DERIVATION OF WALK-IN COOLER AND FREEZER PERFORMANCE STANDARD EQUATIONS AS THEY PERTAIN TO THE ANSI/AHRI STANDARD 1250 AND 1251

A THESIS IN
Mechanical Engineering

Presented to the Faculty of the University
of Missouri-Kansas City in partial fulfillment of
the requirements for the degree

MASTER OF SCIENCE

by
BRYAN CURTIS SARTIN

B.S., University of Missouri – Rolla, 2008

Kansas City, Missouri
2012

© 2012

BRYAN CURTIS SARTIN

ALL RIGHTS RESERVED

DERIVATION OF WALK-IN COOLER AND FREEZER PERFORMANCE

STANDARD EQUATIONS AS THEY PERTAIN TO THE

ANSI/AHRI STANDARD 1250 AND 1251

Bryan Curtis Sartin, Candidate for the Master of Science Degree

University of Missouri-Kansas City, 2012

ABSTRACT

The ANSI/AHRI[1] Standard 1250/1251 Performance Rating of Walk-In Coolers and Freezers strives to standardize refrigeration equipment performance rating. According to the Standard, refrigeration equipment is tested in a laboratory setting using a defined test method. An existing algorithm is used to calculate the Annual Walk-in Energy Factor (AWEF) which serves as a comparison of energy efficiency between equipment. In this work the algorithm was modified in an attempt to more closely approximate actual refrigeration system operation.

To calculate the AWEF, a walk-in box load profile and a ratio of the equipment capacity to the refrigeration load are assumed by the Standard. An extensive literature

---

[1] American National Standard Institute (ANSI)/ Air-Conditioning, Heating, and Refrigeration Institute (AHRI).

review of ninety-eight articles was performed to address these assumptions. Information was categorized and analyzed for each load component, including lighting, occupancy, product, infiltration, conduction, and miscellaneous loads. Additional information was collected on refrigeration system design and operation. A model load profile was developed from which a revised AWEF algorithm was obtained.

Simulations were performed on four walk-in refrigeration units to validate the revised calculation method. Raw results show improved correlation of compressor runtime, because a specific compressor runtime was targeted, reducing variation between hourly simulations and the 1250 calculation from -29.9% with the AHRI 1250 calculation (Becker et al. 2011) to 1.1% with the Proposed 1250 calculation. AWEF correlation between hourly simulations and the 1250 calculation degraded from -7.4% with the AHRI 1250 calculation (Becker et al. 2011) to 15.9% with the Proposed 1250 calculation. Plotting the results for the AWEF and compressor runtime correlation versus variation in the box load, between the hourly simulation and Proposed 1250 calculation results, revealed an issue with the compressor runtime calculation. At a box load variation of zero, the Proposed 1250 AWEF correlation is improved to -2.4%, and the Proposed 1250 compressor runtime correlation is degraded to -14.6%. If a specific compressor runtime had not been targeted, the AWEF correlation for each simulation set would have been improved. To summarize, the Proposed 1250 equations yield an improved AWEF calculation but do not accurately calculate the corresponding compressor runtime.

APPROVAL PAGE

The faculty listed below, appointed by the Dean of the School of Computing and Engineering have examined a thesis titled "Derivation of Walk-in Cooler and Freezer Performance Standard Equations as they Pertain to the ANSI/AHRI Standard 1250 and 1251", presented by Bryan C. Sartin, candidate for the Master of Science degree, and certify that in their opinion it is worthy of acceptance.

Supervisory Committee

Bryan Becker, Ph.D., P.E., F.ASME
Department of Civil and Mechanical Engineering

Ganesh Thiagarajan, Ph.D, P.E.
Department of Civil and Mechanical Engineering

Gregory King, Ph.D.
Department of Civil and Mechanical Engineering

CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

LIST OF ABBREVIATIONS

Φ :	Time related door cycling factor
ξ_H :	Mass fraction of water vapor in the outdoor air (%)
ξ_C :	Mass fraction of water vapor in the indoor air (%)
τ :	Dimensionless time
ρ_H:	Outside (warm) air density (lb/ft^3)
ρ_C:	Inside (cold) air density (lb/ft^3)
ρ_p:	Bulk product density (lb/ft^3)
ρ' :	Ratio of cold air density to hot air density
$\bar{\rho}$:	Average density of the cold interior and warm exterior bodies of air (lb/ft^3)
η_{def} :	Defrost efficiency (%)
θ:	Dimensionless temperature
θ_{amb}:	Ambient dimensionless temperature
θ_{jet}:	Air curtain jet orientation, degrees
β:	Volumetric thermal expansion coefficient (°F^{-1})
μ:	Dynamic viscosity (lb/ft-s)
$\Delta\omega$:	Difference between the specific humidities of air streams (lb vapor/lb dry air)
ω_C:	Humidity ratio of the cold air inside the refrigerated space (lb vapor/lb dry air)
ω_H:	Humidity ratio of the air outside the refrigerated space (lb vapor/lb dry air)
ω_{on}:	Air-on humidity ratio (lb vapor/lb dry air)
ω_{ws}:	Coil surface humidity ratio (lb vapor/lb dry air)
α_1:	13.5, Velocity constant
α_2:	7.67, Velocity constant
a_p:	Product specific surface area (ft^2/ft^3)
A_{floor}:	Floor area of refrigerated space (ft^2)
A_{inf}:	Cross-sectional area of the opening (door) to the refrig. space (ft^2)
A_{TDB}:	The approach of the dry fluid cooler to the dry-bulb temperature (°F)
A_{TWB}:	The approach of the evaporative fluid cooler to the wet-bulb temperature (°F)
A_w:	Liquid water activity coefficient
ACH:	Air changes per hour (1/hr)
AWEF:	Annual Walk-in Energy Factor, Btu/W-hr
b_o:	Jet width at $\bar{x}$ = 0 (ft)
b:	Thickness of air curtain jet (ft)
BF:	Coil bypass factor
BL(T_j):	Heat removed from walk-in box at T_j, not including heat generated by operation of the refrigeration system, W-h

$\dot{BLH}(T_j)$: Non equipment related walk-in box load high at T_j (Btu/h)

$\dot{BLL}(T_j)$: Non-equipment related walk-in box load low at T_j (Btu/h)

$\bar{B}$: Dimensionless width
B: Width (distance from door to back wall) of cold room
c_a: Condenser specific heat capacity (Btu/h-lb-°F)
C_1: 14.4 Btu-min/lb-°F-hr, constant
C_2: Infiltration coefficient dependent of T_{room} ($ft^{1/2}$/s-hr)
C_3: Constant, 1.427 x 10^8 min/ft^3-hr
C_4: Temperature dependent, high load coefficient
C_5: Non-temperature dependent, high load coefficient
C_6: Temperature dependent, low load coefficient
C_7: Non-temperature dependent, low load coefficient
C_8: Percentage of time the refrigeration system operates in high load conditions (%)
C_{ac}: Optimum horizontal air curtain constant
C_{cap}: Compressor capacity correction
C_d: Infiltration correction factor
C_{fan}: Fan factor to account for operation below the minimum condensing temperature (°F)
C_{ir}: Rate of fresh air change per minute (ft^3/min)
C_p: Specific heat of air at constant pressure (Btu/lb-°F)
C_{resp}: 0.658 Btu/grains, Respiration load constant
C_s: Infiltration scaling factor, equal to unity for rectangular openings
C_t: Infiltration traffic coefficient, 1.0919 (ft^3/movement-°$F^{1.76}$)
Cap_{Bal}: Balanced capacity of condensing units and evaporator units that compose the system, Btu
Cap_{comp}: Actual compressor capacity (Btu/hr)
$Cap_{comp,r}$: Rated compressor capacity at the rated conditions (Btu/hr)
Cap_{cond1}: Condenser capacity at point 1
Cap_{cond2}: Condenser capacity at point 2
$Cap_{cond,SST}$: Condensing unit capacity at the design saturated suction temperature (Btu/hr)
Cap_{econd}: Actual evaporative condenser capacity (Btu/hr)
$Cap_{econd,n}$: Nominal evaporative condenser capacity (Btu/hr)
Cap_{evap}: Actual evaporator capacity (Btu/hr)
$Cap_{evap,1TD}$: Evaporator capacity at 1°F T.D. (Btu/hr)
$Cap_{evap,r}$: Rated evaporator capacity at design point (Btu/hr)
CPR: Critical price ratio (%)
CEC: Compressor energy consumption (kW-h)
CFA: Required coil face area (ft^2)
CR: Compressor runtime percentage (%)
D: Diffusivity of the component vapor or gas in the mixture (ft^2/s)
D_{door}: Doorway depth (ft)

D_f: Doorway flow factor
D_t: Doorway open-time factor
$\dot{DF}$: Defrost system power consumption (W)
E: Thermal barrier or infiltration protective device effectiveness (%)
$E_{ref}(T_j)$: Total energy input of refrigeration systems at T_j (W-h)
EER: Energy efficiency ratio (Btu/W-h)
$\dot{EF}_{co}$: Evaporator fan power consumption during compressor off period (W)
$\dot{E}_{ss}(T_j)$: Steady state power consumption at Tj, including power usage of compressor(s), condenser fans, and evaporator fans (W)
f: Respiration coefficient (grains CO_2/lb air-hr-°F^g)
f_d: Frequency of the door cycling (1/s)
F_d: Blockage factor of door
F_{load}: Load factor to compensate for changes in interior store conditions
F_m: Density correction factor
F_{min}: Minimum fraction of design load (0.66 for medium temp, 0.80 for low temp)
F_o: Occupancy percentage (hr/hr)
F_p: Protective device correction factor
F_{pp}: Percentage of product that is vapor-barrier packaged
F_τ: Fractional open time correction factor
F_t: Traffic infiltration correction factor
F_{Tamm}: 0.77, Tamm's equation correction factor applied by Chen (Chen et al. 1999)
F_v: Density correction factor #2
FHR: Fan heat added per ton of net room load (tons/ton)
FHF: Heat of fan power (Btu-h/ft^2)
FP: Fan power (Bhp/ft^2)
Gr: Grashop number
g_{resp}: Respiration exponent
g: Gravity, 32.21 ft/s^2
g_c: Gravitational constant, 32.174 lbm-ft/lbf-s^2
Gr: Grashop number
GRL: Gross room load (tons)
Δh_{41}: Actual enthalpy difference between compressor suction and evaporator inlet (superheated suction – subcooled liquid, Btu/lb)
$\Delta h_{41,r}$: Rated enthalpy difference between compressor suction and evaporator inlet (superheated suction – subcooled liquid, Btu/lb)
Δh_{sub}: Latent heat of sublimation (Btu/lb)
h: Enthalpy (Btu/lb)
h_{1e}: Operating enthalpy of the refrigerant exiting the evaporator (Btu/lb)
$h_{1e,r}$: Rated enthalpy of the refrigerant exiting the evaporator (Btu/lb)
h_{1s}: Enthalpy of the refrigerant suction vapor (Btu/lb)
h_3: Enthalpy of the refrigerant liquid (Btu/lb)

h_{4e}: Operating enthalpy of the refrigerant entering the evaporator (Btu/lb)
$h_{4e,r}$: Rated enthalpy of the refrigerant entering the evaporator (Btu/lb)
h_C: Enthalpy of the air inside (cold) of the cold room (Btu/lb)
$h_{C,mix}$: Enthalpy of the air-vapor mixture inside (cold) of the cold room (Btu/lb mixture)
h_{cor}: Enthalpy of the air in the corridor (Btu/lb)
h_{fg}: Latent heat of evaporation for water at the product temperature (Btu/lb water)
h_g: Enthalpy of water vapor (Btu/lb vapor)
h_H: Enthalpy of the air outside (warm) of the cold room (Btu/lb)
$h_{H,mix}$: Enthalpy of the air-vapor mixture of the warm outside of the refrigerated space (Btu/lb mixture)
h_{mix}: Enthalpy of the air-vapor mixture (Btu/lb mixture)
H: Height of the opening (door) to the refrigerated space (ft)
H_1: Height of the cold room (ft)
HEP: Heat equivalent of fan power, assumed to be 3000 Btu-h/Bhp
HRF: Heat rejection factor as a function of the outside wet bulb and the refrigerant saturated condensing temperature
J: Lag factor
j: Bin number
k: Thermal conductivity of air (Btu-ft/h-ft^2-°F)
k_m: Overall mass transfer coefficient (s/ft)
L_{door}: Door seal length (ft)
LSHR: Load Sensible Heat Ratio
m_{cs}: Slope of condenser capacity vs. sat. suction temperature, Btu/hr-°F
m_{pj}: Amount of product that enters the cold store during the jth day (lb)
m_p: Total stored product weight (lb)
m_{pal}: Amount of product that enters the cold store on one pallet (lb)
m_{tot}: Summation of refrigerant mass flow over the test period (lb)
$\dot{m}_{CO2}$: Mass flow of carbon dioxide (grains CO_2/lb air-hr)
$\dot{m}_{da}$: Dry air mass flow rate of moisture (lb/hr)
$\dot{m}_{\inf}$: Mass flow of the infiltrating air (lb/hr)
$\dot{m}_M$: Modified flow rate of infiltration air to a cold room (lb/min)
$\dot{m}_o$: Critical mass flow rate of moisture (lb/hr)
$\dot{m}_r$: Reverse mass flow rate of moisture (lb/hr)
$\dot{m}_{ss}$: Steady state mass flow of the infiltrating air for a permanently open doorway (lb/hr)
$\dot{m}_{-w}$: Rate of moisture loss (lb water/s-lb product)
$\dot{m}_{wv}$: Rate of mass transfer of a gas or vapor (lb/hr)

$\dot{M}_{ss}$:	Steady dimensionless modified rate of infiltration
$\dot{M}_{M}$:	Dimensionless modified rate of infiltration
n_j:	Bin hours, hrs
n_{pj}:	Amount of product that exits the cold store during the jth day (lb)
N:	Capacity fan speed correlation exponent
N_d:	Box load high to design refrigeration capacity ratio
N_{do}:	Number of door openings in t_{tot}
N_{evap}:	Number of evaporator units
N_f:	Frequency non-dimensional parameter
N_o:	Physical non-dimensional parameter
N_t:	Forklift traffic frequency (movements/hr)
NRL:	Net room load (tons)
OR:	Opening ratio
p_{sat}:	Saturated pressure (psi)
$P_{avg,day}$:	The average daytime cost of electricity ($/kWh)
$P_{avg,shift}$:	The average off-peak cost of electricity during which load shifting would be performed ($/kWh)
Pr:	Prandtl number
q':	Specific load (tons per ft door width)
q''_{floor}:	Heat flux through the floor (Btu/hr-ft^2)-
$\dot{q}_{et}$:	Heat loss due to evaporation process (Btu/hr-lb product)
$\dot{q}_{p}$:	Occupancy load per person (Btu/hr-person)
$\dot{q}_{resp}$:	Specific load due to respiration (Btu/hr-lb product)
Q_{10}:	Ratio of the respiration rate at one temperature to a rate at a 18°F higher temperature
Q_{Inf}:	Infiltration load (Btu)
$Q_{Inf/24}$:	Infiltration load over 24 hr period (Btu/24 hr)
Q_{tot}:	Total load (Btu)
$\dot{Q}_{DF}$:	Defrost power consumption contributed to the box load (Btu/hr)
$\dot{Q}_{def,in}$:	The actual energy required to melt the frost off the coil (Btu/hr)
$\dot{Q}_{Inf}$:	Infiltration load into the refrigerated space (Btu/h)
$\dot{Q}_{Inf-D}$:	Infiltration load through doors (Btu/h)
$\dot{Q}_{Inf-W}$:	Infiltration load through walls (Btu/h)
$\dot{Q}_{L}$:	Latent Heat Flow (Btu/hr)

$\dot{Q}_{melt}$: Theoretical energy required to melt the frost (Btu/hr)

$\dot{Q}_{resp}$: Product respiration heat load (Btu/hr)

$\dot{Q}_{s}$: Sensible Heat Flow (Btu/hr)

$\dot{Q}_{ss}(T)$: Steady state refrigeration capacity as a function of temperature to which the condenser is rejecting heat (Btu/hr)

$\dot{Q}_{tot}$: Average refrigeration load over a period t_{tot} (Btu/hr)

R_{T1}: Respiration rate at a temperature T_1 (Btu/hr-ton)

$R_{T1+10°C}$: Respiration rate at a temperature 18°F higher than T_1 (Btu/hr-ton)

RER: Heat transferred from the air to the refrigerant, Btu-h/ft^2

RH: Relative humidity (%)

RT_{Bal}: Balanced compressor runtime (%)

RPM_a: Actual fan speed (rpm)

RPM_r: Rated fan speed (rpm)

s: Ratio of warm air density to cold air density

S: Seven-eighths cooling time (hr)

Sh: Sherwood number

Sc: Schmidt number

SHR: Sensible heat ratio

SST_1: Saturated suction temperature at point 1 (°F)

SST_2: Saturated suction temperature at point 2 (°F)

t: Time (s)

t_1: Time (days)

$t_{1/2}$: Half cooling time of the product in the given conditions (days)

t_a: Time to open door (s)

t_b: Time the door was fully opened (s)

t_c: Time for the door to close (s)

t_d: Period of the cycle between door openings (s)

t_{oc}: Opening time of the door for one pallet passage (s)

t_{open}: Time that the door is open, compensated for opening and closing periods (s)

t_{dt}: The total time during which defrost was occurring (s)

t_{tot}: Total time period (s)

ΔT: Temperature difference between the interior box and exterior temperatures (°F)

ΔT_{ref}: Reference temperature difference between the interior box and exterior temperatures (°F)

T_1: Ambient temperature, condition 1 (°F)

T_2: Ambient temperature, condition 2 (°F)

T_{ain}: Coil entering air temperature (°F)

T_{amb}: Ambient outside dry-bulb temperature (°F)

T_{aout}: Coil leaving air temperature (°F)

T_C: Temperature of the cold air inside the refrigerated space (°F)
T_{con}: The minimum condensing temperature (°F)
T_{cond}: Condensing temperature (°F)
T_{cor}: Corridor (entry way to refrigerated space) temperature (°F)
T_{cs}: Coil surface temperature (°F)
T_{csc}: Coil surface critical surface temperature (°F)
T_d: Refrigeration system design ambient temperature (°F)
T_{evap}: Evaporator temperature (°F)
T_H: Temperature of the warm air outside the refrigerated space (°F)
T_j: Ambient temperature for bin j (°F)
T_{on}: Air-on condition temperature (°F)
T_{off}: Air-off condition temperature (°F)
T_{pc}: Temperature at center of cooling object (°F)
T_{pini}: Initial temperature of the product (°F)
$T_{\bar{p}}$: Mass average product temperature (°F)
T_{ref}: Reference temperature (°F)
T_{sll}: Suction line loss temperature (°F)
TD_{DB}: Temperature difference between the condensing temperature and the ambient dry-bulb temperature (°F)
$TD_{Eapprox.}$: Approximation of the difference between the interior box and saturated suction temperatures (°F)
TD_{Ebal}: Balanced or actual operating temperature difference between the interior box and saturated suction temperatures (°F)
TD_{Edes}: Temperature difference between the interior box and saturated suction temperatures at evaporator design point (°F)
TD_W: Temperature difference between the condensing temperature and the water temperature (°F)
TD_{WB}: Temperature difference between the condensing temperature and the wet-bulb temperature (°F)

u: Mean velocity in $\bar{x}$ direction (ft/s)
u_C: Velocity profile for cold air leaving the refrigerated space (ft/s)
u_{FV}: Coil face velocity (ft/s)
u_H: Velocity profile for warm air entering the refrigerated space (ft/s)
u_{mf}: Maximum velocity of the velocity profile during infiltration through a doorway (ft/s)
u_o: Mean velocity at nozzle outlet (ft/s)
u_{op}: Optimum horizontal air curtain jet exit velocity (ft/s)
u_{ref}: Elsayed reference velocity in x direction (ft/min)
U: Dimensionless velocity component in x direction
UA: Sensible Capacity Rating (Btu/hr-°F)
$\dot{V}_{AT}$: Air infiltration rate when the doors are closed, air tightness, crack infiltration (ft^3/min)

$\dot{V}_{AT}/L$: Air infiltration rate when the doors are closed per length of crack, air tightness (ft^3/min-ft)

$\dot{V}_{\inf}$: The general infiltration rate compensating for door and crack infiltration (ft^3/min)

$\dot{V}_{nt}$: The average infiltration rate without traffic passing through the doorway (ft^3/min)

$\dot{V}_{p}$: The infiltration rate with the protection device operating on the doorway (ft^3/min)

$\dot{V}_{tr}$: The infiltration rate with traffic passing through at a defined frequency (ft^3/min)

$\dot{V}_{u}$: The infiltration rate with the unprotected doorway (ft^3/min)

V_t: Infiltration per forklift movement, including entry and exit (ft^3/movement)

V: Volume of the refrigerated space (ft^3)

υ_{ain}: Design specific volume of air at coil entrance air temperature (ft^3/lb)

υ_{ainr}: Operational specific volume of air at coil entrance air temp. (ft^3/lb)

υ_{aout}: Specific volume of air at coil exit air temperature (ft^3/lb)

υ_{H}: Specific volume of warm air outside of the refrigerated space (ft^3/lb)

υ_{C}: Specific volume of cold air inside of the refrigerated space (ft^3/lb)

υ_{comp}: Specific volume of inlet gas based on the actual conditions (ft^3/lb)

υ_{compr}: Specific volume of inlet gas based on manufacturer's rated conditions (ft^3/lb)

υ_{Cr}: 10.95, Coil rated specific volume of air inside refrigerated space

υ_{Er}: Specific volume of refrigerant in the evaporator at rated conditions (ft^3/lb)

υ_{E}: Specific volume of refrigerant in the evaporator (ft^3/lb)

υ_{g}: Specific volume of water vapor (ft^3/lb water vapor)

ν: Kinematic viscosity (ft/s^2)

W: Width of the door (ft)

w_{gap}: Gap width (in)

$\bar{x}$: Coordinate position along airflow path, parallel to air flow (ft)

x_C: Height above floor of velocity U_C (ft)

x_H: Height above floor of velocity U_H (ft)

X: Dimensionless distance along the x direction

Y: Dimensionless distance along the y direction

$\bar{y}$: Coordinate position along airflow path, perpendicular to air flow (ft)

z: Door height to density ratio relationship (ft)

ACKNOWLEDGEMENTS

First, I will like thank Dr. Bryan Becker for his guidance and support over the past three years. From directing me to my topic of research to assistance with analysis of results, Dr. Becker has been key to this work. In addition, I would like to express my gratitude to the other members of my thesis committee, including Dr. Ganesh Thiagarajan and Dr. Gregory King. The time that these professors took to assist with the development of this work is a testament of the dedication of the Civil and Mechanical Engineering Department at the University of Missouri – Kansas City to the development of their students. Finally, I would to thank my father, John Sartin, for spending the time to proof this work.

DEDICATION

I would like to dedicate this work to my family. Mandy, thank you for your intermittent patience. Your willingness to provide me time to work on my research and abrupt reversals to frustration with the amount of time that it was taking showed both your faith in my development and longing for our time together. Matthew, you have grown up throughout my Master's research from six months to now three and a half years old. You are turning into quite the little man, and I am looking forward to rough-housing with you a lot more in the coming years. Avery, your cheerful personality is what keeps me going. Just like your grandmother Jean and your mother, you project positive energy to all those around you, please never change.

CHAPTER 1

INTRODUCTION

Walk-in Refrigeration

Walk-in coolers and freezers are defined as refrigerated spaces operating above freezing or below freezing, respectively, and ranging in size from 50 to 3000 ft^2, with ceiling heights ranging from 8 to 30 ft (CEC 2007; CEC 2008; Congress 2007). Walk-ins are used in numerous settings, including offices, retail, grocery stores, schools, hotels, hospitals, and warehouses (Patel et al. 1993), with the majority of applications classified as commercial use.

The commercial sector uses 5.6 Quads (1 Quad = 10^{15} Btu/yr) of energy in the United States, accounting for 15.8% of the total useful energy utilized by the nation (DOE 2004; Goetzler et al. 2009; Westphalen et al. 1996). Commercial refrigeration uses 4.1 to 7% of the commercial sector's energy, equaling 0.23 to 0.35 Quads of energy consumption (Goetzler et al. 2009; Westphalen et al. 1996).

Commercial refrigeration usage may be broken down further to analyze what contribution walk-ins have on energy consumption. Supermarket refrigeration, which contributes to 47 to 56% of a supermarket's total energy use (Arias 2005; Arias and Lundqvist 2006; Christensen and Bertilsen 2004; Fricke and Becker 2011; Goetzler et al. 2009; Huan 2008; Sugiartha et al. 2009; Walker 2001), accounts for the greatest portion of commercial refrigeration at 32.9%. The ratio of supermarket walk-in energy use to total supermarket refrigeration energy use is cited by researchers as ranging from 12.3 to 30.2%, with an average value of 24.3% (Goetzler et al. 2009; Henderson and Khattar 1999; Walker 2001; Walker et al. 1990). Walk-ins used in applications other than supermarkets account for 18.2% of commercial refrigeration energy use (Westphalen et al. 1996). Considering the above information, walk-in coolers and freezers consume 26.2% of commercial refrigeration energy use, 1.5% of commercial building energy use, and 0.2% of the nation's energy use. This is equivalent to 76 trillion Btu/yr of energy use or 242 trillion Btu/yr of primary energy use. Primary energy use is calculated based on a heat rate of 10,867 Btu/kWh to take into account losses in generation and transmission of electricity (Patel et al. 1993).

The refrigeration industry is focused on reducing energy use from both an environmental impact and operating cost perspective. Opportunities for reduction in energy use have been identified by numerous researchers, ranging from 10 to 49.4% of the current consumption. Methods of reducing energy use by refrigerated spaces, discussed in the previously noted research, include application of current energy efficient equipment and construction, technologies used by other industries but not yet applied to walk-in refrigeration, and potential future research developments (Goetzler et al. 2009; Huan 2008; Patel et al. 1993; SCE 2008; Walker 2001; Westphalen et al. 1996).

The government in California began imposing construction standards for walk-ins to drive energy efficient designs in 2004 with the California Code of Regulations Title 20. Federal regulations followed, including the Energy Independence and Security Act (EISA) of 2007 (Congress 2007). The most recent federal regulatory information can be found in the Final Rule of the Energy Conservation Program's Test Procedures for Walk-In Coolers and Freezers (DOE 2011). To ensure that governmental regulations were attainable, the Air-Conditioning, Heating, and Refrigeration Institute (AHRI) developed standards for performance rating of walk-in refrigeration equipment (AHRI 2009b; AHRI 2009c).

ANSI/AHRI 1250/1251 Standard

The AHRI Standard for Performance Rating of Walk-In Coolers and Freezers (AHRI Standard 1250/1251) was published in 2009 (AHRI 2009b; AHRI 2009c). As implied by their titles, this pair of documents was put in place to standardize the rating of the mechanical refrigeration equipment used in walk-in coolers and freezers. Manufacturers are to test their equipment (walk-in systems, condensing units, and unit coolers) in a specific manner as defined by the Standard and then use an algorithm to calculate the Annual Walk-in Energy Factor (AWEF).

To determine the AWEF value for a given piece of equipment, the unit should first be tested according to the method of test defined by the AHRI Standard (AHRI 2009b; AHRI 2009c). Indoor condensing units are to be tested at a rating temperature of 90ºF and outdoor condensers are to be tested at temperatures of 95, 59, and 35ºF. AHRI defines the AWEF as the ratio of the annual refrigeration load less the heat added to the space by the refrigeration equipment, divided by the annual refrigeration system energy usage (Equation 1). With the components of Equation 1 listed in Equations 2 and 3.

$$AWEF = \frac{\sum_{j=1}^{n} BL(T_j)}{\sum_{j=1}^{n} E_{ref}(T_j)} \tag{1}$$

$$E_{ref}(T_j) = \left\{ \begin{array}{l} 0.33 \cdot \left(\dot{E}_{ss}(T_j) - \dot{EF}_{co} \right) \left(\dfrac{\dot{BLH}(T_j) + 3.412 \cdot \dot{EF}_{co} + \dot{Q}_{DF}}{\dot{Q}_{ss}(T_j) + 3.412 \cdot \dot{EF}_{co}} \right) + 0.67 \cdot \\ \left(\dot{E}_{ss}(T_j) - \dot{EF}_{co} \right) \left(\dfrac{\dot{BLL}(T_j) + 3.412 \cdot \dot{EF}_{co} + \dot{Q}_{DF}}{\dot{Q}_{ss}(T_j) + 3.412 \cdot \dot{EF}_{co}} \right) + \dot{EF}_{co} + \dot{DF} \end{array} \right\} \cdot n_j \tag{2}$$

$$BL(T_j) = \left[0.33 \cdot \dot{BLH}(T_j) + 0.67 \cdot \dot{BLL}(T_j)\right] \cdot n_j \qquad (3)$$

The Standard assumes that the refrigeration load profile for walk-ins may be approximated by a period of elevated refrigeration load (high load period) and a period of reduced refrigeration load (low load period). The low load period comprises 16 hours of the day whereas the high load period comprises the remaining 8 hours. The box load levels are designated by AHRI as a function of the refrigeration system capacity at the design conditions. The generalized equations for the box load variables are presented in Equations 4, 5, 6, and 7. The box loads do not include heat added to the space by the refrigeration equipment, including defrost and evaporator fan operation.

Walk-in Cooler Box Loads

$$\dot{BLH}(T_j) = 0.65 \cdot \dot{Q}_{ss}(T_d) + 0.05 \frac{\dot{Q}_{ss}(T_d)(T_j - 35)}{60} \qquad (4)$$

$$\dot{BLL}(T_j) = 0.03 \cdot \dot{Q}_{ss}(T_d) + 0.07 \frac{\dot{Q}_{ss}(T_d)(T_j - 35)}{60} \qquad (5)$$

Walk-in Freezer Box Loads

$$\dot{BLH}(T_j) = 0.55 \cdot \dot{Q}_{ss}(T_d) + 0.25 \frac{\dot{Q}_{ss}(T_d)(T_j + 10)}{105} \qquad (6)$$

$$\dot{BLL}(T_j) = 0.15 \cdot \dot{Q}_{ss}(T_d) + 0.25 \frac{\dot{Q}_{ss}(T_d)(T_j + 10)}{105} \qquad (7)$$

The box load equations break the load into two components, a load driven by the ambient temperature and a consistent load. AHRI developed a model load profile using the

AHRI Load Spreadsheet (AHRI 2009a) to derive the coefficients associated with the box load equations.

An initial investigation by Becker et al. (Becker et al. 2011) of the AHRI Standard 1250/1251 revealed potential issues with the model load that AHRI used to develop the box load equations and variations between the AHRI Standard 1250/1251 performance calculations and those developed through hourly simulations.

This work proposes some modifications to the equations that define the AHRI Standard 1250/1251 (AHRI 2009b; AHRI 2009c). This investigation included the following tasks:

- Review of Current Work
- Literature Review and Industry Investigation
- Development of Proposed Model Load Profile
- Development of Proposed AHRI Standard 1250/1251 Equations
- *eQuest* Simulations
- Proposed 1250 Calculation Method Discussion
- Conclusions

CHAPTER 2

REVIEW OF CURRENT WORK

To begin, the current status of information directly related to the AHRI Standard 1250/1251 (AHRI 2009b; AHRI 2009c) was reviewed. This review of information included the Standard itself, AHRI Load Spreadsheet (AHRI 2009a), and the ARTI Project 9002 Final Report (Becker et al. 2011).

The AHRI Standards 1250/1251 define a performance rating method to be utilized by manufacturers, designers, installers, contractors and end-users. The standard defines method of tests for fixed capacity matched systems, dual capacity matched systems, and variable capacity matched systems with condensers located indoors or outdoors, and is applicable to coolers and freezers. Component methods of tests are defined for fixed capacity condensing units (indoor and outdoor condensers) and evaporators, applicable to coolers and freezers. Information collected from these performance tests is applied to a set of equations to determine the Annual Walk-in Energy Factor (AWEF) for a given set of refrigeration equipment.

The AHRI Load Spreadsheet (AHRI 2009a) uses an array of equations to determine the box refrigeration load. The box conduction load is calculated using a traditional, simplified method which analyzes only the conduction heat transfer, and ignores the effects of transferring that heat to the space via convection. By using a square walk-in footprint, only three different surfaces are considered, including the ceiling, floor, and walls. Similar to the AHRI 1250/1251, this spreadsheet only analyzes walk-in boxes situated inside a temperature controlled building.

The infiltration calculation was done by the AHRI Load Spreadsheet using some simplified equations to define the sensible and latent portions of the load (Equations 8 and 9) with the Gosney Olama equation determining the flow rate through the unprotected door (Equation 12).

$$\dot{Q}_s = 1.1 \cdot \dot{V}_{\text{inf}} \cdot \left(T_H - T_C\right) \tag{8}$$

$$\dot{Q}_L = 5500 \cdot \dot{V}_{\text{inf}} \cdot \left(\omega_H - \omega_C\right) \tag{9}$$

$$\dot{V}_{\text{inf}} = \left(1 - E\right)\left(1 - F_d\right)\dot{V}_u + F_d \, \dot{V}_{AT} \tag{10}$$

$$\dot{V}_{AT} = \frac{\dot{V}_{\text{inf}} \, w_{gap} L_{door}}{A_{\text{inf}} \cdot 12 \, \frac{in}{ft}} \tag{11}$$

$$\dot{V}_u = 0.221 \cdot A_{\text{inf}} \left(1 - \frac{\rho_H}{\rho_C}\right)^{\frac{1}{2}} \left(gH\right)^{\frac{1}{2}} F_m \cdot \frac{60s}{\min} \tag{12}$$

Product loading into the refrigerated space is done at one point in time for all walk-ins as defined by the AHRI Load Spreadsheet. The refrigeration load associated with this product introduction to the space is assumed to affect the space in a linear fashion, pulled down over an eight hour period. The calculation assumes the load is solely sensible for coolers and freezers. Therefore the product must be packaged so that water vapor may not pass through the barrier.

Miscellaneous loads include lighting, occupancy, and vehicle operation. Lighting load is calculated based on Wattage per space footprint and a lighting schedule. The lighting load is reduced by a certain percentage to apply only the light power converted to heat as a load on the space. Occupancy load is defined by Equation 13 and vehicle load is calculated based on the vehicle power usage and operating schedule.

$$\dot{q}_p = F_o \cdot (-11.5 \cdot T_C + 1295) \tag{13}$$

From the model load profiles of large and small walk-in coolers and freezers, the AHRI Standard 1250/1251 box load equations are developed. The small and large cooler box load equations are averaged, as are the equations for the walk-in freezers, to develop a typical walk-in cooler and freezer box load equation set. These equations were applied to the development of the AHRI Standard 1250/1251.

The AHRI Load Spreadsheet contains calculations using both SI and IP units. The only difference in these calculations is the box interior temperature. As defined in the standard, walk-in freezers operate at -23ºC (AHRI 2009c) or -10ºF (AHRI 2009b), and walk-in coolers operate at 2ºC (AHRI 2009c) or 35ºF (AHRI 2009b). These values do not directly equate from Celsius to Fahrenheit, resulting in solution variations of up to 17.7%. The standard box load equations were developed using the SI unit calculation method for both the AHRI Standard 1250 and 1251.

The ARTI Project 9002 Final Report performed a model load investigation similar to that done in this report, analysis of monitored field sites, simulations of walk-in cooler and freezer performance, and analysis of the AHRI Standard 1250/1251 load equations. The model load investigation revealed a few differences between the AHRI model load profile and the literature, including the large walk-in box door size and number, omission of crack infiltration, product loading for the small walk-in box, walk-in freezer relative humidity, and walk-in freezer R-value. Total door area was analyzed as a function of the space floor area, with AHRI large walk-in door areas 1.5% of the refrigerated space floor area. According to Becker et al. (Becker et al. 2011) the literature review revealed an average relationship between the floor area and door area of 9.3%. Infiltration was analyzed by the AHRI Load Spreadsheet (AHRI 2009a) only as a function of door openings. In reality there is constant infiltration to the space through cracks around doors and through any cracks left in between insulated panels during the installation of the walk-in. Product loading for the small cooler was overstated at approximately 3 times the value noted by the Department of Energy (DOE 2010b) for a similar walk-in cooler size. Walk-in cooler relative humidity was specified at 90% by AHRI, but the literature revealed a slightly lower value of 73.6%. The AHRI Load Spreadsheet freezer uses 32 h-ft^2-˚F/Btu construction. When Becker et al. compared this to insulation details reported by other researchers, an average value of 26 h-ft^2-˚F/Btu was found.

The AHRI 1250/1251 Standard specifies an indoor condensing unit ambient temperature of 90ºF. Becker et al. (Becker et al. 2011) recommends a value of 80ºF to more closely approximate the conditions that indoor condensing units operate under.

Comparison of monitored field data to parameters defined in the AHRI Standard 1250/1251 shows that 'low loads' correlate. The 'high load' defined by AHRI 1250/1251 appears to be overstated by 32.5 to 84.7% when compared to the field data. The time spent at 'high load' and 'low load' operation does not agree either. AHRI 1250/1251 assumes 2/3rds of time at 'low load' and 1/3rd of the time at 'high load' for both walk-in coolers and freezers, but field data suggests that the load profile is more evenly split between high and low load conditions. AHRI 1250/1251 assumes 33.2% more time in low load than monitored data for coolers and 23.6% more time in low load than monitored data for freezers.

Additional comparisons were made to the refrigeration load per square foot of monitored field walk-in coolers and freezers. The AHRI 1250/1251 small cooler area-specific refrigeration load correlated well with field data at 27.6% larger. The AHRI 1250/1251 large cooler refrigeration load is less than half that depicted by measured data, however. AHRI 1250/1251 freezer loads were 19.9 and 10.7% of the measured data for the small and large freezers, respectively.

Following eQuest simulation of typical walk-in coolers and freezers the Annual Walk-in Energy Factor (AWEF) was calculated for seven climate zones with the condenser indoors and outdoors. In addition, AWEFs were calculated according to the test method specified in AHRI 1250/1251. The freezer runtimes seem appropriate, with the test method closely matching the values obtained through simulation of the refrigeration system's operation in an actual climate zone. The AHRI 1250/1251 small freezer compressor runtime, as compared to the climate zone simulations, was 3.3% lower, 15.7% lower, and 6.4% lower for condenser indoor; box in, condenser outdoor; and box out, condenser outdoor conditions, respectively. Large freezer runtime comparisons were similar, at 5.0% lower, 30.3% lower, and 26.3% lower, for the condenser indoor; box in, condenser outdoor; and box out, condenser outdoor conditions, respectively. Walk-in cooler runtime comparisons were more drastic. The AHRI 1250/1251 small cooler runtime, as compared to the climate zone simulations, was 45.8% lower, 52.1% lower, and 44.2% lower for the condenser indoor; box in, condenser outdoor; and box out, condenser outdoor conditions, respectively. The large cooler runtime was determined to be 42.1% lower, 51.0% lower, and 47.7% lower for the condenser indoor; box in, condenser outdoor; and box out, condenser outdoor conditions, respectively.

CHAPTER 3

LITERATURE REVIEW AND INDUSTRY INVESTIGATION

An extensive literature review was performed to analyze walk-in cooler and freezer load information available. Information was gathered from numerous sources, including published literature, presentations, and discussions with industry professionals. While this investigation was focused on defining a model load profile for walk-in coolers and freezers in commercial refrigeration applications, the study was not confined to that type equipment. A walk-in load profile is composed of numerous items, including infiltration, conduction, product load, occupancy load, lighting load, and miscellaneous loads. Studies done on these subjects in other applications of refrigeration can be carefully applied to walk-in coolers and freezers.

The primary assumption of this investigation is that the occurrence of specific model load characteristics in literature correlate to the occurrence of these same model load characteristics in industry. Therefore, an analysis of information presented in the literature will appropriately represent the actual application of walk-in refrigeration systems in the field.

Literature is categorized by industry and load type. Industries include industrial, commercial, and general refrigeration. Industrial refrigeration, as defined by this work, includes refrigeration systems supporting the production and/or storage of bulk products. These facilities would typically include warehouses, truck loading docks, pull-down rooms, and ripening rooms. Commercial refrigeration refers to refrigeration of spaces in supermarkets, restaurants, convenience stores, and other miscellaneous small-scale applications. Typically, units in a commercial application have a much higher turnover of products than their industrial counterparts. General refrigeration refers to information that applies to both industrial and commercial refrigeration.

Review of General Refrigeration

Refrigeration System Design

(Huan 2008)

Huan (Huan 2008) investigates energy reduction opportunities for industrial, commercial, and residential refrigeration applications. In the food industry alone, 50% of energy used is applied to refrigerating perishable products. A significant number of opportunities for improvement, including improving product pull-down, reducing infiltration, and properly sizing equipment, are noted.

Production and refrigerated warehouses are typically operated at about -13°F to minimize food quality loss. Refrigerated distribution centers operate from -4 to 0°F. The energy use for a range of refrigerated stores is presented in Table 1.

Table 1. Energy Consumption for Different Sizes of Cold Stores

Room size (ft^3)	Energy Consumption (kWh/ft^3-yr)
353,147	2.83
35,315	5.66
3,532	16.99
353	42.48

Source: (Huan 2008)

(Bansal and Jain 2007)

Cascade systems are used for numerous low-temperature refrigeration applications, including blast freezing, cold storages, and liquefaction of gases at temperatures ranging from -148 to -22°F. A temperature difference ranging from 4.5 to 9°F is commonly used in the cascade condenser to balance condenser first cost and system energy consumption. The high temperature circuit uses a high boiling point refrigerant such as R-22, R-124a, R-507, ammonia, propane, propylene, or hydrocarbons whereas the low temperature side uses CO_2, HFC-23, or R-508B.

While the use of these systems is wide spread, their application requires careful consideration, as available tools to optimize the system are limited. In the comparison of low temperature refrigerants, CO_2 is a clear winner for suction temperatures above -58°F due to its low cost and thermal properties. Bansal and Jain (Bansal and Jain 2007) found ammonia to be the most effective refrigerant for the high pressure side of the cascade system. The system attained a COP of 1.52 when configured with a low mass flow ratio of 0.31, condenser temperature of 86°F, low-side evaporator temperature of -49°F, and cascade heat exchanger approach of 9°F.

(SCE 2009)

It is difficult to operate a refrigeration system efficiently when there are only mechanical controls in place. Configuring sensors and controls to analyze how the system is operating gives the opportunity to optimize the system. This article analyzes the benefits of an internet based monitoring system (SCE 2009). A Southern California Edison (SCE) customer site was chosen to test this monitoring and control system. The facility was a large dry goods distribution warehouse (21,000 ft^2) with numerous zones cooled to a medium temperature. Attached to the facility are small independent walk-in coolers and freezer.

One cooler inside the warehouse is 5,820 ft^2 and contains cheese and other dairy products. It is composed of four separate rooms, 9 compressors, and 17 evaporators interconnected by doors for forklift travel. A second cooler (36°F) is used to pre-sort items prior to loading trucks at the shipping dock. This space also serves as temporary storage after unloading a truck.

The independent walk-ins include one freezer and two coolers. The walk-in freezer, with a setpoint temperature of 0°F (204 ft^2), is located between two walk-in coolers (96 and 300 ft^2). All of these units utilize glass merchandising doors. The walk-in freezer utilizes electric defrost. Prior to monitoring/control defrost occurred four times per day.

Some efficiency measures implemented as a result of this analysis include a significant reduction in defrost, installation of destratification fans in the large cooler, ECM evaporator fan motors, and anti-sweat heater controls. In the end, compressor runtimes were reduced by 26.7% (Table 2) and evaporator fan runtime was reduced by 57.4%. Simultaneous with these energy savings, temperature control improved with the average space temperature lowered 0.9°F.

Table 2. Compressor Runtime Improvements due to Monitoring/Control System

	Compressor Run-Time (%)		
	Prior to Control	With Control	Difference
Retail Cooler A	27.1	18.5	-31.7
Retail Freezer	52.2	34.5	-33.9
Retail Cooler B	35.9	22.7	-36.8
Cooler 1 Z1	31.7	29.0	-8.5
Cooler 1 Z2	37.7	29.4	-22.0
Cooler 2 Z1	38.2	15.2	-60.2
Cooler 2 Z2	19.7	18.0	-8.6
Cooler 3 Z1	0.7	21.4	--
Cooler 3 Z2	16.0	14.9	-6.9
Cooler 3 Z3	39.0	20.7	-46.9
Cooler 4 Z1	18.9	18.3	-3.2
Cooler 4 Z2	20.6	13.4	-35.0
Averages	28.1	21.3	-26.7

Source: (SCE 2009)

General Load Details

(Sherif et al. 2002)

The overall goal of this work is to improve the efficiency of cold stores, with a focus on the controlling coil frosting. By controlling the box conditions relative to the coil surface temperature, the speed and thermal resistance of frost that forms may be confined. As a follow-up to previous work (Sherif et al. 2001), Sherif performed testing related to the application of coils in industry and developed a model for frost formation on coils.

The experimental refrigeration system used a top feed, hot-gas defrosting method, R-22 refrigerant, and refrigeration components with a capacity of 2.6 tons at -40°F saturated suction temperature and 95°F ambient conditions. The freezer is 155” x 124” x 87” tall, has three 7.25” thick walls composed of 2 x 4 wood studs on 2 ft centers distributed within 3.5” molded beads polystyrene insulation with an R-value of 25.64 h-ft^2-

°F/Btu, a front wall composed of one layer of insulation for an R-value of 13.33 h-ft^2-°F/Btu, and an R-value of 35.71 h-ft^2-°F/Btu for the ceiling. The freezer has two 30 7/8” x 66.5” tall doors with good rubber seals, constructed in a similar manner to the front wall but with an additional ¾” layer of plywood beneath the outside aluminum cladding for an R-value of 14.29 h-ft^2-°F/Btu. Each door has one 13” x 13” insulated, double glazed, ½” air space window with an R-value of 1.96 h-ft^2-°F/Btu. The floor is 16 gage galvanized sheet metal on a 6 millimeter layer of visquene with 4” of molded bead polystyrene beneath, and ½” of extruded polystyrene below that. The entire box is on an insulated basement floor 1 ft below grade.

Each door uses 16 ft of heating cable at 3 Watts per foot at 40°F. Two light fixtures are in the space, each using two 60 Watt fluorescent bulbs.

Heat pipes installed upstream and downstream of a dehumidification coil reduce the sensible cooling load on the dehumidification coil and then reheat the exiting air using no external energy. During defrost, damper door positioning was tested to determine the effects on defrost efficiency. Partial dampering was coincident with an 18% improvement in defrost efficiency whereas full dampering increased defrost efficiency by 43%.

Coil operation under supersaturated box conditions produced very dense, snow-like frost which grew quickly but was slow to remove during the defrost cycle. As may be expected, sub-saturated operation had very little frost formation. Frost formed at higher face velocities (greater than 700 fpm) is more likely to be transferred to the drain pan during defrost, resulting in a higher defrost efficiency.

(PG&E 2007)

California Title 24 dictates building energy efficiency standards, including standards for refrigerated warehouses (250,000 ft^2 or more). Pacific Gas and Electric Company (PG&E) discusses changes to Title 24 refrigerated warehouse standards such as insulation requirements, fan and compressor controls, and interior lighting levels (PG&E 2007). Benefits from following Title 24 are promised at 0.3 kWh/ft^2 for insulation improvements, 6.2 kWh/ft^2 for evaporator fan controls, 1.4 kWh/ft^2 with use of evaporative condensers, and 4.1 kWh/ft^2 for compressor controls. These cost savings were determined through simulation of two prototypical warehouses, defined in Table 3 and Table 4. Walk-in refrigerated spaces (3,000 ft^2 or less) are defined by Title 20, and some of those details are also discussed in this report.

Table 3. Prototypical Large Refrigerated Warehouse Model Description

Model Parameter	Value
Shape	Rectangular, 400 ft x 230 ft
Floor area	Freezer: 40,000 ft^2, cooler: 40,000 ft^2, dock: 12,000 ft^2
Number of floors	1
Floor to ceiling height	30 ft
Exterior wall construction	Insulated metal panel
Ext. wall R-value	Cooler and dock: R-20, freezer: R-26
Infiltration rate	Cooler and freezer: 0.1 ACH, dock: 0.3 ACH
Roof construction	Insulated low mass roof
Roof R-value	Cooler and dock: R-23, freezer: R-46
Roof absorptivity	0.80
Lighting power density	0.6 W/ft^2
Equipment power density	0.7 W/ft^2 (forklifts and plug loads)
Operating schedule	24/7
No. people	184 max
Evaporator type	Constant volume, continuous fan operation
Evaporator size (CZ13)	Cooler: 392 ft^2/ton, freezer: 295 ft^2/ton, dock: 218 ft^2/ton
Evaporator cfm (CZ13)	Cooler: 4.3 cfm/ft^2, freezer: 4.79 cfm/ft^2, dock: 7.9 cfm/ft^2
Compressor type	Ammonia screw comp. w/slide valve capacity control
Compressor configuration	Parallel equal, 3 compressors per suction group
Suction groups	Low temperature: -20°F, medium temperature: 30°F
Room temperature	Cooler: 40°F, freezer: -10°F, dock: 40°F
Evaporator fan power	0.15 W/cfm
Condenser type	Evaporative condenser
Min condensing temp	85°F
Condenser fan and pump	330 Btu/hr-W
Cond. design approach	23°F (CZ13, design wet bulb = 73°F)

Source: (PG&E 2007)

Table 4. Prototypical Small Refrigerated Warehouse Model Description

Model Parameter	Value
Shape	Rectangular, 200 ft x 130 ft
Floor area	Freezer: 10,000 ft^2, cooler: 10,000 ft^2, dock: 6,000 ft^2
Number of floors	1
Floor to ceiling height	30 ft
Exterior wall construction	Insulated metal panel
Ext. wall R-value	Cooler and dock: R-20, freezer: R-26
Infiltration rate	Cooler and freezer: 0.1 ACH, dock: 0.3 ACH
Roof construction	Insulated low mass roof
Roof R-value	Cooler and dock: R-23, freezer: R-46
Roof absorptivity	0.80
Lighting power density	0.6 W/ft^2
Equipment power density	0.7 W/ft^2 (forklifts and plug loads)
Operating schedule	24/7
No. people	52 max
Evaporator type	Constant volume, continuous fan operation
Evaporator size (CZ13)	Cooler: 550 ft^2/ton, freezer: 380 ft^2/ton, dock: 360 ft^2/ton
Evaporator cfm (CZ13)	Cooler: 3 cfm/ft^2, freezer: 3.5 cfm/ft^2, dock: 4.7 cfm/ft^2
Compressor type	HFC reciprocating compressor with unloader
Compressor configuration	Parallel equal, 2 compressors per suction group
Suction groups	Low temperature: -20°F, medium temperature: 30°F
Room temperature	Cooler: 40°F, freezer: -10°F, dock: 40°F
Evaporator fan power	0.15 W/cfm
Condenser type	Air-cooled condenser
Min condensing temp	85°F
Condenser fan and pump	53 Btu/hr-W
Cond. design approach	15°F cooler and dock, 10°F freezer

Source: (PG&E 2007)

To appropriately define each parameter, this study investigated current best practice. Both the current best practice and the "Savings by Design Baseline" values are listed for insulation levels (Table 5), evaporator control (Table 6), condenser control (Table 7), compressor control (Table 8), lighting requirements (Table 9), and general system control (Table 10).

Table 5. Refrigerated Warehouse Shell, Design Baseline and Common Practice

	Savings by Design	Common Practice	ASHRAE Recommendation
Freezer Ceiling	R-46	R-31 to 50	R-45 to 50
Freezer Exterior Wall	R-26	R-32 to 56	R-35 to 40
Freezer Floor	R-30	R-18 to 30	R-27 to 32
Cooler Ceiling	R-23	R-24 to 40	R-30 to 35
Cooler Walls	R-20	R-23 to 40	R-25
Dock Ceiling	R-23	Same as facility	R-30 to 35
Dock to Outdoor Wall	R-20	Same as facility	R-25
Underfloor Heating	No electric	Heat recovery-glycol	None

Source: (PG&E 2007)

Table 6. Refrigerated Warehouse Evaporator, Design Baseline and Common Practice

	Savings by Design	Common Practice
Evaporator fan speed control	Constant volume, constant operation	Constant volume, constant operation
Evaporator design approach temperature	10°F	Variable based on humidity required
Evaporator fan power (W/cfm)	Not addressed	No opinion

Source: (PG&E 2007)

Table 7. Refrigerated Warehouse Condenser, Design Baseline and Common Practice

	Savings by Design	Common Practice
Condenser type	Not addressed	Evaporative condensers in ammonia facilities
Air-Cooled condenser fan speed control	Cycling one-speed fans	Cycling one-speed fans
Air-Cooled cond. design approach temperature	10 to 15°F depending on suction temp.	10 to 15°F depending on suction temp.
Air-Cooled condenser fan power	53 Btu/W-hr at 10°F approach temperature	No comment
Evaporative condenser fan speed control	Two speed fan	Two speed fan
Evaporative cond. design approach temperature	18 to 25°F based on design wet bulb temperature	18 to 20°F
Evaporative condenser fan and pump power	330 Btu/W-hr at 100°F SCT and 70°F wet bulb	No comment

Source: (PG&E 2007)

Table 8. Refrigerated Warehouse Compressor Plant, Design Baseline and Common Practice

	Savings by Design	Common Practice
Compressor capacity modulation	Not addressed	Slide valves on screw compressors, multiple compressor racks on reciprocating plants
Compressor oil cooling	Not addressed	Unclear

Source: (PG&E 2007)

Table 9. Refrigerated Warehouse Lighting, Design Baseline and Common Practice

	Savings by Design	Common Practice
Lighting power density	0.6 W/ft^2	0.4 to 1.2 W/ft^2 depending on application
Lighting controls	Not addressed	No control

Source: (PG&E 2007)

Table 10. Refrigerated Warehouse Refrigeration System Control, Design Baseline and Common Practice

	Savings by Design	Common Practice
Suction pressure control	Not addressed	Fixed
Condensing temperature control	85°F minimum cond. temp, fixed	Fixed
Defrost control	Not addressed	Timer control

Source: (PG&E 2007)

Title 24 specifies that a reflectance of 0.20 is necessary on these structures. This is to combat the up to 40°F temperature increase that sun radiation can cause on the roof. Evaporator fan motor research was presented. The median value for specific fan power is 0.14 W/cfm, with 0.8 W/cfm established as the standard for constant volume air handlers for comfort air conditioning. Work by Northwest Energy Efficiency Alliance was discussed, which showed that variable speed fans obtain energy savings of 24 to 78% with no impact on room conditions. These savings increase as the evaporator coil is oversized. A wide number of analyses were done in relation to compressor and condenser sizing and control. The highest savings were seen with floating head pressure down to 70°F, operating at a 5°F dry bulb offset (air cooled) or 9°F wet bulb offset (evaporative), and utilizing variable speed condenser fans.

In regard to Title 20 and walk-in coolers and freezers, the following recommendations are made:

- Wall and ceiling: R-36 freezer, R-28 cooler
- Floor: R-36 freezer

Electric underfloor heating is not typically utilized in warehouse applications but may be used in smaller walk-in applications. This should be thermostatically controlled to limit energy use. Recommendations are made for control methods, as well as a lighting reduction to 0.6 W/ft^2.

Product Load

(Gortner et al. 1948)

Variations in the air temperature in a refrigerated freezer occur during its operation. Large temperature variations may occur if refrigeration equipment breaks down, control logic response lags, during excessive loads while freezing new product, and lack of partitioning of spaces. This work is focused on defining the sensitivity of the relationship between product quality and storage temperature variation.

Products used to analyze the effects of temperature fluctuation include strawberries, beans, peas, and pork. Typical storage temperatures for pork are specified as -5 to -15°F. Test temperatures ranged from 0 to 20°F.

The result of this analysis was that product quality was not significantly different for product cycled between 0 and 20°F and product maintained at 10°F. One thing specifically noted was the need to keep storage temperatures below 0°F to obtain the optimum product longevity.

(Hustrulid and Winters 1943)

Hustrulid and Winters performed testing to analyze the effect of fluctuating storage temperatures on the quality of frozen fruits and vegetables. Snap beans, corn, peas, strawberries, cantaloupe, and raspberries were put in either a constant temperature locker (30 ft^3) or a locker controlled with oscillating air temperature for six to nine months.

Hustrulid states that no difference is apparent between food subjected to oscillating or constant temperatures. Food exposed to oscillating temperature was varied from -20 to 0°F. However, Hustrulid failed to note the temperature of the constant temperature freezer.

(Woodroof and Shelor 1947)

Woodroof presented information related to freezing foods for storage and the effects of storage temperature fluctuation on food quality. He first presented work done by others, then differentiated his work, and finally discussed his results and conclusions.

Zero degrees Fahrenheit is generally accepted as the most desirable temperature for holding frozen fruits. Tressler and Evers echoed this, stating that 0°F should be maintained at all times and that certain fruits such as peaches should be held at -5°F (Tressler and Evers 1947), as storage life can be increased with lower temperature storage. Bauernfeind and Siemers make a similar recommendation of storing peaches between -10 and 0°F (Bauernfeind and Siemers 1945). Wiegand states that 0°F is necessary for locker plants whereas commercial plants require lower temperatures ranging from -10 to 0°F (Wiegand 1931). However, Lutz recommended storing frozen dewberries with temperatures as high as 10 to 15°F (Lutz et al. 1934). Plagge also suggested slightly higher storage temperatures of 0 to 10°F (Plagge 1938). Diehl and Berry stated that 15°F is the practical upper limit for frozen storage of most plant products (Diehl and Berry 1933). Winters conducted experiments which showed that higher storage temperatures lower the quality of the food (Winters 1942).

Hustrulid and Winters (Hustrulid and Winters 1943) reported that no difference in food quality is apparent for reasonable fluctuations in temperature. Opposing this stance, Woodroof, Birdseye (Birdseye 1946), Tressler, and Plagge all explicitly state that storage temperature fluctuation will decrease the quality of food.

(Becker and Fricke 1996a)

Transpiration and respiration are two processes occurring in products over time. Transpiration is loss of moisture through the product skin via convective mass transfer to the surroundings. Respiration is defined as the chemical process by which fruits and vegetables convert stored sugars into carbon dioxide, water, and heat. Respiration adds a continuous heat load to the refrigerated space and affects the transpiration rate. Transpiration effects the relative humidity maintained by the space and the defrost requirements.

Becker presents details on respiration and transpiration for various fruits and vegetables.

(Love and Cleland 2007)

Chiller cabinets are stand-alone refrigeration units used by convenience stores or supermarket chains to keep small displays of products cool. Typical products for such units are packaged beverages, such as soda or bottled water. In addition to keeping products cool, items delivered to such a unit for display are usually at room temperature initially.

The focus of this article is to model the pull down of a chiller cabinet, predicting the interior temperature distribution over time.

A model was developed and then a prototype unit was tested to validate the model. The display cabinet was 21 ft^3 and contained 600 drink cans. It operated at a set point air temperature of 36.5°F using 1300 ft^3/min and a 9 cc compressor utilizing R-134a. Over a 26 hour period it was designed to pull product from 95°F down to the storage temperature.

Simulations of the prototypical unit were done to analyze the effects of modifying certain components that directly impact the refrigeration capacity of the unit. Results included product pull-down periods ranging from 18.8 to 24.9 hours.

Evaporator Application and Defrost

(Nelson)

Nelson gives a general discussion about defrost methods, including water, electric, and hot gas defrost, highlighting the positive aspects of hot gas defrost. Defrost efficiency is defined by Equation 14.

$$\eta_{def} = \frac{\dot{Q}_{melt}}{\dot{Q}_{def,in}} \qquad (14)$$

(Cole 1989) noted that typical defrost systems only operate at a 15 to 20% efficiency, with 60% of the heat lost to the room via convection and radiation, 20% used to heat the metal in the evaporator, and 5% lost due to hot gas bypass at the end of defrost.

A defrost comparison for a 32°F and -10°F room is made. In a colder room there is a greater convective component resulting in a 63% loss to the room compared to a 46% loss to a 32°F room. The defrost duration is 30 minutes. One item noted by Nelson that is not immediately apparent is the fact that increasing the hot gas temperature is not necessarily the best method, as this increases the percentage of heat lost to the refrigerated space.

The minimum time to melt frost was established at 8 and 10 minutes by Cole (Cole 1989), but industrial facilities have hot gas defrost durations in excess of 30 minutes. Reducing the defrost period to its optimum duration reduces the amount of heat lost to the space. Allowing the frost to build longer on the coil before defrost also reduces heat lost to the refrigerated space during the defrost cycle, however this is a balance between the evaporator performance throughout operation and the benefit from increasing the defrost efficiency.

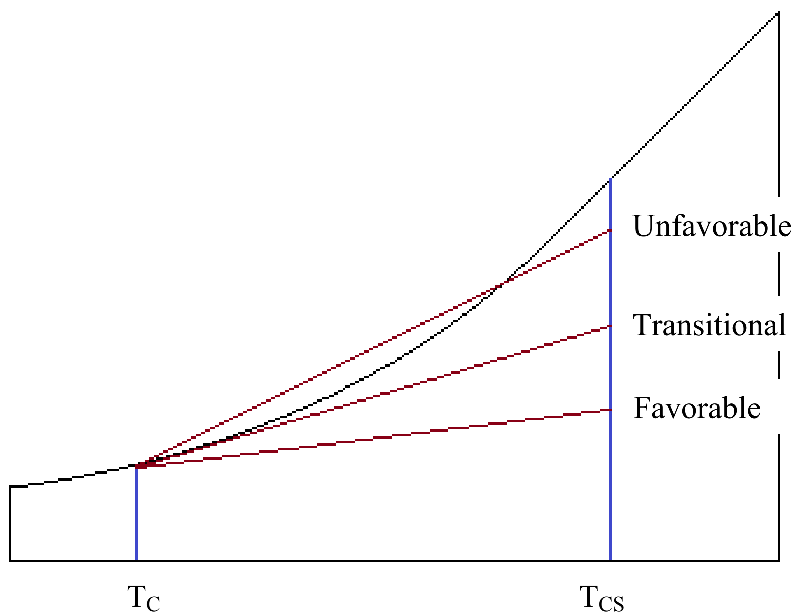
In a cost savings analysis performed by Nelson room temperatures analyzed include 32, 0, -10, and -30°F. Typical defrost efficiencies are noted as 32, 18, 17, and 14%. Nelson predicts that through optimization, these efficiencies may be improved by 91 to 186%.

(O'Hagan et al. 1993)

O'Hagan gathered finned coil performance data under frosting conditions. In an attempt to obtain applicable results, air flow rate was reduced during frosting conditions due to the increased pressure drop over the coil, and a wide range of sensible heat ratios (SHRs) were tested, as coils are often located in areas where they can experience particularly low local SHR. At variations of the parameters noted above, the performance of the coil was measured for comparison.

The concept of unfavorable frost formulation noted by Smith (Smith 1989a) was adopted by O'Hagan. When the line representing the temperature and humidity of the air passing through the coil, crosses the saturation line of the psychrometric chart, unfavorable amounts of frost will form from airborne ice-crystal precipitation. This frost is of a particularly low density, insulating the coil and reducing air flow rate by a significant amount.

Figure 1. Defrost Classifications



Source: (O'Hagan et al. 1993)

The test facility used was 7.87 ft tall, 8.86 ft wide, and 13.78 ft long and constructed of 5.9 inch thick polystyrene sandwich panel. Major system components included, a HCFC-22 test coil, dry heat addition system, and a wet heat addition system.

The coil performance is defined in Equation 15. This was developed to allow for a direct comparison of trials with different SHRs, as presented in Table 11.

$$UA = \frac{\dot{Q}_s}{TD_{Ebal}} = \dot{Q}_s \left[\frac{\ln\left[\left(T_{on} - T_{evap}\right)/\left(T_{off} - T_{evap}\right)\right]}{\left(T_{on} - T_{off}\right)} \right] \qquad (15)$$

Table 11. Experimental Conditions for Coil Performance Trials

Trial	T_{on} (°F)	RH_{on} (%)	SHR	ϕ_s (Btu/hr)
1	31.6	68	0.84	27640
2	31.3	68	0.83	26960
3	29.1	72	0.78	29000
4	30.6	83	0.71	22860
5	29.1	82	0.71	22180
6	31.5	82	0.71	22180
7	32.0	76	0.70	33440
8	32.4	77	0.70	33780
9	30.9	78	0.70	31390
10	31.3	81	0.70	33440
11	29.5	87	0.70	29340
12	30.7	89	0.63	26960
13	28.8	87	0.62	25930
14	31.8	87	0.58	28320
15	32.2	93	0.54	31730
16	32.4	93	0.54	31730

Source: (O'Hagan et al. 1993)

Test result analysis showed variability of frost formation at low SHR levels. Previous work on coil performance with frosting has indicated slight initial performance enhancement prior to degradation due to the insulating effect of the frost. No enhancement was seen by O’Hagan et al. Air flow restriction for the same amount of frost deposited was greater at lower SHR values. After operating at low SHR values, transitioning to higher SHR conditions offers some recovery to coil performance.

(Cleland and O'Hagan 2003)

The heat transfer performed by an evaporator diminishes as water condenses on the coil and turns into frost. This frost continues to build and reduce performance until a defrost cycle removes it. In sizing an evaporator to maintain certain interior conditions, careful attention should be paid to the psychrometrics of the air passing through the coil.

Sherif (Sherif et al. 2001) reported experimental results on coils operating at typical low-temperature refrigerant conditions of -40°F evaporator temperature and 17°F air-on temperature. Cleland reports results from testing of frosting on coils in rooms with air-on temperatures of approximately 32°F (28.8 to 32.9°F) at moderate temperature differences, a wide range of the sensible heat ratios (0.63 to 0.78), and an evaporator temperature ranging from 9 to 12.6°F.

As defined by Smith (Smith 1992; Smith 1989b) air conditions should pass below the saturation curve on the psychrometric chart (Figure 1). To avoid undesirable frost formation, coils should be sized so that the coil surface temperature is greater than the critical coil surface temperature as defined by Equations 16 and 17.

$$\omega_{ws} \approx \frac{1.9283 \times 10^{7}}{\exp\left(\frac{6071.67}{T_{cs} + 271.511}\right) - 3.106705 \times 10^{7}} \tag{16}$$

$$T_{\mathrm{csc}} = T_{on} - \frac{\omega_{on} - \omega_{ws}}{\left(\frac{1}{SHR} - 1\right)\frac{c_a}{\Delta h_{sub}}} \tag{17}$$

For a discrete number of conditions, the critical surface temperature was calculated (Table 12). Due to the relationship of these parameters, any one parameter may be determined given the other two.

Table 12. Critical Conditions to Avoid Unfavorable Frost During Evaporator Operation with Air-On Temperature of 32°F

R_{on} (%)	T_{sc} (°F)	SHR_c
60.0	2.5	0.81
64.4	5.0	0.79
70.0	8.2	0.77
80.0	13.8	0.72
80.2	14.0	0.71
86.1	17.6	0.68
90.0	20.1	0.65
91.4	21.2	0.64
95.0	24.1	0.62
95.8	24.8	0.61

Source: (Cleland and O'Hagan 2003)

(Mago and Sherif 2005)

Frosting of coils during operation reduces performance of evaporators. This study analyzes this effect on coils operating near saturated or under supersaturated conditions. Operation near saturated conditions is a common practice. Items discussed include the two types of frost formations, issues with using hot-gas defrost, and some methods to improve the energy efficiency of the process.

Defrost efficiency is the ratio of the energy required to melt the ice to the energy put into the defrost cycle (Equation 14).

$$\eta_{def} = \frac{\dot{Q}_{melt}}{\dot{Q}_{def,in}} \qquad (14)$$

The experimental facility is a small walk-in freezer with an industrial sized freezer coil (4 fins per inch) installed in it. The space is isolated from infiltration through use of good seals on the doors and the filling of all cracks. To determine the heat transfer properties of the space, the enclosure was tested in a clean, dry, and empty state. Some test conditions that were utilized are listed in Table 13.

Table 13. Experimental Conditions for Various Tests Performed

Experiment	Entering Air		Exit Air		Space		Coil Face
	Dry Bulb (°F)	RH (%)	Dry Bulb (°F)	Degree of Sat. (%)	Temp (°F)	RH (%)	Velocity (fpm)
Demarcation Scenario 1	17.1	64	2.8	92			
Demarcation Scenario 2	17.1	84	2.8	102			
Demarcation Scenario 3	17.1	92	3.0	105			
Demarcation Scenario 4	17.1	99	3.0	109			
Hot Gas Test Initial					3.0	75.6	768
Hot Gas Case A					6.8 to 10.4		
Hot Gas Case B					5		
Heat Pipe Test	37.4	50	17	88			755
High Face Vel. Initial	6	75					315, 728, 1280

Source: (Mago and Sherif 2005)

Demarcation testing showed definite differences between frost forming during sub-saturated conditions and super-saturated conditions, with super-saturated frost being snow-like. Two mechanisms drive frost formation. The first is the relationship between the free stream humidity and the coil surface temperature. The second driving force is due to water condensing out of the air and being convected into the coil. During super-saturated conditions water vapor transitions directly to ice crystals, which are easily deposited once they are convected into the evaporator coil.

Hot gas defrosting was tested during super and sub-saturated conditions. Sub-saturated testing (Case B) had little frosting of the coil, even with a coil surface temperature of -7.6°F. The super-saturated test (Case A) resulted in an evaporator coil completely blocked by ice after four hours of operation. When defrosted, ice fell off in chunks onto the floor as opposed to draining into the pan, with a defrost efficiency of only 22%.

Energy reduction techniques investigated include application of heat pipes, dampers, and high face velocities. When heat pipes are applied to cooling dehumidifying processes they reduce energy usage by approximately 25%. Dampering a coil during the defrost cycle reduces the amount of defrost heat that is added to the space. Defrost efficiencies have been shown to increase by 43% (Mago and Sherif 2002) through the use of dampering. Higher face velocities during operation result in higher defrost efficiencies during the defrost cycle. Defrost efficiency increased from 18 to 70.4% with a change in face velocity from 315 to 1280 fpm. A typical well-designed hot gas system is noted to require defrost for only about 30 minutes every day.

Infiltration Load

(Takahashi and Inoh 1963)

Takahashi discusses the different types of air curtains utilized by cold rooms, including outside down-blow type, inside down-blow-circulation type, and inside lateral-blow-circulation type. Each type of air curtain was tested to analyze what method achieved the highest decrease in infiltration. In conclusion, most air curtains reduced heat loss through doorways to about 20-40% (the air curtains were 80-60% efficient), respectively, almost independent of the type of air curtain applied.

(Gosney and Olama 1975)

Gosney and Olama developed their well-known definition of infiltration rate (Equation 18) in this article. Gosney discusses the current available information on infiltration load calculation, including a table in Chapter 23 of the 1972 Handbook of Fundamentals (ASHRAE 1972) and a graph in *Food Engineering* from an undisclosed source (Unattributed 1969). Additional work cited in this article includes anemometric method testing performed by Fritzsche and Lilienblum (Fritzsche and Lilienblum 1968), Shaw (Shaw 1971), and Brown and Solvason (Brown and Solvason 1962).

$$\dot{m}_{ss} = 0.221 \cdot 3600 \frac{s}{hr} A_{\inf} \rho_C (1 + \Delta\omega) \left(1 - \frac{\rho_H}{\rho_C}\right)^{1/2} (gH)^{1/2} \left[\frac{2}{1 + (\rho')^{1/3}}\right]^{3/2} \tag{18}$$

$$\rho' = \frac{\rho_C}{\rho_H} \tag{19}$$

The primary assumption of the testing method utilized was that infiltration is a function of density differences, with the temperature or compositions of the bodies of air not effectively contributing to the results. Carbon dioxide was used to simulate the dense cold air through a tracer gas method. Model doors were used for testing, with validation testing done on full sized doorways. Based on a theory of Brown and Solvason and work by Tamm (Tamm 1966) and Fritzsche and Lilienblum, the volume flow rate of air through an open door was derived as Equation 20.

$$\dot{V}_u = \frac{C_d}{3} A_{\inf} \left(1 - \frac{\rho_H}{\rho_C}\right)^{1/2} (gH)^{1/2} \left[\frac{2}{1 + (\rho_C / \rho_H)^{1/3}}\right]^{3/2} \cdot 60 \frac{\text{sec}}{\text{min}} \tag{20}$$

The need for the correction factor was identified by Fritzsche and Lilienblum, who were building on Tamm's efforts. This correction factor is noted as being more than a discharge coefficient, as it also adjusts for some important thermal effects. Work by Fritzsche and Lilienblum is not directly applied by those in industry because issues with their method of test were discovered following publication, with imbalances in flow ranging from 5 to 20%.

The scale cold store test simulated a 108°F temperature difference between spaces with an 8.2 ft tall door. Results include a relationship between various parameters defining flow characteristics (Equation 21 with supporting Equations 22, 23, and 24).

$$\frac{Sh}{(Sc)F_v} = 0.2144(Gr)^{0.5012} \tag{21}$$

$$Sh = \frac{\dot{m}_{wv}\, H}{A_{\inf} D(\xi_H - \xi_C)\rho} \cdot \frac{hr}{3600s} \tag{22}$$

$$Sc = \frac{v}{D} \tag{23}$$

$$F_v = \left[\frac{2}{1 + (\rho_H / \rho_C)^{1/3}}\right]^{3/2} \tag{24}$$

Testing results were compared to work done by others using a plot of the Equation 21 versus the Grashof number. Brown and Solvason's results were slightly higher for unknown reasons, but their method of calorimetric testing was questioned. Shaw's results are comparable to Gosney and Olama's results for Grashof greater than 2 x 10^9 despite different testing methods. Below this, Grashof number values tend to a constant Nusselt number of around 10^4, which may be attributed to imbalanced room pressures during Shaw's testing. Fritzsche and Lilienblum's results have a higher slope, which is due to a large leak in their test facility that went unnoticed until after their results were published. One final thing that Gosney et al. presents is the fact that his equation slightly overpredicts infiltration as an assurance for cold store designers. Equation 21 is simplified into Equation

25 to give a more usable form. From this, equations defining water and heat transfer are developed (Equations 26 and 27), where Equation 27 defines a factor to compensate for cycling of the doorway, combining Equations 28, 29, and 30.

$$\dot{m}_{wv} = 0.221 \cdot A_{\inf} \cdot (\xi_H - \xi_C)\rho_C \left(1 - \frac{\rho_H}{\rho_C}\right)^{1/2} (gH)^{1/2} \left[\frac{2}{1 + (\rho_C / \rho_H)^{1/3}}\right]^{3/2} \Phi \cdot 3600 \frac{\sec}{hr} \quad (25)$$

$$\dot{Q}_{\inf-D} = 0.221 \cdot A_{\inf} \cdot (h_H - h_C)\rho_C \left(1 - \frac{\rho_H}{\rho_C}\right)^{1/2} (gH)^{1/2} \left[\frac{2}{1 + (\rho_C / \rho_H)^{1/3}}\right]^{3/2} \Phi \cdot 3600 \frac{\sec}{hr} \quad (26)$$

$$\Phi = \left(1 - \frac{0.0112}{\left(\frac{v t_{open}}{H^2}\right)^{0.3565}}\right)\left(t_{open} f_d\right)^{0.1} \quad (27)$$

$$\Phi = \left(1 - \frac{0.0112}{N_o^{0.3565}}\right) N_f^{0.1} \quad (28)$$

$$N_f = t_{open} f_d = \frac{t_{open}}{t_d} \quad (29)$$

$$N_o = \frac{v t_{open}}{H^2} \quad (30)$$

(Longdill and Wyborn 1978)

Infiltration is a large load on refrigerated spaces at 1.54 x 10^6 Btu/h for enclosed loading docks and as high as 4.09 x 10^6 Btu/h when there is no enclosed loading space attached to a refrigerated storage facility. Air curtains are often utilized to reduce this infiltration load. Longdill and Wyborn's study was focused on developing a basic method to optimize air curtain performance to allow for easier application by cold store designers.

The infiltration flow pattern for a typical doorway has fast air movement at the bottom of the door leaving the refrigerated space, high air flow at the top of the door entering the refrigerated space, and a transition region in between. The maximum velocity for such a door may be represented by Equation 31. Again, this is the maximum velocity found at the top and bottom of the door, with the average velocity being much lower. Profiles for the entering and exiting air flows are given by Equations 32, 33, and 34 (Tamm 1969).

$$u_{mf} = \left[\frac{2gH(1-s)}{1+\sqrt[3]{s}} \right]^{1/2} \tag{31}$$

$$u_C = \left[2g(z - x_C)(1-s) \right]^{1/2} \tag{32}$$

$$u_H = \left[2g(x_H - z)\left(\frac{1}{s} - 1 \right) \right]^{1/2} \tag{33}$$

$$z = \frac{H}{1+\sqrt[3]{s}} \tag{34}$$

Longdill performed tracer-gas method tests, with and without air curtains in operation, to analyze the effectiveness of the unit on a 6250 ft^3 freezer with a 69.8°F outdoor temperature. A vertical air curtain was tested on a 1.74 x 5.41 ft doorway. Variations in jet velocity and jet thickness were studied at store temperatures of -4 to 32°F. The optimum jet velocity is a function of the curtain-jet thickness and the refrigerated space temperature. This optimum point is somewhat sensitive to variations in flow rate, approximated in Table 14. Air curtains operating with 80% efficiency would represent a 682,400 Btu/hr savings for this storage freezer.

Table 14. Optimum Efficiency Point of Vertical Air Curtain and Sensitivity to Jet Velocity Variation

Jet Width (in)	Room Temperature (°F)	η, Peak Efficiency (%)	U_{je}, Optimum Jet velocity (ft/s)	η Sensitivity $\delta\eta/\delta U_{je}\big\vert_{\Delta U=10\%}$
5/8	-4	80.0	51.0	- 8%
1 ½	32	74.5	26.0	- 3%
1 ½	-4	82.0	40.5	- 4%
4 ¾	-4	81.5	33.5	- 1%
3 ¼	-4	78.5	29.5	- 2%

Source: (Longdill and Wyborn 1978)

The effects of jet orientation, jet velocity, and freezer temperature were analyzed for a horizontal air curtain on a 4.00 x 6.43 ft doorway. The optimum jet angle from the plane of the door was 10 degrees at 32°F and 25 degrees at -4°F, with flow oriented into the refrigerated space at the bottom of the door and away from the refrigerated space at the top of the door. Infiltration through horizontal air curtains was defined by Longdill with Equation 35 where K_2 equals 0.78 for θ = 10° and 0.99 for θ = 25°.

$$u_{op} = C_{ac}\left[\frac{1}{4S\sin\theta_{jet}}\frac{W}{b}\right]^{1/2} u_{mf} \qquad (35)$$

Information gathered on the effect of wind on air curtains was presented by Longdill (Table 15).

Table 15. The Effect of Wind on Optimized Curtain Efficiency

Curtain Type	Curtain Number	Curtain Efficiency % No Wind	0.5 ft/s Draught	Wind Load (ft/s)	Side Wind	Direct Wind
Vertical	1	77	35	22.0	2	6
Vertical	2	82	33	22.0	3	16
Vertical	3	83	31	22.0	-9	16
Vertical	4	79	-	22.0	-7	27
Vertical	5	68	15	22.0	1	19
Horizontal	6	82	30	17.1	13	0
Horizontal	6	82	30	8.5	37	1

Source: (Longdill and Wyborn 1978)

(Jones et al. 1983)

ASHRAE handbook data often underestimates moisture infiltration, which may enter through ducts passing through unconditioned spaces, by natural convection at openings, and/or by upstream diffusion through outflowing air. This paper presents a method to estimate infiltration in a more complete manner and offers solutions to reduce these modes of infiltration.

To perform the testing for this work a 12 x 6 x 6 ft test chamber was constructed of plywood, supported by a wood frame. The chamber was lined with sheet metal to ensure a vapor barrier between the exterior conditions and the chamber construction materials. The exterior of the chamber was encased in 2 inch foam insulation. Jones et al. (Jones et al. 1983) applied the results of this testing to define moisture infiltrating into a refrigerated space.

Moisture added to a cold room is more than would be expected due to reverse flows (Equation 36 or Equation37). Equation 36 is accurate within 20% assuming the following conditions are met:

- The temperature difference between spaces is greater than 5°F
- There are no sources of turbulence directed at either side of the opening
- The length of the opening is short compared to its height and width
- The opening is on a vertical surface

Equation 37 is applicable for aspect ratios (length to height) greater than 0.5.

$$\dot{m}_r = \frac{\Delta\omega}{C_s}\left(0.26\,\dot{m}_o - 0.33\,\dot{m}_{da}\right) \tag{36}$$

$$\dot{m}_r = \Delta\omega\left(0.10\,\dot{m}_o - 0.26\,\dot{m}_{da}\right) \tag{37}$$

(Pham and Oliver 1983)

Pham and Oliver performed research detailed in this report (Pham and Oliver 1983), that is referenced throughout the field of refrigeration. In this work, Pham analyzes the effectiveness of a correlation derived by Tamm ((Tamm 1965), Equation 38) using velocity traversing and a tracer gas method test. In addition, the effect of forklift traffic and air curtains are analyzed.

$$\dot{V}_u = \frac{2WH}{3}\sqrt{\frac{2gH(1-s)}{\left(1+s^{1/3}\right)^3}}\cdot 60\,\frac{s}{\min} \tag{38}$$

Too validate Tamm's equation, multiple test freezers were employed, ranging from a 6250 ft^3 facility with a 12.47 inside height, 3.94 x 5.25 ft test doorway, and a test temperature difference of 35.1°F, to a 1,306,643 ft^3 facility with an inside height of 69.6 ft, test doorway dimensions of 9.84 x 11.81 ft, and a temperature difference of 61.2°F across the doorway. Forklift traffic passed through the large doorway at one pass per minute. Test conditions and results summarized in Table 16.

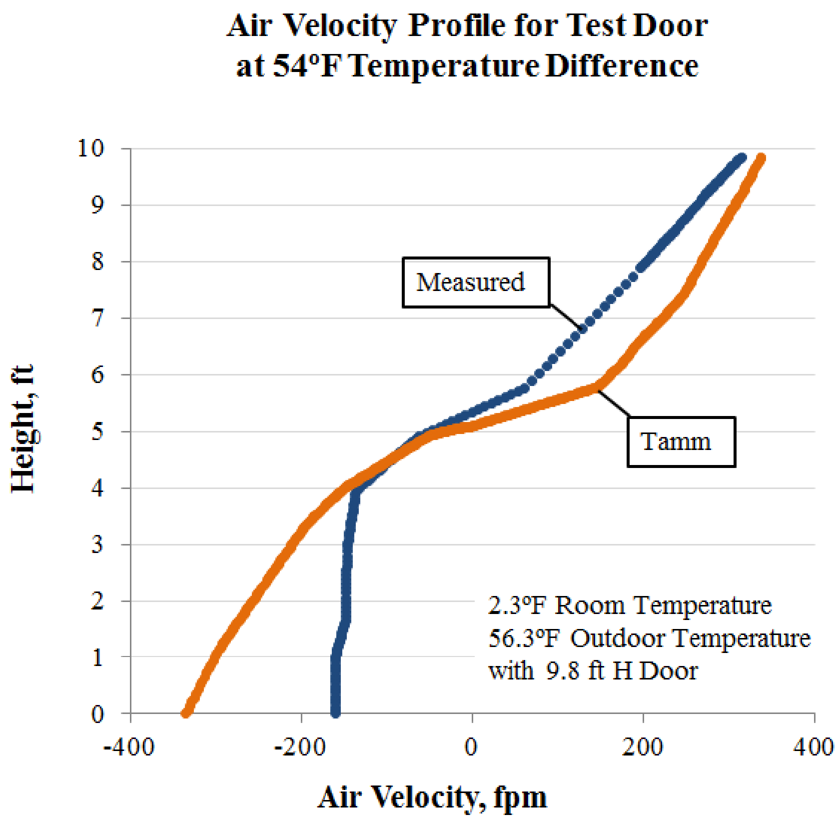
Table 16. Air Interchanges Across Cold Store Doors

Store height (ft)	Door width (ft)	Door height (ft)	Inside temp (°F)	Outside temp (°F)	Measured interchange (cfm)	Interchange from Equation 38
27.6	10.0	9.0	2.3	56.3	6,822.8	10,403.7
27.6	10.0	9.0	2.3	56.3	6,865.2	10,403.7
16.7	8.9	9.5	-0.4	59.0	7,225.4	10,297.8
21.3	9.8	11.8	1.4	44.6	9,195.9	13,560.8
69.6	5.8	7.0	-4.0	57.2	2,585.0	4,322.5
12.5	3.5	6.5	23.9	59.0	1,334.9	1,758.7

Source: (Pham and Oliver 1983)

With the above data, Pham suggests applying an empirical factor to Tamm's equation of 0.68. Pham's testing also included profile testing on a 9.84 ft tall door, with a 2.3°F inside temperature and a 56.3°F outside temperature (Figure 2). Tamm overpredicts the flow rate at the bottom of the doorway, with the measured data profile from the midpoint of the doorway down being a mostly constant profile.

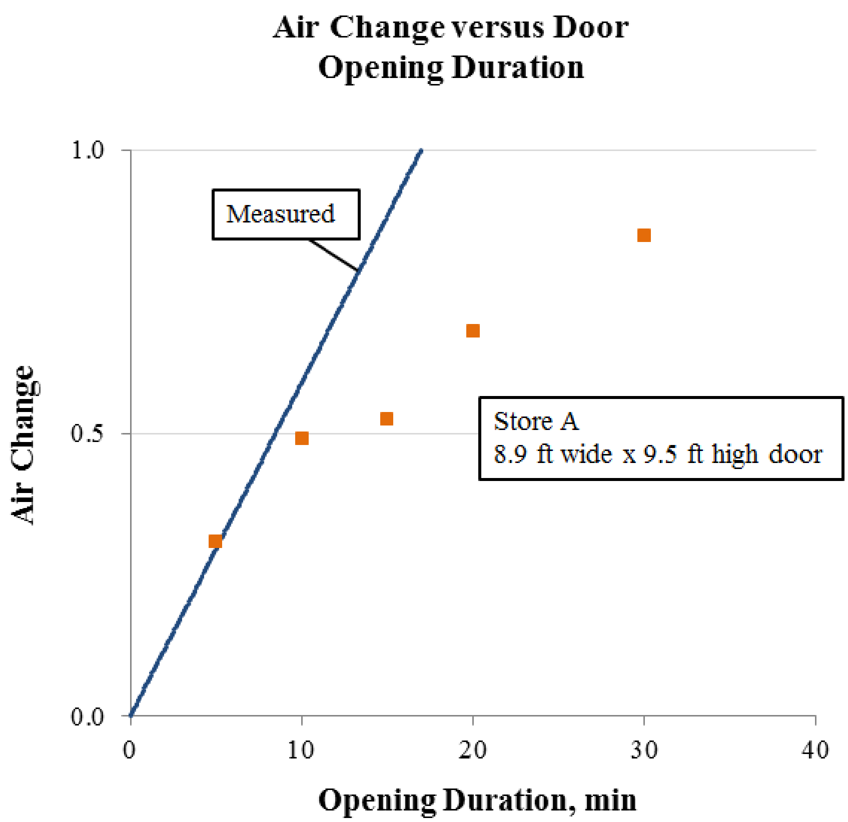
Figure 2. Air Velocity Profile for Test Door at 54°F Temperature Difference



Source: (Pham and Oliver 1983)

Tracer gas testing was performed on four conventional, palletized refrigerated storage facilities, with gross volumes of 141,259 ft^3, 197,762 ft^3, 388,461 ft^3, and 1,341,957 ft^3. Doors were enclosed by a loading area or exposed to the outdoors. Infiltration protection devices included no protection, horizontal or vertical air curtains, plastic strip curtains, and a combination of air curtain and strip curtain protections. One interesting finding was that infiltration decays with time (Figure 3). The cause of this was attributed to internal partitioning of air flow as opposed to a rise in the internal box temperature. Product stacked around the doorways and throughout the cold store, limit or restrict air flow to the doorway. A reduction factor of about 0.53 has been observed by Pham et al. for unprotected doorways in more than one situation.

Figure 3. Air Change versus Door Opening Duration



Source: (Pham and Oliver 1983)

Door protective devices tested included air curtains, strip curtains, and a combination. Their performance and the effect of forklift traffic are summarized in Table 17.

Table 17. Effectiveness of Door Protective Devices and the Effect of Forklift Traffic on 9.84 x 11.81 ft Doorway

	Efficiency (%)	Increase in Infil. (%)
Vertical non-recirculating air curtain, 4.5 inch slot, 1969 fpm jet	79 ± 3	
Horizontal recirculating air curtain, 2.4 inch slot, 1339 fpm jet	76 ± 3	
Plastic strip curtain	93 ± 1	
Horizontal air curtain (as noted above) and plastic strip curtain	91 ± 1	
Forklift with pallet, one pass per minute, unprotected doorway	21 ± 10	- 21 ± 10
Forklift with pallet, one pass per minute, air and/or plastic curtains		43 ± 62
Internal circulation fans off for forklift passage		- 9 ± 6

Source: (Pham and Oliver 1983)

(Hendrix et al. 1989)

Hendrix et al. analyzed the available correlations for infiltration into cold rooms using measured data from four test rooms at two refrigerated warehouses (Hendrix et al. 1989). Rooms 1 and 2 are 25,000 ft^2 (250 x 100 ft) and 30 ft tall for a volume of 750,000 ft^3. Construction is of pre-fabricated, insulated panels, with three central floor-to-ceiling storage racks and additional racks around the perimeter. The space is cooled by two centrally mounted unit coolers rated at 52,000 Btu/h each (7-8°F approach and 63,885 cfm) which maintain -15°F for one room and 0°F for the other. Two 7 x 10 ft pneumatically actuated horizontal sliding insulated doors are positioned on one end of each room, protected with vestibules and strip curtains. On the other end of each room there is a similar door equipped with strip curtains but no vestibule.

