidity levels. However, after review of the parametric analysis results, it was observed that indoor humidity levels in the produce zone were rising to unacceptable levels during much of the summer months in many locations. After consultation with the Project Monitoring Subcommittee, the base case was revised to include humidity control in the produce zone.

These four changes to system configuration, system operation, and modeling assumptions had significant impacts on the base case supermarket performance. In particular, adding mechanical subcooling and reducing condensing temperature reduced refrigeration system energy consumption by half. Figure 28 shows the annual end use energy consumption of the revised base case supermarket prototype in the nine US locations. By comparison with the results of the original base case in Figure 19, all locations show a 35%-40% reduction in building energy consumption. Refrigeration system energy consumption, including the compressors, condenser fans, and refrigerated case energy for lights, fans, and anti-sweat heaters, remains near 70% of total site energy consumption. HVAC system energy use, including air distribution fans, cooling, and heating, accounts for 6%-15% of total energy consumption.

The changes in system parameters also affect the indoor conditions, though less dramatically. Figure 29 shows the average space temperature in the main sales zone throughout the year for the nine locations. Figure 30 shows similar results for the space relative humidity. In general, space temperatures are higher and space relative humidities are lower than the original base case. These results are due to the change in the latent case credits, which are higher in the revised base case, producing more dehumidification and less sensible cooling by the refrigerated cases in the sales zone.

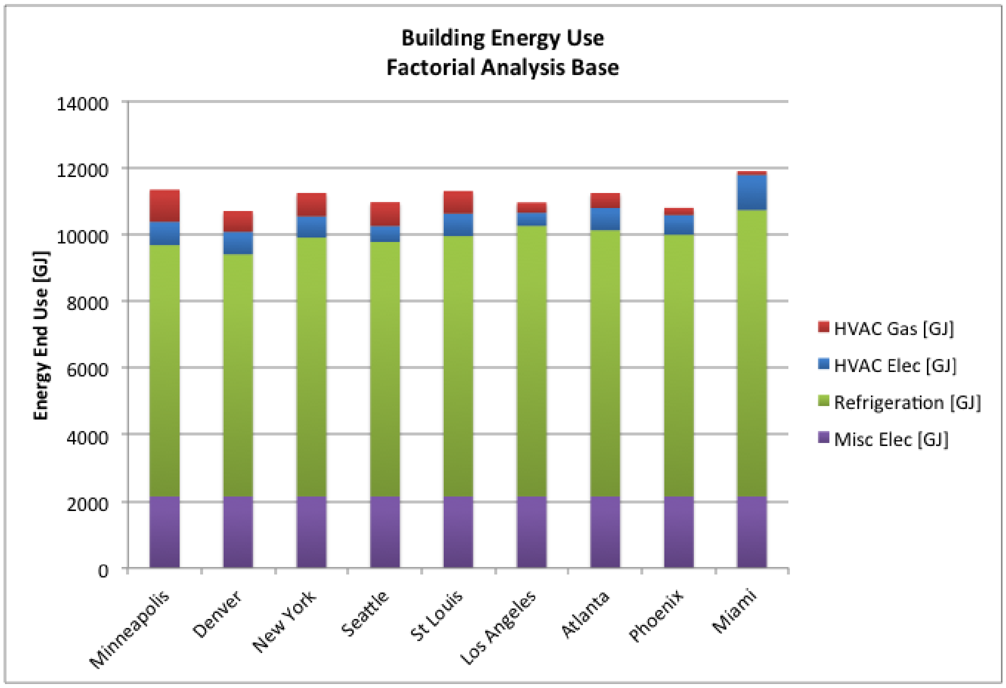


Figure 28 Revised base case energy performance in US locations

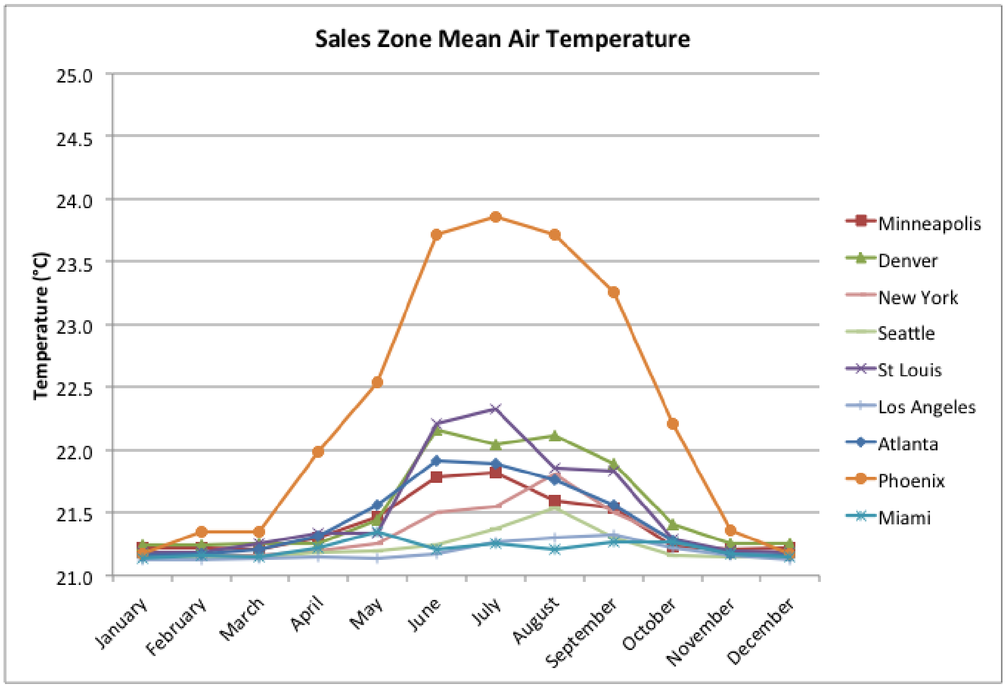


Figure 29 Mean air temperature in sales zone, revised base case, US locations

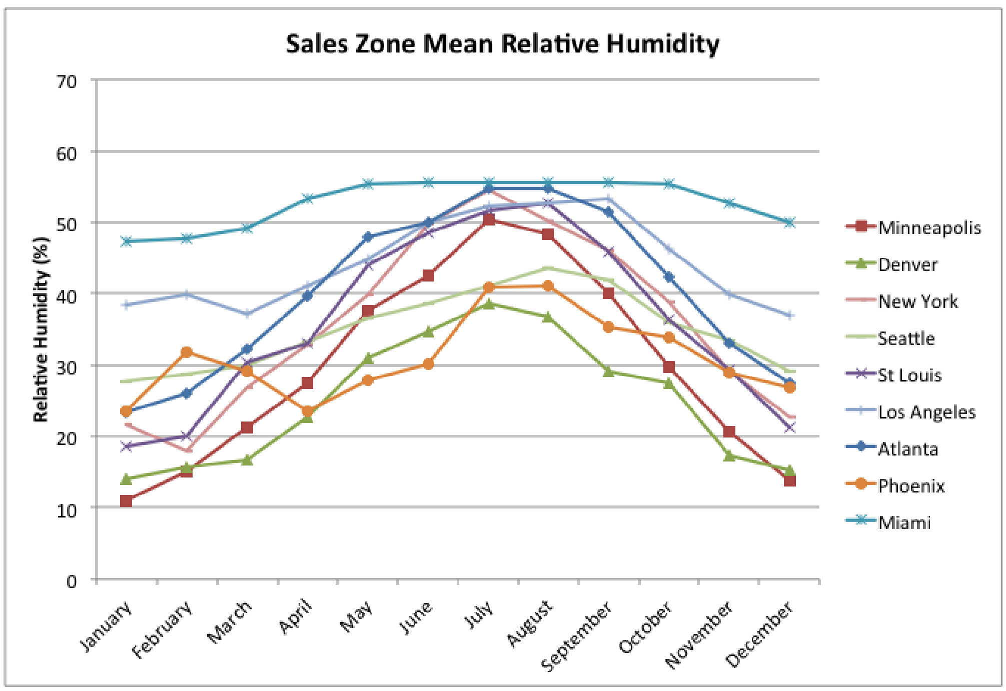


Figure 30 Mean relative humidity in sales zone, revised base case, US locations

7.4 HVAC and Refrigeration Factorial Analysis

7.4.1 Design

The first factorial analysis explores the effects of several key HVAC and refrigeration system variables. The variables were chosen for this analysis based on the results of the parametric analysis, the opportunities of design engineers to affect HVAC and refrigeration system characteristics, the uncertainty in refrigeration system characteristics, and modest variations in location. The parametric analysis shows that HVAC system factors have very little impact on total building energy consumption, largely because HVAC system energy use is such a small fraction of total consumption. Nevertheless, with supermarkets consuming large amounts of energy, even small percentage changes can translate to measurable changes in energy costs.

Seven factors were chosen for this analysis, identified in Table 16. The base values of the factors represent the revised base case supermarket prototype. HVAC system design options that are commonly considered in supermarkets with an eye to lowering refrigeration system energy consumption are reduced airflow, demand controlled ventilation, and a heat pipe to improve HVAC system latent performance. For this analysis, the test value of the airflow is 30% less than they base design value, consistent with the reduction from 0.7 cfm/ft^2 to 0.5 cfm/ft^2 of the parametric analysis. The use of demand controlled ventilation is expected to reduce ventilation requirements at times of low occupancy, reducing the need for heating and cooling. The heat pipe is expected to increase the latent cooling capability of the HVAC system. While the HVAC energy consumption is expected to increase due to the reduction in sensible cooling capacity, the improved latent performance is expected to lower indoor humidity levels, which will reduce refrigeration system energy consumption.

Table 16: Factors for fractional factorial analysis

Factor	Base	Test
Airflow	Design value	Reduce by 30%
Ventilation (DCV)	Constant design value	Demand controlled
Heat Pipe	None	Wrap-around heat pipe
Latent Case Credit (LHR)	Manufacturers data	Increase latent case credit by 50%
Refrigeration Capacity (CC1)	Nominal	Increase capacity by 30%
Case Line Up (CC2)	Nominal	30% of medium temp converted to low temp coffin cases
Location	Atlanta	Los Angeles

The two most significant refrigeration system factors on total building, refrigeration, and HVAC energy are the capacity of the refrigeration system and the lineup of cases between medium and low temperature cases. Increasing the refrigeration capacity by 30% will clearly increase store energy consumption, though the additional cooling effect of the cases will reduce store cooling requirements. The effect of case lineup will be explored by converting 30% of the medium temperature doored cases to low temperature coffin cases, while maintaining the total refrigeration system capacity at is design value. The lineup variations are expected to affect refrigeration energy use due to the changes in defrost and antisweat heater operation, even though the capacity of the system is unchanged. The lineup variations will also affect HVAC energy consumption due to the different sensible and latent case credit characteristics of low and medium temperature cases.

The latent case credit fraction can have a significant impact on the space conditioning performance and is highly uncertain and variable among case types. The test value of this variable is 50% greater than the base value. For example, approximately 6% of the load on a doored refrigerated case may be associated with latent heat removal under environmental design conditions of 23.9°C and 55% RH. In the test case, this latent case credit (or latent heat removal factor) would be increased to 9% of the design refrigeration load, effectively reducing the sensible heat removal by a corresponding amount.

Location was identified to specifically explore interactions between climate and other variables.

A full factorial analysis for this set of factors would require $2^7=128$ simulations. We elected to reduce the number of simulations to 64 through a fractional factorial design of resolution V. The design of the set of simulations is given in Table 17.

Table 17: 2_{VII}^{7-1} fractional factorial design

Run No.	Air flow	DC V	Heat pipe	LH R	CC1	CC2	Loca tion	Run No.	Air flow	DC V	Heat pipe	LH R	CC1	CC2	Loca tion
1	Base	Base	Base	Base	Base	Base	Test	**33**	Base	Base	Base	Base	Base	Test	Base
2	Test	Base	Base	Base	Base	Base	Base	**34**	Test	Base	Base	Base	Base	Test	Test
3	Base	Test	Base	Base	Base	Base	Base	**35**	Base	Test	Base	Base	Base	Test	Test
4	Test	Test	Base	Base	Base	Base	Test	**36**	Test	Test	Base	Base	Base	Test	Base
5	Base	Base	Test	Base	Base	Base	Base	**37**	Base	Base	Test	Base	Base	Test	Test
6	Test	Base	Test	Base	Base	Base	Test	**38**	Test	Base	Test	Base	Base	Test	Base
7	Base	Test	Test	Base	Base	Base	Test	**39**	Base	Test	Test	Base	Base	Test	Base
8	Test	Test	Test	Base	Base	Base	Base	**40**	Test	Test	Test	Base	Base	Test	Test
9	Base	Base	Base	Test	Base	Base	Base	**41**	Base	Base	Base	Test	Base	Test	Test
10	Test	Base	Base	Test	Base	Base	Test	**42**	Test	Base	Base	Test	Base	Test	Base
11	Base	Test	Base	Test	Base	Base	Test	**43**	Base	Test	Base	Test	Base	Test	Base
12	Test	Test	Base	Test	Base	Base	Base	**44**	Test	Test	Base	Test	Base	Test	Test
13	Base	Base	Test	Test	Base	Base	Test	**45**	Base	Base	Test	Test	Base	Test	Base
14	Test	Base	Test	Test	Base	Base	Base	**46**	Test	Base	Test	Test	Base	Test	Test
15	Base	Test	Test	Test	Base	Base	Base	**47**	Base	Test	Test	Test	Base	Test	Test
16	Test	Test	Test	Test	Base	Base	Test	**48**	Test	Test	Test	Test	Base	Test	Base
17	Base	Base	Base	Base	Test	Base	Base	**49**	Base	Base	Base	Base	Test	Test	Test
18	Test	Base	Base	Base	Test	Base	Test	**50**	Test	Base	Base	Base	Test	Test	Base
19	Base	Test	Base	Base	Test	Base	Test	**51**	Base	Test	Base	Base	Test	Test	Base
20	Test	Test	Base	Base	Test	Base	Base	**52**	Test	Test	Base	Base	Test	Test	Test
21	Base	Base	Test	Base	Test	Base	Test	**53**	Base	Base	Test	Base	Test	Test	Base
22	Test	Base	Test	Base	Test	Base	Base	**54**	Test	Base	Test	Base	Test	Test	Test

Run No.	Air flow	DC V	Heat pipe	LH R	CC1	CC2	Loca tion		Run No.	Air flow	DC V	Heat pipe	LH R	CC1	CC2	Loca tion
23	Base	Test	Test	Base	Test	Base	Base		**55**	Base	Test	Test	Base	Test	Test	Test
24	Test	Test	Test	Base	Test	Base	Test		**56**	Test	Test	Test	Base	Test	Test	Base
25	Base	Base	Base	Test	Test	Base	Test		**57**	Base	Base	Base	Test	Test	Test	Base
26	Test	Base	Base	Test	Test	Base	Base		**58**	Test	Base	Base	Test	Test	Test	Test
27	Base	Test	Base	Test	Test	Base	Base		**59**	Base	Test	Base	Test	Test	Test	Test
28	Test	Test	Base	Test	Test	Base	Test		**60**	Test	Test	Base	Test	Test	Test	Base
29	Base	Base	Test	Test	Test	Base	Base		**61**	Base	Base	Test	Test	Test	Test	Test
30	Test	Base	Test	Test	Test	Base	Test		**62**	Test	Base	Test	Test	Test	Test	Base
31	Base	Test	Test	Test	Test	Base	Test		**63**	Base	Test	Test	Test	Test	Test	Base
32	Test	Test	Test	Test	Test	Base	Base		**64**	Test	Test	Test	Test	Test	Test	Test

The results of the factorial analysis should be interpreted by comparing the absolute magnitudes of the R_n values, and then the R_{nm} values, where a large magnitude indicates a significant variable or interaction. The values should also be compared in their correct sign to determine whether the effect is an increase or decrease in energy use. The scale of the graphs is a ratio of the energy values, and is dimensionless.

7.4.2 Refrigeration Energy Results

Figure 31 shows the effect of the seven factors and their interactions on refrigeration energy consumption. The most significant factor is clearly the capacity of the refrigeration system (CC1). While the CC1 analysis involves a 30% increase in capacity, this did not translate directly to a 30% increase in refrigeration energy use since the cases do not always run at rated conditions, and because the refrigeration electricity includes items such as lights, defrost heaters, and antisweat heater, which are not in continuous operation.

Converting 30% of the medium temperature cases to low temperature cases (CC2) increases refrigeration energy use approximately 13%, largely because low temperature coffin cases interact differently with the zone and have greater defrost and antisweat heater needs. Increasing the latent case credit by 50% (LHR) reduces the refrigeration energy consumption by about 5% due to changes in the indoor temperature and humidity. The location had very little impact on the refrigeration energy (<1%) since the annual average outdoor temperature between Atlanta and Los Angeles is almost identical, despite the significant seasonal differences.

The changes in the HVAC system produced almost imperceptible changes in the refrigeration energy consumption. Reducing airflow and adding a heat pump are both expected to reduce indoor humidity levels with corresponding reductions in refrigeration energy. However, they also tend to increase indoor temperature levels, which increase refrigeration energy.

There were relatively few large interaction effects in the refrigeration analysis. The greatest interactions occur between CC1 and CC2. The interaction indicates that the combined effect of increasing refrigeration capacity and reallocating cases with more low temperature cases increases refrigeration energy use by 2% more than the sum of the individual effects. It also suggests that the exact impact of CC1 depends on the value of CC2, and vice versa.

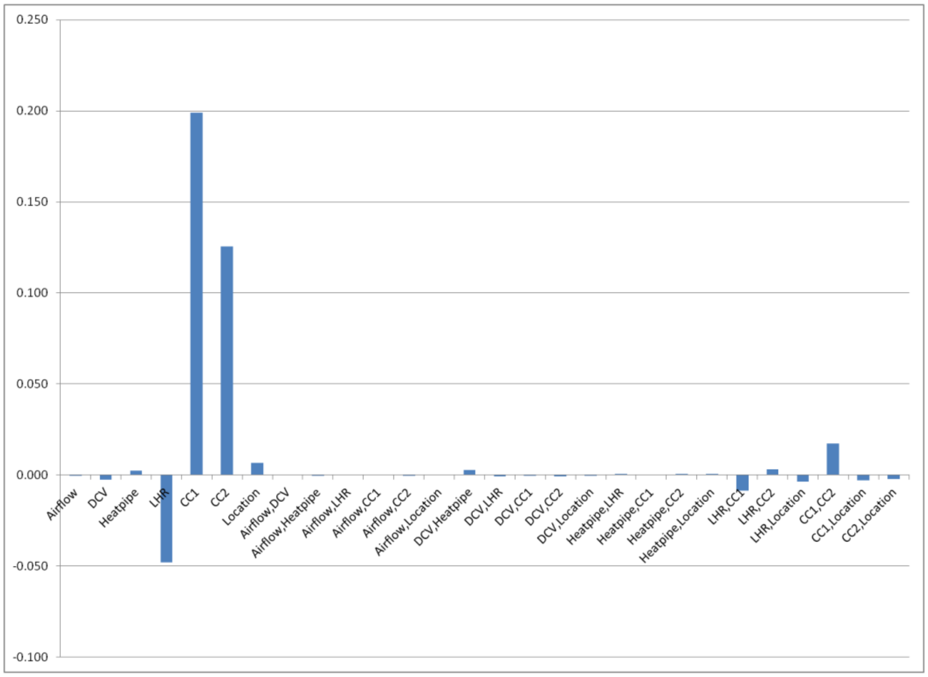


Figure 31. FFA1 Refrigeration - Main Effects and Interactions

7.4.3 HVAC Cooling Energy Results

Figure 32 show the results for HVAC cooling energy for the seven factors and their interactions. The largest single factor on cooling electricity was location, which produced a relative impact of -140%. The very large impact, < 100%, reflects that the change in cooling energy use between Atlanta and Los Angeles was greater than the average of the two cities. On average, Atlanta used 129 GJ of cooling while Los Angeles only required 24 GJ.

Refrigeration loads being added or subtracted also increased and decreased the amount of cooling being done by the refrigeration system, and hence what needed to be picked up by the cooling system. Hence, increasing refrigeration capacity and increasing the latent case credit reduce cooling energy. Converting medium temperature cases to low temperature cases (CC2) increases cooling energy needs. Since the refrigeration capacity of a low temperature case needs to meet additional defrost and antisweat heater needs, there is less energy available to sensibly cool the zone.

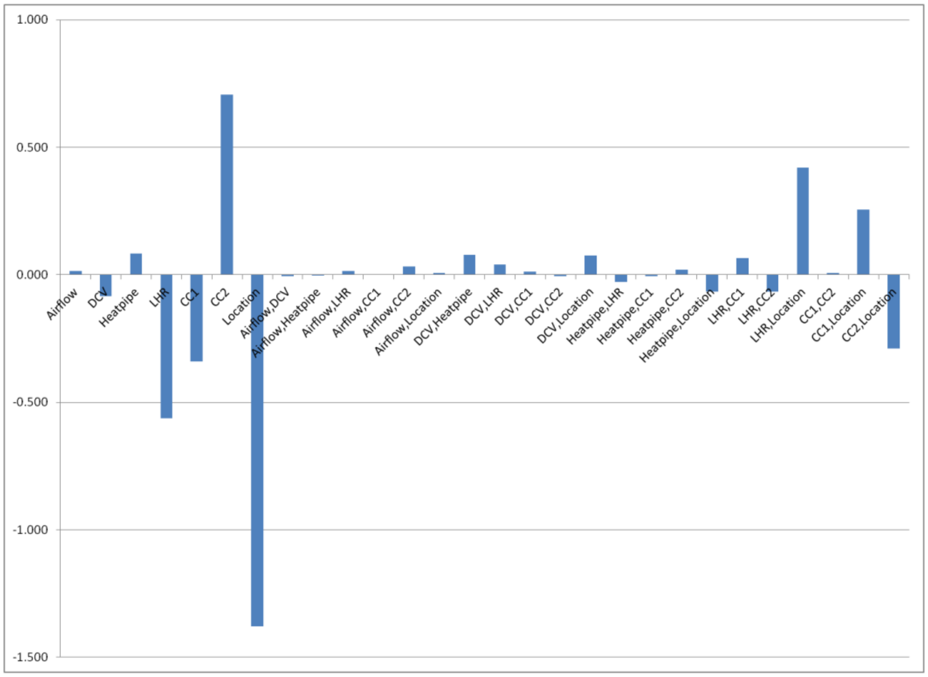


Figure 32. FFA1 HVAC Cooling - Main Effects and Interactions

Compared to the cooling energy impacts produced by refrigeration system changes, HVAC system changes have relatively little effect. Reducing airflow rates to improve HVAC latent cooling performance has almost no effect. Adding demand controlled ventilation reduces cooling by almost 10%, but adding a heat pipe actually increases cooling energy by reducing sensible capacity.

While many of the interactions are large, the most significant interactions include the effect of location. However, the magnitude of these interaction effects are distorted by the large impact of location and the nonlinear effects it imposes. For example, the relative impact of the LHR-Location interaction is 40%, showing that the combined effect of increasing latent case credit and changing to Los Angeles uses 40% greater cooling energy than the sum of the individual effects. Note that the individual effects of latent case credit is -57% and that of location is -140%, suggesting that increasing LHR and moving to Los Angeles combined would reduce cooling energy by about 157%. Similarly, the large interaction also suggests that the impact of LHR depends on location, and vice versa.

7.4.4 HVAC Fans

Figure 33 shows the relative impacts of the seven variables and their interactions on HVAC system fan energy. The largest main effects were associated with the airflow parameter and location. The impact of the airflow factor is an expected direct reduction of fan energy. The location change to Los Angeles reduces fan energy due to the lower heating and cooling loads. Similarly changing the refrigeration loads by increasing capacity and changing the case lineup also changed the overall store loads and therefore the fan sizes.

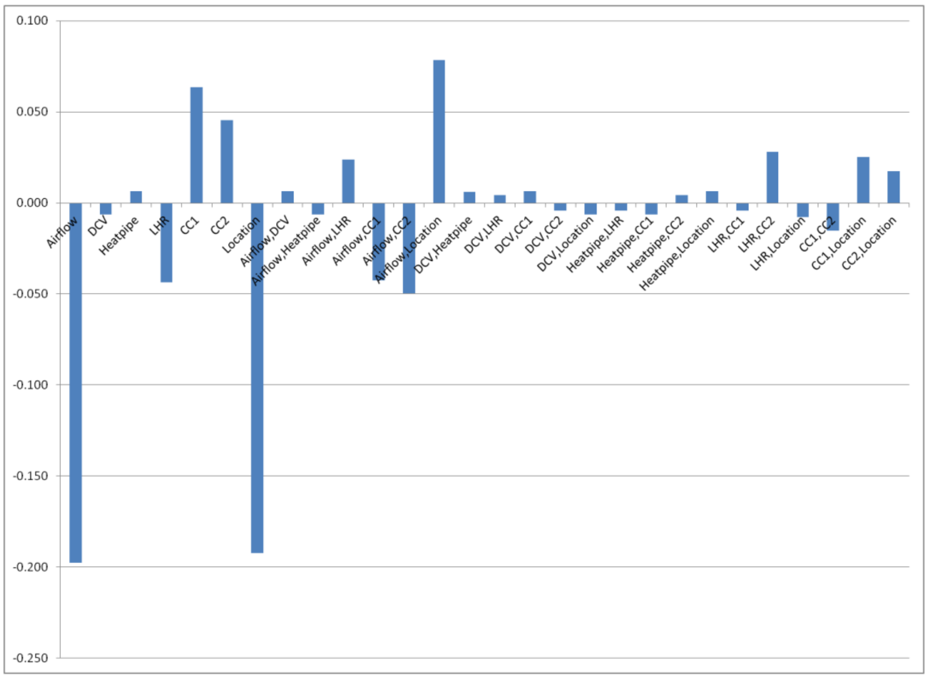


Figure 33. FFA1 HVAC Fans - Main Effects and Interactions

The largest interactions involved airflow and location, which had the greatest main effects. In general the interactions involving airflow tend to have positive values, which compared to the large negative values of the relative impact of the main airflow effect, show that the combined impact of airflow and another factor tend to produce less energy savings than simply adding the impacts of the two individual factors. The same applies to interactions with location. The notable exceptions are the interactions between airflow and the refrigeration capacity and lineup, which identify even greater fan savings when the airflow reduction is combined with the refrigeration system factors.