The second facility has a freezer space, ice cream freezer space, and refrigerated dock space. The freezer is 150 x 180 ft (27,000 ft^2) with a 30 ft ceiling (810,000 ft^3). operating at -10°F. Construction and rack spacing is similar to the other facility. One end of the freezer connects to a smaller ice cream freezer (40 x 180 ft, 7200 ft^2, 180,000 ft^3), operating at -15°F. There are two 8 x 12 ft tall openings to the loading dock protected by air curtains, a conditioned vestibule, and bi-fold strip curtains. Refrigeration is provided by three unit coolers positioned at the end of the freezer, with an additional refrigeration unit in the ice cream freezer. The loading dock operates at 40°F using ceiling mounted unit coolers and is 4500 ft^2 with a 17 ft ceiling (76,500 ft^3). It has six truck bays and an additional 8 x 12.67 ft high doorway protected by an air curtain leading to a dry storage area.

Ambient conditions during testing ranged from dry bulb of 37 to 85°F and relative humidity of 20 to 89%. These conditions corresponded to temperature differences from 13 to 90°F. General findings of the study include strip curtain effectiveness greater than 90% and vestibule effectiveness of nearly 75%. Effectiveness values previously determined by others were presented in Table 18.

An additional finding was the fact that transient behavior is not important during door opening events. Hendrix et al. states that D_f for an unobstructed doorway should always be equal to 1 whereas data collected by Gosney and Olama (Gosney and Olama 1975) suggest a value of 0.5 to 0.6 for the first 20 to 30 seconds that the door is open. Hendrix extends the insignificance of transient effects to long door open periods. In this study, no transient effects were observed for door open periods up to 45 minutes.

Table 18. Previously Reported Effectiveness Values for Infiltration Protection Devices

	Effectiveness (%)	Source
Strip curtain	93	(Pham and Oliver 1983)
Vertical non-circulating air curtain	54 to 81	(Takahashi and Inoh 1963)
Vertical non-circulating air curtain	68 to 83	(Longdill and Wyborn 1978)
Vertical non-circulating air curtain	79 ± 3	(Pham and Oliver 1983)
Vertical double non-recirculating air curtain	60 to 93	(Van Male 1983)
Vertical recirculating inside room air curtain	36 to 80	(Takahashi and Inoh 1963)
Vertical recirculating inside doorway air curtain	58	(Takahashi and Inoh 1963)
Horizontal recirculating air curtain	59	(Takahashi and Inoh 1963)
Horizontal recirculating air curtain	82	(Longdill and Wyborn 1978)
Horizontal recirculating air curtain	76 ± 3	(Pham and Oliver 1983)
Combined air and strip curtains	91 ± 1	(Pham and Oliver 1983)

Source: (Hendrix et al. 1989)

Infiltration methods analyzed by Hendrix et al. include work by Brown and Solvason (Brown and Solvason 1962) in Equation 39, Tamm (Tamm 1966) in Equation 40, Fritzsche and Lilienblum (Fritzsche and Lilienblum 1968) in Equation 41, Gosney and Olama (Gosney and Olama 1975) in Equation 42, and Jones, Beck, and Steele (Jones et al. 1983) in Equation 43. When the infiltration rate is plotted versus any of the contributing factors, all analytical models trend in a similar manner. The main difference is that these trends are offset from each other by up to 80%, with Tamm consistently predicting the highest infiltration, followed by Brown et al., Jones et al., Fritzsche et al., and Gosney et al., with Fritzsche et al. and Gosney et al. switching positions as the lowest infiltration prediction depending on the value of the independent variable.

$$\dot{Q}_{\inf} = 0.343 A_{\inf} (gH)^{1/2} \left[\frac{\rho_C - \rho_H}{\bar{\rho}} \right]^{1/2} \left(1 - 0.498 \left(\frac{D_{door}}{H} \right) \right) \bar{\rho} (h_H - h_C) \cdot 3600 \frac{s}{hr} \quad (39)$$

$$\dot{Q}_{\inf} = 0.333 A_{\inf} (gH)^{1/2} \left[\frac{\rho_C - \rho_H}{\rho_C} \right]^{1/2} \left(\frac{2}{1 + (\rho_H / \rho_C)^{1/3}} \right)^{3/2} \rho_C (h_H - h_C) \cdot 3600 \frac{s}{hr} \quad (40)$$

$$\dot{Q}_{\inf} = 0.333(0.48 + 0.004(T_H - T_C)) A_{\inf} (gH)^{1/2} \left[\frac{\rho_C - \rho_H}{\bar{\rho}} \right]^{1/2} \ldots$$

$$\ldots \left(\frac{2}{1 + (\rho_H / \rho_C)^{1/3}} \right)^{3/2} \rho_C (h_H - h_C) \cdot 3600 \frac{s}{hr} \quad (41)$$

$$\dot{Q}_{\inf} = 0.221 A_{\inf} (gH)^{1/2} \left[\frac{\rho_C - \rho_H}{\rho_C} \right]^{1/2} \left(\frac{2}{1 + (\rho_C / \rho_H)^{1/3}} \right)^{3/2} \rho_C (h_H - h_C) \cdot 3600 \frac{s}{hr} \quad (42)$$

$$\dot{Q}_{\inf} = 0.1733 A_{\inf} (gH)^{1/2} \Delta\omega \left(1 + \frac{1}{\Delta\omega} \right) \left(\frac{T_H}{T_C} - \frac{T_C}{T_H} \right)^{1/2} \rho_C (h_H - h_C) \cdot 3600 \frac{s}{hr} \quad (43)$$

(Foster et al. 2003)

Numerous analytical models are compared, along with CFD modeling, to determine the best method for predicting airflow through doorways.

Empirical models discussed include Brown and Solvason ((Brown and Solvason 1962), Equation 39), Tamm ((Tamm 1966), Equation 40), Fritzsche and Lilienblum ((Fritzsche and Lilienblum 1968), Equation 41), Gosney and Olama ((Gosney and Olama 1975), Equation 42), and Pham and Oliver ((Pham and Oliver 1983), Equation 44).

$$\dot{Q}_{\inf} = 0.226 A_{\inf} (gH)^{1/2} \left[\frac{\rho_C - \rho_H}{\rho_C} \right]^{1/2} \left(\frac{2}{1 + (\rho_H / \rho_C)^{1/3}} \right)^{3/2} \rho_C (h_H - h_C) \cdot 3600 \frac{s}{hr} \quad (44)$$

Testing was completed using a cooler with internal dimensions of 15.75 x 19.0 x 12.47 ft high, operating a 33.8°F. Air flow was measured through 4.46 x 10.50 ft, 3.28 x

10.50 ft, and 1.42 x 2.26 ft openings with an average outdoor temperature of 68°F. A leakage rate of 0.33 cfm was detected on the closed door.

Brown and Tamm models were found to overpredict infiltration by 52.1 to 122.7%. Fritzsche and Gosney models give the closest predictions for the 3.28 ft wide door, but the Gosney and Olama equation (Equation 18) was found to be the closest comparison to experimental results overall. CFD only has an advantage to this empirical method for the case in which infiltration is changing with time.

(Elsayed 1998)

Elsayed discussed the various excepted models of infiltration, checked their accuracy, studied the effect of Grashof number, studied the effect of the room height relative to the door height, and predicted a doorway flow factor.

Infiltration load is heat added to the refrigerated space by mass flow of warm air into the space and cold air out of the space (Equation 45). The infiltration for a given door may be approximated by applying factors for the open period, flow factor, and door flow protective devices in Equation 46 (ASHRAE 1994).

$$\dot{Q}_{\mathrm{inf}} = \dot{m}_{\mathrm{inf}}\left(h_H - h_C\right) \tag{45}$$

$$\dot{m}_{\mathrm{inf}} = \dot{m}_{ss}\, D_t D_f \left(1 - E\right) \tag{46}$$

Correlations analyzed by this study include relationships developed by Gosney and Olama (Gosney and Olama 1975), Jones et al. (Jones et al. 1983), and Cole (Cole 1984). These relationships are given in Equations 47, 48, and 49, respectively.

$$\frac{\dot{m}_{ss}}{W} = 0.221\rho_C\left(1-\frac{\rho_H}{\rho_C}\right)^{0.5} g^{0.5}H^{1.5}\left[\frac{2}{1+\left(\rho_C/\rho_H\right)^{1/3}}\right]^{1.5}\cdot 3600\frac{s}{hr} \tag{47}$$

$$\frac{\dot{m}_{ss}}{W} = 0.173\bar{\rho}\left[gH\left(\frac{T_H}{T_C}-\frac{T_C}{T_H}\right)\right]^{0.5} H \cdot 3600\frac{s}{hr} \tag{48}$$

$$\frac{\dot{m}_{ss}}{W} = 0.0603\,\bar{\rho}\left(T_H-T_C\right)^{0.5} H^{1.5}\cdot 3600\frac{s}{hr} \tag{49}$$

Elsayed points out that Equations 47 and 48 are derived assuming non-viscous one-dimensional flow whereas Equation 49 is derived assuming a zero dimensional flow profile. These correlations do not take into account the aspect ratio of the door or its geometric relationship to the room that flow is entering.

A correlation (Equation 50) was derived assuming a two-dimensional flow profile, Boussinesq approximation, and isothermal walls, with supporting Equations 51 through 55. The sensible heat factor is also defined by Equation 56. Sensible heat transfer is approximated by Equation 57.

$$\dot{M}_M = \frac{\dot{m}_M}{\rho\cdot u_{ref}WH_1} = -\int_0^{OR}\left(U\theta dY\right)_{x=\bar{B}} + \frac{1}{Gr^{1/2}\,\mathrm{Pr}}\int_0^{OR}\left(\frac{d\theta}{dX}dY\right)_{x=\bar{B}} \tag{50}$$

$$U=\frac{u}{u_{ref}},\ \theta=\frac{T-T_H}{T_H-T_C},\ Y=\frac{y}{H_1},\ X=\frac{x}{H_1},\ \bar{B}=\frac{B}{H_1} \tag{51}$$

$$u_{ref}=\sqrt{g\beta\Delta T_{ref}H_1} \tag{52}$$

$$Gr=\frac{\rho^2 g\beta\left(T_H-T_C\right)H_1^3}{\mu^2} \tag{53}$$

$$H_1=\frac{H}{OR} \tag{54}$$

$$\Pr = \frac{\mu \cdot C_p}{k}\left(3600\frac{s}{hr}\right) \tag{55}$$

$$SHR = \frac{h(T_H, \omega_C) - h(T_C, \omega_C)}{h(T_H, \omega_H) - h(T_C, \omega_C)} \tag{56}$$

$$\dot{Q}_s = \dot{m}_M \; C_p (T_H - T_C) \cdot 60 \frac{\min}{hr} \tag{57}$$

Combining Equations 50 through 57, Equation 58 is derived.

$$\dot{Q}_{\inf} = \rho \cdot u_{ref} W H_1 \left[-\int_0^{OR} (U\theta dY)_{x=\bar{B}} + \frac{1}{Gr^{1/2}\Pr}\int_0^{OR}\left(\frac{d\theta}{dX}dY\right)_{x=\bar{B}}\right]\frac{C_p(T_H - T_C)}{SHF} \cdot 60\frac{\min}{hr} \tag{58}$$

The flow factor is defined by Equation 59 and supporting Equation 60. A correlation is suggested based on steady state flow from the Elsayed correlation (Equation 61). ASHRAE suggests a value for D_f of 0.8 to 1.1, depending on the temperature difference between the refrigerated space and the outdoor air (ASHRAE 1994).

$$D_f = \int_0^{\tau} \dot{M}_M \cdot d\tau \,/\, \dot{M}_{ss} \tag{59}$$

$$\tau = \frac{t \cdot u_{ref}}{H_1} \cdot \frac{\min}{60s} \tag{60}$$

$$D_f = 6\tau^{-0.526} \tag{61}$$

Typical heights of cold rooms were cited as ranging from 6.56 to 13.12 ft. Small heights of 0.03 to 0.06 ft may be used to represent gaps in the closed door if not well sealed.

The study determined that the Grashof number and the opening ratio affect the infiltration rate. Secondly, the infiltration flow rate for rooms with equal height (H_1) and depth (B), $10^5 \leq Gr \leq 10^{10}$, and $0.25 \leq OR \leq 0.5$ increases for a certain period, decreases for a period of time, and then stabilizes at steady state. It was found that the Gosney Olama equation (Equation 47) overpredicts the steady state flow rate for rooms with aspect ratios (H_1/B) equal or greater than one. This study depicts much higher values for D_f than those suggested by ASHRAE (ASHRAE 1994). Elsayed found that his correlations for infiltration rate and heat transfer correlated well with data in literature to which his results were compared.

(Chen et al. 2002; Cleland et al. 2004)

Infiltration is a significant load for all refrigerated spaces. A modified model was developed to define infiltration loading through door openings based on measured data on rapid-roll and sliding doors. Testing was performed on seven different commercial facilities ranging from 26,000 to 424,000 ft^3 in size and operating at temperatures ranging from -7 to 38°F (Table 19). Additional variables considered include infiltration protection methods and forklift movement rates.

Table 19. Description of Refrigerated Warehouses and Doors Tested

Site	Number of Doors	Space Temp (°F)	Volume (ft^3)	Main Door (Type, W x H)
A1	1+1P	30	26,000	SD: 5.91 x 7.87 ft
A2a	2+1P	3	35,000	SD: 5.91 x 9.02 ft
A2b	2+1P	30	35,000	SD: 5.91 x 9.02 ft
A3	2+2P	3	78,000	SD: 8.53 x 9.84 ft
A4	2+4P	37	328,000	SD: 8.86 x 9.84 ft
A5	1+2P	-4	214,000	RRD: 7.87 x 10.17 ft
A6	1+2P	-4	424,000	RRD: 7.87 x 10.83 ft
A7	1+1P	5	30,000	SD: 7.87 x 10.0 ft

Source: (Cleland et al. 2004)

A literature review by Cleland presents research performed by others and recommendations by ASHRAE. Tamm (Tamm 1965) developed a theoretical equation (Equation 40) for infiltration which was modified by Gosney and Olama (Gosney and Olama 1975). Gosney's equation (Equation 18) is what ASHRAE recommends for the infiltration calculation (ASHRAE 2002). Longdill tested a 3.9 ft wide by 5.2 ft high doorway in a 6250 ft^3 cold room to validate Tamm's equation. Longdill, Pham and Oliver (Pham and Oliver 1983), and Downing and Meffert (Downing and Meffert 1993) defined air curtains to be 49 to 83% effective. Strip curtain effectiveness ranges from 86 to 96% with no traffic (Downing and Meffert 1993; Hendrix et al. 1989; Pham and Oliver 1983) and 82 to 92% with one forklift passage per minute (Downing and Meffert 1993). For non-protected doorways infiltration is decreased by traffic by 21 ± 10% (Downing and Meffert 1993). Downing et al. found that doors with strip curtain protection had an increase in infiltration of 32 ± 45% with forklift traffic.

The benefits of the rapid-roll doors are the focus of this study. Rapid-roll doors shorten the interval that the door is open for each opening event through the use of magnetic sensor automation to detect forklift movements. In order to achieve this quick action, doors are constructed of light materials, and do not have a good thermal resistance value. Like any doorway, they are susceptible to leakage through cracks around the doorway. Downing and Meffert found these doors to be 93% effective when closed and 79 to 85% effective with one traffic entry and exit per minute. The doors analyzed in Downing and Meffert's study were fully open for 8 to 20 seconds, taking 1 to 2 seconds to open/close. The doors analyzed in Cleland's study have similar operating characteristics. A5 takes 1.7 seconds to open, is open for 15.7 to 17.0 seconds, and then closes in 3.1

seconds. Door A6 takes 2.9 seconds to open, is open for 8.8 to 15.4 seconds, and then closes in 4.9 seconds. There is a transient period of air flow as the door is opened, but the effect of this lag period is assumed insignificant because the door is open longer than 10 seconds.

Slow-acting sliding doors, utilizing plastic strip curtains, were also studied. The time to open or close this type door is between 4 and 10 seconds, with a fully open time of 60 seconds on average. These doors, like rapid-roll doors, are susceptible to poor sealing. Due to installation methods, the sliding doors in this test had a ½” gap that was constantly allowing warm air to enter the space. Forklift movements studied ranged from 1 to 70 movements per hour.

The effectiveness of different protective devices is summarized below and in Table 20. These results correlate well with previous work by others:

- Strip curtain: 92% effectiveness
- Strip curtain with one strip missing: 80%
- Strip curtain with small gaps between strips: 87%

Table 20. Measured Airtightness, Door Flow Factor, Strip-Curtain Effectiveness and Air Infiltration Due to Forklift Traffic

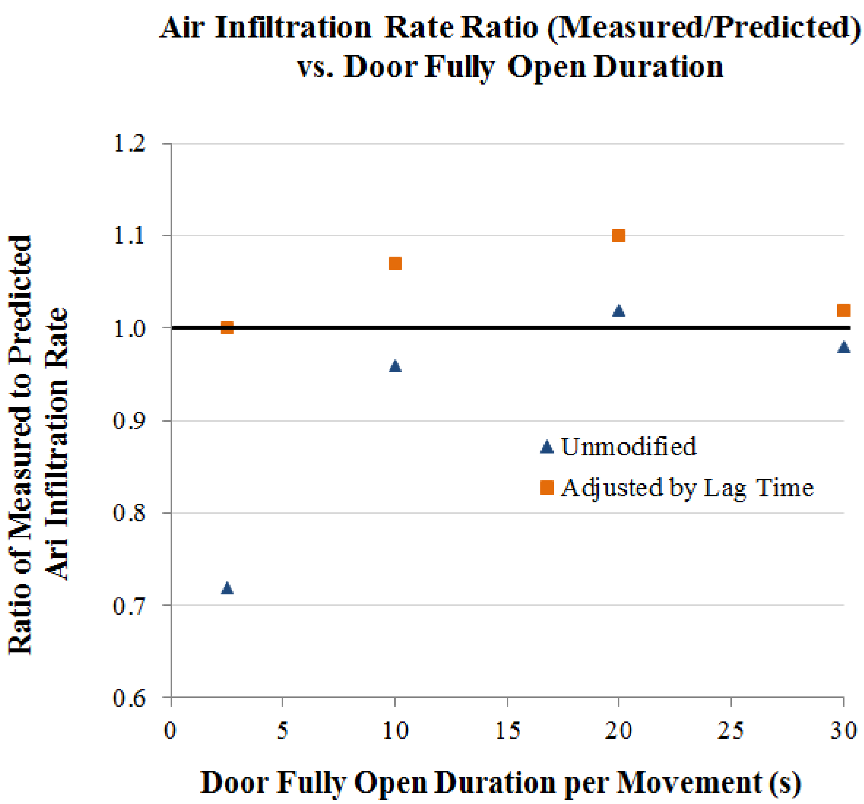
Site	T_C (°F)	ΔT (°F)	$\dot{V}_{AT}/L$ (cfm/ft)	D_f	E	V_t (ft^3/mov)	Std Dev Q_f	Conditions
A1	31.3	32.6	0.3	0.97				No traffic without strip curtain
A2a	5.7	59.0	0.5			1260	410	Open door with strip curtain
A2b	30.2	40.0	0.4			620	150	Open door with strip curtain
A3	2.8	55.6	0.3	0.88	0.87	1530	180	Open door, small gaps in strip curtain, enclosed door
A4	37.9	27.7	0.3			550	180	Open door with strip curtain
A5	-1.1	66.2	1.8	1.27	0.92	1500	170	With strip curtain
						-1910	630	Without strip curtain
A6	-7.2	76.1	2.6	1.31	0.80	2390	180	0.6 in gap in door seal, one strip missing in strip curtain
			1.9					Seal gap covered w/strip curtain
A7	7.0	58.1	1.0			1300	220	0.8 in door seal gap, strip curt
						1460	240	0.8 in door seal gap, open door, with strip curtain
			0.8					Seal gap covered w/strip curtain

Source: (Cleland et al. 2004)

In the development of a modified model for infiltration, door flow factor, fractional door open time, and traffic effects were discussed. Door flow factors are quite variable depending on the type of door. A value between 0.8 and 1.1, in agreement with ASHRAE (ASHRAE 2002), is suggested. A fractional door open time factor (Equation 62) was applied to the infiltration calculation for doors that are repeatedly opened and closed. Results analyzing the measured and predicted air flow rates are shown in Figure 4. This data suggests that for doors open 10 seconds or more, the air flow is fully developed. Hendrix et al. (Hendrix et al. 1989) and Azzouz and Duminil (Azzouz and Duminil 1993) both suggest lag times of less than 3 seconds.

$$D_t = N_{do}\left[\left(t_a + t_c\right)/2 + t_b\right]/t_{tot} \qquad (62)$$

Figure 4. Effect of Door Opening Duration on the Ratio of Measured and Predicted Air Flow with a Rapid-Roll Door without Strip Curtain



Source: (Cleland et al. 2004)

The effect of forklift traffic is not something that has been quantified in equation form other than a door protection modification. Plotting the relationship between infiltration and the temperature difference across the door, Cleland et al. developed an equation to define this relationship (Equation 63). This equation applies to doors similar to the size tested with forklift traffic in this study (7.87 ft wide x 10.0 to 10.83 ft high) with fixed strip curtains. One forklift movement is equivalent to an entry and exit, not just a single passage and each forklift movement is with a pallet measuring 3.6 x 3.3 wide x 6.6 ft tall. Forklift size and speed were not documented. Equation 63 was developed based on data from a sliding door, but no significant difference was noted between doors kept permanently open between forklift movements and those controlled by pull-cord, so potentially this could be applied to other door types.

$$V_t = C_t \Delta T^{1.76} \tag{63}$$

$$\dot{V}_{\text{inf}} = \dot{V}_{AT}\left(1 - D_t\right) + \dot{V}_u D_f D_t \left(1 - E\right) + \dot{V}_{tr} \tag{64}$$

$$\dot{V}_{tr} = \frac{V_t N_t}{60 \frac{\min}{hr}} \tag{65}$$

$$\dot{V}_u = 0.221 \cdot WH\left(gH\right)^{1/2}\left(1 - s\right)^{1/2}\left(\frac{2}{1 + s^{-1/3}}\right)^{3/2} 60 \frac{s}{\min} \tag{66}$$

The modified infiltration model (Equation 64) was used to compare energy savings for different door types and infiltration protection devices, where the doors open to a loading dock at 41°F and 80% relative humidity. Conclusions include the tradeoffs between infiltration, forklift productivity, and door selection. To maintain forklift productivity (above 40 movements per hour) a fast-acting door with protection would be the best combination. However, at low traffic frequency this type door is hard to justify, as it allows more conduction and crack infiltration than a comparable sliding door. Strip curtains enable significant energy reductions, lowering energy costs by \$0.40/hr for 20 forklift movements/hour, even when a fast-acting door is being used.

Review of Industrial Refrigeration

Twenty-eight articles were reviewed that presented information related directly to the field of industrial refrigeration. Industrial refrigeration in this work refers to refrigeration applied on a large scale, like a refrigerated warehouse or as part of a product's preparation for resale, such as a harvest temperature pull-down facility, blast freezer, or cooling of manufacturing equipment or a data center.

Specific topics include refrigeration system design, general load details, and load information related to lighting, product, and infiltration. While each piece of literature reviewed in this section focuses solely on industrial refrigeration, the majority of the content applies to both industrial and commercial refrigeration, and therefore to walk-in coolers and freezers.

Refrigeration System Design

(Cooper 1973)

Cooper presents information related to general design of cold storages, including selection of compressors, refrigerants (Ammonia, R12, R22, and R502), and handling equipment. Examples are given of cold store applications, including the storage of single farm-crops for four to six months. Application examples include storing seed potatoes, bulbs, grain or rice drying and storage, and fruit ripening of bananas, canned pears, plums, and peaches. Air storage temperature varies for each product type. Pears require 29°F air temperature whereas temperatures up to 50°F can be used for peaches, figs, grapes, avocadoes, and apricots. Some additional storage temperatures include 39 to 41°F for oranges, 46 to 48°F for mangoes, and 55°F for green bananas. Relative humidities should generally be 90% but somewhat higher for leafy vegetables, such as lettuce, cabbage, and celery. A cold store may receive up to 10% of its product storage capacity at one time. As a crop storage facility the refrigeration system must remove a large amount of heat from the product (example: 80°F to 32°F in 3 days).

Typical construction types at the time of this publication included brick and concrete, with typical roof construction of corrugated asbestos on steel-framed trusses. Modern cold stores are single-story. Pallets/bins of product may be stacked 23 to 26 ft high. Room is required above the product to allow airflow. For example, in a room that is 16 ft tall bins should only be stacked up to 13 ft high. An analysis of one example store showed that 64% of the available floor area could be utilized for storage. A typical stacking density is 13.73 lb/ft^3.

Three example stores are discussed, presenting information on construction, operation, and loading of refrigerated stores. First, the article discusses an R-12 refrigerant cold room in a semi-tropical region at an altitude of 2300 to 2600 ft, with a design dry bulb of 95°F. The cold room is to operate at interior temperatures of 29.3 and 46.4°F alternatively for gas ripening of fruit crops such as mangoes or bananas.

An example load calculation was performed for another cold store. The design load was 84 Btu/h-ft^2, including conduction, infiltration, product load (respiratory activity of product), and miscellaneous loads from electric motors and lights. Respiration is typically assumed to be 1/3 latent and 2/3 sensible heat. The equipment was selected based on an 18 hr run period.

Finally, a specification of a refrigeration plant was supplied. It was to use a direct expansion refrigeration system utilizing R-12. The space is composed of two rooms with internal dimensions of 29.5 x 30.5 x 15 ft to hold 220 tons of fruits and vegetables. Defrost is not specified for this system, as interior temperature will be above a 35°F. The exterior conditions are shaded, with a 90°F dry bulb and 70% relative humidity. Ten percent of warm product is added to each room per day. The product is lowered to storage temperature in three days with a 1% weight loss. Air infiltration is assumed to be 12 air changes per day. Internal circulation is greater than 30 air changes each hour. The design load is 67900 Btu/h (75.5 Btu/h-ft^2), maintaining 28.9°F and 90% relative humidity. Equipment temperatures are as follows: 16.7°F evaporator temperature, 131°F condensing temperature. The lighting load is 200 W per room, or 0.22 W/ft^2.

(Krack 1977)

Krack Corporation is at the center of industrial refrigeration. They have a selection guide similar to that provided by Heatcraft for commercial refrigeration called RLE-278 (Krack 1977).

Conduction is analyzed in great detail by this manual. The absolute value used as a heat transfer coefficient is dependent upon the material reflectivity, degree of roughness, attitude (vertical or horizontal), length and the air velocity over the surface. Combining these components, one can obtain the surface film conductance. Surface film conductance for indoor walls may be estimated at 1.60 and 6.0 for outdoor walls not exposed to winds in excess of 15 mph. In practice, the surface film coefficient is often ignored due to its small effect on the outcome of sizing the equipment.

Floor conditions are difficult to approximate for a large industrial facility. While ground temperatures are readily available for numerous locations, their application with a refrigerated space is a complicated matter. Freezers typically have heated floors. This actually simplifies the sizing method. It is usual practice to assume a factor of 1 Btu/ft^2-°F - 24 hrs for freezer floors with conventional insulation.

Commercial coolers and freezers are shielded from solar effects if they are constructed inside of the commercial space. Industrial facilities typically do not have this luxury and the roof of refrigerated volume is exposed to direct sunlight. Krack estimates the load by adding degrees to the temperature differential between the interior box and exterior conditions as defined in Table 21. When the facility is in close proximity to a highly reflective surface, such as sand or water the values noted in Table 21 should be increased by 50%.

Table 21. Solar Radiation Allowance

		°F to be Added to Normal TD			
Luminance	Surface Type	East Wall	South Wall	West Wall	Flat Roof
Dark	Slate Roofing Tar Roofing Black Paints	8	5	8	20
Medium	Unpainted Wood Brick, Red Tile Dark Cement Red, Grey, or Green Paint	6	4	6	15
Light	White Stone Light Colored Cement White Paint	4	2	4	9

Source: (Krack 1977)

Depending on the application, door openings can vary widely according to Krack. When specific data is not available the manual suggests using a 24 hr period air change value corresponding to the size of the refrigerated space (Table 22). For refrigerated spaces with heavy usage, the air change estimate should be doubled. On the other hand, if the space is used for long term storage and there are infrequent door openings, these values should be reduced by up to 40%.

Table 22. Average Air Changes per 24 Hrs for Medium Temperature (Above 32°F) and Low Temperature (Below 32°F) Rooms Due to Infiltration and Door Openings.

	Medium Temp	Low Temp		Medium Temp	Low Temp
Volume (ft^3)	Air Changes/24 hr		Volume (ft^3)	Air Changes/24 hr	
200	44.0		5000	7.2	5.6
250		29.0	6000	6.5	5.0
300	34.5	26.2	8000	5.5	4.3
400	29.5	22.5	10000	4.9	3.8
500	26.0	20.0	15000	3.9	3.0
600	23.0	18.0	20000	3.5	2.6
800	20.0	15.3	25000	3.0	2.3
1000	17.5	13.5	30000	2.7	2.1
1500	14.0	11.0	40000	2.3	1.8
2000	12.0	9.3	50000	2.0	1.6
2500		8.1	75000	1.6	1.3
3000	9.5	7.4	100000	1.4	1.1
4000	8.2	6.3	200000	0.9	

Source: (Krack 1977)

Krack also specifies an alternative, more precise method for determining the load due to infiltration. If the frequency and duration of door opening is known the infiltration load can be determined using the following equation (Equation 67). The constant 527 is derived as displayed, where 24 hr refers to the period for which the equation applies, 0.075 lb/ft^3 is the air density assumed, and 60 min/hr is a conversion. The values 100 ft/min, 7 ft, and 60°F are used in the velocity calculation. According to Krack, the average air velocity in either half of a door 7 feet high at a 60°F temperature differential is 100 ft/min.

$$Q_{Inf/24} = 527\sqrt{H \cdot (T_H - T_C)}(h_H - h_C)F_\tau \cdot \frac{A_{\inf}}{2} \quad (67)$$

$$527\frac{lb}{ft^{5/2}(°F)^{1/2}} = 24hr \cdot 0.075\frac{lb}{ft^3} \cdot 60\frac{\min}{hr} \cdot \frac{100\frac{ft}{\min}}{\sqrt{7ft \cdot 60^\circ F}} \quad (68)$$

To adjust infiltration calculations for altitude of the site where the refrigerated space will be located, Krack assumes that relative humidity is constant, enthalpy and humidity ratio increase 2% and 5% respectively for every 1000 ft increase in altitude. The specific volume for a given dry bulb and humidity ratio is inversely proportional to atmospheric pressure.

Product pull down may occur over a 2 to 24 hour period according to Krack, depending on the application and conditions. Products do not uniformly give up their heat after being introduced to a refrigerated space. Heat is given off to space at a higher rate initially. As the temperature of the product drops, so does heat transfer rate to the surrounding refrigerated space (15 to 25% reduction). Load factors are applied when the product loading is a significant component of the load profile to ensure accurate unit sizing.

Typically refrigerated long term storage facilities will be preceded by a blast freezer or cooler to reduce product temperature prior to its introduction to the space. Facilities that do not have this luxury must respond to product pull-down while also operating at a steady long term temperature. Krack recommends flooded and recirculated refrigerant systems to allow for wide control variance of the evaporator TD.

Products in medium temperature environments respire throughout the period when they are stored. Krack provides an extensive list of perishable product storage data. Any moisture lost by the product reduces its quality. To reduce respiration and prolong the storage life of products, storage rooms should be controlled within 1°F. Meat and fish have no continuing life process and therefore generate no heat in storage.

Supplemental loads may include motors, heaters, lights, people, forklifts, and other miscellaneous heat sources. Occupancy load is a function of the number of people in space, the work being performed, and the room temperature. As a good estimate, Krack suggests the values in Table 23 to estimate the load added to the space by each person in the space. For short duration occupancy the heat load should be increased by 20%.

Table 23. Occupancy Heat.

Room Temperature (°F)	Heat per Person Btu/24 hr
50	17300
40	20200
30	22800
20	25200
10	28800
0	31200
-10	33600

Source: (Krack 1977)

Typically, lighting is 1 to 1.5 W/ft^2 for storage areas. If the space is used for active working the lighting load is larger, at 2.5 to 3 W/ft^2. Forklifts may be estimated at 4 to 5hp.

Most industrial facilities need defrost to remove frozen condensate from the evaporator coils. Krack assumes 25% of the heat used during the defrost cycle is added to the space whereas the remainder of the heat is taken away in the coil condensate. Defrost cycles vary based on the application. A defrost cycle explicitly stated by Krack occurs 4 times a day at 15 to 20 minute duration (beef carcass chilling).

Safety factor correction of 5 to 10% to the calculated hourly load is suggested. Numerous examples are presented for calculating refrigeration loads, including blast freezers, fruit chilling and storage, beef carcass chilling, beer cooler and storage, nut storage, distribution centers, milk cooler, and ice cream hardening freezer. A number of these examples are applicable to an analysis of the model load profile for walk-in coolers and freezers, as they are smaller than 3000 ft^2 in plan area. Information presented includes the temperatures and conditions surrounding and inside the refrigerated space; the box construction; refrigeration loads, including product load, occupancy load, equipment load, lighting load, and infiltration load; and equipment operation details, such as defrost cycling and load contribution, and evaporator fan motor load. Details on footprint, box height, and door size for spaces larger than what would qualify as a walk-in refrigerated space were excluded from the model load profile analysis.

Evaporator operation is selected around the desired operating conditions of the box. Walk-ins with meats should have low air flows and moderate evaporator TDs ranging from 10-12°F. If the product must be pulled down from a relatively high initial temperature (50 to 70°F pull-down) the coil TD will approach 18-20°F initially and drop with the temperature of the product. Storage rooms for nuts need a lower relative humidity (65%) than does a meat storage facility (85%). As a result, a higher evaporator TD should be utilized at around 12°F.

When sizing equipment, the frequency and duration of defrost required will affect how closely the equipment size can align with the refrigeration load. Spaces with coil temperatures above 32°F will have no frost accumulation and can be sized for a compressor runtime of 24 hrs at peak load. Medium temperature refrigerated spaces with positive defrost systems can have a compressor runtime of 20 to 22 hours, as defrost will be necessary for 2 to 4 hours of the day. Low temperature refrigerated spaces can typically be sized for 18 hr operation. Off cycle defrost is utilized for space temperature of 35°F or higher (assuming that the coil temperature is below 32°F). This type system can be sized for 16 hr operation, as 8 hours will be required in off cycle to defrost the coil.

Typical refrigeration loads were specified as a function of room size. Small beef carcass chilling rooms (appropriate size to be considered a walk-in cooler) usually operate at 300 Btu/hr-ft^2 (range of 288 to 465 Btu/hr-ft^2) whereas large industrial facilities operate from 171 to 184 Btu/hr-ft^2. From reviewing the information in the Krack manual it is apparent that this is the trend, the larger the facility is the lower the refrigeration ton/ft^2 requirement. Pork chilling rooms have similar load requirements, but, from analysis of the information presented by Krack, they should be upsized (relative to beef carcass rooms of the same size) by 10-20%.

Krack presents typical room load less product loading at 95°F ambient corresponding to given footprints in their manual. They also give typical product loads for walk-in coolers and freezers. These were combined to analyze typical loads for walk-ins. The loads for coolers range from 30.4 (1600 ft^2 with 8 ft ceiling at 36°F interior box temperature) to 145.3 Btu/hr-ft^2 (36 ft^2 with 10 ft ceiling at 28°F interior box temperature). The average walk-in cooler load was 62.6 Btu/hr-ft^2. Walk-in freezer loads ranged from 30.6 (1600 ft^2 with 8 ft ceiling at -20°F interior box temperature) to 169.6 Btu/hr-ft^2 (36 ft^2 with 10 ft ceiling at -10°F interior box temperature), with an average load of 69.3 Btu/hr-ft^2.

Krack presents a table of heat gain factors for common insulating and building materials. The K factors (Btu/hr-ft^2-in-°F) associated with the insulating materials are listed in Table 24.

Table 24. Insulating K-Factors.

Material	K-factor (Btu/hr-ft^2-in-°F)
Foamglass	0.380
Corkboard	0.300
Expanded Polystyrene or Fiberglass	0.240
Extruded Polystyrene	0.185
Slab Urethane	0.160
Foamed in-Place Urethane Panels	0.130

Source: (Krack 1977)

(Stoecker 1998)

The *Industrial Refrigeration Handbook* by Stoecker (Stoecker 1998) covers a wealth of information related to warehouses, blast cooling/freezing, hydrocooling, vacuum cooling, cheese aging, beer and wine fermentation and aging. Of interest to this study was information relevant to building a model load profile for the typical commercial application of walk-in coolers and freezers.

The preservation of the food being processed and/or stored is very important to those in the industrial refrigeration field. Typically, increasing the product's life by improving its storage conditions comes at significant capital and reoccurring costs. Significant research has been done on optimum storage temperature, pull-down time, and storage temperature range to attempt to reduce any money wasted on preserving the product.

The proper storage temperature for different foods has been developed by those in the industry over a number of years. Taking advantage of some products' thermal properties allows for improved storage life. Some fruits for example, may be stored slightly below 32°F without freezing due to the fruits' sugar content. Some meats and poultry products are best stored at temperatures around 30°F to take advantage of similar properties. Some recommended unfrozen storage temperatures are provided in Table 25. Additional interior condition examples include docks at 39 to 46°F and meat processing rooms at 50 to 57°F. When low relative humidities are required, as is the case with seed storage, cooling and dehumidification is performed with a moderate refrigerant temperature and then the air is reheated to the load-ratio line of 50% relative humidity. Reheat coils may also be needed in locations were the outdoor temperature drops, resulting in infrequent evaporator operation. To keep the relative humidity from rising, the coils cool the space to condense water out of the air, followed by a defrost cycle.

Table 25. Recommended Storage Temperatures for Several Unfrozen Food Products

Product	Storage Temperature (°F)	Relative Humidity (%)	Storage Life
Apples (most varieties)	30 to 37	90 to 95	3 to 8 mnths
Avocados	38 to 40		
Bananas	54 to 58	90 to 95	
Beans, Green	40 to 45	90 to 95	7 to 10 days
Broccoli	32	95 to 100	10 to 14 days
Cabbage	32		
Cheese	32 to 50		
Dough, storage	37 to 41		
Dough, proofing	80		
Honeydew melon	41 to 50	90 to 95	2 to 3 wks
Lettuce	32 to 34	95 to 100	2 to 3 wks
Milk	33 to 40		
Peaches	29 to 31	90 to 95	2 to 4 wks
Pears	29 to 31	90 to 95	2 to 7 mnths
Peas, Green	32	95 to 98	1 to 3 wks
Potatoes, early	37 to 39	90 to 95	4 to 5 mnths
Poultry	30 to 35		
Seeds	32 to 50	50 to 65	10 to 12 mnths
Strawberries	31 to 32	90 to 95	5 to 7 days
Tomatoes	38 to 50	90 to 95	4 to 7 days

Source: (Stoecker 1998)

Foods can be frozen in several minutes or several hours with today's methods. This reduces ice crystals on the interior of the product, improving what the end customer receives. Methods include air blast freezing, contact freezing, immersion freezing, belt freezers, spiral freezers, and cryogenic freezing using liquid carbon dioxide or nitrogen. Frozen storage temperatures are typically between -10 and -5°F with -10°F cited as the most frequent. The lower the temperature the better, from a product quality point of view. This comes at an operating and capital cost, however, so warehouses do not want to cool excessively. Bakery goods are stored at 5 to 14°F due to their low moisture content and ice cream and fish are stored at or below -20°F.

Chilling facilities used to pull-down and store beef and pork carcasses remove 58 to 65°F in one night. Birds 4 pounds or less are cooled by 40°F in 4 hours or less. Birds less than 6 lbs in weight must be cooled within 6 hrs whereas any larger birds must be cooled within 8 hrs. Products lose water during processing and storage due to respiration. Carcasses lose about 1% of their weight during processing.

A cooler's product load is composed of the required reduction in temperature and the removal of the heat of respiration. Refer to Table 26 for information for product heat of respiration cited at various interior box temperature conditions. Freezers have three components to their product load, including reducing the temperature of the product to the freezing point, removing the latent heat of fusion, and reducing the product temperature to the storage temperature.

Table 26. Heat of Respiration of Several Unfrozen Fruits and Vegetables

	Heat of Respiration (Btu/lb-day)			
Product	32°F	41°F	59°F	68°F
Apples	0.253 to 0.451	0.559 to 0.792	1.5 to 3.4	1.86 to 3.85
Beans	1.95 to 3.85	3.21 to 6.72		23.3 to 29.8
Celery	0.793	1.21	4.11	7.11
Lettuce	1.00 to 1.86	1.46 to 2.2	4.25 to 4.5	6.61
Peaches	0.451 to 0.703	0.702 to 1.01	3.66 to 4.67	6.52 to 11.3
Pears	5.21 to 8.32	8.72 to 10.7		38.4
Strawberries	1.35 to 1.95	1.8 to 3.66	7.82 to 10.2	11.3 to 21.6

Source: (Stoecker 1998)

Table 27. Specific Heats and Latent Heats of Several Food Products

Product	Water Content (% mass)	High Freezing Point (°F)	Specific Heat Above Freezing (Btu/lb)	Specific Heat Below Freezing (Btu/lb)	Latent Heat of Freezing (Btu/lb)
Apples	84	30	0.902	0.453	121
Chicken	74	27	0.843	0.423	107
Peas	74	31	0.792	0.423	107
Ham	56	29	0.735	0.368	81
Salmon	64	28	0.783	0.392	92
Sirloin beef	56		0.735	0.368	81
Strawberries	90	31	0.938	0.471	130

Source: (Stoecker 1998)

Product loading varies considerably for each site. A warehouse storing seasonal products has a moderate turnover of 8 to 12 per year whereas grocery distribution sees a turnover of 30 per year. In an effort to reduce inventories and increase the shelf life at the supermarket, distribution centers continue to increase the turnover rate up to 100. Typically in storage facilities, the product load is small, as it is either delivered at the storage temperature or precooled in an adjacent facility.

Conduction is a significant component of the load for any refrigerated space. Building materials (Table 28, Table 29) and the insulation thickness are an economics and application exercise, to determine what makes the most sense for the life of the facility. The most popular method for insulating is the use of prefabricated insulating materials which connect together with tongue and groove, spanning up to 30 ft (Table 31). Convection coefficients applicable for various surface orientations, wind direction, and heat flow directions are listed in Table 30.

Table 28. Thermal Properties of Some Insulating Materials

Material	Conductivity (Btu/hr-ft-°F)	Dens. (lb/ft^3)	Rel. Water Vapor Permeability	Flame Spread	Compressive Strength (psi)	Rel. Cost
Cellular foamglass	0.029	8.5	0	0	100	High
Cellular polyurethane	0.013	2.0	Med	30-75	25	Med
Expanded polystyrene	0.020	1.0	Med	If melted burns	12-17	Low
Extruded polystyrene	0.015	5.4	Med	5-15	8-40	Med
Glass fiber	0.021	2.3	High	15-20	N/A	Low
Polyisocyanurate	0.012	2.0	Med	25	20-30	Med

Source: (Stoecker 1998)

Table 29. Conductance and Resistance of Some Materials

Material	Conductance (Btu/hr-ft^2-°F)	Resistance (hr-ft^2-°F/Btu)
8 in cinder concrete block	0.58	1.72
Hollow clay tile, 2 cells	0.66	1.51
Built-up roofing, 0.394 in	3	0.33
Asphalt roll roofing	6.5	0.154

Source: (Stoecker 1998)

Table 30. Convection Coefficients for Various Surface Orientations and Air Flow Conditions

Orientation and Condition	Convection Coefficient (Btu/h- ft^2-°F)
15 mph, winter, any orientation	6
7.5 mph, summer, any orientation	4
Still air:	
Vertical surface, heat flow horizontal	1.46
Horizontal surface, heat flow upward	1.63
Horizontal surface, heat flow downward	1.08

Source: (Stoecker 1998)

Table 31. Thermal Resistances and Recommended Applications of Prefabricated Wall and Ceiling Panels

Thickness (in)	Thermal Resistance (h- ft^2-°F/Btu)	Recommended Temperature Application
2	16.6	Ambient
3	20.8	Down to 32°F
4	33.3	Down to -20°F
5	41.6	Down to -50°F
6	50.0	Down to -70°F

Source: (Stoecker 1998)

In determining what the appropriate model load profile is, it is important to understand which ambient conditions are the most representative of the design point. Stoecker specifies the following locations as typical design points:

- Atlanta: 94°F summer 1% dry bulb (DB), 74°F coincident wet bulb (WB), 78°F summer 1% WB
- Chicago: 94°F summer 1% DB, 75°F coincident WB, 78°F summer 1% WB
- Dallas: 102°F summer 1% DB, 75°F coincident WB, 78°F summer 1% WB

- Los Angeles: 94°F summer 1% DB, 70°F coincident WB, 72°F summer 1% WB
- New York: 92°F summer 1% DB, 74°F coincident WB, 76°F summer 1% WB

Ground temperature is a very important component in determining an appropriate model load. Stoecker states that the sub-floor temperature is typically between 41 and 50°F for refrigerated facilities. Heating is required to maintain 50°F so that the facility does not freeze and heave up. A glycol recirculating loop is the most prevalent system used in industry. The supply temperature needs to be no higher than 50 to 60°F. It is typically assumed that the sub-floor temperature is 50°F or that the floor heating load is about 2 Btu/h-ft^2.

During the day, sun is radiating on the surfaces that form the exterior of the refrigerated storage facility. Depending on the exterior material properties, the temperature of a roof may rise to 120 to 130°F during the day.

Infiltration is a well analyzed component of the refrigeration load. Gosney Olama is the most widely used air infiltration approximation method utilized by industrial refrigeration professionals (Equation 18). One method of determining the infiltration to a space is to take the expected turnover rate and apply an expected door open duration for each product movement. A typical door open time may be assumed at 10 to 15 seconds. Reductions to the infiltration to a space may be achieved by applying air curtains or plastic strip curtains to the opening. Air curtains can remove 65 to 80% of infiltration load compared to an open door (Hayes and Stoecker 1969). Plastic strip curtains are 90% effective when there is no traffic and 85% effective with traffic.

Internal loads include lighting, occupancy, and equipment loads. Lighting levels vary about as much as product loads, depending on the application. Meat processing rooms require high lighting levels to ensure employee safety at 1.5 W/ft^2. Warehouses have much lower light intensity at 0.48 W/ft^2. Additional internal loads for meat processing rooms are also high, as there is a heavy concentration of workers (Table 32) and electrical equipment operating. Warehouses have a lower concentration of employees, but they do have forklift loads.

Table 32. Heat Equivalence of Occupants

Space temp (°F)	Heat rate (Btu/hr-person)
50	720
32	920
14	1,130
-4	1,330

Source: (Stoecker 1998)

Defrost is needed on any coil which operates at a temperature at or below 32°F. Defrost adds refrigeration load to the space (4% of the total load in the example presented in the text), but it is required to effectively cool the space. Defrost efficiency is defined as the ratio of the heat to melt the frost off the coil to the heat provided to the system.

Methods of defrost discussed by Stoecker include air, electric, water, and hot gas. In medium temperature applications, where the coil is still below 32°F, air defrost may be utilized. The refrigeration system shuts off for a period of time to allow the space to heat and defrost the coil. Typically defrosting the coil in this manner requires a significant amount of time, unless the fan continues to operate. Another method used is to insulate the coil and then bring in a supply of warm air from another source to quickly melt the frost on the coil.

Electric defrost is common in small commercial applications due to its low first cost and simplicity. The operational cost for this method is the highest compared to the other alternatives, so large applications typically do not utilize it.

Water defrost is the second most popular industrial defrosting method. To defrost the evaporator coil, hot water is sprayed on the coils. This method is utilized when the refrigeration system does not have enough evaporators to utilize hot gas. The ideal application of water defrost is in a spiral freezer chamber, operating in a production mode.

Hot gas is the most efficient (typically 20%) and most widely used defrost method. The defrost cycles are very quick using this method, allowing for a higher daily compressor runtime and, therefore, smaller equipment selections. During a hot gas defrost cycle the evaporator valves are switched, so that flow to the evaporator coil from the liquid header is shut off, and compressor discharge is brought in through the evaporator discharge line to heat the coil.

R22 and R502 are the most widely used refrigerants but are being phased out. Ammonia is also widely used but highly toxic. Packaged refrigeration units which contain the evaporator, compressor, and condenser typically use halocarbon refrigerant and have air-cooled condensers. Packaged units are popular in refrigerated spaces of less than about 10,000 ft^2 in area and in central plants with areas greater than 20,000 ft^2. Selecting a compressor capable of delivering full capacity is the lowest first cost choice. Using multiple compressors to split the load offers lower energy usage, as compressors can drop off as the load diminishes, and improved flexibility. If one compressor goes down in a system with ten compressors, the system can still maintain 90% of its cooling capacity until the damaged compressor can be replaced or repaired.

Evaporator and condenser fans are heavy energy users in refrigerated systems. They use 30% of total energy use over a year (10-15% of full load power). The mechanical components in a refrigeration system offer waste heat that can sometimes be utilized for other purposes within the overall system. Examples of heat sources in a two stage refrigeration plant are listed in Table 33, with potential uses listed in Table 34.

Table 33. Typical Temperatures of the Heat Sources in a Two-Stage Refrigeration Plant

Source	Type of Compressor	Temperature (°F)
Water from Head Cooling	Reciprocating	85 to 95
Oil to be Cooled, 1st stage	Screw	140 to 150
Oil to be Cooled, 2nd stage	Screw	150 to 170
Desuperheating, 1st stage	Reciprocating	175 to 195
Desuperheating, 2nd stage	Reciprocating	200 to 240
Desuperheating, 1st stage	Screw	120 to 130
Desuperheating, 2nd stage	Screw	150 to 170
Condensing refrigerant	--	60 to 100

Source: (Stoecker 1998)

Table 34. Waste Heat Applications and Temperature Levels

Potential Use	Min Refrigerant Temperature (°F)
Underfloor heating	50 to 60
Water defrost	72 to 80
Space heating in winter	80 to 110
Humidity control reheat coils	90 to 100
Process use preheat water	110 to 120

Source: (Stoecker 1998)

Refrigeration loads relative to their application are compared in Table 35. It should be noted that these values seem low compared to those presented by others. That may be due to the fact that industrial facilities are much larger in size than the walk-in facilities that are the focus of this thesis.

Table 35. Refrigeration Load for Several Applications

Type of Space	Refrigeration Load (Btu/ft^3-hr)
Frozen food storage	0.72 to 1.44
High-rise freezer storage	0.24 to 0.72
Produce storage	0.96 to 1.32
Shipping dock	1.32 to 2.40
Process area	1.92 to 6.00

Source: (Stoecker 1998)

(Zhang and Groll 2005)

Zhang conducted a survey related to refrigerated warehouses. He reached out to industry with questions related to the size of their facility (Table 36), number of compressor stages (Table 38), refrigerant (Table 37), intercooler (Table 39) or economizer type/use, control scheme and equipment, number of staff and their duty time in machine rooms, compressor type, compressor speed control, and evaporator type.

Table 36. Physical Sizes of 62 Public Refrigerated Warehouses

Volume (x 10^6 ft^3)	Plant Number
0.36 to 0.99	11
1.00 to 1.99	7
2.00 to 3.99	21
4.00 to 5.99	8
6.00 to 7.99	8
8.00 to 9.99	2
10.0 to 14.4	5

Source: (Zhang and Groll 2005)

Table 37. Refrigerants Used in Public Refrigerated Warehouses

Refrigerants	Responses to Questionnaire	1998 Directory of IARW
R-717	58	306
R-12	1	44
R-22	5	0
R-134a	1	0
R-404a	1	1
R-502	0	1
R-507	1	0
Total	67	352

Sources: (IARW 1999; Zhang and Groll 2005)

Table 38. Compression Stages in 62 Public Refrigerated Warehouses

	Number of Plants	Percentage (%)
Plants using single stage only	12	19.7
Plants using two stage only	36	59.0
Plants using both two stage and single stage	13	21.3
Plants that did not answer to compression stage	1	

Source: (Zhang and Groll 2005)

Table 39. Types of Intercoolers in 49 Public Refrigerated Warehouses

	Number of Plants	Percentage (%)
Plants with shell-and-coil intercoolers only	24	61.5
Plants with flash-type intercoolers only	15	38.5
Plants with both kinds of intercoolers	6	
Plants with other intercoolers	6	

Source: (Zhang and Groll 2005)

Control types utilized by the 62 plants include computer integral control (40 plants), partial auto-control using no computer (25 plants), and unknown (2 plants). Automatic control methods are known to reduce plant energy usage by up to 40% (Zhang et al. 1999). Seventeen plants had one staff in the machine room, 12 had two individuals, and 26 plants has three staff working in the machine room. The duty time associated with the machine room personnel varied from daytime during the week only (37 plants) to 24 hours a day for (16 plants).

Compressor types varied between piston and screw type. Twelve plants used only piston type, 23 used screw compressors only, and 27 plants utilized both piston and screw compressors in their operation. A total of 226 piston compressors were reported along with 213 screw compressors. Of the piston compressors, 25 of them were semi-hermetic, 30 were integral 2-stage, 3 operated with a VFD, and 7 used a microprocessor for speed control. Of the screw compressors reported, 8 screw compressor were semi-hermetic, 69 were single screw, 73 operated with an economizer, 142 used variable volume control, 16 operated with a VFD, and 185 utilized a microprocessor.

(Stegmann 2000)

Low temperature refrigeration means different things to different industries. In commercial refrigeration, low temperature refers to operating conditions below 32°F whereas in air conditioning, chilled water or glycol temperatures around 32°F are considered low temperature. When Stegmann states low temperature in this article it refers to temperatures lower than those seen in industrial refrigeration and above cryogenic temperatures, falling in the range of -58 to -148°F. Applications include food, pharmaceutical, chemical processing, laboratory environmental chambers, and thermal storage equipment.

Compression schemes for these systems include stock hermetic compressors in an autocascade cycle, single-stage economized screw, two-stage single refrigerant systems utilizing either screw or reciprocating compressors, and two or three-circuit cascade systems. Auto cascade systems are self-contained systems with multiple stages of cooling occurring simultaneously using a specific refrigerant mixture. Single-stage economized screw compressor systems are the simplest low temperature system, operating down to about -60°F. Two-stage single refrigerant system limitations depend on the refrigerants chosen (Table 40). Two-circuit cascade systems are the most common and suitable all the way down to -148°F. Three-circuit cascade systems require an additional temperature reduction to meet certain design conditions. These systems are rare.

Table 40. Minimum Low Temperatures for Two-stage Single Refrigerant Systems

Refrigerant	Min Low Temperature (°F)
HCFC-22	-90
R-507	-90
R-717	-60

Source: (Stegmann 2000)

(USDA 2010)

Every two years the United States Department of Agriculture (USDA) conducts a survey of refrigerated warehouses. The primary information gathered includes the crop, warehouse capacity, warehouse usable space, and whether it is a public or private entity. This information is broken down for analysis based on the location and the other parameters aforementioned. Working space, chill rooms, and curing rooms in meat storage facilities are not included in the storage statistics.

Public general storages are facilities where individuals store food and then pay the owner a certain rate. Apple and pear storages are refrigerated facilities that are only used to store those products. Controlled atmosphere is used to improve the storage duration for some products by regulating the oxygen and carbon dioxide in the environment. Coolers are defined by the study as operating from 0 to 50°F with freezers being any space operating below 0°F. This cooler vs. freezer temperature definition is in stark contrast to the industry standard of 32°F.

Refrigerated storage is rising at a tremendous rate, primarily due to new construction, growing 14% from 2007 to 2009. The five states with the largest refrigerated warehouse capacities (x 10^6 ft^3) include California with 495, Florida at 274, Pennsylvania at 227, Georgia with 218, and Texas with 198. Refrigerated warehouse average usable space is about 82% of the gross space, with freezers comprising about 77% of the total usable space reported for coolers and freezers.

General Load Details

(Kun et al. 2007)

This report analyzes heat reclaim on a Chinese refrigerated warehouse. The ambient temperature specified is 122°F with storage conditions, including 5, 64.4, and 71.6°F.

(Ashby et al. 1979)

Energy can be purchased at a lower cost during off peak hours. Because industrial refrigerated spaces use a significant amount of electricity it is advantageous for these facilities to utilize energy predominantly during off-peak hours. This article discusses the opportunity to save money by operating at two temperatures over the course of the day. During off peak hours the refrigeration system would work extra to pull the storage room temperature down below the typical set point. This way during higher energy rate portions of the day, the refrigeration system is not required to remove as much heat from the space.

Drawbacks to operating in this manner include increases in conduction and infiltration loading due to a larger temperature differential between the box temperature and the exterior temperature, increased equipment capacity and control requirements, and potential damage to the stored product.

Ashby et al. analyzed box temperature variation for a storage freezer containing packaged okra, peas, and strawberries. A control and three different temperature variations were compared. Comparisons were made in regard to energy consumption and product quality.

Details pertinent to developing a model load profile include information presented on typical freezer storage conditions and box construction. The typical freezer storage condition used as the control for this study was -9.4°F. The walk-in boxes utilized in this study had walls constructed of an 0.7" air gap formed from 24 gage aluminum foil fastened to horizontal wood strips, ceiling air spaces of 1.6", and a floor insulated by foam plastic. Thermal conductance values for the box are listed below:

- Walls: 0.041 Btu/h-ft^2-°F
- Ceiling: 0.032 Btu/h-ft^2-°F
- Floor: 0.042 Btu/h-ft^2-°F

(Cleland 1983)

Cleland develops a simulation model for design of a meat processing plant with several independent loads (Cleland 1983). He cites work by Marshall and James who simulated a two-stage compression ammonia plant (Marshall and James 1975) and his previous development of a model for freezing fish on a fishing boat (Cleland et al. 1982). To reduce the complexity of the model, Cleland assumed that equipment with rapid response could be modeled with steady state equations. Differential equations were only used for equipment with long time constants.

The meat processing plant that is the focus of this article has five types of refrigeration applications, including cutting rooms (air temperatures from 46 to 50°F), beef chillers (air temperatures from 32 to 41°F), cold stores (air temperatures from -4 to 5°F), lamb carcass freezers (air temperatures from -22 to -4°F), and beef carton freezers (air temperatures from -40 to -13°F).

The door open percentage was assumed to be 3% with a randomly generated schedule. Product freezing was over a 24 hour period.

(Navy 1986)

This design manual captures design criteria for Naval specification of cold storage facilities (Navy 1986). It is written for facilities larger than what would be considered walk-ins but a good portion of the information is transferrable.

This manual presents design conditions for a wealth of locations. The Navy uses the dry bulb temperature from 2.5 percent column and the wet bulb temperature from the 5 percent column in design of refrigerated storage facilities.

All facilities are specified to require some type of slab freeze protection. For facilities 5000 ft^2 or less and a winter design dry bulb temperature of greater than 32°F, either ventilation pipes or heated ethylene glycol solution piping below slab are used. Facilities larger than 5000 ft^2 and facilities with winter design temperature below freezing use heated ethylene glycol in circulated pipes.

Typical storage conditions are given in Table 41, corresponding to various food storage applications.

Table 41. Typical Storage Conditions for Various Applications

Room	Design Temperature (°F)	Relative Humidity (%)	Product Stored
Meat	32 to 34	85	Fresh meats, fresh fish, chicken, short term storage
Dairy	40	80	Cheese, eggs, butter
Perishable Foods	35 to 40	90	Fruits, vegetables, with storage temperatures of 30 to 40°F
Film (less than 1 yr)	45 to 55	40	Unprocessed photographic film
Film (long term)	-10 to 0	85	New film in sealed package
Freezer (short term)	0	90	Frozen package goods
Freezer (long term)	-20 to -10	90 to 95	Frozen products: meat, fish, poultry, ice cream, vegetables, etc.
Dry Storage	50	55 to 65	Fruits, vegetables and miscellaneous products with storage temperatures of 45 to 60°F
Ice	25	--	Bagged ice
Medical (cooler)	40	50	Antibiotics, blood vaccines
Medical (frozen)	-4	85	Bone and blood
Environmental	-55 to high T	--	Altitude chambers

Source: (Navy 1986)

Typical thermal properties that can be used to calculate heat transfer are included in Table 42. Some recommended R-values for different applications are noted in Table 43. One additional item that is specified, is the reduced insulating value of wood studs used to support insulation. The manual suggests a 10% increase in the U-value of the walls. Solar loads are calculated according to the 1977 ASHRAE method in Chapters 25 and 27 of the Fundamentals (ASHRAE 1977).

Table 42. Typical "U" Factors for Walls and Roofs

Insulation Thickness (in)	U-Value (Btu/hf-ft2-°F)				
	Polyurethane	Polystyrene		Fiber Glass	Cellular Glass
	k = 0.14	k = 0.20	k = 0.25	k = 0.23	k = 0.16
2	0.069	0.083	0.100	0.093	0.132
3	0.048	0.059	0.071	0.066	0.097
4	0.037	0.045	0.056	0.052	0.076
5	0.030	0.037	0.045	0.042	0.063
6	0.025	0.031	0.038	0.036	0.054

Source: (Navy 1986)

Table 43. Recommended R-Values for Various Applications

Type of Facility	Temperature Range (°F)	R-Values (hr-ft^2-°F/Btu)		
		Floors	Walls/ Suspended Ceilings	Roofs
Cooler	40 to 50	Perimeter insulation only	17 to 30	25 to 35
Chill Cooler	28 to 35	15 to 20	25 to 35	30 to 40
Holding Freezer	-10 to 20	30 to 35	35 to 40	40 to 50
Blast Freezer	-40	35 to 40	50 to 60	50 to 60

Source: (Navy 1986)

Product loading is a function of the weight of the product and the difference between the initial temperature of the product being loaded and the temperature of the space. An example of a product load given, was for an automatic ice-making plant at 10 tons/day at 13 ft^2 storage space/ton. After being frozen it is subcooled to 25°F. The Navy notes typically cooling time as 18 to 24 hrs.

Infiltration is sometimes a significant portion of the load on a refrigerated cooler or freezer. To reduce these effects air curtains are suggested, which theoretically provide a 70 to 80% reduction in air flow. A reduction of 30 to 40% is suggested as being more realistic for design calculations. This takes into account the effects of traffic and draughts on the air curtain efficiency.

In refrigerated rooms there is a risk of freezing the ground below the floor. If this occurs the concrete with heave upward. The Navy specifies that this is only a concern if the space is at a temperature below 15°F and the room width is over 10 to 15 ft.

To optimize system efficiency, different temperature boxes are served by different compressor groups. The closer the suction group matches the optimum evaporator saturated suction temperature, the higher efficiency the system will have. Generally temperature difference between the room and evaporator saturated suction temperature is 5 to 10°F for freezers and cooler storage of meat and produce, 10 to 15°F for medium temperature storage of items such as milk, and 15 to 20°F for coolers used for meat and produce preparation.

Sizing refrigeration equipment should be done on a 16 hour basis if air defrost is being utilized. Electric, hot gas, and water defrost systems allow for equipment selection at 20 hr operation.

Various refrigerants are used in these systems. The application of these refrigerants is suggested to correspond to Table 44.

Table 44. Comparative Refrigerant Performance

Refrigerant	Bhp/ton	Compression Ratio	Refrig. Circ. (lb/min-ton)	Typical SST (°F)
R-717	0.989	4.94	0.422	-10 to 40
R-12	1.002	4.08	4.00	30 to 50
R-22	1.011	4.03	2.86	30 to 50
R-502	1.079	3.75	4.38	-40 to 0

Source: (Navy 1986)

(Adre and Hellickson 1989)

Adre performed simulations of cold rooms to analyze energy usage throughout the year, with a focus on opportunities to reduce operating costs. The study identified that the continuous operation of evaporator fans was a significant unnecessary cost. On the cases analyzed, approximately 37 to 52% of the zone refrigeration energy use is due to evaporator fan operation.

For the analysis, a cold store built in 1984 and operated by the Duckwall-Pooley Fruit Company in Hood River, Oregon was used. It is composed of one large common storage area (57.1 x 97.1 x 32.8 ft) with a 4365 bin capacity and three controlled atmosphere rooms (63.0 x 43.5 x 32.8 ft) with a 1245 bin capacity. Each bin measures 4 x 4 x 2 ft in outer dimensions and hold 790 lbs of fruit.

Assumptions and details related to setting up the analysis are as follows:

- One wall of the cold storage is next to the engine room which operates at 50 to 82°F.
- Horizontal heat transfer through the perimeter wall is assumed negligible.
- Ground temperature was assumed to be at a 2 ft depth.

- Indoor relative humidity of 95%.
- There is one door to move bins in/out.
- Average daily temperature and monthly humidity ratio were used to calculate infiltration heat gain.
- Defrost occurred 1 to 8 times a day in 10 minute durations. This was the only period of time that the evaporator fans did not operate.
- Lighting load varied depending on the number of bins loaded, from 0.86 to 2.15 kW for 11 hours per day.
- Equipment and occupancy loads are not explicitly defined but their duration is specified at 50% of an 8 hour day.
- Product loading is not explicitly defined.
- 16 hour compressor runtime, allowing time for the defrost cycle necessary for a 35°F room.

Adre uses "Sol-air" temperatures as he called them, for outdoor conditions defined as the temperature of the outdoor air, including the effect of solar radiation. The sol-air temperatures were calculated using an emissivity of 0.15 for light colored surfaces, resulting in approximate increase in roof temperature of 36°F (88.6 Btu/ft^3 of radiation from the sun).

(Altwies 1998; Altwies and Reindl 1999)

Altwies' work is focused on the potential benefits to a refrigerated warehouse operator from demand shifting the refrigeration load (Altwies 1998; Altwies and Reindl 1999). The main cost of operating a refrigerated warehouse is the energy usage. Energy demand is higher when the ambient temperature is higher, and therefore the load peaks during the daytime and diminishes at night. The cost of electricity follows this same trend as more end-users are calling for electricity in the daytime. By running the refrigeration equipment during the night time to drive down the storage temperature, Altwies suggests a refrigerated storage facility can operate at a lower cost even though more energy will be expended. Two options were analyzed, including full demand shifting and load leveling. Full demand shifting pulls the space temperature down throughout the night, so that no refrigeration equipment operation is necessary during the day to maintain the desired peak storage temperature. Load leveling involves operating the refrigeration equipment at a consistent rate throughout the day, over cooling at night and under cooling during the daytime. A computer model of a refrigerated facility was developed and then correlated with test data from the site.

The effect of temperature fluctuation on food quality has been a debate in the field of industrial refrigeration for some time. This is a concern for this storage methodology and its feasibility is defended by Altwies by summarizing previous research in Table 45. If fruits and vegetables are frozen at the given temperatures, the associated percentage loss of vitamin C may be expected over one year (IIR 1986):

- 10 to 14°F storage temperature – 80 to 90% Vitamin C loss
- -4 to -14°F storage temperature – 10% Vitamin C loss
- -22°F storage – practically no Vitamin C loss

Altwies suggests storage at or below 0°F, as this is a good balance between food preservation and operating costs. From the analysis of previous work she also concludes that temperature swings below 0°F have little effect on the quality of the food.

Table 45. Summary of Previous Frozen Food Quality Research

Product	Investigator	Conditions	Summary
Cauliflower	(Aparicio-Cuesta and Garcia-Moreno 1988)	-5°F (10 hrs) to 39°F (14 hrs) for 10 days, sealed in polyethylene bags	Freeze/thaw. Vitamin C losses of 35% over control, no color change, loss of flavor, softer
Strawberries, Blackberries, Raspberries, Peaches	(Woodroof and Shelor 1947)	-20 to 0°F and 0 to 10°F for up to 12 months, raspberries in sealed 3-lb metal cans, peaches in 1-lb waxed cartons, strawberries and blackberries in 10-lb moisture-proof bags inside corrugated boxes	Color, flavor, and aroma of low-temp cycled strawberries only slightly less pleasing than those at constant -10°F, differences in blackberries were less than with strawberries, peaches more sensitive to fluctuations, high-temp fluctuations worse than constant 10°F in most cases
Snap beans, Corn, Peas, Strawberries, Raspberries, Cantaloupe	(Hustrulid and Winters 1943)	-20 to 0°F (12 to 24 hrs) for minimum of 6 months, unknown packaging	No influence on appearance or palatability compared with control group at constant temperature
4 Varieties of Peas	(Boggs et al. 1960)	-5 to 5°F, -10 to 10°F, and 0 to 20°F (24 hr sinusoidal cycling), retail packaging	Fluctuation of temperature per se had little or no effect on deterioration
Pork roasts, Strawberries, Snap beans, Peas	(Gortner et al. 1948)	0°F (6 days) to 20°F (6 days) for 1 year, wrapped in cellophane (pork) or commercial packaging	Quality changes were equivalent to control foods held at constant 10°F
Okra, Peas, Strawberries	(Ashby et al. 1979)	-11°F (12 hrs) to 0°F (12 hrs) for 1 year, plastic pouches or paperboard cartons placed inside corrugated boxes	No significant changes

Source: (Altwies 1998)

The refrigerated facility that is modeled is a low temperature facility near south central Wisconsin (Design conditions of 98/78°F) with four warehouses where vegetables (Table 46) are processed, packaged, and stored at -5°F. At full capacity this facility can hold 50 million pounds of product in its 200,000 ft^2. Three of the warehouses hold bulk storage boxes (64 ft^3) made of heavy duty cardboard and a plastic liner. The fourth warehouse has products ready for shipment in retail packaging (1 lb bags or small boxes) that are situated inside medium sized cardboard boxes (2 x 1 x 1 ft), which are then stacked on 4 ft pallets and stacked 8 ft high. Additionally, the pallets of material are wrapped with plastic. Packaging of these products is important, because the more resistance the packaging has, the less sensitive the product quality will be to fluctuations in the room conditions over the course of the day.

Table 46. Summary of Vegetables Stored in the Low Temperature Facility

Vegetable	Bulk (% of total)	Cased (% of total)	Bulk Mass (lbm)	Cased Mass (lbm)
Baby Lima Beans	4.2		4905	
Broccoli	2.8	3.5	33136	16363
Carrots	10.7	3.3	126774	15441
Corn (cut & cob)	39.1	23.9	460631	112581
Green Beans	10.3	10.8	122034	50787
Peas	19.6	15.0	231405	70410
Total	86.7	56.4	978885	265582

Source: (Altwies 1998)

In developing a computer model to analyze the energy usage of the facility, product, conduction, infiltration, internal, and equipment loads were all taken into account.

In addition, details on equipment energy use and performance and electricity costs were included.

Product loading was developed using the property data in Table 47 and an undisclosed rate of loading provided by the warehouse.

Table 47. Property Data for Vegetables, Air, and Water Applied to the Model for the Low Temperature Facility

Vegetable	% Water (mass)	% Water Frozen at 4°F	Specific Heat Below Freezing (Btu/lb-°F)	Density (lbm/ft_3)	Thermal Conductivity (Btu/h-ft-°R)
Baby Lima Beans	67		0.401		
Broccoli	90		0.471		0.223
Carrots	88	94	0.465		0.387
Corn (cut & cob)	74		0.423		
Green Beans	89		0.468		
Peas	74	89	0.423	43.7	0.3178
Average	80	92	0.442	43	0.309
		Air	0.240	0.087	0.013
		Ice	0.466	57.4	1.41
		Water		62	0.034

Source: (Altwies 1998)

The transmission load calculation took into account the weather trends over a year, the effect of the sun on the outside surface temperature during the day, and the floor glycol heating loop load (Altwies assumes a constant floor temperature of 67.5°F). Construction details are listed in Table 48. The thermal mass of the building construction is important when attempting to demand shift. Altwies showed that thermally massive construction will delay the effects of changing outdoor conditions, reducing the fluctuations in the transmission load, as compared to lighter construction with the same insulation value using her model through comparisons of heat flux.

Table 48. Warehouse Construction Details

Warehouse:	A & B	C	D
Size (ft^2)	99069	25963	79869
Wall U-value (Btu/h-ft2-°F)	0.036	0.046	0.046
Roof U-value (Btu/h-ft2-°F)	0.036	0.035	0.035
Thermal mass	Light, primarily insulation	Heavyweight, prefab concrete and insulation	
Product to Air Volume Ratio	0.525	0.694	0.396
Installed Evaporator Capacity (tons)	80	120	319
Operating Set Point (°F)	-5	-5	-5

Source: (Altwies 1998)

Information related to the infiltration load of the facility was gathered from the operator in the form of timer settings for automatic doors and from on-site observation of typical door usage. Air velocity measurements were taken in an open doorway to generate an air velocity profile. From this information, a load was calculated and compared to the Chapter 27 ASHRAE 1990 method (ASHRAE 1990), 1994 ASHRAE method (ASHRAE 1994), and a method derived by Jones and applied by Cole (Cole 1987). The 1990 ASHRAE method (Equation 69) was found to best fit the test data and applied to the model.

$$\dot{Q}_{\inf-D} = \left[0.221 \cdot A_{\inf}\left(h_H - h_C\right)\rho_C\left(1-\frac{\rho_H}{\rho_C}\right)^{1/2}\right](gH)^{1/2} F_m D_t D_f \cdot 3600\frac{s}{hr} \qquad (69)$$

Facility personnel supplied information on internal and equipment loads to Altwies. Internal loads include occupancy, lighting, forklift, and other miscellaneous loads. Equipment loads include evaporator fan motors and defrost cycle contribution to the space.

For her research, Altwies altered the current -5°F setpoint to a temperature swing from -12.5 to -2.5°F over the course of the day, with the air temperature at -12.5°F at night and -2.5°F during the daytime by demand load shifting. Full demand load shifting requires idling refrigeration equipment for the entire on-peak period (10-14 hrs in length).

Altwies concluded with three options, including full demand load shifting, full load shifting and load leveling, and load leveling. Given a $155,000 annual cost for electricity the load strategies will save $82,000 (53%), $58,000 (37%), and $1000 (<1%), respectively.

(Manske 1999)

A pumped liquid overfeed ammonia cold storage warehouse near Milwaukee, WI was optimized through modeling the facility with Engineering Equation Solver (Klein and Alvarado 1999) and analyzing the typical operation scheme. Methods for reducing energy usage included, refrigerant temperature control, refrigerant flow control, condenser fan control, evaporator fan control, compressor configuration, refrigerant system configuration, condenser size, defrost control, and infiltration reduction.

Set point temps of 23°F and -12°F are maintained in the intermediate and low pressure receiver vessels respectively. These refrigerant temperatures provide for the freezer with a space temperature of 0°F and the cooler with a space temperature of 34°F. The banana and tomato ripening rooms are cooled using liquid following the high pressure receiver to maintain conditions of 45 to 65°F. Four degrees Fahrenheit of superheat is added by the evaporators and 15°F of subcooling is performed by an evaporative condenser.

Condenser fan control schemes include on/off motor cycling, two speed motor cycling, and variable speed control of the motors. Five schemes of part load evaporator operation include cycling the refrigerant flow to the evaporator while the fan motors stay on, on/off fan motor cycling, two speed motor cycling, variable frequency drive control of evaporator motors, and controlling refrigerant temperature supplied to the evaporator. As an alternative to the current single stage compression system, a two-stage and a thermosiphon system were analyzed.

Building construction is considered lightweight for all spaces and specified in Table 49. Outside of the freezer there is a warming room for workers to spend their breaks in.

Table 49. Conditioned Space Summary

Space	Area (ft^2)	Temp (°F)	Relative Humidity	R-Value Roof	R-Value Wall	R-Value Perimeter
Freezer	54000	0	80	33.09	32.85	n/a
Cooler	32000	34	87	25.09	24.85	1
Cooler Dock	5700	45	65	25.09	24.85	1
Banana Rooms	4000	56 to 64	80		Unavailable	
Tomato Rooms	4000	45 to 55	80		Unavailable	

Source: (Manske 1999)

This facility uses a single screw and a reciprocating compressor operating in parallel to a common discharge line. Compressor capacity adjustment was done in accordance with Equation 70 to account for variations in superheat, subcooling, mass flow, and suction or discharge temperatures.

$$Cap_{comp} = Cap_{comp,r} \frac{\upsilon_{compr}}{\upsilon_{comp}} \frac{\Delta h_{41}}{\Delta h_{41,r}} \tag{70}$$

Screw compressors may be unloaded down to 10% by proper slide positioning. Reciprocating compressors are unloaded by reducing the number of cylinders acting on the refrigerant. Both of these relationships, between load and power usage, were modeled using a linear regression for application in the modeling software.

Flow lifted by the compressors was cooled with an evaporative condenser, which is rated according to Equation 71, where heat rejection factor (HRF) is a function of the wet bulb temperature and the saturated condensing temperature. The coefficient N varies between 0.5 for laminar flow and 0.8 for turbulent flow (Mitchell and Braun 1998). The manufacturer of the evaporative condenser, used for this analysis, suggested a value of 0.76. Capacity is reduced to 35% by operating the condenser in dry cooling mode, which is required at outside temperatures below 31°F.

$$Cap_{econd} = \frac{Cap_{econd,n}}{HRF} \left(\frac{RPM_a}{RPM_r} \right)^N \tag{71}$$

Typically evaporator fans are operated all the time. With a focus on reducing energy usage of the modeled facility, Manske applied part load control of the evaporator fans. This is modeled using Equation 72, which is very similar to Equation 71. The manufacturer suggested a value between 0.5 and 0.6 for the exponent N whereas Manske applied an experimental value of 0.65 that was obtained through a personal communication.

$$Cap_{evap} = Cap_{evap,r} \left(\frac{RPM_a}{RPM_r} \right)^N \tag{72}$$

The loads applied to the systems were analyzed in great detail to effectively approximate the potential savings through different system control schemes. The banana and tomato room load is given as having an annual average of 40.5 tons. The design load for this room was 90.3 tons, giving an idea of how much equipment providers oversize such installations.

This freezer uses time controlled hot gas defrosting operating two to three times a day. The cooler uses hot gas and electric defrost methods. Hot gas blow-by fraction is 10.75% for the cooler and freezer. Fifty percent or more (ASHRAE 1994) of the defrost energy is lost to the refrigerated space. The hot gas defrost load is 180,000 Btu/hr for the cooler and 72,000 Btu/hr for the freezer. An additional 60,000 Btu/hr is added to the cooler by the electric defrost. One hundred twenty thousand Btu/hr is added to the loading dock by air defrost. These loads are mostly constant with a slight increase during the summer.

Infiltration loading was developed using Equation 73, with latent and sensible components separated using psychrometrics.

$$\dot{V}_u = 4.88\sqrt{H \cdot \Delta T}\,\frac{A_{\inf}}{2}F_\tau \qquad (73)$$

The door between the freezer and the outside was open 4 minutes/hour. The door between the cooler and freezer and the cooler and the outside was open 10 minutes/hour. The loading dock to cooler door was assumed to be open 15 minutes/hour. The door between the warehouse and the cooler was open 12 minutes per hour. The outside loading dock doors were open 5 minutes per hour.

Heat is added to the freezer by the warming rooms at 120,000 Btu/hr for most of the year and 156,000 Btu/hr for the months of June through August. An additional consistent heat load is added by the heated freezer floors at 2.5 Btu/h-ft^2. All interior loads, except for lights and warming room loads, are at design levels for second shift (3 pm to 12 am) and at half levels during all other times. Additional miscellaneous loads are summarized in Table 50.

Table 50. Internal Space Loads

Space	Source	Sensible (Btu/hr)	Latent (Btu/hr)	Loading Multiplier for First Shift Operation
Freezer	People	4830	9730	0.5
	Product	0	0	1
	Forklift	118000	0	0.5
	Lights	83060	0	1
	Sub-Floor Heating	135000	0	1
Cooler	People	3450	6950	0.5
	Product	0	1201	1
	Forklift	118000	0	0.5
	Lights	73700	0	1
Dock	People	3450	6950	0.5
	Product	0	0	1
	Forklift	55500	0	0.5
	Lights	27528	0	1

Source: (Manske 1999)

To achieve energy improvements, Manske recommends extensive instrumentation. Electric power monitoring of compressors, evaporator fans, and condenser fans along with miscellaneous other equipment will allow analysis of the control schemes utilized. The instantaneous refrigeration load should be calculated and logged from evaporator enthalpy change and refrigerant mass flow. Necessary instrumentation is summarized in Table 51.

Table 51. Recommended Performance Monitoring Measurements

Location	Minimum Measurement	Preferred Measurement	Usefulness
Condenser	Pressure		Condenser Fan Control
	Mechanical Room Power	Fan and Pump Power	Electricity Consumption
Compressor(s) Suction	Dry Suction Line Pressure	Individual Machine Suction Flange Pressure	Capacity, Horsepower, and Mass Flow Calculations
Compressor(s) Discharge	Discharge Line Pressure	Individual Machine Discharge Flange Pressure	Capacity, Horsepower, and Mass Flow Calculations
Compressor(s)	Part Load Operation		Capacity, Horsepower, and Mass Flow Calculations
Compressor(s)	Mechanical Room Power	Individual Machine Power Draws	Electricity Consumption
Evaporator	Coil Refrigerant Temperature/Pressure, Inlet Air Temperature	Supply and Coil Refrigerant Temperature/Pressure, Inlet and Outlet Air Temperature	Coil Capacity Calculations
Evaporator		Static Pressure Drop or Time Clock Measuring Liquid Solenoid Feed Time	Defrost Demand Determination
Evaporator	Evaporator Fan Power		Electricity Consumption
Receiver Vessels		High Pressure Liquid Supply Line Mass Flow	Load Calculations
Cold Space	Temperature		Control
Outdoor Air	Dry Bulb, Wet Bulb		Condenser Fan Control
Entire System	Electric Bill		Monthly Analysis

Source: (Manske 1999)

Manske compared thirteen different system configurations to analyze the cost reduction opportunities (Table 52). In relation to the capital cost to implement some of these changes, improvements for the most part were small. The best improvement predicted (28.8% savings) is through the use of a split system with an elaborate monitoring system to optimize system performance. This would be tied in with variable speed control of condenser fans, two-speed control of evaporator fans, and optimized defrost cycling. The split system puts the banana and tomato ripening room on its own circuit to reduce the inefficiency of cooling the space with a 23°F suction temperature. The ratio between the peak energy usage and off-peak energy usage for each system configuration was very similar with on-peak energy usage, ranging from 36.5 to 37.0% of the total energy usage.

Table 52. Energy Savings Comparisons between Optimization Techniques

System Design/Optimization Option	Electric Cost ($/yr)	Electric Savings ($/yr)	Electrical Savings (%)
Current System	89667		
Smaller Condenser	87659	2008	2.2
Two-stage Compression	90453	-786	-0.9
Split System	82610	7057	7.9
Split System with Thermosiphon	80655	9012	10.1
Bi-Level, Fixed Head Pressure	92242	-2575	-2.9
VFD Condenser Fans, Min Head Pressure	87458	2209	2.5
VFD Condenser Fans, Opt Head Pressure	85811	3856	4.3
On/Off Evaporator Fans, Opt. Receiver Temp	89131	536	0.6
Two-speed Evaporator Fans, Opt. Receiver T	76850	12817	14.3
Hot Gas Defrost Control	85892	3775	4.2
Warehouse Door Infiltration 50% Reduction	81816	7851	8.8
Optimized Split System – VFD Condenser Fans, Two-speed Evap., Reduced Defrost	63872	25795	28.8

Source: (Manske 1999)

(Stoeckle 2000)

Stoeckle investigates using the product in a refrigerated warehouse as thermal storage to allow reduced compressor runtime during peak electricity cost hours (Stoeckle 2000). A computer model was developed from which different operating schemes were analyzed. A forecasting method to estimate next day demand shifting benefits was developed.

The distribution warehouse modeled consisted of two refrigerated spaces. The freezer was 100,000 ft^2 (316.2 x 316.2 ft), 31.5 ft in height (3,121,000 ft^3 volume), and maintained at 0°F. A 20,000 ft^2 (316.2 x 62.96 ft), 12 ft high loading dock (237,773 ft^3 volume) is attached to this main storage building, which maintains a 34°F temperature. The freezer has 43% of its area covered with product, using a typical static pallet rack installation. Five doors connect the dock to the freezer (14 ft x 10 ft each). Thirty truck bays connect the dock to the outdoors, at 9 ft wide x 8 ft height dimensions each. The infiltration load associated with the opening of these doors is based on Downing and Meffert's method (Equations 74 and 75), assuming freezer doors are open 4 min/hour and each dock door is open 2 min/hour.

$$\dot{Q}_{\inf-D} = \left[4512.82 \frac{ft^{1/2}}{hr} \cdot A_{\inf} \cdot \rho_C \left(1 - \frac{\rho_H}{\rho_C} \right)^{1/2} (H)^{1/2} F_m (1-E) \right] (h_H - h_C) \qquad (74)$$

$$F_m = \left[\frac{2}{\left(1 + \frac{\rho_C}{\rho_H} \right)^{1/3}} \right]^{\frac{3}{2}} \qquad (75)$$

Two freezer constructions are analyzed, including a thermally massive wall and roof construction and a lightweight wall. Both walls had a R-value of 21.74. Demand shifting savings are typically improved with heavier constructions as a result of the load lagging effect that such constructions have due to their thermal capacity. The dock is assigned an R-value of 10.53 h-ft^2-°F/Btu. The convective coefficient used for the interior of the facility is 0.75 Btu/hr-ft^2-°F.

A consistent load that the space sees is a heating loop used to ensure the sub-floor does not freeze. A constant temperature of 67.5°F is assumed at the transition from the dirt to the 12 inch concrete slab and 2 in of insulation. This sub-floor temperature drives a heat flow of 2.48 Btu/h-ft^2, but the value of 3.6 Btu/h-ft^2, suggested by Jekel (Jekel and Reindl 2000), was applied to the model.

Additional loads include lighting for the freezer (0.45 W/ft^2) and the dock (2.5 W/ft^2), equipment load generated by fork lifts operating in the freezer which adds 18.17 tons, and occupancy loading which adds 2.25 tons of heat to the space. The product arrival temperature is assumed to be at storage temperature, removing that component from the load profile.

Some equipment details were noted. The evaporator was sized for a 5.4°F reduction in air temperature with an 8.1°F temperature difference for the freezer and the dock spaces. It is never explicitly stated, but it appears that this system utilizes an evaporative condenser, as the condenser performance is driven by the wet bulb temperature. The minimum temperature difference between the condenser refrigerant and wet bulb temperature is 18°F. The type of defrost was not specified but a typical value of 20% defrost efficiency was noted (Jekel and Reindl 2000).

The two warehouses were analyzed for over a year under typical refrigeration system operation. Results are compared in Table 53, showing very similar energy use. The freezer with massive wall construction required 258 tons of refrigeration capacity whereas the lightweight wall construction freezer required 344 tons.

Table 53. Yearly Cooling Load Comparison for Light and Massive Construction Types

	Sensible Cooling Load / kWh	Latent Cooling Load / kWh	Total Cooling Load / kWh	Total Electricity Demand / kWh
Massive Wall Warehouse				
Freezer	4060160	155121	4215281	1647379
Dock	550649	213893	764542	575260
Total	4610809	369014	4979823	2222640
Lightweight Wall Warehouse				
Freezer	4061038	156438	4217476	1609134
Dock	540927	212045	752972	573138
Total	4601965	368483	4970448	2182273

Source: (Jekel and Reindl 2000)

Demand shifting strategies first discussed by Altwies (Altwies 1998) include full demand shifting, load leveling, and a combination of the two strategies. Full demand shifting cools the refrigerated space enough that refrigeration equipment does not operate during the on-peak period. Load leveling strives to operate the refrigeration equipment at a consistent rate in order to overcool the space during off-peak hours and maintain a temperature below the maximum space temperature during on-peak hours. The third option attempts to take advantage of the off-peak energy costs while still allowing on-peak operation. This method allows for reduced energy costs without the need for as much refrigeration capacity. A fourth strategy discussed precools the space for a few hours prior to throttling back refrigeration equipment operation to maintain a desired temperature throughout high price periods.

During the load shifting simulations, temperature was allowed to rise above the typical freezer temperature of 0°F and brought down to -22°F during off-peak hours. To achieve savings with full load demand shifting, additional compressor capacity was required to drive the storage temperature down during off-peak hours. The massive wall freezer obtained savings of $7000 to $15000 using these methods, for a yearly operational savings of 4 to 10%. Precooling did not show reduction in energy costs because the heavy compressor operation, prior to the floating period, essentially negated any on-peak savings. ***(Stoeckle et al. 2002)***

Demand load shifting has potential to reduce operating cost by utilizing the products stored in the refrigerated space as a thermal battery and taking advantage of lower energy costs during off-peak periods. This paper presents a simple method for determining if demand load shifting will offer a reduction in operational costs for a specific warehouse on a daily basis (Stoeckle et al. 2002). The method includes predicting the maximum allowable floating duration based on the outside air temperatures for the next day and establishing the floating period that yields the greatest operating cost savings. A simple price ratio criterion was found to be a good indicator of energy cost savings by demand shifting.

Real-time pricing rate structures were obtained from Pacific Gas and Electric, Alabama Power, Southern Company, and Niagara Mohawk Power. Not only does electricity cost vary throughout the day, but it also varies from month to month, with peak electricity costs in the summer months.

Whether demand shifting will offer reduced energy costs depends primarily on the real-time price, with the equipment performance being a less significant factor. To determine whether a given day will be better served by conventional operation or by load demand shifting, Equation 76 should be analyzed. Stoeckle et al. found that the critical price ratio is 2.2 in numerous cases. When performing this analysis, if the critical price ratio is above 2.2, the refrigeration system will operate at a lower cost if the load is shifting to the off-peak hours.

$$CPR = \frac{P_{avg,day}}{P_{avg,shif}} \tag{76}$$

Using this method the model showed that annual savings of $12,581 were realized with a maximum product temperature of -0.4°F. Annual savings with a maximum product temperature of 3.2°F were $17,976.

Operating costs savings through demand shifting are relatively small compared to the annual refrigeration system cost. Installation of additional refrigeration capacity results in projected savings in the range of 7% to 11%.

(Magoo 2003)

Refrigerated storage requires significant amounts of energy. During off peak hours, energy is available at a lower cost. Magoo investigates the opportunity for shifting refrigeration system operation into the low cost electricity periods of the day by utilizing the thermal mass of the products being stored in high temperature warehouses. Coolers are limited on the amount of load shifting that they can perform. If the temperature is pulled down too low during off-peak periods, some products could begin to freeze, which would damage them.

A distributed warehouse is modeled and analyzed using TRNSYS. Three approaches are used for modeling the thermal effects of the product. The first approach assumes that the product is a consistent temperature throughout at any given time. The second approach analyzes the product using a three-dimensional finite difference approach. A third approach uses a two-dimensional finite difference model.

Magoo categorizes cold storage into five classifications, including controlled atmosphere storage, coolers, high-temperature freezers, low-temperature storage, and low-temperature storage with extra capacity for pulling down product temperature upon arrival. Controlled atmosphere coolers are used for long term storage of fruits and vegetables. By maintaining certain conditions, the longevity of the stored product may be prolonged. Coolers operate at temperatures above 32°F, high temperature freezers operate from 26.6 to 28.4°F, and low-temperature storage rooms operate from -20.2 to -9.4°F. Product received by some low temperature storage facilities is above freezing and must be pulled down through the latent heat of fusion. This requires significant additional capacity to quickly transition these products to a frozen state. Freezers store agricultural products, meat, fish, poultry, fruit juices, processed and canned foods, and ice cream. Coolers store milk, dairy products like cheese and butter, eggs and egg products at temperatures close to freezing. The closer these products are to freezing the longer their quality will be maintained (ASHRAE 1998). These units have a much higher product turnover than freezers, as products can only be stored for a short period of time at temperatures above freezing before the product quality begins to degrade. Additional spaces include refrigerated docks which are used to reduce infiltration into the associated cooler or freezer to which it is adjoined and reduce temperature loss from products that are received. These facilities are typically maintained at 50°F if adjoined to a cooler.

Magoo discusses freezer and cooler construction details as cited by ASHRAE (ASHRAE 1998). R-values suggested are listed in Table 54.

Table 54. Suggested Resistance Values for Refrigerated Storage Facilities

	R-Value (ft^2-h-°F/Btu)	
	Freezer	Cooler
Floors	27.26 to 31.80	19.87
Walls	35.21 to 39.75	23.85 to 31.80
Roofs	44.86 to 49.97	35.21 to 39.75

Source: (ASHRAE 1998)

Typically, insulated warehouse walls are built with insulation sandwiched between either reinforced concrete or sheet metal on either side. Common insulation materials include polyurethane, polystyrene, and Styrofoam.

Freezers have floor heating to keep the concrete from heaving. The most popular method is a grid of pipes directly below the concrete slab and insulation. Either hot air or glycol is circulated through the pipes to maintain the slab temperature. Coolers do not require sub-floor heating, but insulation is still typically placed under the concrete slab.

The cooler analyzed in this work was developed by Stoeckle (Stoeckle 2000). It is 100,000 ft^2 with each side wall 314.8 ft in length. The height of the walls is 31.5 ft for a volume of 3,121,000 ft^3. A dock adjoins this space and runs the entire length, with dimensions of 314.8 x 62.96 ft and 12.01 ft height for a dock space of 20,000 ft^2 and volume of 237,773 ft^3. The dock is maintained at 50°F whereas the cooler is maintained at

34°F. The space is maintained using a saturated suction temperature of 19°F and discharge temperature of 95°F. If the space temperature varies by 0.09°F from the setpoint, cooling is initiated.

Wall and roof construction is heavyweight for the cooler, composed of insulation between two layers of two inch thick concrete. Three different insulation thickness' were analyzed by this work, including 5.71 inches at 27.93 ft^2-h-°F/Btu, 4.29 inches at 21.19 ft^2-h-°F/Btu, and 5.08 inches at 24.88 ft^2-h-°F/Btu. The dock included in the simulations is composed of the following items from the outside to inside: two inches of concrete, 1.85 inches of insulation, and 5.98 inches of concrete, with an R-value of 10.25 ft^2-h-°F/Btu. The dock and cooler have a common floor composed of 11.8 inches of concrete covered with 1.38 inches of insulation for an R-value of 8.28 ft^2-h-°F/Btu. No floor heating is utilized, with the slab resting directly against soil, which is assumed to be at a constant temperature of 50 °F.

Five 10 x 14 ft doors connect the cooler to the dock. The dock has 30 doors which open to the outdoor air. Each door is 9 x 8 ft. Product is assumed to arrive at storage temperature so there is effectively no product load. The product is assumed to cover 60% of the floor space on average. This is important, because the amount of product in the cooler will affect the capability of the system to load shift.

Infiltration loading is calculated using the Gosney Olama equation (Equation 18). Door openings were assumed to be one per hour for each dock door with a 60 second total door open period. The doorway flow factor is set at 0.8 per ASHRAE (ASHRAE 1998). According to common practice, cooler to dock doors were left open continuously with a door flow factor of 0.8.

Internal loads include lighting, occupancy, and equipment. Lights in the cooler are assumed to have a power density of 0.45 W/ft^2. Seventy percent of the power is radiative heat gain with the rest being conductive. This is important because this analysis is interested in heat transfer in the refrigerated space on a time dependent basis. Stoecker (Stoecker 1998) recommends a similar value of 0.46 W/ft^2 for general purpose illumination in food storage warehouses. The dock had a power density of 2.5 W/ft^2.

People load is calculated according the 1998 ASHRAE method (ASHRAE 1998), defined by Equation 77. Seven people occupy the dock and three occupy the cooler at all times. Sixty percent of the occupancy load is radiative.

$$\dot{q}_p = 928.10 - 11.37 \cdot T_C \qquad (77)$$

Fork trucks are used in the warehouse and dock, adding 15.44 tons and 2.73 tons, respectively. Seventy percent of these loads are radiative.

Milk was analyzed with a lumped capacitance model and a zone air temperature set to 36°F. Milk has recommended safe storage temperatures from 33.1 to 39.9°F and a convective heat transfer coefficient of 0.75 Btu/h-ft^2-F (ASHRAE 1998). Cheese was modeled by a three-dimensional finite difference model with an allowable temperature range of 30 to 50°F.

Increasing the U-value of the walls did increase the energy usage of the refrigerated cooler but did not affect the percentage savings during demand-shifting. Results for the three product modeling techniques were compared. The three-dimensional finite difference method arrived at higher energy usage than the lumped capacitance model. Full demand shifting was found to be possible in both cases. The lumped capacitance model for milk requires a pre-cooling setpoint of 37°F to achieve this, and the finite difference model for cheddar cheese showed that a precooled setpoint of 41°F is required.

(Roy 2010)

Industrial refrigerated warehouses' primary operating cost is the cost of electricity to maintain the desired conditions for products. This article discusses Sobeys warehouse of Trois-Rivières. To reduce energy use, this warehouse utilizes an ammonia central chiller and glycol secondary loop, improved insulation, natural lighting, monitored fresh air supply, and miscellaneous other energy trimming efforts.

The refrigerated portion of the warehouse is 120,000 ft^2, maintained at 39°F. Heating is provided by a radiant floor to both avoid freezing of the sub-floor and improve employee comfort. Building insulation is R-30 for the walls and R-40 for the ceiling. The roof is reflective (white) to reduce the solar load. T5 high output fluorescent lights are used throughout the warehouse to achieve optimum energy efficient lighting. This is supplemented by natural lighting brought in through R-7.5 windows on the south side of the building. Fresh air was reduced by 75% through the use of CO_2 sensors that monitor the space conditions. When outdoor temperatures are below 10°F (844 hours during the year for this location), the refrigeration system operates without compressors using a thermosiphon cycle. During the sizing of equipment, the thermal mass of the product being stored (44,092 lb of milk products) was taken into account, reducing the refrigeration equipment size (transient effects were not considered).

This system offers energy savings of $144,896 annually with a simple seven year payback.

Lighting Load

(Rea 2000)

The Iesna Lighting Handbook Reference and Application presents details on lighting as it applies to numerous industries and applications (Rea 2000). This edition of the handbook has a heightened focus on light quality, discussing design criteria such as eye-source-task geometry, flicker, color, and glare. The handbook gives very little information in regard to refrigeration. It cites that refrigeration storage areas reach as low as -20°F and that warehouse type buildings use fluorescent lighting along the aisles.

Product Load

(Aparicio-Cuesta and Garcia-Moreno 1988)

Long term storage conditions and their effect on food quality are very important to those working in industrial refrigeration. Aparicio-Cuesta et al. presented a report on frozen cauliflower in which the conditions were varied for different samples over a 30 month storage period. The product was kept at a temperature of -7.6°F. Storage in a display freezer was also analyzed for 60 days at -0.4°F.

(Wade 1984)

Wade discusses details of refrigeration design related to the cooling of harvested products from field temperature, including system capacity, air flow management, and pertinent equations. Just because a refrigeration system has sufficient capacity does not mean it will be able to cool the product appreciably faster. If flow is not passing through the product it will not be reduced from its initial temperature to the storage temperature in a 24 hour period no matter what the refrigeration system capacity is. Industrial refrigeration applications typically use pressure-cooling, to ensure a steady flow rate over the product to draw off heat. As an example of product cooling time opportunity, an apple exposed to free airstream at 32°F may be cooled from 68 to 41°F in 2 hours when the surface heat transfer coefficient is 1.94 Btu/h-ft^2-°F. It takes 48 hours to achieve a similar reduction in the center of a stack of packaged apples 1 ft wide.

The time at which the product is seven-eighths cooled is defined as follows. In Equation 78, the lag factor, *J*, is a function of the geometry, thermal conductivity, thermal diffusivity, and surface heat transfer coefficient of the produce. The lag factor is significant when pressure-cooling is being applied because the surface heat transfer coefficient is typically much larger than the thermal conductivity of the product.

$$S = \frac{-\ln(8J)(T_{pc} - T_C)}{dT_{pc} / dt} \qquad (78)$$

If pressure-cooling is not being applied and heat transfer is most limited by the surface heat transfer coefficient, the lag factor does not have a significant effect on the cooling rate. The average cooling rate gives satisfactory results since the load is primarily a conduction load.

Cooling times are typically specified as 24 hours, with the compressor running time between 16 to 24 hr/day. If a pallet is packaged and stored so that there is no flow rate available to it the cooling rate will be substantially less. Cooling times of S = 48 to 79 hours apply if heat transfer is only by conduction. Small stacks of open bulk bins can achieve a 16 to 28 hour cool down if they are properly packaged and stored.

(Devres and Bishop 1995)

Devres et al. built a computer model to estimate energy usage and product moisture loss for given conditions (Devres and Bishop 1995). For this study an actual potato store in the Cambridge region of the United Kingdom was analyzed. Model predictions were compared to actual store measurements to analyze the model effectiveness. This information is important to industrial refrigeration because they are typically conflicting parameters which must be balanced to maintain a successful refrigerated storage operation.

Devres et al. used the following temperature relationship assumptions for the model when measured data was not available:

- Corridor temperature = Ambient temperature - 18°F
- Ground temperature = 41°F
- Ceiling temperature = Ambient temperature + 18°F

For sun load hours an additional temperature difference was added to the walls to account for solar radiation. The structure has a light colored exterior to which 3.6, 5.4, and 5.4°F are added for the south, west, and east walls, respectively.

The interior temperature of 39.2°F (90% relative humidity) is maintained by an evaporator operating at 28.4°F, with a T.D. of approximately 10°F. According to Devres, product storage should be kept within 2 to 5°F of the product freezing point to achieve the best product storage life. With these operating conditions, a defrost system will be needed, as 25% of the condensed water will turn to frost on the coil. Devres uses a defrost efficiency of 50%.

The facility being analyzed has a product load that represents over 50% of total daily load with a half cooling time of 10 days. Product is added to the store over 20 to 30 days and is later unloaded over four to five months. The product's temperature at any given time may be estimated by Equation 79.

$$T_{\bar{p}} = \left(T_{pini} - T_C\right)\exp\left(\frac{\ln(0.5)t_1}{t_{1/2}}\right) + T_C \qquad (79)$$

Devres et al. models the load added by packaging material and respiration heat load. Infiltration is calculated for the door and via crack infiltration. For calculating the infiltration through doorways and through walls, the following equations are applied. This set of equations utilizes the Fritzsche and Lilienblum correction factor, C_2, which takes into account the contraction of the flow, friction, and thermal effects (Fritzsche and Lilienblum 1968).

$$\dot{Q}_{Inf-D} = C_2 t_{open} \rho_C A_{\inf} \sqrt{H\left(1 - \frac{T_C}{T_{cor}}\right)}\left(h_{cor} - h_C\right) \qquad (80)$$

$$C_2 = 3130 + 26.1\left(T_{cor} - T_C\right) \qquad (81)$$

$$t_{open} = \frac{\left(m_{pj} + n_{pj}\right)}{m_{pal}} 2t_{oc} \qquad (82)$$

$$\dot{Q}_{Inf-W} = C_3 C_{ir} V \frac{2}{\upsilon_H - \upsilon_C}\left(h_H - h_C\right) \qquad (83)$$

(CIGR 1999)

Volume 4 of the *CIGR Handbook of Agricultural Engineering* discusses crop quality and its relationship to handling, drying, and storage. Crops of specific interest include grains, potatoes, onions, cassava, yam, edible aroids, fruits, vegetables, grapes, olives, and coffee.

To preserve product quality it is typically best to store the product at low temperatures to slow metabolic processes. Some metabolic processes are exceptions to this rule. Potato respiration rate is slowed by a reduction in temperature. Formation of sugars occurs counter to this trend, forcing storage temperatures to stay above 50°F . Grain storage is performed at 50°F and 75% relative humidity in tropical climates. Sorting and handling of products is typically done at temperatures around 55°F. Various storage temperatures are noted in Table 55. Typically cold stores should be maintained above 32°F.

Table 55. Storage Temperature for Various Crops

Cultivar or Common Name	Storage Temperature (°F)	Relative Humidity (%)	Length of Storage
Seed potatoes	35.6 to 39.2		
Consumer potatoes	39.2 to 41.0		
French fry/dried potatoes	41.0 to 46.4		
Chip industry	44.6 to 50.0		
Starch and derivatives	42.8		
Onion bulbs	26.6 to 35.6	65 to 85	6 to 9 months
Spring (green) onion	32.0 to 33.8	90 to 100	1 to 3 weeks
Cassava	32.0 to 41.0	85 to 90	1 to 6 months
D. trifida yam	37.4		1 month
D. alata yam	54.5 to 62.6		2 to 6 months
D. cayenensis yam	55.4	95	< 4 months
Water yam, greater yam	86.0	60	> 1 month
Elephant yam	50.0		> 1 month
White yam, guinea yam	60.8	80	> 1 month
Yellow yam, 12-month yam	60.8	80	2 months
Cush-cush, Indian yam	60.8 to 64.4	60 to 65	> 1 month
Lesser yam, Chinese yam	77.0		2 months
Tannia	44.6 to 50.0	80	4 to 5 months
Taro	40.0 to 55.9	70 to 90	4 to 6 months

Source: (CIGR 1999)

After harvesting, some crops are put through a regiment of conditions to prepare them for long term storage. Onion bulbs are dried at a high temperature (86 to 89.6°F) using a high airflow rate (254 ft^3/min-ton) for three to five days to remove surface water.

Airflow and temperature are reduced (101.7 ft^3/min-ton and 78.8 down to 59.0°F at 65 to 75% relative humidity) for about 20 days to remove surface moisture and dry, cure, and seal the skin. Bulbs may be stored for up to 10 months if properly cured and stored at 32 to 41°F and 65 to 75% relative humidity. Bulbs are warmed above the dew point of the air for seven days prior to removal to avoid condensation of moisture during grading. After the cooling or pull-down period during which the field heat is removed from the crop, product is often moved to another facility for storage. This allows each refrigeration system to be designed for the specific function of pull-down or storage, reducing overall energy usage and improving equipment effectiveness.

Reading this resource puts into perspective the difficulty of delivering a product to a customer. For example, typical storage losses for Cassava root under ideal conditions is 6 to 7% per week. Not only must storage temperature be low enough to reduce respiration effect, but the temperature must typically be above a threshold to avoid other issues, such as bruising of tubers at temperatures below 50°F.

The temperature dependency of the respiration rate is expressed by Equation 84. Typically Q_{10} is between 2.0 and 2.5 for a temperature range from 35 to 77°F.

$$Q_{10} = \frac{R_{T1+10°C}}{R_{T1}} \qquad (84)$$

The carbon dioxide production and the associated heat load of products is described by Equations 85 and 86, respectively (Becker and Fricke 1996b; Hardenburg et al. 1986) in correspondence to Table 56.

$$\dot{m}_{CO2} = f\left(T_{\bar{p}}\right)^{g_{resp}} \qquad (85)$$

$$\dot{q}_{resp} = C_{resp}\,\dot{m}_{CO2} \tag{86}$$

Table 56. Coefficients for Carbon Dioxide Production by Commodities

	Skin Mass Transfer Coefficient, k_s (grains/ft^2-s-ksi)			Respiration Coefficients		
	Low	Mean	High	VPL	f (Btu/grain-°F^g)	g_{resp}
Carrots	314.3	1542.1	3568.5	0.99	3.074×10^{-3}	1.793
Onions	--	8.778	--	0.98	2.254×10^{-5}	2.538
Potatoes	--	6.277	--	0.98	1.050×10^{-3}	1.769
Sugar beets	89.86	332.1	863.0	0.96	5.280×10^{-4}	1.888

Source: (Becker and Fricke 1996b)

Respiration of product also depends on the atmosphere in which it is stored. Most root products have significantly lower respiratory activity if the external oxygen concentration drops below 10%. Examples of the effect of temperature and oxygen on the respiratory rate are given in Table 57. In addition to removing moisture from the product, respiration in a properly designed storage facility can be expected to remove dry matter at 1.5% of the products' weight.

Table 57. Effect of Temperature and Oxygen Concentration on the Respiratory Rate of Various Commodities

	Carbon Dioxide Production (grains/lb-hr)					
	In Air			In 3% O_2		
	32°F	50°F	68°F	32°F	50°F	68°F
Beetroot						
Storing	.028	.077	.133	.042	.049	.070
Bunching with leaves	.077	.154	.280	.049	.098	.224
Carrots						
Storing	.091	.133	.231	.049	.077	.175
Bunching with leaves	.245	.518	.847	.196	.378	.595
Onion	.021	.049	.056	.014	.028	.028
Potato						
Main crop	.042	.028	.042	.035	.021	.028
Immature	.070	.140	.280	.070	.126	.210

Source: (Kays 1991)

The handbook offers some rules of thumb for the design of refrigerated storage systems ranging from ammonia to halocarbon systems. Generally storage systems are designed with R-30 to 60 for the ceiling, R-20 to 40 for the walls, and R-10 for the floors. Typical loads are from 0.96 to 1.32 Btu/h-ft^3 for refrigerated storage and 1.32 to 2.42 Btu/h-ft^3 for shipping docks. A conduction detail specifically outlined by this work is related to the sun’s effect on the conduction load. A dark, flat roof may be 76°F warmer than the outside air temperature whereas a light colored surface would be only 56°F warmer. Designers add 10 to 20% to the calculated peak load to ensure equipment performance at worst case conditions.

Refrigerated storage requires adequate ventilation to reduce temperature stratification or isolation. Most storage facilities are designed to have airflow of 100 ft^3/min-ton. After the product has reached its storage temperature the airflow is reduced to 20 to 40 ft^3/min-ton with relative humidities typically maintained above 90%.

If evaporators for cold storage are oversized they can operate at a higher temperature. This reduces the amount of moisture removed from the air. It is good practice to size the coil large enough to operate at 4 to 8°F colder than the return air temperature.

(Chourasia and Goswami 2007)

Harvested products require various steps to properly store them depending on the product type. Potatoes require their storage facility to quickly pull-down their temperature from field to storage temperature in order to reduce the respiration losses that occur at elevated temperatures. During the transient cooling period, product cooling contributes about two-thirds of the refrigeration load. In this study, various combinations of cooling rates and loading patterns were investigated to analyze which gave the optimum product quality and minimum refrigeration load. The best configuration was a half cooling time of 6.72 day (30 days to storage temperature) in combination with a linearly decreasing product loading rate over 30 days.

Typically, product load calculations are done assuming a constant load over the entire transient period. This is an appropriate assumption when products are added to the space in a periodic manner. In the application of cooling products from harvest, this is not the case, as the actual load profile varies with time (Cleland 1983).

Potatoes are harvested, cured for 1 day, and then cooled to a final temperature of 71.6 to 77°F. A commercial potato cold store has a capacity of 4000 tons of which 3 to 4% is typically added every day during the harvest season. Once at storage temperature, they are stored anywhere from 1 to 8 months.

Research performed by Devres and Bishop (Devres and Bishop 1995) used the half cooling to represent the product-pull down rate. Chourasia et al. used Equation 79 to define the product temperature at any given time. Moisture loss, which is detrimental to the product, is defined by Equation 87 and heat loss due to this moisture loss is defined by Equation 88.

$$T_{\bar{p}} = \left(T_{pini} - T_C\right)\exp\left(\frac{\ln(0.5)t_1}{t_{1/2}}\right) + T_C \tag{79}$$

$$\dot{m}_{-w} = k_m p_{sat}\left(A_w - \frac{RH}{100}\right)\frac{a_p}{\rho_p} g_c 144 \frac{in^2}{ft^2} 3600 \frac{s}{hr} \tag{87}$$

$$\dot{q}_{et} = \dot{m}_{-w} h_{fg} \tag{88}$$

Cooling rates analyzed are listed in Table 58. Loading rates were either linearly decreasing over a range of 10 to 30 days, constant over 30 days, or a replication of an actual product load from a cold store. The actual loading pattern was primarily a linearly decreasing profile to the 20th day (415 to 20 tons per day) after which it was mostly constant at 15 tons per day.

Table 58. Cooling and Rates Analyzed

Test or Simulation	Cooling Rate Pull-Down Period (days)	$t_{1/2}$ (days)
C-1	10	2.24
C-2	20	4.48
C-3	30	6.72
Devres and Bishop (Devres and Bishop 1995)		10
Chourasia, product near surface (Chourasia et al. 2003)		5

Source: (ASHRAE 1998)

Simulation C-3 had the lowest energy usage due to a less difficult pull-down period. This reduction in energy use is at the sacrifice of product quality however, as moisture loss increased 24% from C-1 to C-3. Another interesting fact is that the heat given off by the product increases for longer pull-down periods. Cooling rates C-1, 2, and 3 correspond to product heat output of 40.75, 45.83, and 50.90 Btu/lb. The more time it takes to cool the product, the more time it spends at a higher metabolic rate.

Infiltration Load

(Hayes and Stoecker 1969)

Hayes et al. gives a detailed discussion of air curtains. Presented research is on high-velocity, non-recirculatory air curtains. A theoretical flow profile with a focus on the flow deflection is compared to measured values with positive correlation. Flow profiles may be modeled by the Equations 89 to 91.

$$\text{Transition zone, } \bar{x} < 5.2\mathrm{b_o}: \ \frac{u}{u_o} = \frac{1}{2}\left[1 + \frac{2}{\sqrt{\pi}} \int_0^x e^{-t^2} dt\right] \tag{89}$$

$$\text{where, } x = \alpha_1 \frac{\bar{y} + b_o/2}{\bar{x}} \tag{90}$$

$$\text{Fully developed zone, } \bar{x} > 5.2\mathrm{b_o}: \ \frac{u}{u_o} = \frac{\sqrt{3}}{2} \sqrt{\frac{b_o \alpha_2}{\bar{x}}} \left[1 + \tanh^2\left(\alpha_2 \frac{\bar{y}}{\bar{x}}\right)\right] \tag{91}$$

Specific details noted that are applicable to this study include some storage and air curtain details. Typical freezer storage conditions were noted as 14°F, at 75°F outdoor temperatures. Access doors protected by air curtains were typically 7 ft high for this study with an expected air curtain effectiveness of 60 to 85%.

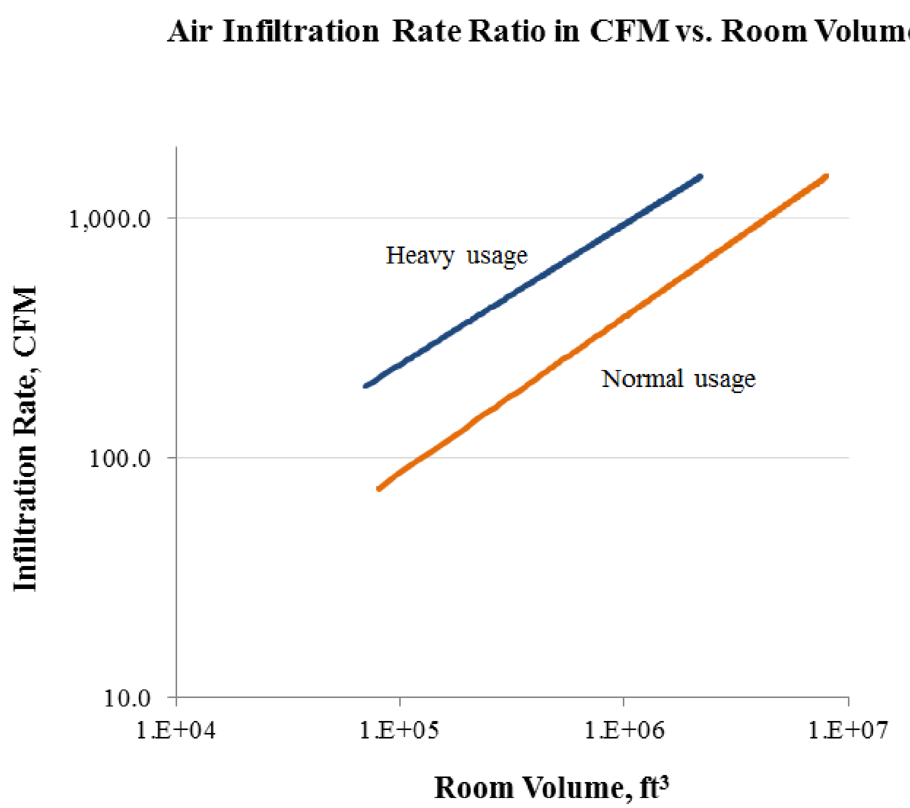
(Cole 1987)

Cole discusses infiltration in industrial facilities and its effects if not controlled closely. Available infiltration calculation methods are discussed.

Cole presents an air velocity estimation method utilized by Krack (Krack 1977). Equation 92 estimates the air velocity with a simple analysis. Another simple analysis which has a higher degree of vagueness is that presented by ASHRAE and repeated in Figure 5.

$$u = 4.88\sqrt{H \cdot \Delta T} \qquad (92)$$

Figure 5. Air Infiltration Rate in CFM vs. Room Volume



Source: (ASHRAE 1981)

Gosney and Olama (Gosney and Olama 1975) developed a mathematical expression (Equations 18 and 19) for infiltration using data gathered with table top testing. The scale models utilized carbon dioxide and air to represent the differing densities whose exchange was monitored through the scaled opening.

$$\dot{m}_{ss} = 0.221 \cdot 3600 \frac{s}{hr} A_{\inf} \rho_C (1 + \Delta\omega) \left(1 - \frac{\rho_H}{\rho_C}\right)^{1/2} (gH)^{1/2} \left[\frac{2}{1 + (\rho')^{1/3}}\right]^{3/2} \qquad (18)$$

$$\rho' = \frac{\rho_C}{\rho_H} \qquad (19)$$

Jones et al. (Jones et al. 1983) studied the same phenomenon but with a focus on the latent load. Through the use of full scale laboratory tests and applying assumptions of a rectangular opening and equivalent mass flows, the mass flow of water vapor was defined according to Equation 93.

$$\dot{m}_{wv} = 0.1733 \cdot 3600 \frac{s}{hr} \Delta\omega \rho_C \left[g\left(\frac{T_H}{T_C} - \frac{T_C}{T_H}\right)\right]^{1/2} WH^{3/2} \qquad (93)$$

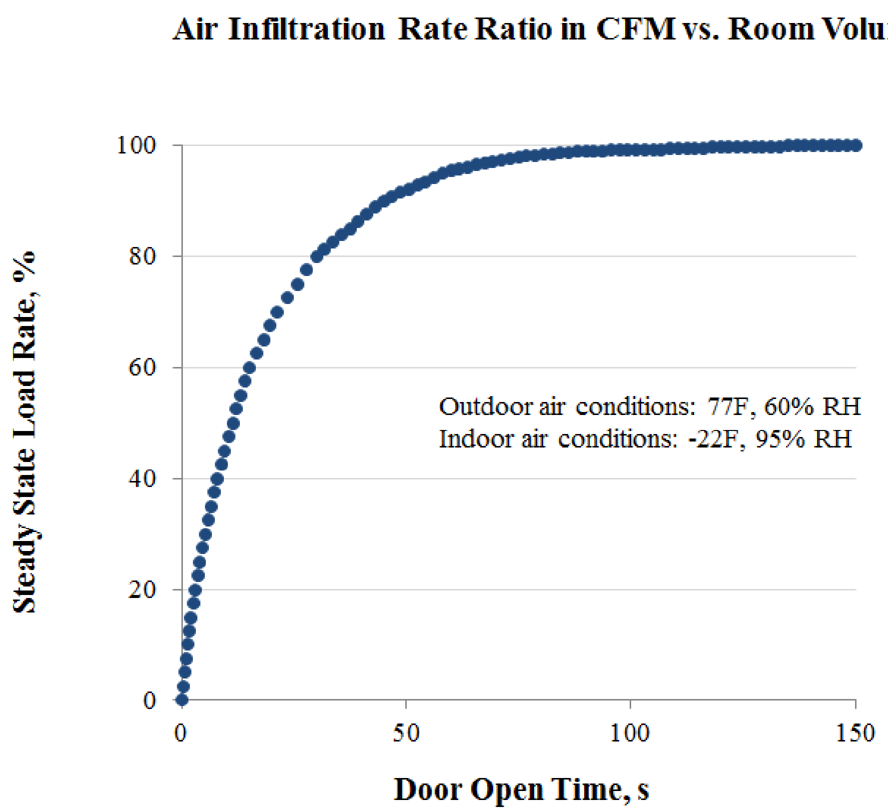
Cole combined the work of Gosney, Olama, and Jones to determine the total flow of air and moisture (Equation 94) and the associated load (Equation 95).

$$\dot{m}_{ss} = \frac{0.1733\Delta\omega}{\upsilon_c} \left(1 + \frac{1}{\Delta\omega}\right) \left[g\left(\frac{T_H}{T_C} - \frac{T_C}{T_H}\right)\right]^{1/2} WH^{3/2} \cdot 3600 \frac{s}{hr} \qquad (94)$$

$$q' = \frac{0.295(1 + \Delta\omega)}{\upsilon_c} \left[\frac{T_H}{T_C} - \frac{T_C}{T_H}\right]^{1/2} H^{3/2} (h_H - h_C) \qquad (95)$$

In a discussion of the equations defining infiltration, it is noted that the height of the opening is the most significant geometric parameter. The width is only a scaling parameter whereas the height has an exponential relationship with the load. This fact should be considered in the design and installation details on doorways to be used for refrigerated storage. Application of the above equations to the actual site is discussed. The duration of the door being open and the amount of disruption incurred by the air movement affects the percentage of air exchange that actually occurs. Cole presents a figure (Figure 6) from an undisclosed source that presents the transient effects of air flow.

Figure 6. Steady State vs. Transient Heat Gain Through an Open Door Assuming 10 ft High by 6.6 ft Wide



Source: (Cole 1987)

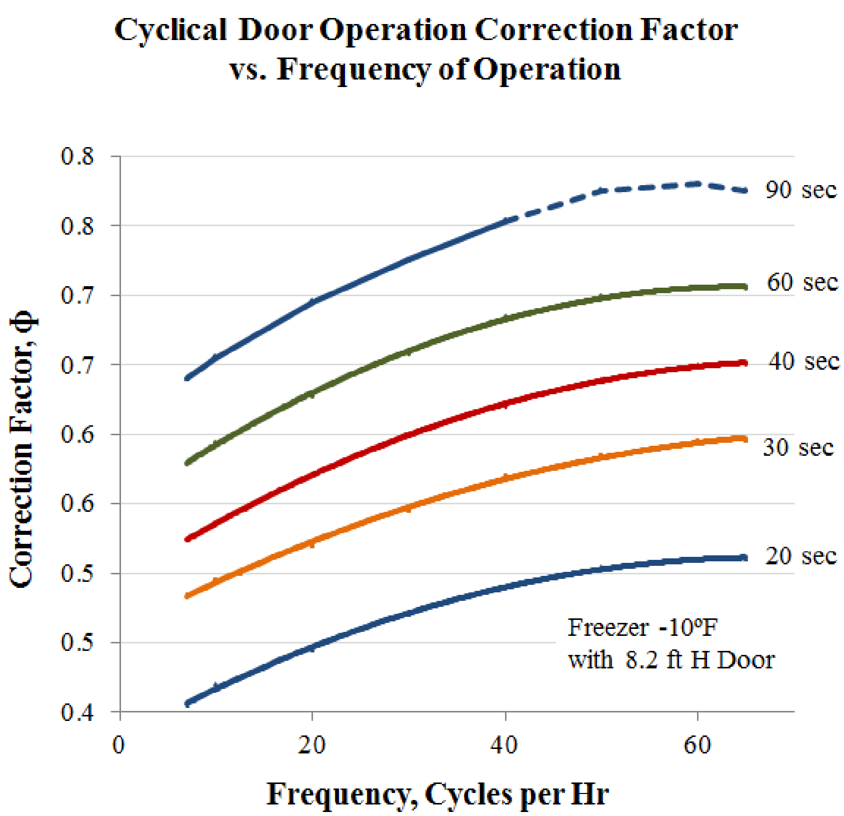
Gosney and Olama (Gosney and Olama 1975) derived a time related factor, ϕ, which varies based on the duration and frequency of door openings and physical parameters such as door height and room temperature. This parameter is displayed graphically in Figure 7 and its definition is repeated below in Equations 28, 29, and 30.

$$\Phi = \left(1 - \frac{0.0112}{N_o^{0.3565}}\right) N_f^{0.1} \quad (28)$$

$$N_f = t_{open} f_d = \frac{t_{open}}{t_d} \quad (29)$$

$$N_o = \frac{v t_{open}}{H^2} \quad (30)$$

Figure 7. Cyclical Door Operation Correction Factor vs. Frequency of Operation for a Freezer with a 8.2 ft High Door Held at -10°F



Source: (Cole 1987)

Additional factors effecting infiltration include internal partitioning (Pham and Oliver 1983) and doorway traffic, which is analyzed by many researchers. Internal partitioning is a function of room geometry and product stacking around the door. Effectively, air flow from the refrigerated space is restricted resulting in the infiltration load dropping off after several minutes of a door being open. Traffic through a doorway reduces infiltration by disrupting the air flow. One traffic passage per minute reduces infiltration by 15%. The lag time that it takes the flow to re-establish is a function of many variables and ranges from 15 seconds to 1 minute. Practitioners have observed these theories in action for continuously open (20 minutes or more) unprotected doorways. The realized load is less than the calculated value due to doorway traffic and partitioning.

Cole summarizes research on door infiltration reduction, including information on air and strip curtains presented by Pham and Oliver (Pham and Oliver 1983). Non-recirculating vertical air curtains have been shown to be 79% efficient whereas strip curtains are 91% efficient. Traffic passing through a strip curtain will reduce efficiency by up to 50%, but this is still an improvement on an open door's infiltration.

(Chen et al. 1999)

Chen performed testing on the infiltration through freight doors used on industrial refrigeration facilities. Door testing included rapid-roll doors, strip curtains, crack infiltration, and the effect of forklift traffic on infiltration (Chen et al. 1999).

Chen modifies an equation presented by Tamm to define infiltration (Tamm 1965) and utilizes an equation by Downing to approximate the effect of door protective device effectiveness (Downing and Meffert 1993) (Equations 96 through 100).

$$\dot{V}_{\inf} = 0.67 \cdot W \cdot H \sqrt{2gH \frac{1-s}{\left(1+s^{1/3}\right)^3}} \cdot 60 \frac{s}{\min} \cdot F_{Tamm} F_P F_t F_\tau + \dot{V}_{AT}\left(1-F_\tau\right) \qquad (96)$$

$$F_t = \frac{\dot{V}_{tr}}{\dot{V}_{nt}} \qquad (97)$$

$$F_\tau = \frac{N_{do}\left[0.5\left(t_a + t_c\right) + t_b\right]}{t_{tot}} \qquad (98)$$

$$F_P = \left(100 - E\right)/100 \qquad (99)$$

$$E = \left(1 - \frac{\dot{V}_p}{\dot{V}_u}\right) \cdot 100 \qquad (100)$$

Tamm's equation was validated by Longdill by testing it on a 3.94 x 5.25 ft high doorway in a 6250 ft^3 cold room (Longdill et al. 1974). Pham and Oliver utilized a slightly modified version of Tamm's equation with infiltration testing performed on 6250 to 1306640 ft^3 rooms with room heights from 12.5 to 69.6 ft (Pham and Oliver 1982; Pham and Oliver 1983). Door sizes tested ranged from 3.54 x 6.50 ft to 9.84 x 11.81 ft.

Single-plastic strip curtain effectiveness reported by numerous authors is noted as follows:

- 86-96% (Downing and Meffert 1993)
- 93% (Pham and Oliver 1983)
- 92 to 96% (Hendrix et al. 1989)
- 93% with rapid roll doors (Downing and Meffert 1993)
- 32 ±45% increase in infiltration due to one forklift entry and exit per minute (Downing and Meffert 1993)

- 79 to 85% with rapid roll doors and one forklift entry/exit per minute. The door was fully open for 6 to 10 seconds and took 1 to 2 seconds to open or close for each traffic pass. (Downing and Meffert 1993)

Pham and Oliver also investigated the effect of forklift traffic on an unprotected doorway. With one forklift entry and exit per minute, infiltration was decreased by 21% compared to an open door with no traffic.

Chen's testing was performed on a single-story refrigerated storage facility with dimensions of 78.7 x 118.1 x 23.0 ft. It operates between -4 and 3.2°F and is constructed of 7.87 inches of polystyrene insulated sandwich panels attached to a steel portal frame. The tested rapid-roll door was 7.87 x 10.83 ft in size, cycling open in 1.7 seconds and closing over a period of 3.1 seconds. The typical open period during passage of a forklift is 16.4 seconds when the door actuation is controlled by sensors. Tests were performed with door open times of 10, 20, 30, 40, and 50 seconds, with forklift traffic at 8 to 150 passes/hr.

Research on the lag time for flow through an open door to reach steady state includes work done by Azzouz and Duminil, who found that only 1.5 seconds are needed to reach steady state flow for a 9.19 x 9.84 ft doorway (Azzouz and Duminil 1993). This result was verified by Chen's testing, as it was determined that the lag time is less than 2 seconds.

Effectiveness values developed based on the research are presented in Table 59.

Table 59. Effectiveness of Different Infiltration Reduction Techniques on Rapid Roll Doors

	% Effectiveness
No protection	0%
No protection with traffic at 1 entry/exit per minute	44%
Strip curtain protection (good condition)	92%
Strip curtain protection (one strip missing)	75%
Strip curtain protection with traffic at 1 entry/exit per minute	62%
Door closed (air-tightness)	99%

Source: (Chen et al. 1999)

(Reindl and Jekel 2008)

Reindl discusses the use of CO_2 tracer gas for measuring infiltration in refrigerated spaces and its application to blast freezers. Blast freezers operate at very low temperatures ranging from -45 to -20°F. The opportunity of reducing freezer energy use by better understanding the infiltration rate has great potential (Reindl and Jekel 2008).

Previously noted infiltration rates for blast freezing systems range from 0.7 to 2.5 air changes per hour. For the facility tested in this article, the rate was calculated at an average of 1.58 air changes per hour, which is considered above an acceptable level.

Review of Commercial Refrigeration

Information gathered from 46 sources is presented in this section, all referring primarily to commercial applications of refrigeration. In this work, commercial refrigeration is defined as refrigeration systems serving an end user in a retail setting. Examples of this type system include a walk-in freezer at a grocery store, a display case at a gas station, or a refrigerated coffin case at a movie theatre.

Literature is organized based on its content and is categorized into seven groups, including refrigeration system design, general load details, ground temperature, evaporator application and defrost, lighting loads, infiltration loads, and ambient and box operating conditions.

Refrigeration System Design

(Knudsen and Pachai 2004)

The Technical University of Denmark did an energy comparison between CO_2 cascade systems and R404a systems (Knudsen and Pachai 2004). Four supermarkets were instrumented for data collection, including two conventional plants and two cascade plants. Data collected includes outside conditions, power consumption, compressor and condenser capacity utilization, suction and discharge temperature, expansion valve operation, superheat at the evaporator, and air temperature in the refrigerated space. The study concluded that cascade systems have power usage similar to a state of the art conventional system.

Information presented in the report that pertains to this investigation included walk-in box sizes. Medium temperature room volumes in the test stores were given as 1230, 844, 1117, and 1337 ft^3. Low temperature room volumes in the test stores were given as 343, 189, 421, and 309 ft^3.

(Walker 1992)

The field of commercial refrigeration is always trying to improve the efficiency of its refrigeration systems. Walker field tested a Safeway store in Menlo Park, CA and then applied different strategies of reducing energy usage (Walker 1992). Items utilized to reduce energy usage include high efficiency display case fan motors, anti-sweat heater controls, and vinyl strip curtains on doors (6 in wide vinyl strips connected above the door and hanging down to the bottom of the door). Both conventional and multiplex systems were tested. Test conditions are listed below:

Conventional Test Conditions:

- Outside Dry Bulb: 60.6°F average, 52.8°F minimum, 68.3°F maximum
- Outside Dew Point: 45.1°F average, 34.4°F minimum, 51.9°F maximum
- Outside Wet Bulb: 52.3°F average, 45.8°F minimum, 58.4°F maximum
- Inside Dry Bulb: 73.6°F average, 71.7°F minimum, 74.4°F maximum
- Inside Dew Point: 44.9°F average, 38.5°F minimum, 49.2°F maximum

Multiplex Test Conditions:

- Outside Dry Bulb: 62.4°F average, 54.7°F minimum, 68.4°F maximum
- Outside Dew Point: 48.2°F average, 44.7°F minimum, 54.7°F maximum
- Outside Wet Bulb: 54.7°F average, 50.3°F minimum, 57.8°F maximum
- Inside Dry Bulb: 72.7°F average, 71.3°F minimum, 74.9°F maximum
- Inside Dew Point: 45.7°F average, 43.1°F minimum, 47.5°F maximum

Conventional (stand-alone) refrigeration systems utilize a single semi-hermetic compressor for each display case circuit or walk-in box. The stand-alone system includes all the necessary refrigerant piping, control valves, receiver, electrical components, and air-cooled condenser mounted on the same base or skid. Head pressure is monitored by a liquid line thermostat set to a 90°F minimum condensing temperature and controlled by cycling the fan used by the condenser.

For medium and high temperature fixtures, off-cycle defrost is commonly used whereas for very low and low temperature applications, electric defrost is employed.

A wealth of information is presented on the test store walk-in boxes (Table 60), as is information on the compressor selected for the store.

Table 60. Refrigerated Walk-in Boxes in the Test Store

Description	Floor Area (ft^2)	Design Load (Btu/h)	Evaporator Temperature (°F)	Discharge Air Temperature (°F)
Meat cooler	480	26300	20	30
Produce cooler	437	21750	31	40
Dairy cooler	540	21750	27	36
Meat prep	800	48000	29	50
Fish cooler	96	7800	20	30
Meat holding	120	9200	21	30
Deli box	70	6275	26	37
Bakery retard	80	6400	27	36
Freezer	500	29900	-22	-12
Bakery freezer	81	7100	-22	-12

Source: (Walker 1992)

In addition to an analysis of refrigeration loads for low and very low temperature display cases, Walker presents an analysis of refrigeration loads for low temperature walk-in boxes. Results are presented in Table 61.

Table 61. Analysis of Low Temperature Walk-in Box Test Results

Description	Freezer	Bakery Freezer
Average Refrigeration Load (Btu/h)	36400	10400
Average Load per Box Area (Btu/h-ft^2)	72.7	128.4
Average Evaporator Temp. (°F)	-19.3	-16.9
Average Discharge Air Temp. (°F)	0.5	2.6
Dew Point Variation During Test (°F)	34.1 to 53.1, 45.3 average	
Dry Bulb Variation During Test (°F)	69.4 to 77.3, 73.0 average	
Load Variation due to Humidity (Btu/h)	3112	968
Load Variation due to Humidity (%)	8.6	9.3
Load Variation due to Dry Bulb (Btu/h)	179	769
Load Variation due to Dry Bulb (%)	0.5	7.4

Source: (Walker 1992)

(Patel et al. 1993)

Patel et al. performs a very involved analysis of commercial building appliances in this report for the United States Department of Energy (Patel et al. 1993). The goal of the report was to highlight opportunities for reduction in energy use in the country's infrastructure. As a next step, Patel et al. presented current technologies, and new and emerging technologies that may be applied to possibly reduce the energy usage.

Refrigeration systems discussed in this report in detail include central and unitary refrigeration systems. Unitary systems include display cases with all refrigeration equipment self-contained. Heat is rejected directly from the unit into the building interior air. Frozen food is maintained at a temperature of 0°F in these cases with an evaporator temperature of approximately -20°F and a COP around 1. Medium temperature coolers operate at interior temperatures between 35 and 40°F, with evaporator temperatures of approximately 20°F and a COP of around 1.75.

Unitary systems operate at an interior ambient dry bulb temperature of 90°F, as the condenser is located inside, coincident with the ANSI/AHRI Standard 1250/1251 (AHRI 2009b; AHRI 2009c) rating method. Typically, central refrigeration systems serve numerous applications at the same time, maintaining low, medium, and high temperature spaces. Major components of the system include the case or walk-in being cooled with evaporator; the compressor; and the air, water, or evaporative-cooled condenser. Mechanical components are typically located some distance from the case or walk-in being cooled, connected by refrigerant piping. Frozen food interior temperatures range from -5 to 0°F, with evaporator temperatures of ranging from -30 to -20°F. Medium temperature applications use 32 to 45°F air maintained by an evaporator coil operating at 10 to 20°F SST. Patel specifies that product loading in these units is typically low, as product is transferred to the case or walk-in at near storage temperature. Standard rating conditions for outdoor condensers used for central refrigeration systems are at 95°F dry bulb and 75°F wet bulb. In addition, it is typical for HFCF-22 to be used in both frozen food and refrigerated food and beverage unitary systems.

Of the total 2.9 Quads (Quad = 10^{15} Btu) of energy used in commercial buildings, refrigeration uses 26% (755 x 10^{12} Btu/yr). Refrigeration energy usage is second only to water heating, which consumes 33% of the total energy. Drilling down into refrigeration itself, Patel identified that central refrigeration systems used in supermarkets and for large walk-in refrigerators and freezers use 490 x 10^{12} Btu/yr (66% of refrigeration baseline) , display cases and other unitary refrigeration equipment use 149 x 10^{12} Btu/yr (20%), and ice makers use 105 x 10^{12} Btu/yr (14%). There is substantial opportunity to reduce the energy usage in commercial refrigeration. Current technologies are available, according to

Patel, that could reduce the energy used by 12% (91 x 10^{12} Btu/yr). Technologies are available in other industries which could be modified so that they could be applied to the refrigeration industry with a potential to reduce energy use by 24% (179 x 10^{12} Btu/yr). In the distant future, the researchers predict new technologies could cut energy use by 49% (373 x 10^{12} Btu/yr).

Central refrigeration systems are currently being installed with variable capacity control, floating head pressure control, water cooled condensers, and mechanical subcooling. There is untapped opportunity to reduce domestic energy usage by 76 x 10^{12} Btu/yr through application of these technologies across the board.

Technologies that could be quickly adapted to central refrigeration systems to improve efficiency include fully floating head pressure control and two-stage R22 with intercooling and subcooling. Application of these technologies has an energy reduction potential of 72 x 10^{12} Btu/yr. Unitary system energy use could be reduced by 79 x 10^{12} Btu/yr if high efficiency compressors, higher insulation levels, increased evaporator and condenser face area, and water cooled condensers were applied.

Potential advanced technologies for central refrigeration systems and unitary systems include advanced vapor cycle with multi-stage mechanical cooling and high-efficiency refrigerants (110 x 10^{12} Btu/yr potential for central refrigeration systems and 35 x 10^{12} Btu/yr for unitary systems), reverse Stirling cycle (141 and 25 x 10^{12} Btu/yr), advanced absorption cycle (246 and 75 x 10^{12} Btu/yr).

Refrigeration systems are expected to be operational all year. Due to variations in load however, the equipment is not always operating. Average runtimes are presented in Table 62 along with inventory, performance, and energy usage data. The column noted as

"U.S. Annual Energy Use" takes into account the conversion to primary energy input at the power plant using a conversion factor of 11,200 Btu per kWh.

Table 62. Unitary and Centralized System Inventory, Performance, and Energy Usage Data

System Type	Building Type	1990 Est. Unit Inventory	Unit Runtime (%)	Typ. Rated Cap (kW)	Typ. COP	1990 ADL Energy (10^8 kWh/yr)	U.S. Annual Energy Use (10^{12} Btu/yr)
Unitary	Office	275267	50	1.4	1.5	16.9	18.9
Unitary	Retail	576985	50	1.4	1.5	35.4	39.6
Unitary	Restaurant	540672	50	1.4	1.5	33.2	37.1
Unitary	Grocery	274371	70	1.4	1.5	23.6	26.4
Unitary	Schools	216089	40	1.4	1.5	10.6	11.9
Unitary	Hotel	122839	50	1.4	1.5	7.53	8.44
Unitary	Hospitals	93250	50	1.4	1.5	5.72	6.40
Central	Restaurant	125280	50	3	1.6	16.5	18.4
Central	Supermarket	187000	70	25	1.5	286	320
Central	Convenience	90000	70	5	1.8	55.0	61.6
Central	Hotel	256170	50	2.5	1.8	28.1	31.4
Central	Hospital	129643	50	3	1.6	17.0	19.1
Central	Warehouse	225841	50	3.5	1.3	34.6	38.8

Source: (Patel et al. 1993)

(Sezgen and Koomey 1995)

Commercial End-Use Planning System (COMMEND 4.0) is used by Sezgen and Koomey (Sezgen and Koomey 1995) to forecast refrigeration end uses. In addition to describing the model, the authors present commercial refrigeration information such as types of applications, types of refrigeration systems and their performance, and market data such as equipment market saturation.

Some information included in this report is directly applicable to the current study. Typical walk-in coolers operate at 35°F box temperature, 20°F evaporating temperature, and 70 Btu/h-ft^2 base refrigeration load for storage of meat, dairy, deli meats and cheese, and produce. Walk-in freezers typically operate at 0°F box temperature with a base refrigeration load of 90 Btu/h-ft^2. Ice cream freezers operate at a case temperature of -12°F with a -33°F evaporating temperature. Walk-in freezer application in groceries, restaurants, and warehouses is on the order of 14, 3, and 3 ft^2/1000 ft^2 of building, respectively. Walk-in cooler use is slightly higher at 45, 11, and 3 ft^2/1000 ft^2 of building in these same applications, respectively. Typical walk-in sizes are noted as 50 x 30 x 20 ft tall, 20 x 10 x 8 ft tall, and 250 x 200 x 40 ft tall for grocery, restaurant, and warehouse applications, respectively.

(Westphalen et al. 1996)

Primary energy use for commercial refrigeration is estimated at 990 trillion Btu annually, with supermarket refrigeration accounting for 33% of the energy use. Another 18% of commercial refrigeration energy use is from the operation of non-supermarket walk-ins. This report identifies opportunities for energy efficiency improvements and estimates savings for successful implementation of the options proposed (Table 63).

Table 63. Identified Energy Savings Potential

	Savings Potential ($x\ 10^{12}$ Btu)	Payback Range (yrs)
Evaporator Fan ECM Motor	85	0.5 to 3
ECM/Variable Speed Compressor	48	2 to 5
High-Efficiency Compressor	39	0.5 to 2
High-Efficiency Fan Blades	30	0.1 to 1
Condenser Fan ECM Motor	25	0.5 to 8
Floating Head Pressure	25	0.3 to 3
Electronic Ballasts	24	1 to 2.5
Non-Electric Anti-sweat	20	1 to 1.5
Thicker Insulation	20	1 to 1.5
Ambient Sub-cooling	12	2 to 11
Hot Gas Defrost	10	1.5 to 3
Liquid-Suction Heat Exchangers	10	4 to 14
Evaporative Condensers	10	--
Anti-sweat Heater Controls	10	2 to 6
Ice Machine Process Improvements	9	1 to 6
Evaporator Fan Shutdown	7	1 to 2
External Heat Rejection	6	7
Economizer Cooling	6	20
Heat Reclaim	3	2 to 5
Defrost Control	3	3
Mechanical Subcooling	2	5

Source: (Westphalen et al. 1996)

Supermarkets utilize numerous types of equipment to keep products at the proper storage temperature. Display cases and walk-ins operate via a centralized refrigeration system. Self-contained refrigeration systems are distributed throughout the store to supply cooling to merchandisers, certain display cases, and refrigerators/freezers similar to those used in a residence.

Some details presented about walk-ins include the use of 3 or more inches of insulation (R-27 typical) contained by steel or fiberglass panels. Freezer floors generally use insulation. Typically these units do not require anti-sweat heaters. Lighting (1 W/ft^2) in walk-ins is turned off when they are not being occupied, which can be half of the time. Evaporator fans typically have shaded-pole motors and operate 100% of the time. Refrigerants were historically CFC-12 and R-502, but this is transitioning to HFC blends, such as R404A. Merchandising doors are sometimes placed along one wall to allow customers to access product in the space. Some walk-ins utilize electric defrost. This defrost power usage may range from 19.5 to 29 W/ft^2. Typical loads for walk-ins include 60 Btu/h-ft^2 for coolers and 80 Btu/h-ft^2 for freezers.

In addition to being used in supermarkets, walk-ins are used in restaurants, convenience stores, cafeterias, food wholesalers, produce and fruit farms, small meat packagers, small ice cream companies, florists, research laboratories, and warehouses. The majority of these applications are stand-alone units with a unitary condensing unit (packaged system). Another option is a split system where the compressor and condenser are piped together in the field. Ten to sixty percent of these installations reject heat to the interior of the building as opposed to placing the condenser on the roof.

As part of this study, an analysis was made of a prototypical supermarket with two medium temperature racks (15°F SST/115°F SDT) and two low temperature racks (-25°F/110°F SDT). The refrigerant system condensers reject heat to an ambient temperature of 100°F. The prototypical freezer is 80 ft^2 and maintains a space temperature of -9°F, and the cooler is 24 x 10 ft, maintaining a 35°F space temperature. The cooler has an oversized compressor to assist in pulling down the temperature of product brought into the space. A detailed presentation of information on the prototypical freezer and cooler (Table 64) reveals that they are the same units used in the analysis (Westphalen et al. 1996).

Common storage temperatures for products include:

- High temperature: 35°F and above for produce and flowers.
- Medium temperature: 10 to 15°F for meats and seafood; 15 to 25°F for dairy, produce, beverage, and meat walk-ins; 25 to 35°F for dairy and produce walk-ins and prep rooms.
- Low temperature: -25 to -15°F for frozen foods.
- Very low temperature: -35 to -25°F for ice cream and frozen bakery.

Numerous opportunities to better utilize energy were identified for supermarkets. Current technologies that can be applied include evaporative condensers, floating or very low head pressure, ambient or mechanical subcooling, heat reclaim, hot gas defrost, variable speed motors for fans, liquid-suction heat exchanger, and anti-sweat heater controls. New technologies include electronic ballasts lighting, high efficiency fan motors, high efficiency fan blades, increased insulation, coil improvements, and defrost control. Advanced technologies include alternative refrigerants, engine-driven compressors, absorption refrigeration, and chemisorption refrigeration.

Opportunities for walk-in cooler and freezer energy reduction are visited separately from supermarket refrigeration. Current technologies that may be applied include hot gas defrost and thicker insulation. New technologies include floating head pressure, ambient subcooling, external heat rejection, economizer cooling, high efficiency fan blades and motors, electronic ballasts, anti-sweat heater control, and evaporator fan cycling. Advanced technologies that may be applied include non-electric anti-sweat and demand defrost control.

Table 64. Characteristics of the Baseline Cooler and Freezer

	Walk-in Cooler	Walk-in Freezer
Floor Size (ft^2)	240	80
W x D x H (ft)	24 x 10 x 8.5	8 x 10 x 7.58
Wall Thickness (in)	4	4
Wall R-Value	28.6	30
Merchandising Doors (ft)	(10) 2 x 6.14	
Access Door W x H (ft)	3 x 6.5	3 x 6.5
Refrigerant	R404A	R404A
Compressor HP	5	1.25
Compressor Type	Semi-Hermetic Recip.	Semi-Hermetic Recip.
Ambient Temperature (°F)	95	90
Walk-in Temperature (°F)	35	-10
Condensing Temperature (°F)	105	113
Evaporating Temperature (°F)	35	-26
Compressor Capacity (kBtuh)	45.0	4.9
Compressor Power (W)	3850	1445
EER (Btu/W)	11.7	3.41
Evaporator Fan Power (W)	880	180
Condenser Fan Power (W)	1060	329
Liquid Suction Heat Exchanger	Y	Y
Anti-sweat Wattage (W)	300	230
Defrost Wattage (W)		1500
Pan Heater Wattage (W)		500
Defrost and Pan Heater Control		Time Initiated / Temp Term.
Percentage of Total Load (%)		
Evaporator Fans	30	27
Coil Defrost	-	10
Pan Heater	-	3
Lighting	6	5
Wall Conduction	14	48
Glass Door Conduction	45	-
Infiltration	5	7

Source: (Westphalen et al. 1996)

(Sami and Tulej 1996)

Sami and Tulej (Sami and Tulej 1996) analyzed HFC23/HCFC22/HFC152A as a substitute for CFC12, CFC502, and HCFC22 in regard to environmental impact, ozone depletion potential, global warming potential, flammability, toxicity, and performance through testing of six walk-in facilities located at Springhill and Dorchester, Canada.. Energy performance and system coefficient of performance are compared, specifically.

Springhill walk-ins included a garbage cooler, meat cooler, bakery freezer, and bakery cooler. The garbage cooler operated with a 9,405 Btu/h capacity evaporator at 30°F evaporating temperature and 15°F temperature difference, utilizing off-cycle defrost. The meat cooler operates with a 16,900 Btu/h electric defrost evaporator at 20°F evaporator temperature. The bakery freezer and cooler each operate with an 8,400 Btu/h electric defrost evaporator at 20°F evaporator temperature. The Dorchester site walk-ins have no information given regarding the system type or operation.

(Sand et al. 1997)

This report compared current and alternative refrigeration, air conditioning, and appliance technologies with regard to the total equivalent warming impact (TEWI), safety, health concerns, initial and operating costs, regional energy considerations, and ease of maintenance.

Current supermarket refrigeration systems require a significant amount of refrigerant, corresponding to high leakage rates. Supermarket alternatives to CFCs and HCFCs include mixtures of HFCs. Direct expansion is the traditional design, with alternatives available such as secondary loop and distributed systems. Secondary loop systems pump cold brine to display cases. This system reduces refrigerant charge (down to 10% of a conventional system in some cases), reduces refrigerant leakage rate, and isolates refrigerant from the sales floor. Distributed systems break the system into smaller units that serve cases in close proximity.

High temperature refrigeration for air conditioning and for cooling of product prep rooms is at air temperatures of around 50°F. Medium temperature refrigeration is used to keep meat, fish, and dairy cases and walk-in coolers for meats and produce at air temperatures from 28 to 45°F. Low temperature refrigeration refers to freezers and ice cream cases and walk-ins. Temperatures for these applications range from -25 to 0°F.

In order to compare system performance, a number of assumptions were made that were thought to be typical for a supermarket application. Evaporator temperatures were assumed to be fixed for all medium and low temperature cases at 20 and -25°F , respectively. Condenser calculations are done using a constant temperature of 97°F for an entire year based on weather data from three locations.

Typical energy usage of supermarket refrigeration systems in the U.S. is 300,000 Btu/h for low temperature refrigeration and 900,000 Btu/h for medium temperature refrigeration. The average refrigerant charge for a supermarket is 2000 to 2500 lbs.

Historically, CFC-12, HCFC-22 and R-502 were used in supermarkets, but new installations typically use the following alternatives. Medium temperature refrigerants include R-404A, R-507, R-134a, R-407a, and R-410A. Low temperature alternatives include R-404A, R-507, R-407A, and R-407C. TEWI values are compared for these refrigerants in Table 65.

Table 65. TEWI for Low and Medium Temperature Direct Expansion Refrigerant System in North America

	TEWI		
	Direct Expansion	Distributed	Secondary Loop
Low Temperature			
R-404A	3,923,600	2,130,200	2,207,200
R-507	3,922,711	2,130,200	2,207,200
R-407A	3,112,100	2,107,900	2,241,300
R-407C	2,980,000	2,152,000	2,301,500
R-717	--	--	2,096,252
Medium Temperature			
R-404A	5,497,300	3,430,300	3,422,400
R-507	5,427,500	3,356,100	3,345,500
R-134a	4,161,400	3,392,700	3,473,800
R-410A	4,125,900	3,196,700	3,260,100
R-717	--	--	3,253,100

Source: (Sand et al. 1997)

(Kimber 1998)

Nevada Energy Control Systems, Inc. has configured a product to control evaporator operation, reducing energy costs while maintaining system cooling performance. Kimber (Kimber 1998) conducted field testing of installations (listed below) of these controllers to determine the energy usage improvement due to their implementation.

- Safeway Store #309, Fremont, CA
- Trader Joe's Store #70, Sacramento, CA
- McDonald's Restaurant, Stockton, CA
- Walnut Creek School District, Walnut Creek, CA
- Cameron Park Liquors, Cameron Park, CA

Provided information included average daily energy consumption of the evaporator fans and the compressor, the number and duration of personnel door openings, the ambient temperature and humidity inside and outside of the walk-in box, and the compressor duty cycle data.

(Walker 2001)

Walker discusses the currently accepted refrigeration system design, and its drawbacks. He discusses advanced systems and their benefits and drawbacks. A low charge refrigeration system which is integrated with the HVAC system is analyzed and compared to currently available systems. Comparisons are made on a basis of energy consumption and total equivalent warming impact (TEWI). The combined energy usage of the store refrigeration and HVAC systems are also compared, framing the full picture of the efficiency of a refrigeration system.

Supermarkets are the largest commercial electricity end-user, with a typical store (35,000 ft^2) accounting for 2 million kWh annually. Of this 2 million kWh, a significant portion is due to refrigeration, comfort cooling, and heating (10-20 % of the store energy usage is HVAC). Typically, multiplex systems are utilized to gain the optimum efficiency, flexibility, and reliability. With a heightened focus on the total environmental impact of refrigerated systems, some supermarkets have moved toward advanced systems, which use less refrigerant but often sacrifice energy efficiency, including low-charge multiplex, secondary loop, distributed, and self-contained.

There is significant opportunity to improve current advanced system designs to improve energy efficiency. Shorter suction lines on secondary loop, distributed, and advanced self-contained systems can translate into lower compressor power usage. Low head pressure operation to reduce energy usage (Walker 2001) may be achieved with distributed and self-contained systems utilizing scroll compressors, and self-contained systems with capacity control.

HVAC system operation and energy usage is directly related to the type of refrigeration system that is being utilized. During space cooling, the refrigeration system is removing heat from the supermarket with exception of the heat convected from the equipment in the machine room. Space heating is the dominant HVAC load because the refrigeration equipment in the store continues to operate, combatting the space heating. Reclaiming waste heat (14-20%) from the refrigeration system may be done by desuperheating discharge gas. This is typically used for water or space heating. Additional reduction in space heating load may be achieved by utilizing a water source heat pump to claim the full potential of the refrigeration system heat rejection and apply it to space heating.

Walk-ins in supermarkets are constructed of pre-fabricated panels that are field assembled. Doors are wide enough to allow pallet jack passage, which is typically accompanied by the door being left open for the loading or unloading period. Infiltration is the largest load for these units, combated in some cases through the use of vinyl strips hung in the doorways. In a supermarket, walk-ins are typically used to hold a specific product type. Meat and produce coolers are also used to prepare food for display.

Multiplex refrigeration uses centralized racks of compressors to serve various cases and walk-ins. Low temperature racks typically operate at -20°F SST, medium temperature racks typically operate at 20°F, and satellite compressors are used to serve off-temperature refrigeration. Each rack is composed of three to four compressors, allowing for capacity control, for reduced energy use and more stable operating conditions. Reciprocating compressors allow operation at a saturated discharge temperature of 70°F. Utilizing scroll compressors would allow operation at discharge temperatures of 40 and 60°F for low and medium temperature refrigeration, respectively, for a potential compressor energy savings of 10% (Walker and Deming 1989) when coupled with condenser bypass controls which also reduce the required refrigerant charge (up to 1/3 reduction in charge).

Compressors are typically reciprocating, semi-hermetic due to relative reliability and energy efficiency. Scroll compressors are typically only used for distributed systems where noise levels are a concern. Condensers are air cooled in most cases, 8 to 10 fpi with a 500 fpm face-velocity, and use multiple fans for capacity control. Evaporative-cooled condensers offer reduction in energy use but come with disadvantages of water treatment and consumption.

Distributed systems are composed of compressor cabinets with racks of scroll compressors which are configured to allow capacity control. This system's benefits include better saturated suction temperature matching, reduced pressure losses through close coupled equipment (SST at 1 to 2°F lower than the saturated temperature at the evaporator, compared to 2 to 4°F for a multiplex system; and return gas rise of 5 to 15°F, compared to 40 to 65°F for a multiplex system) , reduced refrigerant charge (810 to 900 lb per store), and lower minimum condensing temperature of 55 to 60°F. A store-encompassing glycol/water (25 to 33% glycol) loop (300 to 350 gpm) removes heat from each distributed cabinet. Evaporative fluid coolers are often used with this type system to help overcome the additional temperature differential added to the water loop by heat exchange from the distributed refrigeration cycles.

Secondary loop systems utilize a centralized direct expansion compressor rack and a brine or water loop (4 to 6 ft/s, 300 to 500 gpm) that puts the load of the refrigerated cases and walk-ins on the compressors. Multiple loops allow for multiple saturated temperatures for the evaporators in the cases or walk-ins. A hot secondary fluid may be used for defrosting coils which shortens defrost time and reduces energy use (1/2 of defrost energy usage for multiplex systems).

Typical parameters of multiplex, distributed, and secondary loop refrigeration systems were discussed above. Simulations performed by Walker utilized the parameters noted in Table 66. In addition to the values noted in the table, the simulated secondary loop refrigeration system used a fluid temperature difference of 7°F and a chiller approach (fluid leaving – evaporator) of 5°F. Simulated multiplex and distributed refrigeration systems used hot gas defrost for low temperature refrigeration and off-cycle for medium temperature. The secondary loop system uses warm glycol for low and medium temperature case and walk-in defrost.

Table 66. Parameters Used For Refrigeration System Analysis

System	System Pressure Drop (T_{evap}-SST, °F)	Return Gas Temperature (°F)	Min Condensing Temp (°F)	Liquid Refrigerant Temperature (°F)	
				No Subcool	Subcooled
Multiplex	3	45	70	T_{con} - 10	45
Low-charge Multiplex			40 LT, 60 MT		
Distributed	2	T_{evap} + 15	60	T_{con} - 10	(T_{con}–15 to 20)
Secondary	0	T_{evap} + 10	70	T_{con} - 10	40

Source: (Walker 2001)

Self-contained refrigeration systems utilize small reciprocating compressors and unit mounted air-cooled condensing units that reject display case heat into the store. These systems require substantially less refrigerant (estimated 100 to 300 lb for a typical store) and have lower leakage levels due to the benefits of factory assembly. Drawbacks include noise and heat added to the sales floor. This system could be improved by water cooling each self-contained unit with a fluid loop and using scroll compressors to reduce noise levels.

Eighty percent of the load on refrigerated cases is sensible and latent loads removed from the sales floor. During space heating, the HVAC system must resupply this heat continuously to the store. During space cooling, the HVAC system load is reduced by this amount. Typically, equipment used for supermarket HVAC include vapor compressor air conditioners, and gas duct heaters for space heating.

For analysis of different refrigeration – HVAC systems, a load factor (Equation 101) is applied to the design load at temperatures between 40 and 85°F to determine the refrigeration load for each temperature bin. The temperatures of 40 and 85°F are selected because they typically frame suitable conditions for maintaining an indoor temperature from 68 to 75°F.

$$F_{load} = 1 - \left(1 - F_{\min}\right)\frac{85 - T_{amb}}{85 - 40} \qquad (101)$$

Table 67. Heat Rejection System Specifications for Refrigeration Modeling

System	Temperature Difference (T_{cond}-T_{amb})	Design Load/Fan Power (Btu-hr/kW)
Air-Cooled Condenser	LT TD = 10°F MT TD = 15°F	34000 LT 65000 MT
Evaporative Condenser	TD_{WB} = 12°F	70500 Condenser fan and spray pumps
Water-Cooled Condenser Wet Fluid Cooler	TD_W=10°F A_{TWB}=12°F	70500 Tower fan and spray pumps
Water-Cooled Condenser Dry Fluid Cooler	TD_W=10°F A_{TDB}=12°F	65000 Tower fan only

Source: (Walker 2001)

Numerous condenser schemes are listed in Table 67. Air-cooled condensers are most common because of the lack of maintenance required. Liquid temperature on non-subcooled systems is assumed to be 10°F less than the condensing temperature. Mechanically subcooled systems typically have liquid temperatures of 40°F. Condenser fans operate continuously until the minimum condensing temperature is achieved. Fan energy during operation below the minimum condensing temperature is estimated by multiplying installed fan power by a fan factor defined by Equations 102 and 103.

$$\text{If } T_{amb} \geq 30°\text{F}, \; C_{fan} = 1 - 0.75\left(\frac{T_{con} - TD_{DB} - T_{amb}}{T_{con} - TD_{DB} - 30}\right) \quad (102)$$

$$\text{If } T_{amb} < 30°\text{F}, \; C_{fan} = 0.25 \quad (103)$$

Compressor performance data is provided by manufacturers at specific rating conditions. To adjust for a new condition, apply the correction factor listed as Equation 104.

$$C_{cap} = \frac{\upsilon_{Cr}}{\upsilon_C} \frac{\left(h_{1e} - h_{4e}\right)}{\left(h_{1e,r} - h_{4e,r}\right)} \qquad (104)$$

The refrigeration system modeled for the analysis included the cases and walk-ins listed in Table 69. R404a was used for the low temperature racks in the multiplex system and all of the systems in the distributed refrigeration system. R22 was used for the medium temperature racks in the multiplex system. The distributed refrigeration system utilizes 10 different mechanical cabinets, resulting in closer matching of the design saturated suction temperature. The secondary loop refrigeration system used four loops at -20, 10, 20, and 30°F loop temperatures, each with their own chiller system. Low temperature loops use Pekasol 50 whereas medium temperature loops use a propylene glycol/water mixture. The refrigerant used in the centralized mechanical room is R507.

Locations chosen for the refrigeration analysis include Worchester, MA; Washington, DC; Memphis, TN; and Los Angeles, CA (Table 68). Indoor design conditions are 75°F and 55% relative humidity. Load considerations applicable to display cases and walk-ins include conduction, solar roof loading, fan heating, ventilation, infiltration, occupancy, lighting, and miscellaneous loads. Loads are analyzed in the report for the macro-store level, with walk-in and case loads developed independently (Table 69).

Table 68. Design Ambient Conditions for Supermarket Analysis

Location	Dry-bulb (°F)	Wet-bulb (°F)	Specific Humidity (lb/lb)
Worchester, MA	88	77	0.0176
Washington, DC	93	78	0.0174
Memphis, TN	98	80	0.0180
Los Angeles, CA	93	72	0.0120

Source: (Walker 2001)

The analysis compared systems, including multiplex-air cooled condenser, low charge multiplex-air cooled condenser, low charge multiplex-evaporative-cooled condenser, distributed-air cooled condenser, distributed-water cooled via fluid cooler, secondary loop-evaporative-cooled condenser, and advanced self-contained-water cooled via fluid cooler. Lowest energy usage for the Worchester and Washington climates was with the distributed systems. Lowest energy usage for Memphis and Los Angeles was with a secondary loop system with evaporative condensing. The low charge multiplex, distributed, and secondary loop refrigeration systems were very close to each other in energy usage. The energy benefits of low-charge refrigeration systems are highlighted in Table 70.

Table 69. Description of Modeled Cases and Coolers

Sys.	Description	Length No. of Doors or Floor Size (ft)	Discharge Air Temp (°F)	Evaporator Temp (°F)	Design Load (Btu/h)
1	Ice Cream Door Cases	17 doors	-12	-19	26180
2	Ice Cream Door Cases	17 doors	-12	-19	26180
3	Ice Cream Door Cases	19 doors	-12	-19	29260
4	Ice Cream Door Cases	20 doors	-12	-19	30800
5	Frozen Meat	28	-10	-20	12100
6	Grocery Freezer	42 x 15	-10	-20	39900
7	Frozen Fish	12	-12	-19	3300
9	Frozen Food Door Cases	15 doors	-5	-11	21375
10	Frozen Food Door Cases	16 doors	-5	-11	29300
11	Frozen Food Door Cases	15 doors	-5	-11	21375
12	Frozen Food Door Cases	16 doors	-5	-11	22800
13	Bakery Freezer	12 x 10	-5	-11	9800
14	Deli Freezer	8 x 10	-5	-11	7000
16	Meat Cases	40	24	15	54800
17A	Meat Cases	36	22	17	15120
17B	Fish Cases	12	22	17	12660
18	Meat Cases	30	24	18	12580
19A	Bakery Cooler	6 x 8	28	18	10560
19B	Deli Cases	20	32	18	29900
19C	Deli Cases	8	30	25	3360
19D	Deli Cases	16	26	20	7040
19E	Deli Cooler	12 x 11	36	29	9100
19F	Cooler	6 x 8	25	15	10560
19G	Cheese Cooler	6 x 14	25	15	18480
20A	Deli cooler	6 x 8	25	15	10560
20B	Deli/Meat Cases	32	32	18	46880
21	Meat Cooler	40 x 15	30	22	35800
22	Meat Box	15 x 10	30	22	11400
24A	Dairy/Deli Cases	36	32	21	54000
24B	Dairy/Deli Cases	40	32	21	60000
25A	Dairy	12	32	18	18000
25B	Beverage	36	32	18	53820
25C	Beverage	24	32	18	36000
26A	Produce	24	32	21	35880
26B	Produce	56	39	21	43120
26C	Produce	82	39	21	62440
26D	Floral Cooler	12 x 11	38	32	11675
26E	Floral Case	13		21	12350
27A	Bakery Retarder	12 x 8	36	28	7300
27B	Dairy Cooler	40 x 20	36	28	54125
28A	Meat Prep	40 x 15	50	35	74250
28B	Produce Cooler	36 x 14	38	30	27200

Source: (Walker 2001)

Table 70. Low-Charge System Energy Savings for Supermarket Analysis in Washington, DC

System	Heat Rejection Method	% Savings
Multiplex	Air-Cooled Condenser	--
Low-Charge Multiplex	Air-Cooled Condenser	4.3
Low-Charge Multiplex	Evaporative Condenser	11.6
Distributed	Air-Cooled Condenser	11.9
Distributed	Water-Cooled Condenser, Evaporative Rejection	11.3
Secondary Loop	Evaporative Condenser	10.4
Advanced Self-Contained	Water-Cooled Condenser, Evaporative Rejection	-7.3
Secondary Loop	Water-Cooled Condenser, Evaporative Rejection	1.8

Source: (Walker 2001)

Energy efficiency is not the only contributing factor for selection of a refrigeration system. The total equivalent warming impact (TEWI) for each system is summarized in Table 71.

Table 71. Total Equivalent Warming Impact for Analyzed Systems in Washington, DC

System	Heat Rejection Method	Charge (lb)	Refrig.	Leak (%)	Annual Energy (kWh)	TEWI (10^4 ton CO_2)		
						Direct	Indir.	Total
Multiplex	Air-Cooled	3000	R404a/	30	976800	15.01	10.49	25.50
	Evaporative	3000	R22	30	896400	15.01	9.63	24.64
Low-Charge	Air-Cooled	2000	R404a/	15	935200	5.00	10.05	15.05
Multiplex	Evaporative	2000	R22	15	863600	5.00	9.28	14.28
Distributed	Air-Cooled	1500	R404a	10	860500	3.67	9.24	12.91
	Water-Cooled	900	R404a	5	866100	1.10	9.30	10.40
2ndary Loop	Evaporative	500	R507	10	875200	1.25	9.41	10.66
	Water-Cooled	200	R507	5	959700	0.25	10.32	10.57
Self-Contained	Water-Cooled	100	R404a	1	1048300	0.02	11.27	11.29

Source: (Walker 2001)

By combining the energy usage of the refrigeration system and the supermarket HVAC system and by taking into account their relationship, Walker developed a comparison of the combined operating cost of multiplex, multiplex with heat reclaim, and distributed with water source heat pump systems (Table 72). It is not surprising that the distributed system integrated into a WSHP has much lower energy usage compared to a multiplex system integrated with an HVAC system, as the distributed system showed an 11.3% savings compared to the multiplex system in the previous analysis (Table 70).

Table 72. Estimated Annual Combined Operating Cost of Supermarket Refrigeration and HVAC Systems in Washington, DC

	Annual Operating Cost ($)				
	Energy:		Physical Use:		
System	Refrig.	HVAC	Refrig.	Water	Total
Multiplex	69353	33869	3375	0	106597
Multiplex and Heat Reclaim	72193	27297	3375	0	102865
Distributed and Water Source Heat Pump	61493	22669	349	1616	86127

Source: (Walker 2001)

(Nagaraju et al. 2001)

A unique photo-voltaic cold store, utilizing R-12 refrigerant, was tested by Nagaraju et al. (Nagaraju et al. 2001) to analyze its performance. The 750.9 ft^3 (15.1 x 7.1 x 7.1 ft tall) facility maintains an interior temperature of 5°F to preserve 10 tons of fish. It is located inside another building, which maintains a temperature less than 86°F although the temperature does vary. Calculated loads include infiltration at 842.8 Btu/h (0.5 ACH) and a product load of 22,046 lb at 0.9°F temperature drop per hour, equivalent to 8,632.7

Btu/h (0.435 Btu/lb-°F). Nagaraju et al. concludes by stating that the system performed according to its intended design, meeting the cold store demands.

(Marchese 2002)

Marchese gives a general discussion about sizing walk-in coolers and freezers, as it pertains to commercial refrigeration. Included in this discussion are the typical loads that compose the total refrigeration demand to maintain the desired operating conditions of a refrigerated space. Loads include heat transmission, air infiltration, product load, and supplemental loads added by occupants and equipment operating in the space.

(Walker and Baxter 2002)

Multiplex refrigeration systems' display cases and walk-ins are served with refrigerant from a central plant, with multiple racks operating at various saturation temperatures. These compressor racks are cooled by a centralized condenser(s) mounted above the machine room, where the racks are housed. Multiplex systems are currently the most sought after type of refrigeration system due to their efficiency and performance advantages over a conventional refrigeration system. The main drawback, that supermarkets have been willing to except up to this point, is the amount of refrigerant charge required for these systems. Distributed refrigeration systems reduce the required refrigerant charge by locating direct expansion systems to serve display cases and walk-ins in local proximity. Heat rejection is done via water cooled condensers which are plumbed to air cooled condensers on the roof. A few drawbacks to this system type include higher compressor power, as a less efficient scroll compressor must be used to achieve lower noise levels and increased operating condensing temperatures due to the need for a fluid loop, which adds an additional temperature rise.

Two Price Chopper supermarkets in Marlborough, MA were tested to compare the actual performance of multiplex and distributed supermarket refrigeration systems. Both stores consisted of 52,000 ft^2 of floor space and were newly constructed, opening in 1997. The multiplex store that was tested as part of this work had air-cooled condensers, R404a for the low temperature rack, and R22 for the two medium temperature racks. As is typically done, the low temperature air cooled condenser was sized for a 10°F temperature difference and the medium temperature condenser was sized for a 15°F temperature difference, both with a minimum condensing temperature of 70°F . The design loads were 422,825 and 899,953 Btu/hr for the low and medium temperature racks, respectively, with one of the medium temperature racks pulling a subcool load off the low temperature rack of 361,670 Btu/hr.

The distributed refrigeration store uses three water-source heat pumps for store HVAC that reclaim waste heat from the refrigeration system. The refrigeration system rejects heat to the water loop for store heating when needed and the heat pumps reject heat back to the fluid loop during store space cooling. The minimum condensing temperature was set at 60°F to maintain the 50°F fluid loop temperature required to meet store heating demands. The refrigeration system is composed of 10 compressor cabinets located out in the store with loads of 404,845 and 1,010,936 Btu/hr for low and medium temperature refrigeration, respectively.

Energy (Table 73) and TEWI (Table 74) comparisons were made of the two systems. Because the distributed system is tied into the store spacing cooling/heating system, a more complete analysis would have taken into account the overall energy performance of each store. The distributed system uses more energy over the course of a year but has a much smaller impact on the environment.

Table 73. Energy Consumption Comparison Between Multiplex and Distributed Refrigeration Systems

	Energy Consumption (kWh/day)			
	Distributed	Multiplex	Difference	% Difference
May to August				
Low Temperature Compressor	1306.4	1290.2	16.2	1.2
Medium Temperature Compressor	1594.0	1201.2	392.8	24.6
Heat Rejection	702.7	608.4	94.3	13.4
Total	3603.2	3100.5	502.7	14.0
Nov. to February				
Low Temperature Compressor	863.2	957.9	-94.7	-11.0
Medium Temperature Compressor	951.3	635.9	315.4	33.2
Heat Rejection	316.1	364.4	-48.3	-15.3
Total	2130.5	1958.2	172.3	8.1

Source: (Walker and Baxter 2002)

Table 74. TEWI Analysis for the Field Test Results

	Refrigerant Leakage (lb)		Energy (kWh)	TEWI (lb CO_2)		
	R-404a	R-22		Direct	Indirect	Total
May to August						
Multiplex	67	133	381300	445455	546405	991860
Distributed	17		443169	54884	635064	689945
Difference						299930
Nov. to February						
Multiplex	67	133	234960	445455	336699	782154
Distributed	17		255600	54884	361633	416517
Difference						365637

Note: Multiplex leak rate estimated at 20%/yr, Distributed leak rate estimated at 5%/yr.

Source: (Walker and Baxter 2002)

(Christensen and Bertilsen 2004)

Due to growing demand for refrigeration systems that have a lower impact on the environment, there are numerous new types of systems being implemented across the world. This article discusses a 7750 ft^2 supermarket (small compared to a typical 25,000 ft^2 U.S. supermarket) in Fakta Beder, Denmark that implemented a propane-CO_2 cascade refrigeration system. Propane is used as the refrigerant for the high temperature level, with a 6.8°F evaporating temperature and a 77°F condensing temperature. Carbon dioxide is used as the refrigerant at the low temperature level, which operates at a -25.6°F evaporating temperature and a 14°F condensing temperature. The low temperature carbon dioxide loop and a medium temperature brine loop operating from 17.6 to 28.4°F, reject heat onto the propane circuit, which uses a dry cooler to reject heat to the atmosphere.

Dimensions, air temperatures, and capacities of the walk-ins and refrigerated cabinets in the supermarket are given in Table 75. The energy used by this refrigeration system accounted for about 56% of the total energy usage in July of 2004.

Table 75. Walk-in and Display Cabinet Details for Denmark Supermarket

	Air-On Temperature (°F)	Capacity (Btu/h)	Dimensions, LxWxH (ft)
Glass door walk-in cooler	28.4	15,355	16.7 x 10.8 x 8.5
Glass door walk-in freezer	-14.8	14,331	11.8 x 10.8 x 8.0
Glass door walk-in ice cream freezer	-18.4	9,554	6.9 x 10.8 x 8.0
Open cooler display case	28.4	49,135	32.8 in length
Doored freezer display case	-16.6	7,507	12.3 x 3.3 x 3.1
Doored freezer display case	-16.6	2,730	6.4 x 3.3 x 3.1
Doored cooler display case	28.4	4,777	12.3 x 3.3 x 3.1
Doored cooler display case	28.4	2,388	6.4 x 3.3 x 3.1
Total		105,777	

Source: (Christensen and Bertilsen 2004)

Christensen concluded that this system performed at an efficiency level equivalent to a conventional R404a system. The system's capital costs were, however, 12 to 20% greater than a conventional system installation. Larger systems, which are more typical, would likely be only 10% more expensive than a conventional system.

(Sekhar et al. 2004)

Sekhar et al. (Sekhar et al. 2004) analyzed the performance of an ozone friendly mixture of HFC134a, HC600a, and HC290 in a walk-in cooler. Results were compared to R-12, with the mixture using 28.6% less energy (6-10% improved COP). Testing was done on a 7.55 x 7.55 x 9.19 ft tall walk-in with an average interior temperature of 38.8°F. The unit was insulated with 5.91 inches of mineral wool.

(Sekhar and Lal 2005)

In a continuation of previous work (Sekhar et al. 2004), Sekhar and Lal analyzed the performance of a blend of HFC134a, HC600a, and HC290 and a conventional CFC refrigerant (R-12) (Sekhar and Lal 2005). The refrigerants that compose this blend have their own drawbacks when utilized in a system independently. When combined per this investigation, the refrigerant mixture in medium and low temperature applications used 10 to 30% and 5 to 15% less energy than R-12, respectively.

Testing was performed on domestic refrigerators, deep freezers, vending machines, and walk-in coolers. The walk-in cooler used was the same unit used in his previous work and operated at the same setpoint temperature (Sekhar et al. 2004).

(Arias 2005; Arias and Lundqvist 2006)

Supermarkets use a significant amount of energy, through necessity, to keep products cold during storage. In Sweden, supermarkets use about 3% of the nation's energy. Forty seven percent of this energy is used for refrigeration. The industry is always looking for methods to improve the operation of equipment in such a way as to reduce energy consumption while maintaining the performance and flexibility of today's supermarket multiplex systems. Two new methods for achieving this include the use of heat recovery and floating condensing temperatures.

Drawbacks to heat recovery systems include requiring a minimum liquid temperature to be maintained (100 to 106°F) and calling for the greatest load during the winter, when the refrigeration system is operating at a reduced load. Operating the refrigeration system to meet these needs reduces the efficiency of the system but when coupled with the building heating, the store overall energy usage is reduced.

Floating condensing systems allow the condensing temperature to follow the ambient conditions. At lower outdoor temperatures, the system operates at decreased energy consumption due to the lower system pressures. This type system requires sophisticated controls and different components, such as special valving, to work effectively. Arias developed a computer model to analyze energy efficiency, environmental impact, and economy of refrigeration systems. The model was validated with field testing.

Field testing was conducted at supermarkets in the cities of Sala, Hjo, Farsta Centrum, Hedemora, Västerås, Taby Centrum, and Kista Centrum. Provided supermarket test site information is included in Table 76. Typical store conditions are noted as 71.6°F and 65% relative humidity.

Table 76. Field Test Supermarket Details

	#1	#2	#3	#4	#5	#6	#7
City	Sala	Hjo	Farsta Centrum	Hedemora	Västerås	Täby Centrum	Kista Centrum
Area (ft2)	29,063	7,750	19,375	21,528	23,681	29,063	29,063
Refrigerant System Type	Cascade, secondary	Cascade, secondary and distributed	Distributed w/chiller	Parallel, secondary	Cascade DX and secondary loop	Cascade secondary distributed and self-contained	Cascade secondary distributed and self-contained
Low Temp. (tons)	9.95	3.70	56.87 combined	8.25	9.10	9.10	5.97
Med.Temp. (tons)	36.96	17.06		21.33	36.96	54.03	39.81
Cooler Disp. Cases	10	10	29	6	14	11	15
Freezer Disp. Cases	3	4	8	3	5	5	6
Walk-ins	7	7	11	11	5	8	10
Med.Temp Refrigerant	R404A	R404A	R404A	R404A	R404A	R404A	R404A
Low Temp Refrigerant	R404A	R404A	R404A	R404A	R404A	R404A	R404A
Brine 1	Pekasol 50/60	Ethyl alcohol	Ethyl alcohol	Propylene Glycol	Propylene Glycol	Propylene Glycol	Ethyl alcohol
Brine 2	Pekasol 50/90	Ethyl alcohol	Pekasol 50/90	CO_2			
Coolant Fluid	Pekasol 50/60			Propylene Glycol	Propylene Glycol	Propylene Glycol	Propylene Glycol
Heat Recovery	Y	Y	N	Y	Y	Y	N

Source: (Arias 2005)

For simulation purposes some energy gain assumptions were made. Lighting was assumed at 1.4 W/ft^2 and heat gain from people was assumed at 682 Btu/hr. Infiltration calculations utilized an air change method using data from Granryd (Granryd 2003), presented in Table 77.