7.4.5 HVAC Electricity

HVAC electricity is the sum of the cooling and fan energy, which were discussed in the previous two sections. Figure 34 shows the effects of the several factors and their interactions on the combined HVAC electricity use. The results are basically a summation of cooling and fan impacts. One interesting result is that increasing the refrigeration capacity by 30% (CC1) has almost no effect, since the increased fan energy required by the heating capacity was offset by the decreased cooling energy from the larger refrigeration capacity.

The largest interactions were around airflow and location, with the biggest being airflow with location. Despite the CC1 not being a large main effect, the interaction with location was one of the larger effects.

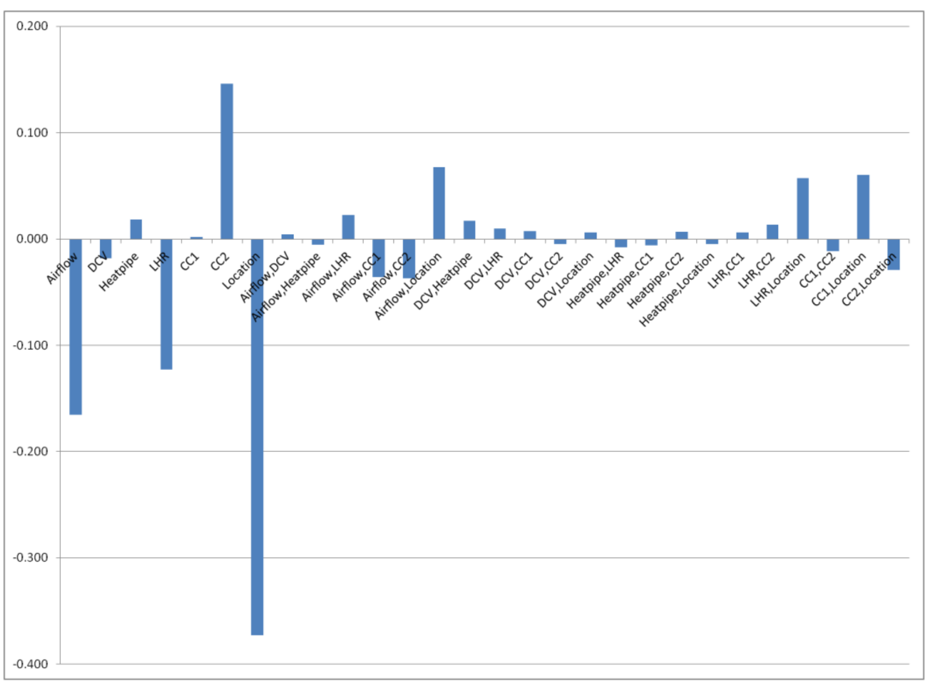


Figure 34. FFA1 HVAC Electricity - Main Effects and Interactions

7.4.6 HVAC Gas

Figure 35 shows the impact on gas energy consumption for HVAC heating and dehumidification reheat. The most significant effect is the change in case lineup to use more low temperature coffin cases, which reduces heating energy by 58%. Los Angeles has a milder climate with both lower heating loads and dehumidification requirements, thereby reducing heating. Increasing the latent case credits reduces HVAC dehumidification needs and the associated reheat energy. Adding refrigeration capacity directly increases heating requirements.

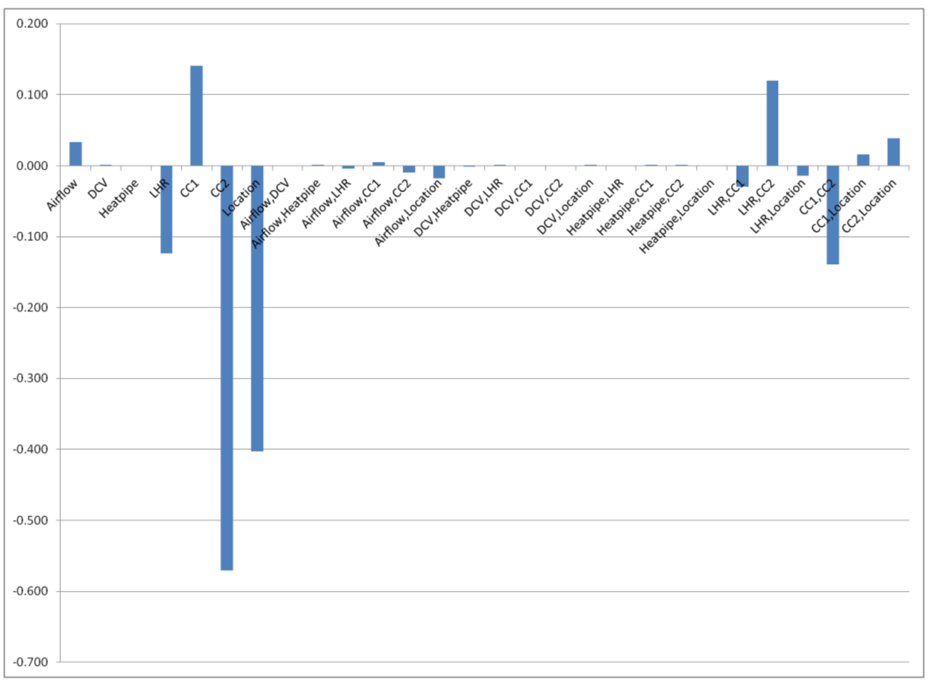


Figure 35. FFA1 HVAC Gas - Main Effects and Interactions

7.4.7 Building Source Energy Results

Building Source Energy is the sum of the total building electricity and the HVAC gas use, weighted by source energy factors of 1.047 for gas and 3.34 for electricity [16]. The results are almost identical to results of building electricity use, which is omitted for economy. Figure 36 shows the results.

The main effects are dominated by the refrigeration system, which account for about 70% of energy consumption. While it is expected that increasing refrigeration capacity by 30% (CC1) would have a direct impact, it is somewhat more surprising that simply converting 30% of the sales area medium temperature cases to low temperature coffin cases, without increasing the total refrigeration system capacity, causes a 10% increase in supermarket source energy use. Similarly, an uncertainty or variations in the latent case credit of the refrigerated cases can affect energy consumption by as much as 5%.

By comparison, HVAC system design options have very little effect on supermarket energy consumption, producing changes of less than 1%. Among the HVAC option, airflow reduction offers the greatest opportunities with total energy savings of less than 1%, though the relative impact of interactions between airflow and the other factors suggest that the savings could be between 0.3% - 1.0%.

The change in location from Atlanta to Los Angeles has less than 2% effect.

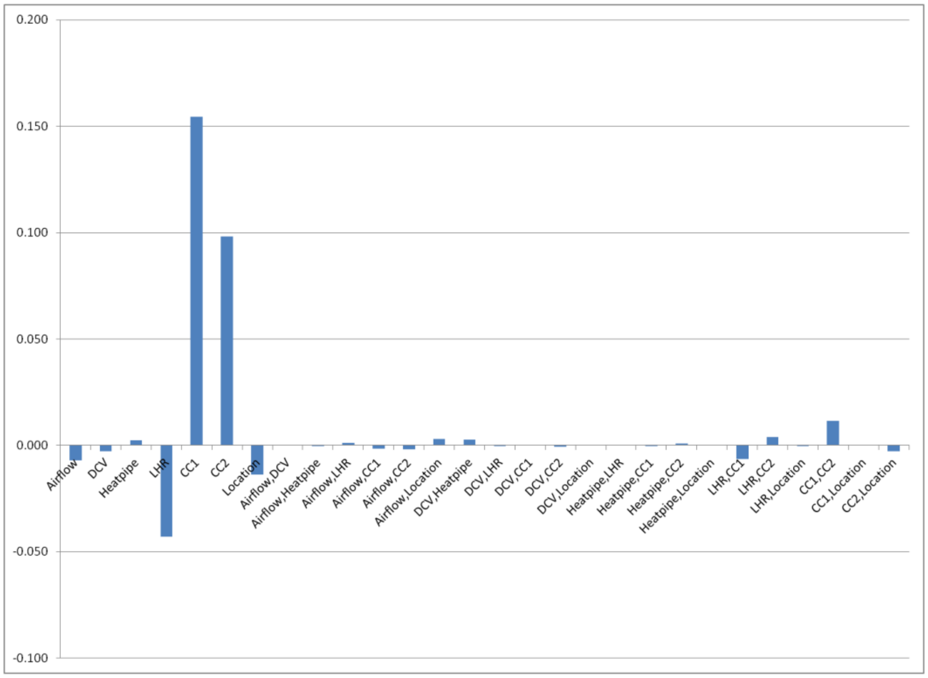


Figure 36. FFA1 Building Energy - Main Effects and Interactions

7.4.8 Summary

The dominant variables seen in this study were the latent case credit, refrigeration capacity, and case line up. This tended to indicate that the refrigeration variables were far more dominant over the HVAC effects, and that the initial assumptions of the project around the ability of the HVAC systems to have significant impacts on total building energy use may have been overestimated.

Despite this, because the quantities of energy being used in supermarkets are so large, even relatively small savings can be a cost saving to the store operator, so may still be worth pursuing.

7.5 Zoning Factorial Analysis

The previous factorial analysis focused on interactions between refrigeration and HVAC systems and showed that HVAC system design options have very little effect on building energy use. The previous analysis did not include two of the zone-level factors that are expected to have an impact on the latent heat balance in supermarkets: relative humidity setpoint and humidity zoning layout. A second, smaller factorial analysis is performed to highlight the effect of these zone level variables.

The zone factorial analysis combines indoor humidity setpoint and zone layout with the dominant variables from the first study, i.e., the latent case credit, refrigeration capacity, and case line up. The zone layout and humidity setpoint are considered in a single case. The base value of the factor is a single sales zone with a humidity setpoint of 55% RH. The test value of the factor splits the sales zone into two separate zones with separate air handlers – one for the main sales area and a second for the freezer aisles of the store. While the first zone continues to use a 55% RH setpoint, this second zone has a humidity setpoint of 40% RH. Referring to the layout of the supermarket in Figure 17, the new zone encloses the portion of the sales area comprising the doored freezer cases.

7.5.1 Design

Five factors were considered in the analysis: refrigeration capacity (CC1), case lineup (CC2), latent case credit (LHR), humidity zoning, and location. A fractional factorial analysis of resolution V using sixteen runs was conducted. The study was conducted three times with different location pairs.

Table 18 gives the factors used in the analysis and shows the design of the analysis.

In general, the results for the Atlanta and Los Angeles case for the factors previously investigated were not significantly different from the previous factorial analysis, which was to be expected. We will focus here on the results for the Atlanta and New York analysis which are more similar climates and cover the populous eastern US. The results for Miami and Denver offer a comparison of very contrasting climates with significantly different ambient conditions.

Table 18: Factors for zoning fractional factorial analysis

Factor	Base	Test
Latent Case Credit (LHR)	Manufacturers data	Increase latent fraction by 50%
Refrigeration Capacity (CC1)	Nominal	Increase capacity by 30%
Case Line Up (CC2)	Nominal	30% of medium temp converted to low temp coffin cases
Humidity	Single Sales zone @ 55% RH	Separate Sales zone for low temperature cases at 40%RH setpoint, main Sales zone at 55% RH
Location-Run 1	Atlanta	Los Angeles
-Run 2	Atlanta	New York
-Run 3	Miami	Denver

Table 19: 2_V^{5-1} fractional factorial design

Run No.	LHR	CC1	CC2	Humidity	Location
1	Base	Base	Base	Base	Test
2	Test	Base	Base	Base	Base
3	Base	Test	Base	Base	Base
4	Test	Test	Base	Base	Test
5	Base	Base	Test	Base	Base
6	Test	Base	Test	Base	Test
7	Base	Test	Test	Base	Test
8	Test	Test	Test	Base	Base
9	Base	Base	Base	Test	Base
10	Test	Base	Base	Test	Test
11	Base	Test	Base	Test	Test
12	Test	Test	Base	Test	Base
13	Base	Base	Test	Test	Test
14	Test	Base	Test	Test	Base
15	Base	Test	Test	Test	Base
16	Test	Test	Test	Test	Test

7.5.2 Refrigeration Energy Results

Figure 37 and Figure 38 show the effect of the five factors on refrigeration energy use for the Atlanta-New York and Miami-Denver pairs of locations. The largest interaction was between the capacity increase and the case lineup, however in Miami and Denver the interactions with the location also became more significant, due to the more significant climate differences. The humidity and zoning factor reduced refrigeration energy consumption by just over 2% due to the lower environmental humidity and temperatures in the areas of the freezer cases. While still small compared to the 20% and 13% effect of the refrigeration system factors, it offers the first real opportunity to influence refrigeration energy consumption. The modest interaction between humidity and location suggest that the exact savings are somewhat dependent on location.