Table 77. Number of Air Exchanges from Experiments

Room Volume (ft^3)	Number of Air Changes per 24 Hours	
	Cooler	Freezer
247	30.0	38.0
353	24.5	31.5
706	17.0	21.5
1,413	11.5	14.5
3,531	7.0	9.0
17,657	2.7	3.5
35,315	2.7	2.5
105,944	1.05	1.35

Source: (Granryd 2003)

(Zhang 2006)

Supermarkets use a large amount of energy, consuming 8.7 kWh/ft^2 on average. Most commonly, a parallel rack direct expansion is used for an HFC refrigerant, such as R404a, in quantities of 3100 to 5100 lb. Efforts to reduce environmental impact of such facilities have led to the development of new types of supermarket refrigeration systems such as distributed, self-contained, glycol secondary loop, and CO_2 secondary loop/cascade systems. Distributed systems are similar to parallel rack, with the exception that distributed systems have multiple racks spread throughout the store. Each rack serves a group of walk-ins and/or display cases that are in close proximity to it, reducing refrigerant charge and piping lengths of the system. Self-contained systems are display cases or walk-ins that have their own compressor and condenser, cooled by water that loops the store and then exchanges the heat with the atmosphere through a rooftop unit. This system reduces refrigerant charge but sacrifices efficiency of operation. Secondary glycol systems have a centrally located direct expansion system that cools a glycol circuit. The glycol circuit loops the store to cool the refrigerated spaces. Because of the additional steps of heat exchange, this system also results in an increased energy cost. Secondary systems using CO_2 in place of glycol have much higher efficiency and therefore lower energy cost, by taking advantage of a phase change. The primary drawback to this system is its capital cost.

Zhang developed models to analyze each of these system types. Outputs compared include annual energy consumption, Total Equivalent Warming Impact (TEWI), and life cycle cost. Comparative studies include work done by Walker and Baxter (Walker and Baxter 2003) and Arias and Lundqvist (Arias and Lundqvist 2005).

Two typical supermarkets (30139 and 54896 ft^2) were specified for modeling the different systems. Both supermarkets had display cases and walk-ins with an indoor condition of 75.2°F (55% relative humidity). The refrigeration load is varied based on the outdoor temperature's effect on interior conditions utilizing Equation 101.

$$F_{load} = 1 - \left(1 - F_{\min}\right)\frac{85 - T_{amb}}{85 - 40} \qquad (101)$$

Models were developed for six different system configurations at three site locations utilizing the loads and operating characteristics noted in Table 78 and Table 79. Air cooled condensers were sized with a 10°F difference between the condensing temperature and the ambient dry-bulb (TD) for low temperature systems and 15°F for medium temperature systems. During low ambient conditions, condenser fans are cycled to back up refrigerant and maintain head pressure (minimum condensing temperature of 70°F). The condensing temperature of the low temperature distributed system is allowed to drop to 50°F. Parasitic losses apply mainly to the parallel rack system, including 3°F for low temperature racks and 2°F for medium temperature racks. The glycol system has a 7°F glycol temperature rise for each display case.

Table 78. Rack Refrigeration System Configuration for 30139 ft^2 Store

	SST (°F)	Design Refrigerated Load (Btu/h)
Medium-temp, Rack A	19	511390
Low-temp, Rack B	-25	185260

Source: (Zhang 2006)

Table 79. Distributed Refrigeration System Configuration for 30139 ft^2 Store

	SST (°F)	Design Refrigerated Load (Btu/h)
Medium-temp, Unit A	16	223680
Medium-temp, Unit B	19	307060
Low-temp, Unit D	-25	82040
Low-temp, Unit E	-12	74820

Source: (Zhang 2006)

Energy consumption results show that distributed systems with scroll compressors have the lowest energy usage over a year (Table 80). Energy consumption includes that of the compressors, condenser fans, and pumps.

Table 80. Annual Energy Consumption of Refrigeration Systems Analyzed

	Rack	Distributed	Self-contained	Glycol Secondary Loop	CO_2 Secondary Loop	CO_2 Cascade
St. Louis	506	462	561	584	514	511
Boston	465	420	516	540	476	474
Dallas	582	543	646	665	587	582

Source: (Zhang 2006)

Analyzing the breakdown of energy usage (Table 81), it would appear that compressors use the most energy. This is misleading, as evaporator motor energy usage is not included in the energy analysis. In actuality, compressors are the second largest contributor to the energy usage of a refrigeration system.

Table 81. Breakdown of Annual Energy Consumption (MWh)

Systems	Compressors	Secondary Loop Pumps	Condenser/ Fluid Cooler Fans	Total
Rack	448	N/A	58	506
Distributed	408	N/A	54	462
Self-contained	503	N/A	58	561
Glycol Secondary Loop	489	36.4	58	584
CO_2 Secondary Loop	453	3.5	58	514
CO_2 Cascade	448	1.5	58	507

Source: (Zhang 2006)

The annual operating cost analysis took into account the energy used and the refrigerant used, assuming a leakage rate. This comparison to a standard parallel rack system shows cost savings for two of the five systems. Distributed, CO_2 secondary loop, and CO_2 cascade systems operated at higher operating costs of 12.7, 5.3, and 6.6%, respectively. Self-contained and glycol secondary loops had lower annual costs of 3.1 and 8.3%, compared to the parallel rack system. The TEWI analysis, as seen in Table 82, shows that the self-contained, glycol secondary loop, CO_2 secondary loop, and CO_2 cascade systems are all good solutions for reducing the environmental impact of refrigeration systems. This analysis is based on a conversion factor of 1.433 lb CO_2/kWh.

Table 82. TEWI of Various Refrigeration Systems

						TEWI (lb CO_2 x 10^5)		
Systems	Refr.	Charge (lb)	Leak	GWP[†]/ lb	Annual (kWh)	Direct	Indirect	Total
Rack	R404a	2500	0.15	1757	506044	14.527	7.252	21.779
Distributed	R404a	1300	0.10	1757	461987	5.036	6.620	11.657
Self-contained	R404a	300	0.01	1757	561200	0.116	8.042	8.158
Glycol	R404a	500	0.04	1757	583608	0.775	8.363	9.138
Secondary Loop	Propylene/ Potassium		0.1	0		0		
CO_2	Propane	500	0.04	9	513954	0.004	7.365	7.370
Secondary Loop	CO_2	1500	0.1	0.5		0.001		
CO_2 Cascade	Propane	500	0.04	9	507043	0.004	7.266	7.271
	CO_2	1500	0.1	0.5		0.001		

Source: (Zhang 2006)

[†] GWP: Global Warming Potential

(Hwang et al. 2007)

(Hwang et al. 2007) compared the performance of R-404a and R-410a to R-290 (propane) for walk-in refrigeration systems. Hydrocarbons, such as propane have low global warming potential but are highly flammable.

A theoretical analysis showed that R-290 was superior at higher condensing temperatures (19% and 6% better than R-404a and R-410a, respectively at 122°F SCT) and at lower temperatures it was outperformed by the HFCs to which it was compared by around 4%.

Experimental testing was performed in a psychrometric test facility on a low temperature and medium temperature refrigeration system. The low temperature refrigeration system had a capacity of 13,650 Btu/hr and utilized a -20.2°F saturated evaporating temperature and the medium temperature refrigeration system had a capacity of 37,535 Btu/hr and operated with saturated evaporating temperatures from -4 to 32°F.

In conclusion, based on equivalent capacity, R-290 performed at an 11 to 12% and 4 to 9% improved COP compared to R-404a and R-410a, respectively. At part load conditions it performed with an improved COP of 5 to 10% as compared to both HFC refrigerants.

(Sugiartha et al. 2009)

In the United Kingdom it is typical for direct expansion to be used to maintain space temperatures in the refrigerated spaces. The energy required to perform this cooling accounts for an average of 40% of supermarket energy use. This study analyzes a gas-turbine trigeneration system used for a 30,139 ft^2 supermarket in the south of England, relative to a conventional system.

The average electrical demand of the store is 395 kW (59 kW for low temperature refrigeration and 99 kW for medium temperature refrigeration). The average heat demand of the store is 55 kW.

The study concluded that gas turbine heat diversion to absorption chillers of 85% or less, a trigeneration plant performs at a lower system energy usage than a conventional system.

(Royal 2010)

Grocery store refrigeration systems run all day long and account for one-third to half of the store's electricity usage. As a waste product, the refrigeration system generates a constant supply of low quality heat.

A typical direct expansion system operates low temperature coils at -20°F with a 4 to 6°F superheat and 30°F return gas temperature. Medium temperature evaporator coils operate using a saturated suction temperature of 20°F, superheat of 5 to 8°F, and a return gas temperature of around 50°F. The heat recovery potential for this type system is summarized in Table 83.

Water heating is a common form of heat recovery. With the waste heat available, electric heat is seldom required to obtain the desired potable hot water temperature. Some systems utilize a holdback valve on the refrigeration system to maintain a condensing temperature sufficient to provide the desired waste heat temperature. This should be avoided, as the energy required to maintain the refrigeration system at these conditions is more than what would be require to heat the potable water with electric heat.

Another method of heat recovery is pre-heating the outdoor air supplied to the store. Based on the minimum condensing temperatures listed in Table 83, no space heating is required from a HVAC using this form of heat recovery at outdoor temperatures above 47.5°F.

A third method of heat recovery is using the refrigeration system waste heat in a heat pump configuration with the HVAC system. These systems are difficult to implement, as the peak HVAC heating requirement corresponds with the lowest condensing temperature and therefore, the lowest heating potential.

Table 83. Summary of Heat Recovery Potential for Various System Conditions with R-404A and R-407A Refrigerants

Condensing	Compressor	Mass Flow	Percent of Design	Heat Recovery
R-404A, 378,000 Btu/h Full Load, -20°FSST – Low Temperature – Desuperheating				
65	157.0	5,791	85%	135,182
75	167.0	6,412	88%	154,400
85	179.0	7,140	91%	181,509
95	192.0	8,008	94%	217,454
105	205.0	9,061	97%	263,331
115	218.0	10,397	100%	324,283
R-404A, 378,000 Btu/h Full Load, -20°FSST – Low Temperature – Full Condensing				
Condensing	Compressor	Mass Flow	Percent of Design	Heat Recovery
65	157.0	5,791	85%	504,503
75	167.0	6,412	88%	545,929
85	179.0	7,140	91%	596,603
95	192.0	8,008	94%	657,603
105	205.0	9,061	97%	730,107
115	218.0	10,397	100%	819,514
R-404A, 889,000 Btu/h Full Load, +20°FSST – Medium Temperature – Full Condensing				
Condensing	Compressor	Mass Flow	Percent of Design	Heat Recovery
65	115.0	10,204	70%	888,992
75	128.0	11,774	76%	1,002,507
85	141.0	13,580	82%	1,134,656
95	153.0	15,685	88%	1,287,939
105	166.0	18,176	94%	1,464,490
115	178.0	21,231	100%	1,673,508
R-407A, 378,000 Btu/h Full Load, -20°FSST – Low Temperature – Full Condensing				
Condensing	Compressor	Mass Flow	Percent of Design	Heat Recovery
65	190.0	4,602	85%	494,032
75	204.0	5,016	88%	534,032
85	219.0	5,494	91%	580,323
95	235.0	6,032	94%	633,730
105	250.0	6,666	97%	693,743
115	267.0	7,412	100%	767,212
R-407A, 889,000 Btu/h Full Load, +20°FSST – Medium Temperature – Full Condensing				
Condensing	Compressor	Mass Flow	Percent of Design	Heat Recovery
65	128.0	8,305	70%	770,192
75	143.0	9,459	76%	868,218
85	159.0	10,763	82%	978,387
95	174.0	12,219	88%	1,097,125
105	188.0	13,899	94%	1,226,168
115	203.0	15,841	100%	1,375,440

Source: (Royal 2010)

(Morris 2012)

Kevin Morris' presentation on supermarket refrigeration equipment selections gives details on walk-in coolers and freezers, and methods used to properly size equipment for these units (Morris 2012).

Walk-in cooler and freezer details include information on interior conditions, the relationship between interior conditions and evaporator selection, and example box sizes and corresponding loads. Walk-ins that contain items such as meat require high humidity (90% typical) to ensure minimum weight loss. To achieve this, evaporators are oversized and utilize a low air flow rate, resulting in a tight air side change in temperature of 7-9°F. On the other end of the spectrum, in rooms where people are constantly working to prepare foods or where candy or film are stored, low relative humidities in the range of 50-65% are maintained. To achieve this, high air flow rates are utilized with smaller evaporator units operating at 17-22°F air side temperature drops. A typical guideline for supermarkets is an evaporator delta T of 10-12°F for low temperature and 15°F for medium temperature applications.

Example walk-ins ranged in size from 30x30x8' to 20x12x8', with freezer box temperatures of 0 or -10°F and cooler temperatures of 35°F . A product load of 1000 lb of meat at 40°F was given as an example for a 40x20x8' freezer.

A system balancing routine is utilized to determine the actual balanced operating point of a given set of equipment. Condenser capacity at multiple saturated suction temperatures (SST) and the same design ambient temperature, evaporator quantity, evaporator capacity, and delta T are analyzed with the equations below to obtain the balanced capacity, evaporator delta T, and SST, and the runtime hours.

$$m_{cs} = \frac{Cap_{cond1} - Cap_{cond2}}{SST_1 - SST_2} \quad (105)$$

$$TD_{Ebal} = \frac{m_{cs}\left(T_C - T_{sll}\right) + Cap_{cond1} - m_{cs} \cdot SST_1}{N_{Evap}\dfrac{Cap_{evap}}{TD_{Edes}} + m_{cs}} \quad (106)$$

$$Cap_{Bal} = \frac{TD_{Ebal} \cdot N_{Evap} \cdot Cap_{evap}}{TD_{Edes}} \quad (107)$$

$$RT_{Bal} = \frac{Q_{tot}}{Cap_{Bal}} \quad (108)$$

General Load Details

(Walker et al. 1990)

Supermarket refrigeration accounted for 59% of the total store energy consumption in 1990 (Walker et al. 1990). With a great potential for reduction in energy usage, supermarket refrigeration systems are a constant target for optimization. Walker compared conventional and multiplex refrigeration systems used for supermarket refrigeration. The conventional refrigeration system used air cooled condensers, fixed head pressure control, and electric defrost. Multiplex configurations included air-cooled and evaporative condensers, floating and fixed head pressure control, ambient and mechanical sub-cooling, and electric and hot gas defrost. To allow for application by industry, the report focuses on the operating and capital costs of each system type for various operating requirements.

A display case line-up or circuit consists of several display cases that have parallel liquid and suction refrigerant piping. Conventional refrigeration systems use a single compressor for each display case lineup or walk-in box. Each circuit in the store is isolated from all of the others with dedicated refrigerant piping, control valves, receiver, electrical components, condenser, and compressor. Multiplex refrigeration systems use semi-hermetic compressors typically, mounted on a rack. Display case lineups and walk-ins that are operating at similar conditions share a liquid line and discharge line.

An equal parallel compressor rack is a type of multiplex refrigeration system in which all compressors are of the same size, in terms of refrigeration capacity. Unequal parallel compressor racks allow the system to closely match the required refrigeration load during off-design periods, achieve the highest possible suction pressure, and higher system efficiency.

Multiplex systems use floating head pressure to reduce compressor power. Variation in the head pressure must be limited however, as it typically has a negative effect on the ability of the expansion valve to properly feed refrigerant into the evaporator coil and could potentially cause flashing of the refrigerant in the liquid lines. A minimum condensing temperature of 70°F is typically recommended. In conventional systems with a single compressor, floating head pressure can cause system instability issues. As the head pressure drops, the capacity of the compressor increases, causing long off-cycle periods. To avoid this the minimum condensing temperature for conventional systems must be set much higher, usually between 90 and 95°F. This is referred to as fixed head pressure control.

Six sites were used as part of the simulations, including Atlanta, GA; Boston, MA; Los Angeles, CA; Milwaukee, WI; Phoenix, AZ; and St. Louis, MO. A Safeway Stores, Inc. field test site in Menlo Park, CA (Store #990) was used as a baseline to build the simulation from. The system has three racks, operating at five suction temperatures (SST), including a low (3 compressors at -23°F SST) and very low (1 compressor at -33°F SST) temperature rack (Rack A), a medium temperature rack at 16°F SST (Rack B), and a high temperature rack at 26°F SST (Rack C). A satellite compressor operates at 9°F SST on Rack C to refrigerate the meat display cases. As is typically done, this refrigeration system utilizes hot gas defrost.

This investigation was limited to single-stage vapor compression, using semi-hermetic reciprocating compressors. R12 was utilized for medium and high temperature refrigeration and R502 was used for low temperature circuits (both being phased out under the Montreal protocol).

Evaporative condensers were sized for a condensing temperature of 100°F. Air cooled condensers are applied with a design temperature differential (Condenser temperature minus design ambient dry bulb) of 10°F for low temperature racks and 15°F for medium and high temperature racks. As mentioned before, minimum condensing temperature for the multiplex refrigeration systems is set at 70°F during floating head pressure operation. Head pressure is also controlled so that it raises prior to initiation of each defrost cycle. Using oversized condensers reduces the condensing temperature. However, the savings of operating at a lower head pressure are typically offset by increased fan energy consumption of the oversized condensers.

First the conventional system was modeled. It utilized 23 single compressors with individual air-cooled condensers with fixed head pressure control set at 90°F. The 23 compressors are categorized as 1 very low, 5 low, 12 medium, and 5 high temperature. The conventional system has small air cooled condensers mounted to the compressor base in the machine room. Suction pressure is held between set points by cycling of the compressors. Head pressure is controlled above a minimum condensing temperature of 90°F by a liquid line thermostat, which cycles the condenser fans.

Conventional systems typically use either off-cycle or electric defrost. Off-cycle is used for medium or high temperature refrigeration circuits whereas electric defrost is utilized for low temperature cases or boxes.

Table 84. Refrigerated Cases and Walk-in Boxes in the Modeled Supermarket Rated at the Inside Ambient of 75°F Dry Bulb and 55% Relative Humidity

Description	Case Type	Total Case Length (ft)	Floor Area (ft2)	Design Load (Btu/h)	Eva Temp (°F)	Discharge Air Temp (°F)
Ice cream	Wide aisle tub	76	--	32300	-33	-20
Frozen food	26 door reach-in	68	--	39520	-23	-5
Frozen food	26 door reach-in	68	--	39520	-23	-5
Frozen food	Wide aisle tub	64	--	21600	-23	-10
Freezer	Walk-in box	--	500	29900	-22	-12
Bakery freezer	Walk-in box	--	81	7100	-22	-12
Cheese case	Single deck	39	--	13074	18	37
Produce	Multi-deck	24	--	17400	19	37
Dairy	Multi-deck	48	--	61200	19	32
Dairy	Multi-deck	40	--	51000	19	32
Deli	Multi-deck	36	--	58680	16	32
Beverage	Multi-deck	48	--	61200	19	32
Cheese	Single deck	22	--	6400	18	37
Produce	Single deck	52	--	22100	19	37
Meat cooler	Walk-in box	--	480	26300	20	30
Fish cooler	Walk-in box	--	96	7800	20	30
Meat holding	Walk-in box	--	120	9200	21	30
Meat case	Multi-deck	36	--	58500	9	22
Meat	Multi-deck	24	--	39000	9	22
Fish	Service	16	--	6246	21	27
Produce cooler	Walk-in box	--	437	21750	31	40
Dairy cooler	Walk-in box	--	540	27825	27	36
Meat prep	Walk-in box	--	800	48000	29	50
Deli box	Walk-in box	--	70	6275	26	36
Bakery retard	Walk-in box	--	80	6400	27	--

(Walker et al. 1990)

The building was modeled according to the following specification:

- Hours of operation: 24 h/day
- Conditioned air volume: 358,000 ft^3
- Total floor space: 43,139 ft^2
- Sales floor space: 25,600 ft^2
- Building construction quality: tight
- Building walls U-value (ASHRAE 8-in concrete block, filled insulation): 0.103 Btu/h-ft^2-°F
- Building roof U-value (ASHRAE No. 17, steel sheet with 2-in insulation): 0.093 Btu/h-ft^2-°F
- Storefront window U-value: 0.917 Btu/hr-ft^2-°F
- Storefront window area (30% of north wall area): 450 ft^2
- Ventilation flow rate: 21,000 cfm (return air), 2,200 cfm (fresh air)
- Internal sensible gain due to lighting and equipment: 3.24 W/ft^2
- Internal moisture generation: 10 lbm/h
- Building thermal capacitance (ASHRAE heavy construction classification): 666,000 Btu/°F
- Air volume used for moisture capacitance: 358,000 ft^3
- Walls: Inside convection coefficient: 1.4 Btu/h-ft^2-°F
- Window: U-value (not including inside/outside convection): 14.2 Btu/h-ft^2-°F
- Inside convection coeff: 1.4 Btu/h-ft^2-°F

- Window dimensions: 5 ft high, 90 ft long
- People heat gain: Sensible: 315 Btu/h-person
- Latent: 325 Btu/h-person
- Floor U-Value (ASHRAE 12in concrete slab with 12-in insulation): 0.0238 Btu/h-ft^2-°F
- 661 ft of refrigerated cases, 2512 ft^2 of walk-in boxes
- HVAC: central air handler capable of 21,000 ft^3/min
- Air cooled condensers were set at 10°F temperature difference (condenser temperature - design ambient dry bulb) for very low and low temperature circuits whereas 15°F was used for medium and high temperature circuits.
- The evaporative condenser utilized a design condensing temp of 100°F.

The multiplex system parallels the field site with 13 parallel compressors with 1 very low, 3 low, 6 medium, and 3 high temperature compressors. Floating head pressure is utilized by this system along with ambient sub coolers and mechanical sub coolers for very low and low temperature racks. Hot gas defrost is utilized for very low and low temperature racks. The entire system is cooled by a remotely located evaporative condenser

The conventional and multiplex defrost schedules were provided in great detail. The number of defrosts per day ranges from 1 to 4 with an average of 1.63 times per day for walk-in freezers and 0 to 4 with an average of 2.58 times per day for walk-in coolers. Defrost duration over the course of the day ranged from 0.5 to 3.33 hrs, with an average duration of 1.17 hrs for walk-in freezers and 0 to 3.47 hrs, with an average duration of 1.98 hrs for walk-in coolers.

The lighting load assumed was 3.24 W/ft^2. Occupancy load was set as 50 people from 3 am to 6 am and 200 people from 3 pm to 6 pm, with the number of people ramping linearly between these two points. For refrigerated cases and walk-in box details see Table 84. Suction temperatures are the same as those outlined in the details about the field site.

(Anonymous 2004)

This article gives a general discussion of walk-in coolers and freezers. Walk-ins may be pre-engineered or built on site. They may be located indoors or outdoors and may reject heat using a packaged refrigeration system connected to the box or using a remote condenser. Footprints range from 5 x 6 ft to an entire warehouse with some being multiple stories. These units store 28 lb of product/ft^3 of available storage volume on average.

A common walk-in box temperature is 35°F whereas a freezer is typically maintained at -10°F. Construction materials are typically urethane insulated panels encased in sheet metal with resistance values of around R-30 for 4" of insulation. Five to six inches of insulation are available. Insulated panels are mass produced in sizes of 1, 2, and 4 ft wide by 7.5, 8.5, and 9.5 ft tall, confining the possible walk-in box sizes. Doors are 24 to 34 inches wide

When combination units are designed, the freezer door opens to the cooler to reduce heat loss. Air curtains or vinyl door strips are used by some walk-ins to reduce infiltration to the space.

(PG&E 2004)

Pacific Gas and Electric Company (PG&E) analyzes the current state of walk-ins in California in this report. The report characterizes typical walk-ins and proposes some standards to improve the efficiency of refrigeration equipment in the jurisdiction.

Walk-ins range from less than 50 ft^2 to 2,000 ft^2, with ceiling heights ranging from 8 to 30 ft. Walk-ins may be placed inside the building that it is serving or just outside the building. Small walk-ins typically only have one access door but often have several glass display doors for customer access to products. Construction is typically prefabricated panels of polyurethane, polystyrene, or fiberglass with an insulation thickness of 3.5 to 5.5 inches. Wood or high density polyethylene are sometimes used for structural integrity of the panels.

Recommended standards include automatic door closers, reach-in doors with anti-sweat heater control, evaporator fan control, and high efficiency lighting.

(Heatcraft 2008)

The Heatcraft Engineering Manual (Heatcraft 2008) is a well-trusted source of information on sizing a walk-in for numerous applications in the commercial refrigeration industry. There is a wealth of specific information that can be applied as well as general approximations for when the details related to a project are somewhat unknown.

Walk-ins are composed of various materials. To assist in determining the conduction load, Heatcraft provides a table of heat loads based on these variables. Insulation and building material details are included in Table 85.

Table 85. Insulation and Building Material Conductivity for Walk-In Conduction Load Calculation

Material	K (Btu-hr/in-°F)
Cork or Mineral Wool	0.30 to 0.38
Glass Fiber or Polystyrene	0.26
Sprayed Urethane	0.16
Foamed in Place Urethane	0.12
Air	4.65
Vermiculite	0.47
Sawdust	0.45
Mac. Paper	0.28
Styrofoam	0.24

Source: (Heatcraft 2008)

The conduction load associated with the materials that the box is constructed of is fairly straightforward from a calculation point of view. The designer uses the difference between the outside temperature and the box temperature and the resistance to heat transfer to determine the load. One variable that must be somewhat approximated due to its complexity, is the radiant heat load received by the walk-in box from its surroundings. The primary radiant load is due to the sun and is approximated by Heatcraft using a temperature differential that is added to the outdoor temperature, based on the position and emissivity of each outside surface (Table 86).

Table 86. Allowance for Sun Effect (°F)

	Allowance for Sun Effect (+ °F)			
Surface Type	East Wall	South Wall	West Wall	Flat Roof
Dark Colored Surfaces: Slate roofing, tar roofing, or black paints	8	5	8	20
Light Colored Surfaces: White stone, light colored cement, or white paint	4	2	4	9
Medium Colored Surfaces: Unpainted wood, brick, red tile, dark cement, or red, gray, or green paint	6	4	6	15

Source: (Heatcraft 2008)

Another conduction component is the heat load from the ground. Cooler heat gain is based on the temperature differential between the local ground temperature and the storage room temperature. Freezer floors, however, are commonly heated to prevent the ground below the freezer from freezing and heaving up. A minimum slab temperature is 40°F but because the effects of a sub-floor freezing would be so costly, most walk-ins use a slab temperature of 55°F. An extensive list of ground and design temperatures is provided in the Heatcraft Manual.

Product load can be calculated easily by hand if the details on the quantity of product that will be received daily is known, along with the temperature at which it will be received. If no such information is available Heatcraft presents a typical average daily product load based on the cooler or freezer's size (Table 87). Storage requirements and miscellaneous product properties are presented by Heatcraft in great detail.

The time that it takes for product to be pulled down to storage temperature is closely tied to the equipment selection for walk-ins. Heatcraft notes a 24 hour time period as a typical pull-down duration, and gives an example of a storage room design operating at 1°F/hr pull-down.

Table 87. Walk-in Coolers and Freezers Typical Average Daily Product Loads

Volume (ft^3)	Average Daily Product Loads (lbs.) for Coolers	Average Daily Product Loads (lbs.) for Freezers
500 to 3000	6200 to 8000	1600 to 2000
3000 to 4600	8000 to 11000	2000 to 2500
4600 to 8100	11000 to 17000	2500 to 4000
8100 to 12800	17000 to 26000	4000 to 6200
12800 to 16000	26000 to 33000	6200 to 7500
16000 to 20000	33000 to 40000	7500 to 9500
20000 to 28000	40000 to 56000	9500 to 13000
28000 to 40000	56000 to 66000	13000 to 17000
40000 to 60000	66000 to 110000	17000 to 25000
60000 to 80000	110000 to 150000	25000 to 34000
80000 and up	150000 and up	34000 and up

Source: (Heatcraft 2008)

Infiltration calculation methods are simple or more involved depending on the amount of information known about the application. If the number of minutes that the door will be open per day is known, the following equation may be applied.

$$Q_{Inf} = 4.88\frac{ft^{1/2}}{{}^{\circ}F^{1/2}\min}\frac{\sqrt{H\cdot(T_H-T_C)}(1-E)(h_H-h_C)t_{open}}{\upsilon_H}\cdot\frac{A_{\inf}}{2}\cdot\frac{\min}{60s} \qquad (109)$$

This is a similar equation to that presented by Krack (Equation 67), except for the fact that Krack assumes a 24 hr period and a constant outside air density of 0.075 lb/ft^3.

If details required to calculate infiltration using Equation 109 are not available, designers use a typical value for the average air changes in a 24 hour period. Heatcraft provides these values as a function of the volume of the space (Table 88). The values in Table 88 are the same as those in Table 22 (Krack 1977) with the following exceptions. Heatcraft interpolates to some intermediate values and extrapolates to some larger space sizes that are omitted by Krack. Heatcraft also omits some intermediate values that Krack includes. The only values that do not agree are air changes for the 200,000 ft^3 cooler, for which Krack specifies 0.9 air changes per 24 hours whereas Heatcraft specifies 1.1. To allow for easy calculation of the heat load Heatcraft also provides a table of data to convert air changes to heat loss based on the inside and outdoor conditions.

Lighting for storage spaces are typically 1 to 1.5 W/ft^2. Cutting or processing rooms may have 2 to 3 W/ft^2. Equipment loads include lift trucks, evaporator fans, heaters, etc. Lift trucks are typically battery powered for refrigerated spaces due to the desire to keep the required air changes to a minimum. A conservative estimate for the load added to the space by lift trucks in larger refrigerated spaces is 0.16 Btu/hr-ft^3 for coolers and 0.20 Btu/hr-ft^3 for freezers. Occupancy loads vary depending on the facility type and the temperature at which it operates (Table 89).

Table 88. Average Air Changes per 24 Hrs for Medium Temperature (Above 32°F) and Low Temperature (Below 32°F) Rooms Due to Infiltration and Door Openings.

	Medium Temp	Low Temp		Medium Temp	Low Temp
Volume (ft^3)	Air Changes/24 hr		Volume (ft^3)	Air Changes/24 hr	
200	44.0	33.5	8000	5.5	4.3
250	38.0	29.0	10000	4.9	3.8
300	34.5	26.2	15000	3.9	3.0
400	29.5	22.5	20000	3.5	2.6
500	26.0	20.0	25000	3.0	2.3
600	23.0	18.0	30000	2.7	2.1
800	20.0	15.3	40000	2.3	1.8
1000	17.5	13.5	50000	2.0	1.6
1500	14.0	11.0	75000	1.6	1.3
2000	12.0	9.3	100000	1.4	1.1
3000	9.5	7.4	150000	1.2	1.0
4000	8.2	6.3	200000	1.1	0.9
5000	7.2	5.6	300000	1.0	0.85
6000	6.5	5.0			

Source: (Heatcraft 2008)

Table 89. Heat Equivalent of Occupancy

Box Temperature (°F)	Btu/24 hr-person
50	17280
40	20160
30	22800
20	25200
10	28800
0	31200
-10	33600

Source: (Heatcraft 2008)

Glass doors used for customer access to products stored in display walk-in coolers and freezers increase conduction to the walk-in box and introduce additional paths for infiltration. Heatcraft offers approximation for the total load added by use of glass doors in Table 90.

Table 90. Glass Door Loads

Box Temperature (°F)	Btu/door
35	1060
30	960
0	1730
-10	1730
-20	1730

Source: (Heatcraft 2008)

When the main components of heat load are tabulated, a 10% safety factor is typically applied to compensate for minor omissions. The peak calculated load is then spread over a 16 to 22 hour condensing unit runtime. Thirty-five degree Fahrenheit rooms often use off-cycle defrosts, requiring 6 to 8 hours for defrost (16-18 hour condensing unit runtime). Blast coolers and freezers with electric or hot gas defrost operate at 18 hour runtime. Rooms used for frozen storage are typically sized for 18 to 20 hour operation whereas medium temperature rooms (25 to 34°F) with positive defrost and high temperature rooms (50°F or above) are sized for 20 to 22 hour runtime.

Rooms above 32°F typically have an evaporator sized at a 10 to 12°F T.D. to maintain an interior relative humidity of 80 to 85%. If a higher relative humidity is needed (meat cutting room for example) the evaporator size should be increased, pulling down the T.D. of the coil. Storage rooms operating below freezing typically use a 10°F or lower T.D.. Evaporator temperature differences and their associated applications are summarized below (Table 91). Evaporator temperature differences may be approximated by Equation 110.

$$TD_{Eapprox} = \frac{Cap_{cond,SST}}{Cap_{evap,1TD}} \tag{110}$$

Table 91. Recommended Evaporator Temperature Difference per Various Applications

T.D. (°F)	Approximate RH (%)	Application
7 to 9	90	Low product moisture loss. Vegetable, produce, flower, unpackaged ice and chill rooms.
10 to 12	80 to 85	General storage and convenience store coolers, packaged meats and vegetables, and fruits.
12 to 16	65 to 80	Moderate relative humidity requirements. Beer, wine, pharmaceuticals, potatoes, onions, tough skin fruit, and short term packaged products.
17 to 22	50 to 65	Humidity must remain low to avoid damage to product. Prep/cutting rooms, beer warehouses, candy or film storage and loading docks.

Source: (Heatcraft 2008)

Another component of evaporator unit design is the air flow rate that they operate at. Air change needs for various applications are summarized in Table 92. The desired air changes may be converted to air flow using Equation 111.

$$\dot{V}_{\inf} = ACH \cdot V \frac{hr}{60\min} \tag{111}$$

Table 92. Recommended Evaporator Induced Room Air Changes/Hour

Type of Application	Recommended Air Change/Hour Range
Holding Freezer	40 to 80
Packaged Holding Center	40 to 80
Cutting Rooms	20 to 30
Meat Chill Room	80 to 120
Boxed Banana Ripening	120 to 200
Vegetables and Fruit Storage	30 to 60
Blast Freezer	150 to 300
Work Areas	20 to 30
Unpackaged Meat Storage	30 to 60

Source: (Heatcraft 2008)

(SCE 2008)

eQuest, an energy modeling tool, was used by Southern California Edison (SCE 2008) to model walk-in coolers and freezers for development of Title 20 standards for walk-in boxes. Energy efficiency focused improvements suggested by this report for application to the Title 20 standard include strip curtain or spring hinged door infiltration reduction, high efficiency lighting or controls, freezer floor insulation (R-28 or greater), floating head pressure control, low head pressure capable compressors, variable speed evaporator fans, temperature termination defrost control, and anti-sweat heater wattage limits with humidity monitoring controls.

Four different walk-in types (250 ft2, 500 ft2, 1000 ft^2, and 2,500 ft^2) were simulated as coolers and freezers in a baseline configuration and with the improvements noted above to determine the cost savings potential assuming a year round ground temperature of 50°F. Details pertaining to the calculation of the walk-in load are listed in Table 93.

Table 93. Refrigerated Walk-In Modeling Details

Cooler temperature (°F)	35
Freezer temperature (°F)	5
Product Load (%)	70
Lighting Load (W/ft^2)	1.0
250 ft^2 cooler infiltration rate (ACH)	1.00
500 ft^2 cooler infiltration rate (ACH)	0.68
1000 ft^2 cooler infiltration rate (ACH)	0.46
2500 ft^2 cooler infiltration rate (ACH)	0.25
250 ft^2 freezer infiltration rate (ACH)	0.78
500 ft^2 freezer infiltration rate (ACH)	0.53
1000 ft^2 freezer infiltration rate (ACH)	0.36
2500 ft^2 freezer infiltration rate (ACH)	0.21

Source: (SCE 2008)

Strip curtain protected doors are assumed to have a 75% reduction in infiltration by this study. Freezer floor insulation of R-28 is compared to the standard 6" concrete slab (R-1.2). A film coefficient of 0.61 hr-ft^2-°F/Btu (ASHRAE 2005) is applied.

(Goetzler et al. 2009)

Navigant discusses current equipment used for different applications in commercial refrigeration. Commercial refrigeration energy usage accounts for 4.1 to 6.3% of primary energy used by commercial buildings (DOE 2006; DOE 2008). Annual refrigeration energy usage was broke down into various types of refrigeration equipment. Supermarket refrigeration accounts for 56% of the total commercial refrigeration energy usage, with non-supermarket walk-ins accounting for an additional 12%. Opportunities to reduce energy usage are presented and compared to the current state.

The quantity of various units in the domestic U.S. is presented for calculation of energy savings (Table 94). The quantities of retailers that utilize this equipment are listed in Table 95. This information is very useful in understanding how common these various applications are.

Table 94. 2008 Installed Base and Total Energy Consumption by Equipment Type

	Installed Units (x 1000)	Total Primary Energy Consumption (TWh/yr)
Supermarket Refrigeration Systems:		
Display Cases	2,100	214
Compressor Racks	140	375
Condensers	140	50
Walk-in Coolers	152	Combined
Walk-in Freezers	76	Walk-ins:
Combination Walk-ins	17	51
Other Refrigeration Systems:		
Walk-in Coolers (non-supermarket)	468	79
Walk-in Freezers (non-supermarket)	234	52
Combination Walk-ins (non-supermarket)	53	17
Food Preparation and Service Equipment	1,516	55
Reach-in Coolers and Freezers	2,712	106
Beverage Merchandisers	920	45
Ice Machines	1,491	84
Refrigerated Vending Machines	3,816	100
Total		1,225

Source: (Goetzler et al. 2009)

Table 95. 2007 Number of Stores and Average Sales in the Grocery Industry

Store Type	Number of Stores (x 1000)	U.S. Annual Sales ($ Billions)
Supermarkets	35.0	535.4
Convenience	145.9	306.6
Grocery	13.7	18.2
Wholesale Clubs	1.2	101.5
Military Convenience Stores	0.4	2.2
Total	196.2	963.9

Source: (Goetzler et al. 2009)

Typical walk-in sizes are cited as ranging from 80 to 2600 ft^2 in floor area with 8 ft ceilings. Walls and ceilings may be R-28 through application of blown polyurethane foam, expanded polystyrene, or extrude polystyrene. Walk-ins always have at least one door, which must be heated with anti-sweat heaters in freezers. Lighting loads range from 0.9 to 1.2 W/ft^2. R404A is a common refrigerant for these refrigeration systems. Typical evaporator operating temperatures for certain products are listed in Table 96. A cited electric defrost energy usage is 25 W/ft^2 of refrigerated space. Two prototypical walk-ins are presented which were used to estimate the baseline energy usage (Table 64). Coincidently, these units are the same as those used by Westphalen et al. (Westphalen et al. 1996) in a similar analysis.

Table 96. Evaporator Operating Ranges for Various Products

	Temperature Range (°F)	Applications
High Temperature	35 and above	Produce, Flowers
Medium Temperature	10 to 15	Meats, Seafood
Medium Temperature	15 to 25	Dairy, Produce, Beverages, Meat Walk-ins
Medium Temperature	25 to 35	Dairy and Produce Walk-ins, Prep Rooms
Low Temperature	-25 to -15	Frozen Foods
Very Low Temperature	-35 to -25	Ice Cream, Frozen Bakery

Source: (Goetzler et al. 2009)

(DOE 2010b)

The DOE Preliminary Technical Support Document (TSD) (DOE 2010b) presents a preliminary analysis of the current market of walk-in coolers and freezers. In order to understand the full picture, the technologies' energy use, initial cost, and cost to manufacture were all taken into account. In consideration of improving the energy efficiency of walk-in coolers and freezers, the effect on consumers and manufacturer's was analyzed. Four types of walk-ins were analyzed, including non-display and display walk-in coolers and freezers. Three different footprints were considered, including small (10 x 8 x 7.6 ft tall), medium (12 x 20 x 9.5 ft tall), and large (25 x 30 x 12 ft tall), but an industry discussion revealed that the grouping may have not been appropriate. Small walk-ins typically range from 200 to 400 ft^2. The height of the space significantly changes the design, with heights ranging from 8 to 30 ft, where 30 ft is more applicable to a warehouse construction. Six equipment classes were analyzed, including dedicated refrigeration system with condenser indoors, dedicated refrigeration system with condenser outdoors, and multiplex refrigeration systems for low or medium temperature applications.

An overview of walk-in coolers and freezers revealed that coolers operate above 32°F and freezers operate below. These boxes typically use automatic door closers for a tight closure, some method of infiltration protection, and insulation better than R-25 for coolers and R-32 for freezers, with freezers using R-28 or better for floor insulation. Insulated walls used for these sizes of walk-ins are typically prefabricated panels with a foam core and a metal skin on either side.

In the analysis of walk-in performance, a baseline system was developed, and multiple variations to system design were simulated to analyze the effect of each component on the overall energy efficiency and first cost of the system. The baseline envelopes used 4" of wall and ceiling insulation, composed of extruded polystyrene or polyurethane. Freezer floors are insulated to R-28, as noted as typical above, and coolers have no floor insulation. Display walk-ins have glass display doors with an R-value of 2.3 hr-ft-°F/Btu. Walk-ins were assigned a certain number of doors based on their size and type. Passage doors had a glass area of 0.9 ft^2 and door dimensions of 3 x 7 ft. Freight doors were 7 x 9 ft tall for small and medium walk-ins and 7 x 12 ft tall for large walk-ins. Display doors are 2.5 x 6.3 ft tall in this study.

Table 97. Number of Doors for Each Walk-in Type

	Display Doors	Passage Doors	Freight Doors
Non-display small	0	1	0
Non-display medium	0	1	1
Non-display large	0	2	1
Display small	3	1	0
Display medium	8	1	0
Display large	50	2	0

Source: (DOE 2010b)

No infiltration reduction devices, door systems, or anti-sweat heater control are utilized. Lighting is done with 5 ft T8 fluorescents. Design option alternatives that were analyzed in this study are noted below. The refrigeration equipment and control methods used to serve the walk-ins were also varied to analyze the effect on the overall efficiency and first cost.

- Wall/ceiling insulation thickness: 4”, 4.4”, 5”, 6”, and 7”.
- Floor insulation: none, 4”, 5”, and 6”.
- Insulation materials: extruded polystyrene/polyurethane (5.90 hr-ft-°F/Btu-in), vacuum insulation panels (29.15 hr-ft-°F/Btu-in), and hybrid panels (75% vacuum insulated, 25% standard foam insulation).
- Display Door R-values: 2.3, 3.7, 8.1, and 12.5 hr-ft-°F/Btu.
- Sealants for pre-fabricated panel joints: gasket, gasket and caulking, and advanced tongue and groove.
- Infiltration protection devices: none, strip curtain, and air curtain
- Door systems: none and vestibules (98% effectiveness).
- Anti-sweat heater: no control or control
- Lighting: T8 fluorescent, compact fluorescent, or LED
- Other controls: no control, lighting sensors, and door opening control

Using finite element analysis and specific walk-in operating temperatures, DOE determined floor heat flux for various levels of floor insulation. Results varied based on the floor area and are presented in Table 98.

Table 98. Average Heat Flux for Various Footprints of Walk-in Coolers (35°F) and Freezers (-10°F)

Floor Area (ft^2)	Average Heat Flux ($Btu/h\text{-}ft^2$)		
Cooler Floor R-Value:	0	22.42	28.03
36	8.61	1.48	1.21
71.4	7.31	1.43	1.18
80	6.90	1.41	1.17
240	4.40	1.31	1.10
750	2.97	1.13	0.97
1200	3.04	1.18	1.01
Freezer Floor R-Value:	22.42	28.03	33.64
36	3.15	2.59	2.20
48	3.11	2.56	2.18
71.4	3.04	2.51	2.14
180	2.88	2.40	2.06
500	2.54	2.16	1.88
1200	2.51	2.14	1.86

Source: (DOE 2010b)

Parameters used by DOE for their calculations, which are pertinent to this study, include information related to the loads of the walk-in. Freezers were operated at -10°F and 60% relative humidity and coolers were operated at 35°F and 60% relative humidity with an ambient exterior temperature of 75°F and 40% relative humidity. The concrete floor temperature was assumed to be 60°F for coolers and 65°F for freezers. Associated with the transmission load as well, DOE specified an external film coefficient of 0.68 $hr\text{-}ft^2\text{-}°F/Btu$, internal film coefficient of 0.25 $hr\text{-}ft^2\text{-}°F/Btu$, and floor film coefficient of 0.87 $hr\text{-}ft^2\text{-}°F/Btu$. Infiltration details specified include the door flow factor of 0.8, display door opening schedule of 72 per day at 8 seconds per opening, passage door opening schedule of 60 per day at 12 seconds per opening with an additional 15 minutes per day that the door stands open, and a freight door opening schedule of 60 openings per day at 12 seconds per opening with an additional 15 minutes per day that the door stands open. The anti-sweat heater provided 1 W of heat per foot of display door perimeter, with 70% of that heat transferred into the refrigerated space. Defrosts are 15 minutes in duration at 4 defrosts per day.

(DOE 2010c)

This presentation (DOE 2010c) is a summary of the Technical Support Document (TSD) for energy conservation in walk-in coolers and freezers (DOE 2010b) and correspondence with industry experts following the TSD being presented (DOE 2010a).

Walk-ins are noted as any refrigerated space less than 3,000 ft^2. Walk-in coolers operate above 32°F and freezers operate at or below 32°F. Typical outdoor conditions are noted as 90°F for indoor condensing units and 35, 59, and 95°F for outdoor condensing units. Foam is specified as the most common insulating material. Four inch thick foam panels can vary significantly in R-value depending on what foam material is used, ranging from 16 to 32 hr-ft^2-°F/Btu. Steel which is used as the casing has an R-value of 0.16 hr-ft^2-°F/Btu. Vacuum insulated panels have R-values ranging from 120 to 200 hr-ft^2-°F/Btu. The initial R-value deteriorates over time as gases diffuse from the foam and it takes on water from the air.

DOE computer simulations of heat transfer through uninsulated cooler floors developed the following two equations for heat flux (Equations 112 and 113) as a function of floor area, assuming a sub-floor temperature of 65°F. A floor R-value of 28 hr-ft^2-°F/Btu is required for freezers and a ground temperature of 60°F is assumed.

$$A_{floor} < 750\ ft^2:\ q''_{floor} = 33.153 A_{floor}^{-0.364} \qquad (112)$$

$$A_{floor} \geq 750\ ft^2:\ q''_{floor} = 0.002 A_{floor} + 2.84 \qquad (113)$$

Some assumed schedules are presented for lighting and anti-sweat heaters. Lights are on 75% of the time when no timers or other shut-off systems are in place. A unit that does have lights that automatically shut off may only use lights half of the time. Anti-sweat heaters are on all the time in a typical walk-in, but through the use of heater controls this may be reduced to half the time for freezers and 25% of the time for walk-in coolers.

(Becker et al. 2011)

The Final Report, prepared by Becker for the Air-Conditioning and Refrigeration Technology Institute (ARTI) in 2011, consolidated information from a number of sources (Becker et al. 2011). Information presented includes information of walk-in size, construction, infiltration, and infiltration reduction

Data from monitored field sites is analyzed in this report (Table 99). Field sites are in Massachusetts and Rhode Island and include one freezer and three coolers. One cooler is for a convenience store whereas the other walk-ins are all utilized by a restaurant. In addition to data on unit performance, the box operating conditions, box construction, walk-in size and configuration, door size and type, anti-sweat heater loads, defrost loads and cycles, lighting, product load, occupancy load, infiltration, and equipment loads are all compared. Due to a current focus on energy efficiency, a number of articles discussed by Becker et al. were related to new standards being put in place for future construction or testing of new concepts to reduce energy usage of supermarket refrigeration. Unfortunately, using this information to define a model load for a rating standard would not be appropriate. The majority of equipment that is in service or will be brought into service in the near future will not be the new energy efficient equipment. Typically, such items are more expensive than standard equipment and require significant advantage or a government mandate to incentivize selecting them.

Table 99. Runtime Analysis of Walk-in Coolers and Freezers

	Walk-in Coolers					Walk-in Freezers	
Statistic	Tedeschi 11 Door	Chili's Beer	Chili's Food	Field Average	AHRI 1250/1251	Chili's Food	AHRI 1250/1251
Min Hourly Runtime %	9.5	1.8	8.4	6.6		24.0	
Max Hourly Runtime %	39.4	32.0	82.6	51.3		84.2	
Ave. Hourly Runtime %	24.8	13.1	44.0	27.3		45.7	
Ave Low Load RT%	16.4	7.7	25.8	16.6	10	33.3	40
Operation at Low Load %	45.8	55.2	50.0	50.3	67	54.2	67
Ave High Load RT%	32.3	19.6	61.9	37.9	70	60.4	80
Operation at High Load %	54.2	44.8	50.0	49.7	33	45.8	33

Source: (Becker et al. 2011)

Door opening and closing events were recorded for three field sites, as was the defrost cycle detail (Table 100). The Tedeschi 11-door cooler had an average of 9.3 openings per day of 4.8 minute average duration. Chili's food cooler had an average of 22.0 openings per day of 1.0 minute average duration. Chili's food freezer had an average of 85.0 openings per day of 1.2 minute average duration. The AHRI Load Spreadsheet (AHRI 2009a) was compared to the field door data with 55.5 openings per day of 0.41 minute average duration.

Table 100. Analysis of Walk-in Cooler and Freezer Defrost Cycles

Statistic	Tedeschi 11 Door Cooler	Chili's Beer Cooler	Chili's Food Cooler	Cooler Average	Chili's Food Freezer
Average number of defrosts per day	1.75	1.25	4.25	2.42	3.00
Average duration of cycles (min)	41.6	24.3	34.6	33.5	37.8

Source: (Becker et al. 2011)

Becker et al. validated eQuest by comparing simulation results to field data obtained from a PG&E instrumented freezer at the Food Service Technology Center (FSTC). Three trials were analyzed with an average difference between eQuest results and the physical test data of 3.0%, with eQuest overestimating the refrigeration system's energy usage.

Seven locations were used for climate zone simulations in eQuest. This exercise allowed comparison of the field site performance of walk-ins across the United States. Climate zone locations include Miami, FL; San Antonio, TX; San Francisco, CA; Kansas City, MO; Omaha, NE; Billings, MT; and International Falls, MN. Outside of a few initial simulations, which included the validation simulation, four additional simulation setups were utilized, including a small cooler, small freezer, large cooler, and large freezer. Freezers that were simulated utilized a heated sub-floor maintaining a temperature of 50°F.

Becker analyzed the model load profile to determine if the Air-Conditioning, Heating, and Refrigeration Institute (AHRI) was using appropriate assumptions on how a typical walk-in cooler and freezer would be constructed and how it would be loaded. This information is also included in the model load analysis of this thesis.

(Blackmore 2012)

Tim Blackmore from Heatcraft did a presentation where he discussed methods used to size walk-in coolers and freezers (Blackmore 2012). The focus of the presentation was on walk-ins for supermarkets, of which numerous examples of walk-in coolers and freezers were given. Included with these example walk-ins, were details on walk-in dimensions, interior box temperature, exterior temperature, personnel door size, infiltration calculation, product load details, lighting load details, and equipment load details.

Typical load components were noted as follows:

- Lights: 1 to 1.5 W/ft^2 for normal storage; 2.5 to 3 W/ft^2 for docks/workrooms
- Motors: 6 hp/100000 ft3 coolers; 8 hp /100000 ft3 freezers; Fork-trucks between 4 to 5 hp
- People: 4 person/100000 ft3
- Spray K = .16 Btuh/in/F TD; Foam = .12; Styrofoam = .26

The infiltration load calculation is done utilizing an equation that is seen in a number of publications (Heatcraft 2008; Krack 1977). This is the same equation as utilized by Heatcraft (Equation 109), and varies from Krack (Equation 67) because it refers to a heat load (Btu/h) as opposed to a quantity of heat over a 24 hr period (Btu) and does not assume a constant air density.

$$Q_{Inf} = \frac{4.88\sqrt{H \cdot (T_H - T_C)}(1-E)(h_H - h_C)t_{open}}{\upsilon_H} \cdot \frac{A_{\inf}}{2} \cdot \frac{\min}{60s} \quad (109)$$

This infiltration equation estimates a load as opposed to calculating a flow rate with the Gosney Olama equation (Equation 18). To obtain a flow rate, one must multiply $Q_{\inf}$ by the specific volume of the incoming air and the specific enthalpy difference between the interior box air and the ambient air and divide by the quantity of the time that the door is open.

Ground Temperature

The ground temperature has a significant effect on the box heat load, especially for walk-in coolers. Walk-in freezers typically use some method to heat the walk-in floor to keep the ground from freezing and heaving upward. A higher ground temperature simply reduces the amount of heat that must be added below the walk-in to maintain the desired soil temperature.

(DOE 2010a)

A DOE Transcript (DOE 2010a) was put together to document a discussion with the public on the proposed rules to define test procedures for walk-in coolers and freezers. A wealth of information is contained in this document, as industry professionals were the primary attendants. Of particular interest to this work, is the information presented in relation to ground temperature for walk-ins. Typically, industry professionals assume a ground temperature of 55°F for walk-in design sizing. A ground temperature of 60°F may be assumed if a more conservative sizing is desired.

(Newell 2011)

Ground heat transfer is an important factor in building design. A three-dimensional, transient model was used to determine what level of insulation was necessary for the topic project. In the model a region of Earth extending 98.4 ft horizontally from the building perimeter and 16.4 ft deep was defined with a ground temperature of 50°F.

Soil is a dynamic substance with varying moisture content throughout the year. Varying soil properties and freezing of the soil were not incorporated into the model except for analysis of a range of soil thermal diffusivities (3.3 to 8.6 x 10^{-6} ft^2/s).

(Somrani et al. 2008)

Somrani et al analyzed two methods for determining ground heat transfer gains for ice rinks, with a goal of improving the design of these facilities in the future. These methods include an analytical method using an inter-zone temperature estimation technique and an implicit finite difference model analyzing heat transfer on a time-varying basis.

Loads on ice rink cooling systems include conductive loads from the floor, piping, pumps, and ice resurfacing; convective loads from the air to ice surface; and radiant loads between the ice surface and the ceiling (including light fixtures).

Soil temperatures and property assumptions for the model development were stated. The water table temperature (16.4 ft below the surface) is assumed to be 50°F throughout the year. The soil temperature near the surface fluctuates around 57.2°F (±19.2°F). Soil thermal diffusivity was assumed to be 6.94 x 10^{-6} ft^2/s.

(Chuangchid and Krarti 2001)

This paper analyzes floor heat load as it applies to residential and commercial radiant floor heating applications. Similar to Somrani et al. (Somrani et al. 2008), slab and soil temperatures are determined using the inter-zone temperature profile estimation technique. An identical water table temperature assumption is made.

(Smith 1989a)

Smith analyzed the psychrometrics of evaporator coil selection for a freezer operating with exit temperatures of -15 to -5°F. His calculations built toward solving for the required coil face area (CFA) (Equations 114 to 119).

$$CFA = 12{,}000 \frac{Btu-h}{ton} \cdot \frac{GRL}{RER} \tag{114}$$

$$GRL = NRL(1+FHR) \tag{115}$$

$$RER = \frac{C_1 \cdot u_{FV}(1-BF)(T_{ain}-T_{cs})}{\upsilon_{aout} \cdot LSHR} \tag{116}$$

$$BF = \frac{T_{aout}-T_{cs}}{T_{ain}-T_{cs}} \tag{117}$$

$$FHR = \frac{FHF}{RER-FHF} \tag{118}$$

$$FHF = FP \cdot HEP \frac{\upsilon_{ainr}}{\upsilon_{ain}} \tag{119}$$

(Faramarzi 1999)

Faramarzi discusses the components of the refrigeration load seen by a display case. Each load component is analyzed on a theoretical basis and opportunities are noted for improving the efficiency of display cases.

Display case cooling load components include transmission heat gain, radiation heat gain, infiltration sensible and latent heat load, lighting and motor heat gain, anti-sweat heater and defrost heat gain, product pull down load, and product latent heat of respiration.

Defrost types vary depending on the case temperature. Medium temperature cases with evaporator temperatures around 30°F use off-cycle defrost, which essentially means that refrigerant flow is shut off to the case. This allows the coil temperature to rise above 32°F to melt the frost. Reverse air defrost is used to speed up this process. This method pulls air from the store to melt frost. Neither of these methods require an auxiliary heat source in the classical sense.

Low temperature applications require an auxiliary heat source for defrost. Electric defrost works by using elements located near the evaporator. During defrost the fans and refrigerant flow are shut off and the elements are turned on. Hot gas defrost back-flows compressor discharge gas into the evaporator to defrost the coil from the inside. This method adds up to 85% more heat than required to melt the ice. The defrost period is controlled by either a timer or a temperature sensor.

Anti-sweat heaters typically operate continuously in low temperature or glass door cases, keeping ice from forming on some surfaces in the display case.

An 8 ft open vertical meat case was tested at ambient room conditions of 75°F dry bulb and 55% relative humidity. The resulting load distribution was 3% evaporator fans, 4% conduction loading, 12% radiant load, 8% lighting load, and 73% infiltration load. The infiltration load is the most significant portion of an open display case heat load even when an air curtain is utilized. To ensure the most energy efficient operation of an open case, proper air curtain flow is essential. Product should not block the flow path, and air curtains should be in good condition and orientation.

(Sujau et al. 2006)

A great deal of research has gone into optimizing the defrost cycle, as it is a significant load on the system. Factors influencing the formation of frost on a coil include the coil design, box operating conditions, and the efficiency of the defrost system. An additional variable is the frequency and duration of each defrost cycle.

Sujau et al. performed testing on a 10.8 x 14.4 x 9.8 ft tall walk-in box with electric defrost (4.6 kW) to determine the effect of defrost cycle period and frequency on the overall system efficiency. Defrost intervals of 6 to 30 hrs were tested. The walk-in was entered through a 3.9 x 7.9 ft door fitted with a strip curtain.

Lighting Load

(Kassa et al. 2004)

With a focus on consumer safety, a study of 57 food service establishments was performed to analyze the efficacy of qualitative light levels. Locations tested within each establishment included walk-ins and food preparation counters. Results are compared to the Federal Food Code (FDA 2001) specified as 10 foot-candles at 30 inches above the floor and 50 foot-candles on food preparation surfaces.

Of the 55 walk-in coolers tested, 12 used fluorescent lighting and 43 used incandescent. Measurements were taken in the center of the walk-ins. The geometric mean of the results from the fluorescent lit walk-ins was 15.3 foot candles, ranging from 6.4 to 85.5 ft-c. Seventy five percent of the coolers met the standard. The geometric mean of the results from the incandescent lit walk-ins was 3.4 ft-c, ranging from 1 to 16.7 ft-c. Only 7% of incandescently lit walk-ins met the standard. Analysis of 185 food preparation counters revealed a geometric mean of 38.7 ft-c, ranging from 2.9 to 196.8 ft-c. Only 23% of establishments met the standard.

Infiltration Load

(Fricke and Becker 2011)

Supermarkets utilize open and glass-doored display cases. Glass-doored display cases have a significant energy use advantage, but many retailers resist using them because they believe that these type display cases also reduce purchases, as the customer must open a door to get to the product. In support of using glass doored display cases, Fricke cites the fact that 50% of supermarket energy usage is for refrigeration of foods (Westphalen et al. 1996) and that 70% of the load on open display cases is due to infiltration (Faramarzi et al. 2002). Faramarzi found that installation of glass doors on a display case reduced the refrigeration load by 68%, reduced compressor power by 87%, and simultaneously reduced product temperature by 6°F. This allows for smaller central refrigeration systems and higher food quality.

Fricke performed testing on two similar 25,000 ft^2 supermarkets to highlight the differences in performance and sales due to installation of glass-doored display cases. The first store is in Osawatomie, KS with average sales of $80,000 per week. This location utilizes new doored cases to store dairy products (48 ft, 20 door) and alcoholic beverages (12 ft, 6 door). The second store is located in Wamego, KS with weekly sales of $140,000. This store displays alcoholic beverages in a new 24 ft open multi-deck case.

An analysis was done of the test data (Table 101) where the daily refrigeration load was defined by Equation 120, and compressor power (Equation 121) is determined via the energy-efficiency ratio (EER) of the refrigeration system, which is calculated using the adjusted dew-point temperature. The adjusted dew point was determined as the saturated temperature of the refrigerant at the evaporator pressure less 2°F.

$$\dot{Q}_{tot} = \frac{m_{tot}(h_{1s} - h_3)}{t_{tot} - t_{dt}} \frac{3600s}{hr} \tag{120}$$

$$CEC = \dot{Q}_{tot} \frac{t_{tot} - t_{dt}}{1000 \frac{W}{kW} \cdot EER} \cdot \frac{hr}{3600s} \tag{121}$$

Table 101. Test Result Comparison of Doored and Non-doored Display Cases

	Doored Display Case (Store #1)	Open Display Case (Store #2)
Compressor Electric Energy (kWh/day-ft)	0.46	1.76
Lighting Electric Energy (kWh/day-ft)	0.47	0.22
Evap. Fan Electric Energy (kWh/day-ft)	0.18	0.24
Anti-sweat Electric Energy (kWh/day-ft)	0.61	--
Total Electric Energy (kWh/day-ft)	1.71	2.21
Beverage Sales Increase Post Installation	27%	29%
Dairy Sales Increase Post Installation	-2.8%	-0.5%

Source: (Fricke and Becker 2011)

Open door case energy use varies depending on the store conditions. The system uses 25% more energy when the store relative humidity is increased from 20 to 45%. No correlation between door openings on doored cases and energy consumption was determined. Door openings were recorded for Store #1 at 4792 from April 13, 2009 to June 3, 2009 with a mean duration of 31 seconds, ranging from 1 seconds to 30 minutes, 42 seconds.

Ambient and Box Conditions

(Mitchell et al. 1992)

Store air conditioning and air dehumidification techniques are investigated by Mitchell et al. (Mitchell et al. 1992). Operating performance, operating cost, and installation costs are calculated and compared for multiple system types and configurations.

Information pertinent to this study included details on the conditions inside supermarkets. These conditions will affect both the conduction and infiltration load on the refrigerated space. Supply air inlet temperatures ranged from 60 to 70°F during the cooling season and were around 72°F during the winter months. Mitchell analyzed supermarket HVAC systems maintaining 58°F at 55% relative humidity and 71°F at 50% relative humidity.

(Thompson and Spinoglio 1996)

This report provides insight into the types of refrigerated rooms available for perishable product storage, suggested operating conditions for different products, and some crude sizing information (Thompson and Spinoglio 1996).

Walk-in boxes are noted to range from 6 x 8 ft in floor plan to 20 x 20 ft. Thompson et al. suggests that 12 to 18 inches be left above the product to allow good air flow through the facility, and that fiberglass insulation, polystyrene, or urethane foam board be used to obtain an R-value of at least 19 h-ft^2-°F/Btu. The discussion on types of refrigerated storage facilities is not confined to stationary buildings. In addition to the prefabricated and owner built cold rooms discussed, Thompson et al. discusses refrigerated transportation options, including rail cars, highway vans, and cargo containers.

Suggested storage temperatures for various fruits and vegetables are presented. These commodities are broken down into three groups, including those that are to be stored at temperatures of 32 to 36°F and 90 to 98% relative humidity, products stored at 45 to 50°F and 85 to 95% relative humidity, and finally items to be stored at temperatures ranging from 55 to 65°F and a relative humidity of 85 to 95%. Higher humidity conditions are obtained by oversizing the evaporator coil to allow a higher saturated suction temperature and thereby reduce the amount of condensate.

One set of product loading information is provided through an example cold room. The 16 x 19 ft walk-in operates at 33°F and has 581 boxes of product turnover every 24 hours.

(Henderson and Khattar 1999)

Typically, refrigerated storage conditions correspond to relatively high humidities. Moisture in the box air is condensed onto the evaporator coil during system operation. Not only does this increase the heat load on the coil by adding latent heat of vaporization, but it reduces coil performance by limiting air flow and insulating the heat transfer surfaces.

Field tests were performed on two supermarkets to investigate the relationship between humidity, defrost, and energy usage. The first store (33,400 ft^2) is in Minneapolis, operates 24 hours a day, and contains 2,849 ft^2 of walk-in coolers and freezers (Table 103). The initial system utilized hot gas, time-terminated defrost and 10 kW of anti-sweat heaters. The second store (50,000 ft^2) is in Indianapolis and operates 24 hours a day. This supermarket contains 3,685 ft^2 of total walk-in footprint (Table 102). The initial system uses electric, pressure terminated defrost. A 9kW anti-sweat heater is used to keep the door free to move. Daily average space humidity ranges from 30 to 50%, with ambient temperatures ranging from 59 to 89°F.

Modifications to the systems' control methods were performed to analyze the effects. These modifications included, temperature-terminated defrost and anti-sweat heater control. Test data was analyzed to understand the relationship between energy use, and ambient temperature and space humidity.

Energy usage trends linearly with the outside ambient conditions down to about 60 or 70°F. Below these conditions energy usage is still reducing but at a slightly flatter rate. Energy usage is essentially constant at ambient conditions of 20°F or less. The effect of space humidity is a little more difficult to determine, but a multi-linear regression was developed to simplify the analysis. At a 22% relative humidity, every 1% increase in the space relative humidity increases energy usage by 10 kWh/day (0.4% change in energy usage / 1% change in relative humidity for Store B).

Analyzing the benefit to the system by using temperature-terminated defrost presented some additional information about the effect of humidity on the system performance. At a relative humidity of 36%, the temperature-terminated system operates with the same efficiency as the time-terminated system. Above this humidity level the temperature-terminated system actually uses more energy, suggesting that the time-terminated system may be set at too short of a duration.

Table 102. Store A Refrigeration System Details

Zone	Description	Typ. Case Temp (°F)	Case Load (Btu/h)	Defrost	Anti-sweat Htr (W/ft)
A01	30 ft ice cream coffin, rear	-12	20,670	Hot gas	
A02	30 ft ice cream coffin, front	-18	20,670	Hot gas	
A03	9 door 22.5 ft frozen food, front	-3	14,400	Hot gas	38
A04	23 door 57.5 ft frozen food, West front	-6	35,800	Hot gas	38
A05	10 door 25 ft frozen food, West rear	-9	16,000	Hot gas	50
A06	15 door 37.5 ft frozen food, aisle front	-10	13,200	Hot gas	52
A07	10 door, 25 ft frozen food, aisle rear	-14	20,800	Hot gas	53
A08	36 ft frozen meat	-10	19,800	Hot gas	
A09	(2) 6 ft dual temperature coffins, meat	13	3,300	Hot gas	
A10	4 ft five deck, produce dressing	36	4,040	Offcycle	120
A11	8 door 20 ft bakery and dual temp.	-9	12,800	Hot gas	25
A12	258 ft^2 walk-in meat freezer, rear	-8	19,200	Hot gas	
A13	258 ft^2 main storage freezer	-2	19,600	Hot gas	
A14	6 ft Traulsen case, deli backroom	-7	6,000	Electric	
A14a	4 ft five deck salad dressing case	31	N/A	Offcycle	
B15	24 ft fresh meat, West	24	29,020	Hot gas	20
B17	36 ft smoked meat	32	22,620	Hot gas	
B18	32 ft five deck dairy, rear	32	51,360	Offcycle	
B19	48 ft five deck dairy, front	35	77,040	Offcycle	
B20	12 ft salad bar	42	14,250	Offcycle	
B22	28 ft service deli display	26	11,760	Offcycle	14
B23	(3) 18 ft produce islands	41	20,100	Offcycle	
B24	516 ft^2 walk-in produce cooler	37	14,500	Offcycle	
C25	24 ft fresh meat, East	23	38,040	Hot gas	20
C26	567 ft^2 walk-in meat cooler	32	38,600	Hot gas	
C27	32 ft produce case, North	33	25,600	Offcycle	
C28	44 ft produce case, East	33	25,600	Offcycle	
C29	24 ft five deck deli/bakery	32	38,520	Offcycle	
C30	24 ft service meat display	29	5,440	Offcycle	
C31	196 ft^2 walk-in coolers, deli and floral	36	7,000	Offcycle	
C33	344 ft^2 walk-in dairy cooler	41	9,000	Offcycle	
C34	710 ft^2 meat prep. Area	54	23,400	Offcycle	
D36	24 ft deli cheese display case	N/A	41,670	Offcycle	
Totals:	716 ft and 2,849 ft^2		719,800		9,840 W

Source: (Henderson and Khattar 1999)

Table 103. Store B Refrigeration System Details

Rack	Description	SST (°F)	Case Load (Btu/h)	Anti-sweat Heater (W/ft)	Fans & Const. Heaters (W/ft)	Case Lights (W/ft)
1L	(4) 8 ft & (1) 12 ft five deck dairy	20	75,768		20.1	57.0
1U	(3) 12 ft five deck dairy	20	61,992		20.1	63.3
2L	40 ft x 16 ft walk-in dairy cooler	-	42,200			
2L	24 ft x 16 ft walk-in produce cooler	20	32,100			
2U	(3) 8 ft five deck deli cases	15	41,400		61.3	132.3
3L	(2) 12 ft five deck deli cases	15	41,400		61.3	132.3
3U	26 ft x 16 ft walk-in meat cooler	-	23,800			
3U	22 ft x 10 ft meat holding cooler	20	15,200			
4L	30 ft x 16 ft meat freezer	-25	32,900			
4U	12 ft dual temperature meat coffin	-	4,020		26.8	
4U	12 ft dual temperature seafood coffin	-28	4,020		26.8	
5U	17 door 42.5 ft frozen food merch.	-20	24,650	81.4	32.2	32.5
5L	20 door 50 ft frozen food merch.	-20	29,000	81.9	32.2	30.4
6L	17 door 42.5 ft frozen food merch.	-20	24,650	81.4	32.2	32.5
6U	8 door 20 ft frozen seafood merch.	-	13,200	81.7	32.2	31.6
6U	3 door 7.5 ft frozen nutrition merch.	-25	4,950	243.8	38.3	33.7
7U	9 door 22.5 ft frozen food merch.	-	13,050	35.8	32.2	33.2
7U	4 door 10 ft frozen bakery merch.	-20	5,800	346.2	32.2	36.8
7L	17 door 42.5 ft ice cream merch.	-25	28,050		38.4	32.5
8L	(3) 6 ft promo end units, coffin	-35	7,200		32.6	
8U	10 ft x 8 ft seafood freezer	-25	8,900			
9U	10 ft x 16 ft deli freezer	-25	14,500			
9L	(4) 12 ft single-deck meat cases	-	17,040		9.6	
9L	(4) 12 ft fresh meat coffin cases	-	12,480		26.8	
9L	(3) 8 ft dairy promo, coffin	15	6,240		27.3	
10U	8 ft pizza display cases	-	3,800		34.5	
10U	(4) 8 ft cheese display cases	-	12,160			
10U	6 ft Tyler entrée case	15	3,800			
10L	(3) 16 ft European-style deli cases	-	17,120		57.5	19.2
10L	(4) service & (2) self-serve cases, 30 ft	-	13,800			
10L	10 ft deli counter w/refrig. and freezer	-	16,500			
10L	8 ft x 10 ft bakery retarder	-	7,500			
10L	10 ft x 10 ft deli cooler	20	8,700			
11U	(2) 8 ft service meat display	-	2,720			
11U	(2) 8 ft service seafood display	-	1,600			11.5
11U	8 ft x 8 ft seafood cooler	20	6,400			2.9
11L	200 ft^2 irregular shaped floral cooler	-	16,900			
11L	8 ft cut flower five deck	20	11,500			
12	(10) produce cases w/ air screen, 112 ft	-	82,880		12.9	23.0
12	(2) 8 ft five deck juice cases	20	27,552		20.1	54.6
13	(2) 12 ft and (5) 8 ft produce islands	-	62,400		16.4	
13	8 ft and (2) 12 ft juice-bar style cases	-	14,080		10.8	
13	12 ft back counter w/refrigeration	20	23,925			
14	715 ft^2 meat prep. area	-	35,800			
14	146 ft^2 service meat prep. area	35	7,300			
Total:	938 ft and 3,685 ft^2		960,947	18,745	22,325	23,529

Source: (Henderson and Khattar 1999)

At 55% relative humidity the anti-sweat heater operates at full capacity to keep surfaces above dew point. Through a simple heat transfer and psychrometric analysis, the rate at which the heater drops off with reduced humidity levels was increased from 3.5 kWh/day-RH% to 7.8 kWh/day-RH% for Store A and from 4.6 kWh/day-RH% to 9.1 kWh/day-RH% for Store B.

(Wichman and Braun 2009)

Wichman and Braun (Wichman and Braun 2009) experimented with the effectiveness of applying fault detection and diagnostics to a refrigeration system. The diagnostic method determines fault conditions through a logical approach to common issues and resulting effects that may be strongly linked to those issues. Through an understanding of what issues will likely happen in the system and by adding some logic to the system for calculation of system conditions based on only system temperatures, the number of sensors required was reduced to the minimum of 13.

A walk-in cooler and freezer were used to analyze the detection methodology by simulating system issues, including compressor valve leakage, liquid-line restriction, condenser fouling, evaporator fouling, ice accumulation, low refrigerant charge, and high refrigerant charge. The walk-in cooler operates with R-22 to maintain an interior temperature of 37.4°F and the freezer uses R-404a. The ambient temperatures simulated include 55, 75, and 95°F.

(Edwards 2010)

Comments made by Southern California Edison, San Diego Gas and Electric, and Sacramento Municipal Utility District to the “DOE Notice of Proposed Rulemaking Test

Procedures for Walk-In Coolers and Walk-In Freezers" are summarized in this report (Edwards 2010).

Proposed test conditions include:

- Cooler box temperature of 55°F
- Freezer box temperature of 20°F
- Infiltration temperature difference of 70°F
- Ambient dry bulbs of 35, 59, and 95°F
- Saturated suction temperatures of -40, -22, 18 and 35°F
- Condensing temperatures of 110°F with a 65°F superheat

Comments included information primarily related to system temperatures. Return gas temperature is around 41°F for walk-in coolers and 5°F for freezers. A cooler operates between 32 and 55°F, with a total chilled area of less than 3000 ft^2. Walk-in freezers are also defined as being smaller than 3000 ft^2 and operating at temperatures below freezing.

The capacity of the system may best be approximated by the enthalpy difference between the liquid line and 10°F superheated gas leaving the evaporator coil. The superheated gas temperature is typically 35°F for coolers and -10°F for freezers.

CHAPTER 4

DEVELOPMENT OF PROPOSED MODEL LOAD PROFILE

Following review of the current work and a thorough analysis of the available literature, information pertinent to developing an appropriate model load profile was compiled. Due to the volume of information and sources, the comparisons were reduced into different facets of the model load, including site conditions, box conditions, box conduction, infiltration, infiltration protection, product, lighting and occupancy, and miscellaneous loads. This analysis was performed for walk-in coolers and freezers with the exception of site conditions, infiltration protection, and door details, as they apply in a similar manner to both walk-in coolers and freezers.

The primary assumption of this work is that the occurrence of particular situations in real applications in industry is in a similar proportion to the number occurrences of that same situation in published works. Results were therefore windowed and averaged to determine what the most typical occurrence was. Using this as a guide, values for each component of the model load are proposed.

Walk-in Boxes

As mentioned previously, there are some parameters which are applicable to both walk-in coolers and freezers. In the present work, site conditions, infiltration protective devices, and door details are applied to the calculation of a model load in the same manner for walk-in coolers and freezers.

Site Conditions

Site conditions include the outdoor temperature and relative humidity, solar loading on the exterior of the walk-in box, ground temperature, and the indoor conditions of the facility that the walk-in is serving. Ground conditions will be discussed individually for walk-in coolers and freezers, as they vary for each application. Fifty-two articles presented information related to the site conditions that affect walk-in coolers and freezers, including (Adre and Hellickson 1989), (Altwies 1998), (Altwies and Reindl 1999), (Anonymous 2004), (Arias 2005), (Arias and Lundqvist 2006), (Becker et al. 2011), (Blackmore 2012), (Christensen and Bertilsen 2004), (CIGR 1999), (Cleland 1983), (Cooper 1973), (Devres and Bishop 1995), (DOE 2010b), (DOE 2010c), (Faramarzi 1999), (Foster et al. 2003), (Goetzler et al. 2009), (Gosney and Olama 1975), (Hayes and Stoecker 1969), (Heatcraft 2008), (Henderson and Khattar 1999), (Hendrix et al. 1989), (Huan 2008), (Kimber 1998), (Krack 1977), (Kun et al. 2007), (Longdill and Wyborn 1978), (Love and Cleland 2007), (Manske 1999), (Mitchell et al. 1992), (Nagaraju et al. 2001), (Navy 1986), (Patel et al. 1993), (PG&E 2007), (Pham and Oliver 1983), (Roy 2010), (SCE 2008), (SCE 2009), (Sekhar et al. 2004), (Sekhar and Lal 2005), (Stoecker 1998), (Stoeckle 2000), (Stoeckle et al. 2002), (Sujau et al. 2006), (Walker et al. 1990), (Walker 1992), (Walker 2001), (Walker and Baxter 2002), (Westphalen et al. 1996), (Wichman and Braun 2009), and (Zhang 2006).

All of the information discerned from the literature related to site conditions, whether reported from the field of industrial or commercial refrigeration, is applicable to this study. The raw data was analyzed, averaged for each source, and then averaged overall (Table 104 and Table 105). Through comparison of the results with the values included in the initial model load for the AHRI 1250/1251 Standard Load Spreadsheet (AHRI 2009a), new values are proposed for developing a model load for the AHRI 1250/1251 Standard (AHRI 2009b; AHRI 2009c).

Outdoor ambient temperatures, as noted in Table 104, refer to the air temperature that the outdoor condenser rejects heat to. For the case where the walk-in box is located outside, outdoor ambient conditions are coincident with the conduction load experienced by three of the four walls that surround the refrigerated space. When the box is located inside, conduction load is associated with the indoor site conditions (Table 105). The outdoor dry bulb noted by the literature ranged from 35 to 122°F, with an average value of 81.8°F. The ambient outdoor conditions at which the AHRI Load Spreadsheet was initially analyzed seem appropriate and will remain unchanged. The effect of sun loading on the space was altered from 12 hours at a 15°F increase in the roof temperature to 8 hours at a 20°F increase in roof temperature. The proposed roof temperature increase more closely approaches the average roof temperature increase determined from literature, of 33.2°F. The average temperature was not matched however due to the common application of highly reflective roof surfaces and the fact that the value in literature is coincident with the peak sun load.

The AHRI Load Spreadsheet (AHRI 2009a) assumed typical indoor (store) design conditions of 75°F dry bulb and 50% relative humidity (Table 105). The literature notes dry bulb temperatures ranging from 50 to 90°F for various applications, with an average temperature of 73.5°F, correlating well with the AHRI Load Spreadsheet value of 75°F. The relative humidities noted in literature also correlate well with the AHRI Load Spreadsheet value of 50%, ranging from 30 to 65%, with an average value of 49.6%. Values supplied by Becker et al. (Becker et al. 2011) from the FSTC test site for indoor ambient temperature and relative humidity were excluded from the range and average at the top of Table 105 because they refer to conditions at which the indoor condenser operates.

Table 104. Outdoor Site Conditions for Walk-in Coolers and Freezers

Outdoor Site Conditions for Walk-in Coolers and Freezers (1 of 6)					
	Outdoor Ambient Temp (°F)	Coincident Wet Bulb (°F)	Outdoor Ambient RH (%)	Sunlight directly on roof?	Roof Temp (°F)
Range	35 to 122	49.0 to 83.0	20 to 89		Ambient + 2 to 76°F
Average	81.8	69.2	54.4		Ambient + 33.2°F
Proposed	80, 95, 110			N – 5pm to 10am, Y – 10am to 5pm	Ambient + 20°F
(AHRI 2009a)	80, 95, 110			N – 8pm to 8am, Y – 8am to 8pm	Roof T = Ambient T + 15°F if there is direct sunlight
(Adre and Hellickson 1989)	100.4	69.8	20.8	Yes; Sol-air temp	36°F increase with 3300 kJ/m3 of global radiation; 0.15 for light-colored surfaces. South facing exterior wall: 27°F with 3300 kJ/m3 of global radiation.
(Altwies 1998)	South central Wisconsin; Design 998/78°F				
(Altwies and Reindl 1999)	North central U.S.; Design 98/78°F	78	40.6		

Outdoor Site Conditions for Walk-in Coolers and Freezers (2 of 6)					
	Outdoor Ambient Temp (°F)	Coincident Wet Bulb (°F)	Outdoor Ambient RH (%)	Sunlight directly on roof?	Roof Temp (°F)
(Becker et al. 2011)	Lincoln, RI (Chili's Restaurant #1070); Miami , FL 76.9F; San Antonio, TX 69.5; San Francisco, CA 58.0; Kansas City, MO 54.7; Omaha, NE 51.2; Billings, MT 48.2; International Falls, MN 38.2				
(Blackmore 2012)	90				
(Christensen and Bertilsen 2004)	Fakta Beder, Denmark				
(CIGR 1999)					Ambient + 76 (dark, flat roof); Ambient + 56 (light colored)
(Cleland 1983)	New Zealand				
(Cooper 1973)	89.6, 90	81.3	70		Ambient + 75.6 (dark, flat roof); Ambient + 55.8 (light colored)

Outdoor Site Conditions for Walk-in Coolers and Freezers (3 of 6)					
	Outdoor Ambient Temp (°F)	Coincident Wet Bulb (°F)	Outdoor Ambient RH (%)	Sunlight directly on roof?	Roof Temp (°F)
(Devres and Bishop 1995)			79.7		Ambient + 18 (roof); Ambient + 3.6, 5.4, and 5.4 (south, west, and east walls)
(DOE 2010c)	35, 59, 95				
(Goetzler et al. 2009)	90 (freezer), 95 (cooler)				
(Gosney and Olama 1975)	77	67.2	60		
(Hayes and Stoecker 1969)	75				
(Heatcraft 2008)	95				Ambient + 2 to 20 (7.58 average, Table 86)
(Henderson and Khattar 1999)	Minneapolis, MN (Store A), Indianapolis, IN (Store B); 59 to 89				
(Hendrix et al. 1989)	37 to 85		20 to 89		
(Huan 2008)	68	63.9	80		
(Kimber 1998)	59.0 (Safeway)				

Outdoor Site Conditions for Walk-in Coolers and Freezers (4 of 6)					
	Outdoor Ambient Temp (°F)	Coincident Wet Bulb (°F)	Outdoor Ambient RH (%)	Sunlight directly on roof?	Roof Temp (°F)
(Krack 1977)	95, 100	78.9, 83.0	50	Y / N	2 to 20 allowance based on construction and orientation, 7.6 average (Table 21); Add 50% if adjacent to highly reflective surfaces
(Kun et al. 2007)	122				
(Longdill and Wyborn 1978)	69.8				
(Love and Cleland 2007)	95				
(Manske 1999)	Milwaukee, WI		Milwaukee, WI		
(Navy 1986)	Dry bulb 2.5 percent; Wet bulb 5 percent			Yes. According to ASHRAE (1977)	
(Patel et al. 1993)	95	75	39.1		
(PG&E 2007)				Reflectance of 0.20 specified	40 increase
(Pham and Oliver 1983)	56.3 (#1 and 2); 59 (#3 and 6); 44.6 (#4); 57.2 (#5)				

Outdoor Site Conditions for Walk-in Coolers and Freezers (5 of 6)					
	Outdoor Ambient Temp (°F)	Coincident Wet Bulb (°F)	Outdoor Ambient RH (%)	Sunlight directly on roof?	Roof Temp (°F)
(Roy 2010)	Trois-Rivieres, Quebec			Yes. Reflective white roof.	
(SCE 2008)	Hourly weather for Baltimore, MD (54.7 average)	49.0	Hourly weather for Baltimore, MD (67.1 average)		140°F on 100°F ambient days
(Stoecker 1998)	1% Summer Design: 94 (Atlanta), 94 (Chicago), 102 (Dallas), 94 (Los Angeles), 92 (New York)	74 (Atlanta), 75 (Chicago), 75 (Dallas), 70 (Los Angeles), 74 (New York)	39.3 (Atlanta), 41.7 (Chicago), 28.6 (Dallas), 29.9 (Los Angeles), 43.2 (New York)	Yes	120 to 130 during the day
(Stoeckle et al. 2002)	Madison, WI				
(Walker et al. 1990)	Atlanta, GA; Boston, MA; Los Angeles, CA; Milwaukee, WI; Phoenix, AZ; St. Louis, MO				
(Walker 1992)	61.5 average	53.5 average	53.8 average		
(Walker 2001)	Worcester, MA (88); Washington, DC (93); Memphis, TN (98); Los Angeles, CA (93)	MA (77); DC (78); TN (80); CA (72)			

Outdoor Site Conditions for Walk-in Coolers and Freezers (6 of 6)

	Outdoor Ambient Temp (°F)	Coincident Wet Bulb (°F)	Outdoor Ambient RH (%)	Sunlight directly on roof?	Roof Temp (°F)
(Walker and Baxter 2002)	Marlborough and Webster, MA		Marlborough and Webster, MA		
(Westphalen et al. 1996)	100 (typical), 90 (prototypical freezer), 95 (prototypical cooler)				
(Wichman and Braun 2009)	55, 75, 95				
(Zhang 2006)	St. Louis, MO; Boston, MA; Dallas, TX		St. Louis, MO; Boston, MA; Dallas, TX		

Table 105. Indoor Site Conditions for Walk-in Coolers and Freezers

Indoor Site Conditions for Walk-in Coolers and Freezers (1 of 2)		
	Indoor (Store) Amb. Temp (°F)	Indoor (Store) Ambient RH (%)
Range	50 to 90	30 to 65
Average	73.5	49.6
Proposed	75	50
(AHRI 2009a)	75	50
(Adre and Hellickson 1989)	Wall 1: 50 to 82.4°F machine room	
(Altwies and Reindl 1999)	Warehouses adjacent to each other	
(Anonymous 2004)	Freezer door opens to cooler (typical)	
(Arias 2005; Arias and Lundqvist 2006)	71.6 (typical)	65 (typical)
(Becker et al. 2011)	Indoor walk-in box and condensing unit at 80F, 70, 75 (FSTC)	`41, 34, 55 (FSTC)
(DOE 2010b)	75	40 (weighted national average)
(DOE 2010c)	90	
(Faramarzi 1999)	75	55
(Foster et al. 2003)	68	
(Henderson and Khattar 1999)	59 to 89	30 to 50%
(Kimber 1998)	69.9 (Safeway), 57.7 (McDonald’s), 61.3 (Walnut Creek), 61.7 (Cameron Park Liquor)	
(Manske 1999)	Adjacent to worker warming room; Cooler shares one wall with freezer, one wall with the dock, and one wall with a heated space, else outdoors	

Indoor Site Conditions for Walk-in Coolers and Freezers (2 of 2)

	Indoor (Store) Amb. Temp (°F)	Indoor (Store) Ambient RH (%)
(Mitchell et al. 1992)	71, 58	50, 55
(Nagaraju et al. 2001)	86	
(SCE 2009)	Wall 1 and 3 are connected to walk-in coolers	
(Sekhar et al. 2004), (Sekhar and Lal 2005)	89.6	
(Stoeckle 2000)	Freezer (0°F) shares wall with 34°F loading dock; Else Outdoors; Values excluded from range and average.	
(Stoeckle et al. 2002)	Freezer Wall 1 adjacent to dock (34°F); Dock Wall 1 adjacent to freezer (0°F); Values excluded from range and average.	
(Sujau et al. 2006)	59	55
(Walker 1992)	73.2 average	36.9 average
(Walker 2001)	75 (typical), 68 to 75	55, 45, 40
(Zhang 2006)	75.2	55

Infiltration Protection

Operators of refrigerated spaces are always looking for opportunities to reduce the parasitic load on the space. In a number of applications, infiltration to the space is a considerable portion of the load. A number of researchers have worked to develop an understanding of what methods of infiltration protection work the best in different situations. Effectiveness of different infiltration methods, as found in the literature, are presented in Table 106.

Authors who have researched the effectiveness of infiltration protection devices include (Becker et al. 2011), (Chen et al. 1999), (Cleland et al. 2004), (Cole 1987), (Downing and Meffert 1993), (Hayes and Stoecker 1969), (Hendrix et al. 1989), (Longdill and Wyborn 1978), (Navy 1986), (Pham and Oliver 1983), (SCE 2008), (Stoecker 1998), (Takahashi and Inoh 1963), and (Van Male 1983). Their work gives information on infiltration reduction for open door with fork lift traffic, internal circulation, air curtains, strip curtains, swing-type plastic hinged doors, and rapid roll doors. The AHRI Load Spreadsheet (AHRI 2009a) assumes an infiltration protection effectiveness of 85%. This value is in the range of air curtains or strip curtains and was therefore re-used in the modified model load.

Table 106. Comparison of Door Blockages and Infiltration Protective Device Effectiveness

Comparison of Door Blockages and Infiltration Protective Device Effectiveness (1 of 3)	
Infiltration Type or Contributor	Effectiveness and Comments
Proposed	85%
(AHRI 2009a)	85%
Open Door Infiltration	
(Cole 1987)	Traffic passage frequency of one entrance/exit per minute or higher reduces the load by 15%
(Chen et al. 1999)	44% decrease in infiltration with traffic for unprotected rapid roll doors
(Pham and Oliver 1983)	Forklift traffic decreases infiltration by 21% for unprotected doorways
Internal Circulation Effects on Infiltration	
(Pham and Oliver 1983)	Internal circulation reduces air change by 9%.
Air Curtain Infiltration	
(Hayes and Stoecker 1969)	60 - 85%
(Longdill and Wyborn 1978), (Pham and Oliver 1983), (Downing and Meffert 1993)	49 to 83%
(Navy 1986)	30-40% (70-80% theoretically)
(Stoecker 1998)	65-80%
(Takahashi and Inoh 1963)	60-80%
(Longdill and Wyborn 1978)	horizontal: 82% no wind
(Longdill and Wyborn 1978)	horizontal: 0% direct wind > 2.5 m/s
(Longdill and Wyborn 1978)	horizontal: 13 to 37% with side wind at 5.2 to 2.6 m/s.
(Longdill and Wyborn 1978)	Horizontal Recirc: 82%

Comparison of Door Blockages and Infiltration Protective Device Effectiveness (2 of 3)	
Infiltration Type or Contributor	Effectiveness and Comments
Air Curtain Infiltration (Cont.)	
(Pham and Oliver 1983)	Horizontal, recirc, 60mm slot, 22.3 fps: 76%
(Takahashi and Inoh 1963)	Horizontal Recirc: 59%
(Pham and Oliver 1983)	Horizontal air curtain and plastic strip curtain: 91%
(Longdill and Wyborn 1978)	Vertical Non-circulating: 68-83%
(Pham and Oliver 1983)	Vertical, non-recirc, 115mm slot, 32.8 fps: 79%
(Takahashi and Inoh 1963)	Vertical Non-circulating: 54-81%
(Takahashi and Inoh 1963)	Vertical Recirc: 36-80% inside room
(Takahashi and Inoh 1963)	Vertical Recirc: 58% inside doorway
(Van Male 1983)	Vertical Double Non-recirc: 60-93%
(Pham and Oliver 1983)	Forklift traffic at one passage (entry and exit) per minute increases infiltration +105/-19% for air curtain doorways
Strip Curtain Infiltration	
(Becker et al. 2011)	75%
(Chen et al. 1999)	92% (good condition)
(Cleland et al. 2004)	92% (good condition)
(Downing and Meffert 1993)	86 to 96%
(Hendrix et al. 1989)	92 to 96%
(Pham and Oliver 1983)	93%
(Pham and Oliver 1983)	93% (no traffic)
(Pham and Oliver 1983), (Hendrix et al. 1989)	86 to 96% (no traffic)
(SCE 2008)	75%

Comparison of Door Blockages and Infiltration Protective Device Effectiveness (3 of 3)	
Infiltration Type or Contributor	Effectiveness and Comments
Strip Curtain Infiltration (Cont.)	
(Stoecker 1998)	90% (no traffic)
(Chen et al. 1999)	75% (one strip missing)
(Cleland et al. 2004)	80% (one strip missing)
(Cleland et al. 2004)	87% (small gap between strips)
(Chen et al. 1999)	Fork lift traffic at one passage (entry and exit) per minute: 62%
(Downing and Meffert 1993)	82 to 92% with one forklift entry/exit per minute
(Pham and Oliver 1983), (Hendrix et al. 1989)	82 to 92% (one forklift entry/exit per minute)
(Pham and Oliver 1983)	Forklift traffic at one passage (entry and exit) per minute increases infiltration +105/-19% for strip curtain doorways
(Stoecker 1998)	85% (with traffic)
(Pham and Oliver 1983)	Horizontal air curtain and plastic strip curtain: 91%
Swing-type Plastic Hinged Doors	
(SCE 2008)	95%
Rapid Roll Doors	
(Downing and Meffert 1993)	93% without traffic
(Downing and Meffert 1993)	79 to 85% with one entry/exit per minute (fully open 8 to 20 seconds with 1-2 second open/close)
(Cleland et al. 2004)	Defined via modified Gosney and Olama equation (Gosney and Olama 1975)
(Chen et al. 1999)	Defined via modified Tamm equation (Tamm 1965)
(Chen et al. 1999)	Door closed (air tightness): 99%

Door Details

Doors are used on walk-in coolers for personnel access, customer access, and pallet entry. Thirty authors presented door information, summarized in Table 107. Articles include (Anonymous 2004), (Aparicio-Cuesta and Garcia-Moreno 1988), (Becker et al. 2011), (Chen et al. 1999), (CIGR 1999), (Cleland et al. 2004), (Devres and Bishop 1995), (DOE 2010b), (DOE 2010c), (Foster et al. 2003), (Goetzler et al. 2009), (Hayes and Stoecker 1969), (Hendrix et al. 1989), (Huan 2008), (Kimber 1998), (Krack 1977), (Longdill et al. 1974), (Longdill and Wyborn 1978), (Magoo 2003), (Manske 1999), (Patel et al. 1993), (Pham and Oliver 1983), (Sherif et al. 2002), (Stoecker 1998), (Stoeckle 2000), (Stoeckle et al. 2002), (Sujau et al. 2006), (Walker 2001), (Walker and Baxter 2002), and (Westphalen et al. 1996).

Types of doors presented in the literature include reach-in doors for customer access to products, passage doors for personnel entry, and freight doors for pallet access with a hand truck or forklift. Walk-ins with reach-in doors have 3 to 76 doors per walk-in, with an average number of 18.5 doors. The number of passage doors per walk-in ranged from 1 to 2 in the literature, with an average of 1.3 doors. Passage doors range in size from 1.7 x 5.2 ft to 4 x 7.9 ft (width x height). The average door size dimensions are 3.2 x 6.4 ft. Some passage doors have a window in the door, ranging from 0.9 to 1.8 ft^2 in area, with an average area of 1.2 ft^2. Passage doors are typically of the same construction as the walk-in walls. Included in the list of passage door constructions is a range of reach-in door resistance values provided by the Department of Energy (DOE 2010c). These values are excluded from the range and average cited in the header of Table 107. Freight doors are used on refrigerated spaces depending on their application. Information on freight door size, presented in Table 107, was not filtered to remove information specific to larger refrigerated spaces, such as refrigerated warehouses, and should be interpreted with regard to applicability. Some of the values for door size presented by Foster et al. (Foster et al. 2003) were excluded from the range and average values for freight door height and width, as they refer to different partially open positions of a single freight door used during testing.

The AHRI Load Spreadsheet (AHRI 2009a) used one door for all walk-ins analyzed. Small walk-ins used a personnel door measuring 4 x 7 ft whereas large walk-ins used a freight door measuring 6 x 10 ft. Taking into account the findings from the literature, only the width of the small door was modified. The width of the door was reduced from 4 ft to 3 ft. As was depicted by the literature review, AHRI assumed the doors were of the same construction as the walk-in walls. This approach was not altered for determination of the modified model load.

Table 107. Door Details for Walk-in Coolers and Freezers

Door Details for Walk-in Coolers and Freezers (1 of 9)								
	Number of Reach-In Doors	Number of Passage Doors	Height of Passage Door (ft)	Width of Passage Door(ft)	Passage Door Total Glass Area (ft^2)	Passage Door R-value (h-ft^2-°F/Btu)	Height of Freight Door (ft)	Width of Freight Door (ft)
Range	3 to 76	1 to 2	5.2 to 7.9	1.7 to 4	0.9 to 1.8	14.3 to 32	7.0 to 14	4.5 to 14
Average	18.5	1.3	6.4	3.2	1.2	18.6	10.4	8.6
Proposed	0	1 sm	7 am	3 sm		32 (freezer), 25 (cooler)	10 lrg	6 lrg
(AHRI 2009a)	0	1 sm	7 sm	4 sm		32 (freezer), 25 (cooler)	10 lrg	6 lrg
(Anonymous 2004)		Swing (typical)		2 to 2.83				
(Aparicio-Cuesta and Garcia-Moreno 1988)	Yes							

Door Details for Walk-in Coolers and Freezers (2 of 9)

	Number of Reach-In Doors	Number of Passage Doors	Height of Passage Door (ft)	Width of Passage Door(ft)	Passage Door Total Glass Area (ft^2)	Passage Door R-value (h-ft^2-°F/Btu)	Height of Freight Door (ft)	Width of Freight Door (ft)
(Becker et al. 2011)	11 (Tedeschi Food Shop #110)	1 (Chili's Freezer, Small and Large Freezer, Tedeschi Food Shop #110, Chili's Beer and Food Cooler, Small and Large Cooler)	6.67 (FSTC, Small Freezer and Cooler); 7 (Prototypical)	3.25 (FSTC, Small Freezer and Cooler); 4 (Prototypical)		4" Extruded polystyrene sandwich panel, R-17.24 (Small Freezer and Cooler); R-32 (Large Freezer); 25 R-value (Large Cooler)	10 (Large Freezer and Cooler)	6 (Large Freezer and Cooler)
(Chen et al. 1999)			6.5	3.54			10.83, 11.81	7.87, 9.84
(CIGR 1999)				2 to 2.5				8 to 10

Door Details for Walk-in Coolers and Freezers (3 of 9)

	Number of Reach-In Doors	Number of Passage Doors	Height of Passage Door (ft)	Width of Passage Door(ft)	Passage Door Total Glass Area (ft^2)	Passage Door R-value (h-ft^2-°F/Btu)	Height of Freight Door (ft)	Width of Freight Door (ft)
(Cleland et al. 2004)		1 (A1, A2a, A2b, A7, (Longdill and Wyborn 1978); 2 (A3, A5, A6)	5.2 (Longdill and Wyborn 1978)	3.9 (Longdill and Wyborn 1978)			10.17 (A5); 10.83 (A6); 7.87 (A1); 10 (A7); 9.02 (A2); 9.84 (A3); 9.84 (cooler)	7.87 (A5, A6, and A7); 5.91 (A1 and A2); 8.53 (A3); 8.86 (cooler)
(Devres and Bishop 1995)							12.14	13.12
(DOE 2010b)	0, 3 ,8 , 50	1 sm, 1 med, 2 lrg	7	3	0.9 sm, 0.9 med, 1.8 lrg		9 med, 12 lrg	7 med, 7 lrg
(DOE 2010c)						0.3 to 0.43 (reach-in doors)		

Door Details for Walk-in Coolers and Freezers (4 of 9)								
	Number of Reach-In Doors	Number of Passage Doors	Height of Passage Door (ft)	Width of Passage Door(ft)	Passage Door Total Glass Area (ft^2)	Passage Door R-value (h-ft^2-°F/Btu)	Height of Freight Door (ft)	Width of Freight Door (ft)
(Foster et al. 2003)							2.26 (excluded from average and range), 10.5	7.55; 4.46, 3.28, 1.41 (excluded from average and range)
(Goetzler et al. 2009)	10	1	6.5	3				
(Hayes and Stoecker 1969)			7					
(Hendrix et al. 1989)							Freezer: 10 (#1 and 2), 12 (#3); Cooler: 12.67 (to Dry stores)	Freezer: 7 (#1 and 2), 8.5 (#3); Cooler: 8 (to Dry stores)
(Huan 2008)							9.84	9.19

Door Details for Walk-in Coolers and Freezers (5 of 9)								
	Number of Reach-In Doors	Number of Passage Doors	Height of Passage Door (ft)	Width of Passage Door(ft)	Passage Door Total Glass Area (ft^2)	Passage Door R-value (h-ft^2-°F/Btu)	Height of Freight Door (ft)	Width of Freight Door (ft)
(Kimber 1998)	13 (Safeway), 9 (Trader Joe's), 13 (Cameron Park Liquor)	1 (Safeway), 1 (Trader Joe's), 1 (McDonald's), 2 (Walnut Creek), 1 (Cameron Park Liquor)		3 (McDonald's)				
(Krack 1977)						3" insulation	11.5, 8	4.5, 6
(Longdill et al. 1974)			5.25	3.94				
(Longdill and Wyborn 1978)		2	5.41; 6.43	1.74; 4				
(Magoo 2003)							8 (outdoors); 14 (cooler)	9 (outdoors); 10 (cooler)
(Manske 1999)							8	10

Door Details for Walk-in Coolers and Freezers (6 of 9)								
	Number of Reach-In Doors	Number of Passage Doors	Height of Passage Door (ft)	Width of Passage Door(ft)	Passage Door Total Glass Area (ft^2)	Passage Door R-value (h-ft^2-°F/Btu)	Height of Freight Door (ft)	Width of Freight Door (ft)
(Patel et al. 1993)						Double glazing on glass doors		
(Pham and Oliver 1983)			6.50 (#6)	3.54 (#6)			11.81 (#4), 9.02 (#1 and 2), 9.51 (#3) , 6.99 (#5)	9.84 (#4), 10.01 (#1 and 2), 8.86 (#3), 5.81 (#5)

Door Details for Walk-in Coolers and Freezers (7 of 9)

	Number of Reach-In Doors	Number of Passage Doors	Height of Passage Door (ft)	Width of Passage Door(ft)	Passage Door Total Glass Area (ft^2)	Passage Door R-value (h-ft^2-°F/Btu)	Height of Freight Door (ft)	Width of Freight Door (ft)
(Sherif et al. 2002)		2	5.54	2.57	1.17	Outside to Inside: Aluminum sheet (AS), 3/4" Plywood, 3.5" Polystyrene molded beads with 2x4s on 2 ft centers (PS), 1/16" AS with Window of insulated double glazing with 1/2" air space for R-value of 14.29		
(Stoecker 1998)							7 to 14	5 to 14

Door Details for Walk-in Coolers and Freezers (8 of 9)

	Number of Reach-In Doors	Number of Passage Doors	Height of Passage Door (ft)	Width of Passage Door(ft)	Passage Door Total Glass Area (ft^2)	Passage Door R-value (h-ft^2-°F/Btu)	Height of Freight Door (ft)	Width of Freight Door (ft)
(Stoeckle 2000)							14 (freezer to dock); 8 (outdoors)	10 (freezer to dock); 9 (outdoors)
(Stoeckle et al. 2002)							8 (outdoors); 14 (freezer to dock)	9 (outdoors); 10 (freezer to dock)
(Sujau et al. 2006)		1	7.9	3.9				
(Walker 2001)	17-20 (ice-cream cases); 15-16 (food cases)							
(Walker and Baxter 2002)	Freezer: 76 (Reach-in #1), 23 (Reach-in #2), 52 (Reach-in #3); Cooler: 46 (Reach-in dairy)							

Door Details for Walk-in Coolers and Freezers (9 of 9)

	Number of Reach-In Doors	Number of Passage Doors	Height of Passage Door (ft)	Width of Passage Door(ft)	Passage Door Total Glass Area (ft^2)	Passage Door R-value ($h\text{-}ft^2\text{-}°F/Btu$)	Height of Freight Door (ft)	Width of Freight Door (ft)
(Westphalen et al. 1996)	20% of walk-in freezers have merchandising doors; 30% of walk-in coolers have merchandising doors, 10 (prototypical cooler)	At least 1	6.5	3				

Walk-in Coolers

In order to define a model load profile for walk-in coolers, information gathered from literature that related to the various components of a model load was analyzed to determine what is typical in application. Load components in addition to those analyzed above include box interior conditions, box conduction details, infiltration, product loading, lighting and occupancy, defrost details, and other miscellaneous loads.

Box Interior Conditions

Box interior conditions for the purposes of this study include the box interior temperature, box interior relative humidity, and the box dimensions. To ensure results are applicable to walk-in coolers, only interior dimensions for spaces that would be considered walk-ins ($\leq$ 3000 ft^2) are included in the analysis.

Forty-six authors presented operating temperatures and relative humidities of walk-in coolers. These reports include (Adre and Hellickson 1989), (Anonymous 2004), (Aparicio-Cuesta and Garcia-Moreno 1988), (Becker et al. 2011), (Blackmore 2012), (Chen et al. 2002), (Christensen and Bertilsen 2004), (CIGR 1999), (Cleland 1983), (Cleland et al. 2004), (Cooper 1973), (Devres and Bishop 1995), (DOE 2010b), (Edwards 2010), (Goetzler et al. 2009), (Gortner et al. 1948), (Heatcraft 2008), (Henderson and Khattar 1999), (Hendrix et al. 1989), (Krack 1977), (Kun et al. 2007), (Love and Cleland 2007), (Magoo 2003), (Manske 1999), (McHugh 2010), (Morris 2012), (Navy 1986), (Nelson 2011), (Kimber 1998), (Patel et al. 1993), (PG&E 2007), (Roy 2010), (Sand et al. 1997), (SCE 2008), (SCE 2009), (Sekhar and Lal 2005), (Sekhar et al. 2004), (Sezgen and Koomey 1995), (Stoecker 1998), (Stoeckle 2000), (Stoeckle et al. 2002), (Sujau et al.

2006), (Thompson and Spinoglio 1996), (USDA 2010), (Westphalen et al. 1996), and (Wichman and Braun 2009). Information is summarized in Table 108.

Thirty-five articles presented information on box dimensions, including (Adre and Hellickson 1989), (Anonymous 2004), (Arias 2005), (Becker et al. 2011), (Blackmore 2012), (Christensen and Bertilsen 2004), (Cooper 1973), (DOE 2010a), (DOE 2010b), (DOE 2010c), (Elsayed 1998), (Foster et al. 2003), (Goetzler et al. 2009), (Heatcraft 2008), (Henderson and Khattar 1999), (Kimber 1998), (Knudsen and Pachai 2004), (Krack 1977), (Love and Cleland 2007), (McHugh 2010), (Morris 2012), (Patel et al. 1993), (PG&E 2004), (SCE 2008), (SCE 2009), (Sekhar et al. 2004), (Sekhar and Lal 2005), (Sezgen and Koomey 1995), (Sujau et al. 2006), (Thompson and Spinoglio 1996), (Walker et al. 1990), (Walker 1992), (Walker 2001), (Walker and Baxter 2002), and (Westphalen et al. 1996). Interior box dimensions are listed in Table 109.

The calculation of the AHRI 1250 (AHRI 2009b)/1251 (AHRI 2009c) walk-in cooler box load equations used interior conditions of 35°F and 90% relative humidity. In reviewing the literature, cooler box interior temperatures ranged from 0 to 86°F, with relative humidities ranging from 40 to 100%. Considering the corresponding average values of 38.6°F dry bulb and 80.0% relative humidity, it was determined that AHRI used an appropriate value for the dry bulb temperature. As noted by Becker et al. (Becker et al. 2011), the relative humidity is overstated by AHRI and will be reduced to 80% for the proposed model load.

Two walk-in sizes were used to develop the AHRI 1250 (AHRI 2009b)/1251 (AHRI 2009c) walk-in cooler equations. The small walk-in was 8 x 8 ft with an 8 ft ceiling and the large walk-in was 50 x 50 ft in floor plan with a 20 ft ceiling. Box sizes seen in the literature ranged from 5 x 6 ft to 50 x 63 ft boxes. Floor areas ranged from 30 to 3000 ft^2, with volumes ranging from 7 to 89,841 ft^3. The average values for box width, length, height, floor area, and volume were 14.9 ft, 24.1 ft, 12 ft, 702.5 ft^2, and 8,863.7 ft^3, respectively. Values used initially by AHRI to develop the model load seem appropriate and will remain unchanged for this study.

Table 108. Box Interior Conditions for Walk-in Coolers

Box Interior Conditions for Walk-in Coolers (1 of 5)		
	Box Interior Temperature (°F)	Box Interior Relative Humidity (%)
Range	0 to 86	40 to 100
Average	38.6	80.0
Proposed	35	80
(AHRI 2009a)	35	90
(Adre and Hellickson 1989)	32	95
(Anonymous 2004)	35	
(Aparicio-Cuesta and Garcia-Moreno 1988)	32 to 39	
(Becker et al. 2011)	37.5 (Tedeschi Cooler), 38 (Chili's Beer and Food Cooler); 35 (Prototypical); 35 (Small and Large Cooler)	
(Blackmore 2012)	35	
(Chen et al. 2002; Cleland et al. 2004)	37; 41 (loading dock)	80 (loading dock)
(Christensen and Bertilsen 2004)	28.4	

Box Interior Conditions for Walk-in Coolers (2 of 5)		
	Box Interior Temperature (°F)	Box Interior Relative Humidity (%)
(CIGR 1999)	50 (grain storage), 55 (sorting and handling of some products); 35.6 to 39.2 (seed potatoes); 39.2 to 41 (consumer potatoes); 41 to 46.4 (French fries/dried potatoes); 44.6 to 50 (chip industry); 42.8 (starch and derivatives); 32 (most common), 32 to 41 (onion curing and storage, cassava), 26.6 to 35.6 (onion bulbs), 32 to 33.8 (spring onion), 37.4 (D. trifida yam), 54.5 to 62.6 (D. alata yam), 55.4 (D. cayenesis yam), 86 (water yam), 50 (elephant yam), 60.8 (white and yellow yam), 60.8 to 64.4 (Indian yam), 77 (chinese yam), 44.6 to 50 (tunnia), 40 to 55.9 (taro)	75; 90 - 100 (perishable commodities); 65 to 75 (onion bulbs), 90 to 100 (spring onion), 85 to 90 (cassava), 95 (D. cayenesis yam), 60 (water yam), 80 (white and yellow yam, tannia), 60 to 65 (Indian yam), 70 to 90 (tannia)
(Cleland 1983)	46.4 to 50 (cutting rooms); 32 to 41 (beef chiller)	
(Cooper 1973)	32 (most fresh fruits and vegetables); 29.3 to 46.4 (example); 29.3 (pears); 50 (peaches, figs, grapes, apricots, avocadoes); 39.2 to 41 (oranges); 46.4 to 48.2 (mangoes); 55.4 (green bananas)	90 or higher for leafy vegetables.
(Devres and Bishop 1995)	39.2	90
(DOE 2010a)	32 to 45, 32 to 41, 32 to 55	
(DOE 2010b)	35	60
(DOE 2010c)	35	
(Edwards 2010)	55	
(Foster et al. 2003)	32	
(Goetzler et al. 2009)	35 (baseline); 10 to 15 (meats, seafood), 15 to 25 (dairy, produce, beer, juice, meat walk-in coolers), 35 and above (produce, flowers)	
(Gortner et al. 1948)	41	

Box Interior Conditions for Walk-in Coolers (3 of 5)		
	Box Interior Temperature (°F)	Box Interior Relative Humidity (%)
(Heatcraft 2008)	Heatcraft Table 7; 38.4 average; 30 (walk-in)	Heatcraft Table 7; 85.3 average; 80 to 85 typical
(Henderson and Khattar 1999)	37 (produce), 32 (meat), 36 (deli, floral), 41 (dairy), 54 (meat prep)	
(Hendrix et al. 1989)	40	
(Krack 1977)	36 (#1), 35 (milk); Krack Table 15 (flowers and nursery); 28 - 40 (storage cooler), 28 - 34 (meat chill), 35 - 45 (work rooms); Krack Table 28 (cheese); 40 (beer); 28 to 32 (beer); 28 (nuts); 32 to 60 (Krack Table 42)	65 (beer and nut storage); 85 (fruit)
(Kun et al. 2007)	64.4, 71.6	
(Love and Cleland 2007)	36.5	
(Magoo 2003)	32 or higher (cooler); 50 (cooler dock); 34 (cooler)	
(Manske 1999)	56 to 64 (banana rooms); 45 to 55 (tomato rooms); 34 (cooler); 45 (cooler dock)	80; 87 (cooler); 65 (cooler dock)
(McHugh 2010)	32 to 55	
(Morris 2012)	35	90 (minimum weight loss), 80 to 85 (general storage), 65 to 80 (beer, wine, drugs), 50 to 65 (prep/cutting rooms, candy, film)
(Navy 1986)	32 to 34 (meat); 40 (dairy); 35 to 40 (perishable); 45 to 55 (film); 50 (fruits, vegetables); 40 (medical); 40 to 50 (cooler); 28 to 35 (chill cooler)	85 (meat); 80 (dairy); 90 (perishable); 40 (film); 55 to 65 (fruits, vegetables); 50 (medical)

Box Interior Conditions for Walk-in Coolers (4 of 5)		
	Box Interior Temperature (°F)	Box Interior Relative Humidity (%)
(Nelson 2011)	32	
(Kimber 1998)	39.4 (Safeway), 40.0 (McDonald's), 37.2 (Walnut Creek), 46.8 (Cameron Park Liquor)	81.8 (Safeway), 76.3 (McDonald's)
(Patel et al. 1993)	32 to 45	
(PG&E 2007)	40	
(Roy 2010)	39	
(Sand et al. 1997)	50 (prep rooms); 28 to 45 (Medium temp for meat, fish, and dairy cases and walk-in coolers for meats and produce)	
(SCE 2008)	35	
(SCE 2009)	36.2 (warehouse); 40.6 (cooler A); 38 (cooler B)	
(Sekhar and Lal 2005) (Sekhar et al. 2004)	38.8	
(Sezgen and Koomey 1995)	35	
(Stoecker 1998)	32 to 37 (apples); 38 to 40 (avocadoes); 56 to 58 (bananas); 32 (cabbage, broccoli, peas); 32 to 45 (cheese, lettuce); 29 to 31 (pears); 30 to 35 (poultry); 31 to 32 (strawberries); 38 to 50 (tomatoes); 32 to 50 (seeds); honeydew (41 to 50); 31 to 32 (peaches); 40 to 45 (beans); 37 to 39 (potatoes); beef (39); 33 to 40 (milk); 37 to 80 (dough); 50 to 57 (meat processing); 39 to 46 (docks)	90 to 95% (typical); 90 to 100 (lettuce); 90 to 98 (peas); 50-65 (seed)
(Stoeckle 2000)	34 (storage cooler)	
(Stoeckle et al. 2002)	34	
(Sujau et al. 2006)	35.1 (low load), 36.7 (high load)	82.3 (low load), 70.0 (high load)

Box Interior Conditions for Walk-in Coolers (5 of 5)		
	Box Interior Temperature (°F)	Box Interior Relative Humidity (%)
(Thompson and Spinoglio 1996)	32 to 65	85 to 98
(USDA 2010)	0 to 50	
(Westphalen et al. 1996)	35 (prototypical); 35 and above (produce and flowers); 10 to 15 (meats and seafood); 15 to 25 (dairy, produce, beverage, and meat walk-ins); 25 to 35 (dairy and produce walk-ins and prep rooms)	
(Wichman and Braun 2009)	37.4	

Table 109. Box Interior Dimensions for Walk-in Coolers

Box Interior Dimensions for Walk-in Coolers (1 of 5)					
	Box Width (ft)	Box Length (ft)	Box Height (ft)	Floor Area (ft^2)	Box Volume (ft^3)
Range	5 to 50	6 to 63	6.6 to 32.8	30 to 3000	7 to 89,841
Average	14.9	24.1	12.0	702.5	8863.7
Proposed	8, 50	8, 50	8, 20	64, 2500	512, 50000
(AHRI 2009a)	8 sm, 50 lrg	8 sm, 50 lrg	8 sm, 20 lrg	64 sm, 2500 lrg	512 sm, 50000 lrg
(Adre and Hellickson 1989)	43.5 (CA)	63 (CA)	32.81	2738 (CA)	89841 (CA)
(Anonymous 2004)	5 to 9	6 to 19	7.5, 8.5, or 9.5 inside	30 to 171	
(Arias 2005)					7, 10, 20, 40, 100, 500, 1000, 3000
(Becker et al. 2011)	9 (Tedeschi Food Shop #110); 6.25 (Chili's Restaurant #1070 Beer Cooler); 7.5 (Chili's Food Cooler); 15.8 (Prototypical); 7.42 (Small Cooler); 50 (Large Cooler)	32 (Tedeschi Food Shop #110); 7 (Chili's Restaurant #1070 Beer Cooler); 17 (Chili's Food Cooler); 15.8 (Prototypical); 9.42 (Small Cooler); 50 (Large Cooler)	8 to 30; 9 (Chili's Restaurant #1070 Beer Cooler); 8.5 (Chili's Food Cooler); 12 (Prototypical); 7.92 (Small Cooler); 20 ft (Large Cooler)	288 (Tedeschi Food Shop #110); 43.75 (Chili's Restaurant #1070 Beer Cooler); 127.5 (Chili's Food Cooler); 249.6 (Prototypical); 69.9 (Small Cooler); 2500 (Large Cooler)	393.8 (Chili's Restaurant #1070 Beer Cooler); 1083.8 (Chili's Food Cooler); 2995.2 (Prototypical); 553.6 (Small Cooler); 50000 (Large Cooler)

Box Interior Dimensions for Walk-in Coolers (2 of 5)					
	Box Width (ft)	Box Length (ft)	Box Height (ft)	Floor Area (ft^2)	Box Volume (ft^3)
(Blackmore 2012)	20, 8 (convenience store)	30, 28 (convenience store)	16, 8 (convenience store)	600, 224 (convenience store)	9600, 1792 (convenience store)
(Christensen and Bertilsen 2004)	10.83 (walk-in with glass doors)	16.73 (walk-in with glass doors)	8.53 (walk-in with glass doors)	181.2	1,545.5
(Cooper 1973)	32.81	32.81	14.76	1,076.5	15,889.1
(DOE 2010a)				< 3000	
(DOE 2010b)	8.0 sm, 12 med, 25 lrg	10 sm, 20 med, 30 lrg	7.6 sm, 9.5 med, 12 lrg	80 sm, 240 med, 750 lrg	608 sm, 2280 med, 9000 lrg
(DOE 2010c)				< 3000	
(Elsayed 1998)			6.56 to 13.12		
(Foster et al. 2003)	15.74	19.03	12.47	299.67	3736.0
(Goetzler et al. 2009)	10	24	8.5	240; 80 to 250 (typical)	2,040
(Heatcraft 2008)	6, 8 (convenience), 14 (beef)	30, 28 (convenience), 16 (beef)	22, 8 (convenience, beef)	180, 224 (convenience, beef)	3960, 1792 (convenience, beef)

Box Interior Dimensions for Walk-in Coolers (3 of 5)					
	Box Width (ft)	Box Length (ft)	Box Height (ft)	Floor Area (ft^2)	Box Volume (ft^3)
(Henderson and Khattar 1999)				Store A: 516 (produce), 567 (meat), 196 (deli, floral), 344 (dairy), 710 (meat prep). Store B: 640 (dairy), 384 (produce), 416 (meat), 220 (meat), 80 (bakery), 100 (deli), 64 (seafood), 200 (floral), 715 (meat prep), 146 (service meat prep)	
(Kimber 1998)	9.5 (Trader Joe's), 9.0 (McDonald's), 13.0 (Walnut Creek)	28.3 (Trader Joe's), 12.0 (McDonald's), 24.3 (Walnut Creek)	8.5 (Trader Joe's), 8.0 (McDonald's), 8.7 (Walnut Creek)	268.9 (Trader Joe's), 108.0 (McDonald's), 315.9 (Walnut Creek)	5,695 (Safeway), 2286 (Trader Joe's), 864 (McDonald's), 2,739 (Walnut Creek), 1,880 (Cameron Park Liquor)
(Knudsen and Pachai 2004)					1230, 844, 1117, 1337
(Krack 1977)	10 (#1), 6 to 12 (beer), 20 (beef)	12 (#1), 6 to 30 (beer), 30 (beef)	8 (#1), 10 (beer); 16 - 24 (storage cooler), 10 - 12 (work rooms), 16 (beef)	120 (#1), 36 to 360 (beer), 600 (beef)	960 (#1), 360 to 3600 (beer), 9600 (beef)

Box Interior Dimensions for Walk-in Coolers (4 of 5)					
	Box Width (ft)	Box Length (ft)	Box Height (ft)	Floor Area (ft^2)	Box Volume (ft^3)
(Love and Cleland 2007)					21
(McHugh 2010)				Walk-in cooler has chilled area <3000 ft^2	
(Morris 2012)	16, 15	25, 35	8, 10	400, 525	3200, 5250
(Patel et al. 1993)				2038	
(PG&E 2004)					50 to 2000
(SCE 2008)				250 sm, 500, 1000, 2500 lrg	
(SCE 2009)				96 (cooler A); 300 (cooler B)	
(Sekhar and Lal 2005) (Sekhar et al. 2004)	7.6	7.6	9.2	57.8	531.4
(Sezgen and Koomey 1995)	30, 10	50, 20	20, 8	1500, 200	30000, 1600
(Sujau et al. 2006)	10.8	14.4	9.8	155.5	1524.1
(Thompson and Spinoglio 1996)	6; 10; 16 (example room); 20; 9	8; 12; 19 (example room); 20; 45			
(Walker 1992; Walker et al. 1990)				480 (meat); 437 (produce); 540 (dairy); 800 (meat prep); 96 (fish); 120 (meat holding); 70 (deli); 80 (bakery retard)	

Box Interior Dimensions for Walk-in Coolers (5 of 5)					
	Box Width (ft)	Box Length (ft)	Box Height (ft)	Floor Area (ft^2)	Box Volume (ft^3)
(Walker 2001)	6 (bakery, cooler #1, cheese and deli #2); 12 (deli and floral); 15 (meat #1 and meat prep); 10 (meat #2); 14 (produce); 20 (dairy)	8 (bakery, cooler #1, and deli #2); 12 (deli and floral); 14 (cheese); 36 (produce); 40 (meat #1, dairy, and meat prep); 15 (meat #2)		48 (bakery, cooler #1, and deli #2); 132 (deli and floral); 84 (cheese); 600 (meat #1 and meat prep); 150 (meat #2); 800 (dairy); 504 (produce)	
(Walker and Baxter 2002)				140 (floral walk-in); 2195 (walk-in coolers); 1010 (meat prep); 520 (produce prep); 857 (dairy); 153 (deli); 48 (raw); 110 (fish); 984 (produce); 1056 (meat); 120 (bakery)	
(Westphalen et al. 1996)	10	24	8.5	Meat cooler: 400, Other coolers: 2600; Prototypical: 240	2040

Box Conduction

The box conduction load is defined by the wall, ceiling, and floor construction and the exterior and interior conditions of the walk-in. Site conditions, with exception of the ground temperature, are noted in Table 104 and Table 105, and interior box conditions are listed in Table 108. Box construction details, as found in the literature, are noted in Table 111, with some additional conduction load details analyzed in Table 112.

Information on ground conditions for walk-in coolers was presented by (Adre and Hellickson 1989), (Chuangchid and Krarti 2001), (Cooper 1973), (Devres and Bishop 1995), (DOE 2010a), (DOE 2010b), (DOE 2010c), (Heatcraft 2008), (Krack 1977), (Magoo 2003), (Navy 1986), (Roy 2010), (SCE 2008), (Somrani et al. 2008), (Stoecker 1998), (Stoeckle 2000), and (Stoeckle et al. 2002). Information presented by (Chuangchid and Krarti 2001) and (Somrani et al. 2008) is not included in this discussion, as it does not directly apply to walk-ins. This information is summarized in Table 110.

Walk-in cooler construction details were presented by (Adre and Hellickson 1989), (ASHRAE 1998), (Anonymous 2004), (Becker et al. 2011), (Blackmore 2012), (CIGR 1999), (Cooper 1973), (DOE 2010a), (DOE 2010b), (DOE 2010c), (Goetzler et al. 2009), (Heatcraft 2008), (Krack 1977), (Magoo 2003), (Manske 1999), (Navy 1986), (Patel et al. 1993), (PG&E 2007), (Roy 2010), (SCE 2008), (Sekhar et al. 2004), (Sekhar and Lal 2005), (Stoecker 1998), (Stoeckle 2000), (Stoeckle et al. 2002), (Sujau et al. 2006), (Thompson and Spinoglio 1996), and (Westphalen et al. 1996). Convective film coefficients related to calculation of the conduction load for walk-in coolers were presented by (DOE 2010b), (Krack 1977), (SCE 2008), and (Stoecker 1998).

Floor conditions are specified by some authors as a heat load per footprint and by others as a ground temperature which is applied to calculate the heat load. Floor heat loads ranged from 0.97 to 8.61 Btu/hr-ft^2, with an average value of 3.29 Btu/hr-ft^2. Coolers typically do not have floor heating installed, as sub-floor freezing is less of a concern. The local ground temperature is therefore applied for determining the floor heat load. Values in literature ranged from 40 to 80°F, with an average value of 57.6°F. Additional works that should be noted but were too involved for application in this work include (Somrani et al. 2008), (Schulak and Horvay 2002), (Chuangchid and Krarti 2001), (ISO 1998), (Krarti et al. 2004), (Krarti 1999), and (Krarti et al. 1988). The AHRI Load Spreadsheet (AHRI 2009a) uses a value of 50°F, which through comparison to literature values is low. Increasing the ground temperature associated with the model load profile from 50°F to 55°F should better represent typical ground temperatures seen by walk-in boxes. When applied to the model load calculation, this ground temperature correlated with a floor heat load of 1.28 Btu/hr-ft^2.

Box construction typically varies slightly for the ceiling, walls, and floor of a walk-in cooler. Ceiling R-values ranged from 7.9 to 60 h-ft^2-°F/Btu for walk-in coolers, with an average value of 28.4 h-ft^2-°F/Btu. This was comparable to the results found for the walls of a cooler, which ranged from 7.9 to 42.5 h-ft^2-°F/Btu, with an average value of 23.9 h-ft^2-°F/Btu. The floor R-value was the exception, as it appears that about half of the time walk-in coolers do not have insulated floors. The cited values ranged from on slab to 42.5 h-ft^2-°F/Btu, with an average value of 14.0 h-ft^2-°F/Btu. AHRI had assumed R-values of 25 h-ft^2-°F/Btu for the walls, ceiling, and floor. This insulation level was maintained for the walls and ceiling in the modified model load but was reduced to 15 h-ft^2-°F/Btu for the floors, to more closely match the literature findings.

The conduction load is tied to the space via convection on the interior and exterior surfaces with the exception of the floor. A majority of designers and the AHRI Load Spreadsheet (AHRI 2009a) ignore the effects of convection, but for this analysis the convection film coefficients were applied to the conduction load calculation. Values differed for the ceiling, wall, and floor, ranging from 0.17 to 0.87 h-ft^2-°F/Btu. Values of 0.5 h-ft^2-°F/Btu will be used for the wall and ceiling calculation, and 0.6 h-ft^2-°F/Btu will be used for the floor calculation.

Table 110. Floor Conditions for Walk-in Coolers

Floor Conditions for Walk-in Coolers (1 of 2)		
	Floor Heat Load (Btu/hr-ft2)	Ground Temperature (°F)
Range	0.97 to 8.61	40 to 80
Average	3.29	57.6
Proposed	1.28	55
(AHRI 2009a)	0.6	50
(Adre and Hellickson 1989)		Average soil temperature at 51 cm
(Cooper 1973)		79.7
(Devres and Bishop 1995)		41
(DOE 2010a)		55 or 60
(DOE 2010b)	R-22.42, R-28.03, R-33.64: 8.61, 1.48, 1.21 (36 ft2); 7.31, 1.43, 1.18 (71.4 ft2); 6.9, 1.41, 1.17 (80 ft2); 4.4, 1.31, 1.1 (240 ft2); 2.97, 1.13, 0.97 (750 ft2); 3.04, 1.18, 1.01 (1200 ft2); (DOE finite element analysis)	DOE Test Procedure: 60
(DOE 2010c)	for A ≤ 750 ft2: qfloor = 33.153 A^-0.364; for A > 750 ft2: qfloor = 0.0002 A + 2.84 (floorless coolers)]	65

Floor Conditions for Walk-in Coolers (2 of 2)		
	Floor Heat Load (Btu/hr-ft2)	Ground Temperature (°F)
(Heatcraft 2008)		Ground temperatures ranging from 40 to 80, 66.3 average
(Krack 1977)		55
(Magoo 2003)	No heating	50 under slab
(Navy 1986)	Facilities > 5000 ft^2 or < 32°F: heated by glycol; Other facilities may use ventilation pipes as alternative heating method	
(Roy 2010)	Yes	
(SCE 2008)		50
(Stoecker 1998)		Typically 41 to 50°F below slab
(Stoeckle 2000)	3.6 (Jekel and Reindl 2000)	67.5 below slab
(Stoeckle et al. 2002)	3.6 (Jekel and Reindl 2000)	

Table 111. Box Construction for Walk-in Coolers

Box Construction for Walk-in Coolers (1 of 6)			
	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
Range	7.9 to 60	7.9 to 42.5	On slab to 42.5
Average	28.4	23.9	14.0
Proposed	25	25	15
(AHRI 2009a)	25	25	25
(Adre and Hellickson 1989)			Floor + 51 cm of soil
(ASHRAE 1998)	35.21 to 39.75	23.85 to 31.8	19.87
(Anonymous 2004)	4" Urethane insulation sandwiched between aluminum, SS, high-density urethane or fiberglass (typical)	4" Urethane insulation sandwiched between aluminum, SS, high-density urethane or fiberglass (typical)	4" Urethane insulation sandwiched between aluminum, SS, high-density urethane or fiberglass (typical)
(Becker et al. 2011)	4 inch extruded polystyrene sandwich panel, R-17.24 (Small Cooler); 25 R-value sandwich panel (Large Cooler)	3.5, 4 or 5.5 inch thick insulated panels (typical); R-28 (Title 20); 6 inch Polystyrene foam insulated panels (Prototypical); 4 inch extruded polystyrene sandwich panel, R-17.24 (Small Cooler); 25 R-value sandwich panel (Large Cooler)	12 in soil, 6 inch concrete, 4 inch extruded polystyrene sandwich panel (Small Cooler, R-18.9); 12 in soil, 6 inch concrete, 25 R-value insulating material (Large Cooler, R-27.03)

Box Construction for Walk-in Coolers (2 of 6)

	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
(Blackmore 2012)		Spray K = 0.16 Btu / hr-in-F, Foam = 0.12, Styrofoam = 0.26; 4 inch Styrene (Convenience store)	6" concrete slab (Convenience store)
(CIGR 1999)	Insulated with rigid board, foam materials, or loss fill. 60 R-value common Range of 30 to 60.	0.30 W/m^2-K typical; 0.25 W/m^2-K without mechanical cooling; 0.20 W/m^2-K ceilings, 0.25 W/m^2-K walls, 0.505 W/m^2-K floors; Walls insulated with fiberglass batting, rigid urethane foam boards or sprayed on foam. 20 to 40 R-value	
(Cooper 1973)	R-42.53, 6" thick	Brick and concrete or light steel frames and panels of pre-cast polyurethane bonded between steel or aluminum outer coating and plywood or other permeable materials as inner face. 42.53 h-ft^2-ºF/Btu; 5 inches thick	42.53 h-ft^2-ºF/Btu; 5 inches thick
(DOE 2010a)		No foam insulation degradation seen in practice	
(DOE 2010b)	4” XPS or PU, R-24 (independent testing lab)	4” XPS or PU, R-24 (independent testing lab)	Uninsulated
(DOE 2010c)		4" Foam: 16 to 32 h-ft^2-ºF/Btu; 4" Vacuum insulated panel: 120 to 200 h-ft^2-ºF/Btu (excluded from average for Table 111). Insulating value degrades over time due to diffusion of gases and moisture absorption.	

Box Construction for Walk-in Coolers (3 of 6)			
	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
(Goetzler et al. 2009)	28; minimum of 4 in of blown polyurethane foam	28; minimum of 4 in of blown polyurethane foam; 28.6; 4 in thick (baseline)	28; minimum of 4 in of blown polyurethane foam
(Heatcraft 2008)	Walk-in: 4 in Styrene (R = 16.7, k = 0.24)	1 to 10 inches of cork/mineral wool, glass fiber/poly styrene, sprayed urethane, foamed in place urethane; Heatcraft Table 13 - General Standard for Insulation Thickness in Storage Rooms; 2 to 4 inches of styrofoam or polyurethane. Walk-in: 4 in Styrene (R = 16.7, k = 0.24)	Walk-in: 6" concrete slab
(Krack 1977)	3-4" foamed in place urethane; 3" foamglass to 6" foamed in place urethane (7.9 to 23.1 h-ft^2-°F/Btu, 14.9 average)	3-4" foamed in place urethane; 3" foamglass to 6" foamed in place urethane (7.9 to 23.1 h-ft^2-°F/Btu, 14.9 average)	6 in concrete on grade, no insulation or 4 or 6" styrene
(Magoo 2003)	same as walls	Insulation: 21.19 to 27.93 with a 2 in layer of concrete on either side (cooler); Outside to inside: 2 in concrete, 1.85 in insulation, 5.98 in concrete; 10.25 R-value (dock)	Outside to inside: 11.8 in concrete, 1.38 in insulation; 8.28 R-value (cooler); Outside to inside: 11.8 in concrete, 1.38 in insulation; 8.28 R-value (dock)
(Manske 1999)	25.09 lightweight	24.85 lightweight	1 (perimeter)

Box Construction for Walk-in Coolers (4 of 6)

	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
(Navy 1986)	25 to 35 (40 to 50°F box temp); 30 to 40 (28 to 35°F)	17 to 30 (40 to 50°F box temp); 25 to 35 (28 to 35°F)	Perimeter only (40 to 50°F box temp); 15 to 20 (28 to 35°F)
(Patel et al. 1993)		2 inches of foam insulation on all walls	
(PG&E 2007)	R-24 to R-40 (range) of wood frame plywood with 4" of blown insulation; R-23, 0.8 absorptivity (prototypical cooler/dock)	R-23 to R-40 (range) of either 4" to 5" expanded urethane metal clad panels or sandwiched concrete panels; R-30 (typical for cooler or dock); R-20 (prototypical cooler/dock)	Uninsulated concrete on grade
(Roy 2010)	40	30 with natural lighting through high insulated R-7.5 translucent windows	
(SCE 2008)	> R-28	> R-28	
(Sekhar et al. 2004), (Sekhar and Lal 2005)	5.9 inches of mineral wool	5.9 inches of mineral wool	5.9 inches of mineral wool

Box Construction for Walk-in Coolers (5 of 6)			
	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
(Stoecker 1998)		Combining a structural material with insulation and applying an abrasion resistant coating is a traditional method. Insulation bonded to metal skin and tongue and grooved together is becoming more popular. Common materials (Btu/hr-ft-ºF): Cellular foamglass (.029), Cellular polyurethane (.013), Expanded polystyrene (.02), Extruded polystyrene (.015), Glass fiber (.021), Polyisocyanurate (.012); Recommended temperature application for thickness of insulation: Ambient at 2 inches (R-value of 16.6), Ambient to 32°F at 3 inches (R-value of 20.8)	
(Stoeckle 2000)		10.53 (loading dock)	
(Stoeckle et al. 2002)		21.74, Thermally lightweight (foam-core sandwich) and massive (concrete) studied	Concrete slab on grade with an imbedded glycol heating loop
(Sujau et al. 2006)	5.9 inches of polystyrene sandwich panel	5.9 inches of polystyrene sandwich panel	5.9 inches of polystyrene sandwich panel
(Thompson and Spinoglio 1996)		Foam-board insulation encased in metal; 2 x 6 framing for recommended R-19 fiberglass, polystyrene, or urethane insulation	

Box Construction for Walk-in Coolers (6 of 6)			
	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
(Westphalen et al. 1996)		4 to 5 inches; R value 28.6 to 35.75; 3 to 4 inches of blown polyurethane foam typical R-27; 4 inches of insulation at R-28.6 (prototypical)	

Table 112. Convection Film Coefficients for Walk-in Coolers

Convection Film Coefficients for Walk-in Coolers			
	External Equivalent Convective Film Coefficient (h-ft^2-°F/Btu)	Internal Equivalent Convective Film Coefficient (h-ft^2-°F/Btu)	Floor Equivalent Convective Film Coefficient (h-ft^2-°F/Btu)
Range	0.17 to 0.68	0.25 to 0.68	0.61 to 0.87
Average	0.35	0.52	0.70
Proposed	0.5	0.5	0.6
(AHRI 2009a)			
(DOE 2010b)	0.68	0.25	0.87
(Krack 1977)	0.17 (outdoor, winds < 15 mph)	0.63 (still air)	
(SCE 2008)			0.61 (ASHRAE 2005)
(Stoecker 1998)	0.17 (15 mph winter, any orientation); 0.25 (7.5 mph summer, any orientation)	0.68 (still air, vertical surface, heat flow horizontal)	0.61 (still air, horizontal surface, heat flow upward)

Infiltration

Components for calculation of the infiltration load include the difference in temperature of the interior and exterior spaces, the schedule of door openings and their duration, and the calculation method utilized. Space interior temperatures were analyzed in Table 108, and exterior and interior site conditions are listed in Table 104 and Table 105, respectively. Door opening schedules and infiltration calculation methods are compared in Table 113.

Twenty-eight authors presented information related to infiltration, including (Arias 2005), (Becker et al. 2011), (Blackmore 2012), (Christensen and Bertilsen 2004), (Cleland 1983), (Cleland et al. 2004), (Cole 1987), (Cooper 1973), (Devres and Bishop 1995), (DOE 2010b), (Elsayed 1998), (Foster et al. 2003), (Fricke and Becker 2011), (Heatcraft 2008), (Hendrix et al. 1989), (Kimber 1998), (Krack 1977), (Magoo 2003), (Manske 1999), (Patel et al. 1993), (PG&E 2007), (Roy 2010), (SCE 2008), (Stoecker 1998), (Stoeckle 2000), (Stoeckle et al. 2002), and (Westphalen et al. 1996).

Primary infiltration calculation methods used in the literature include Gosney and Olama (Equation 25), Downing and Meffert (Equation 74), Heatcraft (Equation 109), Krack (Equation 67), and Air Change Method (Table 22, Table 77, and Table 88). Gosney and Olama is the most widely used and most accepted. Coincidently, this is the method used in the AHRI Load Spreadsheet (AHRI 2009a). Door opening schedules and the duration of door openings were all captured from literature. The number of door openings and their duration are difficult to compare due to variations in the test period and complexity of some schedules. When analyzed as an open door percentage, literature values for the door being open ranged from 1.3 to 100%. The average door open percentage was 12.7%. The AHRI Load Spreadsheet door opening schedule equates to an open percentage of 1.18% for the small cooler and 2.78% for the large cooler. The average open door percentage was skewed by numerous door opening schedules where the door was open all the time. After analyzing the available literature, a 5.4% open door percentage was chosen for the proposed model load.

Table 113. Infiltration Details for Walk-in Coolers

Infiltration Details for Walk-in Coolers (1 of 8)				
	Infiltration Calculation	Miscellaneous Infiltration Calculation Information	Number of Door Openings	Duration of Door Openings
Range		Average Open Door Percentage: 1.3% to 100%	9.3 to 960	8.8 to 316.5 seconds
Average		Average Open Door Percentage: 12.7%	174.2	75 9 seconds
Proposed	Gosney Olama	Average Open Door Percentage: 5.4%	13 per hour - 7 am to 7 pm	30 seconds
AHRI Load Spreadsheet (2009a)	Gosney Olama Equation (ASHRAE Refrigeration Handbook); Air Density Factor: 0.97	Average Open Door Percentage: 1.18% (sm), 2.78% (lrg)	30 openings per hour – 6am to 7am, 2 openings per hour – 7am to 7pm (sm); 32 openings per hour – 6am to 7am, 4 openings per hour – 7am to 7pm (lrg)	30 seconds per opening – 6am to 7am, 5 seconds per opening – 7am to 7pm (sm); 30 seconds per opening – 6am to 7pm (lrg)
(Arias 2005)		30 Air changes per 24 hrs for 247 ft^3 room, 24.5 for 353, 17 for 706, 11.5 for 1413, 7 for 3531, 2.7 for 17657, 2.7 for 35315, 1.05 for 105944		

Infiltration Details for Walk-in Coolers (2 of 8)

	Infiltration Calculation	Miscellaneous Infiltration Calculation Information	Number of Door Openings	Duration of Door Openings
(Becker et al. 2011)	Heatcraft Air Changes (Small and Large Cooler)		9.3 (Tedeschi Cooler), 22 (Chili's Food Cooler)	288 s (Tedeschi Cooler), 60 s (Chili's Food Cooler)
(Blackmore 2012)	Vel = 4.88 * sqrt(H * TD)(1-%) Delta h / specific volume; Air change method; 4.9 avg air changes (9600 ft3 box size)			
(Christensen and Bertilsen 2004)			Store operation is 12 hrs per day	
(Cleland 1983)			3% of the time	

Infiltration Details for Walk-in Coolers (3 of 8)

	Infiltration Calculation	Miscellaneous Infiltration Calculation Information	Number of Door Openings	Duration of Door Openings
(Cleland et al. 2004)		Good sliding door seals leak at 0.26 to 0.52 ft^3 / min-ft	Door opening frequency typically cited as 1 per minute. Loading dock doors at 40 movements per hour where a movement is defined as a forklift entering and then exiting through the doorway.	Rapid-roll doors: Fully open 8 to 20 seconds, 1-2 seconds open/close (Downing and Meffert 1993), Equivalent fully open time of 17.5 seconds typical; Sliding doors: 4 to 10 second open/close, Equivalent fully open time is typically 30 seconds; 17, 15.7, 15.4, 8.8 (test points rapid-roll)

Infiltration Details for Walk-in Coolers (4 of 8)

	Infiltration Calculation	Miscellaneous Infiltration Calculation Information	Number of Door Openings	Duration of Door Openings
(Cole 1987)	Modified Gosney Olama	Flow drops off after several minutes due to partitioning effect of product stacked around door; Krack Corporation calculation method: V = 4.88 (H) (TD)0.5; Figure 17 - Air infiltration rate in cfm vs. room volume; 18 - Air infiltration in air changes vs. room volume; ϕ (Time Related Factor), Gosney Olama: 0.5 typical for high traffic door operated cyclically; Lag Time: 15 to 60 seconds		
(Cooper 1973)	Air change method	12 empty volume air changes per day assumed		
(Devres and Bishop 1995)			56 openings per day	20 seconds / opening

Infiltration Details for Walk-in Coolers (5 of 8)				
	Infiltration Calculation	Miscellaneous Infiltration Calculation Information	Number of Door Openings	Duration of Door Openings
(DOE 2010b)	Gosney Olama Equation (ASHRAE Refrigeration Handbook)	Door Flow Factor: 0.8 (ASHRAE Fundamentals); Flow between panel joints: 0.13 ft3/hr per ft2 external surface (DOE research)	60 openings per day for passage and freight doors (DOE Test Procedure)	12 seconds per opening for passage and freight doors. Passage and freight doors stand open an additional 15 minutes per day (DOE Test Procedure)
(Elsayed 1998)		10 or 20 mm can represent a gap in a closed door that is not well sealed. Two-dimensional model citing Gosney (1975), Jones (1983), and Cole (1984)		
(Foster et al. 2003)	CFD	Closed door leakage rate of 3.6 cfm; Lag time of 1.6 seconds, drop off time of 29 seconds		10, 20, 30, 40 seconds; 3.28 ft door takes 6 seconds to open/close
(Fricke and Becker 2011)			6.3 openings per hour	31 seconds (mean)

Infiltration Details for Walk-in Coolers (6 of 8)				
	Infiltration Calculation	Miscellaneous Infiltration Calculation Information	Number of Door Openings	Duration of Door Openings
(Heatcraft 2008)	Average air change or Heatcraft infiltration calculation: [(4.88)sqrt(door height) (area/2) (minutes open) sqrt(temp diff F) (enthalpy incoming - box air)] [(1-X)]/specific volume of incoming air	Table 88; Table 90- Glass Door Loads; Table 92		
(Hendrix et al. 1989)		Door Flow Factor: 1; lag time of less than 3 seconds		
(Kimber 1998)			72.8 openings per day (Safeway passage door)	6.4 hours per day (Safeway passage door)
(Krack 1977)	Air change method or infiltration calculation: [(527)sqrt(door height*TD)*A/2*(minutes open) (delta h)]	Air changes (Table 22)		480 min open / 24 hrs; 5 min/hr (beef carcass)

Infiltration Details for Walk-in Coolers (7 of 8)

	Infiltration Calculation	Miscellaneous Infiltration Calculation Information	Number of Door Openings	Duration of Door Openings
(Magoo 2003)	Gosney (1975)	Door Flow Factor: 0.8 (ASHRAE Fundamentals)	1/hr per door (outdoor dock)	Always open, Normal practice to keep dock-cooler doors open at all times if temperature difference is less than 50°F (cooler); Dock: Open 30 seconds per passage + Open-close time of 30 seconds = 60 seconds per opening (outdoor); Doors to cooler are always open
(Manske 1999)				10 min/hr each (cooler/outside), 15 min/hr each (cooler/dock), 12 min/hr each (warehouse/cooler), 5 min/hr each (dock/outside) for second shift and half of these values for first shift:

Infiltration Details for Walk-in Coolers (8 of 8)				
	Infiltration Calculation	Miscellaneous Infiltration Calculation Information	Number of Door Openings	Duration of Door Openings
(Patel et al. 1993)		Significant portion of load		
(PG&E 2007)	Air change method	0.1 ACH		
(Roy 2010)		1800 cfm; Refrigerated warehouses typically don't have ventilation systems		
(SCE 2008)		1.00 air changes per hour (sm), 0.68 ACH, 0.46 ACH, 0.25 ACH (lrg) (Heatcraft Refrigeration)		
(Stoecker 1998)	Gosney and Olama			10 to 15 seconds (typical)
(Stoeckle 2000)	(Downing and Meffert 1993)			2 min/hr each (outside); 4 min/hr each (freezer)
(Stoeckle et al. 2002)	(Downing and Meffert 1993)			4 min/hr ea (freezer); 2 min/hr ea (outdoors)
(Westphalen et al. 1996)		Good door sealing systems		

Product

Another load that varies greatly from walk-in to walk-in is the product load. Depending on the application, some walk-ins pull-down product temperatures to the storage temperature whereas others just maintain product temperature. Some coolers must continuously combat product respiration whereas others store only packaged products or non-perishable products that do not evaporate-off moisture. The length of storage is fairly consistent (1 to 20 days) with coolers, as the duration of product storage is limited by its shelf life.

Through researching the available literature, information on product type (Table 114), pull-down characteristics (Table 114), product loading (Table 114), and product thermal properties (Table 115), was obtained. Thirty researchers contributed information to this study, including (Adre and Hellickson 1989), (Anonymous 2004), (Becker and Fricke 1996a), (Becker et al. 2011), (Blackmore 2012), (Chourasia and Goswami 2007), (CIGR 1999), (Cooper 1973), (Devres and Bishop 1995), (DOE 2010b), (Goetzler et al. 2009), (Heatcraft 2008), (Henderson and Khattar 1999), (Krack 1977), (Love and Cleland 2007), (Magoo 2003), (Manske 1999), (Navy 1986), (Patel et al. 1993), (PG&E 2007), (Roy 2010), (SCE 2008), (SCE 2009), (Stoeckle et al. 2002), (Stoecker 1998), (Thompson and Spinoglio 1996), (USDA 2010), (Wade 1984), (Walker and Baxter 2002), and (Westphalen et al. 1996).

Coolers store many different products, ranging from fruits and vegetables to film reels. The type of product strongly affects what storage conditions the box will operate at (Table 108), which is a factor in all heat load calculations. As noted previously, the amount of the heat that must be removed from the incoming product to reach the storage condition varies greatly for each application. Research showed that product pull-down temperature differences range from 0 to 68°F, with an average value of 23.8°F. AHRI (AHRI 2009a) used a value of 10°F for the pull-down temperature difference. This value was not altered for the modified model load, as a disproportional number of references were for harvest applications where field heat must be removed from the product. Product loading schedules also vary widely, but due to the fact that product temperature pull-down is over an extended period ranging from 8 to 720 hrs (115.7 hr average) for coolers, the load may be assumed to be constant over the entire day. The average value is skewed by the large number of applications where crops are pulled down from field temperature to storage temperature over numerous weeks following harvest. A more typical value would be 24 hrs or more. Assuming that the load may be applied as a constant value of the entire day to the model load, the product load was analyzed on a loading ratio basis. Information in the literature showed a daily loading ratio ranging from 1.23 to 4.06 lbs of product per ft^3 of refrigerated space, with an average value of 1.91. The larger values of product loading were coincident with smaller coolers. The AHRI Load Spreadsheet used values of 12.11 and 1.60 lbs of product per ft^3 for the small and large coolers, respectively. The smaller cooler product loading seems high considering the data obtained from the literature, resulting in an adjustment of the product loads to 4.0 and 1.8 lbs of product per ft^3 of refrigerated space for the small and large coolers, respectively.

A wealth of information was obtained from the literature on the thermal properties of various products. The AHRI Load Spreadsheet (AHRI 2009a) uses a product specific heat of 0.9 Btu/lb-°F. After analyzing the literature, this value remained unchanged for the modified model. Additionally, it is apparent that it is common for products be stored so that their respiration heat load is added to the space. For this study, it is assumed that half of the products stored are unpackaged and therefore add a respiration heat load to the space, with a respiration coefficient of approximately 0.0292 Btu/lb-hr.

Table 114. Product Loading for Walk-in Coolers

Product Loading for Walk-in Coolers (1 of 8)				
	Product Type	Product Pull-Down Temp. Difference (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3 of refrigerated space)
Range		0 to 68		1.23 to 4.06
Average		23.8		1.91
Proposed	Fruits and Vegetables (50% packaged)	10		4 sm, 1.8 lrg
(AHRI 2009a)	Fruits and Vegetables	10	6,200/8 lb per hour – 6am to 2pm (sm); 80,000/8 lb per hour – 6am to 2pm (lrg)	12.11 sm, 1.60 lrg
(Adre and Hellickson 1989)			Pull-down and heat of respiration; Capacity of 1650 tons of fruit which can be loaded in 14 days. Fruit enters at 95°F or less	
(Anonymous 2004)			28 lbs of solid food per 1 ft^3 of open storage area	
(Becker et al. 2011)	Fruit / vegetables, snap beans (Prototypical, Small Cooler, Large Cooler)	10 (Prototypical, Small and Large Cooler)	6480 lb/day (Prototypical); 57 lb/hr with 24 hr pull-down (Small Cooler); 80,000 lb/hr 6am to 7am with 8hr pull-down (Large Cooler)	2.16 (Prototypical), 2.47 (Small Cooler), 1.6 (Large Cooler)
(Blackmore 2012)	Beer and milk (Convenience store)	50 (beer), 5 (milk)	2000 lb of beer, 200 lb of milk (Convenience store), 24 hr pull-down	1.23 (Convenience Store)

Product Loading for Walk-in Coolers (2 of 8)

	Product Type	Product Pull-Down Temp. Difference (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3 of refrigerated space)
(Chourasia and Goswami 2007)	Potatoes in gunny bags made of meshed jute threads	35.1	Contributes to 2/3rds of refrigeration load during transient cooling periods of cold storage. 6.72 day half cool time, 10 to 30 day pull down time. Storage for 1 to 8 months. 25.97 to 54.89 W/ton average load. 4000 ton capacity. The daily loading rate should be 3-4% of the total loading capacity.	
(CIGR 1999)		3.6	3500 J/kg-K (pull-down); 1000-4200 J/kg-K (respiration); Temperatures and storage lengths given (onions, spring onions, cassava, tubers, yams, taro, tannia)	
(Cooper 1973)	Fresh fruits and vegetables, seed potatoes, bulbs, grain-drying, rice drying, fruit ripening or coloring for bananas, canned pears, plums and peaches	48.6	Single farm crops four to six months. Product stacked 13.1 ft high in a 16.4 ft room. Product covers 65% of floor area and 46.3% of volume. 220 kg/m^3 of product per insulated volume. Receive 10% of nominal capacity of warm product on each day during harvest season, taking three days to cool from 80.6°F to 32°F. Respiratory activity of fruit at 250 kcal/ton-day	1.37

Product Loading for Walk-in Coolers (3 of 8)				
	Product Type	Product Pull-Down Temp. Difference (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3 of refrigerated space)
(Devres and Bishop 1995)			More than 50% of total daily load; Half cooling time of 10 days; Capacity of 1,400,000 kg of potatoes; 4 to 5 months to unload product.	
(DOE 2010b)		10		4 sm, 2 med, 2 lrg
(Goetzler et al. 2009)	Meat, Seafood, Produce, Dairy, Deli, Beer/Juice, Flowers			

Product Loading for Walk-in Coolers (4 of 8)

	Product Type	Product Pull-Down Temp. Difference (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3 of refrigerated space)
(Heatcraft 2008)	Heatcraft Table 7; Banana ripening room: Bananas shipped in fiberboard cartons (10 x 16 x 22") holding 42 lb net (47 lb gross) with 864 boxes in a carload lot.		6,200 - 8,000 lbs/day (500 - 3,000 ft^3); 8,000 - 11,000 lbs/day (3000 - 4,600 ft^3); 11,000 to 17,000 lbs/day (4,600 - 8,100 ft^3); 17,000 - 26,000 lbs/day (8,100 - 12,800 ft^3); 26,000 - 33,000 lbs/day (12,800 - 16,000 ft^3); 33,000 - 40,000 lbs/day (16,000 - 20,000 ft^3); 40,000 - 56,000 lbs/day (20,000 - 28,000 ft^3); 56,000 - 66,000 lbs/day (28,000 - 40,000 ft^3); 66,000 - 110,000 lbs/day (40,000 - 60,000 ft^3); 110,000 - 150,000 lbs/day (60,000 - 80,000 ft^3); Heatcraft Table 9 - Heat Loads of Keg and Bottled Beer; Heatcraft Table 10 - Carcass Heat Loads; Product Loading Density from 13 to 57 lb/ft^3 (Heatcraft Tab le 7); Banana ripening room: 1644 lb turnover per week; 1°F/hr pull down rate; Pull down in 24 hours, average load; Heatcraft Table A - Product Cooling Loads for Walk-In Coolers	4.06 lb/day-ft^3 (500 - 3,000 ft^3); 2.5 (3000 - 4,600 ft^3); 2.20 (4,600 - 8,100 ft^3); 2.06 (8,100 - 12,800 ft^3); 2.05 (12,800 - 16,000 ft^3); 2.02 (16,000 - 20,000 ft^3); 2.00 (20,000 - 28,000 ft^3); 1.79 (28,000 - 40,000 ft^3); 1.76 (40,000 - 60,000 ft^3); 1.86 (60,000 - 80,000 ft^3); Heatcraft Table 9 - Heat Loads of Keg and Bottled Beer; Heatcraft Table 10 - Carcass Heat Loads
(Henderson and Khattar 1999)			Store A: Deli, floral, dairy, meat prep; Store B: Dairy, produce, meat, bakery, deli, seafood	

Product Loading for Walk-in Coolers (5 of 8)

	Product Type	Product Pull-Down Temp. Difference (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3 of refrigerated space)
(Krack 1977)	Fruit, beef carcass, beer	10 (milk); 0 to 68 (Krack Table 42); 20 (beer); 66 (beef carcass); 44 (beer)	10 to 15 day storage period; 20% of load (#1), 37.6% of load (300 gallons of milk / day) with 10 hr pull-down (milk); 20% daily turn of 900 cases (27.4% of load); 1ºF / hr typical; 20% daily inventory turn with 20°F product temperature drop (beer cooler); 15ºF, 24 hr pull-down (beef carcass); 24 hr pull-down (beer storage)	Average product loads: 1.2 to 2.7 Btu/hr-ft^3 (30,000 to 250 ft^3, 1.7 Btu/hr-ft^3 average); Krack Table 42 (specific product loads)
(Love and Cleland 2007)	600 - 12 fl oz soda cans	58.5	22.7 to 24.9 hr pull-down time according to pull-down tests	

Product Loading for Walk-in Coolers (6 of 8)

	Product Type	Product Pull-Down Temp. Difference (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3 of refrigerated space)
(Magoo 2003)	Milk or cheese		Cooler: 60% of cooler floor space covered with pallets of stored products. Distribution type where products are constantly shipped/received. Assumed to arrive at storage temperature. 2600 lb/pallet, 35.25" x 42" pallet dimensions. Industry: Products stored near freezing: milk, dairy products like cheese and butter, eggs and egg products. Vegetables and fruits are typically stored at temperatures well above freezing for short periods of time. There is a constant turnover of stored products.	
(Manske 1999)			Banana and tomato rooms: 10 - 15% per day; Cooler: Sensible: 0 Btu/h (assumes product arrives at storage temperature), Latent: 1201; Dock: None	
(Navy 1986)	Meat, Dairy, Perishable food, Film, Fruits, Vegetables, Medical		18 to 24 hrs pull-down time	
(Patel et al. 1993)		0	Minimal; Typically fresh and frozen products are received at storage temp.	

Product Loading for Walk-in Coolers (7 of 8)

	Product Type	Product Pull-Down Temp. Difference (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3 of refrigerated space)
(PG&E 2007)	Fresh fruit commodities such as grapes			
(Roy 2010)	Milk product		44092 lb. Sizing doesn't account for transient states or thermal mass.	
(SCE 2008)			70% of capacity; 21,840 Btu/hr (sm), 43,680 Btu/hr, 86,520 Btu/hr, 195,720 Btu/hr (lrg)	
(SCE 2009)	Cheese and other dairy products (warehouse cooler)			
(Stoecker 1998)	Beef and pork carcasses	58, 65, 40	Beef and pork carcass chilling facilities remove 58 to 65°F in one night; 4 pound birds are cooled 40°F in 4 hours or less	
(Stoeckle et al. 2002)		0	Loading dock where product is assumed to arrive at storage temperature	
(Thompson and Spinoglio 1996)	Fruits, vegetables, cut flowers		3 ft aisle with 12" to 18" from top of product to ceiling; 7 day storage; 50% turnover every day (example room)	

Product Loading for Walk-in Coolers (8 of 8)

	Product Type	Product Pull-Down Temp. Difference (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3 of refrigerated space)
(USDA 2010)	Apples/pear storage are stored exclusively in 14.37% of refrigerated warehouse facilities		Usable Percentages: 57.59 Min State Ave, 91.85 Max State Ave, 82.29 Median State Ave, 81.70 Average General, 84.20 Average Apples/Pears, 82.06 Average Overall; Apples/Pears: 0.43 Bushels/ft^3 Usable Storage, 0.36 Bushels/ft^3 Gross Storage	
(Wade 1984)		27	An apple exposed to free airstream at 32°F is cooled from 68 to 41°F in 2 hrs. 48 hrs to reduce center stack temperature of 1 ft wide packaged apples. Cooling times of 48 to 79 hrs observed in the center of a pallet load of unpackaged fruit in boxes cooling by conduction only. Small stacks of open bulk bins cool in 16 to 28 hrs. Open stacks of lidded bins with 1 to 8% ventilation cool in 66 to 99 hrs.	
(Walker and Baxter 2002)	Floral; Meat; Produce; Dairy; Deli; Raw; Fish; Meat Prep; Produce; Meat; Bakery			
(Westphalen et al. 1996)	Meats, seafood, dairy, produce, beer/juice, flowers			

Table 115. Product Sensible and Latent Thermal Properties for Walk-in Coolers

Product Sensible and Latent Thermal Properties for Walk-in Coolers (1 of 4)								
	Product Specific heat Above Freezing (Btu/lb-F)	Product Moisture Loss during storage (%)	Product Heat of Respiration at 32°F (Btu/lb/day)	Product Heat of Respiration at 41°F (Btu/lb/day)	Product Heat of Respiration at 50°F (Btu/lb/day)	Product Heat of Respiration at 59°F (Btu/lb/day)	Product Heat of Respiration at 68°F (Btu/lb/day)	Product Heat of Transpiration
Range	0.50 to 0.91	1	0.25 to 8.32	0.56 to 10.7	0.86 to 2.34	1.02 to 10.2	1.27 to 38.4	
Average	0.80	1	1.26	1.68	1.25	3.22	7.35	
Proposed	0.9							
(AHRI 2009a)	0.9							
(Adre and Hellickson 1989)			Apples, average of many cultivars: 0.252 - 0.451; Pears: 0.288-0.396	Apples, average of many cultivars: 0.558-0.793; Pears, late ripening: 0.648-1.531; Pears, early ripening: 0.811-1.712	Pears, late ripening: 0.865-2.2072; Pears, early ripening: 1.081-2.342	Apples, average of many cultivars: 1.495-3.405; Pears, late ripening: 3.062-4.684; Pears, early ripening: 3.783-5.944	Apples, average of many cultivars: 1.856-3.855; Pears, late ripening: 3.603-8.106; Pears, early ripening: 4.323-9.907	
(Becker and Fricke 1996a)			Coefficients given					Coefficients given

Product Sensible and Latent Thermal Properties for Walk-in Coolers (2 of 4)

	Product Specific heat Above Freezing (Btu/lb-F)	Product Moisture Loss during storage (%)	Product Heat of Respiration at 32°F (Btu/lb/day)	Product Heat of Respiration at 41°F (Btu/lb/day)	Product Heat of Respiration at 50°F (Btu/lb/day)	Product Heat of Respiration at 59°F (Btu/lb/day)	Product Heat of Respiration at 68°F (Btu/lb/day)	Product Heat of Transpiration
(Becker et al. 2011)	0.91 (Small Cooler); 0.90 (Large Cooler)							
(CIGR 1999)			Coefficients given	0.737 (39.2°F); 0.778 (42.8°F)	0.860 (46.4°F); 0.901 (50°F)	1.024 (57.2°F)	1.269	
(Cooper 1973)	0.9	1% weight loss						
(Devres and Bishop 1995)	3600 J/kg K (potato), 2720 (packaging)							
(DOE 2010b)	0.9							
(Heatcraft 2008)	Estimate Spec. Heat = 0.20 + (0.008 X % water); Heatcraft Table 7		Heatcraft Table 7	Heatcraft Table 7 (at 40°F)		Heatcraft Table 7 (at 60°F)		

Product Sensible and Latent Thermal Properties for Walk-in Coolers (3 of 4)

	Product Specific heat Above Freezing (Btu/lb-F)	Product Moisture Loss during storage (%)	Product Heat of Respiration at 32°F (Btu/lb/day)	Product Heat of Respiration at 41°F (Btu/lb/day)	Product Heat of Respiration at 50°F (Btu/lb/day)	Product Heat of Respiration at 59°F (Btu/lb/day)	Product Heat of Respiration at 68°F (Btu/lb/day)	Product Heat of Transpiration
(Krack 1977)	Krack Table 21		Meat and fish have no continuing life process and therefore generate no heat in storage					
(Magoo 2003)	0.502 Btu/lbm-°F (cheese)							

Product Sensible and Latent Thermal Properties for Walk-in Coolers (4 of 4)								
	Product Specific heat Above Freezing (Btu/lb-F)	Product Moisture Loss during storage (%)	Product Heat of Respiration at 32°F (Btu/lb/day)	Product Heat of Respiration at 41°F (Btu/lb/day)	Product Heat of Respiration at 50°F (Btu/lb/day)	Product Heat of Respiration at 59°F (Btu/lb/day)	Product Heat of Respiration at 68°F (Btu/lb/day)	Product Heat of Transpiration
(Stoecker 1998)		1 % (beef carcass)	0.253 to 0.451 (apples); 1.95 to 3.85 (beans); 0.793 (celery); 1.00 to 1.86 (lettuce); 0.451 to 0.703 (peaches); 5.21 to 8.32 (pears); 1.35 to 1.95 (strawberry)	0.559 to 0.792 (apples); 3.21 to 6.72 (beans); 1.21 (celery); 1.46 to 2.2 (lettuce); 0.702 to 1.01 (peaches); 8.72 to 10.7 (pears); 1.8 to 3.66 (strawberries)		1.5 to 3.4 (apples); 4.11 (celery); 4.25 to 4.5 (lettuce); 3.66 to 4.67 (peaches); 7.82 to 10.2 (strawberry)	1.86 to 3.85 (apples); 23.3 to 29.8 (beans); 7.11 (celery); 6.52 to 11.3 (peaches); 38.4 (pears); 11.3 to 21.6 (strawberries)	

Lighting and Occupancy

Lighting and occupancy loads are considered in the same section of this report, as their schedules typically coincide with each other. Through analyzing the literature, load details related to lighting and occupancy were found in the following works, (Adre and Hellickson 1989), (Arias 2005), (Becker et al. 2011), (Blackmore 2012), (Cooper 1973), (Devres and Bishop 1995), (DOE 2010b), (DOE 2010c), (Goetzler et al. 2009), (Heatcraft 2008), (Kassa et al. 2004), (Krack 1977), (Magoo 2003), (Manske 1999), (PG&E 2007), (Roy 2010), (SCE 2008), (Stoecker 1998), (Stoeckle 2000), (Sujau et al. 2006), and (Westphalen et al. 1996).

Lighting loads found in the literature ranged from 0.08 to 9.83 W/ft^2,with an average value of 1.50 W/ft^2. The AHRI Load Spreadsheet (AHRI 2009a) used a lighting power usage of 1.56 W/ft^2 for the small cooler and 1 W/ft^2 for the large cooler. Ninety eight percent of the lighting load is assumed to affect the space of the small cooler whereas only 85% is assumed to affect the space of the large cooler due to the use of fluorescent lighting. The modified model load uses a lighting load of 1 W/ft^2, but unlike AHRI, all of the lighting energy usage is a load on the space.

The quantity of people in the space that AHRI assumed seems appropriate at 1 person for the small cooler and 2 for the large cooler. The percentage of time that people occupy the space ranged in the literature from 3 to 100% of the time, with an average value of 34.7% of the time. Lights were on for a period of 4 to 100% of the time, with an average value of 54.7%. AHRI (AHRI 2009a) assumed that the lights were on 3.75 and 12.5% of the time for the small and large cooler, respectively. People are assumed to occupy the space for 5.83 and 12.5% of the time for the small and large cooler, respectively. The proposed model load will use occupancy and lighting percentages of 20 and 30%, respectively. A value ranging from 30 to 50% seems more acceptable from the collected data, but hesitation is made to use this, as unfortunately there was some ambiguity in some of the literature as to whether the occupancy percentage referred to a percentage of the day or a percentage of the working day.

Table 116. Lighting and Occupancy Load Details for Walk-in Coolers

Lighting and Occupancy Load Details for Walk-in Coolers (1 of 6)					
	Lighting Power	Lighting Schedule	Occupancy (Number of People)	Occupancy Time	Occupancy Load (W)
Range	0.08 to 9.83 W/ft^2	4 to 100% of time	69.9 to 210,000 ft^3/person	3 to 100% of time	
Average	1.50 W/ft^2	54.7% of time	32,176 ft^3/person	34.7% of time	
Proposed	1 W/ft^2	30% of time	1 person (sm), 2 people (lrg)	20% of time	
(AHRI 2009a)	1.56 W/ft^2 sm, 1 W/ft^2 lrg (Light Power Converted to Heat: 98% sm, 85% lrg)	30 minutes per hour – 6am to 7am, 2 minutes per hour – 7am to 7pm (sm); 60 minutes per hour – 6am to 7am, 10 minutes per hour – 7am to 7pm (lrg)	1 person – 6am to 7pm (sm); 2 people - 6am to 7pm (lrg)	60 minutes per hour – 6am to 7am, 2 minutes per hour – 7am to 7pm (sm); 60 minutes per hour – 6am to 7am, 10 minutes per hour – 7am to 7pm (lrg)	
(Adre and Hellickson 1989)	0.47 W/ft^2 (CA); 0.58 W/ft^2 (common storage)	4 fixtures (<20 bins of fruit), 8 fixtures (20 to 100 bins of fruit), 10 fixtures (101 of more bins of fruit. 11 operating hours		50% of 8 hour day	
(Arias 2005)	1.39 W/ft^2				200 W per person

Lighting and Occupancy Load Details for Walk-in Coolers (2 of 6)

	Lighting Power	Lighting Schedule	Occupancy (Number of People)	Occupancy Time	Occupancy Load (W)
(Becker et al. 2011)	1 W/ft^2 (Prototypical, Small and Large Cooler)	6 min/hr 8am to 6pm (Small Cooler); 60 min/hr 6am to 7am, 10 min/hr 7am to 7pm (Large Cooler)	1 person (Prototypical, Small Cooler); 2 people (Large Cooler)	4.8 min / hr 8am to 6pm Monday to Saturday, 2.4 min / hr 10am to 6pm Sunday (Prototypical); 6 min/hr 8am to 6pm (Small Cooler); 60 min / hr 6am to 7am, 10 min / hr 7am to 7pm (Large Cooler)	80.6 W/person latent, 80.6 W/person sensible (Small Freezer); 410.3 W/person (Large Freezer)
(Blackmore 2012)	1 to 1.5 W/ft^2 (normal storage), 2.5 to 3 W/ft^2 (docks/workrooms)		4 person / 100,000 ft^3		
(Cooper 1973)	0.61 W/ft^2, 0.22 W/ft^2				
(Devres and Bishop 1995)		150 seconds per entry, 56 entries per day		9.7% of the time	
(DOE 2010b)	0.31 W/ft^2 sm; 0.08 W/ft^2 med; 0.09 W/ft^2 lrg				

Lighting and Occupancy Load Details for Walk-in Coolers (3 of 6)					
	Lighting Power	Lighting Schedule	Occupancy (Number of People)	Occupancy Time	Occupancy Load (W)
(DOE 2010c)		On 75% of time (no control), 50% of time (timers or auto-shut off)			
(Goetzler et al. 2009)	1.2 W/ft^2 (walk-in merchandiser); Typical: 1 W/ft^2 Incandescent (0.9 to 1.2 W/ft^2)	Operate 66% (walk-in merchandiser) of the time			
(Heatcraft 2008)	1 to 1.5 W/ft^2. Cutting or processing rooms can be double the wattage.				1 person per 24 hours for each 25,000 ft^3; Table 89

Lighting and Occupancy Load Details for Walk-in Coolers (4 of 6)					
	Lighting Power	Lighting Schedule	Occupancy (Number of People)	Occupancy Time	Occupancy Load (W)
(Kassa et al. 2004)	22% Fluorescent, 78% Incandescent lighting; Fluorescently lit walk-ins at 15.3 foot candles (geometric mean of 6.4 to 85.5 range); Incandescently lit walk-ins at 3.43 ft-c (geometric mean of 1 to 16.7 range); Prep counters at 38.7 ft-c (geometric mean of 2.9 to 196.8 range)				
(Krack 1977)	1 W/ft^2 (beef carcass, nut storage)		4 person / 840000 ft^3 (fruit storage); 2 person / 9600 ft^3 (beef carcass)		

Lighting and Occupancy Load Details for Walk-in Coolers (5 of 6)					
	Lighting Power	Lighting Schedule	Occupancy (Number of People)	Occupancy Time	Occupancy Load (W)
(Magoo 2003)	Cooler: 0.45 W/ft^2; 70% radiative, 30% convective; Dock: 2.5 W/ft^2; 70% radiative, 30% convective		3 (cooler), 7 (dock)		Cooler: 266 W (ASHRAE 1998) per person; 60% radiative, 40% convective; Dock: 212 W (ASHRAE 1998) per person; 60% radiative, 40% convective
(Manske 1999)	Cooler: 0.68 W/ft^2; Dock: 1.42 W/ft^2	16 hrs/day (first and second shift)			Sensible: 505.6 W (first), 1011.1 (second); Latent: 1018.5 (first), 2036.9 (second)
(PG&E 2007)	0.4 to 1.2 W/ft^2; 0.6 (prototypical);	24/7 operation	80 max (cooler), 24 max (dock)	24/7 operation	
(Roy 2010)	T5-HO fluorescent with natural lighting				
(SCE 2008)	Proposed: 1.0 W/ft^2 (Heatcraft Refrigeration); High efficacy fluorescent lighting	75% of time		25% of time	

Lighting and Occupancy Load Details for Walk-in Coolers (6 of 6)					
	Lighting Power	Lighting Schedule	Occupancy (Number of People)	Occupancy Time	Occupancy Load (W)
(Stoecker 1998)	.48 W/ft^2 for storage, 1.5 W/ft^2 for processing areas				210 W/person at 50°F, 270 W/person at 32°F
(Stoeckle 2000)	2.5 W/ft^2 (loading dock)				
(Sujau et al. 2006)	2.88 W/ft^2				
(Westphalen et al. 1996)	1 W/ft^2 (incandescent box lighting). 9.83 W/ft display lighting (4 - 60inch T12 50W fluorescent lamps with 2 per ballast)	On when occupied by employees. 50% of time. Display lighting on 66% of the time.	1	50% of time	

Miscellaneous Loads

Miscellaneous loads include all other heat loads seen by walk-in coolers. The primary types of miscellaneous loads are summarized by this study and include vehicle energy usage, evaporator fan power, and door heater loads.

Authors from which information was gathered include (Adre and Hellickson 1989), (Becker et al. 2011), (Blackmore 2012), (DOE 2010b), (DOE 2010c), (Goetzler et al. 2009), (Heatcraft 2008), (Krack 1977), (Magoo 2003), (Manske 1999), (PG&E 2007), (Roy 2010), (SCE 2008), (SCE 2009), (Sujau et al. 2006), and (Westphalen et al. 1996).

Larger coolers typically have at least one battery powered forklift in use but smaller coolers do not. This is consistent with the ARHI Load Spreadsheet (AHRI 2009a). The amount of time that forklifts operate and the quantity of forklifts in the space were consolidated for this analysis by looking at power usage on a space square footage basis. Literature values ranged from 0.08 to 1.12 W-hr/ft^3-day, with an average value of 0.59 W-hr/ft3-day. AHRI used a value of 0.75 W-hr/ft^3-day for the vehicle power usage for the large cooler whereas this work proposes a value of 1.00 W-hr/ft3-day. The small cooler will continue to be viewed as not utilizing forklifts as done in the AHRI initial model load.

Most walk-ins utilize door heaters to avoid frosting up of passage ways or reach-in doors. In the examples found in the literature, 8 to 15 W/ft of heating is used around the perimeter of the doorways, with an average power usage of 11.5 W/ft. AHRI did not utilize passage door heating, but this work proposes 8 W/ft.

Table 117. Miscellaneous Load Details for Walk-in Coolers

Miscellaneous Load Details for Walk-in Coolers (1 of 3)					
	Number of Vehicles	Vehicle Energy Usage	Evaporator Fan Operation	Passage Door Heater Power	Miscellaneous Heat Load
Range		0.08 to 1.12 W-hr/ft^3-day		8 to 15 W/ft	300 to 20,000 W
Average		0.59 W-hr/ft^3-day		11.5 W/ft	5895 W
Proposed		1.00 W-hr/ft^3-day (lrg)		8 W/ft	
(AHRI 2009a)	0 sm; 1 lrg	0 sm; 37.3 kW at 60 minutes per hour – 6am to 7am (lrg)			
(Adre and Hellickson 1989)	2	26 kW per forklift @ 60% load factor; 1/2 of the 8 hour day (1/2 of time spent outside the refrigerated space)			
(Becker et al. 2011)	1 (Large Cooler)	0.746 W-hr/ft^3-day (Large Cooler)	always on		
(Blackmore 2012)		4.48 kW/100,000 ft^3; 2.98 to 3.73 kW per forklift			
(DOE 2010b)				8 W/ft; 24 hr/day	
(DOE 2010c)				Always on typically	

Miscellaneous Load Details for Walk-in Coolers (2 of 3)

	Number of Vehicles	Vehicle Energy Usage	Evaporator Fan Operation	Passage Door Heater Power	Miscellaneous Heat Load
(Goetzler et al. 2009)				300 W Always on; Passage door heaters typically used.	
(Heatcraft 2008)	Generally only battery operated lift trucks used; 1 motor horsepower for each 16,000 ft^3 of storage.	2.3 to 4.4 kW over the period of operation; Heatcraft Table 14 - Heat gain due to operation of battery operated lift truck			Heatcraft Table 11 - Heat Equivalency of Motors; Banana ripening room: 6 to 20 kW per half carload to bring load up to temperature before ripening process is initiated
(Krack 1977)	1 hp / 26250 ft3 storage (fruit storage)	Forklifts may be estimated at 3.0 to 3.7 kW; 40 min/hr for 8 hours operating time			
(Magoo 2003)		Cooler: 54.3 kW total; 70% radiative, 30% convective; Dock: 9.6 kW total; 70% radiative, 30% convective			
(Manske 1999)		Cooler: 17.3 kW (first); 34.6 (second); Dock: 8.1 kW (first); 16.3 (second)	cycle with load		

Miscellaneous Load Details for Walk-in Coolers (3 of 3)

	Number of Vehicles	Vehicle Energy Usage	Evaporator Fan Operation	Passage Door Heater Power	Miscellaneous Heat Load
(PG&E 2007)		0.7 W/ft^2 (forklifts and miscellaneous equipment)			
(Roy 2010)		92% (80% typical) battery charger efficiency			
(SCE 2008)					Anti-sweat heaters
(SCE 2009)	Yes (warehouse)				
(Sujau et al. 2006)					Varies: 3100 W Sensible, 230 Latent to 5100 Sensible, 340 Latent
(Westphalen et al. 1996)			Always on		Anti-sweat heater typically not used; 300 W total for merchandising doors, 100% on (prototypical)

Defrost

The AHRI 1250/1251 Standard (AHRI 2009b; AHRI 2009c) assumes no defrost is used on walk-in coolers. This is not always the case, however, with fifteen authors giving examples of coolers utilizing defrosting systems. These works include (Adre and Hellickson 1989), (Becker et al. 2011), (Cooper 1973), (Devres and Bishop 1995), (Goetzler et al. 2009), (Heatcraft 2008), (Henderson and Khattar 1999), (Krack 1977), (Manske 1999),(Nelson 2011), (Stoecker 1998), (Sujau et al. 2006), (Walker et al. 1990), (Walker 1992), (Walker 2001), and (Westphalen et al. 1996).

Available literature stated that both off-cycle and electric defrost are typical for coolers operating with an evaporator temperature below freezing. From analyzing the data in Table 118, electric defrost appears to be most often used for high humidity applications or coolers that operate very close to freezing, such as a meat cooler. Defrost power ranged from 19.5 to 29.6 W/ft^2, with an average value of 25.7 W/ft^2. Adding refrigeration systems that do not use defrost or use an off-cycle defrost to this analysis results in an average defrost power of 11.04 W/ft^2. The analysis of the data on defrost duration and frequency, much like the analysis of the defrost power usage, only includes data about refrigeration systems that use defrost. For the calculation of the annual walk-in energy factor (AWEF), defrost will occur twice a day for 1 hour periods.