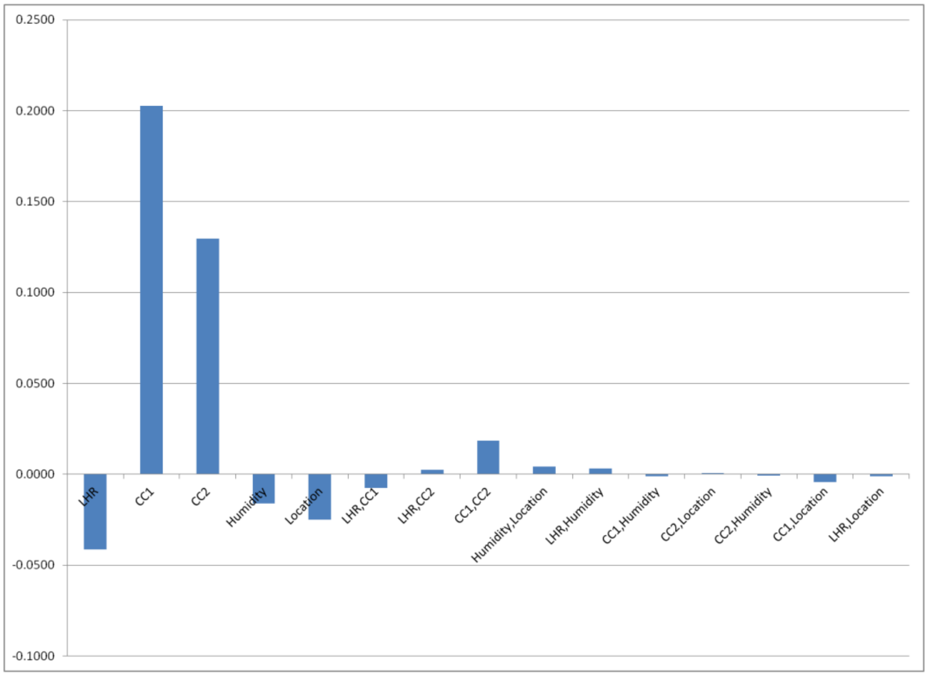


Figure 37. FFA2 Refrigeration, Atlanta & NY - Main Effects and Interactions

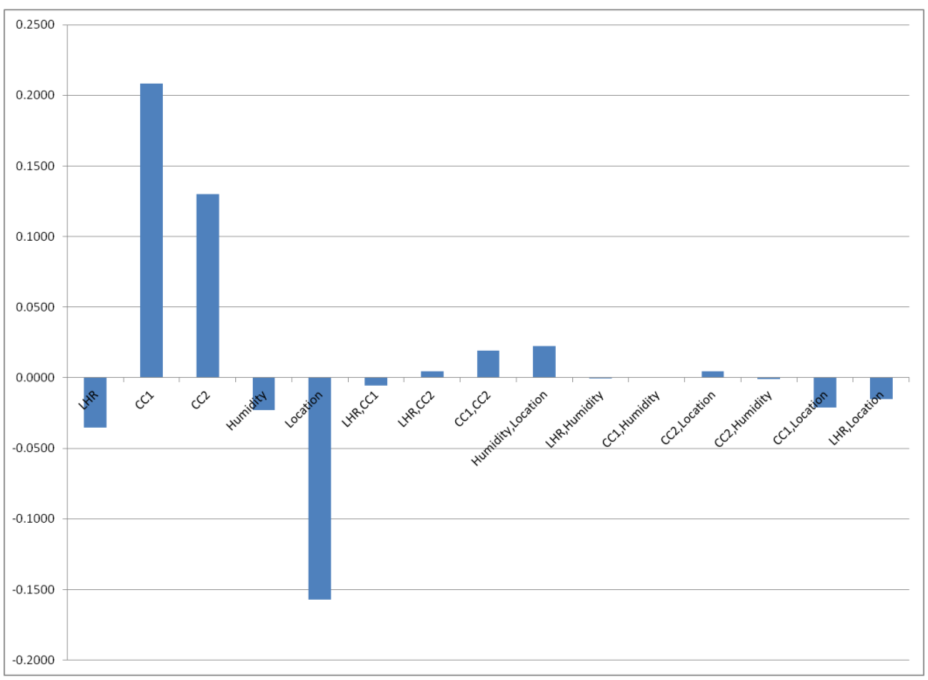


Figure 38. FFA2 Refrigeration, Miami & Denver - Main Effects and Interactions

7.5.3 HVAC Cooling Energy Results

Figure 39 and Figure 40 show the effect of the five factors on cooling energy use for the Atlanta-New York and Miami-Denver pairs of locations. As in the previous factorial analysis, cooling energy use is strongly influenced by many factors, with the potential for an overwhelming influence of location. While the climatic differences between Atlanta and New York produce only 2.5% difference in refrigeration energy use, cooling energy changes by 30%. Cooling in Denver is very small, producing a relative impact of 175% compared to Miami. Refrigeration system characteristics have similar influence to that previously discussed.

The main new information here is the impact of humidity zoning. While maintaining a lower humidity in the freezer aisles reduced refrigeration energy use by about 2%, it increases cooling energy by 80%, though the exact increase is clearly dependent on location. For the Miami-Denver analysis, the average increase is only 60% largely due to the small cooling needs in Denver. In all cases, a closer examination revealed that the large cooling increase was due to two competing factors. First, the lower humidity in the freezer zone required more cooling for dehumidification and subsequent reheat. Second, by isolating the freezer cases to a smaller zone, their cooling effect on the rest of the sales area was diminished, which increased the need for sensible cooling in the main sales zone.

In both figures, the humidity-location interaction is relatively large, reinforcing that the impact of humidity zoning and location influences are coupled. Similar interactions are shown among humidity, location, and the refrigeration parameters. While the impacts of all five variables on cooling energy are large, the size of the impact of any one variable is significantly influenced by the values of the other variables.

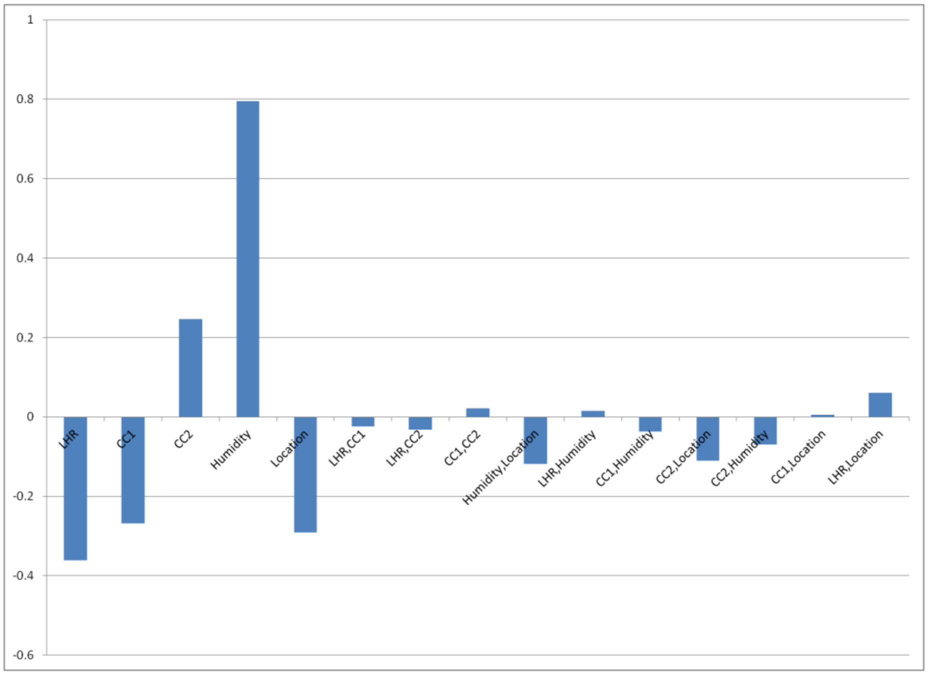


Figure 39. FFA2 HVAC Cooling, Atlanta & NY - Main Effects and Interactions

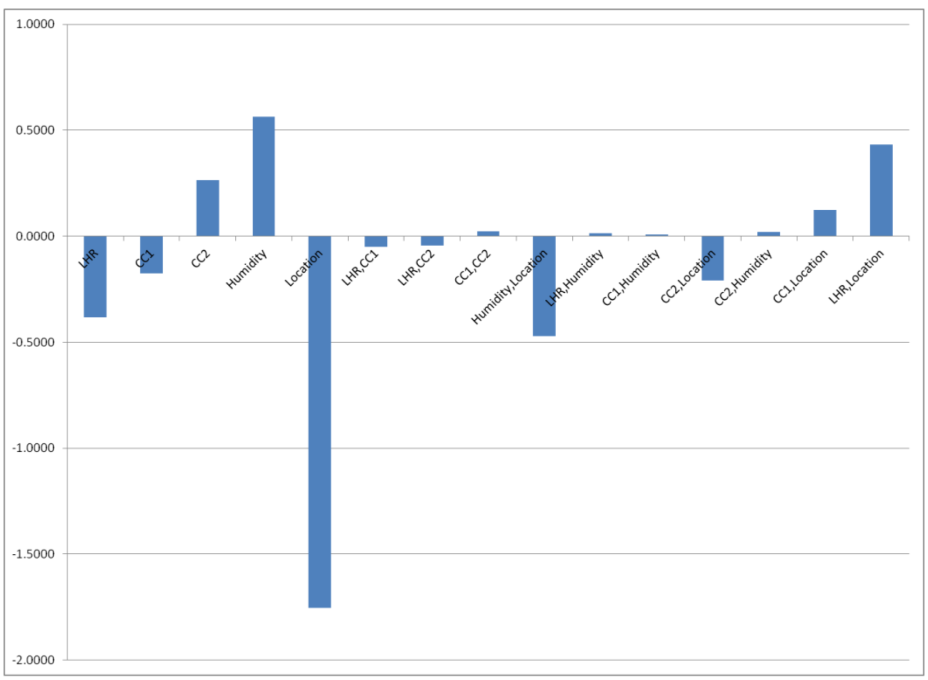


Figure 40. FFA2 HVAC Cooling, Miami & Denver - Main Effects and Interactions

7.5.4 HVAC Fan Energy Results

Figure 41 and Figure 42 show the effect of the five factors on fan energy use for the Atlanta-New York and Miami-Denver pairs of locations. The main conclusion is that location dominates the fan energy use. The reason for the influence is that space heating needs typically dictate fan sizing, which then influences fan energy throughout the year.

For the Atlanta-New York results, the influence of location is less than 2% and no factor or interaction affects fan energy by more than 5%. However, the higher heating requirements of Denver increase fan energy use by over 40% in the comparison with Miami.