Table 118. Defrost Details for Walk-in Coolers

Defrost Details for Walk-in Coolers (1 of 6)					
	Defrost Type	Defrost Power (W/ft^2)	Defrost Heat Space Load (Btu/h)	Defrost Efficiency	Defrost Schedule
Range		19.5 to 29.6			0.8 to 8 times per day; 0.5 to 4 hr/day
Average		25.7			3.0 times per day; 1.3 hr/day
Proposed					2 times per day; 2 hr/day
(AHRI 2009a)					
(Adre and Hellickson 1989)					1 to 8 per day; 10 minutes long
(Becker et al. 2011)					1.75 cycles / day at 41.6 min / cycle (Tedeschi Cooler), 1.25 cycles / day at 24.3 min / cycle (Chili's Beer Cooler), 4.25 cycles / day at 34.6 min / cycle (Chili's Food Cooler)
(Cooper 1973)		No defrost needed for rooms above 35.6°F			

Defrost Details for Walk-in Coolers (2 of 6)

	Defrost Type	Defrost Power (W/ft^2)	Defrost Heat Space Load (Btu/h)	Defrost Efficiency	Defrost Schedule
(Devres and Bishop 1995)		Assume 25% of the condensed water turns into frost on the coil		50%	
(Goetzler et al. 2009)		25 W/ft^2			
(Heatcraft 2008)	Off-cycle				
(Henderson and Khattar 1999)	Off-cycle (typical), Hot gas (meat)				
(Krack 1977)					4 times per day; 15 to 20 min / cycle (beef carcass)

Defrost Details for Walk-in Coolers (3 of 6)					
	Defrost Type	Defrost Power (W/ft^2)	Defrost Heat Space Load (Btu/h)	Defrost Efficiency	Defrost Schedule
(Manske 1999)	Hot gas + electric (cooler), Air (loading dock)		Cooler: 180 MBH June to Sept, 144 MBH otherwise (hot gas); Cooler: 60 MBH June to Sept, 48 MBH otherwise (electric, cooler); Dock: 120 MBH June to Sept, 60 MBH otherwise (air)	50% (ASHRAE 1994)	Hot gas defrosted two to three times a day on a time scheduled basis. Evaporator defrost cycles are staggered.

Defrost Details for Walk-in Coolers (4 of 6)

	Defrost Type	Defrost Power (W/ft^2)	Defrost Heat Space Load (Btu/h)	Defrost Efficiency	Defrost Schedule
(Nelson 2011)	Types include: water, electric, hot gas with hot gas being the preferred method		15 - 20% to melt ice, 60% lost to room via convection/radiation, 20% required to heat and cool the metal of the evaporator, and 5% lost due to hot gas bypassing the defrost regulator at the end of the defrost (Cole 1989). 32F room: 46% room loss, 32% melt frost, 18% heat and cool metal, 4% warm melt	max theoretical defrost efficiency: 60 to 70% (Cole 1989); Examples: 32% (32ºF room typical)	Typical: 30 minutes, Optimized: 8 to 10 minutes
(Stoecker 1998)	Air or electric (most common)				
(Sujau et al. 2006)		4.6 kW			6, 8, 12, 18, 24, and 30 hr intervals

Defrost Details for Walk-in Coolers (5 of 6)

	Defrost Type	Defrost Power (W/ft^2)	Defrost Heat Space Load (Btu/h)	Defrost Efficiency	Defrost Schedule
(Walker et al. 1990)	Off-cycle, electric				Conventional Medium Temp Off Cycle: 55 min at 00:00, 12:00 (#3); 60 min at 00:00, 12:00 (#24); 40 min at 00:00, 06:00, 12:00, 18:00 (#25); 40 min at 00:30, 06:30, 12:30, 18:30 (#9); 40 min at 04:00, 13:30, 20:30 (#10); 40 min at 04:30, 12:30, 19:30 (#11); 40 min at 01:30, 07:30, 13:30, 19:30 (#12); 40 min at 02:30, 08:30, 14:30, 20:30 (#13); 60 min at 01:00, 07:00, 13:00, 19:00 (#15); 36 min at 02:00, 10:00, 17:00 (#20A and B). Meat satellite Off Cycle: 40 min at 02:00, 08:00, 14:00, 20:00 (#2, #27A and B). High Temp Off Cycle: 54 min at 05:00, 17:00 (#17); 50 min at 01:00 (#26); 50 min at 00:00 (#23); two systems have no defrost. Multiplex Medium Temp Off Cycle: 52 min at 05:30, 11:30, 17:30, 23:30 (#B3); 30 min at 02:30, 14:30 (#B24); 52 min at 03:30, 09:30, 15:30, 21:30 (#B25); 52 min at 04:30, 10:30, 16:30, 22:30 (#B9); 50 min at 01:30, 09:30, 18:30 (#B10); 52 min at 02:30, 10:30, 17:30 (#B11); 52 min at 05:30, 11:30, 17:30, 23:30 (#B12); 48 min at 00:30, 06:30, 12:30 (#B13); 30 min at 10:30, 22:30 (#B15); 36 min at 07:30, 14:30, 23:30 (#B20A and B). High Temp Off Cycle: 48 min at 01:00, 07:00, 13:00, 19:00 (#B2, 27A and B); 54 min at 03:30 (#B17); 50 min at 00:00 (#B26); 50 min at 23:00 (#B23); two systems have no defrost

Defrost Details for Walk-in Coolers (6 of 6)

	Defrost Type	Defrost Power (W/ft^2)	Defrost Heat Space Load (Btu/h)	Defrost Efficiency	Defrost Schedule
(Walker 1992)	Off-cycle commonly; time terminated				
(Walker 2001)	Off-cycle (multiplex and distributed), warm glycol (secondary loop)				
(Westphalen et al. 1996)	Electric (meat walk-in)	19.5 W/ft^2 (meat); 19.5 to 29 W/ft^2 (typical)			4.2% duty cycle (20 min every 8 hrs)

Loading Summary

Through a literature analysis and comparison to the AHRI Load Spreadsheet (AHRI 2009a) a new model load profile was developed for walk-in coolers. Box conditions, construction, and size were not altered from the AHRI Load Spreadsheet, but convection coefficients were added to the conduction analysis to improve its accuracy. Review of the literature showed that the ground temperature used by AHRI was understated at 50°F. This was increased to 55°F to better match typical ground temperatures.

The infiltration calculation utilized by the AHRI Load Spreadsheet is sound, and the door sizes are close to the correct size. The percentage of time that doors were open for the walk-in coolers was increased from 1.18% and 2.78% for the small and large coolers, respectively, to 5.4% for both coolers.

The product load used by AHRI used a pull-down temperature of 10°F and a pull-down period of 8 hours, with product loading of 12.11 and 1.60 lbs of product per ft^3 for the small and large coolers, respectively. The literature revealed that a pull-down period of 24 hours was more appropriate than the 8 hr period utilized by AHRI and that the small cooler load was overstated. Cooler loads were adjusted to 4.0 and 1.8 lbs of product per ft^3 of refrigerated space for the small and large coolers, respectively. The product specific heat was maintained for the product load calculation, but a respiration heat load was also added to the space.

Lighting power load per square foot was only adjusted slightly from 1.56 W/ft^2 and 1W/ft^2 for the small and large cooler, respectively, to 1 W/ft^2 for both spaces. AHRI assumed 98% of the lighting load affects the space for the small cooler whereas only 85% is assumed to affect the space for the large cooler due to the use of fluorescent lighting. For the modified model all of the lighting load is a heat load added to the refrigerated space. The number of people in the space at any given time was maintained from the AHRI Load Spreadsheet, but the schedule of lighting and occupancy were both increased. The lighting percentage was increased from 3.75 and 12.5% for the small and large cooler, respectively, to 30% of the time. The occupancy schedule used by AHRI was 5.83 and 12.5% for the small and large cooler. This was also modified to a percentage of 20%.

Defrost on medium temperature refrigerated spaces is common. Defrost will be added to the AWEF calculation, occurring twice a day for 1 hour periods. Miscellaneous loads were adjusted by increasing the vehicle heat addition to the space from 0.75 to 1.00 W-hr/ft^3-day for the large cooler. In addition, passage door heaters were added at 8 W/ft of door perimeter.

Walk-In Freezers

The literature that was located presented a wealth of information on typical engineering and operation of walk-in freezers. The available sources were analyzed, compared to each other, and then compared to the model load defined by AHRI in the AHRI Load Spreadsheet (AHRI 2009a). Modifications to the model load were developed based on this analysis.

Box Interior Conditions

Box interior conditions refer to the temperature and humidity levels maintained within the walk-in box and the interior dimensions of the space. Both commercial and industrial refrigeration application sources were investigated by this study. Commercial refrigeration, for the purpose of this study, refers to installations at grocery stores, restaurants, and gas stations. These installations are smaller in size and often interface with customers who access the refrigerated space through glass doors. Industrial refrigeration refers to large refrigeration systems used for production, processing, and storage of frozen products. Information on commercial refrigeration is directly applicable to this study whereas industrial refrigeration information is not always applicable. Industrial warehouse dimensions were excluded from the analysis of the typical interior dimensions of a refrigerated space to ensure applicability of the results to this study.

Information related to interior conditions for a walk-in freezer was presented by fifty-seven authors, including (Altwies 1998), (Altwies and Reindl 1999), (Anonymous 2004), (Aparicio-Cuesta and Garcia-Moreno 1988), (Ashby et al. 1979), (Bauernfeind and Siemers 1945), (Becker et al. 2011), (Blackmore 2012), (Boggs et al. 1960), (Chen et al. 1999), (Christensen and Bertilsen 2004), (Cleland 1983), (Cleland and O'Hagan 2003), (Cleland et al. 2004), (Diehl and Berry 1933), (DOE 2010a), (DOE 2010b), (DOE 2010c), (Foster et al. 2003), (Goetzler et al. 2009), (Gortner et al. 1948), (Gosney and Olama 1975), (Hayes and Stoecker 1969), (Heatcraft 2008), (Henderson and Khattar 1999), (Hendrix et al. 1989), (Huan 2008), (Hustrulid and Winters 1943), (IIR 1986), (Krack 1977), (Kun et al. 2007), (Lutz et al. 1934), (Mago and Sherif 2005), (Magoo 2003), (Manske 1999), (McHugh 2010), (Morris 2012), (Nagaraju et al. 2001), (Navy 1986), (Nelson 2011), (O'Hagan et al. 1993), (Patel et al. 1993), (PG&E 2007), (Pham and Oliver 1983), (Plagge 1938), (Sand et al. 1997), (SCE 2008), (SCE 2009), (Sezgen and Koomey 1995), (Stoecker 1998), (Stoeckle 2000), (Tressler and Evers 1947), (USDA 2010), (Westphalen et al. 1996), (Wiegand 1931), and (Woodroof and Shelor 1947). Results of the literature search for box interior conditions are found in Table 119.

Box interior dimensions are discussed by thirty-five works, including (Anonymous 2004), (Arias 2005), (Becker et al. 2011), (Blackmore 2012), (Christensen and Bertilsen 2004) , (DOE 2010a), (DOE 2010b), (DOE 2010c), (Edwards 2010), (Elsayed 1998), (Foster et al. 2003), (Goetzler et al. 2009), (Heatcraft 2008), (Henderson and Khattar 1999), (Huan 2008), (Jones et al. 1983), (Knudsen and Pachai 2004), (Krack 1977), (Longdill and Wyborn 1978), (McHugh 2010), (Morris 2012), (Nagaraju et al. 2001), (Navy 1986), (O'Hagan et al. 1993), (Patel et al. 1993), (PG&E 2004), (Pham and Oliver 1983), (SCE 2008), (SCE 2009), (Sezgen and Koomey 1995), (Sherif et al. 2002), (Walker et al. 1990), (Walker 1992), (Walker 2001), (Walker and Baxter 2002), and (Westphalen et al. 1996). The information gathered from these reports is summarized in Table 120.

The box interior temperature varies for walk-in freezers based on the application and the product being stored, ranging from -40 to 32.4°F, according to the literature. The average storage temperature is -0.3°F. A majority of the storage temperatures cited as most typical were either 0 or -10°F, which prompted an adjustment to the AHRI Load Spreadsheet (AHRI 2009a) storage temperature from -10 to -5°F. Box interior relative humidities range from 50 to 95%, with an average value of 77.9%. The value used by AHRI is on the low end of typical relative humidities found in the literature. The modified model load will use a relative humidity of 80%.

Walk-in freezers are constructed in various sizes ranging from 30 to 3000 ft^2 in the literature. Box dimensions (width x length x height) varied from 5 x 6 x 6 ft tall to 50 x 50 x 69.6 ft tall, with noted box volumes ranging from 7 to 50,000 ft^3. AHRI used freezer dimensions of 8 x 8 x 8 ft tall and 50 x 50 x 20 ft tall for the small and large freezers analyzed, respectively. Through the literature review, it appears that these box dimensions are appropriate to analyze the range of box sizes.

Table 119. Box Interior Conditions for Walk-in Freezers

Box Interior Conditions for Walk-in Freezers (1 of 4)		
	Box Interior Temperature (°F)	Box Interior RH (typical) (%)
Range	-40 to 32.4	50 to 95
Average	-0.3	77.9
Proposed	-5	80
(AHRI 2009a)	-10	50
(Altwies 1998)	0°F is typical to balance food preservation and operating costs; -5 (test facility)	
(Altwies and Reindl 1999)	-5	
(Anonymous 2004)	-10	
(Aparicio-Cuesta and Garcia-Moreno 1988)	-7.6 (cauliflower); -0.4 (display freezer)	
(Ashby et al. 1979)	-11 to 0, -9.4 (typical)	
(Bauernfeind and Siemers 1945)	-10 to 0 (peaches)	
(Becker et al. 2011)	10 (Chili's Freezer); 0 (FSTC); -10 (Small Freezer, Large Freezer)	
(Blackmore 2012)	-10 (beef freezer)	
(Boggs et al. 1960)	-10 to 20	
(Chen et al. 1999)	0; 3.2 to -4	
(Christensen and Bertilsen 2004)	-14.8 (walk-in with glass doors); -18.4 (ice-cream with glass doors)	

Box Interior Conditions for Walk-in Freezers (2 of 4)		
	Box Interior Temperature (°F)	Box Interior RH (typical) (%)
(Cleland 1983)	-4 to 5 (cold stores); -22 to -4 (lamb carcass freezers); -40 to -13 (beef carton freezers)	
(Cleland and O'Hagan 2003)	28.8 to 32	68 to 93
(Cleland et al. 2004)	-4 (A5 and 6); 3 (A2a and A3); 5 (A7); 30 (A1, A2b)	
(Diehl and Berry 1933)	15 is practical upper limit for frozen storage	
(DOE 2010a)	32 or lower	
(DOE 2010b)	-10	60
(DOE 2010c)	32 or lower, -10	
(Edwards 2010)	20	
(Foster et al. 2003)	-4	
(Goetzler et al. 2009)	-10 (baseline); -20 to 10 (typical)	
(Gortner et al. 1948)	0 to -20; -5 to -15 (pork)	
(Gosney and Olama 1975)	-22	85
(Hayes and Stoecker 1969)	14	
(Heatcraft 2008)	Heatcraft Table 7; -1.6 average	Heatcraft Table 7; 65 average
(Henderson and Khattar 1999)	-8 (meat), -2 (main storage)	
(Hendrix et al. 1989)	-15 (#1); 0 (#2); -10 (#3); -15 (#4)	

Box Interior Conditions for Walk-in Freezers (3 of 4)

	Box Interior Temperature (°F)	Box Interior RH (typical) (%)
(Huan 2008)	-13 (bulk and production stores); 0 to -4 (distribution stores);	90
(Hustrulid and Winters 1943)	-20 to 0, 5 typical	
(IIR 1986)	10 to 14 (80 to 90% Vitamin C loss), -4 to -14 (10% Vitamin C loss), -22 (practically no Vitamin C loss)	
(Krack 1977)	-15 average	
(Kun et al. 2007)	5	
(Lutz et al. 1934)	10 to 15 (frozen dewberries)	
(Mago and Sherif 2005)	6.8 to 10.4	75
(Magoo 2003)	High temperature freezer: 28.4 to 26.6; low temperature storage: -9.4 to -20.2	
(Manske 1999)	0	80
(McHugh 2010)	20	
(Morris 2012)	-10, 0	90 (minimum weight loss), 80 to 85 (general storage), 65 to 80 (beer, wine, drugs), 50 to 65 (prep/cutting rooms, candy, film)
(Nagaraju et al. 2001)	5	
(Navy 1986)	0 to -10 (film); 0 (short term, packaged); -10 to -20 (long term; meat, fish, poultry, ice cream, vegetables); 25 (ice), -4 (medical); -40 (blast freezer)	85 (film); 90 (short term, packaged); 90 to 95 (long term; meat, fish, poultry, ice cream, vegetables); 85 (medical)

Box Interior Conditions for Walk-in Freezers (4 of 4)

	Box Interior Temperature (°F)	Box Interior RH (typical) (%)
(Nelson 2011)	0, -10, -30	
(O'Hagan et al. 1993)	29.12 to 32.36	68 to 93
(Patel et al. 1993)	-5 to 0	
(PG&E 2007)	-10	
(Pham and Oliver 1983)	2.3 (#1 and 2); -0.4 (#3); 1.4 (#4); -4 (#5); 23.9 (#6)	
(Plagge 1938)	0 to 10	
(Sand et al. 1997)	0 to -25 (Freezers and ice cream cases)	
(SCE 2008)	-10, 5	
(SCE 2009)	8.9	
(Sezgen and Koomey 1995)	0 (typical), -12 (ice cream)	
(Stoecker 1998)	-5 to -10 (typical); -22 (ice cream and fish); 5 to 14 (bakery goods)	
(Stoeckle 2000)	0 (distribution warehouse)	
(Tressler and Evers 1947)	-5 (peaches)	
(USDA 2010)	Less than 0	
(Westphalen et al. 1996)	-10; -9 (prototypical); -25 to -15 (frozen foods), -35 to -25 (ice cream and frozen bakery)	
(Wiegand 1931)	-10 to 0 (commercial plants), 0 (locker plants)	
(Woodroof and Shelor 1947)	-20 to 10; 0 typical	

Table 120. Box Interior Dimensions for Walk-in Freezers

Box Interior Dimensions for Walk-in Freezers (1 of 4)					
	Box Width (ft)	Box Length (ft)	Box Height (ft)	Floor Area (ft^2)	Box Volume (ft^3)
Range	5 to 50	6 to 50	6 to 69.55	30 to 3000	7 to 50,000
Average	13.3	18.2	11.4	604.5	4677.1
Proposed	8, 50	8, 50	8, 20	64, 2500	512, 50000
(AHRI 2009a)	8 sm, 50 lrg	8 sm, 50 lrg	8 sm, 20 lrg	64 sm, 2500 lrg	512 sm, 50000 lrg
(Adre and Hellickson 1989)	43.5 (CA); 57.1 (common storage)	63 (CA); 97.11 (common storage)	32.81	2738 (CA); 5544 (common storage)	89841 (CA); 181,885 (common storage)
(Anonymous 2004)	5 to 9	6 to 19	7.5, 8.5, or 9.5 inside	30 to 171	
(Arias 2005)					7, 10, 20, 40, 100, 500, 1000, 3000
(Becker et al. 2011)	7.25 (Chili's Freezer); 89 inch (FSTC); 7.42 (Small Freezer); 50 (Large Freezer)	10.25 (Chili's Freezer); 113 inch (FSTC); 9.42 (Small Freezer); 50 (Large Freezer)	8 to 30; 8.5 (Chili's Freezer); 95 inch (FSTC); 7.92 (Small Freezer); 20 (Large Freezer)	50 to 3000; 69.84 (FSTC), 74.3 (Chili's Food Freezer); 69.9 (Small Freezer); 2500 (Large Freezer)	552.9 (FSTC), 631.6 (Chili's Food Freezer); 553.6 (Small Freezer); 50,000 (Large Freezer)
(Blackmore 2012)	24	20	12	480	5,760
(Christensen and Bertilsen 2004)	10.83 (walk-in); 6.89 (ice cream)	11.81 (walk-in); 10.83 (ice cream)	8.04	127.9 (walk-in); 74.6 (ice cream)	1028.3 (walk-in); 599.9 (ice cream)

Box Interior Dimensions for Walk-in Freezers (2 of 4)					
	Box Width (ft)	Box Length (ft)	Box Height (ft)	Floor Area (ft^2)	Box Volume (ft^3)
(DOE 2010a)				< 3000	
(DOE 2010b)	6.0 sm, 9.0 med, 20.0 lrg	8.0 sm, 20.0 med, 25.0 lrg	7.6 sm, 9.5 med, 12.0 lrg	48 sm, 180 med, 500 lrg	364.8 sm, 1710.0 med, 6000.0 lrg
(DOE 2010c)				< 3000	
(Elsayed 1998)			6.56 to 13.12		
(Foster et al. 2003)	15.75	19.02	12.47	299.7	3736.0
(Goetzler et al. 2009)				80, 1000 (Westphalen et al. 1996)	
(Heatcraft 2008)	12 (ice cream hardening), 20 (beef)	14 (ice cream hardening), 24 (beef)	8 (ice cream hardening), 12 (beef)	168 (ice cream hardening), 480 (beef)	1344 (ice cream hardening), 5760 (beef)
(Henderson and Khattar 1999)				Store A: 258 (meat and main); Store B: 480	
(Huan 2008)					35314.7, 3531.5, 353.1
(Jones et al. 1983)	6	12	6	72	432
(Knudsen and Pachai 2004)					343, 189, 421, 309

Box Interior Dimensions for Walk-in Freezers (3 of 4)

	Box Width (ft)	Box Length (ft)	Box Height (ft)	Floor Area (ft^2)	Box Volume (ft^3)
(Krack 1977)	16 (Pull-down), 30 (Ice cream)	32 (Pull-down), 30 (Ice Cream)	10 (Pull-down, Ice cream)	512 (Pull-down), 900 (Ice Cream)	5120 (Pull-down), 9000 (Ice Cream); 250 to 30,000 (Krack Table 41)
(Longdill and Wyborn 1978)					6,251
(McHugh 2010)				Walk-in freezer has chilled area <3000 ft^2	
(Morris 2012)	20, 12 and 12	40, 20 and 12	8	800, 240, and 144	6400, 1920 and 1152
(Nagaraju et al. 2001)	7.1	15.1	7.1	106.4	750.9
(Navy 1986)			12 (ice storage)		
(O'Hagan et al. 1993)	8.86	13.78	7.87	122.1	960.9
(Patel et al. 1993)				2,038	
(PG&E 2004)				50 to 2,000	
(Pham and Oliver 1983)			12.5 to 69.55 (D)		
(SCE 2008)				250 sm, 500, 1000, 2500 lrg	
(SCE 2009)				204	

Box Interior Dimensions for Walk-in Freezers (4 of 4)					
	Box Width (ft)	Box Length (ft)	Box Height (ft)	Floor Area (ft^2)	Box Volume (ft^3)
(Sezgen and Koomey 1995)	30, 10	50, 20	20, 8	1500, 200	30000, 1600
(Sherif et al. 2002)	12.92	10.33	7.25	133.5	967.6
(Walker 1992; Walker et al. 1990)				500 (freezer); 81 (bakery)	
(Walker 2001)	15 (grocery); 10 (bakery); 8 (deli)	42 (grocery); 12 (bakery); 10 (deli)		630 (grocery); 120 (bakery); 80 (deli)	
(Walker and Baxter 2002)				925 (freezer); 100 (meat); 144 (bakery); 1,013 (freezer #2); 506 (freezer #3); 144 (fish); 81 (deli); 180 (ice cream)	
(Westphalen et al. 1996)	8	10	8 (typical), 7.58 (prototypical)	1000 (combined); 80 to 250 (typical); 80 (prototypical)	640 to 2000 (typical)

Box Conduction

The walk-in freezer conduction load is calculated using numerous inputs, including the box interior conditions (Table 119), the surface area of the box (Table 120), the exterior conditions surrounding the box (Table 104 and Table 105), the ground temperature (Table 121), the box construction (Table 122), and the convective coefficient (Table 123).

Information about the floor heat load was presented by, (Adre and Hellickson 1989), (Altwies and Reindl 1999), (Becker et al. 2011), (Cooper 1973), (Devres and Bishop 1995), (DOE 2010a), (DOE 2010b), (DOE 2010c), (Heatcraft 2008), (Krack 1977), (Magoo 2003), (Manske 1999), (Navy 1986), (PG&E 2007), (SCE 2008), (Somrani et al. 2008), (Stoecker 1998), (Stoeckle 2000), and (Stoeckle et al. 2002). Floor heat load was presented in two forms: floor heat load (Btu/hr-ft^2) and ground temperature.

Box construction varies greatly from one walk-in freezer to another. This is highlighted by the information obtained from (Altwies and Reindl 1999), (Anonymous 2004), (Ashby et al. 1979), (ASHRAE 1998), (Becker et al. 2011), (Blackmore 2012), (Chen et al. 1999), (DOE 2010a), (DOE 2010b), (DOE 2010c), (Heatcraft 2008), (Hendrix et al. 1989), (Jones et al. 1983), (Krack 1977), (Magoo 2003), (Manske 1999), (Nagaraju et al. 2001), (Navy 1986), (O'Hagan et al. 1993), (Patel et al. 1993), (PG&E 2007), (SCE 2008), (Sherif et al. 2002), (Stoecker 1998), (Stoeckle 2000), (Stoeckle et al. 2002), and (Westphalen et al. 1996), with R-values ranging up to +202%/-93% from the average value. Convective film coefficients for calculation of the box conduction load for walk-in freezers were noted by (DOE 2010b), (Krack 1977), (SCE 2008), (Sherif et al. 2002), (Stoecker 1998), and (Stoeckle 2000).

Ground temperatures vary based on the location and the method of floor heating utilized by the walk-in freezer. The industrial facilities discussed in the literature utilized glycol loops as a rule, but walk-ins may utilize any of a number of floor heating options, including heated glycol, ducted hot air, or electric heat. Literature ground temperatures ranged from 40 to 79.7°F, with an average value of 56.2°F. Due to a slight underestimation of the floor temperature, the AHRI Load Spreadsheet (AHRI 2009a) ground temperature was increased from 50 to 55°F. Floor heat loads cited by the literature ranged from 1.86 to 5.56 Btu/hr-ft^2, with an average value of 3.08 Btu/hr-ft^2. AHRI (AHRI 2009a) used a value of 1.88 Btu/hr-ft^2 which closely matches the proposed 2.34 Btu/hr-ft^2 (calculated using the conduction equation).

As could be expected, the insulating value of walk-in freezers varies greatly depending on the application and the owner's willingness to invest additional capital to reduce operating cost. Ceiling insulation ranged from 16.7 to 50 h-ft^2-°F/Btu in the literature, with an average value of 33.0 h-ft^2-°F/Btu. Walls were of similar construction, ranging from 3.3 to 83.3 h-ft^2-°F/Btu (28.0 h-ft^2-°F/Btu average). The R-value of the floor is typically lower than the remainder of the box, ranging from no insulation to 35.4 h-ft^2-°F/Btu. The average floor R-value was 25.1 h-ft^2-°F/Btu. The AHRI Load Spreadsheet assumption on ceiling insulation value was in close agreement with this and therefore was not changed. Walls and floor insulation were altered to 28 and 25 h-ft^2-°F/Btu, respectively, to more closely mirror the average value for walk-in freezer construction.

AHRI did not utilize a convective coefficient in the calculation of the conduction load. Numerous authors did utilize convective coefficients, ranging from 0.17 to 1.33 h-ft^2-

°F/Btu. This study will utilize values of 0.5 h-ft^2-°F/Btu for the internal and external convective film coefficient and use a value of 0.6 h-ft^2-°F/Btu for the floor coefficient.

Table 121. Floor Conditions for Walk-in Freezers

Floor Conditions for Walk-in Freezers (1 of 3)		
	Floor Heat Load (Btu/hr-ft^2)	Ground Temperature (°F)
Range	1.86 to 5.56	40 to 79.7
Average	3.08	56.2
Proposed	2.34	55
(AHRI 2009a)	1.88	50
(Adre and Hellickson 1989)		Average soil temperature at 51 cm
(Altwies and Reindl 1999)	67.5°F Floor temperature (average of supply and return glycol temperatures)	67.5 below slab (average of supply and return glycol temperatures)
(Becker et al. 2011)	Heated sub-floor (Small Freezer)	50 minimum (Small Freezer)
(Cooper 1973)		79.7
(Devres and Bishop 1995)		41
(DOE 2010a)		55 or 60
(DOE 2010b)	R-0, R-22.42, R-28.03: 3.15, 2.59, 2.2 (36 ft2); 3.11, 2.56, 2.18 (48 ft2); 3.04, 2.51, 2.14 (71.4 ft2); 2.88, 2.4, 2.06 (180 ft2); 2.54, 2.16, 1.88 (500 ft2); 2.51, 2.14, 1.86 (1200 ft2); (DOE finite element analysis)	DOE Test Procedure: 65
(DOE 2010c)		60
(Heatcraft 2008)		40 (min), 55 (typical) slab temp

Floor Conditions for Walk-in Freezers (2 of 3)		
	Floor Heat Load (Btu/hr-ft^2)	Ground Temperature (°F)
(Krack 1977)	1 Btu/ft2-F-24 hrs (conventional insulation)	55
(Magoo 2003)	Most popular method is a pipe grid system in the base concrete slab directly under the insulation. Hot air or glycol is circulated to keep the slab above freezing.	50 under slab
(Manske 1999)	2.5	
(Navy 1986)	Facilities > 5000 ft^2 or < 32°F: heated by glycol; Other facilities may use ventilation pipes as alternative heating method; Freezing is a concern if storage temperature ≤ 15°F and room width is over 10 to 15 ft.	
(PG&E 2007)	Electric under slab heating is not very common in large warehouses, but may be used in small frozen storage spaces	
(SCE 2008)		50
(Somrani et al. 2008)		57.2°F soil near temperature
(Stoecker 1998)	Underfloor heating fluid supply temp no higher than 50 to 60°F. Heating is required to maintain a soil temperature at about 50°F.	Underfloor heating fluid supply temp no higher than 50 to 60°F. Heating is required to maintain a soil temperature at about 50°F (freezer).
(Stoeckle 2000)	3.6 (Jekel and Reindl 2000)	67.5 below slab

Floor Conditions for Walk-in Freezers (3 of 3)

	Floor Heat Load (Btu/hr-ft^2)	Ground Temperature (°F)
(Stoeckle et al. 2002)	3.6 (Jekel and Reindl 2000)	

Table 122. Box Construction for Walk-in Freezers

Box Construction for Walk-in Freezers (1 of 7)			
	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
Range	16.7 to 50	3.3 to 83.3	On slab to 35.4
Average	33.0	28.0	25.1
Proposed	32	28	25
(AHRI 2009a)	32	32	32
(Altwies and Reindl 1999)	27.78 (A and B), 28.57 C and D)	27.78 (A and B) wood and insulation, lightweight; 21.74 (C and D) concrete and insulation, heavyweight	
(Anonymous 2004)	4" Urethane insulation sandwiched between aluminum, SS, high-density urethane or fiberglass (typical)	4" Urethane insulation sandwiched between aluminum, SS, high-density urethane or fiberglass (typical)	4" Urethane insulation sandwiched between aluminum, SS, high-density urethane or fiberglass (typical)
(Ashby et al. 1979)	31.55 R-value; Air spaces of 4.1 cm	24.69 R-value; Insulation in walls consists of 1.8 cm air spaces formed by 24 gage aluminum foil fastened to horizontal wood strips.	23.66 R-value; Insulation consists of foam plastic
(ASHRAE 1998)	44.86 to 49.97	35.21 to 39.75	27.26 to 31.80

Box Construction for Walk-in Freezers (2 of 7)			
	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
(Becker et al. 2011)	4" Extruded polystyrene sandwich panel, R-17.24 (Small Freezer); 32 R-value (Large Freezer)	3.5, 4 or 5.5 inch thick insulated panels (typical); R-36 (Title 20); 3.5 in urethane foam insulated panels (FSTC); 4" Extruded polystyrene sandwich panel, R-17.24 (Small Freezer); 32 R-value (Large Freezer)	12 in soil, 6 in concrete, 4" Extruded polystyrene sandwich panel (Small Freezer, R-18.9); 12 in soil, 6 in concrete, R-value 32 (Large Freezer, R-34.5)
(Blackmore 2012)		Spray K = 0.16 Btu / hr-in-ºF, Foam = 0.12, Styrofoam = 0.26; 4 inch of foamed in-place urethane (beef freezer)	
(Chen et al. 1999)		200 mm polystyrene sandwich panel attached to a steel portal frame	
(DOE 2010a)		No foam insulation degradation seen in practice	
(DOE 2010b)	4" XPS or PU, R-24 (independent testing lab)	4" XPS or PU, R-24 (independent testing lab)	4" XPS or PU, (R-22.42)
(DOE 2010c)		4" Foam :16 to 32 h-ft^2-ºF/Btu; 4" Vacuum insulated panel: 120 to 200 h-ft^2-ºF/Btu (excluded from average for Table 122). Insulating value degrades over time due to diffusion of gases and moisture absorption.	28

Box Construction for Walk-in Freezers (3 of 7)			
	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
(Heatcraft 2008)	Walk-in: 4 in Urethane (R=25, k=0.16)	1 to 10 inches of cork/mineral wool, glass fiber/poly styrene, sprayed urethane, foamed in place urethane; Heatcraft Table 13 - General Standard for Insulation Thickness in Storage Rooms; 6 in styrofoam, 4.7 in urethane average; Walk-in: 4 in Urethane (R=25, k=0.16)	Walk-in: 4 in Urethane (R=25, k=0.16)
(Hendrix et al. 1989)		Pre-fabricated insulated panels (1, 2, and 3); #3 and #4 share a wall with two openings between the spaces	
(Jones et al. 1983)		Framed plywood, 2 in foam insulation	
(Krack 1977)	3" thick; 6" Polystyrene; 5" corkboard to 6" slab urethane (16.7 to 32.4 h-ft^2-°F/Btu, 25.9 average)	3" Urethane; 6" Polystyrene; 5" corkboard to 6" slab urethane (16.7 to 32.4 h-ft^2-°F/Btu, 25.9 average)	6" concrete sealed with 4" Urethane/ Polystyrene block
(Magoo 2003)	44.86 to 49.97 (ASHRAE 1998), insulated (polyurethane, polystyrene, styrofoam) sandwich with either reinforced concrete or sheet metal on either side.	35.21 to 39.75 (ASHRAE 1998), insulated (polyurethane, polystyrene, styrofoam) sandwich with either reinforced concrete or sheet metal on either side.	27.26 to 31.8 (ASHRAE 1998), insulated (polyurethane, polystyrene, styrofoam) sandwich with either reinforced concrete or sheet metal on either side.
(Manske 1999)	33.09 lightweight	32.85 lightweight	

Box Construction for Walk-in Freezers (4 of 7)

	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
(Nagaraju et al. 2001)	6 inches of extruded polystyrene, plywood panels each side	6 inches of extruded polystyrene, plywood panels each side	6 inches of extruded polystyrene
(Navy 1986)	40 to 50	35 to 40	30 to 35
(O'Hagan et al. 1993)		150 mm polystyrene sandwich panel	
(Patel et al. 1993)		2 inches of foam insulation on all walls	
(PG&E 2007)	R-31 to R-50 from layers of 5" isocyanurate (lrg), 6" expanded urethane metal clad panel (smaller facilities); R-46 Insulated low mass roof, 0.8 absorptivity (prototypical)	R-32 to R-56 (range); R-35 (typical) made of 5 to 6 inches expanded urethane metal clad panels; R-26 insulated panel (prototypical)	R-18 to R-30; Glycol tubes set in a mud slab with 4" of rigid styrene and 6" of reinforced concrete; There is a shift away from glycol to electric to negate issues with leakage
(SCE 2008)	> R-36	> R-36	Proposed: R-28 insulation (Energy Independence and Security Act), 6 inches of concrete at R-1.2 (ASHRAE 2005)

Box Construction for Walk-in Freezers (5 of 7)			
	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
(Sherif et al. 2002)	Outside to Inside: 20ga Galvanized sheet, 3/4" Plywood, 4" Polystyrene molded beads, 1" Extruded polystyrene, 1/16" Aluminum sheet (AS), 3.5" Polystyrene molded beads with 2x4s on 2 ft centers, 1/16" AS for 35.71 R-value	Wall 1: 1/16" Aluminum sheet (AS), 3.5" Polystyrene molded beads with 2x4s on 2 ft centers (PS), 1/16" AS for 13.33 R-value; Walls 2-4: 1/16" AS, 3.5" PS, 1/8" AS, 3.5" PS, 1/6" AS for 25.64 R-value	Outside to Inside: Insulated floor, 1/2" Extruded Polystyrene, 4" Polystyrene molded beads, 6 mm Visquene, 16g Galvanized sheet

Box Construction for Walk-in Freezers (6 of 7)			
	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
(Stoecker 1998)		Combining a structural material with insulation and applying an abrasion resistant coating is a traditional method. Insulation bonded to metal skin and tongue and grooved together is becoming more popular. Common materials (Btu/hr-ft-ºF): Cellular foamglass (.029), Cellular polyurethane (.013), Expanded polystyrene (.02), Extruded polystyrene (.015), Glass fiber (.021), Polyisocyanurate (.012); Recommended temperature application for thickness of insulation: 32 to -20°F at 4 inches (R-value of 33.3), -20 to -50°F at 5 inches (R-value of 41.6), -50 to -70°F at 6 inches (R-value of 50)	
(Stoeckle 2000)	21.7	21.7	
(Stoeckle et al. 2002)		21.74, Thermally lightweight (foam-core sandwich) and massive (concrete) studied	Concrete slab on grade with an imbedded glycol heating loop

Box Construction for Walk-in Freezers (7 of 7)			
	Ceiling R-value (h-ft^2-°F/Btu)	Wall R-value (h-ft^2-°F/Btu)	Floor R-value (h-ft^2-°F/Btu)
(Westphalen et al. 1996)		3 or more inches of polyurethane foam insulation encased by galvanized metal; R-27 (typical), 4 inches R-30 (prototypical); 4 to 5 inches; R value 28.6 to 35.75	Insulation required

Table 123. Convection Film Coefficients for Walk-in Freezers

Convection Film Coefficients for Walk-in Freezers			
	External Equivalent Convective Film Coefficient (h-ft^2-°F/Btu)	Internal Equivalent Convective Film Coefficient (h-ft^2-°F/Btu)	Floor Equivalent Convective Film Coefficient (h-ft^2-°F/Btu)
Range	0.17 to 0.68	0.24 to 1.33	0.61 to 0.87
Average	0.42	0.65	0.70
Proposed	0.5	0.5	0.6
(AHRI 2009a)			
(DOE 2010b)	0.68	0.25	0.87
(Krack 1977)	0.17 (outdoor, winds < 15 mph)	0.63 (still air)	
(SCE 2008)			0.61 (ASHRAE 2005)
(Sherif et al. 2002)	0.61 (still air)	0.24 (750 fpm moving air)	
(Stoecker 1998)	15 mph winter, any orientation: 0.167 Btu/h-ft2-°F; 7.5 mph summer, any orientation: 0.25 Btu/h-ft2-°F	Still air, vertical surface, heat flow horizontal: 0.685; Horizontal surface, heat flow downward: 0.926	Horizontal surface, heat flow upward: 0.613;
(Stoeckle 2000)		1.33 (average of free and forced convection in freezer)	

Infiltration

The infiltration load for walk-in freezers is primarily a function of the door height and width, the temperature difference between the interior and exterior of the box, and the door opening schedule, including the duration and frequency of opening. Twenty-six authors presented their chosen method of infiltration calculation, including (Arias 2005), (Becker et al. 2011), (Blackmore 2012), (Chen et al. 1999), (Christensen and Bertilsen 2004), (Cleland 1983), (Cleland et al. 2004), (DOE 2010b), (Elsayed 1998), (Foster et al. 2003), (Gosney and Olama 1975), (Heatcraft 2008), (Huan 2008), (Krack 1977), (Longdill and Wyborn 1978), (Manske 1999), (Nagaraju et al. 2001), (Patel et al. 1993), (PG&E 2007), (Pham and Oliver 1983), (SCE 2008), (Sherif et al. 2002), (Stoecker 1998), (Stoeckle 2000), (Stoeckle et al. 2002), and (Westphalen et al. 1996).

Four preferred infiltration calculation methods were presented, with the Gosney Olama equation (Equation 18) being the most prevalent. This method was used by AHRI (AHRI 2009a) in the development of the model load equations, and was also used in the modified model load. Given a calculation method and a temperature difference between the inside and outside temperatures, the only other additional component necessary to calculate the infiltration load is the door opening schedule. The door opening schedule is a combination of the door opening frequency and the duration of each opening. To better analyze the amount of time the door is open, the door schedule is analyzed as a percentage of time that the door was open. Door open percentages ranged from 0.6 to 25.8% in the literature, with an average value of 9.0%. The AHRI Load Spreadsheet used an open door percentage of 0.42% for the small freezer and 1.94% for the large freezer. These values understated typical infiltration loads and were increased to 5.4% in the modified model load.

Table 124. Infiltration Details for Walk-in Freezers

Infiltration Details for Walk-in Freezers (1 of 7)				
	Infiltration Calculation	Misc. Details on Infiltration	Number of Door Openings	Duration of Door Openings
Range		Average Open Door Percentage: 0.6 to 25.8%	0.57 to 1800	7.5 to 900 seconds
Average		Average Open Door Percentage: 9.0%	415.1	90.8 seconds
Proposed	Gosney Olama	5.4%	13 per hour - 7 am to 7 pm	30 seconds
(AHRI 2009a)	Gosney Olama Equation (ASHRAE Refrigeration Handbook)	Air Density Factor: 0.96	8 openings per hour – 6am to 7am, 2 openings per hour – 7am to 7pm (sm); 8 openings per hour - 6am to 7am, 4 openings per hour - 7am to 7pm (lrg)	30 seconds per opening – 6am to 7am, 5 seconds per opening – 7am to 7pm (sm); 30 seconds per opening – 6am to 7pm (lrg)
(Arias 2005)	Air change method	38 Air changes per 24 hrs for 247 ft^3 room, 31.5 for 353, 21.5 for 706, 14.5 for 1413, 9 for 3531, 3.5 for 17657, 2.5 for 35315, 1.35 for 105944		

Infiltration Details for Walk-in Freezers (2 of 7)

	Infiltration Calculation	Misc. Details on Infiltration	Number of Door Openings	Duration of Door Openings
(Becker et al. 2011)	(Gosney and Olama 1975); Heatcraft Air Changes (Small and Large Freezer)		85 / day (Chili's Food Freezer); 2 openings / day (4 of 14 days) and 0 openings / day (10 of 14 days)	72 s (Chili's Food Freezer); 900 s (FSTC)
(Blackmore 2012)	Vel = 4.88 * sqrt(H * TD)(1-%) Delta h / specific volume; Air change method			
(Chen et al. 1999)		Tested at freight door open periods of 10, 20, 30, 40, 50 seconds and forklift traffic of 8 to 150 passes/hr. Infiltration with the door closed (air tightness) was only 17% of that for a protected open door; Lag Time: 1.5 seconds (Azzouz and Duminil 1993); less than 2 seconds		Rapid roll doors: fully open for 6 to 10 seconds and took 1 to 2 seconds to open/close for each pass; 1.7 seconds fully closed to open and 3.1 seconds from fully open to closed, activated by magnetic sensors; door open for 16.4 seconds for each pass.

Infiltration Details for Walk-in Freezers (3 of 7)				
	Infiltration Calculation	Misc. Details on Infiltration	Number of Door Openings	Duration of Door Openings
(Christensen and Bertilsen 2004)			Store operation is 12 hrs per day	
(Cleland 1983)				3% of the time
(Cleland et al. 2004)		Good sliding door seals leak at 0.26 to 0.52 ft^3 / min-ft	Door opening frequency typically cited as 1 per minute. Loading dock doors at 40 movements per hour.	Rapid-roll doors: Fully open 8 to 20 seconds, 1-2 seconds open/close (Downing and Meffert 1993), Equivalent fully open time of 35 seconds typical; Sliding doors: 4 to 10 second open/close, Equivalent fully open time is typically 60 seconds

Infiltration Details for Walk-in Freezers (4 of 7)

	Infiltration Calculation	Misc. Details on Infiltration	Number of Door Openings	Duration of Door Openings
(DOE 2010b)	Gosney Olama Equation (ASHRAE Refrigeration Handbook)	Door Flow Factor: 0.8 (ASHRAE Fundamentals); Flow between panel joints: 0.13 ft3/hr per ft2 external surface (DOE research)	60 openings per day for passage and freight doors (DOE Test Procedure)	12 seconds per opening for passage and freight doors. Passage and freight doors stand open an additional 15 minutes per day (DOE Test Procedure)
(Elsayed 1998)		10 to 20 mm can represent a gap in a closed door that is not well sealed. Two-dimensional model citing (Gosney and Olama 1975), (Jones et al. 1983), and (Cole 1984)		

Infiltration Details for Walk-in Freezers (5 of 7)

	Infiltration Calculation	Misc. Details on Infiltration	Number of Door Openings	Duration of Door Openings
(Foster et al. 2003)	CFD	Closed door leakage rate of 3.6 cfm; Interior box temperature rose 63.2F at 6.56 ft and 13.9F at 1.64 ft in a 30 second period with the 6.56 ft tall door open, Interior box temperature rose 32.2F at 6.56 ft and 7.2F at 1.64 ft in a 30 second period with the 1.41 ft tall door open; Lag times range from 0.3 to 1.5 seconds, Drop off times range from 17 to greater than 30 seconds.		10, 20, 30, 40 seconds; 7.55 ft wide door takes 8 seconds to open/close, 3.28 ft wide door takes 6 seconds to open/close
(Gosney and Olama 1975)	Gosney as derived		15 times per hr, 8 hrs per day	30 seconds per opening
(Heatcraft 2008)	Average air change or Heatcraft infiltration calculation: [(4.88)sqrt(door height) (area/2) (minutes open) sqrt(temp diff F) (enthalpy incoming - box air)] [(1-X)]/specific volume of incoming air.	Table 90 - Glass Door Loads; Table 88 - Air Change		

Infiltration Details for Walk-in Freezers (6 of 7)

	Infiltration Calculation	Misc. Details on Infiltration	Number of Door Openings	Duration of Door Openings
(Huan 2008)		Heat: 200 kWh; Water vapor: 108 kg/day (Entering door per day)		Open 1 hr/day overall
(Krack 1977)	Vel = 4.88 * sqrt(H * TD)		Open 15 min / hr for 12 hrs; 4 hr/ 24 hr; 20 min/hr for 12 hrs	
(Longdill and Wyborn 1978)		450 kW for enclosed loading docks; 1200 kW with no enclosed loading space		
(Manske 1999)	(Krack 1992)		4 min/hr each (freezer/warming room), 10 min/hr each (freezer/ cooler) for second shift and half of these values for first shift	
(Nagaraju et al. 2001)	Air change method	0.5 air change per hour (247 W)		
(Patel et al. 1993)	Significant portion of load			
(PG&E 2007)	Air change method	0.3 ACH		

Infiltration Details for Walk-in Freezers (7 of 7)				
	Infiltration Calculation	Misc. Details on Infiltration	Number of Door Openings	Duration of Door Openings
(Pham and Oliver 1983)		Enclosed by loading area or exposed; Air curtain (horizontal or vertical), plastic strip curtain, air and plastic curtains, or no protection; No more than one forklift passes / minute; Internal circulation fans on or off	One door opening per minute	
(SCE 2008)		Proposed: 0.78 air changes per hour (sm), 0.53 ACH, 0.36 ACH, 0.21 ACH (lrg) (Heatcraft 2008)		
(Sherif et al. 2002)		Good rubber seals around doors		
(Stoecker 1998)	Gosney and Olama			Opening/closing speeds for freight doors: 25 in/s to 50 in/s; 10 to 15 sec (typical)
(Stoeckle 2000)	(Downing and Meffert 1993)			4 min/hr each
(Stoeckle et al. 2002)	(Downing and Meffert 1993)			4 min/hr each
(Westphalen et al. 1996)		Good door sealing systems		

Product

Various products are stored in walk-in freezers, including fruits, vegetables, meat, and ice cream. The calculation of the product load for a walk-in freezer storing these items is a function of the product entering temperature, box temperature, loading frequency, and loading quantity. These components of the product load were collected from works by(Altwies 1998), (Altwies and Reindl 1999), (Anonymous 2004), (Aparicio-Cuesta and Garcia-Moreno 1988), (Ashby et al. 1979), (Becker et al. 2011), (Blackmore 2012), (Boggs et al. 1960), (Cleland 1983), (DOE 2010b), (Gortner et al. 1948), (Heatcraft 2008), (Henderson and Khattar 1999), (Hustrulid and Winters 1943), (Krack 1977), (Magoo 2003), (Manske 1999), (Morris 2012), (Nagaraju et al. 2001), (Navy 1986), (Goetzler et al. 2009), (Patel et al. 1993), (PG&E 2007), (SCE 2008), (Stoeckle 2000), (Stoeckle et al. 2002), (USDA 2010), (Walker and Baxter 2002), and (Woodroof and Shelor 1947).

Product pull-down temperature difference ranged from 0 to 130°F, with an average value of 19.7°F. Daily loading ratios for products ranged from 0.16 to 2.47 lb/ft^3-day, with an average value of 0.67 lb/ft^3-day. AHRI (AHRI 2009a) applied a product pull down of 10°F with a loading ratio of 0.4 lb/ft^3-day for the large freezer and 3.1 lb/ft^3-day for the small freezer. The pull-down temperature seems appropriate, but to make the loading ratio more consistent these values will be modified to 0.5 lb/ft^3-day for the large freezer and 1.0 lb/ft^3-day for the small freezer.

Table 125. Product Loading for Walk-in Freezers

Product Loading for Walk-in Freezers (1 of 7)				
	Product Type	Product Pull-Down Temp Diff. (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3–day of refrigerated space)
Range		0 to 130		0.16 to 2.47
Average		19.7		0.67
Proposed	Fruits and Vegetables (100% packaged)	10		1 sm, 0.5 lrg
(AHRI 2009a)	Fruits and Vegetables	10	1,600/8 lb per hour from 6:00am to 2:00pm (sm); 20,000/8 lb per hour from 6:00am to 2:00pm (lrg)	3.1 sm, 0.4 lrg

Product Loading for Walk-in Freezers (2 of 7)				
	Product Type	Product Pull-Down Temp Diff. (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3–day of refrigerated space)
(Altwies 1998)	64 ft^3 heavy duty cardboard boxes lined with a plastic vapor barrier (A, B, C); 1 lb retail packaged bags/boxes (D) packed in small 2' x 1' x 1' cardboard boxes arranged on 4' x 4' x 8' pallets; 86.7% of Bulk Storage: 4.2% Baby Lima Beans, 2.8% Broccoli, 10.7% Carrots, 39.1% Corn (cut and cob), 10.3% Green Beans, 19.6% Peas; 56.4% of Cased Storage: 3.5% Broccoli, 3.3% Carrots, 23.9% Corn (cut and cob), 10.8% Green Beans, 15.0% Peas			
(Altwies and Reindl 1999)	Product stored in 64 ft^3 cardboard boxes	10	50 million lb capacity for warehouses A through D; Product-to-Air volume ratio: 0.525 (A and B), 0.694 (C), 0.296 (D)	
(Anonymous 2004)			28 lbs of solid food per 1 ft^3 of open storage area	

Product Loading for Walk-in Freezers (3 of 7)

	Product Type	Product Pull-Down Temp Diff. (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3–day of refrigerated space)
(Aparicio-Cuesta and Garcia-Moreno 1988)	Cauliflower		30 months (cauliflower); 60 days (display freezer)	
(Ashby et al. 1979)	Okra, peas, strawberries in plastic pouches or paperboard cartons placed inside cardboard boxes		Turnover in 1 year	
(Becker et al. 2011)	Fruits and Vegetables, packaged (Small and Large Freezer)	10 (Small and Large Freezer)	Fixed product load, miscellaneous bagged and boxed food products loaded to approximately 50% of the freezer's volume capacity (FSTC); 57 lb/hr with 24 hour pull-down, packaged (Small Freezer); 20,000 lb/hr 6am to 7am with 8 hr pull-down, packaged (Large Freezer)	2.47 (Small Freezer), 0.4 (Large Freezer)
(Blackmore 2012)	Beef	45ºF (beef)	3000 lb beef / day, 24 hr pull-down	0.52 (beef)
(Boggs et al. 1960)	Peas in retail packaging			
(Cleland 1983)			24 hr pull-down period	

Product Loading for Walk-in Freezers (4 of 7)

	Product Type	Product Pull-Down Temp Diff. (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3–day of refrigerated space)
(DOE 2010b)		10		1 sm, 0.5 med, 0.5 lrg
(Gortner et al. 1948)	Pork roasts, strawberries, snap beans, peas; Pork wrapped in cellophane and fruits and vegetables in commercial packaging		Turnover in 1 year	
(Heatcraft 2008)	Heatcraft Table 7	10	1,600 - 2,000 lbs/day (500 - 3,000 ft^3); 2,000 - 2,500 lbs/day (3000 - 4,600 ft3); 2,500 - 4,000 lbs/day (4,600 - 8,100 ft3); 4,000 - 6,200 lbs/day (8,100 - 12,800 ft3); 6,200 - 7,500 lbs/day (12,800 - 16,000 ft3); 7,500 - 9,500 lbs/day (16,000 - 20,000 ft3); 9,500 - 13,000 lbs/day (20,000 - 28,000 ft3); 13,000 - 17,000 lbs/day (28,000 - 40,000 ft3); 17,000 - 25,000 lbs/day (40,000 - 60,000 ft3); 25,000 - 34,000 lbs/day (60,000 - 80,000 ft3); Product Loading Density (Heatcraft Table 7); Walk-in: 24 hour pull down with average load; Heatcraft Table D - Product Freezing Loads for Walk-In Freezers	1.02 average lbs/day-ft^3 (500 - 3,000 ft^3); 0.59 (3000 - 4,600 ft3); 0.51 (4,600 - 8,100 ft3); 0.49 (8,100 - 12,800 ft3); 0.48 (12,800 - 16,000 ft3); 0.47 (16,000 - 28,000 ft3); 0.44 (28,000 - 40,000 ft3); 0.42 (40,000 - 80,000 ft3)

Product Loading for Walk-in Freezers (5 of 7)

	Product Type	Product Pull-Down Temp Diff. (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3–day of refrigerated space)
(Henderson and Khattar 1999)	Meat			
(Hustrulid and Winters 1943)	Snap beans, corn, peas, strawberries, raspberries, cantaloupe; 6 - 9 months		6 month minimum turnover period; 12 to 24 hour pull-down	
(Krack 1977)	Fish, ice cream	55 (Pull-down); 0 to 130 (Krack Table 42)	Blast freezer cools product in 2 hrs. Product packaged, boxed, and palletized requires 16 hrs to cool. Chill factors utilized to compensate for the non-uniform distribution of product load. The later portion of the chill is 15-25% of the peak. 2000 lb fish/day (Pull-down, 16 hr pull-down period, 45.6% of load); 3.3 gal/ft2 * 900 ft2/day (Ice cream, 10 hr pull-down period, 86.4% of load); Average product loads: 0.19 to 0.45 Btu/hr-ft3 (30,000 to 250 ft3, 0.25 Btu/hr-ft3 average); Krack Table 42 (specific product loads)	

Product Loading for Walk-in Freezers (6 of 7)				
	Product Type	Product Pull-Down Temp Diff. (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3–day of refrigerated space)
(Magoo 2003)			Meat, fish, poultry, fruit juices, processed and canned foods, ice cream, seasonal agricultural crops	
(Manske 1999)		0	None. Assuming product arrives at storage temperature.	
(Morris 2012)	Meat	50 (meat)	1000 lb meat / day	0.156 (meat)
(Nagaraju et al. 2001)	Fish	0.9°F per hour	22,046 lb	
(Navy 1986)	Film, packaged products, meat, fish, poultry, ice cream, vegetables, ice, medical		18 to 24 hrs pull-down time	
(Goetzler et al. 2009)	Frozen food			
(Patel et al. 1993)		0	Minimal; Typically fresh and frozen products are received at storage temperature.	
(PG&E 2007)	Ice cream and holding freezers			
(SCE 2008)			Proposed: 70% of capacity; 57,120 Btu/hr (sm), 113,400 Btu/hr, 226,800 Btu/hr, 633,360 Btu/hr (lrg)	
(Stoeckle 2000)	Peas (Altwies 1998)		Occupies 43% of floor area using static pallet rack installation	

Product Loading for Walk-in Freezers (7 of 7)				
	Product Type	Product Pull-Down Temp Diff. (°F)	Product Loading	Daily Loading Ratio (lbs of product per ft^3–day of refrigerated space)
(Stoeckle et al. 2002)	Frozen peas (Altwies and Reindl 1999)		Product is assumed to arrive at storage temperature	
(USDA 2010)			Usable percentages: 57.59 min state ave, 91.85 max state ave, 81.62 median state ave, 81.70 average overall	
(Walker and Baxter 2002)	Meat, bakery, fish, deli, ice cream			
(Woodroof and Shelor 1947)	3-lb metal cans of raspberries, 1-lb waxed cartons of peaches, 10-lb moisture-proof bags in corrugated boxes of strawberries and blackberries		Turnover in 1 year	

Table 126. Product Sensible and Latent Thermal Properties for Walk-in Freezers

Product Sensible and Latent Thermal Properties for Walk-in Freezers (1 of 2)				
	Specific Heat Above Freezing (Btu/lb-F)	High Freezing Point (°F)	Latent Heat of Freezing (Btu/lb)	Specific Heat Below Freezing (Btu/lb-F)
Range	0.74 to 0.98	27 to 31	79 to 130	0.37 to 0.47
Average	0.84	29.3	101.3	0.44
Proposed				0.5
(AHRI 2009a)			0.5	0.5
(Altwies and Reindl 1999)				0.401 (Baby lima beans), 0.471 (broccoli), 0.465 (carrots), 0.423 (corn, peas), 0.468 (green beans)
(Becker et al. 2011)				0.47 (Small and Large Freezer)
(Blackmore 2012)			115 (Fresh Game), 95 (Beef), 79 (Turkey)	
(DOE 2010b)				0.45
(Heatcraft 2008)	Estimate Spec. Heat = 0.20 + (0.008 X % water); Heatcraft Table 7	Heatcraft Table 7	Estimate Latent Heat = 143.3 X % water; Table 7	Estimate Spec. Heat = 0.20 + (0.008 X % water); Heatcraft Table 7
(Krack 1977)		Krack Table 9		

Product Sensible and Latent Thermal Properties for Walk-in Freezers (2 of 2)				
	Specific Heat Above Freezing (Btu/lb-F)	High Freezing Point (°F)	Latent Heat of Freezing (Btu/lb)	Specific Heat Below Freezing (Btu/lb-F)
(Nagaraju et al. 2001)				0.435
(Stoecker 1998)	.902 (apples); .843 (chicken); .792 (peas); .735 (ham, sirloin beef); .783 (salmon); .938 (strawberries)	30 (apples); 27 (chicken); 31 (peas); 29 (ham); 28 (salmon); 31 (strawberries	121 (apples); 107 (chicken, peas); 81 (ham, sirloin beef); 92 (salmon); 130 (strawberries)	.453 (apples); .423 (chicken, peas); .368 (ham, sirloin beef); .392 (salmon); .471 (strawberries)
(Stoeckle et al. 2002)				0.442

Lighting and Occupancy

Lighting and occupancy loads for walk-in freezers are related, as the lights are typically on when people are occupying the space and off when there is no one in the space. Thirteen authors presented material related to these loads, including (Arias 2005), (Becker et al. 2011), (Blackmore 2012), (DOE 2010b), (DOE 2010c), (Goetzler et al. 2009), (Heatcraft 2008), (Krack 1977), (Manske 1999), (PG&E 2007), (SCE 2008), (Sherif et al. 2002), (Stoecker 1998), (Stoeckle 2000), and (Westphalen et al. 1996).

Lighting for the spaces ranged from 0.08 to 3 W/ft^2, with an average value of 0.99 W/ft^2. AHRI (AHRI 2009a) used a value of 1.56 W/ft^2 to define the lighting load for the small freezer and 1 W/ft^2 for the large freezer. The heat contribution of these lights to the space was assumed at 98% for the small freezer which used incandescent bulbs and 85% for the large cooler which utilized fluorescent lighting. The modified load profile adjusts the lighting load to 1 W/ft^2 for the large and small freezers, consistent with the literature, and assumes all of the lighting energy use is a load on the space. All energy used by the lighting will eventually be dissipated as heat into the space and should therefore be included in the lighting load. AHRI assumed 1 person is in the small freezer, according to the occupancy schedule, and 2 people are in the large freezer. This is consistent with what was noted by the literature and will therefore be unchanged in the modified model load profile.

Schedules for lighting and occupancy both ranged from 4 to 100% of the day with average values of 62.7 and 46.5%, respectively. Based on this information, it is a sound assumption that lighting and occupancy loadings have the same schedule. The AHRI Load Spreadsheet's small freezer assumed lighting and occupancy schedules equating to 3.75%

of the time. The large freezer was occupied with the lights on for 10.42% of the time. Both of these schedules are grossly insufficient when compared to the literature. The proposed model load will use occupancy and lighting schedules equating to 20 and 30% of the time, respectively. As discussed for walk-in coolers, the typical loading schedule defined by the literature of around 50% is not used because there is some ambiguity as to whether the percentage applies to the entire day, normal business hours, or over some other partial day period.

Table 127. Lighting and Occupancy Load Details for Walk-in Freezers

Lighting and Occupancy Load Details for Walk-in Freezers (1 of 4)					
	Lighting Power	Lighting Schedule	Occupancy (number of people)	Occupancy time	Occupancy Load (W)
Range	0.08 to 3 W/ft^2	4 to 100% of time	69.9 to 25,000 ft^3/person	4 to 100% of time	
Average	0.99 W/ft^2	62.7% of time	20,845 ft^3/person	46.5% of time	
Proposed	1 W/ft^2	30% of time	1 person (sm), 2 people (lrg)	20% of time	
(AHRI 2009a)	1.56 W/ft^2 sm, 1 W/ft^2 lrg (Light Power Converted to Heat: 98% sm, 85% lrg)	30 minutes per hour – 6am to 7am, 2 minutes per hour – 7am to 7pm (sm); 30 minutes per hour – 6am to 7am, 10 minutes per hour – 7am to 7pm (lrg)	1 person – 6am to 7pm (sm); 2 people – 6am to 7pm (lrg)	30 minutes per hour – 6am to 7am, 2 minutes per hour – 7am to 7pm (sm); 30 minutes per hour – 6am to 7am, 10 minutes per hour – 7am to 7pm (lrg)	
(Arias 2005)	1.39 W/ft^2				200 W per person
(Becker et al. 2011)	1.43 W/ft^2 (FSTC); 1 W/ft^2 (Small and Large Freezer)	6 min/hr 8am to 6pm (Small Freezer); 30 min/hr 6am to 7am, 10 min/hr 7am to 7pm (Large Freezer)	1 person (Small Freezer); 2 people (Large Freezer)	6 min/hr 8am to 6pm (Small Freezer); 30 min/hr 6am to 7am and 10 min/hr 7am to 7pm (Large Freezer)	80.6 W/person latent, 80.6 W/person sensible (Small Freezer); 410.3 W/person (Large Freezer)

Lighting and Occupancy Load Details for Walk-in Freezers (2 of 4)

	Lighting Power	Lighting Schedule	Occupancy (number of people)	Occupancy time	Occupancy Load (W)
(Blackmore 2012)	1 to 1.5 W/ft2 (normal storage), 2.5 to 3 W/ft2 (docks, workrooms)		4 person / 100,000 ft3		
(DOE 2010b)	0.31 W/ft^2 sm; 0.08 W/ft^2 med; 0.09 W/ft^2 lrg				
(DOE 2010c)		On 75% of time (no control), 50% of time (timers or auto shut-off)			
(Goetzler et al. 2009)	0.93 W/ft^2; Typical: 1 W/ft^2 Incandescent (0.9 to 1.2 W/ft^2)	Operate 50% of the time		50% of the time	
(Heatcraft 2008)	1 to 1.5 W/ft^2. Cutting or processing rooms can be double to wattage.		1 person per 24 hours for each 25,000 ft^3		

Lighting and Occupancy Load Details for Walk-in Freezers (3 of 4)					
	Lighting Power	Lighting Schedule	Occupancy (number of people)	Occupancy time	Occupancy Load (W)
(Krack 1977)	1 to 1.5 W/ft^2				Heat equivalents for short period occupancy should be increased by 20%
(Manske 1999)	0.45 W/ft^2	16 hrs/day (first and second shift)			Sensible: 707.8 W (first), 1415.6 (second); Latent: 1425.8 (first), 2851.7 (second)
(PG&E 2007)	0.4 to 1.2 W/ft^2; 0.6 (prototypical)	24/7 operation	80 max	24/7 operation	
(SCE 2008)	Proposed: 1.0 W/ft^2 (Heatcraft Refrigeration); High efficacy fluorescent lighting	75%		25%	
(Sherif et al. 2002)	Fluorescent (2) 60 W ea; 0.90 W/ft^2	90.2% operating time			
(Stoecker 1998)	.48 W/ft^2				270 W/person at 32°F, 330 W/person at 14°F, 390 W/person at -4°F

Lighting and Occupancy Load Details for Walk-in Freezers (4 of 4)					
	Lighting Power	Lighting Schedule	Occupancy (number of people)	Occupancy time	Occupancy Load (W)
(Stoeckle 2000)	0.45 W/ft^2				7920 W
(Westphalen et al. 1996)	0.9 to 1.0 W/ft^2; Incandescent; 1 W/ft2 (prototypical)	On when occupied by employees. 50% of time.		50% of the time	

Miscellaneous Loads

All other loads on walk-in freezers are analyzed in this section. Miscellaneous loads specifically considered include vehicle operation, evaporator fan operation, and passage door heater power. Information on these loads was collected from (Becker et al. 2011), (Blackmore 2012), (DOE 2010b), (Goetzler et al. 2009), (Heatcraft 2008), (Krack 1977), (Manske 1999), (Nagaraju et al. 2001), (PG&E 2007), (SCE 2008), (Sherif et al. 2002), (Stoeckle 2000), and (Westphalen et al. 1996).

Vehicle energy usage in the refrigerated space ranged from 0.56 to 3.22 W-hr/day-ft^3 in the literature, with an average of 1.48 W-hr/day-ft^3. The AHRI Load Spreadsheet (AHRI 2009a) used a value of 0.38 W-hr/day-ft^3 to define the vehicle energy usage in the large freezer. This parameter will be modified to 1.00 W-hr/day-ft^3 in the proposed model load profile. Passage door heaters were not considered by the AHRI Load Spreadsheet but are specified as ranging from 3 to 11.5 W/ft of door perimeter by the literature, with an average value of 7.5 W/ft. Eight watts per ft will be used in the modified model load.

Table 128. Miscellaneous Load Details for Walk-in Freezers

Miscellaneous Load Details for Walk-in Freezers (1 of 3)					
	Number of Vehicles	Vehicle Energy Usage	Evaporator Fan Operation	Passage Door Heater Power	Miscellaneous Heat Load
Range		0.56 to 3.22 W-hr/day-ft^3		3 to 11.5 W/ft	23 to 45,700 W
Average		1.48 W-hr/day-ft^3		7.5 W/ft	10,446 W
Proposed		1.00 W-hr/day-ft^3 (lrg)		8 W/ft	
(AHRI 2009a)	0 sm, 1 lrg	0 sm; 37.3 kW at 30 minutes per hour – 6am to 7am (lrg)			
(Becker et al. 2011)	1 (Large Freezer)	0.746 W-hr/day-ft^3 (Large Freezer)	always on		
(Blackmore 2012)		5.97 kW/100,000 ft3; 2.98 to 3.73 kW per forklift			
(DOE 2010b)				8 W/ft; 24 hrs/day	2-way Pressure Relief Valve Heater: 23 W, 24 hr operation
(Goetzler et al. 2009)				230 W; Always on	
(Heatcraft 2008)	Generally only battery operated lift trucks used; 1 motor horsepower for each 12,500 ft^3 of storage.	2.3 to 4.4 kW over the period of operation; Heatcraft Table 14 - Heat gain due to operation of battery operated lift truck			Heatcraft Table 11 - Heat Equivalency of Motors

Miscellaneous Load Details for Walk-in Freezers (2 of 3)

	Number of Vehicles	Vehicle Energy Usage	Evaporator Fan Operation	Passage Door Heater Power	Miscellaneous Heat Load
(Krack 1977)		Forklifts may be estimated at 3.0 to 3.7 kW; 40 min/hr operating time			
(Manske 1999)		17.3 kW (first); 34.6 (second)	cycle with load		35.2 kW (warming room load on freezer) typical, 45.7 kW during June to Sept
(Nagaraju et al. 2001)					Varies 700 to 2350 W
(PG&E 2007)		0.7 W/ft^2 (forklifts and miscellaneous equipment)			
(SCE 2008)					Anti-sweat
(Sherif et al. 2002)				16 ft Chromalox per door; 3 W/ft at 40°F. Assuming 50% of heat enters the freezer space; 24 hr/day operation	10 kW Artificial load generator

Miscellaneous Load Details for Walk-in Freezers (3 of 3)					
	Number of Vehicles	Vehicle Energy Usage	Evaporator Fan Operation	Passage Door Heater Power	Miscellaneous Heat Load
(Stoeckle 2000)		63.89 kW total (Manske 1999)			
(Westphalen et al. 1996)			100% on		230 W anti-sweat heaters for access door

Defrost

An effective defrost method is necessary for adequate space cooling. Because of the effect of this component on the refrigeration system, numerous authors have done research or presented practical information on defrost systems, including (Becker et al. 2011), (DOE 2010b), (Goetzler et al. 2009), (Heatcraft 2008), (Henderson and Khattar 1999), (Krack 1977), (Mago and Sherif 2005), (Manske 1999), (Nelson 2011), (SCE 2009), (Stoecker 1998), (Stoeckle 2000), (Walker et al. 1990), (Walker 1992), (Walker 2001), and (Westphalen et al. 1996). According to the literature the most prevalent defrost system for freezers is hot gas, with electric element defrost as the second most utilized.

Electric defrost systems use anywhere from 1.3 to 34.5 W/ft^2 according to the literature, with an average value of 17.8 W/ft^2. Defrost schedules range from 1 to 4.8 times per day, with a total defrost period ranging from 0.5 to 6 hr/day. The average defrost schedule was 2.4 times/day at 1.7 hr/day. Based on this information, a defrost schedule of 2 times per day at 1 hr per defrost cycle (2 hr/day) will be applied to the AWEF calculation.

Table 129. Defrost Details for Walk-in Freezers

Defrost Details for Walk-in Freezers (1 of 5)					
	Defrost Type	Electric Defrost + Drain-down Heater Power	Defrost Heat Space Load (Btu/h)	Defrost Efficiency	Defrost Schedule
Range		1.3 to 34.5 W/ft^2			1 to 4.8 times per day; 0.5 to 6 hr/day
Average		17.8 W/ft^2			2.4 times per day; 1.7 hr/day
Proposed					2 times per day; 2 hr/day
(AHRI 2009a)					
(Becker et al. 2011)	Electric (FSTC)	90 W (FSTC)			3 cycles per day at 37.8 min / cycle (Chili's Food Freezer)
(DOE 2010b)		1.656 kW (sm); 2.756 (lrg)			1 hr / day runtime (15 minute cycles, 4 times per day)
(Goetzler et al. 2009)		25 W/ft^2 (typical); Baseline: 1500 W Defrost, 500 W Pan			Time Initiated/ Temperature Terminated; 4% of the time; 60 min every 24 hrs
(Heatcraft 2008)	Electric, hot gas				

Defrost Details for Walk-in Freezers (2 of 5)

	Defrost Type	Electric Defrost + Drain-down Heater Power	Defrost Heat Space Load (Btu/h)	Defrost Efficiency	Defrost Schedule
(Henderson and Khattar 1999)	Store A: Hot gas; Store B: Electric				
(Krack 1977)				25% of the heat is added to the space	6 hr / 24 hr
(Mago and Sherif 2005)	Hot gas (typical)		35.44 to 95.03 Btu/hr; 2.5% of total industrial refrig. load (typical)	Dampers increase defrost efficiency by 43%. 18 to 70.4% for air rates from 315 to 1280 fpm.	15 minutes every 5 hrs; 30 minutes every 24 hrs (typical)
(Manske 1999)	Hot gas		Heat load: 72 MBH June to Sept, 60 MBH otherwise (hot gas, freezer)	50% (ASHRAE, 1994)	Hot gas defrosted twice a day on a time scheduled basis. Evaporator defrost cycles are staggered.

Defrost Details for Walk-in Freezers (3 of 5)

	Defrost Type	Electric Defrost + Drain-down Heater Power	Defrost Heat Space Load (Btu/h)	Defrost Efficiency	Defrost Schedule
(Nelson 2011)	Types include: water, electric, hot gas - with hot gas being the preferred method		15 - 20% to melt ice, 60% lost to room via convection/radiation, 20% required to heat and cool the metal of the evaporator, and 5% lost due to hot gas bypassing the defrost regulator at the end of the defrost (Cole 1989). -10F room: 63% lost to room, 17% melt frost, 19% heat and cool metal, and 2% warm melt	max theoretical defrost efficiency: 60 to 70% (Cole 1989); Examples: 17% (3), 11%, 28% (2), 44%, 34%, 29%, 17% (-10F room); 14% (-30F room typical); 18% (0F room typical)	Typical: 30 minutes, Optimized: 8 to 10 minutes
(SCE 2009)	Electric				4 times per day (typical)
(Stoecker 1998)	Hot gas, water, or electric		4% of refrigeration load (typical)	20% (hot gas)	Twice a day 20 minutes each

Defrost Details for Walk-in Freezers (4 of 5)

	Defrost Type	Electric Defrost + Drain-down Heater Power	Defrost Heat Space Load (Btu/h)	Defrost Efficiency	Defrost Schedule
(Stoeckle 2000)				20% (Jekel and Reindl 2000)	
(Walker et al. 1990)	Electric, hot gas				Conventional Low Temp Electric: 48 min at 00:30 (#8); 56 min at 05:00 (#1); 60 min at 01:30 (#4); 60 min at 03:30 (#5); 25 min at 02:00, 10:00, an 18:00 (#18); 50 min at 00:00, 06:00, 12:00, 18:00 (#21); Multiplex Low Temp Hot Gas: 34 min at 23:30 (#8); 30 min at 03:30 (#7); 40 min at 00:30 (#4); 40 min at 02:30 (#5); Low Temp Electric: 60 min at 22:30 (#A8); 60 min at 03:30 (#A7); 60 min at 00:30 (#A4); 60 min at 02:30 (#A5); 26 min at 05:30, 13:30, 21:30 (#A18); 40 min at 04:30, 10:30, 16:30, 22:30 (#A21)
(Walker 1992)	Electric				

Defrost Details for Walk-in Freezers (5 of 5)					
	Defrost Type	Electric Defrost + Drain-down Heater Power	Defrost Heat Space Load (Btu/h)	Defrost Efficiency	Defrost Schedule
(Walker 2001)	Hot gas (multiplex and distributed), warm glycol (secondary loop)				
(Westphalen et al. 1996)	Electric; Time Initiated / Temperature Terminated	25 W/ft^2; 1500 W defrost, 500 W pan heater time initiated, temp terminated (prototypical)			4.2% duty cycle (60 min every 24 hrs)

Loading Summary

Following an in-depth analysis of literature related to the loads on walk-in freezers, a proposed model load profile was developed. Some assumptions made by the AHRI Load Spreadsheet (AHRI 2009a) were found to not correlate with information gathered from the literature while other information seemed appropriate.

Walk-in freezer interior conditions were adjusted from -5°F/50% relative humidity to -10°F/80% relative humidity to better match typical operating conditions. The box dimensions and insulating value of the walls, ceiling, and floor were within reasonable agreement with the literature and were therefore not changed. Conduction load components that were changed include ground temperature (increased from 50 to 55°F) and convective coefficients (0.5 h-ft^2-°F/Btu for internal and external convection, 0.6 h-ft^2-°F/Btu for floor convection).

The infiltration calculation used by AHRI (AHRI 2009a) is the same as used by a majority of the researchers. The other component to determine the infiltration load is the percentage of time that the doorway is open. Reviewing the literature, highlighted the fact that the door open percentage was understated by AHRI, at 0.42% for the small freezer and 1.94% for the large freezer. The percentage was increased to 5.4% in the modified model load.

The product load is determined from the amount of product loaded and the amount of heat that must be removed from the product to reduce its temperature to the storage temperature. The product pull-down temperature difference was not modified, but the loading ratio was modified for the large and small freezers. AHRI used loading ratios of

0.4 lb/ft^3-day for the large freezer and 3.1 lb/ft^3-day for the small freezer, but these were modified to 0.5 lb/ft^3-day for the large freezer and 1.0 lb/ft^3-day for the small freezer.

Lighting and occupancy loads are composed of a heat load and a schedule defining when lights are on or when people occupy the space. The lighting heat load was only modified for the small freezer from 1.56 W/ft^2 to 1.0 W/ft^2, to match typical lighting levels and the lighting load for the large freezer. The modified model load assumes that all energy used by the lighting is eventually converted to heat for the space, unlike AHRI which assumed that only a portion of it was converted to heat. The number of people in the space was kept consistent with the AHRI Load Spreadsheet at one person for the small freezer and two for the larger freezer. The schedules for lighting and occupancy were defined by the percentage of time that the heat load is applied. The AHRI occupancy and lighting period was greatly increased from 3.75 and 10.42% for the small and large freezer, respectively, to 20% for the occupancy period and 30% for the lighting period percentage.

Defrost load was not analyzed by the AHRI Load Spreadsheet but is analyzed by this work for calculation of the AWEF. Defrost load is assumed to operate twice a day for 1 hr each time.

Miscellaneous loads include the heat generated by forklifts and passage door heaters. Vehicle loads for the walk-in freezer were modified from an understated value of 0.38 W-hr/day-ft^3 to 1.00 W-hr/day-ft^3 in the proposed model load profile. Passage door heaters were not considered by the AHRI Load Spreadsheet but will be assumed to be 8 W/ft of door perimeter by the proposed model load.

Summary

Following an in-depth literature search, information related to the model load profile was compiled and analyzed for walk-in coolers and freezers. The range and average of each component of the model load was compared to the value used by the AHRI Load Spreadsheet (AHRI 2009a). If the values did not agree, adjustments were made for the proposed model load profile.

Variations between the AHRI Load Spreadsheet and the proposed model load include the consideration of convection, ground temperature, door open percentage, pounds of product loaded per day, occupancy and lighting percentage, lighting power percentage added to the refrigerated space, small walk-in lighting load, freezer interior conditions, vehicle heat load, and the addition of a passage door heater load.

CHAPTER 5

DEVELOPMENT OF PROPOSED AHRI STANDARD 1250/1251 EQUATIONS

Following an extensive review and analysis of information found in literature related to the loads seen by walk-in coolers and freezers, a proposed model load was developed. Using the AHRI Load Spreadsheet (AHRI 2009a) as a starting point, a modified calculation method was developed. Using proven methods, proposed AHRI Standard 1250/1251 (AHRI 2009b; AHRI 2009c) box load equations were determined.

Modifications to Calculation Method

The AHRI Load Spreadsheet (AHRI 2009a) was used as a starting point to develop the modified model load calculation. Variations in the basic calculations include more precise values for air density and humidity content, infiltration sensible and latent heat load calculations via Gosney Olama, addition of respiration load to coolers, modification of lighting heat load calculation, and addition of door heaters

Approximate air densities and moisture levels are used by AHRI for the site and box operating conditions. These values are used to calculate the infiltration sensible and latent loads. The approximate density values are up to 3.5% different than the actual density, and humidity varies by up to 73.9% from actual. To improve the accuracy of the calculation, air density and humidity values were calculated using (Sugar_Engineers 2012) which uses a calculation method based off (ASHRAE 1989).

The infiltration load calculation used by the AHRI Load Spreadsheet used three main steps to arrive at sensible and latent loads. The infiltration rate was determined via Gosney and Olama's infiltration equation (Equation 12), which is widely accepted as the best correlation to actual infiltration flow. An infiltration lag factor of 2.5 seconds is applied to each door opening to arrive at an equivalent open door period. To determine the sensible and latent heat load components Equations 8 and 9 are applied, respectively.

$$\dot{Q}_s = 1.1 \frac{Btu-\min}{ft^3 hr ^\circ F} \cdot \dot{V}_{\inf} \cdot \left(T_H - T_C\right) \tag{8}$$

$$\dot{Q}_L = 5500 \frac{Btu-\min}{ft^3 hr} \cdot \dot{V}_{\inf} \cdot \left(\omega_H - \omega_C\right) \tag{9}$$

The constant equal to 1.1 in Equation 8 represents the product of air density, the specific heat of air at constant pressure (C_p), and a conversion from minutes to hours. One interesting thing to note is that when back calculated, using an excepted value of 0.24 Btu/lb_m-°F for the air specific heat, the air density that composes this constant is not equal to the average of the box and surrounding approximated densities previously cited in the load calculation but is 1.5 or 3.8% smaller, depending on whether the walk-in is operating as a cooler or a freezer. To improve this calculation method, Equation 122 was utilized.

$$\dot{Q}_s = \dot{V}_{\inf} \cdot \left(T_H - T_L\right) \cdot C_p \frac{\rho_H + \rho_C}{2} \cdot 60 \frac{\min}{hr} \tag{122}$$

The constant of 5500 utilized by Equation 9 is composed of a conversion from minutes to hours and the quotient of the water vapor properties of enthalpy and specific volume according to Equation 123. This equation is a fairly rough equation, as the values it is assuming for specific enthalpy and volume are averaged for the refrigerated space conditions and surrounding conditions.

$$5500 \frac{Btu-\min}{ft^3 hr} = \frac{h_g}{\upsilon_g} 60 \frac{\min}{hr} \tag{123}$$

Instead of using Equation 9 to calculate the latent load, a method developed by Gosney and Olama (Gosney and Olama 1975) and presented in Equations 124 and 125 was utilized.

$$\dot{Q}_L = \dot{V}_{\inf}\left(h_{H,mix} - h_{C,mix}\right)\frac{\rho_H - \rho_C}{2}60\frac{\min}{hr} - \dot{Q}_s \qquad (124)$$

$$h_{mix} = \frac{h}{1+\omega} \qquad (125)$$

The product load calculated by the AHRI Load Spreadsheet (AHRI 2009a) was only a sensible heat load. To allow calculation of the latent and sensible heat load due to continuous respiration of products stored in walk-in coolers the usable storage space and product heat of respiration as a function of temperature were added to the calculation set. The latent product load is calculated according to Equation 126, where R_{T1} is a function of the box temperature.

$$\dot{Q}_{resp} = m_p R_{T1}\left(1 - F_{pp}\right) \qquad (126)$$

The lighting load calculation used in the AHRI Load Spreadsheet (AHRI 2009a) is reduced to take into account different lighting efficiencies. According to the calculation, incandescent lights convert 98% of this power into heat whereas fluorescent lights only convert 85% into heat. To determine the refrigerated space lighting load, the lighting power is reduced by 2% for incandescent bulbs and 15% for fluorescent. This approach was altered in the modified calculation method, as all energy added to the space either as heat or radiated light is eventually converted to heat which must be rejected from the refrigerated space.

Passage door heaters are commonly used, as was identified through the literature search. Calculation of the heat load added by passage door heaters was added to the calculation method, based off wattage per linear foot of door heater.

Proposed Model Load Calculation Inputs

Based on the review and analysis of information gathered from the literature, the model load inputs are summarized in Table 130, Table 131, and Table 132. This information was put into a calculation spreadsheet to define the proposed model load profiles for four walk-ins.

Table 130. Proposed Walk-in Cooler and Freezer Model Load Inputs

	Proposed	(AHRI 2009a)	Average	Range
Outdoor Ambient Temperature (ºF)	80, 95, 110	80, 95, 110	81.8	35 to 122
Sun Loading Effect on Roof Temp	8 hrs at Amb + 20ºF	12 hrs at Amb + 15ºF	Ambient + 33.2ºF	Ambient + 2 to 76ºF
Indoor Ambient Temperature (ºF)	75	75	73.5	50 to 90
Indoor Ambient Relative Hum (%)	50	50	49.6	30 to 65
Infiltration Protection (%)	85	85		
Reach-in Door Quantity	0	0	18.5	3 to 76
Passage Door Quantity	1 sm	1 sm	1.3	1 to 2
Freight Door Quantity	1 lrg	1 lrg		
Passage Door Width (ft)	3 sm	4 sm	3.2	1.7 to 4.0
Passage Door Height (ft)	7 sm	7 sm	6.4	5.2 to 7.9
Passage Door Glass Area (ft^2)			1.2	0.9 to 1.8
Freight Door Width (ft)	6 lrg	6 lrg	8.6	4.5 to 14
Freight Door Height (ft)	10 lrg	10 lrg	10.4	7.0 to 14
Door Construction	Same as walls	Same as walls	Typically same as walls	

Table 131. Proposed Walk-in Cooler Model Load Inputs

Proposed Walk-in Cooler Model Load Inputs (1 of 2)				
	Proposed	(AHRI 2009a)	Average	Range
Interior Box Temperature (ºF)	35	35	38.6	0 to 86
Interior Relative Humidity (%)	80	90	80.0	40 to 100
Box Width (ft)	8 sm 50 lrg	8 sm 50 lrg	14.9	5 to 50
Box Length (ft)	8 sm 50 lrg	8 sm 50 lrg	24.1	6 to 63
Box Height (ft)	8 sm 20 lrg	8 sm 20 lrg	12.0	6.6 to 32.8
Box Floor Area (ft^2)	64 sm 2500 lrg	64 sm 2500 lrg	702.5	30 to 3000
Box Volume (ft^3)	512 sm 50,000 lrg	512 sm 50,000 lrg	8863.7	7 to 89,841
Floor Heat Load (Btu/hr-ft^2)	1.28	0.6	3.29	0.97 to 8.61
Ground Temperature (ºF)	55	50	57.6	40 to 80
Ceiling R-value (h-ft^2-°F/Btu)	25	25	28.4	7.9 to 60
Wall R-value (h-ft^2-°F/Btu)	25	25	23.9	7.9 to 42.5
Floor R-value (h-ft^2-°F/Btu)	15	25	14.0	On slab to 42.5
External Equivalent Convective Film Coefficient (h-ft^2-°F/Btu)	0.5		0.35	0.17 to 0.68
Internal Equivalent Conv Film Coef (h-ft^2-°F/Btu)	0.5		0.52	0.25 to 0.68
Floor Equivalent Conv Film Coef (h-ft^2-°F/Btu)	0.6		0.70	0.61 to 0.87
Infiltration Flow Calc Method	(Gosney and Olama 1975)	(Gosney and Olama 1975)	(Gosney and Olama 1975) most accepted	
Open Door Percentage (%)	5.4	1.18 sm 2.78 lrg	12.7	1.3 to 100

Proposed Walk-in Cooler Model Load Inputs (2 of 2)				
	Proposed	(AHRI 2009a)	Average	Range
Number of Door Openings / day	156	54 sm 80 lrg	174.2	9.3 to 960
Duration of Each Door Opening (s)	30	18.9 sm 30 lrg	75.9	8.8 to 316.5
Product Pull-down Temperature (°F)	10	10	23.8	0 to 68
Product Pull-down Period (hrs)	24	8	115.7	8 to 720
Daily Loading Ratio (lb/ft^3 walk-in)	4.0 sm 1.8 lrg	12.11 sm 1.60 lrg	1.91	1.23 to 4.06
Product Spec Heat Above Freezing (Btu/lb-°F)	0.9	0.9	0.80	0.50 to 0.91
Product Respiration Coef. at 35°F (Btu/lb-hr)	0.0292		0.0583	0.0147 to 0.3797
Percentage Product Vapor-Barrier Packaged (%)	50			
Lighting Power (W/ft^2)	1.00	1.56 sm 1.00 lrg	1.50	0.08 to 9.83
Percentage of Light Power Impacting Heat Load (%)	100	98 sm 85 lrg		
Lights On Percentage (%)	30	3.75 sm 12.5 lrg	54.7	4 to 100
People in Space	1 sm 2 lrg	1 sm 2 lrg	1.5	1 to 2
Occupancy Percentage (%)	20	5.83 sm 12.5 lrg	34.7	3 to 100
Vehicle Power (W-hr/ft^3-day)	1.00 lrg	0.75 lrg	0.59	0.08 to 1.12
Door Frame Heater (W/ft)	8		11.5	8 to 15
Defrost Power (W/ft^2)			25.7	19.5 to 29.6
Defrost Frequency (cycles per day)	2		3.0	0.8 to 8
Defrost Duration (hr/day)	2		1.3	0.5 to 4

Table 132. Proposed Walk-in Freezer Model Load Inputs

Proposed Walk-in Freezer Model Load Inputs (1 of 2)				
	Proposed	(AHRI 2009a)	Average	Range
Interior Box Temperature (ºF)	-5	-10	-0.3	-40 to 32.4
Interior Relative Humidity (%)	80	50	77.9	50 to 95
Box Width (ft)	8 sm 50 lrg	8 sm 50 lrg	13.3	5 to 50
Box Length (ft)	8 sm 50 lrg	8 sm 50 lrg	18.2	6 to 50
Box Height (ft)	8 sm 20 lrg	8 sm 20 lrg	11.4	6 to 69.6
Box Floor Area (ft^2)	64 sm 2500 lrg	64 sm 2500 lrg	604.5	30 to 3000
Box Volume (ft^3)	512 sm 50,000 lrg	512 sm 50,000 lrg	4677.1	7 to 50,000
Floor Heat Load (Btu/hr-ft^2)	2.34	1.88	3.08	1.86 to 5.56
Ground Temperature (ºF)	55	50	56.2	40 to 79.7
Ceiling R-value (h-ft^2-°F/Btu)	32	32	33.0	16.7 to 50
Wall R-value (h-ft^2-°F/Btu)	28	32	28.0	3.3 to 83.3
Floor R-value (h-ft^2-°F/Btu)	25	32	25.1	On slab to 35.4
External Equivalent Conv Film Coef (h-ft^2-°F/Btu)	0.5		0.42	0.17 to 0.68
Internal Equivalent Conv Film Coef (h-ft^2-°F/Btu)	0.5		0.65	0.24 to 1.33
Floor Equivalent Conv Film Coef (h-ft^2-°F/Btu)	0.6		0.70	0.61 to 0.87
Infiltration Flow Calc Method	(Gosney and Olama 1975)	(Gosney and Olama 1975)	(Gosney and Olama 1975) most accepted	
Open Door Percentage (%)	5.4	0.42 sm 1.94 lrg	9.0	0.6 to 25.8

Proposed Walk-in Freezer Model Load Inputs (2 of 2)				
	Proposed	(AHRI 2009a)	Average	Range
Number of Door Openings / day	156	32 sm 56 lrg	415.1	0.57 to 1800
Duration of Each Door Opening (s)	30	11.3 sm 30 lrg	90.8	7.5 to 900
Product Pull-down Temperature (°F)	10	10	19.7	0 to 130
Product Pull-down Period (hrs)	24	8	17.1	8 to 24
Daily Loading Ratio (lb/ft^3 walk-in)	1.0 sm 0.5 lrg	3.1 sm 0.4 lrg	0.67	0.16 to 2.47
Product Spec. Heat Belo Freezing (Btu/lb-°F)	0.5	0.5	0.44	0.37 to 0.47
Percentage Product Vapor-Barrier Packaged (%)	100	100		
Lighting Power (W/ft^2)	1.00	1.56 sm 1.00 lrg	0.99	0.08 to 3
Percentage of Light Power Impacting Heat Load (%)	100	98 sm 85 lrg		
Lights On Percentage (%)	30	3.75 sm 10.42 lrg	62.7	4 to 100
People in Space	1 sm 2 lrg	1 sm 2 lrg	1.3	1 to 2
Occupancy Percentage (%)	20	3.75 sm 10.42 lrg	46.5	4 to 100
Vehicle Power (W-hr/ft^3-day)	1.00 lrg	0.38	1.48	0.56 to 3.22
Door Frame Heater (W/ft)	8		7.5	3 to 11.5
Defrost Power (W/ft^2)			17.8	1.3 to 34.5
Defrost Frequency (cycles per day)	2		2.4	1 to 4.8
Defrost Duration (hr/day)	2		1.7	0.5 to 6

Proposed Model Load Hourly Calculations

Applying the inputs listed in Table 130, Table 131, and Table 132 to a calculation spreadsheet, hourly loads were determined for the proposed walk-in freezers and coolers. From the hourly loads that are calculated, the AHRI Standard 1250/1251 (AHRI 2009b; AHRI 2009c) box load equations may be derived. Hourly model load calculations for the small cooler, large cooler, small freezer, and large freezer are presented in Table 133, Table 134, Table 135, and Table 136, respectively. Information is presented in these tables in groups of variables that are used together to calculate certain portions of the refrigeration heat load.

Table 133. Proposed Small Walk-in Cooler Hourly Model Load Calculation

Proposed Small Walk-in Cooler Hourly Model Load Calculation (1 of 3)

SMALL COOLER	AM												PM											
Hour	12:00	1:00	2:00	3:00	4:00	5:00	6:00	7:00	8:00	9:00	10:00	11:00	12:00	1:00	2:00	3:00	4:00	5:00	6:00	7:00	8:00	9:00	10:00	11:00
INPUTS AND INITIAL CALCULATIONS																								
Outdoor Site Conditions																								
Outdoor Ambient Temperature (°F)	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95
Outdoor Ambient Temperature (°C)	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0
Sunlight Loading Effect on Roof Temp (°F)	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20
Schedule - Sunlight Directly on Roof (Y/N)	N	N	N	N	N	N	N	N	N	N	Y	Y	Y	Y	Y	Y	Y	Y	N	N	N	N	N	N
Indoor Site Conditions																								
Indoor Ambient Temperature (°F)	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
Indoor Ambient RH (%)	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50
Indoor Ambient Air Specific Volume (ft3/lb)	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7
Indoor Ambient Air Humidity Ratio (kg water/kg dry air)	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01
Indoor Ambient Enthalpy (Btu/lb dry air)	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11
Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85
Box Interior Conditions																								
Interior Temperature (°C)	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7
Interior Temperature (°F)	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35
Interior RH (typical) (%)	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Interior Air Specific Volume (ft3/lb)	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5
Interior Air Humidity Ratio (kg water/kg dry air)	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003
Interior Air Enthalpy (Btu/lb dry air)	12.064	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06
Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02
Box Interior Dimensions																								
Case Width (ft)	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00
Case Length (ft)	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00
Case Height (ft)	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00
Case Floor Area (ft^2)	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64
Case Volume (ft^3)	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512
Usable Volume % for Storage * Utilization of Storage Area	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%
Usable Volume (ft^3)	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230
Floor Conditions																								
Ground Temperature (°F)	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55
Floor Heat Load (Btu/hr-ft^2)	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78	0.78
Box Construction																								
Box Location (IN, OUT)	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT
Ceiling R-value (h-ft^2-°F/Btu)	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32
Wall R-value (h-ft^2-°F/Btu)	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28
Floor R-value (h-ft^2-°F/Btu)	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25
Convection Film Coefficients																								
External Convective Film Coefficient (h-ft^2-°F/Btu)	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Internal Convective Film Coefficient (h-ft^2-°F/Btu)	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Floor Convective Film Coefficient (h-ft^2-°F/Btu)	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Infiltration Details																								
Door Height (f)	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7
Door Width (f)	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
Door R-value (h-ft^2-°F/Btu)	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28
Protective Device Effectiveness (%)	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85
Door openings in this hour (#/hour)	0	0	0	0	0	0	0	13	13	13	13	13	13	13	13	13	13	13	13	0	0	0	0	0
Time per opening (s/opening)	0	0	0	0	0	0	0	30	30	30	30	30	30	30	30	30	30	30	30	0	0	0	0	0
Open Door Percentage (%)	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	0.0%	0.0%	0.0%	0.0%	0.0%
Crack Infiltration (cfm)	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Proposed Small Walk-in Cooler Hourly Model Load Calculation (2 of 3)

SMALL COOLER

Product Loading																								
Product Type	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits an Vegetabl
Product Original Temperature (°F)	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45
Product Pull-down Temperatuer (°F)	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Product Pull-down Period (hrs)	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24
Daily Loading Ratio (lb/ft^3)	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00
Schedule - Product Loading (%)	0	0	0	0	0	0	0	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	0	0	0	0	0
Product Sensible and Latent Thermal Properties																								
Product Specific heat above freezing (Btu/lb/F)	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
Specific heat below freezing (Btu/lb/F) (if applicable)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Heat of Respiration at box temperatre (Btu/lb/hr)	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
Density (lb/ft3)	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00
Percentage of Product that is packaged	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%
Lighting and Occupancy Details																								
Lighting Power (W/ft^2)	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Schedule - Lights time on during the hour (min)	0	0	0	0	0	0	0	36	36	36	36	36	36	36	36	36	36	36	36	0	0	0	0	0
Schedule - Lights time on during the hour (%)	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	0.0%	0.0%	0.0%	0.0%	0.0%
#People	0	0	0	0	0	0	0	2	2	2	2	2	2	2	2	2	2	2	2	0	0	0	0	0
Schedule - occupancy during the hour (min)	0	0	0	0	0	0	0	24	24	24	24	24	24	24	24	24	24	24	24	0	0	0	0	0
Schedule - Occupancy during the hour (%)	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	0.0%	0.0%	0.0%	0.0%	0.0%
Miscellaneous Load Details																								
#Motorized Vehicles	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Vehicle Power (W-hr/ft^3-day)	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Vehicle Power (hp)	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Minutes operating during the hour (min)	0	0	0	0	0	0	0	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0	0	0	0	0
Vehicle Power (W-hr/ft^3)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Passage Door Heater (W/ft)	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8

CONDUCTION CALCULATION

Box Location (OUT, IN)	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT
Box Temperature (°F)	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35
Outdoor Ambient (°F)	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95
Indoor Ambient (°F)	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
Wall R-value (h-ft^2-°F/Btu)	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29
Walls Total Area (sq. ft.)	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00
Walls Heat Load (Btu/h)	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52	485.52
Roof Temperature (°F)	95	95	95	95	95	95	95	95	95	95	115	115	115	115	115	115	115	115	95	95	95	95	95	95
Ceiling R-value (h-ft^2-°F/Btu)	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33
Ceiling Total Area (sq. ft.)	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00
Ceiling Heat Load (Btu/h)	116.36	116.36	116.36	116.36	116.36	116.36	116.36	116.36	116.36	116.36	155.15	155.15	155.15	155.15	155.15	155.15	155.15	155.15	116.36	116.36	116.36	116.36	116.36	116.36
Ground Temperature (°F)	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55
Floor Total Area (sq. ft.)	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00
Floor R-value (h-ft^2-°F/Btu)	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6
Floor Heat Load (Btu/hr)	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00
Conduction Heat Load (Btu/h)	**652**	**652**	**652**	**652**	**652**	**652**	**652**	**652**	**652**	**652**	**691**	**691**	**691**	**691**	**691**	**691**	**691**	**691**	**652**	**652**	**652**	**652**	**652**	**652**

Proposed Small Walk-in Cooler Hourly Model Load Calculation (3 of 3)

SMALL COOLER

INFILTRATION CALCULATION																								
Outside Temperature T1 (°F)	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
Outside Density ρ1 (lb/ft^3)	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073
Outside Humidity Ratio HR1(kg water/kg dry air)	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092
Outside Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85
Box Temperature T2 (°F)	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35
Box Density ρ2 (lb/ft^3)	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080
Box Humidity Ratio HR2 (kg water/kg dry air)	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034
Box Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02
Air density factor Fm	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98
Door height (ft)	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7
Door width (ft)	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
Door area (ft^2)	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21
Fully-established flow rate through open door (CFM)	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61	1179.61
Crack Infiltration (cfm)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Door openings per hour	0.00	0.00	0.00	0.00	0.00	0.00	0.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	0.00	0.00	0.00	0.00	0.00
Seconds per opening	0.00	0.00	0.00	0.00	0.00	0.00	0.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	0.00	0.00	0.00	0.00	0.00
EFOT (effective open time)(min/hr)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	0.00	0.00	0.00	0.00	0.00
η door (blockage factor of door)	1.000	1.000	1.000	1.000	1.000	1.000	1.000	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	1.000	1.000	1.000	1.000	1.000
η inf. Barrier (blockage factor of inf. Barrier)	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85
Overall air flow rate (CFM)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	17.57	17.57	17.57	17.57	17.57	17.57	17.57	17.57	17.57	17.57	17.57	17.57	0.00	0.00	0.00	0.00	0.00
Sensible (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	773.86	773.86	773.86	773.86	773.86	773.86	773.86	773.86	773.86	773.86	773.86	773.86	0.00	0.00	0.00	0.00	0.00
Latent (Btu/h, Gosney)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	501.97	501.97	501.97	501.97	501.97	501.97	501.97	501.97	501.97	501.97	501.97	501.97	0.00	0.00	0.00	0.00	0.00
Average Latent Heat of Vaporization (Btu/lbm)	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40
Latent (Btu/h, Approximation for verification)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	499.53	499.53	499.53	499.53	499.53	499.53	499.53	499.53	499.53	499.53	499.53	499.53	0.00	0.00	0.00	0.00	0.00
Infiltration Heat Load (Btu/h)	**0**	**0**	**0**	**0**	**0**	**0**	**0**	**1276**	**1276**	**1276**	**1276**	**1276**	**1276**	**1276**	**1276**	**1276**	**1276**	**1276**	**1276**	**0**	**0**	**0**	**0**	**0**
PRODUCT LOAD CALCULATION																								
Product Original Temperature	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45
Box Temperature	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35
Effective product this hour (pull-down calc)	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85
Stored product weight this hour (assumed constant)	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451
Product Specific heat above freezing (Btu/lb/F)	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
Specific heat below freezing (Btu/lb/F) (if applicable)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Heat of Respiration at box temperatre (Btu/lb/hr)	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
Percentage of Product that is packaged	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%
Pull-down Sensible Heat Load (Btu/h)	768	768	768	768	768	768	768	768	768	768	768	768	768	768	768	768	768	768	768	768	768	768	768	768
Respiration Latent & Sensible Heat Load (Btu/h)	94	94	94	94	94	94	94	94	94	94	94	94	94	94	94	94	94	94	94	94	94	94	94	94
Product Heat Load (Btu/h)	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**	**862**
MISCELLANEOUS LOAD CALCULATION																								
Lighting power (W)	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00
Lights time on (h/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.00	0.00	0.00	0.00	0.00
Lighting heat load (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	131.02	131.02	131.02	131.02	131.02	131.02	131.02	131.02	131.02	131.02	131.02	131.02	0.00	0.00	0.00	0.00	0.00
Box Temperature (°F)	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00
Occupancy (#people)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	0.00	0.00	0.00	0.00	0.00
Occupancy time (h/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.00	0.00	0.00	0.00	0.00
Occupancy heat load (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	714.00	714.00	714.00	714.00	714.00	714.00	714.00	714.00	714.00	714.00	714.00	714.00	0.00	0.00	0.00	0.00	0.00
#Vehicles	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Power per vehicle (hp)	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00
Vehicle operating time (h/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Vehicle heat load (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Misc. Heat Load (Btu/h)	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92
Miscellaneous Heat Load (Btu/h)	**546**	**546**	**546**	**546**	**546**	**546**	**546**	**1,391**	**1,391**	**1,391**	**1,391**	**1,391**	**1,391**	**1,391**	**1,391**	**1,391**	**1,391**	**1,391**	**1,391**	**546**	**546**	**546**	**546**	**546**
TOTAL HEAT LOAD (BTU/H)	**2,060**	**2,060**	**2,060**	**2,060**	**2,060**	**2,060**	**2,060**	**4,181**	**4,181**	**4,181**	**4,220**	**4,220**	**4,220**	**4,220**	**4,220**	**4,220**	**4,220**	**4,220**	**4,181**	**2,060**	**2,060**	**2,060**	**2,060**	**2,060**
BOX LOAD (HIGH, LOW)	L	L	L	L	L	L	L	H	H	H	H	H	H	H	H	H	H	H	H	L	L	L	L	L

Table 134. Proposed Large Walk-in Cooler Hourly Model Load Calculation

Proposed Large Walk-in Cooler Hourly Model Load Calculation (1 of 3)

LARGE COOLER

	AM												PM											
Hour	12:00	1:00	2:00	3:00	4:00	5:00	6:00	7:00	8:00	9:00	10:00	11:00	12:00	1:00	2:00	3:00	4:00	5:00	6:00	7:00	8:00	9:00	10:00	11:00
INPUTS AND INITIAL CALCULATIONS																								
Outdoor Site Conditions																								
Outdoor Ambient Temperature (°F)	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95
Outdoor Ambient Temperature (°C)	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0
Sunlight Loading Effect on Roof Temp (°F)	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20
Schedule - Sunlight Directly on Roof (Y/N)	N	N	N	N	N	N	N	N	N	N	Y	Y	Y	Y	Y	Y	Y	Y	N	N	N	N	N	N
Indoor Site Conditions																								
Indoor Ambient Temperature (°F)	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
Indoor Ambient RH (%)	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50
Indoor Ambient Air Specific Volume (ft3/lb)	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7
Indoor Ambient Air Humidity Ratio (kg water/kg dry air)	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01
Indoor Ambient Enthalpy (Btu/lb dry air)	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11
Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85
Box Interior Conditions																								
Interior Temperature (°C)	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7
Interior Temperature (°F)	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35
Interior RH (typical) (%)	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Interior Air Specific Volume (ft3/lb)	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5
Interior Air Humidity Ratio (kg water/kg dry air)	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003
Interior Air Enthalpy (Btu/lb dry air)	12.064	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06	12.06
Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02
Box Interior Dimensions																								
Case Width (ft)	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00
Case Length (ft)	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00
Case Height (ft)	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00
Case Floor Area (ft^2)	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500
Case Volume (ft^3)	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000
Usable Volume % for Storage * Utilization of Storage Area	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%
Usable Volume (ft^3)	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500
Floor Conditions																								
Ground Temperature (°F)	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55
Floor Heat Load (Btu/hr-ft^2)	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28	1.28
Box Construction																								
Box Location (IN, OUT)	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT
Ceiling R-value (h-ft^2-°F/Btu)	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25
Wall R-value (h-ft^2-°F/Btu)	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25
Floor R-value (h-ft^2-°F/Btu)	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15
Convection Film Coefficients																								
External Convective Film Coefficient (h-ft^2-°F/Btu)	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Internal Convective Film Coefficient (h-ft^2-°F/Btu)	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Floor Convective Film Coefficient (h-ft^2-°F/Btu)	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Infiltration Details																								
Door Height (f)	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Door Width (f)	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6
Door R-value (h-ft^2-°F/Btu)	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25
Protective Device Effectiveness (%)	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85
Door openings in this hour (#/hour)	0	0	0	0	0	0	0	13	13	13	13	13	13	13	13	13	13	13	13	0	0	0	0	0
Time per opening (s/opening)	0	0	0	0	0	0	0	30	30	30	30	30	30	30	30	30	30	30	30	0	0	0	0	0
Open Door Percentage (%)	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	0.0%	0.0%	0.0%	0.0%	0.0%
Crack Infiltration (cfm)	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Proposed Large Walk-in Cooler Hourly Model Load Calculation (2 of 3)

LARGE COOLER

Product Loading																								
Product Type	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits anc Vegetable
Product Original Temperature (°F)	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45
Product Pull-down Temperatuer (°F)	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Product Pull-down Period (hrs)	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24
Daily Loading Ratio (lb/ft^3)	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80	1.80
Schedule - Product Loading (%)	0	0	0	0	0	0	0	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	0	0	0	0	0
Product Sensible and Latent Thermal Properties																								
Product Specific heat above freezing (Btu/lb/F)	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
Specific heat below freezing (Btu/lb/F) (if applicable)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Heat of Respiration at box temperatre (Btu/lb/hr)	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
Density (lb/ft3)	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00
Percentage of Product that is packaged	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%
Lighting and Occupancy Details																								
Lighting Power (W/ft^2)	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Schedule - Lights time on during the hour (min)	0	0	0	0	0	0	0	36	36	36	36	36	36	36	36	36	36	36	36	0	0	0	0	0
Schedule - Lights time on during the hour (%)	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	0.0%	0.0%	0.0%	0.0%	0.0%
#People	0	0	0	0	0	0	0	2	2	2	2	2	2	2	2	2	2	2	2	0	0	0	0	0
Schedule - occupancy during the hour (min)	0	0	0	0	0	0	0	24	24	24	24	24	24	24	24	24	24	24	24	0	0	0	0	0
Schedule - Occupancy during the hour (%)	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	0.0%	0.0%	0.0%	0.0%	0.0%
Miscellaneous Load Details																								
#Motorized Vehicles	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
Vehicle Power (W-hr/ft^3-day)	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Vehicle Power (hp)	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Minutes operating during the hour (min)	0	0	0	0	0	0	0	33.52	33.52	33.52	33.52	33.52	33.52	33.52	33.52	33.52	33.52	33.52	33.52	0	0	0	0	0
Vehicle Power (W-hr/ft^3)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.08	0.08	0.08	0.08	0.08	0.08	0.08	0.08	0.08	0.08	0.08	0.08	0.00	0.00	0.00	0.00	0.00
Passage Door Heater (W/ft)	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8

CONDUCTION CALCULATION

Box Location (OUT, IN)	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT
Box Temperature (°F)	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35
Outdoor Ambient (°F)	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95
Indoor Ambient (°F)	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
Wall R-value (h-ft^2-°F/Btu)	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26
Walls Total Area (sq. ft.)	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00
Walls Heat Load (Btu/h)	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54	8461.54
Roof Temperature (°F)	95	95	95	95	95	95	95	95	95	95	115	115	115	115	115	115	115	115	95	95	95	95	95	95
Ceiling R-value (h-ft^2-°F/Btu)	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26	26
Ceiling Total Area (sq. ft.)	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00
Ceiling Heat Load (Btu/h)	5769.23	5769.23	5769.23	5769.23	5769.23	5769.23	5769.23	5769.23	5769.23	5769.23	7692.31	7692.31	7692.31	7692.31	7692.31	7692.31	7692.31	7692.31	5769.23	5769.23	5769.23	5769.23	5769.23	5769.23
Ground Temperature (°F)	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55
Floor Total Area (sq. ft.)	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00
Floor R-value (h-ft^2-°F/Btu)	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6	15.6
Floor Heat Load (Btu/hr)	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13	3205.13
Conduction Heat Load (Btu/h)	**17436**	**17436**	**17436**	**17436**	**17436**	**17436**	**17436**	**17436**	**17436**	**17436**	**19359**	**19359**	**19359**	**19359**	**19359**	**19359**	**19359**	**19359**	**17436**	**17436**	**17436**	**17436**	**17436**	**17436**

Proposed Large Walk-in Cooler Hourly Model Load Calculation (3 of 3)

LARGE COOLER																								
INFILTRATION CALCULATION																								
Outside Temperature T1 (°F)	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
Outside Density ρ1 (lb/ft^3)	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073
Outside Humidity Ratio HR1(kg water/kg dry air)	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092
Outside Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85
Box Temperature T2 (°F)	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35
Box Density ρ2 (lb/ft^3)	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080	0.080
Box Humidity Ratio HR2 (kg water/kg dry air)	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034	0.0034
Box Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02	12.02
Air density factor Fm	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98	0.98
Door height (ft)	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Door width (ft)	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6
Door area (ft^2)	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
Fully-established flow rate through open door (CFM)	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31	4028.31
Crack Infiltration (cfm)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Door openings per hour	0.00	0.00	0.00	0.00	0.00	0.00	0.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	0.00	0.00	0.00	0.00	0.00
Seconds per opening	0.00	0.00	0.00	0.00	0.00	0.00	0.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	0.00	0.00	0.00	0.00	0.00
EFOT (effective open time)(min/hr)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	0.00	0.00	0.00	0.00	0.00
η door (blockage factor of door)	1.000	1.000	1.000	1.000	1.000	1.000	1.000	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	1.000	1.000	1.000	1.000	1.000
η inf. Barrier (blockage factor of inf. Barrier)	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85
Overall air flow rate (CFM)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	60.01	60.01	60.01	60.01	60.01	60.01	60.01	60.01	60.01	60.01	60.01	60.01	0.00	0.00	0.00	0.00	0.00
Sensible (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	2642.69	2642.69	2642.69	2642.69	2642.69	2642.69	2642.69	2642.69	2642.69	2642.69	2642.69	2642.69	0.00	0.00	0.00	0.00	0.00
Latent (Btu/h, Gosney)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1714.18	1714.18	1714.18	1714.18	1714.18	1714.18	1714.18	1714.18	1714.18	1714.18	1714.18	1714.18	0.00	0.00	0.00	0.00	0.00
Average Latent Heat of Vaporization (Btu/lbm)	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40	1062.40
Latent (Btu/h, Approximation for verification)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1705.87	1705.87	1705.87	1705.87	1705.87	1705.87	1705.87	1705.87	1705.87	1705.87	1705.87	1705.87	0.00	0.00	0.00	0.00	0.00
Infiltration Heat Load (Btu/h)	**0**	**0**	**0**	**0**	**0**	**0**	**0**	**4357**	**4357**	**4357**	**4357**	**4357**	**4357**	**4357**	**4357**	**4357**	**4357**	**4357**	**4357**	**0**	**0**	**0**	**0**	**0**
PRODUCT LOAD CALCULATION																								
Product Original Temperature	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45	45
Box Temperature	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35
Effective product this hour (pull-down calc)	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750	3750
Stored product weight this hour (assumed constant)	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000
Product Specific heat above freezing (Btu/lb/F)	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
Specific heat below freezing (Btu/lb/F) (if applicable)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Heat of Respiration at box temperatre (Btu/lb/hr)	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
Percentage of Product that is packaged	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%	50.0%
Pull-down Sensible Heat Load (Btu/h)	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750	33750
Respiration Latent & Sensible Heat Load (Btu/h)	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188	9188
Product Heat Load (Btu/h)	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**	**42938**
MISCELLANEOUS LOAD CALCULATION																								
Lighting power (W)	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00
Lights time on (h/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.00	0.00	0.00	0.00	0.00
Lighting heat load (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	0.00	0.00	0.00	0.00	0.00
Box Temperature (°F)	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00	35.00
Occupancy (#people)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	0.00	0.00	0.00	0.00	0.00
Occupancy time (h/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.00	0.00	0.00	0.00	0.00
Occupancy heat load (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	714.00	714.00	714.00	714.00	714.00	714.00	714.00	714.00	714.00	714.00	714.00	714.00	0.00	0.00	0.00	0.00	0.00
#Vehicles	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	0.00	0.00	0.00	0.00	0.00
Power per vehicle (hp)	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00
Vehicle operating time (h/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.00	0.00	0.00	0.00	0.00
Vehicle heat load (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	0.00	0.00	0.00	0.00	0.00
Misc. Heat Load (Btu/h)	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47
Miscellaneous Heat Load (Btu/h)	**873**	**873**	**873**	**873**	**873**	**873**	**873**	**20,922**	**20,922**	**20,922**	**20,922**	**20,922**	**20,922**	**20,922**	**20,922**	**20,922**	**20,922**	**20,922**	**20,922**	**873**	**873**	**873**	**873**	**873**
TOTAL HEAT LOAD (BTU/H)	**61,247**	**61,247**	**61,247**	**61,247**	**61,247**	**61,247**	**61,247**	**85,652**	**85,652**	**85,652**	**87,575**	**87,575**	**87,575**	**87,575**	**87,575**	**87,575**	**87,575**	**87,575**	**85,652**	**61,247**	**61,247**	**61,247**	**61,247**	**61,247**
BOX LOAD (HIGH, LOW)	L	L	L	L	L	L	L	H	H	H	H	H	H	H	H	H	H	H	H	L	L	L	L	L

Table 135. Proposed Small Walk-in Freezer Hourly Model Load Calculation

Proposed Small Walk-in Freezer Hourly Model Load Calculation (1 of 3)

SMALL FREEZER	AM												PM											
Hour	12:00	1:00	2:00	3:00	4:00	5:00	6:00	7:00	8:00	9:00	10:00	11:00	12:00	1:00	2:00	3:00	4:00	5:00	6:00	7:00	8:00	9:00	10:00	11:00
INPUTS AND INITIAL CALCULATIONS																								
Outdoor Site Conditions																								
Outdoor Ambient Temperature (°F)	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95
Outdoor Ambient Temperature (°C)	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0
Sunlight Loading Effect on Roof Temp (°F)	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20
Schedule - Sunlight Directly on Roof (Y/N)	N	N	N	N	N	N	N	N	N	N	Y	Y	Y	Y	Y	Y	Y	Y	N	N	N	N	N	N
Indoor Site Conditions																								
Indoor Ambient Temperature (°F)	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
Indoor Ambient RH (%)	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50
Indoor Ambient Air Specific Volume (ft3/lb)	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7
Indoor Ambient Air Humidity Ratio (kg water/kg dry air)	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01
Indoor Ambient Enthalpy (Btu/lb dry air)	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11
Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85
Box Interior Conditions																								
Interior Temperature (°C)	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6
Interior Temperature (°F)	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5
Interior RH (typical) (%)	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Interior Air Specific Volume (ft3/lb)	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5
Interior Air Humidity Ratio (kg water/kg dry air)	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
Interior Air Enthalpy (Btu/lb dry air)	-0.691	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69
Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69
Box Interior Dimensions																								
Case Width (ft)	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00
Case Length (ft)	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00
Case Height (ft)	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00	8.00
Case Floor Area (ft^2)	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64	64
Case Volume (ft^3)	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512	512
Usable Volume % for Storage * Utilization of Storage Area	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%
Usable Volume (ft^3)	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230	230
Floor Conditions																								
Ground Temperature (°F)	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55
Floor Heat Load (Btu/hr-ft^2)	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34
Box Construction																								
Box Location (IN, OUT)	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT
Ceiling R-value (h-ft^2-°F/Btu)	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32
Wall R-value (h-ft^2-°F/Btu)	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28
Floor R-value (h-ft^2-°F/Btu)	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25
Convection Film Coefficients																								
External Convective Film Coefficient (h-ft^2-°F/Btu)	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Internal Convective Film Coefficient (h-ft^2-°F/Btu)	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Floor Convective Film Coefficient (h-ft^2-°F/Btu)	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Infiltration Details																								
Door Height (f)	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7
Door Width (f)	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
Door R-value (h-ft^2-°F/Btu)	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28
Protective Device Effectiveness (%)	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85
Door openings in this hour (#/hour)	0	0	0	0	0	0	0	13	13	13	13	13	13	13	13	13	13	13	13	0	0	0	0	0
Time per opening (s/opening)	0	0	0	0	0	0	0	30	30	30	30	30	30	30	30	30	30	30	30	0	0	0	0	0
Open Door Percentage (%)	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	0.0%	0.0%	0.0%	0.0%	0.0%
Crack Infiltration (cfm)	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Proposed Small Walk-in Freezer Hourly Model Load Calculation (2 of 3)

SMALL FREEZER

Product Loading																								
Product Type	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits anc Vegetable
Product Original Temperature (°F)	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5
Product Pull-down Temperatuer (°F)	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Product Pull-down Period (hrs)	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24	24
Daily Loading Ratio (lb/ft^3)	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Schedule - Product Loading (%)	0	0	0	0	0	0	0	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33	0	0	0	0	0
Product Sensible and Latent Thermal Properties																								
Product Specific heat above freezing (Btu/lb/F)	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
Specific heat below freezing (Btu/lb/F) (if applicable)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Heat of Respiration at box temperatre (Btu/lb/hr)	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
Density (lb/ft3)	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00
Percentage of Product that is packaged	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%
Lighting and Occupancy Details																								
Lighting Power (W/ft^2)	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Schedule - Lights time on during the hour (min)	0	0	0	0	0	0	0	36	36	36	36	36	36	36	36	36	36	36	36	0	0	0	0	0
Schedule - Lights time on during the hour (%)	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	0.0%	0.0%	0.0%	0.0%	0.0%
#People	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
Schedule - occupancy during the hour (min)	0	0	0	0	0	0	0	24	24	24	24	24	24	24	24	24	24	24	24	0	0	0	0	0
Schedule - Occupancy during the hour (%)	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	0.0%	0.0%	0.0%	0.0%	0.0%
Miscellaneous Load Details																								
#Motorized Vehicles	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Vehicle Power (W-hr/ft^3-day)	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Vehicle Power (hp)	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Minutes operating during the hour (min)	0	0	0	0	0	0	0	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0	0	0	0	0
Vehicle Power (W-hr/ft^3)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Passage Door Heater (W/ft)	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8

CONDUCTION CALCULATION																								
Box Location (OUT, IN)	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT
Box Temperature (°F)	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5
Outdoor Ambient (°F)	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95
Indoor Ambient (°F)	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
Wall R-value (h-ft^2-°F/Btu)	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29	29
Walls Total Area (sq. ft.)	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00	256.00
Walls Heat Load (Btu/h)	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62	838.62
Roof Temperature (°F)	95	95	95	95	95	95	95	95	95	95	115	115	115	115	115	115	115	115	95	95	95	95	95	95
Ceiling R-value (h-ft^2-°F/Btu)	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33	33
Ceiling Total Area (sq. ft.)	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00
Ceiling Heat Load (Btu/h)	193.94	193.94	193.94	193.94	193.94	193.94	193.94	193.94	193.94	193.94	232.73	232.73	232.73	232.73	232.73	232.73	232.73	232.73	193.94	193.94	193.94	193.94	193.94	193.94
Ground Temperature (°F)	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55
Floor Total Area (sq. ft.)	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00
Floor R-value (h-ft^2-°F/Btu)	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6
Floor Heat Load (Btu/hr)	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00	150.00
Conduction Heat Load (Btu/h)	**1183**	**1183**	**1183**	**1183**	**1183**	**1183**	**1183**	**1183**	**1183**	**1183**	**1221**	**1221**	**1221**	**1221**	**1221**	**1221**	**1221**	**1221**	**1183**	**1183**	**1183**	**1183**	**1183**	**1183**

Proposed Small Walk-in Freezer Hourly Model Load Calculation (3 of 3)

SMALL FREEZER

INFILTRATION CALCULATION

Outside Temperature T1 (°F)	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
Outside Density ρ1 (lb/ft^3)	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073
Outside Humidity Ratio HR1(kg water/kg dry air)	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092
Outside Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85
Box Temperature T2 (°F)	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5
Box Density ρ2 (lb/ft^3)	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087
Box Humidity Ratio HR2 (kg water/kg dry air)	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005
Box Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69
Air density factor Fm	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96
Door height (ft)	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7	7
Door width (ft)	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
Door area (ft^2)	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21
Fully-established flow rate through open door (CFM)	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12	1605.12
Crack Infiltration (cfm)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Door openings per hour	0.00	0.00	0.00	0.00	0.00	0.00	0.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	0.00	0.00	0.00	0.00	0.00
Seconds per opening	0.00	0.00	0.00	0.00	0.00	0.00	0.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	0.00	0.00	0.00	0.00	0.00
EFOT (effective open time)(min/hr)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	0.00	0.00	0.00	0.00	0.00
η door (blockage factor of door)	1.000	1.000	1.000	1.000	1.000	1.000	1.000	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	1.000	1.000	1.000	1.000	1.000
η inf. Barrier (blockage factor of inf. Barrier)	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85
Overall air flow rate (CFM)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	23.91	23.91	23.91	23.91	23.91	23.91	23.91	23.91	23.91	23.91	23.91	23.91	0.00	0.00	0.00	0.00	0.00
Sensible (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	2208.25	2208.25	2208.25	2208.25	2208.25	2208.25	2208.25	2208.25	2208.25	2208.25	2208.25	2208.25	0.00	0.00	0.00	0.00	0.00
Latent (Btu/h, Gosney)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1074.31	1074.31	1074.31	1074.31	1074.31	1074.31	1074.31	1074.31	1074.31	1074.31	1074.31	1074.31	0.00	0.00	0.00	0.00	0.00
Average Latent Heat of Vaporization (Btu/lbm)	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70
Latent (Btu/h, Approximation for verification)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1081.23	1081.23	1081.23	1081.23	1081.23	1081.23	1081.23	1081.23	1081.23	1081.23	1081.23	1081.23	0.00	0.00	0.00	0.00	0.00
Infiltration Heat Load (Btu/h)	**0**	**0**	**0**	**0**	**0**	**0**	**0**	**3283**	**3283**	**3283**	**3283**	**3283**	**3283**	**3283**	**3283**	**3283**	**3283**	**3283**	**3283**	**0**	**0**	**0**	**0**	**0**

PRODUCT LOAD CALCULATION

Product Original Temperature	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5
Box Temperature	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5
Effective product this hour (pull-down calc)	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21	21
Stored product weight this hour (assumed constant)	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451	6451
Product Specific heat above freezing (Btu/lb/F)	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
Specific heat below freezing (Btu/lb/F) (if applicable)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Heat of Respiration at box tempertatre (Btu/lb/hr)	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
Percentage of Product that is packaged	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%
Pull-down Sensible Heat Load (Btu/h)	107	107	107	107	107	107	107	107	107	107	107	107	107	107	107	107	107	107	107	107	107	107	107	107
Respiration Latent & Sensible Heat Load (Btu/h)	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Product Heat Load (Btu/h)	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**	**107**

MISCELLANEOUS LOAD CALCULATION

Lighting power (W)	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00	64.00
Lights time on (h/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.00	0.00	0.00	0.00	0.00
Lighting heat load (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	131.02	131.02	131.02	131.02	131.02	131.02	131.02	131.02	131.02	131.02	131.02	131.02	0.00	0.00	0.00	0.00	0.00
Box Temperature (°F)	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00
Occupancy (#people)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	0.00	0.00	0.00	0.00	0.00
Occupancy time (h/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.00	0.00	0.00	0.00	0.00
Occupancy heat load (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	541.00	541.00	541.00	541.00	541.00	541.00	541.00	541.00	541.00	541.00	541.00	541.00	0.00	0.00	0.00	0.00	0.00
#Vehicles	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Power per vehicle (hp)	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00
Vehicle operating time (h/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Vehicle heat load (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Misc. Heat Load (Btu/h)	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92	545.92
Miscellaneous Heat Load (Btu/h)	**546**	**546**	**546**	**546**	**546**	**546**	**546**	**1,218**	**1,218**	**1,218**	**1,218**	**1,218**	**1,218**	**1,218**	**1,218**	**1,218**	**1,218**	**1,218**	**1,218**	**546**	**546**	**546**	**546**	**546**

TOTAL HEAT LOAD (BTU/H)	**1,835**	**1,835**	**1,835**	**1,835**	**1,835**	**1,835**	**1,835**	**5,790**	**5,790**	**5,790**	**5,829**	**5,829**	**5,829**	**5,829**	**5,829**	**5,829**	**5,829**	**5,829**	**5,790**	**1,835**	**1,835**	**1,835**	**1,835**	**1,835**
BOX LOAD (HIGH, LOW)	L	L	L	L	L	L	L	H	H	H	H	H	H	H	H	H	H	H	H	L	L	L	L	L

Table 136. Proposed Large Walk-in Freezer Hourly Model Load Calculation

Proposed Large Walk-in Freezer Hourly Model Load Calculation (1 of 3)

LARGE FREEZER

	AM												PM											
Hour	12:00	1:00	2:00	3:00	4:00	5:00	6:00	7:00	8:00	9:00	10:00	11:00	12:00	1:00	2:00	3:00	4:00	5:00	6:00	7:00	8:00	9:00	10:00	11:00
INPUTS AND INITIAL CALCULATIONS																								
Outdoor Site Conditions																								
Outdoor Ambient Temperature (°F)	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35
Outdoor Ambient Temperature (°C)	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0
Sunlight Loading Effect on Roof Temp (°F)	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20	20
Schedule - Sunlight Directly on Roof (Y/N)	N	N	N	N	N	N	N	N	N	N	Y	Y	Y	Y	Y	Y	Y	Y	N	N	N	N	N	N
Indoor Site Conditions																								
Indoor Ambient Temperature (°F)	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
Indoor Ambient RH (%)	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50	50
Indoor Ambient Air Specific Volume (ft3/lb)	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7	13.7
Indoor Ambient Air Humidity Ratio (kg water/kg dry air)	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01	0.01
Indoor Ambient Enthalpy (Btu/lb dry air)	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11	28.11
Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85
Box Interior Conditions																								
Interior Temperature (°C)	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6	-20.6
Interior Temperature (°F)	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5
Interior RH (typical) (%)	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80	80
Interior Air Specific Volume (ft3/lb)	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5
Interior Air Humidity Ratio (kg water/kg dry air)	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
Interior Air Enthalpy (Btu/lb dry air)	-0.691	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69
Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69
Box Interior Dimensions																								
Case Width (ft)	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00
Case Length (ft)	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00	50.00
Case Height (ft)	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00	20.00
Case Floor Area (ft^2)	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500	2500
Case Volume (ft^3)	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000	50000
Usable Volume % for Storage * Utilization of Storage Area	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%	45.0%
Usable Volume (ft^3)	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500	22500
Floor Conditions																								
Ground Temperature (°F)	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55
Floor Heat Load (Btu/hr-ft^2)	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34	2.34
Box Construction																								
Box Location (IN, OUT)	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT
Ceiling R-value (h-ft^2-°F/Btu)	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32	32
Wall R-value (h-ft^2-°F/Btu)	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28
Floor R-value (h-ft^2-°F/Btu)	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25
Convection Film Coefficients																								
External Convective Film Coefficient (h-ft^2-°F/Btu)	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Internal Convective Film Coefficient (h-ft^2-°F/Btu)	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Floor Convective Film Coefficient (h-ft^2-°F/Btu)	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Infiltration Details																								
Door Height (f)	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Door Width (f)	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6
Door R-value (h-ft^2-°F/Btu)	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28
Protective Device Effectiveness (%)	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85	85
Door openings in this hour (#/hour)	0	0	0	0	0	0	0	13	13	13	13	13	13	13	13	13	13	13	13	0	0	0	0	0
Time per opening (s/opening)	0	0	0	0	0	0	0	30	30	30	30	30	30	30	30	30	30	30	30	0	0	0	0	0
Open Door Percentage (%)	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	10.8%	0.0%	0.0%	0.0%	0.0%	0.0%
Crack Infiltration (cfm)	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Proposed Large Walk-in Freezer Hourly Model Load Calculation (2 of 3)

Product Loading								
Product Type	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables
Product Original Temperature (˚F)	5	5	5	5	5	5	5	5
Product Pull-down Temperatuer (˚F)	10	10	10	10	10	10	10	10
Product Pull-down Period (hrs)	24	24	24	24	24	24	24	24
Daily Loading Ratio (lb/ft^3)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Schedule - Product Loading (%)	0	0	0	0	0	0	0	8.33
Product Sensible and Latent Thermal Properties								
Product Specific heat above freezing (Btu/lb/F)	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
Specific heat below freezing (Btu/lb/F) (if applicable)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Heat of Respiration at box temperatre (Btu/lb/hr)	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
Density (lb/ft3)	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00
Percentage of Product that is packaged	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%
Lighting and Occupancy Details								
Lighting Power (W/ft^2)	1	1	1	1	1	1	1	1
Schedule - Lights time on during the hour (min)	0	0	0	0	0	0	0	36
Schedule - Lights time on during the hour (%)	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	60.0%
#People	0	0	0	0	0	0	0	2
Schedule - occupancy during the hour (min)	0	0	0	0	0	0	0	24
Schedule - Occupancy during the hour (%)	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	0.0%	40.0%
Miscellaneous Load Details								
#Motorized Vehicles	0	0	0	0	0	0	0	1
Vehicle Power (W-hr/ft^3-day)	1	1	1	1	1	1	1	1
Vehicle Power (hp)	10	10	10	10	10	10	10	10
Minutes operating during the hour (min)	0	0	0	0	0	0	0	33.52
Vehicle Power (W-hr/ft^3)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.08
Passage Door Heater (W/ft)	8	8	8	8	8	8	8	8
CONDUCTION CALCULATION								
Box Location (OUT, IN)	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT
Box Temperature (˚F)	-5	-5	-5	-5	-5	-5	-5	-5
Outdoor Ambient (˚F)	95	95	95	95	95	95	95	95
Indoor Ambient (˚F)	75	75	75	75	75	75	75	75
Wall R-value (h-ft^2-˚F/Btu)	29	29	29	29	29	29	29	29
Walls Total Area (sq. ft.)	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00
Walls Heat Load (Btu/h)	13103.45	13103.45	13103.45	13103.45	13103.45	13103.45	13103.45	13103.45
Roof Temperature (˚F)	95	95	95	95	95	95	95	95
Ceiling R-value (h-ft^2-˚F/Btu)	33	33	33	33	33	33	33	33
Ceiling Total Area (sq. ft.)	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00
Ceiling Heat Load (Btu/h)	7575.76	7575.76	7575.76	7575.76	7575.76	7575.76	7575.76	7575.76
Ground Temperature (˚F)	55	55	55	55	55	55	55	55
Floor Total Area (sq. ft.)	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00
Floor R-value (h-ft^2-˚F/Btu)	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6
Floor Heat Load (Btu/hr)	5859.38	5859.38	5859.38	5859.38	5859.38	5859.38	5859.38	5859.38
Conduction Heat Load (Btu/h)	**26539**	**26539**	**26539**	**26539**	**26539**	**26539**	**26539**	**26539**

Product Loading								
Product Type	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables
Product Original Temperature (˚F)	5	5	5	5	5	5	5	5
Product Pull-down Temperatuer (˚F)	10	10	10	10	10	10	10	10
Product Pull-down Period (hrs)	24	24	24	24	24	24	24	24
Daily Loading Ratio (lb/ft^3)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Schedule - Product Loading (%)	8.33	8.33	8.33	8.33	8.33	8.33	8.33	8.33
Product Sensible and Latent Thermal Properties								
Product Specific heat above freezing (Btu/lb/F)	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
Specific heat below freezing (Btu/lb/F) (if applicable)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Heat of Respiration at box temperatre (Btu/lb/hr)	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
Density (lb/ft3)	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00
Percentage of Product that is packaged	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%
Lighting and Occupancy Details								
Lighting Power (W/ft^2)	1	1	1	1	1	1	1	1
Schedule - Lights time on during the hour (min)	36	36	36	36	36	36	36	36
Schedule - Lights time on during the hour (%)	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%	60.0%
#People	2	2	2	2	2	2	2	2
Schedule - occupancy during the hour (min)	24	24	24	24	24	24	24	24
Schedule - Occupancy during the hour (%)	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%	40.0%
Miscellaneous Load Details								
#Motorized Vehicles	1	1	1	1	1	1	1	1
Vehicle Power (W-hr/ft^3-day)	1	1	1	1	1	1	1	1
Vehicle Power (hp)	10	10	10	10	10	10	10	10
Minutes operating during the hour (min)	33.52	33.52	33.52	33.52	33.52	33.52	33.52	33.52
Vehicle Power (W-hr/ft^3)	0.08	0.08	0.08	0.08	0.08	0.08	0.08	0.08
Passage Door Heater (W/ft)	8	8	8	8	8	8	8	8
CONDUCTION CALCULATION								
Box Location (OUT, IN)	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT
Box Temperature (˚F)	-5	-5	-5	-5	-5	-5	-5	-5
Outdoor Ambient (˚F)	95	95	95	95	95	95	95	95
Indoor Ambient (˚F)	75	75	75	75	75	75	75	75
Wall R-value (h-ft^2-˚F/Btu)	29	29	29	29	29	29	29	29
Walls Total Area (sq. ft.)	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00
Walls Heat Load (Btu/h)	13103.45	13103.45	13103.45	13103.45	13103.45	13103.45	13103.45	13103.45
Roof Temperature (˚F)	95	95	115	115	115	115	115	115
Ceiling R-value (h-ft^2-˚F/Btu)	33	33	33	33	33	33	33	33
Ceiling Total Area (sq. ft.)	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00
Ceiling Heat Load (Btu/h)	7575.76	7575.76	9090.91	9090.91	9090.91	9090.91	9090.91	9090.91
Ground Temperature (˚F)	55	55	55	55	55	55	55	55
Floor Total Area (sq. ft.)	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00
Floor R-value (h-ft^2-˚F/Btu)	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6
Floor Heat Load (Btu/hr)	5859.38	5859.38	5859.38	5859.38	5859.38	5859.38	5859.38	5859.38
Conduction Heat Load (Btu/h)	**26539**	**26539**	**28054**	**28054**	**28054**	**28054**	**28054**	**28054**

Product Loading								
Product Type	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits and Vegetables	Fruits anc Vegetable
Product Original Temperature (˚F)	5	5	5	5	5	5	5	5
Product Pull-down Temperatuer (˚F)	10	10	10	10	10	10	10	10
Product Pull-down Period (hrs)	24	24	24	24	24	24	24	24
Daily Loading Ratio (lb/ft^3)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Schedule - Product Loading (%)	8.33	8.33	8.33	0	0	0	0	0
Product Sensible and Latent Thermal Properties								
Product Specific heat above freezing (Btu/lb/F)	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
Specific heat below freezing (Btu/lb/F) (if applicable)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Heat of Respiration at box temperatre (Btu/lb/hr)	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
Density (lb/ft3)	28.00	28.00	28.00	28.00	28.00	28.00	28.00	28.00
Percentage of Product that is packaged	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%
Lighting and Occupancy Details								
Lighting Power (W/ft^2)	1	1	1	1	1	1	1	1
Schedule - Lights time on during the hour (min)	36	36	36	0	0	0	0	0
Schedule - Lights time on during the hour (%)	60.0%	60.0%	60.0%	0.0%	0.0%	0.0%	0.0%	0.0%
#People	2	2	2	0	0	0	0	0
Schedule - occupancy during the hour (min)	24	24	24	0	0	0	0	0
Schedule - Occupancy during the hour (%)	40.0%	40.0%	40.0%	0.0%	0.0%	0.0%	0.0%	0.0%
Miscellaneous Load Details								
#Motorized Vehicles	1	1	1	0	0	0	0	0
Vehicle Power (W-hr/ft^3-day)	1	1	1	1	1	1	1	1
Vehicle Power (hp)	10	10	10	10	10	10	10	10
Minutes operating during the hour (min)	33.52	33.52	33.52	0	0	0	0	0
Vehicle Power (W-hr/ft^3)	0.08	0.08	0.08	0.00	0.00	0.00	0.00	0.00
Passage Door Heater (W/ft)	8	8	8	8	8	8	8	8
CONDUCTION CALCULATION								
Box Location (OUT, IN)	OUT	OUT	OUT	OUT	OUT	OUT	OUT	OUT
Box Temperature (˚F)	-5	-5	-5	-5	-5	-5	-5	-5
Outdoor Ambient (˚F)	95	95	95	95	95	95	95	95
Indoor Ambient (˚F)	75	75	75	75	75	75	75	75
Wall R-value (h-ft^2-˚F/Btu)	29	29	29	29	29	29	29	29
Walls Total Area (sq. ft.)	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00	4000.00
Walls Heat Load (Btu/h)	13103.45	13103.45	13103.45	13103.45	13103.45	13103.45	13103.45	13103.45
Roof Temperature (˚F)	115	115	95	95	95	95	95	95
Ceiling R-value (h-ft^2-˚F/Btu)	33	33	33	33	33	33	33	33
Ceiling Total Area (sq. ft.)	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00
Ceiling Heat Load (Btu/h)	9090.91	9090.91	7575.76	7575.76	7575.76	7575.76	7575.76	7575.76
Ground Temperature (˚F)	55	55	55	55	55	55	55	55
Floor Total Area (sq. ft.)	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00
Floor R-value (h-ft^2-˚F/Btu)	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6
Floor Heat Load (Btu/hr)	5859.38	5859.38	5859.38	5859.38	5859.38	5859.38	5859.38	5859.38
Conduction Heat Load (Btu/h)	**28054**	**28054**	**26539**	**26539**	**26539**	**26539**	**26539**	**26539**

Proposed Large Walk-in Freezer Hourly Model Load Calculation (3 of 3)

LARGE FREEZER

INFILTRATION CALCULATION																								
Outside Temperature T1 (°F)	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75	75
Outside Density ρ1 (lb/ft^3)	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073	0.073
Outside Humidity Ratio HR1(kg water/kg dry air)	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092	0.0092
Outside Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85	27.85
Box Temperature T2 (°F)	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5
Box Density ρ2 (lb/ft^3)	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087	0.087
Box Humidity Ratio HR2 (kg water/kg dry air)	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005	0.0005
Box Specific Enthalpy of Air-Vapor Mixture (Btu/lb mix)	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69	-0.69
Air density factor Fm	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96	0.96
Door height (ft)	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Door width (ft)	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6
Door area (ft^2)	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
Fully-established flow rate through open door (CFM)	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38	5481.38
Crack Infiltration (cfm)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
Door openings per hour	0.00	0.00	0.00	0.00	0.00	0.00	0.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	13.00	0.00	0.00	0.00	0.00	0.00
Seconds per opening	0.00	0.00	0.00	0.00	0.00	0.00	0.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	0.00	0.00	0.00	0.00	0.00
EFOT (effective open time)(min/hr)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	5.96	0.00	0.00	0.00	0.00	0.00
η door (blockage factor of door)	1.000	1.000	1.000	1.000	1.000	1.000	1.000	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	0.901	1.000	1.000	1.000	1.000	1.000
η inf. Barrier (blockage factor of inf. Barrier)	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85
Overall air flow rate (CFM)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	81.65	81.65	81.65	81.65	81.65	81.65	81.65	81.65	81.65	81.65	81.65	81.65	0.00	0.00	0.00	0.00	0.00
Sensible (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	7541.02	7541.02	7541.02	7541.02	7541.02	7541.02	7541.02	7541.02	7541.02	7541.02	7541.02	7541.02	0.00	0.00	0.00	0.00	0.00
Latent (Btu/h, Gosney)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	3668.71	3668.71	3668.71	3668.71	3668.71	3668.71	3668.71	3668.71	3668.71	3668.71	3668.71	3668.71	0.00	0.00	0.00	0.00	0.00
Average Latent Heat of Vaporization (Btu/lbm)	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70	1073.70
Latent (Btu/h, Approximation for verification)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	3692.32	3692.32	3692.32	3692.32	3692.32	3692.32	3692.32	3692.32	3692.32	3692.32	3692.32	3692.32	0.00	0.00	0.00	0.00	0.00
Infiltration Heat Load (Btu/h)	**0**	**0**	**0**	**0**	**0**	**0**	**0**	**11210**	**11210**	**11210**	**11210**	**11210**	**11210**	**11210**	**11210**	**11210**	**11210**	**11210**	**11210**	**0**	**0**	**0**	**0**	**0**
PRODUCT LOAD CALCULATION																								
Product Original Temperature	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5
Box Temperature	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5	-5
Effective product this hour (pull-down calc)	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042	1042
Stored product weight this hour (assumed constant)	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000	630000
Product Specific heat above freezing (Btu/lb/F)	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
Specific heat below freezing (Btu/lb/F) (if applicable)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Heat of Respiration at box temperatre (Btu/lb/hr)	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
Percentage of Product that is packaged	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%	100.0%
Pull-down Sensible Heat Load (Btu/h)	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208	5208
Respiration Latent & Sensible Heat Load (Btu/h)	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Product Heat Load (Btu/h)	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**	**5208**
MISCELLANEOUS LOAD CALCULATION																								
Lighting power (W)	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00	2500.00
Lights time on (h/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.00	0.00	0.00	0.00	0.00
Lighting heat load (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	5118.00	0.00	0.00	0.00	0.00	0.00
Box Temperature (°F)	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00	-5.00
Occupancy (#people)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	0.00	0.00	0.00	0.00	0.00
Occupancy time (h/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.00	0.00	0.00	0.00	0.00
Occupancy heat load (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1082.00	1082.00	1082.00	1082.00	1082.00	1082.00	1082.00	1082.00	1082.00	1082.00	1082.00	1082.00	0.00	0.00	0.00	0.00	0.00
#Vehicles	0.00	0.00	0.00	0.00	0.00	0.00	0.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	0.00	0.00	0.00	0.00	0.00
Power per vehicle (hp)	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00
Vehicle operating time (h/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.00	0.00	0.00	0.00	0.00
Vehicle heat load (Btu/h)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	14216.67	0.00	0.00	0.00	0.00	0.00
Misc. Heat Load (Btu/h)	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47	873.47
Miscellaneous Heat Load (Btu/h)	**873**	**873**	**873**	**873**	**873**	**873**	**873**	**21,290**	**21,290**	**21,290**	**21,290**	**21,290**	**21,290**	**21,290**	**21,290**	**21,290**	**21,290**	**21,290**	**21,290**	**873**	**873**	**873**	**873**	**873**
TOTAL HEAT LOAD (BTU/H)	**32,620**	**32,620**	**32,620**	**32,620**	**32,620**	**32,620**	**32,620**	**64,247**	**64,247**	**64,247**	**65,762**	**65,762**	**65,762**	**65,762**	**65,762**	**65,762**	**65,762**	**65,762**	**64,247**	**32,620**	**32,620**	**32,620**	**32,620**	**32,620**
BOX LOAD (HIGH, LOW)	L	L	L	L	L	L	L	H	H	H	H	H	H	H	H	H	H	H	H	L	L	L	L	L

From the hourly calculations, average, peak, and min heat loads are compared in Table 137, Table 138, Table 139, Table 140, Table 141, and Table 142 on a heat load and percentage of total heat load basis. Analysis of this information may identify the need for future alterations to the model load. Average heat loads (Table 137 and Table 138) show coolers having fairly similar percentages of conduction loads, with conduction loading being the primary mode of heat loading on the large freezer. Analyzing infiltration loading for the large refrigerated spaces on a percentage basis depicts a very small infiltration load, which leads this author to question whether the infiltration schedule is too small. Comparing infiltration loads in Table 137, however, shows that the large space infiltration loads are about three times larger than the loads for the smaller spaces. Product heat load contribution to the freezers is smaller, on a percentage and heat load basis, as the pull-down portion of the heat load is the same while the coolers have the addition of a respiration heat load. Miscellaneous loads are fairly consistent on a percentage basis for each refrigerated space even though each space has different load components and magnitudes.

Peak loads (Table 139 and Table 140) reveal which loads vary the most throughout the day. The peak load percentage of total load increases for infiltration and miscellaneous loads, as they vary the most through the day, relative to the conduction and product heat loads which are more consistent. This load variation is also evident through analysis of the minimum heat loads seen by the four refrigerated spaces (Table 141 and Table 142).

Table 137. Average Heat Loads for the Proposed Model Walk-ins

	Small Cooler	**Small Freezer**	**Large Cooler**	**Large Freezer**
Walls Heat Load (Btu/h)	485.52	838.62	8461.54	13103.45
Ceiling Heat Load (Btu/h)	129.29	206.87	6410.26	8080.81
Floor Heat Load (Btu/hr)	50.00	150.00	3205.13	5859.38
Total Conduction Heat Load (Btu/h)	**664.81**	**1195.49**	**18076.92**	**27043.63**
Sensible (Btu/h)	386.93	1104.12	1321.35	3770.51
Latent (Btu/h)	250.98	537.16	857.09	1834.35
Total Infiltration Heat Load (Btu/h)	**637.91**	**1641.28**	**2178.44**	**5604.86**
Pull-down Sensible Heat Load (Btu/h)	768.00	106.67	33750.00	5208.33
Respiration Heat Load (Btu/h)	94.08	0.00	9187.50	0.00
Total Product Heat Load (Btu/h)	**862.08**	**106.67**	**42937.50**	**5208.33**
Lighting heat load (Btu/h)	65.51	65.51	2559.00	2559.00
Occupancy heat load (Btu/h)	357.00	270.50	357.00	541.00
Vehicle heat load (Btu/h)	0.00	0.00	7108.33	7108.33
Misc. Heat Load (Btu/h)	545.92	545.92	873.47	873.47
Total Miscellaneous Heat Load (Btu/h)	**968.43**	**881.93**	**10897.81**	**11081.81**
Total Heat Load (Btu/h)	**3133.23**	**3825.36**	**74090.67**	**48938.63**

Table 138. Average Heat Load Percentages for the Proposed Model Walk-ins

	Small Cooler	**Small Freezer**	**Large Cooler**	**Large Freezer**
Walls Heat Load (% of Total Load)	15.5%	21.9%	11.4%	26.8%
Ceiling Heat Load (% of Total Load)	4.1%	5.4%	8.7%	16.5%
Floor Heat Load (% of Total Load)	1.6%	3.9%	4.3%	12.0%
Total Conduction Heat Load (% of Total)	**21.2%**	**31.3%**	**24.4%**	**55.3%**
Sensible (% of Total Load)	12.3%	28.9%	1.8%	7.7%
Latent (% of Total Load)	8.0%	14.0%	1.2%	3.7%
Total Infiltration Heat Load (% of Total)	**20.4%**	**42.9%**	**2.9%**	**11.5%**
Pull-down Sensible Heat Load (% of Total)	24.5%	2.8%	45.6%	10.6%
Respiration Heat Load (% of Total Load)	3.0%	0.0%	12.4%	0.0%
Total Product Heat Load (% of Total)	**27.5%**	**2.8%**	**58.0%**	**10.6%**
Lighting heat load (% of Total Load)	2.1%	1.7%	3.5%	5.2%
Occupancy heat load (% of Total Load)	11.4%	7.1%	0.5%	1.1%
Vehicle heat load (% of Total Load)	0.0%	0.0%	9.6%	14.5%
Misc. Heat Load (% of Total Load)	17.4%	14.3%	1.2%	1.8%
Total Miscellaneous Load (% of Total)	**30.9%**	**23.1%**	**14.7%**	**22.6%**

Table 139. Peak Heat Loads for the Proposed Model Walk-ins

	Small Cooler	**Small Freezer**	**Large Cooler**	**Large Freezer**
Walls Heat Load (Btu/h)	485.52	838.62	8461.54	13103.45
Ceiling Heat Load (Btu/h)	155.15	232.73	7692.31	9090.91
Floor Heat Load (Btu/hr)	50.00	150.00	3205.13	5859.38
Total Conduction Heat Load (Btu/h)	**690.67**	**1221.35**	**19358.97**	**28053.73**
Sensible (Btu/h)	773.86	2208.25	2642.69	7541.02
Latent (Btu/h)	501.97	1074.31	1714.18	3668.71
Total Infiltration Heat Load (Btu/h)	**1275.83**	**3282.56**	**4356.88**	**11209.73**
Pull-down Sensible Heat Load (Btu/h)	768.00	106.67	33750.00	5208.33
Respiration Heat Load (Btu/h)	94.08	0.00	9187.50	0.00
Total Product Heat Load (Btu/h)	**862.08**	**106.67**	**42937.50**	**5208.33**
Lighting heat load (Btu/h)	131.02	131.02	5118.00	5118.00
Occupancy heat load (Btu/h)	714.00	541.00	714.00	1082.00
Vehicle heat load (Btu/h)	0.00	0.00	14216.67	14216.67
Misc. Heat Load (Btu/h)	545.92	545.92	873.47	873.47
Total Miscellaneous Heat Load (Btu/h)	**1390.94**	**1217.94**	**20922.14**	**21290.14**
Total Heat Load (Btu/h)	**4219.52**	**5828.51**	**87575.49**	**65761.93**

Table 140. Peak Heat Load Percentages for the Proposed Model Walk-ins

	Small Cooler	Small Freezer	Large Cooler	Large Freezer
Walls Heat Load (% of Total Load)	11.5%	14.4%	9.7%	19.9%
Ceiling Heat Load (% of Total Load)	3.7%	4.0%	8.8%	13.8%
Floor Heat Load (% of Total Load)	1.2%	2.6%	3.7%	8.9%
Total Conduction Heat Load (% of Total)	**16.4%**	**21.0%**	**22.1%**	**42.7%**
Sensible (% of Total Load)	18.3%	37.9%	3.0%	11.5%
Latent (% of Total Load)	11.9%	18.4%	2.0%	5.6%
Total Infiltration Heat Load (% of Total)	**30.2%**	**56.3%**	**5.0%**	**17.0%**
Pull-down Sensible Heat Load (% of Total)	18.2%	1.8%	38.5%	7.9%
Respiration Heat Load (% of Total Load)	2.2%	0.0%	10.5%	0.0%
Total Product Heat Load (% of Total)	**20.4%**	**1.8%**	**49.0%**	**7.9%**
Lighting heat load (% of Total Load)	3.1%	2.2%	5.8%	7.8%
Occupancy heat load (% of Total Load)	16.9%	9.3%	0.8%	1.6%
Vehicle heat load (% of Total Load)	0.0%	0.0%	16.2%	21.6%
Misc. Heat Load (% of Total Load)	12.9%	9.4%	1.0%	1.3%
Total Miscellaneous Load (% of Total)	**33.0%**	**20.9%**	**23.9%**	**32.4%**

Table 141. Minimum Heat Loads for the Proposed Model Walk-ins

	Small Cooler	**Small Freezer**	**Large Cooler**	**Large Freezer**
Walls Heat Load (Btu/h)	485.52	838.62	8461.54	13103.45
Ceiling Heat Load (Btu/h)	116.36	193.94	5769.23	7575.76
Floor Heat Load (Btu/hr)	50.00	150.00	3205.13	5859.38
Total Conduction Heat Load (Btu/h)	**651.88**	**1182.56**	**17435.90**	**26538.58**
Sensible (Btu/h)	0.00	0.00	0.00	0.00
Latent (Btu/h)	0.00	0.00	0.00	0.00
Total Infiltration Heat Load (Btu/h)	**0.00**	**0.00**	**0.00**	**0.00**
Pull-down Sensible Heat Load (Btu/h)	768.00	106.67	33750.00	5208.33
Respiration Heat Load (Btu/h)	94.08	0.00	9187.50	0.00
Total Product Heat Load (Btu/h)	**862.08**	**106.67**	**42937.50**	**5208.33**
Lighting heat load (Btu/h)	0.00	0.00	0.00	0.00
Occupancy heat load (Btu/h)	0.00	0.00	0.00	0.00
Vehicle heat load (Btu/h)	0.00	0.00	0.00	0.00
Misc. Heat Load (Btu/h)	545.92	545.92	873.47	873.47
Total Miscellaneous Heat Load (Btu/h)	**545.92**	**545.92**	**873.47**	**873.47**
Total Heat Load (Btu/h)	**2059.88**	**1835.15**	**61246.87**	**32620.39**

Table 142. Minimum Heat Load Percentages for the Proposed Model Walk-ins

	Small Cooler	Small Freezer	Large Cooler	Large Freezer
Walls Heat Load (% of Total Load)	23.6%	45.7%	13.8%	40.2%
Ceiling Heat Load (% of Total Load)	5.6%	10.6%	9.4%	23.2%
Floor Heat Load (% of Total Load)	2.4%	8.2%	5.2%	18.0%
Total Conduction Heat Load (% of Total)	**31.6%**	**64.4%**	**28.5%**	**81.4%**
Sensible (% of Total Load)	0.0%	0.0%	0.0%	0.0%
Latent (% of Total Load)	0.0%	0.0%	0.0%	0.0%
Total Infiltration Heat Load (% of Total)	**0.0%**	**0.0%**	**0.0%**	**0.0%**
Pull-down Sensible Heat Load (% of Total)	37.3%	5.8%	55.1%	16.0%
Respiration Heat Load (% of Total Load)	4.6%	0.0%	15.0%	0.0%
Total Product Heat Load (% of Total)	**41.9%**	**5.8%**	**70.1%**	**16.0%**
Lighting heat load (% of Total Load)	0.0%	0.0%	0.0%	0.0%
Occupancy heat load (% of Total Load)	0.0%	0.0%	0.0%	0.0%
Vehicle heat load (% of Total Load)	0.0%	0.0%	0.0%	0.0%
Misc. Heat Load (% of Total Load)	26.5%	29.7%	1.4%	2.7%
Total Miscellaneous Load (% of Total)	**26.5%**	**29.7%**	**1.4%**	**2.7%**

AHRI Standard 1250/1251 Box Load Calculation

From the proposed model load profiles, the AHRI Standard 1250/1251 (AHRI 2009b; AHRI 2009c) box load equations may be derived using the AHRI Load Spreadsheet methodology (AHRI 2009a).

Per the AHRI Standards, a walk-in load profile is composed of operation at a high usage, heavy load condition and low usage, light load condition. These are called high and low loads. The box high and low loads, as they are defined by AHRI and as they will be referenced in this work, refer to the refrigeration load, with exclusion of heat added to the space by the operation of the refrigeration system (defrost and evaporator fans), at the high and low load conditions, respectively. The box load is assumed to be composed of two components, a temperature dependent load component and a non-temperature dependent component (Equations 127 and 128). At the point where the space temperature equals the box surrounding ambient conditions, the driving potential for the temperature dependent portion of the load is assumed to go to zero. Likewise, when the ambient temperature is coincident with the rating point, the box load is assumed to be at the design percentage of the refrigeration system capacity.

$$\dot{BLH}(T_j) = \dot{Q}_{ss}(T_d)\left(C_4 \frac{T_j - T_C}{T_d - T_C} + C_5 \right) \tag{127}$$

$$\dot{BLL}(T_j) = \dot{Q}_{ss}(T_d)\left(C_6 \frac{T_j - T_C}{T_d - T_C} + C_7 \right) \tag{128}$$

The four coefficients that define the box load equations may be determined through the analysis of the model load outputs and application of the convenient conditions at certain ambient temperatures mentioned above. The temperature dependent box load coefficients may be solved by using Equations 129 and 130, which take advantage of the fact that these coefficients define the linear slope between two points on a graph of the box heat load versus the ratio of the space to ambient temperature difference coincident with the box heat load and the design point temperature difference. Values for the box heat loads at two ambient temperatures are obtained using the model hourly load calculation and applied to the equations below to determine the box load coefficients.

$$C_4 = \frac{\dot{BLH}(T_1) - \dot{BLH}(T_2)}{\dot{Q}_{ss}(T_d)\left(\frac{T_1 - T_C}{T_d - T_C} - \frac{T_2 - T_C}{T_d - T_C}\right)} = \frac{N_d}{\dot{BLH}(T_d)} \frac{\dot{BLH}(T_1) - \dot{BLH}(T_2)}{\left(\frac{T_1 - T_C}{T_d - T_C} - \frac{T_2 - T_C}{T_d - T_C}\right)} \tag{129}$$

$$C_6 = \frac{\dot{BLL}(T_1) - \dot{BLL}(T_2)}{\dot{Q}_{ss}(T_d)\left(\frac{T_1 - T_C}{T_d - T_C} - \frac{T_2 - T_C}{T_d - T_C}\right)} = \frac{N_d}{\dot{BLH}(T_d)} \frac{\dot{BLL}(T_1) - \dot{BLL}(T_2)}{\left(\frac{T_1 - T_C}{T_d - T_C} - \frac{T_2 - T_C}{T_d - T_C}\right)} \tag{130}$$

Equations 131 and 132 are achieved at the design ambient temperature, as previously noted. Through manipulating these equations and assuming a ratio of the high box load at the design point and the refrigeration system capacity at the design point, the non-temperature dependent coefficients may be determined (Equations 133 and 134).

$$\dot{BLH}(T_d) = (C_4 + C_5)\dot{Q}_{ss}(T_d) = N_d \dot{Q}_{ss}(T_d) \tag{131}$$

$$\dot{BLL}(T_d) = (C_6 + C_7)\dot{Q}_{ss}(T_d) \tag{132}$$

$$C_5 = N_d - C_4 \tag{133}$$

$$C_7 = \dot{BLL}(T_d)\frac{N_d}{\dot{BLH}(T_d)} - C_6 \tag{134}$$

The box load equations may be simplified to Equations 135 and 136, so that box load values may be determined for any bin temperature (T_j).

$$\dot{BLH}(T_j) = \frac{\dot{BLH}(T_1) - \dot{BLH}(T_2)}{T_1 - T_2}(T_j - T_d) + \dot{BLH}(T_d) \quad (135)$$

$$\dot{BLL}(T_j) = \frac{\dot{BLL}(T_1) - \dot{BLL}(T_2)}{T_1 - T_2}(T_j - T_d) + \dot{BLL}(T_d) \quad (136)$$

Determination of the BLL/BLH

To determine the box load low and high AHRI Standard 1250/1251 (AHRI 2009b; AHRI 2009c) equations, the results from the hourly model load profile are needed, as are assumptions for the additional capacity built into the refrigeration system above the design heat load. Typical system oversizing may be discerned from the literature through applied safety factors and noted compressor runtimes. Three references present typical safety factors applied to the design sizing of refrigeration systems, including (Krack 1977), (Blackmore 2012), and (Heatcraft 2008). Blackmore and Heatcraft specify a 10% safety factor be applied to the calculated design heat load whereas Krack specifies an additional 5 to 10%.

Authors, including (Adre and Hellickson 1989), (Ashby et al. 1979), (Becker et al. 2011), (Blackmore 2012), (Cooper 1973), (Goetzler et al. 2009), (Heatcraft 2008), (Krack 1977), (Morris 2012), (Nelson 2011), (Patel et al. 1993), (Sand et al. 1997), (SCE 2009), (Wade 1984), and (Westphalen et al. 1996), present compressor runtimes. Runtimes for freezers ranged from 40 to 99.07%, with an average value of 66.49%. Cooler compressor runtimes ranged from 0.70 to 91.67%, with an average value of 54.90%.

To select an appropriate design load to design capacity ratio, the model loads were analyzed at various ratios. Compressor runtimes were calculated according to the method first applied by Becker et al. (Becker et al. 2011) for each design load to capacity ratio. The general equation for calculating the compressor runtime based on the AHRI Standard 1250/1251 (AHRI 2009b; AHRI 2009c) is listed as Equation 137.

$$CR = \frac{1}{8760}\sum_{j=1}^{n} \frac{0.33 * \dot{BLH}(T_j) + 0.67 * \dot{BLL}(T_j)}{\dot{Q}_{ss}(T_j)} * n_j \qquad (137)$$

Applying the generalized equations for the box load high and low (Equations 127 and 128) the compressor runtime equation is modified to that listed in Equation 138. A new coefficient, C_8, has been added to the equation, defining the percentage of time that the walk-in box operates in a high load condition. Through information presented in Becker et al. (Becker et al. 2011), a value of 0.50 will be used for C_8. Additional inputs required include the exterior site conditions at the design point, the system capacity at the design conditions, the temperature of the refrigerated space, and a weather profile based off bin hours. The bin weather data used by this work is the same as specified by the AHRI 1250 Standard (AHRI 2009b), modeling Kansas City, Missouri's weather (Table 143).

$$CR = \frac{1}{8760}\sum_{j=1}^{n}\left(\left[\left(\frac{T_j - T_C}{T_d - T_C}\right)\left[C_8(C_4 - C_6) + C_6\right] + C_8C_5 - C_8C_7 + C_7\right]\frac{\dot{Q}_{ss}(T_d)}{\dot{Q}_{ss}(T_j)} n_j\right) \qquad (138)$$

Table 143. AHRI 1250 Condenser Out, Temperature Bins and Corresponding Bin Hours, Kansas City, MO

Bin Temperature (ºF)	Bin Hours (hr)
100.4	9
95	74
89.6	257
84.2	416
78.8	630
73.4	898
68	737
62.6	943
57.2	628
51.8	590
46.4	677
41	576
35.6	646
30.2	534
24.8	322
19.4	305
14	246
8.6	189
3.2	78
-2.2	5

Source: (AHRI 2009b)

$\dot{Q}_{ss}\left(T_j\right)$ is approximated using three points of steady state refrigeration system capacity, coincident with the method of test points ((AHRI 2009b) Equations 18 and 20). (Becker et al. 2011) obtained these values through simulation of four walk-ins. The values for system capacity were applied in this work to determine the compressor runtime. The magnitude of these system capacity values is not important, as there is also a measure of capacity in the numerator of the compressor runtime equation. Using Becker's system capacity values allows direct comparison of compressor runtime results from the AHRI and proposed box load equations, as they use the same backdrop.

$T_j \leq 59°F$

$$\dot{Q}_{ss}\left(T_j\right) = \dot{Q}_{ss}\left(35°F\right) + \left(\dot{Q}_{ss}\left(59°F\right) - \dot{Q}_{ss}\left(35°F\right)\right)\frac{T_j - 35}{59 - 35} \qquad \text{Equation 18 (AHRI 2009b)}$$

$T_j > 59°F$

$$\dot{Q}_{ss}\left(T_j\right) = \dot{Q}_{ss}\left(59°F\right) + \left(\dot{Q}_{ss}\left(95°F\right) - \dot{Q}_{ss}\left(59°F\right)\right)\frac{T_j - 59}{95 - 59} \qquad \text{Equation 20 (AHRI 2009b)}$$

The results of the box load high to design capacity ratio and compressor runtime analysis are presented in Table 144, Table 145, Table 146, and Table 147 for the small and large coolers and freezers. Figure 8 displays the information listed the in tables noted above. Walk-in boxes located indoors with indoor condensing units operate at higher compressor runtimes relative to other box and condenser configurations with the same box load high to capacity ratio. This is due to the operating conditions being more consistent through the period. Coolers operate at higher compressor runtimes, when compared to freezers at the same box load high to capacity ratio. Smaller refrigerated spaces have less variation in the compressor runtime for a given box load high to capacity ratio, compared to larger refrigerated spaces.

From the information presented in the literature about compressor runtime and refrigeration system safety factor, it is apparent that systems are designed according to a particular box load high to capacity ratio, as opposed to being sized with a particular annual compressor runtime in mind. Systems with more unpredictable operating conditions are given more oversize than boxes with a consistent load profile, such as indoor walk-in boxes with indoor condensing units. Through a rough analysis of the literature and the results from the hourly model load analysis of the box load high to capacity ratio, the ratio will be set to 70% for walk-in coolers and 85% for walk-in freezers.

Table 144. Small Cooler Compressor Runtime as a Function of Box Load High to Capacity Ratio

Small Cooler Compressor Runtime			
	CR: Compressor Runtime		
N_d: Box Load High to Capacity Ratio	Indoor Box Indoor Condenser	Indoor Box Outdoor Condenser	Outdoor Box Outdoor Condenser
70	51.49	46.36	41.80
72	52.96	47.69	43.00
74	54.43	49.01	44.19
76	55.90	50.34	45.39
78	57.37	51.66	46.58
80	58.84	52.99	47.77
82	60.31	54.31	48.97
84	61.78	55.64	50.16
86	63.26	56.96	51.36
88	64.73	58.29	52.55
90	66.20	59.61	53.75

Table 145. Small Freezer Compressor Runtime as a Function of Box Load High to Capacity Ratio

Small Freezer Compressor Runtime			
	CR: Compressor Runtime		
N_d: Box Load High to Capacity Ratio	Indoor Box Indoor Condenser	Indoor Box Outdoor Condenser	Outdoor Box Outdoor Condenser
70	45.37	43.39	37.34
72	46.66	44.63	38.41
74	47.96	45.87	39.48
76	49.25	47.11	40.54
78	50.55	48.35	41.61
80	51.85	49.59	42.68
82	53.14	50.83	43.74
84	54.44	52.07	44.81
86	55.73	53.31	45.88
88	57.03	54.55	46.94
90	58.33	55.79	48.01

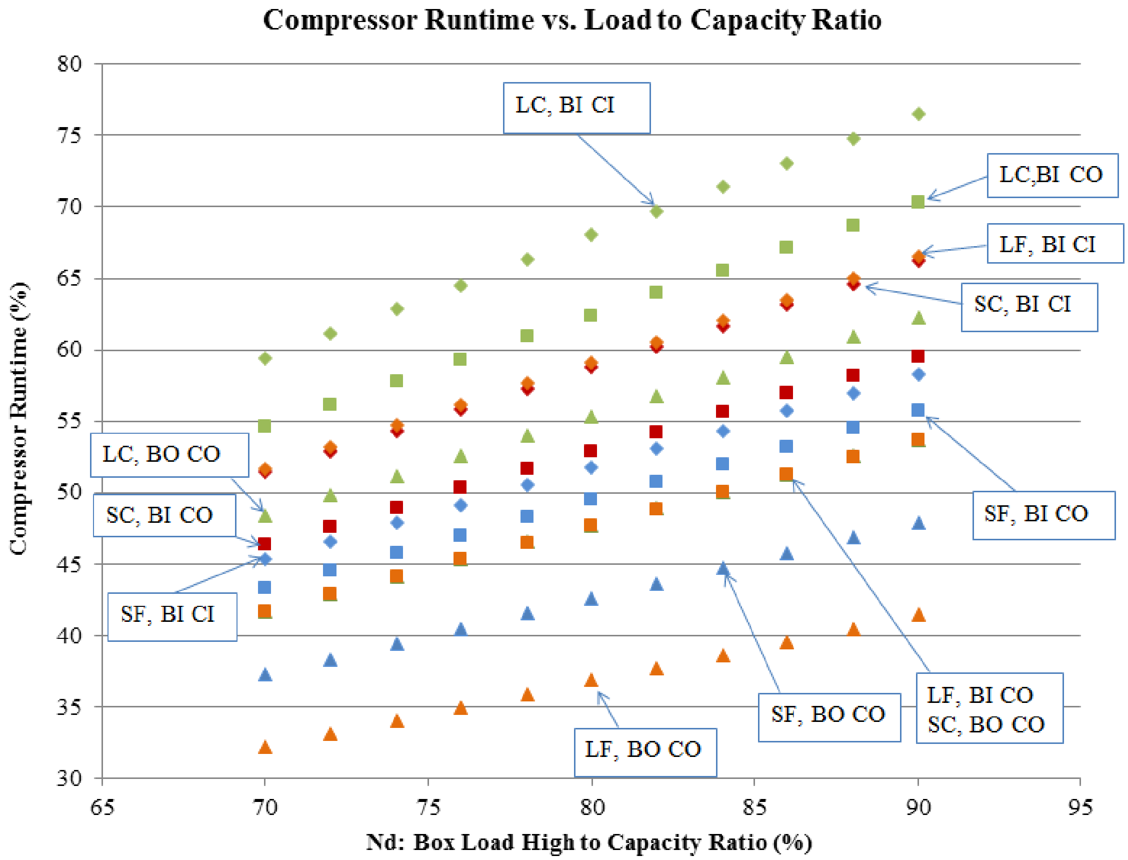
Table 146. Large Cooler Compressor Runtime as a Function of Box Load High to Capacity Ratio

Large Cooler Compressor Runtime			
	CR: Compressor Runtime		
N_d: Box Load High to Capacity Ratio	Indoor Box Indoor Condenser	Indoor Box Outdoor Condenser	Outdoor Box Outdoor Condenser
70	59.51	54.66	48.48
72	61.21	56.22	49.87
74	62.91	57.78	51.25
76	64.61	59.34	52.64
78	66.31	60.90	54.02
80	68.01	62.46	55.41
82	69.71	64.03	56.79
84	71.41	65.59	58.18
86	73.11	67.15	59.56
88	74.81	68.71	60.95
90	76.51	70.27	62.33

Table 147. Large Freezer Compressor Runtime as a Function of Box Load High to Capacity Ratio

Large Freezer Compressor Runtime			
	CR: Compressor Runtime		
N_d: Box Load High to Capacity Ratio	Indoor Box Indoor Condenser	Indoor Box Outdoor Condenser	Outdoor Box Outdoor Condenser
70	51.75	41.76	32.27
72	53.23	42.96	33.19
74	54.71	44.15	34.11
76	56.19	45.34	35.03
78	57.67	46.54	35.96
80	59.15	47.73	36.88
82	60.62	48.92	37.80
84	62.10	50.12	38.72
86	63.58	51.31	39.64
88	65.06	52.50	40.57
90	66.54	53.70	41.49

Figure 8. Compressor Runtime vs. Load to Capacity Ratio Analysis



Plugging the above values for the box load high to capacity ratio into Equations 129 to 134, the box load equations (Equations 127 and 128) for each refrigerated space may be determined. The determined coefficients are listed in Table 148.

$$B\dot{L}H(T_j) = \dot{Q}_{ss}(T_d)\left(C_4 \frac{T_j - T_C}{T_d - T_C} + C_5\right) \tag{127}$$

$$B\dot{L}L(T_j) = \dot{Q}_{ss}(T_d)\left(C_6 \frac{T_j - T_C}{T_d - T_C} + C_7\right) \tag{128}$$

Table 148. Proposed Modified Box Load Equation Coefficients

	C_4	C_5	C_6	C_7	C_8
Small Cooler	0.0855	0.6145	0.0855	0.2573	0.5000
Large Cooler	0.1022	0.5978	0.1022	0.3910	0.5000
Small Freezer	0.1251	0.7249	0.1251	0.1431	0.5000
Large Freezer	0.2334	0.6166	0.2334	0.1915	0.5000

Averaging the coefficients for the small and large refrigerated spaces results in proposed modified Standard AHRI 1250/1251 box load equations, as follows in Equations 139, 140, 142, and 143. These equations are combined with the coefficient C_8 to develop the complete box load definitions in Equations 141 and 144.

Walk-in Cooler Proposed Box Load Equations:

$$\dot{BLH}(T_j) = \dot{Q}_{ss}(T_d)\left(0.094 \cdot \frac{T_j - T_C}{T_d - T_C} + 0.606\right) \tag{139}$$

$$\dot{BLL}(T_j) = \dot{Q}_{ss}(T_d)\left(0.094 \frac{T_j - T_C}{T_d - T_C} + 0.324\right) \tag{140}$$

$$BL(T_j) = \left(0.094 \cdot \frac{T_j - T_C}{T_d - T_C} + 0.465\right) \cdot \dot{Q}_{ss}(T_d) \cdot n_j \tag{141}$$

Walk-in Freezer Proposed Box Load Equations

$$\dot{BLH}(T_j) = \dot{Q}_{ss}(T_d)\left(0.179 \cdot \frac{T_j - T_C}{T_d - T_C} + 0.671\right) \tag{142}$$

$$\dot{BLL}(T_j) = \dot{Q}_{ss}(T_d)\left(0.179 \frac{T_j - T_C}{T_d - T_C} + 0.167\right) \tag{143}$$

$$BL(T_j) = \left(0.179 \cdot \frac{T_j - T_C}{T_d - T_C} + 0.419\right) \cdot \dot{Q}_{ss}(T_d) \cdot n_j \tag{144}$$

Comparison of Proposed Box Load Equations to AHRI Standard 1250/1251 Equations

The AHRI Standard 1250/1251 Equations (AHRI 2009b; AHRI 2009c) are listed below for comparison to the proposed box load equations.

AHRI Walk-in Cooler Box Load Equations:

$$\dot{BLH}(T_j) = \dot{Q}_{ss}(T_d)\left(0.05 \cdot \frac{T_j - T_C}{T_d - T_C} + 0.65\right) \tag{145}$$

$$\dot{BLL}(T_j) = \dot{Q}_{ss}(T_d)\left(0.07 \frac{T_j - T_C}{T_d - T_C} + 0.03\right) \tag{146}$$

$$BL(T_j) = \left(0.063 \cdot \frac{T_j - T_C}{T_d - T_C} + 0.235\right) \cdot \dot{Q}_{ss}(T_d) \cdot n_j \tag{147}$$

AHRI Walk-in Freezer Box Load Equations

$$\dot{BLH}(T_j) = \dot{Q}_{ss}(T_d)\left(0.25 \cdot \frac{T_j - T_C}{T_d - T_C} + 0.55\right) \tag{148}$$

$$\dot{BLL}(T_j) = \dot{Q}_{ss}(T_d)\left(0.25 \frac{T_j - T_C}{T_d - T_C} + 0.15\right) \tag{149}$$

$$BL(T_j) = \left(0.250 \cdot \frac{T_j - T_C}{T_d - T_C} + 0.282\right) \cdot \dot{Q}_{ss}(T_d) \cdot n_j \tag{150}$$

The variation between Equations 139 through 144 and the corresponding Equations 145 through 150 is not simply due to the dynamics of altering the model load profile. Coefficient C_5 decreases for the walk-in cooler (Proposed to AHRI) while C_4 and C_7 increase. Coefficient C_4 decreases for the walk-in freezer (Proposed to AHRI) while C_5 and C_7 both increase. When applied to the consolidated box load equation utilizing the new value for coefficient C_8, all equation coefficients are increased for the walk-in cooler, as is the temperature dependent coefficient for the walk-in freezer. The non-temperature dependent coefficient for the walk-in freezer decreases.

The proposed walk-in cooler box load has a 49.2% higher temperature dependent coefficient and a 97.9% higher non-temperature dependent coefficient when compared to the AHRI walk-in cooler box load. The proposed walk-in freezer box load has a 28.4% smaller temperature dependent coefficient and a 48.6% higher non-temperature dependent coefficient when compared to the AHRI walk-in freezer box load. Essentially, the walk-in cooler box load is higher according to the revised model load whereas the walk-in freezer box load is similar on average, and less affected by the temperature differential between the space and its surroundings.

Analysis of the Proposed Model Load

Through application of Equation 137, compressor runtimes were calculated for the walk-in cooler and freezer, for condenser indoor and outdoor conditions, using the system capacities used by Becker et al., thereby allowing for direct comparison to his results. Compressor runtimes are listed in Table 149. In comparison to runtime information obtained from the literature, the proposed values show better correlation to typical operation of 55 to 65% compressor runtime.

Table 149. Compressor Runtimes for the Proposed and AHRI 1250 Box Load Equations using System Capacities Determined by (Becker et al. 2011)

	Proposed 1250		AHRI 1250	
	Condenser Indoors	Condenser Outdoors	Condenser Indoors	Condenser Outdoors
Small Cooler	55.90	44.69	29.80	23.43
Large Cooler	55.90	45.70	29.80	23.54
Small Freezer	59.80	45.87	53.20	41.89
Large Freezer	59.80	39.63	53.20	36.16

Source: (Becker et al. 2011)

For the proposed model load the compressor runtimes have increased in all cases. Coolers with the condenser locating indoors have a compressor runtime increase of 88% whereas freezer runtimes increased by 12%. Small and large coolers with the condenser outdoors had runtime increases similar to indoor condensing unit coolers at 91% and 94%, respectively. Walk-in freezers with outdoor condensers saw a moderate increase in compressor runtime of 10% for both the small and large freezers.

Comparing the model loads utilized by AHRI and the proposed revision to the Standard, Table 150 and Table 151 were developed. For consistency the AHRI I-P calculation was compared to the corresponding I-P calculation performed for the proposed model load. The average hourly loads for walk-in coolers are comparable for the AHRI Load Spreadsheet and the proposed model load, decreasing 8% for the small cooler and increasing 34% for the large cooler. The main variation in the refrigeration loads, between the two calculations, is the range of the heat loads. Small and large cooler heat loads calculated by AHRI vary from the average by + 325% / -78% and +369% / -70%, respectively. The Proposed Model Load has heat loads for the small and large coolers varying from the average by + 35% / - 35% and + 18% / - 17%, respectively.