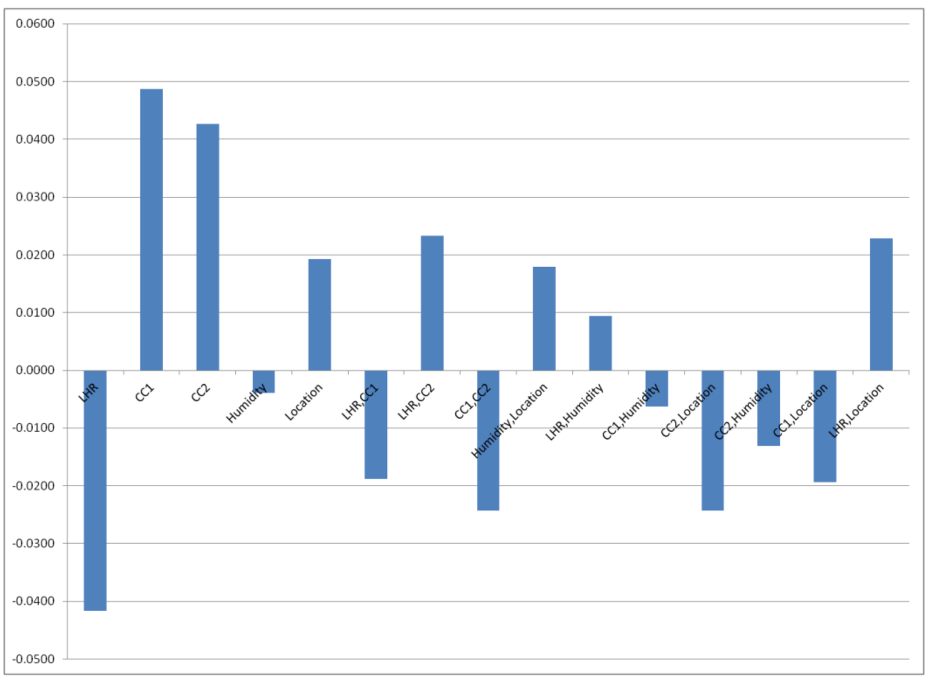


Figure 41. FFA2 HVAC Fans, Atlanta & NY - Main Effects and Interactions

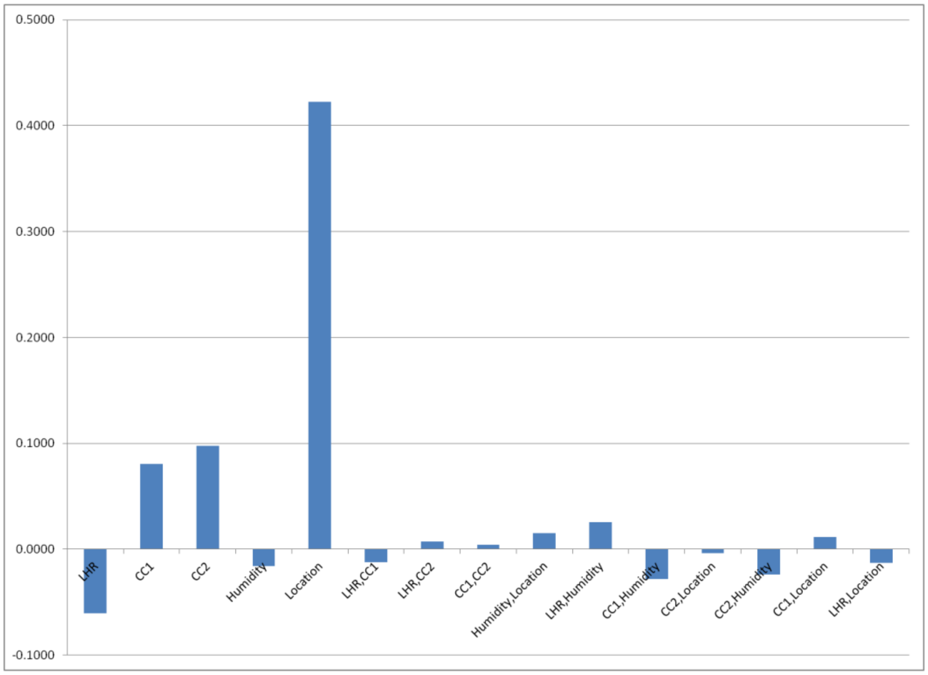


Figure 42. FFA2 HVAC Fans, Miami & Denver - Main Effects and Interactions

7.5.5 HVAC Electricity Results

Figure 43 and Figure 44 show the effect of the five factors on HVAC electricity use for the Atlanta-New York and Miami-Denver pairs of locations. The electricity results merge the cooling and fan results into a single metric and reflect the relative impact of the two factors on HVAC electricity. The coupling of refrigeration and space conditioning are clearly evident, with changes in both the humidity zoning and the refrigeration system parameters exerting significant influence. Again, climatic conditions can have a dominant influence when comparing very humid and dry location.

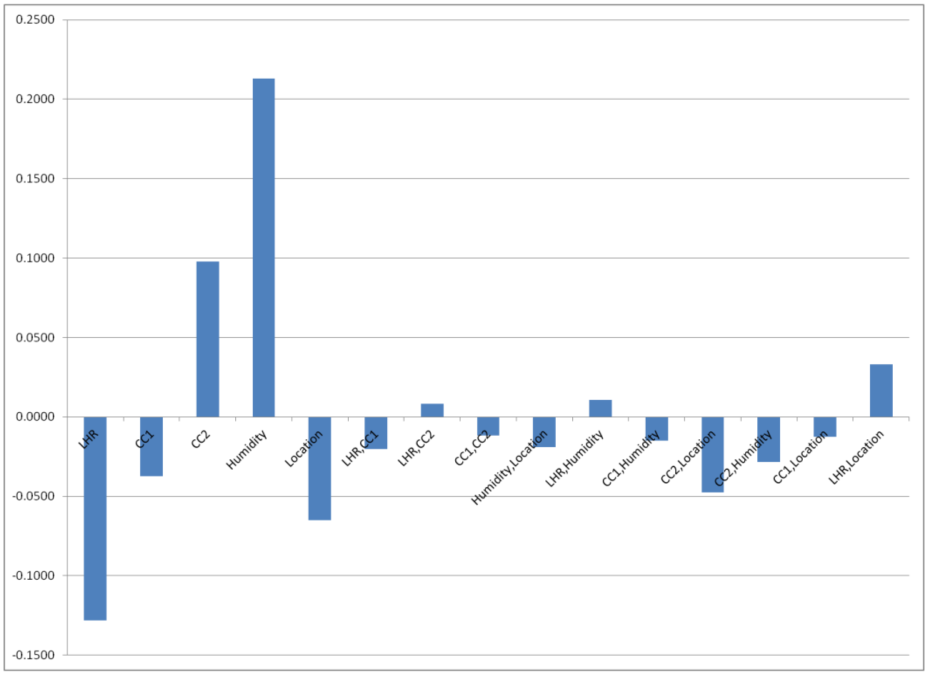


Figure 43. FFA2 HVAC Electricity, Atlanta & NY - Main Effects and Interactions

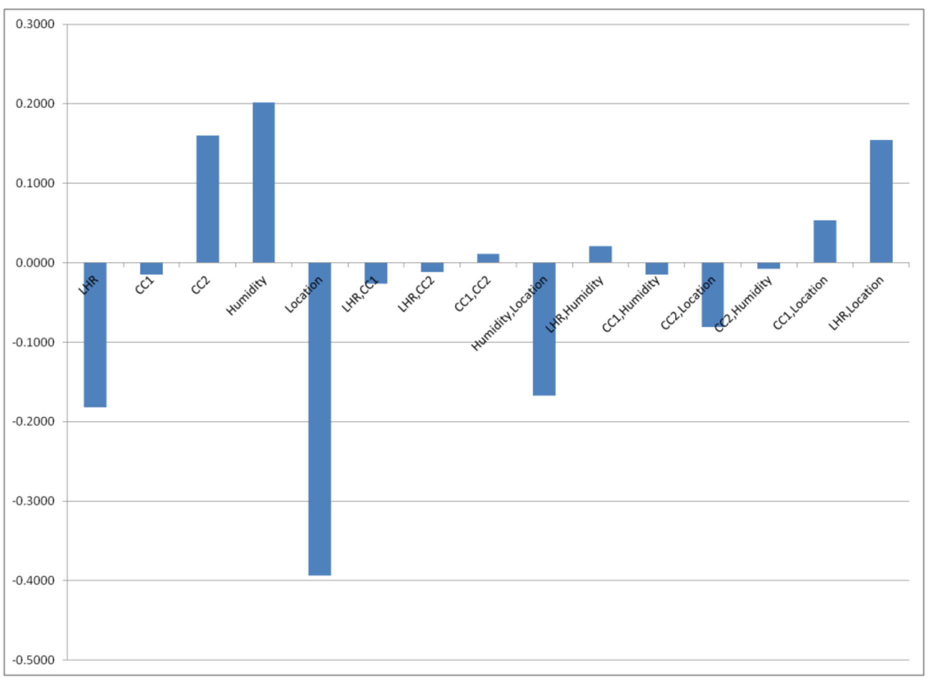


Figure 44. FFA2 HVAC Electricity, Miami & Denver - Main Effects and Interactions

7.5.6 HVAC Gas Energy Results

Figure 45 and Figure 46 show the effect of the five factors on gas energy use for heating and dehumidification reheat for the Atlanta-New York and Miami-Denver pairs of locations. Since most gas is required for heating when the reclaimed condenser heat is insufficient, the main influences are location and refrigeration system factors. The increased heating that might be required in the freezer zone as a result of the rezoning is not seen in the results shown below since the heating load is provided by reclaimed heat from the refrigeration system.

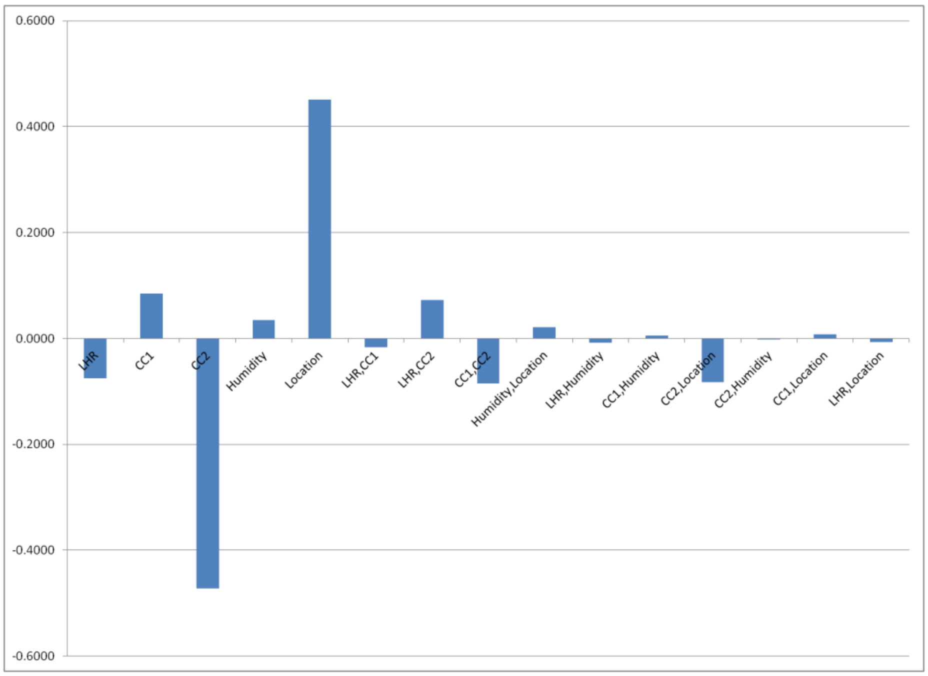


Figure 45. FFA2 HVAC Gas, Atlanta & NY - Main Effects and Interactions

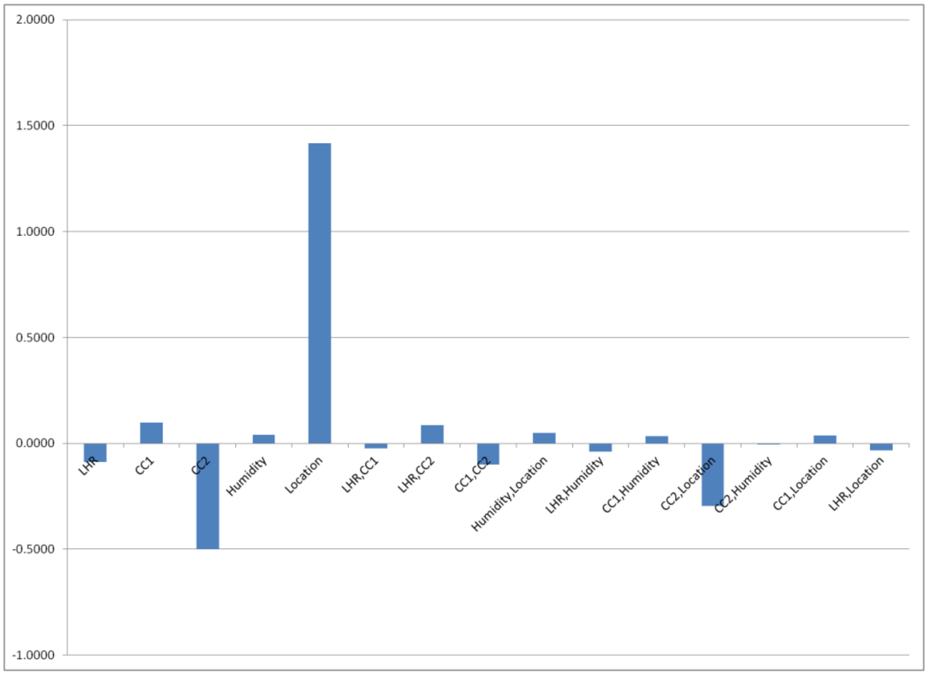


Figure 46. FFA2 HVAC Gas, Miami & Denver - Main Effects and Interactions

7.5.7 Building Source Energy Results

Figure 47 and Figure 48 show the effect of the five factors on building source energy use for the Atlanta-New York and Miami-Denver pairs of locations. In addition, Figure 49 shows the results for the Atlanta-Los Angeles pair of locations. As observed in the previous analysis, refrigeration system factors can influence building energy use by 4% - 15%. The large effects of humidity zoning observed in the HVAC electricity results are drowned out by the much larger refrigeration effects. The net impact of humidity zoning on building energy use is almost imperceptible, independent of location. The modest relative impacts of the interactions involving humidity zoning show that other factors are not significantly influenced by the humidity zoning factor.

The largest interaction was between the capacity increase and the case lineup, however in Miami and Denver the interaction between the capacity increase and the location also became more significant, due to the larger difference in ambient humidity between the two climates.

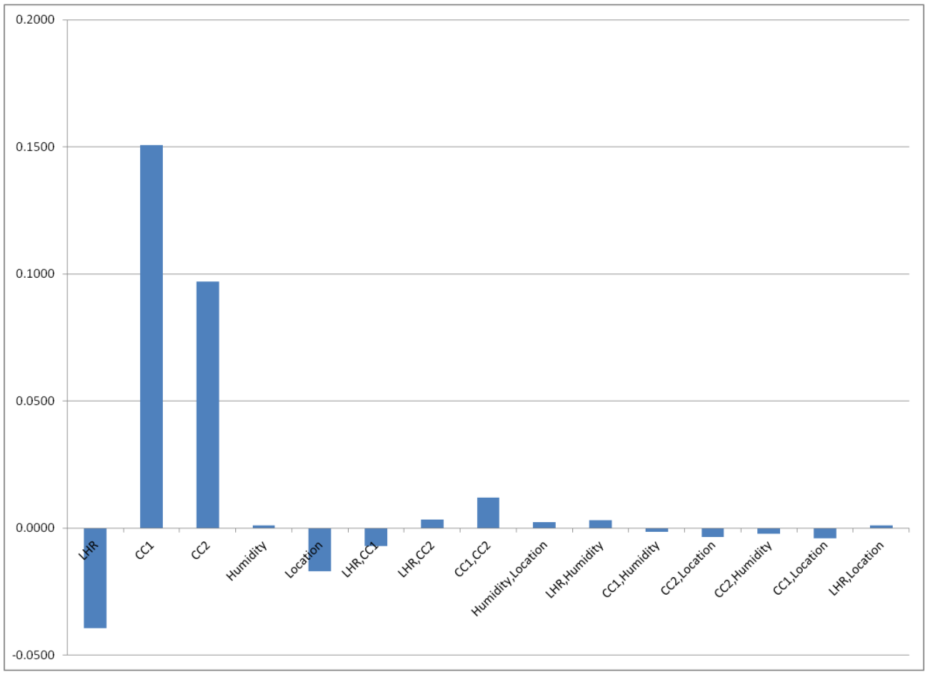


Figure 47. FFA2 Building Energy, Atlanta & NY - Main Effects and Interactions

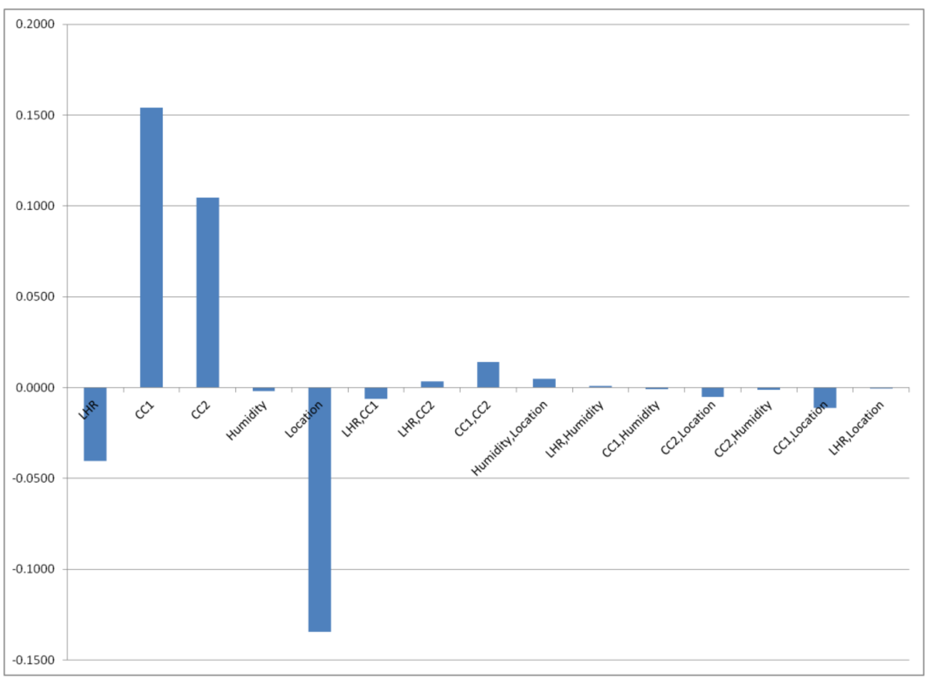


Figure 48. FFA2 Building Energy, Miami & Denver - Main Effects and Interactions

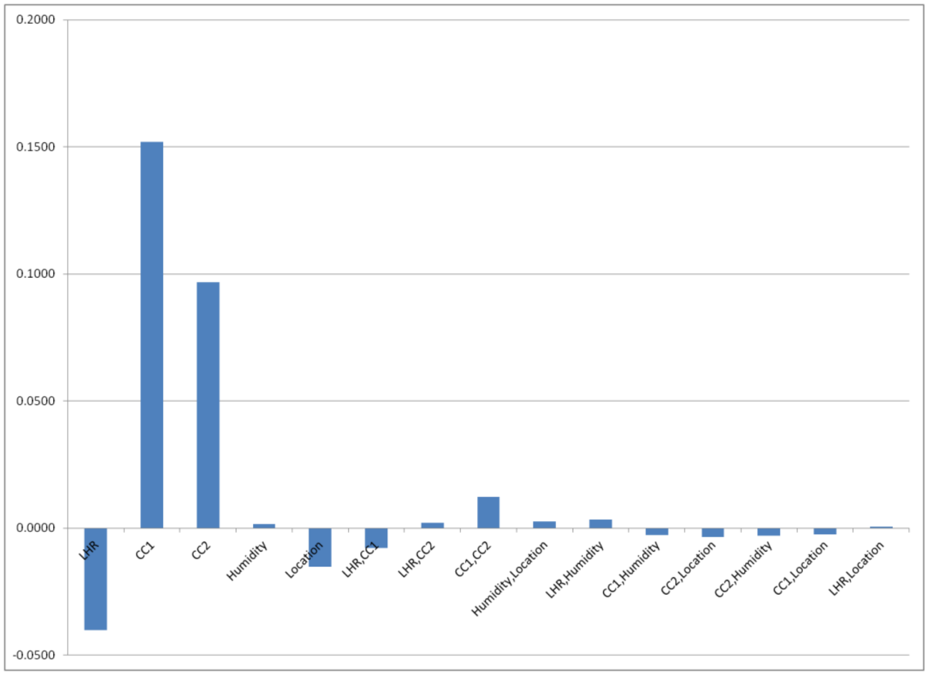


Figure 49. FFA2 Building Energy, Atlanta & LA - Main Effects and Interactions

7.5.8 Summary

The zone factorial analysis was performed to explore the impact of humidity zoning compared with other key factors identified in previous analyses. The results indicate that, while humidity zoning can have a significant influence on the individual energy end use components, the net effect is nearly nil. The dominant variables seen in this study were again the latent case credit, refrigeration capacity, and case line up, reinforcing that the refrigeration variables are far more dominant over the HVAC effects, and that the initial assumptions of the project around the ability of the HVAC systems to have significant impacts on total building energy use may have been overestimated.

Chapter 8

8 Conclusions

8.1 Summary of Project

The overall objective of this project was to provide an assessment of the potential for energy savings due to humidity in supermarkets by optimizing the design and operation of the combined HVAC and refrigeration systems. There is a strong interaction between a supermarket's HVAC system and its refrigerated display cases. Store air exchanges both moisture and heat with the refrigerated cases.

The research was approached in four main sections:

- Selecting an energy modeling program and validating its results
- Developing a Prototypical Store
- Conducting a sensitivity analysis on common supermarket characteristics and then a parametric study of 25 significant factors in supermarket design
- Conducting a fractional factorial analysis on a reduced number of refrigeration and HVAC factors

This study made use of EnergyPlus as the simulation tool for modeling supermarkets, and generally found good agreement with either measured results or expected outcomes on an individual variable level.

The matter of heat reclaim from the refrigeration system was an issue, and one that has since been resolved by the developers of the program, however the resolution has not been included in the results herein.

A prototypical store was developed based on the DOE Commercial Buildings Benchmark model, and surveys of practitioners in the supermarket industry. Expansions to this model were made based on the results of sensitivity analyses undertaken regarding common characteristics.

The prototypical store was modeled in EnergyPlus with hundreds of variations to explore the sensitivity of supermarket performance to the impact of design and operation. The factors included store layout, building envelope design, internal heat gains, store operations, airflow design, refrigeration system size and layout, and HVAC system design and operation. Factorial analyses were conducted to determine the effects of the interactions of a selection of the most important factors.

8.2 Summary of Findings

The research produced a number of findings and conclusions that covered a wide range of issues of supermarket design and operation. While many of the decisions that influence supermarket energy consumption are driven by consumer relations and food marketing, it is important for the refrigeration and HVAC designer to recognize the broader factors.

An informal survey of supermarket industry trends was conducted which included review of prototypical store drawings and designs from several supermarket companies. Several conclusions were drawn from the survey.

- Refrigeration systems in supermarkets are large and growing. There is little indication that modern supermarkets in the near future will reduce the size of refrigeration systems.
- Process loads in the kitchen and bakery areas of supermarkets loads are largely exhausted through hoods and the remainder are usually confined to a single HVAC zone.

- The choice between doored and open refrigerated cases is generally determined by marketing. However it is well documented that doored cases are a more efficient choice and trends indicate a steady adoption of doored cases.

Preliminary analysis included an evaluation of basic supermarket store design options and produced several conclusions.

- Supermarket building envelope performance has little influence on store energy performance. A high performance supermarket is characterized by the performance of its systems, not the building.
- While the size of the supermarket can clearly influence energy use, the size of the refrigeration system has a much greater impact. In general, refrigeration energy use accounts for two-thirds of the total energy consumption in a modern supermarket.
- Through the use of CFD and testing of zone mixing influence, it was shown that the effects of refrigeration on store conditions are quite localized. The "superstore" concept, where a portion of a large retail store comprises a supermarket, can effectively be analyzed using independent models of the supermarket and retail areas without significant interaction.
- When considering a superstore, the designer should treat the supermarket and the larger dry goods area of the store as separate entities. While active dehumidification should be used in the supermarket areas, there is no need to apply active dehumidification to the dry goods for the purpose of maintaining supermarket humidity levels.

The main body of the analysis focused on the refrigeration and HVAC system design and operation.

- Refrigeration system energy consumption is typically an order of magnitude greater than HVAC system energy consumption. This disparity continues to grow as refrigeration systems grow larger while building envelopes improve and lighting systems become more efficient.

- Mechanical subcooling and floating head pressure can dramatically reduce refrigeration system energy consumption.
- Almost all of the store heating needs, for space heating and reheat after dehumidification, can be met by heat reclaim from the large refrigeration system. A saturated condensing temperature of 40°C is generally sufficient to meet the needs, though a detailed optimization was not performed.
- Refrigeration system factors dominate the energy consumption in supermarkets and balance between refrigeration system and space conditioning needs. The most important factors are the overall capacity of the refrigeration system, the case lineup, and the extent to which the cases remove heat and moisture from the supermarket spaces.
- For a given refrigeration system capacity, the layout of cases with the store and the specific types of cases used in the spaces can have a very significant impact on annual energy use of both the refrigeration and HVAC system. Simply changing the case line while maintaining the same refrigeration system capacity can change total building energy consumption by as much at 10% due to case energy use and the interactions with the HVAC systems.
- A well-designed HVAC system is sufficient. Advanced design strategies of airflow reduction, demand control ventilation, and heat pipe heat exchangers will have effects of 0.3-1.0% on total electricity use. It appears that the initial assumptions of the project around the ability of the HVAC systems to have significant impacts on total building energy use were overestimated. While even these small percentages of the energy consumption of a large supermarket may result in significant absolute cost savings, it is unlikely that costly or complex strategies are justified.
- Rezoning the HVAC system and adjusting the humidity setpoints had been expected to have a significant impact. It has long been proposed that lower zone humidity, achieved through higher HVAC system energy use, would reduce refrigeration system energy consumption to achieve a

net store energy savings. Our results indicate net savings within ±0.2% in store energy consumption across the 5 climate zones examined.

- While HVAC system factors do not have significant influence on store energy, relatively small changes or uncertainties in refrigeration system characteristics can have a dominant influence on the sizing and energy use of the HVAC system. HVAC design engineers must work closely with store designers to include consideration of case layout and case performance for proper system sizing. Even modest redesign of case layout or lineup should involve close coordination with HVAC design.
- In almost all locations, the temperature and relative humidity in the sales area were easily maintained below their cooling setpoints of 23.8°C (75°F) and 55% RH. Reducing the humidity setpoint to 40% RH caused significant overcooling and reheat to achieve the setpoint, requiring twice the cooling energy use. By comparison, the refrigeration system energy use was reduced by only 2-3%, reflecting the reduced sensitivity of doored refrigeration case performance to store conditions.
- The effects of climate on supermarket energy consumption are smaller than most other building applications. Location affects HVAC system design, including fan and heating sizing, and influences the indoor temperature and humidity. Location has relatively little effect on the refrigeration electricity use in climates as diverse as Atlanta and Los Angeles because the annual average dry bulb temperature is similar despite the variation in seasonal swings.

8.3 Future Work

Given the outcomes of this research there are some topics that may still be worth further exploration. Specifically these include:

- Building energy simulation tools for supermarket design and analysis would benefit from more sophisticated refrigeration system and refrigerated case models and improved modeling of heat recovery from refrigeration systems for other building applications.
- Economic analysis of the savings that can be made from the systems analyzed, including capital cost assessments to determine payback periods.
- Given that the conclusions have determined that the refrigeration systems are the dominant factor in the outcomes, but they were not the driving force in the planning and design of this project, it may be appropriate for a further study to be conducted focused more solely on the refrigeration in the supermarkets.

Bibliography

1. D. Kosar, O. Dumitrescu, Humidity Effects on Supermarket Refrigerated Case Energy Performance: A Database Review, ASHRAE Transactions, Vol 111, 2005.
2. ASHRAE, ASHRAE Handbook – HVAC Applications, I-P ed, ASHRAE, Atlanta, 2007.
3. R.S. Pitzer, M. Malone, Case Credits and Return Air Paths for Supermarkets, ASHRAE Journal, February 2005.
4. R. Faramarzi, Showcasing energy efficient emerging refrigeration technologies. Emerging Technologies in Energy Efficiency Summit, October 2004.
5. Arias, J. 2005. Energy Usage in Supermarkets - Modeling and Field Measurements. Department of Energy Technology. Stockholm, Sweden, Royal Institute of Technology.
6. Baxter, V.D. 2003. IEA Annex 26: Advanced Supermarket Refrigeration/Heat Recovery Systems, Final Report Volume 1.
7. eQuest. 2006. DOE-2 Volume 2r Dictionary - Refrigeration Systems
8. Khattar, M. K. and H. I. Henderson. 2000. Experiences with Modeling Supermarket Energy Use and Performance. Presented at the IEA Supermarket Refrigeration Workshop, 2-4 October 2000, Stockholm, Sweden.
9. Tassou, S.A. and Ge, Y.T. 2000. Modeling of the Energy and Environmental Impact of Supermarket Environmental Control Systems. Brunel University, Uxbridge, Middlesex, UK.
10. Weather Underground. November 2013.

    http://www.wunderground.com/history/

11. Box, G.E.P., Hunter, W.G., and Hunter, J.S. 1978. Statistics for Experimenters: An Introduction to Design, Data Analysis and Model Building. John Wiley & Sons, Inc.

12. NIST/SE-MATECH. February 2010. National Institute of Standards and Technology, e-Handbook of Statistical Methods. http://www.itl.nist.gov/div898/handbook/index.htm

13. DOE Commercial Buildings Benchmarks V1.2 http://www1.eere.energy.gov/buildings/commercial_initiative/reference_buildings.html

14. USDOE. 2009. EnergyPlus Version 4.0 Documentation.

15. van der Sluis, S. M. 2004. Review of calculation programs for supermarket DX refrigerating systems year-round energy consumption. Presented at the 6th IIR Gustav Lorentzen Conference on Natural Working Fluids, Glasgow, Scotland.

16. Energy Star, 2011. Methodology for Incorporating Source Energy Use http://www.energystar.gov/ia/business/evaluate_performance/site_source.pdf

17. Laurie, L. 2011. Personal communication.

Appendix A

CFD Analysis

Abstract

Supermarkets are energy intensive buildings, and energy savings are available in the operation of the store refrigeration systems by operating the stores at lower humidity levels. This incurs an energy penalty on the HVAC system however. This paper utilizes computational fluid dynamics (CFD) to investigate the anecdotally understood concept that latent heat absorption by the refrigerated cases is localized, with the intention of realizing energy savings by operating the refrigerated case sections of stores at lower humidity levels than the dry goods area, thus reducing the dehumidification load.

Modeling was undertaken of an actual store, and results compared with measured data for validation. Further CFD models were undertaken with different refrigerated case layouts to demonstrate the humidity profiles created by differing layouts.

Results showed good agreement with predictions, that humidity levels in the freezer aisle are lower than in the dry goods area, and that locating more refrigerated cases together enhances this effect. This shows the potential for optimization of the relationship between refrigeration design and HVAC system design for the purposes of energy savings.

Keywords: CFD simulations, Supermarkets, Humidity profiles, Energy Savings

1. Introduction

Supermarkets are the most energy intensive buildings in the commercial sector, and are responsible for approximately 54.5 billion kWh of electricity annually. Refrigeration makes up approximately 50% of this electricity use. While the outdoor condensing temperature is the dominant effect on the refrigeration energy use, the store temperature and humidity can also have significant impact [1].

There is a strong interaction between a supermarket's HVAC system and its refrigerated display cases. Conditioned store air exchanges both moisture and heat with the refrigerated cases. Increases in store humidity impose heavier loads on the refrigeration equipment and causes sweating on products and shelving, as well as frost on evaporator coils [2][3].

Modern supermarkets have a high percentage of refrigerated cases with glass doors, which somewhat reduces the problem of both sensible and latent heat exchange with the environment by reducing the air change rates. However the doors require anti-condensation heaters and not all retailers accept these cases for all display types [2].

However, reducing the store humidity level can have a positive effect on the refrigerated case energy use. Reducing the conditions from 55% to 35% relative humidity (RH) has been demonstrated to produce an 18% decrease in compressor power demand for open case refrigeration [4]. However dehumidification has an energy penalty on the HVAC system, and this penalty can be large when considering the volume of air that a typical supermarket HVAC system deals with.

If it can be proven that the dehumidifying effect of the refrigerated cases was localized, then supermarket HVAC design could be modified such that dehumidified air was provided to the case area, while the dry goods area was maintained at a higher humidity ratio. Typically energy savings can be realized within a comfort band of 30-60% RH [1]. This has the potential to save considerable energy in the refrigeration system, while minimizing the energy penalty on the HVAC system.

Studies have observed humidity profiles in supermarkets, as in Rosario [5], where differences of up to 20% RH were measured at five data points located in one store in a study in Florida, and others have examined the potential for energy savings from reducing humidity levels [1][9], but no CFD verification has been undertaken regarding large scale humidity profiles in supermarkets. CFD modeling has been limited to the modeling of specific display cases. However the results of these scenarios have been observed anecdotally for many years in supermarkets around the world.

The intention of this study was to make use of Computational Fluid Dynamics (CFD) to analyze how much of the dehumidifying effect of the cases was localized, or whether the traditional assumption of a fully mixed space made by energy simulation programs was valid. In order to test this, an actual store was modeled in Phoenics, and the results compared with measured data from testing that had been undertaken in the store. Variations on the layout of this store were also then tested to determine the impact that the layout of refrigeration had on humidity profile.

2. Methodology

2.1 Description of Base Model

The building used for this study was an actual store located in Berkeley Heights, New Jersey. The store had been studied previously and hence data existed regarding the temperature and humidity at various locations through the store for comparison. This data was used to validate the CFD model.

The store was created as a 65m x 42m x 4m high domain, containing refrigerated cases, as well as internal loads. Peripheral back of house spaces, and slightly non-rectangular elements of the actual store were neglected in the creation of the simplified CFD model.

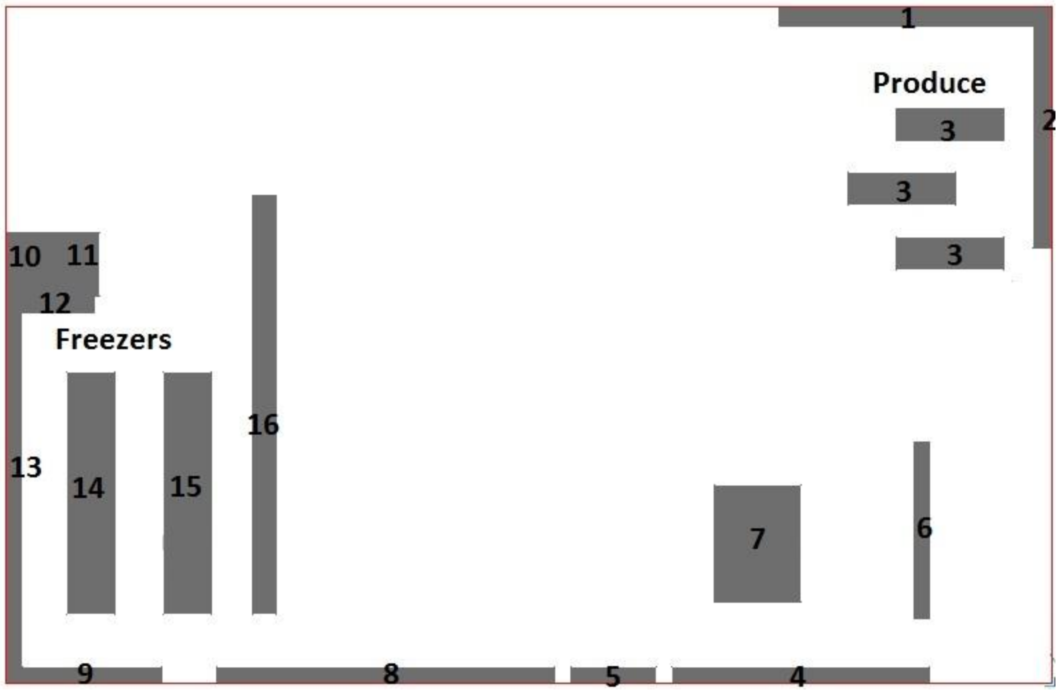


Fig. 1. Plan of the CFD domain showing case layout (numbers refer to case lineup in Table 1)

Refrigerated cases were included in the model based on the capacity of actual cases in the existing store. The cases were merged into simple blocks for the purposes of modeling, however the locations were the same as in the actual store. In order to determine the humidity absorption by the refrigerated cases a latent case credit was calculated for each case based on the case type using the guidelines set out in the ASHRAE handbook [2]. In addition a usage factor was applied to both cases with doors and coffin cases to allow for the reduced impact from these cases as compared to standard multi-deck cases [3]. Cases that were deemed to affect only the back of house conditions were ignored. The latent case credit was converted into a water removal rate by use of the following equation:

$$q = \dot{m}_w \times h_{fg}$$

where h_{fg} is the heat of vaporization of water. The case lineup is included in Table 1, and resulted in a total water removal rate of -12.15g_w/s.

	Description	Capacity (kW)	Latent Case Credit (%)	Latent Case Credit (kW)	Usage Factor	Water Removal (g/s)
1	Produce wall - open	13.7	15	2.06	1	-0.8247
2	Produce wall - open	15.8	15	2.37	1	-0.9478
3	Produce - coffin	15.9	20	3.17	0.5	-0.6342
4	Service deli	3.8	15	0.58	1	-0.2304
5	Service meats	3.8	15	0.57	1	-0.2268
6	Meats - open	16.4	20	3.27	1	-1.3083
7	Cheese - open	6.7	20	1.35	1	-0.5392
8	Meat cases - open	29.0	20	5.80	1	-2.3211
9	Service seafood	1.1	15	0.17	1	-0.0686
10	Bakery cooler - door	3.0	15	0.45	0.35	-0.0634
11	Bakery freezer - door	7.4	15	1.12	0.35	-0.1563
12	Frozen food - door	3.5	20	0.70	0.35	-0.0977
13	Frozen food - door	19.0	20	3.80	0.35	-0.5326
14	Ice cream - coffin	11.6	15	1.74	0.7	-0.4874
15	Frozen food - door	32.8	20	6.56	0.35	-0.9191
16	Dairy - open	34.9	20	6.99	1	-2.7947
	Total					**-12.15**

Table 1. Refrigerated Case Latent Credits, Use Factors, and Water Removal Rates

Target store conditions were 23.9°C and 55% RH, based on the report on the original work done at the store [7], these conditions are fairly typical for many supermarkets.

The lighting levels were taken from the original store construction drawings from 1990 to be 26W/m².

Occupancy loads were included as per ASHRAE 62.1 [6], which suggests 8 people per 100m² for supermarkets, with loads of 75W sensible and 55W latent per person. Infiltration loads for the store were assumed to be approximately 0.5ACH, and for the purposes of the simulation it was assumed that the outside conditions were 28°C and 55% RH.

Thus, in addition to the lighting, sensible loads on the space were 8.7W/m², and total latent additions to the space were equivalent to a water addition rate of $22g_w$/s.

For the purposes of simplification it was assumed that we have adiabatic walls, we neglected external loads and other internal sensible loads since the sensible loads are not the focus of this study. A refinement of this assumption could be the focus of further work. Sufficient sensible loads are present that buoyancy effects exist.

Conditioned air was supplied to the store at 18°C and a humidity condition sufficient to ensure a latent heat balance in the space, which in this case was 9.2 g_w/kg_{air}, and two return grilles were located in the ceiling towards the rear of the store. The total supply air volume was 13.2m³/s.

The airflow pattern in the space was modeled to reflect the actual store, but was simplified. Air distribution was modeled through four rows through the store. The simulation did not model each diffuser, but did look at the split of airflow from front to back of the store. 6m³/s was delivered at the front through a slot diffuser in the front wall, while the rest was through ceiling mounted slots, 4m³/s was delivered one quarter of the way back from the front, 2.2m³/s was delivered just over half way back, and 1m³/s was delivered at the back of the store.

2.2 Model Construction

The models were run in the CFD software Phoenics (v2009), making use of the special purpose program Flair within Phoenics that is designed for indoor environments.

The turbulence model selected was the k-∊ renormalization group (RNG) model as studies have shown that this has been widely used for indoor environments and performs well [8]. The Boussinesq approximation was used for the buoyancy model. The model was run as a steady state problem.

Grid independence was analyzed using the Normalized Root Mean Square method and determined at 126x190x12. Each model took 4000 iterations to reach convergence.

CFD programs make use of numerical techniques to solve equations for mass, momentum, and energy, which can be written in the general form as:

$$\frac{\partial \rho \phi}{\partial t} + \frac{\partial \rho U_j \phi}{\partial x_j} = \frac{\partial}{\partial x_j}\left(\Gamma_{\Phi,eff} \frac{\partial \phi}{\partial x_j}\right) + S_\phi$$

where F represents the value 1, velocity, and temperature in each equation respectively. This equation can also be used as the scalar transport equation to solve for a contaminant.

The humidity was solved by treating the humidity ratio as a contaminant in the scalar transport equation, and then converting this humidity ratio to relative humidity manually for comparison to measured data as required.

2.3 Simulation Descriptions

In addition to the Base Case four further models were simulated to demonstrate the humidity profiles caused by the refrigerated cases. The models were:

- Dehumidification Case
- Middle Location
- Side Location
- Scattered

These models were set up as described in the following sections.

2.3.1 Dehumidification Case

The first variation on the Base Case related to actual modifications that took place in the store. The store was modified during 1995 to include an all-electric desiccant dehumidification system to serve the freezer aisle area. This modification was made as part of a trial to take advantage of the fact that at lower humidity levels savings in refrigeration energy can be realized [1].

This area was supplied air to achieve a target of 21.6°C and 50% RH locally in the freezer area. The target RH in the rest of the store was increased to 60%. The volume of air supplied was 3.3 m^3/s. The airflow for this scenario was modeled as shown in Figure 2, and involved dedicated supply air diffusers in the freezer area. The return air diffuser was also relocated into the dry goods area.

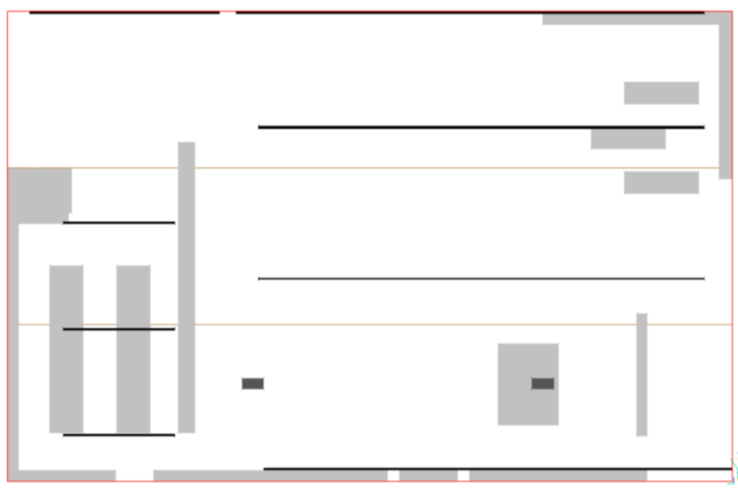


Figure 2: Plan of Inlet and Outlet Airflows in Dehumidification Case

2.3.2 Refrigeration Layout

The other scenarios which were modeled involved moving the refrigerated cases around the store. These layouts were modeled with the original airflow pattern. The layouts were as follows:

- Middle Location – the freezers and other cases located around the freezers were relocated to the centre of the store.
- Side Location - all of the cases except those along the back wall and the bakery cases were relocated to the same side of the store as the produce cases.
- Scattered – all of the cases were scattered around the store so that no cases were grouped together. While this does not represent a realistic scenario, since some cases are always grouped for reasons of refrigerant piping, stocking simplicity, store planning etc., it does serve to illustrate an alternative humidity profile.

3. Data Analysis

3.1 Comparison to Measured Data

The measurements were taken in the store in 1995 after the installation of the dehumidification system. A relative humidity sensor was installed in the freezer aisle to measure conditions, while a second measurement listed as a humidistat measurement was actually an average of several sensors located around the store.

Comparing the measured data to the data from the CFD models gives the results shown in Table 2.

	Measured	CFD
Store Typical	60%	58% - Base Case
		61% - Dehum. Case
Freezer Aisle	50-52%	52% - Dehum. Case

Table 2. Comparison of measured data and CFD model results for outdoor conditions 28°C, 55%RH

It is important to note that since the sensible loads in the space were simplified, the temperature profiles cannot be considered to be accurate to a precise degree. As such, while the humidity ratio values might be accurate, the relative humidity values might swing as much as 5% with a 2°C temperature difference. For this reason, the rest of the results will be considered in the form in which they were calculated, humidity ratio, and future work should look into refining the sensible loads and hence refining the relative humidity profiles.

3.2 Analysis – Main findings

The findings were compared by taking readings at vertical poles in two locations in the freezer aisles, in the produce area, and in the dry goods area in each model, and taking averages of these poles. These averages were then compared between zones and between models, which is shown in Figure 3.

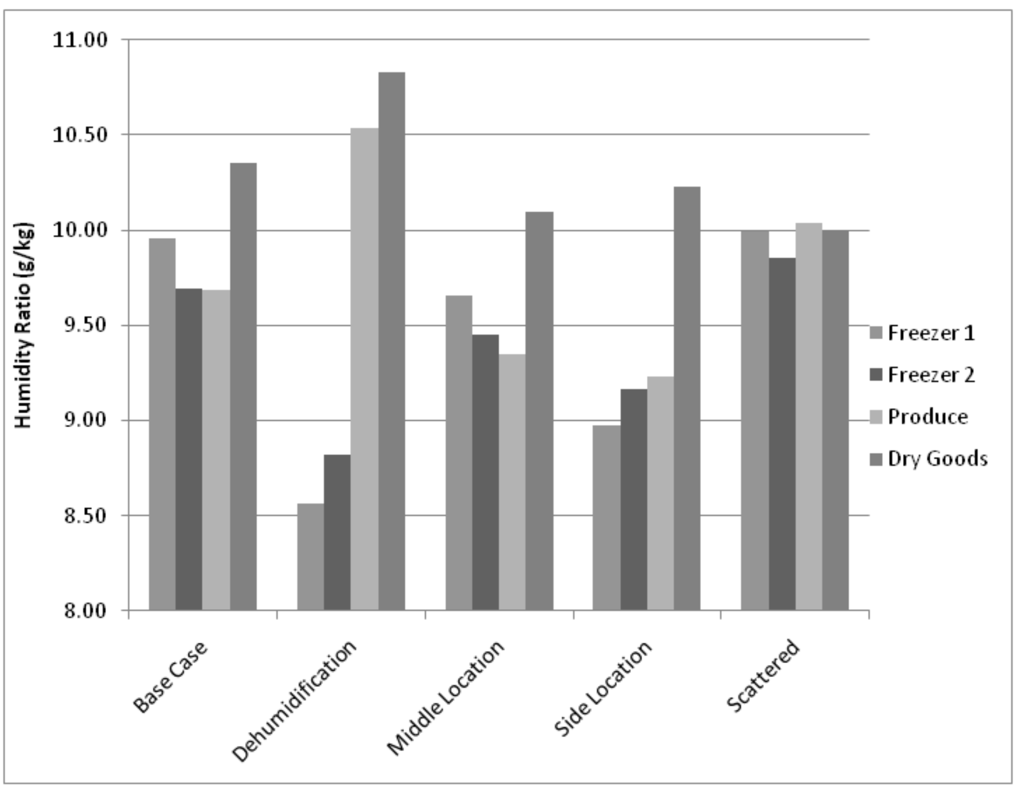


Figure 3. Plot of average humidity readings in different areas of models

The models show easily discernible localized dehumidification effects near to the refrigerated cases, these effects increased when the cases were gathered together in the Side Location model. The Scattered model clearly demonstrates that when the refrigerated cases are distributed around the store, the localized effect of the cases is no longer able to be discerned.

The findings were also compared to determine how much the horizontal average profiles of the store varied between the different models. Table 2 shows this data, examining humidity ratios in and around the occupied zone, and return air humidity and temperature.

It can be seen that despite the varying profiles that exist on a smaller scale the average data shows very close agreement between models.

Measurement	**Base Model**	**Dehum. Model**	**Middle Location**	**Side Location**	**Scattered**
Average HRAT @0m (g/kg)	9.8	9.8	9.7	9.8	9.9
Average HRAT @1.2m (g/kg)	10.0	10.0	9.8	9.9	10.0
Average HRAT @2.4m (g/kg)	10.0	10.1	9.8	10.0	10.0
Return Air HRAT Outlet 1 (g/kg)	10.0	10.1	10.0	9.5	10.0
Return Air HRAT Outlet 2 (g/kg)	9.6	9.4	9.8	10.4	10.0
Return Air Temperature 1 (°C)	24.0	24.4	24.0	24.0	23.9
Return Air Temperature 2 (°C)	23.9	23.6	24.0	24.0	24.0

Table 3. Comparison of average store humidity and temperature profiles between models

3.3 Discussion

The results clearly show that the latent effect of the refrigerated cases is localized when any number of cases is grouped together. This is evidenced the humidity profiles shown in all the Base Case and the

Middle Location model, the effects are shown magnified in the Side Location model, and effectively counteracted by the Scattered model.

It is important to note that the HVAC system design and layout has a significant impact on the magnitude of these effects, and on the energy impacts of these effects.

The dehumidification case which was reviewed was a retrofit case, so it is not surprising that it was not an optimal design, as the dehumidification was provided only to the freezer area. The open cases in the produce area and along the rear wall of the store were then exposed to higher humidity levels than previously, having a negative impact on the energy savings.

The base case store in this scenario has a single roof top air handling unit (RTU) serving the whole store. Often a store would be served by more than one unit, meaning that even if dedicated dehumidification is not provided it may be possible with good zoning to allow the refrigerated area of the store to operate at a lower humidity level than the dry goods area, since the return air will not be mixed together.

Additionally, more specific designs to provide dehumidification to air provided to the refrigerated case area can provide further energy savings in the refrigeration system. By providing dedicated RTUs to the refrigerated case areas with dehumidification and higher humidity levels to RTUs in the dry goods areas savings can be realized. Energy simulation studies would reveal potential optimal HVAC and refrigeration design strategies for energy savings, however practical studies have shown they can be significant. Tests from modifications to HVAC control strategies have shown refrigeration savings of 7.8kWh/day for each 1% drop in RH at one store, and 10kWh/day for each 1% drop in RH at another, at stores with 211kW and 281kW of installed refrigeration respectively [9].

While the latent effect is significant, the sensible cooling effect, which has been neglected in this study, must also be considered. Refrigerated cases also have a sensible cooling effect on their environment, clustering too many cases together may create reheat issues which will have a heating energy penalty.

4. Conclusion

The modeling undertaken confirmed the anecdotal evidence that the latent cooling caused by refrigerated cases in supermarkets does have a localized effect, with humidity levels in refrigerated areas of the stores typically being in the order of $1g_w/kg_{air}$ lower than in the center of the dry goods areas.

Based on previous studies that have been undertaken it would be expected that significant energy savings could be realized by optimizing the relationship between the refrigerated case design and layout, and the HVAC system design and layout, maximizing the refrigeration system savings while minimizing the dehumidification penalty to the HVAC system.

Future CFD work should focus on a more detailed model of the sensible loads to confirm whether the simplifications that have been employed in this modeling have impacted the buoyancy effects, as well as providing more detailed gradients to allow the calculation of accurate relative humidity profiles. In addition it would be interesting to examine the cold aisle effects of clustering the refrigeration, and the potential energy impacts from maintaining this area at comfortable temperatures.

References

[1] D. Kosar, O. Dumitrescu, Humidity Effects on Supermarket Refrigerated Case Energy Performance: A Database Review, ASHRAE Transactions, Vol 111, 2005.

[2] ASHRAE, ASHRAE Handbook – HVAC Applications, I-P ed, ASHRAE, Atlanta, 2007.

[3] R.S. Pitzer, M. Malone, Case Credits and Return Air Paths for Supermarkets, ASHRAE Journal, February 2005.

[4] R. Faramarzi, Showcasing energy efficient emerging refrigeration technologies. Emerging Technologies in Energy Efficiency Summit, October 2004.

[5] L. Rosario, R.H. Howell, Relative Humidity and Temperature and Temperature Measurements and Predictions in Supermarkets, ASHRAE Transactions, Vol. 107, 2001.

[6] ASHRAE, ANSI/ASHRAE Standard 62.1-2004 Ventilation for Acceptable Indoor Air Quality, ASHRAE, Atlanta, 2004.

[7] M. Brandemuehl, Demonstration and Testing of an All-Electric Desiccant Dehumidification System at Grand Union Supermarket #3250, Berkeley Heights, NJ, Joint Center for Energy Management, August 1997.

[8] Q. Chen, Z. Zhai, W. Zhang, Z. Zhang, Evaluation of Various Turbulence Models in Predicting Airflow and Turbulence in Enclosed Environments by CFD: Part 1: Summary of Prevalent Turbulence Models, HVAC&R Research, November 2007.

[9] H.I. Henderson, M. Khattar, Measured Impacts of Supermarket Humidity Level on Defrost Performance and Refrigerating System Energy Use, ASHRAE Transactions, Vol. 105, 1999.