Table 150. Walk-in Cooler Hourly Load Comparisons for the AHRI Load Spreadsheet (AHRI 2009a) and the Proposed Model Load

	AHRI Load Spreadsheet		Proposed Model Load	
	Small Cooler	Large Cooler	Small Cooler	Large Cooler
Peak Hourly Load	**14,547**	**258,314**	**4,220**	**87,576**
Conduction (Btu/h)	794	17,800	691	19,359
Infiltration (Btu/h)	5,757	15,728	1,276	4,357
Product Load (Btu/h)	6,975	90,000	862	42,938
Miscellaneous (Btu/h)	1,060	136,286	1,391	20,922
Ave Hourly Load	**3,421**	**55,120**	**3,133**	**74,091**
Conduction (Btu/h)	774	17,050	665	18,077
Infiltration (Btu/h)	257	1,638	638	2,178
Product Load (Btu/h)	2,325	30,000	862	42,938
Miscellaneous (Btu/h)	65	6,432	968	10,898
Min Hourly Load	**755**	**16,300**	**2,060**	**61,247**
Conduction (Btu/h)	755	16,300	652	17,436
Infiltration (Btu/h)	0	0	0	0
Product Load (Btu/h)	0	0	862	42,938
Miscellaneous (Btu/h)	0	0	546	873

The average hourly loads for walk-in freezers (Table 151) varied a good amount from AHRI to the proposed model load. The small freezer average load increased by 126%, and the large freezer load increased by 36%. Small and large freezer heat loads calculated by AHRI vary from the average by + 244% / -33% and +216% / -29%, respectively. The proposed model load has heat loads for the small and large coolers varying from the average by + 52% / - 52% and + 34% / - 33%, respectively. The variability of the AHRI

Load Spreadsheet model loads result in lower compressor runtimes than are typical in application.

Table 151. Walk-in Freezer Hourly Load Comparisons for the AHRI Load Spreadsheet (AHRI 2009a) and the Proposed Model Load

	AHRI Load Spreadsheet		Proposed Model Load	
	Small Freezer	Large Freezer	Small Freezer	Large Freezer
Peak Hourly Load	**5,835**	**113,805**	**5,829**	**65,762**
Conduction (Btu/h)	1,160	26,563	1,221	28,054
Infiltration (Btu/h)	2,832	7,254	3,283	11,210
Product Load (Btu/h)	1,000	12,500	107	5,208
Miscellaneous (Btu/h)	872	68,660	1,218	21,290
Ave Hourly Load	**1,694**	**35,959**	**3,825**	**48,939**
Conduction (Btu/h)	1,145	25,977	1,195	27,044
Infiltration (Btu/h)	150	2,116	1,641	5,605
Product Load (Btu/h)	333	4,167	107	5,208
Miscellaneous (Btu/h)	65	3,700	882	11,082
Min Hourly Load	**1,130**	**25,391**	**1,835**	**32,620**
Conduction (Btu/h)	1,130	25,391	1,183	26,539
Infiltration (Btu/h)	0	0	0	0
Product Load (Btu/h)	0	0	107	5,208
Miscellaneous (Btu/h)	0	0	546	873

The goal of the AHRI Standard 1250/1251 (AHRI 2009b; AHRI 2009c) is to calculate the AWEF for manufactured refrigeration equipment. (Becker et al. 2011) noted variations of the AHRI 1250 Standard AWEF values and compressor runtimes to simulations of climate zones in his analysis (Table 152).

Table 152. Percentage Variations of AWEF and Compressor Runtimes, AHRI 1250 to Average Climate Zone Simulations

	Small Cooler	Large Cooler	Small Freezer	Large Freezer
AWEF				
Condenser In	-27.9	-10.9	+ 21.3	-2.5
Condenser Out	-28.1	-17.5	+ 10.7	-4.1
Compressor Run (%)				
Condenser In	-45.8	-41.7	-3.3	-5.0
Condenser Out	-49.1	-49.8	-12.0	-29.6

Source: (Becker et al. 2011)

CHAPTER 6

EQUEST SIMULATIONS

Four sets of simulations were performed to analyze the Proposed 1250 Standard equations. eQuest (Hirsch 2006), a whole building energy-use simulation software, was used to simulate each walk-in box and refrigeration system. The simulation software was initially developed to analyze energy efficiency in building design, but a refrigeration module has been developed. The modeling tool allows for a detailed representation of the walk-in construction, walk-in load profile, and refrigeration equipment. eQuest is capable of utilizing various weather data files, including weather data from the National Oceanic and Atmospheric Administration (NOAA), to allow energy calculations over a time period with varying outdoor conditions.

eQuest has been applied by Southern California Edison (SCE 2008) and Becker et al. (Becker et al. 2011) in their respective simulations of walk-in coolers and freezers. Southern California Edison did not present a justification for selection of this modeling software. Becker et al. performed an extensive study in the selection of this modeling tool, analyzing four additional modeling software products. After selection of eQuest as the best tool for modeling walk-in coolers and freezers, the software was validated through simulation of laboratory tests performed by Pacific Gas & Electric Company (PG&E). The refrigerated space was modeled according to its size and construction. Refrigeration equipment was modeled according to the specifications provided by PG&E, as were additional refrigeration loads on the space. Three PG&E laboratory tests were simulated in eQuest by Becker et al. and the resulting refrigeration system energy usage was tabulated. Comparison to the PG&E recorded energy usage showed variance of -0.35 to +4.55%, with an average of +2.69%. Therefore, eQuest is capable of developing simulation results for walk-in refrigeration system operation with good correlation to actual measured data.

As noted previously, four sets of simulations were performed to analyze the Proposed 1250 Standard equations. Each simulation set was for a different walk-in, including a small cooler, large cooler, small freezer, and large freezer, in accordance with the proposed model load (Table 130 through Table 132) and proposed hourly load calculation (Table 133 through Table 136). Each simulation set was composed of six simulations and includes simulations representing tests performed in a laboratory setting to allow calculation of the AWEF according to the Proposed 1250 Standard equations and hourly simulations representing a walk-in operating over the course of a year.

According to the AHRI 1250/1251 Standard (AHRI 2009b; AHRI 2009c), three laboratory tests are required to allow determination of the AWEF for refrigeration equipment operating with an outdoor condenser, and one laboratory test is required for calculating the AWEF for a walk-in refrigeration system operating with an indoor condenser. Laboratory tests of refrigeration equipment require two environmental chambers. One test chamber contains the evaporator for the refrigeration system under test. Heat and humidity is added to the space to simulate the walk-in box load, and the evaporator works to maintain a space setpoint temperature, as defined by the Standard. Heat load is increased until the refrigeration system is operating 100% of the time, but the evaporator is still maintaining the setpoint temperature. The second test chamber contains the refrigeration system’s condensing unit. It is maintained at specific conditions (temperature and humidity according to the Standard) to simulate condenser operating conditions.

From these tests, the system capacity and system electrical usage is recorded. These values are then applied to the Proposed 1250 equations to approximate the refrigeration system's AWEF value. The AWEF represents refrigeration system operation over an entire year, and it is obtained for condenser indoor and outdoor configurations through these four laboratory tests.

Hourly simulations were performed for four walk-in configurations, including a small cooler, large cooler, small freezer, and large freezer. These simulations are meant to represent actual refrigeration operation over an entire year using a weather file to represent varying outdoor conditions at 15 minutes intervals. The walk-in construction, space heat loads, and load schedules setup in the simulations are the same as the corresponding system operating parameters outlined in the proposed model load (Table 130 through Table 132). After performing the simulation of a year of refrigeration system operation, the walk-in box load and refrigeration system electrical consumption is summed over the entire year. In correspondence with the AWEF equation (Equation 1), the annual box load is divided by the system electrical consumption over the course of the year.

For each simulation set, the proposed model load was used to determine the overall space load, from which refrigeration equipment was sized according to industry practice (Heatcraft 2008). Information pertinent to the refrigeration equipment was loaded into the simulation software. Manufacturer compressor information included the mass flow and power usage as a function of saturated suction and discharge temperatures. All refrigeration systems simulated used R-404a refrigerant. The condensing temperature is throttled within 10 ºF of the condenser design temperature with a backflooding setpoint of 65ºF. The eQuest condenser rated capacity varies from the rated capacity noted on the manufacturer's specification sheet, as it also includes the heat of compression added to the system. For the simulation, an air-side heat balance of the condensing unit was performed at the design conditions. Condenser fans are assumed to be single-speed, cycling to maintain the desired operating conditions. Evaporator performance and energy usage is set up by eQuest using fan affinity and design performance parameters. Fan affinity parameters include the static pressure that the fan operates at, fan efficiency, and mechanical efficiency (motor and drive).

Small Cooler Equipment Sizing

The small cooler simulation was set up in accordance with the proposed model load parameters presented in Table 130, Table 131, and Table 133. Items not included in the model load include the defrost load, defrost schedule, and the evaporator fan schedule.

Defrost occurs for an hour twice a day at 8 am and 8 pm. Defrost heat load added to the refrigerated space is assumed to be 60% of the electric usage, in accordance with Cole (Cole 1989). Over the time periods when defrost is occurring the refrigeration system is shut off, including the evaporator fans. Box construction is in accordance with the proposed model load. In addition to the parameters noted in the model load, absorptivity was set at 0.7 and surface roughness was noted as 5, which corresponds to a smooth surface, such as glass or paint on pine (Hirsch 2006).

Equipment was sized according to the industry standard by assuming a 16 hour design condensing unit runtime (Heatcraft 2008; Krack 1977) and then applying a 10% safety factor. The calculation method used was that presented by Heatcraft in their Engineering Manual (Heatcraft 2008).

Table 153. Small Cooler Heatcraft Load Calculation (Heatcraft 2008)

Simulation Set:		Small Cooler		
INPUTS				
Box Dims (L x W x H):		8	8	8
Ambient Temp (F):		95		
Room Temp (F):		35	TD	60
Ground Temp (F), Table 21		55	TD	20
Insulation:	inches thick	type	R-value	Heat Load (Table 1)
Ceiling:			25	41
Walls:			25	41
Floor:			15	70.89
Product Load:				
Lbs / day:		2048		
Cp:		0.90		
Entry temp (F):		45		
UnPackaged Lbs Stored:		3225.50		
Heat of Respiration (Btu/lb/24 hr, Table 8)		0.72		
Miscellaneous Loads:				
Motors (hp):		0.33		
Vehicles (hp):		0		
Lights (W * %On):		19.2		
People (No * %Occup):		0.2		
Other (W):		160		
CALCULATED LOAD COMPONENTS				
Transmission Load				
Ceiling:			2624	
Walls:			10496	
Floor:			4537	
Air Change Load				
Factor (Table 4)		25.64	32688	
Factor (Table 6)		2.49		
Product Load				
(Sensible) lbs/day x Spec Heat x Temp Drop			18432	
(Latent) lbs stored x Factor (Table 8)			2322	
Miscellaneous Loads				
(Motor and Veh)x75000 Btu/hp - 24 hr			24531	
(Lights)x 82			1574	
(People) BTU/24 hrs (Table 12)		21480	4296	
Other (Btu/24 hr)			13102	
Total Load			114603	
x 1.1 SF				
Total Load w SF			126063	
Operating Hrs		16		
Hourly Heat Load (Btu/hr)			7879	

Using the calculated sizing load the condensers and compressors were sized using Emerson Climate Technologies' online product sizing application (Emerson 2012). The evaporators were selected from product information available from Larkin (Larkin 2007).

Equipment Selection:

- One Copeland RST61C1E-CAV compressor: 7,700 Btu/h at 20SST/110SCT.
- One Emerson FJAM-A101-TFC-001 condenser: 7,700 Btu/h at 25SST/123.6SCT/90DB
- One Larkin LCE665 evaporator: 7,475 Btu/h at 20SST/10TD

Small Freezer Equipment Sizing

The small freezer simulation was set up in accordance with the proposed model load parameters presented in Table 130, Table 132, and Table 135. Items not included in the model load include the defrost load and schedule and the evaporator fan schedule.

Defrost occurs for a half hour four times a day at 2 am, 8 am, 2 pm, and 8 pm. Defrost heat load added to the refrigerated space is assumed to be 60% of the electric usage, in accordance with Cole (Cole 1989). Over the time periods when defrost is occurring the refrigeration system is shut off, including the evaporator fans. Box construction is in accordance with the proposed model load. In addition to the parameters noted in the model load, absorptivity was set at 0.7 and surface roughness was noted as 5, which corresponds to a smooth surface, such as glass or paint on pine (Hirsch 2006).

Equipment was sized according to the industry standard by assuming a 18 hour design condensing unit runtime (Heatcraft 2008; Krack 1977) and then applying a 10%

safety factor. The calculation method used was that presented by Heatcraft in their Engineering Manual (Heatcraft 2008).

Table 154. Small Freezer Heatcraft Load Calculation (Heatcraft 2008)

Simulation Set:		Small Freezer		
INPUTS				
Box Dims (L x W x H):		8	8	8
Ambient Temp (F):		95		
Room Temp (F):		-5	TD	100
Ground Temp (F), Table 21		55	TD	60
Insulation:	inches thick	type	R-value	
Ceiling:			32	
Walls:			28	
Floor:			25	
Product Load:				
Lbs / day:		512		
Cp:		0.5		
Entry temp (F):		5		
Miscellaneous Loads:				
Motors (hp):		0.65		
Vehicles (hp):		0		
Lights (W):		64		
People (No):		1		
Other (W):		160		
CALCULATED LOAD COMPONENTS				
Transmission Load				
Ceiling:			4752	
Walls:			21312	
Floor:			3456	
Air Change Load				
Factor (Table 5)		19.76	38243	
Factor (Table 6)		3.78		
Product Load				
lbs/day x Spec Heat x Temp Drop			2560	
Miscellaneous Loads				
(Motor and Veh)x75000 Btu/hp - 24 hr			48865	
(Lights)x 82			5248	
(People) BTU/24 hrs (Table 12)		32400	32400	
Other (Btu/24 hr)			13102	
Total Load			156836	
x 1.1 SF				
Total Load w SF			172519	
Operating Hrs		18		
Hourly Heat Load (Btu/hr)			9584	

Using the calculated sizing load the condensers and compressors were sized using Emerson Climate Technologies' online product sizing application (Emerson 2012). The evaporators were selected from product information available from Larkin (Larkin 2007).

Equipment Selection:

- One Copeland ZF11K4E-PFV compressor: 11,700 Btu/h at -25SST/105SCT.
- One Emerson DJAL-025Z-CFV-001 condenser: 11,300 Btu/h at -25SST/108.6SCT/90DB
- Two Larkin LCE467 evaporators: 6,365 Btu/h ea at -25SST/10TD

Large Cooler Equipment Sizing

The large cooler simulation was set up in accordance with the proposed model load parameters presented in Table 130, Table 131, and Table 134. Items not included in the model load include the defrost load and schedule and the evaporator fan schedule.

Defrost occurs for an hour twice a day at 8 am and 8 pm. Defrost heat load added to the refrigerated space is assumed to be 60% of the electric usage, in accordance with Cole (Cole 1989). Over the time periods when defrost is occurring the refrigeration system is shut off, including the evaporator fans. Box construction is in accordance with the proposed model load. In addition to the parameters noted in the model load, absorptivity was set at 0.7 and surface roughness was noted as 5, which corresponds to a smooth surface, such as glass or paint on pine (Hirsch 2006).

Equipment was sized according to the industry standard by assuming a 18 hour design condensing unit runtime (Heatcraft 2008; Krack 1977) and then applying a 10%

safety factor. The calculation method used was that presented by Heatcraft in their Engineering Manual (Heatcraft 2008).

Table 155. Large Cooler Heatcraft Load Calculation (Heatcraft 2008)

Simulation Set:		Large Cooler		
INPUTS				
Box Dims (L x W x H):		50	50	20
Ambient Temp (F):		95		
Room Temp (F):		35	TD	60
Ground Temp (F), Table 21		55	TD	20
Insulation:	inches thick	type	R-value	Heat Load (Table 1)
Ceiling:			25	41
Walls:			25	41
Floor:			15	70.89
Product Load:				
Lbs / day:		90000		
Cp:		0.90		
Entry temp (F):		45		
Unpackaged Lbs Stored:		315000.00		
Heat of Respiration (Btu/lb/24 hr, Table 8)		0.72		
Miscellaneous Loads:				
Motors (hp):		6.54		
Vehicles (hp):		2.27		
Lights (W):		2500		
People (No*%Occup):		0.4		
Other (W):		256		
CALCULATED LOAD COMPONENTS				
Transmission Load				
Ceiling:			102500	
Walls:			164000	
Floor:			177237	
Air Change Load				
Factor (Table 4)		2	249000	
Factor (Table 6)		2.49		
Product Load				
(Sensible) lbs/day x Spec Heat x Temp Drop			810000	
(Latent) lbs stored x Factor (Table 8)			226800	
Miscellaneous Loads				
(Motor and Veh)x75000 Btu/hp - 24 hr			661217	
(Lights)x 82			205000	
(People) BTU/24 hrs (Table 12)		21480	8592	
Other (Btu/24 hr)			20963	
Total Load			2625309	
x 1.1 SF				
Total Load w SF			2887840	
Operating Hrs		16		
Hourly Heat Load (Btu/hr)			180490	

Using the calculated sizing load the condensers and compressors were sized using Emerson Climate Technologies' online product sizing application (Emerson 2012). The evaporators were selected from product information available from Larkin (Larkin 2007).

Equipment Selection:

- One Copeland 4DH3R22ML-TSK compressor: 194,000 Btu/h each at 25SST/110SCT.
- Six Emerson FFAS-A35Z-CFV-081 condensers: 30,400 Btu/h ea at 25SST/118.9SCT/90DB
- Eight Larkin LCE6200 evaporators: 23,000 Btu/h ea at 20SST/10TD

Large Freezer Equipment Sizing

The large freezer simulation was set up in accordance with the proposed model load parameters presented in Table 130, Table 132, and Table 136. Items not included in the model load include the defrost load and schedule and the evaporator fan schedule.

Defrost occurs for an hour twice a day at 8 am and 8 pm. Defrost heat load added to the refrigerated space is assumed to be 60% of the electric usage, in accordance with Cole (Cole 1989). Over the time periods when defrost is occurring the refrigeration system is shut off, including the evaporator fans. Box construction is in accordance with the proposed model load. In addition to the parameters noted in the model load, absorptivity was set at 0.7 and surface roughness was noted as 5, which corresponds to a smooth surface, such as glass or paint on pine (Hirsch 2006).

Equipment was sized according to the industry standard by assuming a 18 hour design condensing unit runtime (Heatcraft 2008; Krack 1977) and then applying a 10%

safety factor. The calculation method used was that presented by Heatcraft in their Engineering Manual (Heatcraft 2008).

Table 156. Large Freezer Heatcraft Load Calculation (Heatcraft 2008)

Simulation Set:		Large Freezer		
INPUTS				
Box Dims (L x W x H):		50	50	20
Ambient Temp (F):		95		
Room Temp (F):		-5	TD	100
Ground Temp (F), Table 21		55	TD	60
Insulation:	inches thick	type	R-value	
Ceiling:			32	
Walls:			28	
Floor:			25	
Product Load:				
Lbs / day:		25000		
Cp:		0.5		
Entry temp (F):		5		
Miscellaneous Loads:				
Motors (hp):		4.89		
Vehicles (hp):		4		
Lights (W):		2500		
People (No):		2		
Other (W):		256		
CALCULATED LOAD COMPONENTS				
Transmission Load				
Ceiling:			185625	
Walls:			333000	
Floor:			135000	
Air Change Load				
Factor (Table 5)		1.6	302400	
Factor (Table 6)		3.78		
Product Load				
lbs/day x Spec Heat x Temp Drop			125000	
Miscellaneous Loads				
(Motor and Veh)x75000 Btu/hp - 24 hr			666489	
(Lights)x 82			205000	
(People) BTU/24 hrs (Table 12)		32400	64800	
Other (Btu/24 hr)			20963	
Total Load			2017314	
x 1.1 SF				
Total Load w SF			2219045	
Operating Hrs		18		
Hourly Heat Load (Btu/hr)			123280	

Using the calculated sizing load the condensers and compressors were sized using Emerson Climate Technologies' online product sizing application (Emerson 2012). The evaporators were selected from product information available from Larkin (Larkin 2007).

Equipment Selection:

- Four Copeland 3DB3F33KL-TFC compressors: 33,100 Btu/h each at -25SST/105SCT.
- Four Emerson CJDK-0750-TSC-001 condensers: 34,030 Btu/h ea at -25SST/106.2SCT/90DB
- Eight Larkin LCE4174 evaporators: 17,400 Btu/h ea at -20SST/10TD

eQuest Hourly Simulation Setup and Procedure

Hourly simulations were set up for box in-condenser in and box out condenser out configurations. The box in, condenser used a weather file representing 90ºF dry bulb temperature and a 75ºF wet bulb. Because the condenser is located indoors, it was assumed for the simulation that there was no wind or solar loading on the walk-in box. This weather file parallels the AHRI Standard 1250/1251 (AHRI 2009b; AHRI 2009c) definition for indoor condensing unit walk-in box ambient conditions. Box out condenser out climate zone simulations were only performed for Kansas City, MO. The Kansas City weather file is the basis for the algorithm used to make calculations for outdoor condensing unit systems in the Standard (Table 143). Simulation of the Kansas City weather data allowed for a direct comparison to the accuracy of the AHRI walk-in rating methodology outlined by AHRI Standard 1250/1251 (AHRI 2009b; AHRI 2009c). The altitude is set at 1033 ft, corresponding with the altitude of Kansas City, MO (SEL et al. 2006). Ground temperatures for the refrigerated space are set at 55ºF.

Evaporator design air flow and capacity are input directly from the manufacturer's specification. Evaporator fan power usage is not input directly into eQuest, but it is calculated by the simulation program using a fan performance curve, an operating pressure, and equipment efficiencies. Initial guesses for fan efficiency and fan operating pressure are made, and then a simulation is run. The resulting fan power usage is iterated to the manufacturer's specified energy usage through multiple simulations.

eQuest AHRI 1250 Test Simulation Setup and Procedure

AHRI specifies multiple methods of testing equipment in the Standard 1250/1251 (AHRI 2009b; AHRI 2009c). Systems with indoor condensing unit refrigeration systems require performance testing at a condenser entering air temperature of 90 ºF DB (75ºF WB). Rating of refrigeration systems with the condenser outdoors are performed at three ambient conditions, including 95ºF/75ºF, 59ºF/54ºF, and 35ºF/34ºF. Weather files were generated that modeled these conditions, and additionally specify no cloud cover, no precipitation, no wind, and no solar loading.

Testing performed in the laboratory setting to allow calculation of the AWEF is performed during continuous system operation. The heat load on the refrigeration equipment is increased until the system reaches continuous steady state operation. At this point, the refrigeration system is rejecting its maximum potential of heat load. During testing the energy usage and system heat rejection are recorded to allow determination of the AWEF. The simulation of this test method is set up to parallel the physical test.

The box that the refrigeration system is operating has its heat resistance increased to 10,000 h-ft^2-°F/Btu (limited by eQuest) to greatly reduce the heat lost to the surroundings. The majority of space heat loads, including occupancy, equipment, product, defrost, and lighting are removed from the simulation. In their place a singular load is added to represent the heater used to generate the heat load in the AHRI test method. Similar to the method used to determine the evaporator operating parameters, the heater power is iterated until the compressor is operating 100% of the time. Careful attention is paid to the space temperature to ensure that the compressor is operating at 100%, but is not being outperformed by the heat load. From these three simulations the AWEF can be calculated using the proposed box load equations.

eQuest Simulation Results

Table 157. Small Cooler eQuest Field Simulation Results

Small Cooler eQuest Field Simulation Results		
Box Location	Indoor	Outdoor
Ambient	Indoor	Kansas City
Cooling Energy (MBtu)	27.351	25.299
Evaporator Fan Energy (kWh)	1959.264	1958.776
Defrost Heat Load (kWh)	788.400	788.400
Cooling w/o Unit Cooler (MBtu)	17.976	15.925
Electrical Energy (kWh)	8311.74	8006.85
AWEF	2.16	1.99
Average Runtime (%)	54.59	50.28

Table 158. Small Cooler eQuest Indoor Condenser Method of Test Simulation Results

Proposed Model Load AWEF Calculation: Cooler - Condenser Indoors

Data from eQuest Simulation Results:		
Total Cooling Energy:	60.925	MBtu
Total Electric Consumption:	29640.43	kWh
Power Demand for Supply Fans:	0.24399	kW
Power Demand for Defrost:	1.8	kW
Defrost Operation:	2	hr/day
Assumed Defrost Space Load Percentage:	60	%
Miscellaneous Electric Consumption:	15627.84	kWh
Modified Data from eQuest Simulation Results:		
Total Cooling Rate: Q	6954.909	Btu/hr
Energy Consumption: Ess	1599.611	W
Evaporator Fan Power: Efc,off	243.9901	W
Average Defrost Space Load, Q_dot_{DF}	307.08	Btu/hr
Average Defrost Power, DF_dot	150	W
Calculations According to AHRI 1250 Standard:		
Net Refrigeration Capacity: Qss	5815.335	Btu/hr
Box Load High: BLH	4070.734	Btu/hr
Box Load Low: BLL	2430.81	Btu/hr
Load Factor High: LFH	0.783761	
Load Factor Low: LFL	0.537075	
AWEF:	2.521412	

Table 159. Small Cooler eQuest Outdoor Condenser Method of Test Simulation Results

Proposed Model Load AWEF Calculator: Small Cooler – Condenser Outdoors

Data from eQuest Simulation Results:

Ambient Temperature (°F)	Total Cooling Energy (MBtu)	Total Electric Consumption (kWh)	Miscellaneous Electric Usage (kWh)	Power Demand for Supply Fans (kW)	Power Demand for Defrost (kW)
95	59.489	29348.47	15207.36	0.243990068	1.8
59	63.275	29609.18	16319.88		Daily Defrost Run (hrs)
35	65.113	29919.57	16863		2

Modified Data from eQuest Simulation Results:

Ambient Temperature (°F)	qss (Btu/h)	Ess (W)		
95	5651.407621	1614.281963	Evaporator Fan Power: Efc,off	243.9900685
59	6083.599402	1517.043379	Ave. Defrost Space Load, Q_dot_{DF} (Btu/h)	307.08
35	6293.416754	1490.476027	Average Defrost Power, DF_dot (W)	150

Calculations According to Proposed Box Load Equations:

Temp (°F)	Bin Hour (hr)	qss (Btu/h)	Ess (W)	BLH (Btu/h)	BLL (Btu/h)	LFH	LFL	WLH (Btu/h)	WLL (Btu/h)	BL (Btu)	E (W-h)
100.4	9	5587	1629	3985	2391	0.80	0.55	4460	3073	28690	11948.40838
95	74	5651	1614	3956	2362	0.79	0.54	4441	3052	233776	96382.71383
89.6	257	5716	1600	3927	2334	0.77	0.53	4423	3032	804526	328436.4006
84.2	416	5781	1585	3899	2305	0.76	0.52	4404	3011	1290334	521690.2778
78.8	630	5846	1571	3870	2276	0.75	0.51	4385	2990	1936039	775374.3825
73.4	898	5911	1556	3841	2248	0.74	0.50	4366	2969	2733863	1084797.571
68	737	5976	1541	3813	2219	0.73	0.49	4347	2948	2222574	873959.063
62.6	943	6040	1527	3784	2190	0.72	0.48	4327	2926	2816758	1097832.58
57.2	628	6099	1515	3755	2161	0.71	0.48	4307	2905	1857832	719314.8849
51.8	590	6147	1509	3726	2133	0.70	0.47	4286	2882	1728490	667651.7307
46.4	677	6194	1503	3698	2104	0.69	0.46	4264	2859	1963949	756922.8926
41	576	6241	1497	3669	2075	0.68	0.45	4243	2837	1654428	636321.8035
35.6	646	6288	1491	3640	2047	0.67	0.45	4221	2814	1836956	705187.2412
30.2	534	6335	1485	3612	2018	0.66	0.44	4199	2791	1503156	576045.7392
24.8	322	6383	1479	3583	1989	0.65	0.43	4178	2768	897160	343274.4418
19.4	305	6430	1473	3554	1961	0.65	0.43	4156	2745	841045	321352.2603
14	246	6477	1467	3526	1932	0.64	0.42	4134	2722	671294	256176.049
8.6	189	6524	1461	3497	1903	0.63	0.41	4112	2699	510329	194541.8483
3.2	78	6571	1455	3468	1875	0.62	0.41	4090	2675	208374	79363.26502
-2.2	5	6619	1449	3440	1846	0.61	0.40	4068	2652	13214	5029.148385
	8760								Total	25752786	10051603
										AWEF	2.56

Table 160. Small Cooler Proposed Walk-In Compressor Runtimes

Calculation of Annual Compressor Runtime
Cooresponding to Proposed Model Load
Small Cooler

%t, BLH	50%	C8
%t, BLL	50%	(1 - C8)
%Qss, BLH	70%	C4 + C5
%Qss, BLL	42%	C6 + C7

Condenser Indoors

Run %	55.90%

%1, BLH	61%	C5
%2, BLH	9%	C4
%1, BLL	32%	C7
%2, BLL	9%	C6

Condenser Outdoors

q_{ss} (95F)	5651.407621
q_{ss} (59F)	6083.599402
q_{ss} (35F)	6293.416754
Run %	45.90%

Temp [F]	Bin Hour [hr]	CR (t_j)	CR (t_j) * nj
100.4	9	0.57	5.17
95	74	0.56	41.37
89.6	257	0.54	139.88
84.2	416	0.53	220.45
78.8	630	0.52	325.00
73.4	898	0.50	450.90
68	737	0.49	360.15
62.6	943	0.48	448.41
57.2	628	0.46	290.81
51.8	590	0.45	266.53
46.4	677	0.44	298.27
41	576	0.43	247.44
35.6	646	0.42	270.52
30.2	534	0.41	217.92
24.8	322	0.40	128.02
19.4	305	0.39	118.10
14	246	0.38	92.75
8.6	189	0.37	69.36
3.2	78	0.36	27.85
-2.2	5	0.35	1.74
	8760		4020.63

Table 161. Small Freezer eQuest Field Simulation Results

Small Freezer eQuest Field Simulation Results		
Box Location	Indoor	Outdoor
Ambient	Indoor	Kansas City
Cooling Energy (MBtu)	36.324	40.631
Evaporator Fan Energy (kWh)	3918.456	3918.456
Defrost Heat Load (kWh)	1576.800	1576.800
Cooling w/o Unit Cooler (MBtu)	17.573	21.880
Electrical Energy (kWh)	19877.43	19686.18
AWEF	0.88	1.11
Average Runtime (%)	48.75	50.02

Table 162. Small Freezer eQuest Indoor Condenser Method of Test Simulation Results

Proposed Model Load AWEF Calculation: Freezer - Condenser Indoors

Data from eQuest Simulation Results:		
Total Cooling Energy:	78.068	MBtu
Total Electric Consumption:	47315.98	kWh
Power Demand for Supply Fans:	0.487972	kW
Power Demand for Defrost:	3.6	kW
Defrost Operation:	2	hr/day
Assumed Defrost Space Load Percentage:	60	%
Miscellaneous Electric Consumption:	18431.03	kWh
Modified Data from eQuest Simulation Results:		
Total Cooling Rate: Q	8911.872	Btu/hr
Energy Consumption: Ess	3297.369	W
Evaporator Fan Power: Efc,off	487.9719	W
Average Defrost Space Load, Q_dot_{DF}	614.16	Btu/hr
Average Defrost Power, DF_dot	300	W
Calculations According to AHRI 1250 Standard:		
Net Refrigeration Capacity: Qss	6632.752	Btu/hr
Box Load High: BLH	5637.839	Btu/hr
Box Load Low: BLL	2294.932	Btu/hr
Load Factor High: LFH	0.954114	
Load Factor Low: LFL	0.551243	
AWEF:	1.366521	

Table 163. Small Freezer eQuest Outdoor Condenser Method of Test Simulation Results

Proposed Model Load AWEF Calculator: Small Freezer – Condenser Outdoors

Data from eQuest Simulation Results:

Ambient Temperature (°F)	Total Cooling Energy (MBtu)	Total Electric Consumption (kWh)	Miscellaneous Electric Usage (kWh)	Power Demand for Supply Fans (kW)	Power Demand for Defrost (kW)
95	73.132	47017.03	16985.64	0.487971918	3.6
59	81.826	47238.1	19534.8		Daily Defrost Run (hrs)
35	85.982	48239.43	20752.43		2

Modified Data from eQuest Simulation Results:

Ambient Temperature (°F)	qss (Btu/h)	Ess (W)		
95	6069.281643	3428.240868	Evaporator Fan Power: Efc,off	487.9719178
59	7061.747396	3162.477169	Ave. Defrost Space Load, Q_dot$_{DF}$ (Btu/h)	614.16
35	7536.17662	3137.785388	Average Defrost Power, DF_dot (W)	300

Calculations According to Proposed Box Load Equations:

Temp (°F)	Bin Hour (hr)	qss (Btu/h)	Ess (W)	BLH (Btu/h)	BLL (Btu/h)	LFH	LFL	WLH (Btu/h)	WLL (Btu/h)	BL (Btu)	E (W-h)
100.4	9	5920	3468	5218	2159	0.99	0.59	5851	3464	33193	28191.29404
95	74	6069	3428	5159	2100	0.96	0.57	5837	3436	268578	224529.4289
89.6	257	6218	3388	5100	2041	0.94	0.55	5821	3408	917687	755656.751
84.2	416	6367	3349	5042	1983	0.91	0.53	5803	3378	1461033	1185798.982
78.8	630	6516	3309	4983	1924	0.89	0.51	5784	3348	2175663	1741634.202
73.4	898	6665	3269	4924	1865	0.86	0.50	5764	3316	3048501	2408574.245
68	737	6814	3229	4866	1807	0.84	0.48	5742	3283	2458707	1918597.094
62.6	943	6963	3189	4807	1748	0.82	0.47	5719	3250	3090622	2383542.029
57.2	628	7097	3161	4748	1689	0.80	0.45	5692	3214	2021388	1547973.453
51.8	590	7204	3155	4690	1631	0.79	0.44	5660	3176	1864462	1429957.274
46.4	677	7311	3150	4631	1572	0.77	0.43	5628	3137	2099674	1613589.659
41	576	7418	3144	4572	1513	0.75	0.42	5595	3097	1752638	1350283.952
35.6	646	7524	3138	4514	1455	0.74	0.41	5562	3057	1927734	1489693.063
30.2	534	7631	3133	4455	1396	0.72	0.40	5528	3017	1562186	1211516.591
24.8	322	7738	3127	4396	1337	0.71	0.38	5493	2976	923102	718834.228
19.4	305	7845	3122	4338	1279	0.70	0.37	5458	2935	856474	670065.4504
14	246	7951	3116	4279	1220	0.68	0.36	5423	2893	676364	531931.3604
8.6	189	8058	3111	4220	1161	0.67	0.35	5386	2851	508557	402293.3736
3.2	78	8165	3105	4162	1103	0.66	0.34	5350	2809	205305	163452.9197
-2.2	5	8272	3100	4103	1044	0.64	0.33	5313	2766	12867	10316.69235
	8760								Total	27864735	21786432
										AWEF	1.28

Table 164. Small Freezer Proposed Walk-In Compressor Runtimes

Calculation of Annual Compressor Runtime
Cooresponding to Proposed Model Load
Small Freezer

%t, BLH	50%	C8
%t, BLL	50%	(1 - C8)
%Qss, BLH	85%	C4 + C5
%Qss, BLL	35%	C6 + C7

Condenser Indoors

Run %	59.80%

%1, BLH	67%	C5
%2, BLH	18%	C4
%1, BLL	17%	C7
%2, BLL	18%	C6

Condenser Outdoors

q.. (95F)	6069.281643
q.. (59F)	7061.747396
q.. (35F)	7536.17662
Run %	44.70%

Temp [F]	Bin Hour [hr]	CR (t_j)	CR (t_j) * nj
100.4	9	0.62	5.61
95	74	0.60	44.25
89.6	257	0.57	147.58
84.2	416	0.55	229.47
78.8	630	0.53	333.90
73.4	898	0.51	457.41
68	737	0.49	360.85
62.6	943	0.47	443.90
57.2	628	0.45	284.25
51.8	590	0.44	256.80
46.4	677	0.42	283.39
41	576	0.40	231.89
35.6	646	0.39	250.13
30.2	534	0.37	198.86
24.8	322	0.36	115.32
19.4	305	0.34	105.04
14	246	0.33	81.47
8.6	189	0.32	60.18
3.2	78	0.31	23.87
-2.2	5	0.29	1.47
	8760		3915.64

Table 165. Large Cooler eQuest Field Simulation Results

Large Cooler eQuest Field Simulation Results		
Box Location	Indoor	Outdoor
Ambient	Indoor	Kansas City
Cooling Energy (MBtu)	781.665	749.835
Evaporator Fan Energy (kWh)	39187.453	39187.453
Defrost Heat Load (kWh)	15768.000	15768.000
Cooling w/o Unit Cooler (MBtu)	594.149	562.319
Electrical Energy (kWh)	181383.58	166834.94
AWEF	3.28	3.37
Average Runtime (%)	54.59	50.28

Table 166. Large Cooler eQuest Indoor Condenser Method of Test Simulation Results

Proposed Model Load AWEF Calculation: Cooler - Condenser Indoors

Data from eQuest Simulation Results:		
Total Cooling Energy:	1487.559	MBtu
Total Electric Consumption:	630728.3	kWh
Power Demand for Supply Fans:	4.880029	kW
Power Demand for Defrost:	36	kW
Defrost Operation:	2	hr/day
Assumed Defrost Space Load Percentage:	60	%
Miscellaneous Electric Consumption:	391396.8	kWh
Modified Data from eQuest Simulation Results:		
Total Cooling Rate: Q	169812.7	Btu/hr
Energy Consumption: Ess	27320.95	W
Evaporator Fan Power: Efc,off	4880.029	W
Average Defrost Space Load, Q_dot_{DF}	6141.6	Btu/hr
Average Defrost Power, DF_dot	3000	W
Calculations According to AHRI 1250 Standard:		
Net Refrigeration Capacity: Qss	147020.4	Btu/hr
Box Load High: BLH	102914.3	Btu/hr
Box Load Low: BLL	61454.53	Btu/hr
Load Factor High: LFH	0.768044	
Load Factor Low: LFL	0.514732	
AWEF:	3.689806	

Table 167. Large Cooler eQuest Outdoor Condenser Method of Test Simulation Results

Proposed Model Load AWEF Calculator: Large Cooler – Condenser Outdoors

Data from eQuest Simulation Results:

Ambient Temperature (˚F)	Total Cooling Energy (MBtu)	Total Electric Consumption (kWh)	Miscellaneous Electric Usage (kWh)	Power Demand for Supply Fans (kW)	Power Demand for Defrost (kW)
95	1416.955	614082.56	370705.75	4.88002911	36
59	1504.452	626651	396398.44		Daily Defrost Run (hrs)
35	1553.731	637896.38	410861.53		2

Modified Data from eQuest Simulation Results:

Ambient Temperature (˚F)	qss (Btu/h)	Ess (W)		
95	138960.5946	27782.74087	Evaporator Fan Power: Efc,off	4880.02911
59	148948.8366	26284.53881	Ave. Defrost Space Load, Q_dot$_{dr}$ (Btu/h)	6141.6
35	154574.2932	25917.22032	Ave Defrost Power, DF_dot (W)	3000

Calculations According to Proposed Box Load Equations:

Temp (˚F)	Bin Hour (hr)	qss (Btu/h)	Ess (W)	BLH (Btu/h)	BLL (Btu/h)	LFH	LFL	WLH (Btu/h)	WLL (Btu/h)	BL (Btu)	E (W-h)
100.4	9	137462	28007	97978	58791	0.78	0.53	107722	72769	705459	207570.6438
95	74	138961	27783	97272	58086	0.77	0.52	107218	72224	5748244	1677379.199
89.6	257	140459	27558	96567	57380	0.76	0.51	106709	71676	19782217	5726150.427
84.2	416	141957	27333	95862	56675	0.75	0.50	106198	71125	31727590	9111854.775
78.8	630	143455	27109	95156	55969	0.74	0.49	105682	70571	47604615	13567234.99
73.4	898	144954	26884	94451	55264	0.73	0.48	105163	70014	67222050	19015939.55
68	737	146452	26659	93746	54559	0.71	0.47	104641	69454	54650136	15348033.6
62.6	943	147950	26434	93040	53853	0.70	0.47	104115	68892	69260322	19314957.47
57.2	628	149371	26257	92335	53148	0.69	0.46	103581	68324	45681615	12673650.39
51.8	590	150636	26174	91630	52443	0.68	0.45	103033	67746	42501276	11771040.35
46.4	677	151902	26092	90924	51737	0.67	0.44	102483	67167	48290882	13353829.48
41	576	153168	26009	90219	51032	0.67	0.43	101930	66586	40680192	11233815.29
35.6	646	154434	25926	89513	50327	0.66	0.43	101376	66003	45168300	12458278.38
30.2	534	155699	25844	88808	49621	0.65	0.42	100819	65418	36960599	10184045.68
24.8	322	156965	25761	88103	48916	0.64	0.41	100260	64831	22059975	6073249.917
19.4	305	158231	25678	87397	48210	0.63	0.41	99698	64242	20680182	5689626.247
14	246	159497	25596	86692	47505	0.62	0.40	99135	63652	16506234	4539100.912
8.6	189	160762	25513	85987	46800	0.61	0.39	98570	63061	12548305	3449684.552
3.2	78	162028	25431	85281	46094	0.60	0.39	98002	62467	5123647	1408402.731
-2.2	5	163294	25348	84576	45389	0.60	0.38	97433	61872	324912	89319.97266
	8760								Total	633226751	176893165
										AWEF	3.58

Table 168. Large Cooler Proposed Walk-In Compressor Runtimes

Calculation of Annual Compressor Runtime
Cooresponding to Proposed Model Load
Large Cooler

%t, BLH	50%	C8
%t, BLL	50%	(1 - C8)
%Qss, BLH	70%	C4 + C5
%Qss, BLL	42%	C6 + C7

Condenser Indoors

Run %	55.90%

%1, BLH	61%	C5
%2, BLH	9%	C4
%1, BLL	32%	C7
%2, BLL	9%	C6

Condenser Outdoors

q.. (95F)	138960.5946
q.. (59F)	148948.8366
q.. (35F)	154574.2932
Run %	45.99%

Temp [F]	Bin Hour [hr]	CR (t_j)	CR (t_j) * nj
100.4	9	0.57	5.16
95	74	0.56	41.37
89.6	257	0.54	139.98
84.2	416	0.53	220.75
78.8	630	0.52	325.65
73.4	898	0.50	452.10
68	737	0.49	361.33
62.6	943	0.48	450.15
57.2	628	0.46	291.99
51.8	590	0.45	267.41
46.4	677	0.44	299.05
41	576	0.43	247.91
35.6	646	0.42	270.84
30.2	534	0.41	218.03
24.8	322	0.40	128.00
19.4	305	0.39	118.01
14	246	0.38	92.61
8.6	189	0.37	69.21
3.2	78	0.36	27.77
-2.2	5	0.35	1.73
	8760		4029.03

Table 169. Large Freezer eQuest Field Simulation Results

Large Freezer eQuest Field Simulation Results		
Box Location	Indoor	Outdoor
Ambient	Indoor	Kansas City
Cooling Energy (MBtu)	537.634	572.046
Evaporator Fan Energy (kWh)	39180.281	39180.281
Defrost Heat Load (kWh)	15768.000	15768.000
Cooling w/o Unit Cooler (MBtu)	350.143	384.555
Electrical Energy (kWh)	225431.34	210007.44
AWEF	1.55	1.83
Average Runtime (%)	62.50	60.27

Table 170. Large Freezer eQuest Indoor Condenser Method of Test Simulation Results

Proposed Model Load AWEF Calculation: Freezer - Condenser Indoors

Data from eQuest Simulation Results:		
Total Cooling Energy:	864.542	MBtu
Total Electric Consumption:	486985.6	kWh
Power Demand for Supply Fans:	4.87921	kW
Power Demand for Defrost:	36	kW
Defrost Operation:	2	hr/day
Assumed Defrost Space Load Percentage:	60	%
Miscellaneous Electric Consumption:	208838.4	kWh
Modified Data from eQuest Simulation Results:		
Total Cooling Rate: Q	98692.01	Btu/hr
Energy Consumption: Ess	31751.96	W
Evaporator Fan Power: Efc,off	4879.21	W
Average Defrost Space Load, Q_dot_{DF}	6141.6	Btu/hr
Average Defrost Power, DF_dot	3000	W
Calculations According to AHRI 1250 Standard:		
Net Refrigeration Capacity: Qss	75902.54	Btu/hr
Box Load High: BLH	64517.16	Btu/hr
Box Load Low: BLL	26262.28	Btu/hr
Load Factor High: LFH	0.943341	
Load Factor Low: LFL	0.53	
AWEF:	1.640064	

Table 171. Large Freezer eQuest Outdoor Condenser Method of Test Simulation Results

Proposed Model Load AWEF Calculator: Large Freezer – Condenser Outdoors

Data from eQuest Simulation Results:

Ambient Temperature (°F)	Total Cooling Energy (MBtu)	Total Electric Consumption (kWh)	Miscellaneous Electric Usage (kWh)	Power Demand for Supply Fans (kW)	Power Demand for Defrost (kW)
95	804.144	472758.16	191143.16	4.879210388	36
59	936.239	493924.31	229862.58		Daily Defrost Run (hrs)
35	977.72	501617.84	242038.77		2

Modified Data from eQuest Simulation Results:

Ambient Temperature (°F)	qss (Btu/h)	Ess (W)		
95	69007.79443	32147.83105	Evaporator Fan Power: Efc,off	4879.210388
59	84087.13233	30144.03311	Ave. Defrost Space Load, Q_dot$_{DF}$ (Btu/h)	6141.6
35	88822.4063	29632.31393	Average Defrost Power, DF_dot (W)	3000

Calculations According to Proposed Box Load Equations:

Temp (°F)	Bin Hour (hr)	qss (Btu/h)	Ess (W)	BLH (Btu/h)	BLL (Btu/h)	LFH	LFL	WLH (Btu/h)	WLL (Btu/h)	BL (Btu)	E (W-h)
100.4	9	66746	32448	59324	24544	0.98	0.57	65721	37884	377403	263484.6784
95	74	69008	32148	58657	23877	0.95	0.54	65616	37596	3053733	2092097.216
89.6	257	71270	31847	57990	23210	0.92	0.52	65483	37289	10434105	7022098.274
84.2	416	73532	31547	57323	22543	0.89	0.50	65323	36963	16611963	10993677.66
78.8	630	75793	31246	56656	21876	0.86	0.48	65138	36621	24737311	16114818.11
73.4	898	78055	30946	55989	21209	0.83	0.46	64930	36264	34661492	22248655.84
68	737	80317	30645	55321	20542	0.81	0.45	64700	35892	27955526	17698397.26
62.6	943	82579	30344	54654	19875	0.78	0.43	64451	35506	35140409	21963597.18
57.2	628	84442	30106	53987	19207	0.76	0.42	64133	35081	22983202	14254886.12
51.8	590	85508	29991	53320	18540	0.75	0.40	63707	34595	21198952	13164907.27
46.4	677	86573	29875	52653	17873	0.73	0.39	63275	34105	23873320	14851610.67
41	576	87639	29760	51986	17206	0.72	0.38	62839	33611	19927508	12424623.45
35.6	646	88704	29645	51319	16539	0.70	0.37	62398	33114	21918353	13703304.25
30.2	534	89769	29530	50652	15872	0.69	0.36	61953	32614	17762073	11140935.58
24.8	322	90835	29415	49985	15205	0.68	0.35	61503	32110	10495680	6608135.714
19.4	305	91900	29300	49318	14538	0.66	0.34	61049	31603	9738116	6157722.881
14	246	92966	29185	48651	13871	0.65	0.33	60590	31093	7690260	4886598.858
8.6	189	94031	29069	47984	13204	0.64	0.33	60128	30580	5782302	3694352.042
3.2	78	95097	28954	47317	12537	0.63	0.32	59662	30064	2334319	1500476.348
-2.2	5	96162	28839	46650	11870	0.62	0.31	59192	29545	146301	94670.66368
	8760								Total	316822328	200879050
										AWEF	1.58

Table 172. Large Freezer Proposed Walk-In Compressor Runtimes

Calculation of Annual Compressor Runtime
Cooresponding to Proposed Model Load
Large Freezer

%t, BLH	50%	C8
%t, BLL	50%	(1 - C8)
%Qss, BLH	85%	C4 + C5
%Qss, BLL	35%	C6 + C7

Condenser Indoors

Run %	59.80%

%1, BLH	67%	C5
%2, BLH	18%	C4
%1, BLL	17%	C7
%2, BLL	18%	C6

Condenser Outdoors

q.. (95F)	69007.79443
q.. (59F)	84087.13233
q.. (35F)	88822.4063
Run %	42.69%

Temp [F]	Bin Hour [hr]	CR (t_j)	CR (t_j) * nj
100.4	9	0.63	5.65
95	74	0.60	44.25
89.6	257	0.57	146.40
84.2	416	0.54	225.92
78.8	630	0.52	326.38
73.4	898	0.49	444.06
68	737	0.47	348.06
62.6	943	0.45	425.54
57.2	628	0.43	270.90
51.8	590	0.41	243.38
46.4	677	0.39	267.14
41	576	0.38	217.49
35.6	646	0.36	233.45
30.2	534	0.35	184.73
24.8	322	0.33	106.65
19.4	305	0.32	96.73
14	246	0.30	74.71
8.6	189	0.29	54.97
3.2	78	0.28	21.72
-2.2	5	0.27	1.33
	8760		3739.46

Simulation results are summarized in Table 173 and Table 174. AWEF calculations via the Proposed 1250 equations gave similar results to the hourly simulations. Variations between the two methods may be attributed to differences in the calculation methods, the derivation of the Proposed 1250 equations, and limitations in equipment available to build the refrigeration systems. Intuitively, an hourly simulation, calculating energy usage on fifteen minute intervals, will be more representative of actual refrigeration system operation than the Proposed 1250 calculation which uses four laboratory tests and a set of equations to arrive at the refrigeration system AWEF. The simplicity in the Proposed 1250 calculation of the AWEF results in a variance to the performance rating obtained via an hourly simulation of the refrigeration system. In the development of the Proposed 1250 equations for walk-in coolers, for example, coefficients were developed for a small and large cooler, and then the coefficients were averaged to arrive at the walk-in cooler box load coefficients. This adds an inherent variance between the 1250 Proposed equation calculations and results of simulations of the walk-ins used to develop the 1250 Proposed equations.

When selecting refrigeration equipment for a given walk-in load, variation is added to the Proposed 1250 correlation. As the heat load decreases, there is more variation on a percentage basis between the capacity of the refrigeration equipment and the actual load simply due to the larger percentage gap in available component refrigeration capacities at lower loads.

Table 173. eQuest Simulation AWEF Calculations

	Box In, Condenser In	Proposed 1250, Condenser In	Box Out, Condenser Out, Kansas City	Proposed 1250, Condenser Out
Small Freezer	0.88	1.37	1.11	1.28
Small Cooler	2.16	2.52	1.99	2.56
Large Freezer	1.55	1.64	1.83	1.58
Large Cooler	3.28	3.69	3.37	3.58

Table 174. eQuest Simulation Compressor Runtimes

	Box In, Condenser In	Proposed 1250, Condenser In	Box Out, Condenser Out, Kansas City	Proposed 1250, Condenser Out
Small Freezer	48.75	59.80	50.02	44.70
Small Cooler	45.84	55.90	40.19	45.90
Large Freezer	62.50	59.80	60.27	42.69
Large Cooler	54.59	55.90	50.28	45.99

AWEF and compressor runtime variations are displayed in Table 175. Variation in the AWEF values primarily trend with the compressor runtime, suggesting that if the selected refrigeration equipment was oversized precisely the same for all cases, the AWEF variation would be minimized. Comparisons of the percentage variation of AWEF and runtimes obtained via the Standard 1250 Equations and the Proposed 1250 Equations are made in Table 176. AWEF and compressor runtime variations had a higher standard deviation using the proposed method. Average variations were smaller for the compressor runtime but larger for the AWEF values determined via the proposed method. The higher standard deviation is due to a large variation in the proposed coefficients developed in Table 148, as compared with those developed by the AHRI Load Spreadsheet (AHRI

2009a). This not only creates a gap between the correlation of a large or small walk-in with the Standard but also between an indoor and outdoor condensing system. The proposed method does better at targeting the actual compressor runtime but not the AWEF values. AWEF values vary significantly due to a combination of equipment sizing and the calculation method.

Table 175. Proposed 1250 vs. Hourly Weather Simulations (% Increase)
(1250 – Hourly) / Hourly

	AWEF		Compressor Runtimes	
	Condenser In	Condenser Out	Condenser In	Condenser Out
Small Freezer	+ 55.7	+15.3	+ 22.7	- 10.6
Small Cooler	+ 16.7	+ 28.6	+ 21.9	+ 14.2
Large Freezer	+ 5.8	- 13.7	- 4.3	- 29.2
Large Cooler	+ 12.5	+ 6.2	+ 2.4	- 8.5

Table 176. Comparison of AHRI (Becker et al. 2011) to Proposed Percentage Variations of AWEF and Compressor Runtimes, 1250 to Average Climate Zone Simulations
(1250 – Ave Climate Zone Simulation) / Ave Climate Zone Simulation

	AWEF		Compressor Runtimes	
	Condenser In	Condenser Out	Condenser In	Condenser Out
Small Freezer, AHRI 1250	+ 21.3	+ 10.7	- 3.3	- 12.0
Small Freezer, Proposed 1250	+ 55.7	+15.3	+ 22.7	- 10.6
Small Cooler, AHRI 1250	- 27.9	- 28.1	- 45.8	- 49.1
Small Cooler, Proposed 1250	+ 16.7	+ 28.6	+ 21.9	+ 14.2
Large Freezer, AHRI 1250	- 2.5	- 4.1	- 5.0	- 29.6
Large Freezer, Proposed 1250	+ 5.8	- 13.7	- 4.3	- 29.2
Large Cooler, AHRI 1250	- 10.9	- 17.5	- 41.7	- 49.8
Large Cooler, Proposed 1250	+ 12.5	+ 6.2	+ 2.4	- 8.5
Average AHRI 1250	- 5.0	- 9.8	- 24.0	- 35.1
Average Proposed 1250	+ 22.7	+ 9.1	+ 10.7	- 8.5

Plotting the 1250 AWEF calculation values vs. the hourly simulation results, obtained by Becker et al. (Becker et al. 2011) and the current work, shows very good correlation between the calculation method and a simulated year of operation (Figure 9 and Figure 10). The correlation between the calculation method and the hourly simulations was improved for the coolers while the freezer results gave marginally worse correlation in most cases.

Figure 9. AHRI or Proposed 1250 Calculated AWEF vs. Hourly Simulation AWEF Comparison for (Becker et al. 2011) and the Current Work, Condenser Indoors

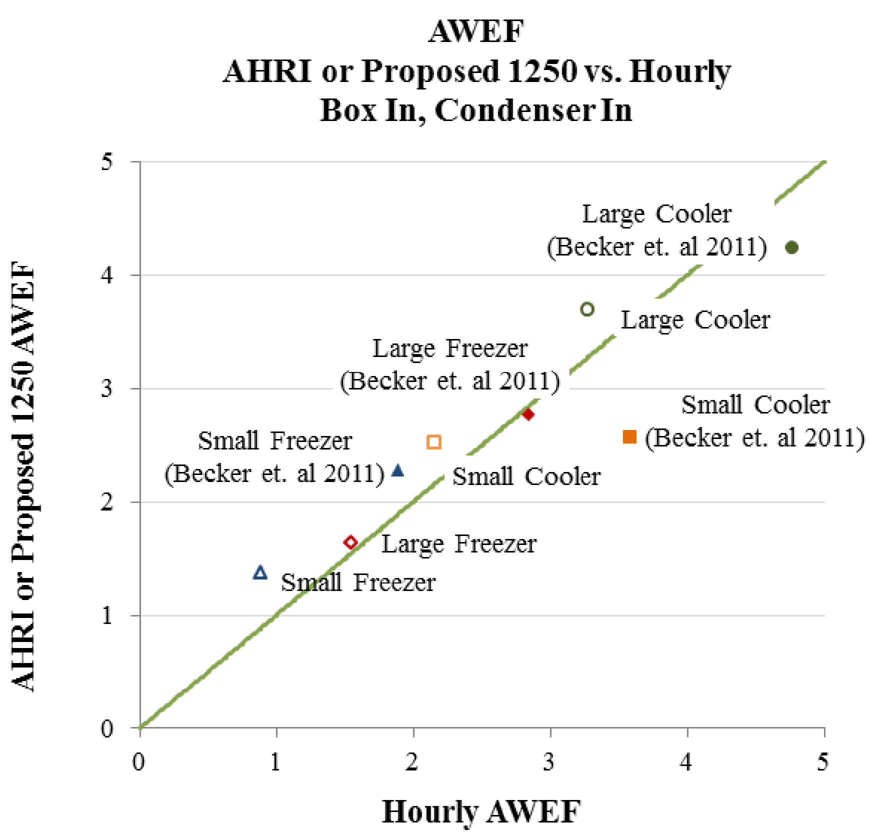
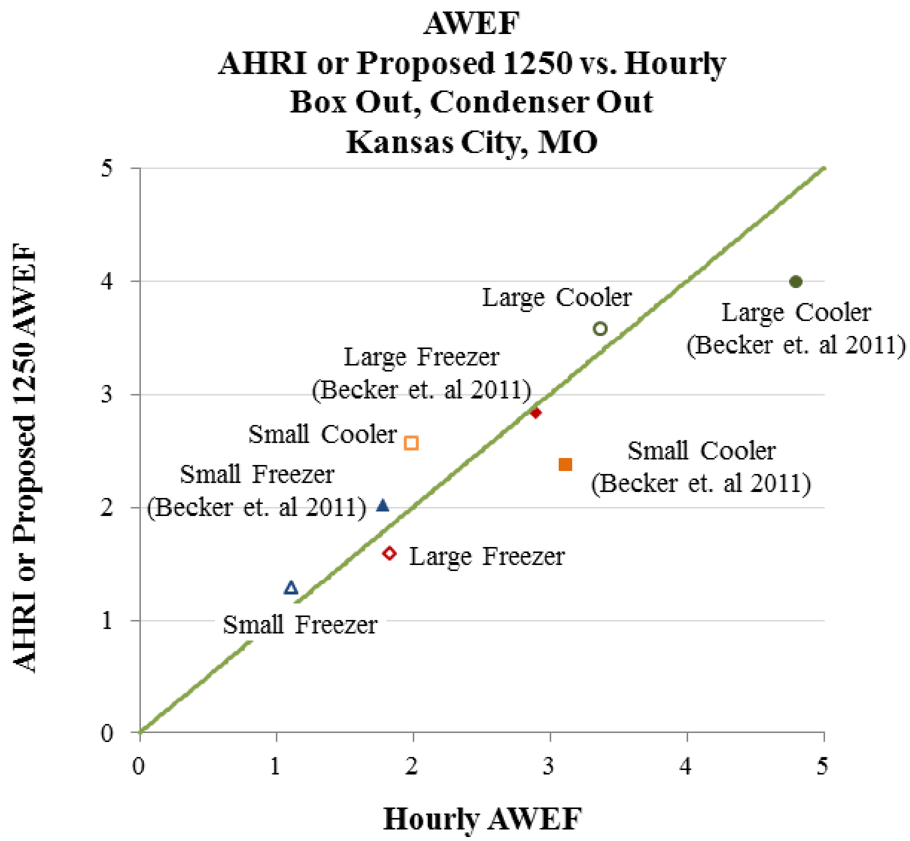


Figure 10. AHRI or Proposed 1250 Calculated AWEF vs. Hourly Simulation AWEF Comparison for (Becker et al. 2011) and the Current Work, Condenser Outdoors



The relationship between the variations in the AWEF and the compressor runtimes determined via the Standard 1250 calculation and the hourly simulations are consistent between the results obtained by Becker et al. (Becker et al. 2011) and those obtained with the Proposed 1250 calculation (Figure 11 and Figure 12). A trend line of this data does not pass through the origin of the chart, however. Assuming that the compressor runtime variance is zero, the AWEF variance would be + 17.7% for the outdoor condenser or +14.4% for the indoor condenser system. Essentially, both the indoor and outdoor calculation methods vary from actual by the same amount.

Figure 11. Percent Difference in AWEF vs. Percent Difference in Runtime for Proposed 1250 and AHRI 1250 (Becker et al. 2011) vs. Hourly Simulations, Box Out Condenser Out

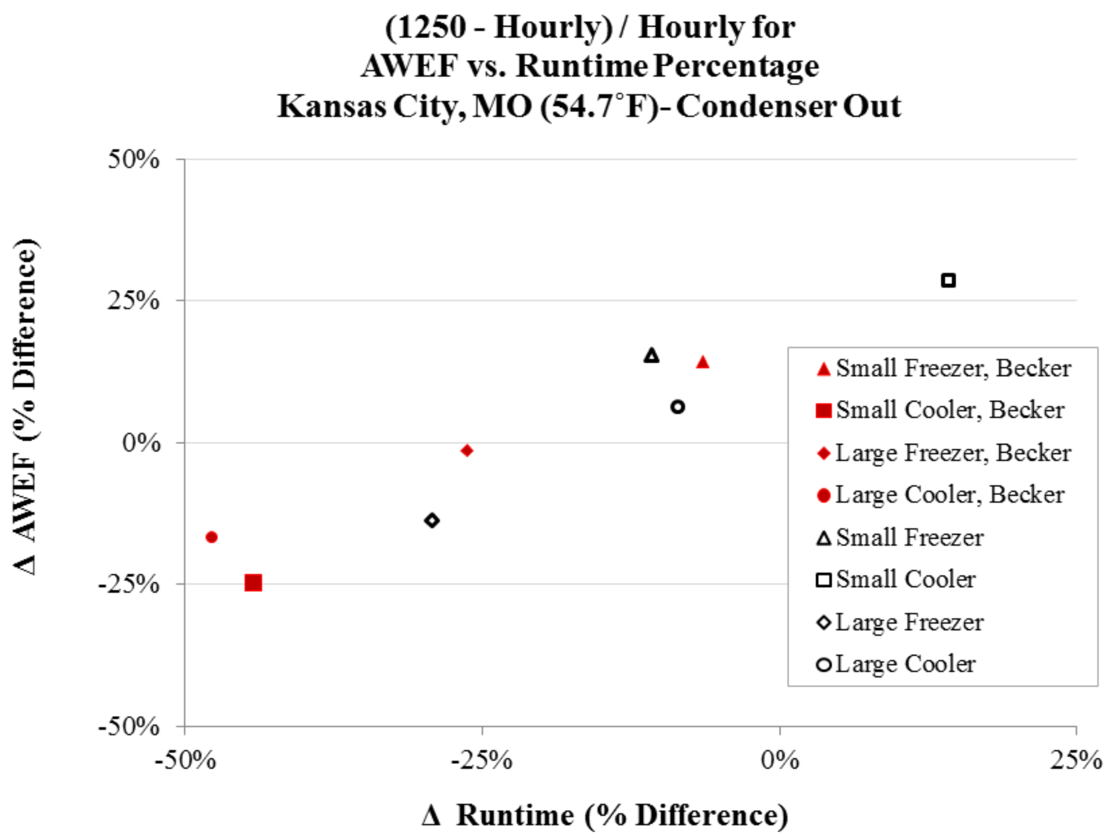
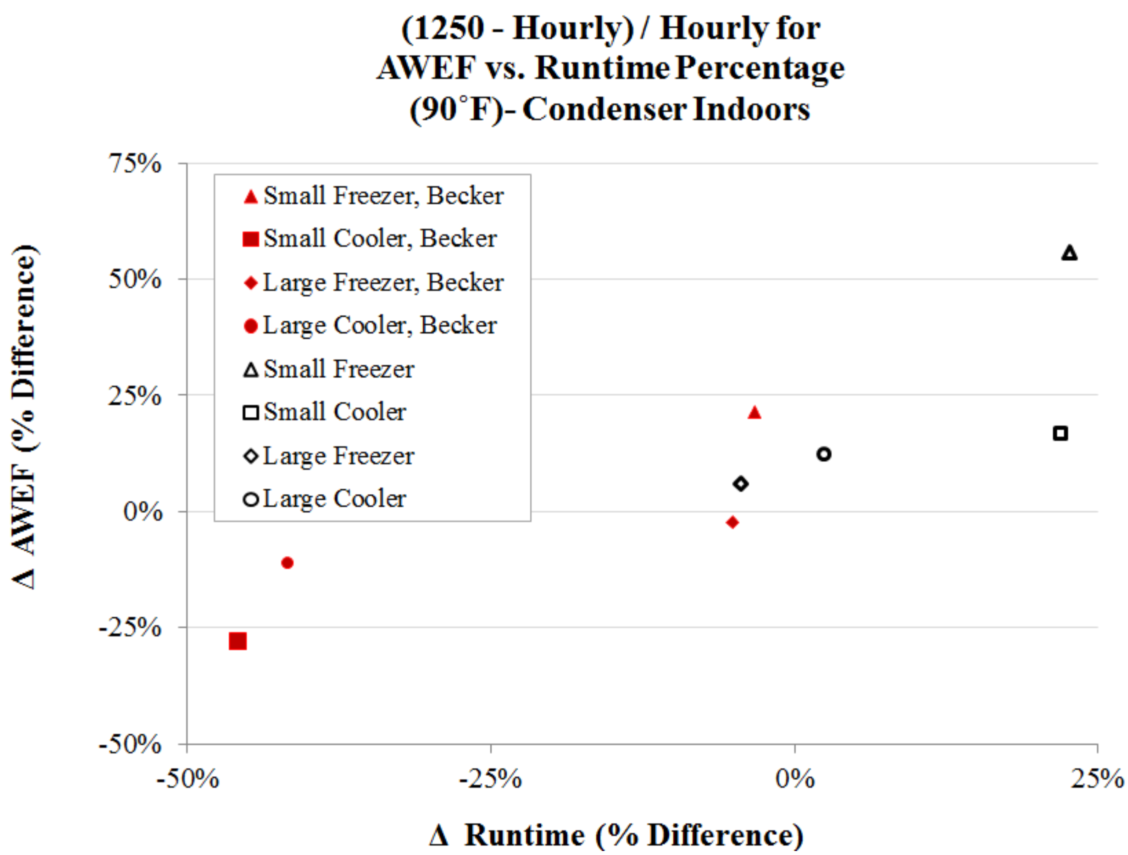
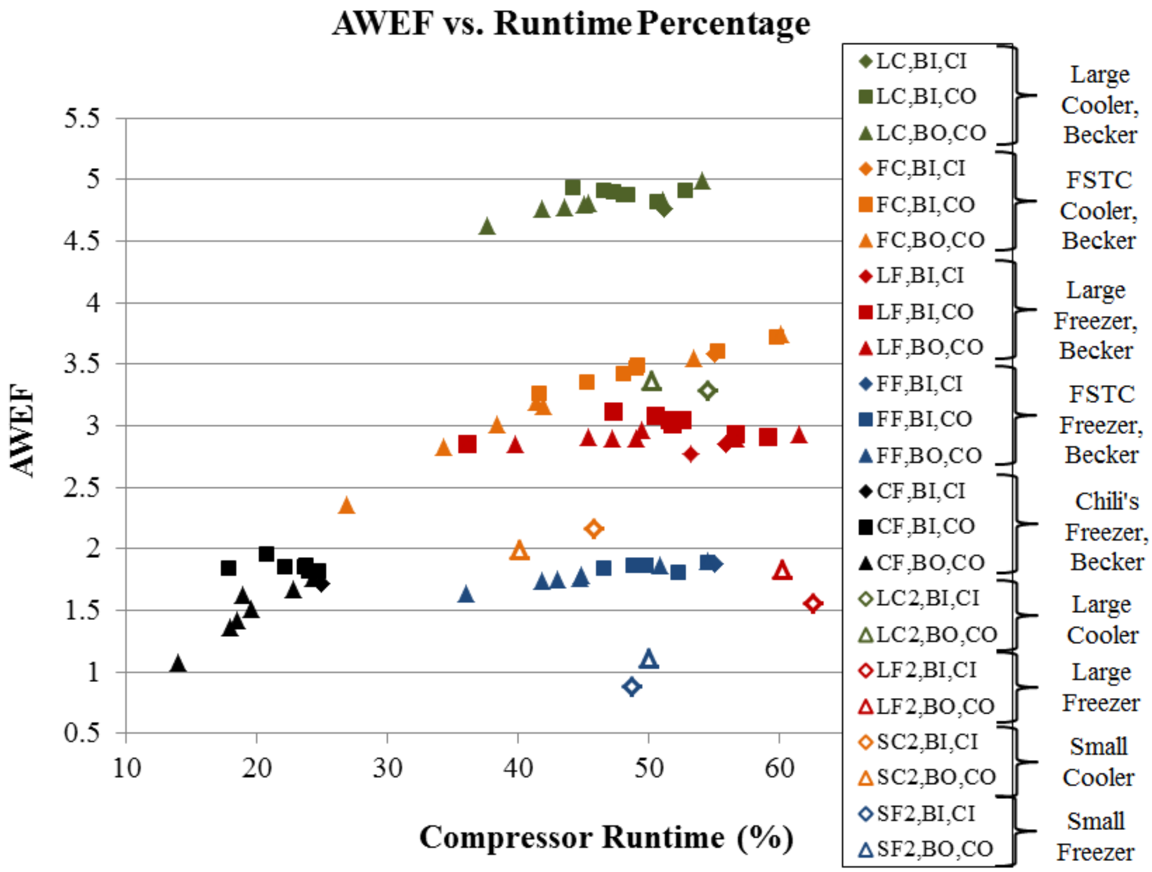


Figure 12. Percent Difference in AWEF vs. Percent Difference in Runtime for Proposed 1250 and AHRI 1250 (Becker et al. 2011) vs. Hourly Simulations, Box In Condenser In



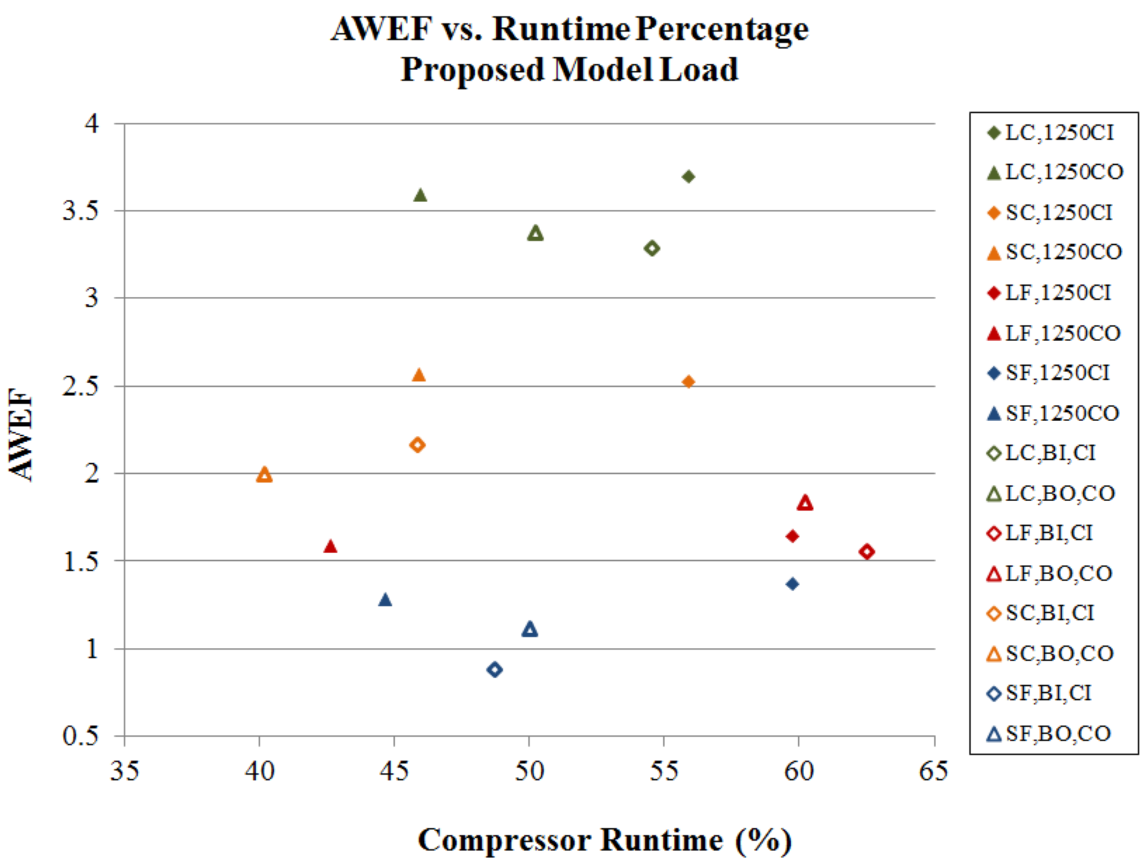
Simulations completed by this work and Becker et al. (Becker et al. 2011) are displayed in Figure 13. These simulations are all hourly simulations which were targeted to represent actual refrigeration system operation. The current work AWEF values have similar trending to Becker et al. results. The current simulations yielded AWEFs of a smaller magnitude, primarily due to the inclusion of defrost systems for the coolers and freezers analyzed. These AWEF values are therefore more representative of what will actually be observed in practice.

Figure 13. AWEF vs. Runtime for eQuest Hourly Simulations of Current Work and (Becker et al. 2011)



AWEF and compressor runtimes are presented in Figure 14 for the Proposed 1250 calculations and hourly simulations. Through analysis of this graph, one can see the large discrepancy between the gap in the condenser in and condenser out compressor runtimes for the Proposed 1250 calculation and the hourly simulations. The compressor runtime equation (Equation 138) is primarily a function of the box load calculations. The only variable changing from the box load calculations for the indoor and outdoor cases is the ambient temperature.

Figure 14. AWEF vs. Runtime Percentage for Proposed 1250 Calculation and Hourly Simulations



CHAPTER 7

PROPOSED 1250 CALCULATION METHOD DISCUSSION

The results of this study and those obtained by Becker et al. (Becker et al. 2011) show similar correlation between the AWEF calculated by the Standard equations, utilized by either study, and the hourly simulations. Using Figure 11 and Figure 12 it was determined that the difference between the hourly simulated AWEF and the Standard calculated AWEF was + 14 to 17% at a compressor runtime difference of zero. Assuming that this is due to a calculation issue with the AWEF, either the box load calculation is too high or the electrical consumption calculation is too low. The box load and energy usage associated with each simulation set was analyzed using Table 177 and Table 178. When this information is plotted (Figure 15) clear relationships are shown. Variation of the calculated AWEFs' correlation with hourly simulations essentially goes to zero when the box load variation equals zero.

As a general rule, the box load determined via the proposed calculation method is larger than the box load obtained via hourly weather data simulations. Some variation is expected from the Standard calculation because the box load equations that define typical cooler and freezer operation were developed by averaging box load definitions for large and small coolers and freezers. Averaging the variations for the large and small units better represents the discrepancy between the Proposed 1250 calculation and hourly weather data simulations.

Table 177. Comparison of the Box Load and Electrical Usage via Hourly Simulations and Standard 1250 Calculations

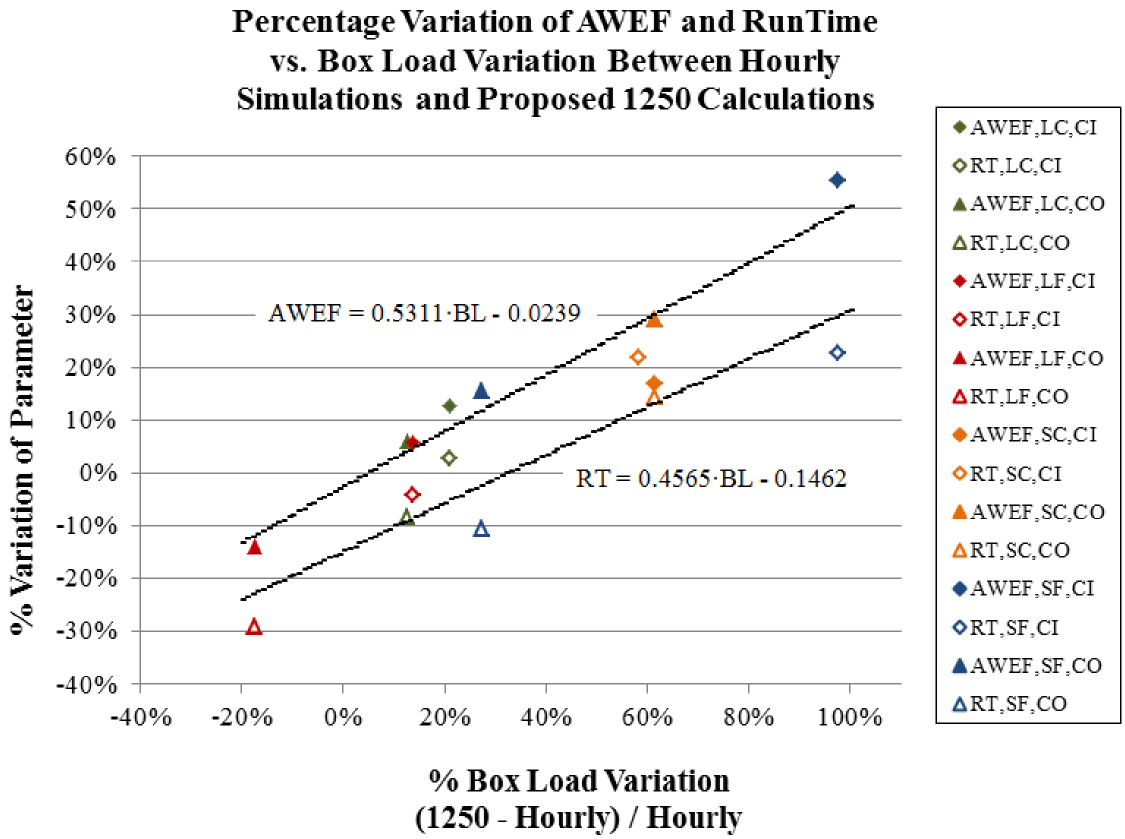
	Box Load (MBtu)	Energy Use (kW-h)	AWEF (Btu/W-h)	Runtime (%)
Box Out, Condenser Out				
Small Freezer, Proposed 1250	27.865	21786.43	1.28	44.70
Small Freezer, Hourly Simulation	21.880	19686.18	1.11	50.02
Small Cooler, Proposed 1250	25.753	10051.60	2.56	45.90
Small Cooler, Hourly Simulation	15.925	8006.85	1.99	40.19
Large Freezer, Proposed 1250	316.822	200879.05	1.58	42.69
Large Freezer, Hourly Simulation	384.555	210007.44	1.83	60.27
Large Cooler, Proposed 1250	633.227	176893.17	3.58	45.99
Large Cooler, Hourly Simulation	562.319	166834.94	3.37	50.28
Box In, Condenser In				
Small Freezer, Proposed 1250	34.746	25426.27	1.37	59.80
Small Freezer, Hourly Simulation	17.573	19877.43	0.88	48.75
Small Cooler, Proposed 1250	28.477	11300.30	2.52	55.90
Small Cooler, Hourly Simulation	17.976	8311.74	2.16	45.84
Large Freezer, Proposed 1250	397.614	242438.07	1.64	59.80
Large Freezer, Hourly Simulation	350.143	225431.34	1.55	62.50
Large Cooler, Proposed 1250	719.935	195114.72	3.69	55.90
Large Cooler, Hourly Simulation	594.149	181383.58	3.28	54.59

Table 178. Percentage Variation of the Box Load and Electrical Usage via Hourly Simulations and Proposed 1250 Calculations Compared to Variations in the AWEF and Runtime

(1250 – Hourly) / Hourly				
	Box Load (MBtu)	Energy Use (kW-h)	AWEF (Btu/W-h)	Runtime (%)
Box Out, Condenser Out				
Small Freezer	+ 27.35	+ 10.67	+ 15.32	- 10.64
Small Cooler	+ 61.71	+ 25.54	+ 28.75	+ 14.21
Large Freezer	- 17.61	- 4.35	- 13.66	- 29.17
Large Cooler	+ 12.61	+ 6.03	+ 6.23	- 8.53
Box In, Condenser In				
Small Freezer	+ 97.72	+ 27.92	+ 55.29	+ 22.67
Small Cooler	+ 58.42	+ 35.96	+ 16.67	+ 21.95
Large Freezer	+ 13.56	+ 7.54	+ 5.81	- 4.32
Large Cooler	+ 21.17	+ 7.57	+ 12.49	+ 2.40
Average Percent Variations				
Freezer	+ 30.25	+ 10.45	+ 15.69	- 5.36
Cooler	+ 38.48	+ 18.77	+ 16.03	+ 7.51

Runtime variation was minimal for the Proposed 1250 calculation, and AWEF variation was similar for both walk-in freezers and coolers at around 16%. The discrepancy between the AWEF and runtime variation was graphically displayed in Figure 15, with AWEF variation approaching zero at a box load variation of 4.5%. Runtime variation approaches zero at a box load variation of 32.0%.

Figure 15. Percentage Variation of AWEF and Runtime vs. Box Load Variation Between Hourly Simulations and Proposed 1250 Calculations
(1250 – Hourly) / Hourly



CHAPTER 8

CONCLUSIONS

Walk-in coolers and freezers consume 26.2% of commercial refrigeration energy use, 1.5% of commercial building energy use, and 0.2% of the nation's energy use annually (Arias 2005; Arias and Lundqvist 2006; Christensen and Bertilsen 2004; DOE 2004; Fricke and Becker 2011; Goetzler et al. 2009; Henderson and Khattar 1999; Huan 2008; Sugiartha et al. 2009; Walker 2001; Walker et al. 1990; Westphalen et al. 1996). This is equivalent to 76 trillion Btu/yr of energy use or 242 trillion Btu/yr of primary energy use (Patel et al. 1993). To improve the energy efficiency of the walk-in construction and operation, the federal government and some state governments have enacted regulations dictating the minimum design criteria.

The ANSI/AHRI 1250/1251 Standards (AHRI 2009b; AHRI 2009c) are referenced by the federal regulations (DOE 2011). Because these standards are being applied to legislature, it is essential that they appropriately represent refrigeration equipment performance. Similar to the Environmental Protection Agency's MPG rating for vehicles, the ANSI/AHRI 1250/1251 AWEF rating for refrigeration equipment is a relative performance rating based on some assumptions. For rating mileage, certain driving conditions, driving methods, and auxiliary loads are assumed. For rating refrigeration equipment performance, certain operating conditions, operating methods, and walk-in construction details are assumed. Both of these standard ratings vary from the actual results based on the correlation of the assumed and actual conditions. Regardless of how precise these rated values are, they should accurately portray which equipment is better from an energy efficiency perspective. Additionally, they should portray this within a small range of error from actual case to allow analysis of the integration of this equipment into the building, for operating cost versus capital cost comparisons of different whole system designs.

An extensive literature search has allowed the development of a detailed list of items contributing to the refrigeration load imposed on walk-in coolers and freezers. In addition to its application to this work, the walk-in load detail provides a foundation for future investigations related to refrigerated storage.

The Proposed 1250 Standard equations yield an AWEF correlation of -2.4% but do not accurately calculate the corresponding compressor runtime, with a variation of -14.6%, when box load variation is removed from the analysis (Figure 15). A new compressor runtime calculation method is needed to more accurately calculate this parameter.

REFERENCES

Adre, N., and Hellickson, M. L. 1989. Simulation of the transient refrigeration load in a cold storage for apples and pears. *ASHRAE Transactions* 32(3):1038-1048.

AHRI. 2009a. AHRI Load Spreadsheet used to develop load profiles found in AHRI Standard 1250/1251. Air-Conditioning, Heating and Refrigeration Institute, Arlington, VA.

AHRI. 2009b. ANSI/AHRI Standard 1250 (I-P), 2009 Standard for Performance Rating of Walk-In Coolers and Freezers. Air-Conditioning, Heating and Refrigeration Institute, Arlington, VA.

AHRI. 2009c. ANSI/AHRI Standard 1251 (SI), 2009 Standard for Performance Rating of Walk-In Coolers and Freezers. Air-Conditioning, Heating and Refrigeration Institute, Arlington, VA.

Altwies, J. E. 1998. Electrical demand reduction in refrigerated warehouses. University of Wisconsin - Madison, Madison, Wisconsin.

Altwies, J. E., and Reindl, D. T. 1999. Passive thermal energy storage in refrigerated warehouses. *20th International Congress of Refrigeration, IIR/IIF*. Sydney.

Anonymous. 2004. Walk-in refrigerators and freezers. *Foodservice Equipment and Supplies* Sep 2004. 57(9):84-85.

Aparicio-Cuesta, M. P., and Garcia-Moreno, C. 1988. Quality of frozen cauliflower during storage. *Journal of Food Science* 53(2):491-493. Food Technologies.

Arias, J. 2005. Energy usage in supermarkets - modelling and field measurements, Royal Institute of Technology.

Arias, J., and Lundqvist, P. 2005. Modeling supermarkets energy usage. *Proc. Vicenze Conf.*:109-116. IIR.

Arias, J., and Lundqvist, P. 2006. Heat recovery and floating condensing in supermarkets. *Energy & Buildings* 38(2):73-81.

Ashby, B. H., Bennett, A. H., Bailey, W. A., Molerratanoud, W., and Kramer, A. 1979. Energy savings and quality deterioration from holding frozen foods at two daily temperature levels. *Transactions of the American Society of Agricultural Engineers*:938-943.

ASHRAE. 1972. *1972 ASHRAE Handbook - Fundamentals*. Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers,

ASHRAE. 1977. *1977 ASHRAE Handbook - Fundamentals*. Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers,

ASHRAE. 1981. *1981 ASHRAE Handbook - Fundamentals*. Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers,

ASHRAE. 1989. *1989 ASHRAE Handbook - Fundamentals*. Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers,

ASHRAE. 1990. *1990 ASHRAE Handbook - Refrigeration*. I-P Edition. Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers,

ASHRAE. 1994. *1994 ASHRAE Handbook - Refrigeration*. I-P Edition. Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers,

ASHRAE. 1998. *1998 ASHRAE Handbook - Refrigeration*. Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, ASHRAE. 2002. *2002 ASHRAE Handbook - Refrigeration*. Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.,

ASHRAE. 2005. *2005 ASHRAE Handbook - Fundamentals*. Atlanta, Georgia: American Society of Heating, Refrigerating, and Air-Conditioning Engineers,

Azzouz, A., and Duminil, J. G. M. 1993. Experimental determination of cold loss caused by opening industrial coldroom doors. *International Journal of Refrigeration* 16:57-66.

Bansal, P. K., and Jain, S. 2007. Cascade systems: past, present, and future. *ASHRAE Transactions* 113:245.

Bauernfeind, J. C., and Siemers, G. F. 1945. Retardation of discoloration in frozen sliced peaches. *Quick Frozen Foods* 7:48.

Becker, B. R., and Fricke, B., A. 1996a. Transpiration and respiration of fruits and vegetables. New Developments in Refrigeration for Food Safety and Quality, International Institute of Refrigeration, Paris, France, and American Society of Agricultural Engineers, St. Joseph, Michigan, 110-121.

Becker, B. R., and Fricke, B. A. 1996b. Simulation of moisture loss and heat loads in refrigerated storage of fruits and vegetables. *Refrigeration Science and Technology Proceedings*. Lexington, KT.

Becker, B. R., Fricke, B. A., and Sartin, B. C. 2011. Performance standards for walk-in refrigerator and freezer systems (ARTI project 9002) final report.

Birdseye, C. 1946. The preservation of foods by freezing. *Refrigerating Engineering (Application Data 22)* 2:51.

Blackmore, T. 2012. Refrigeration load calculations, Heatcraft. University of Missouri - Kansas City, Thermal Science Systems and Components Lecture 6, February 16. Kansas City, Missouri.

Boggs, M. M., Dietrich, W. C., Nutting, M., Olson, R. L., Lindquist, F. E., Bohart, G. S., Neumann, M. J., and Morris, M. J. 1960. Time-temperature tolerance of frozen foods XXI: frozen peas. *Food Technology* 14:181.

Brown, W. G., and Solvason, K. R. 1962. Natural convection through rectangular openings in partitions. I. vertical partitions. *International Journal of Heat and Mass Transfer* 5:859-868.

CEC. 2007. Appliance efficiency regulations. CEC-400-2007-016-REV1, California Energy Commission, Sacramento, CA.

CEC. 2008. 2008 building energy efficieincy standards for residential and nonresidential buildings. CEC-400-2008-001-CMF, California Energy Commission, Sacramento, CA.

Chen, P., Cleland, D. J., Lovatt, S. J., and Bassett, M. R. 1999. Air infiltration into refrigerated stores through rapid-roll doors. *Proceedings of the 20th International Congress of Refrigeration*:925-932. IIR/IIF. Sydney.

Chen, P., Cleland, D. J., Lovatt, S. J., and Bassett, M. R. 2002. An empirical model for predicting air infiltration into refrigerated stores through doors. *International Journal of Refrigeration* 25:799-812.

Chourasia, M. K., Chatterjee, G., Biswas, I., Sahu, R., and Kumar, A. 2003. Cooling parameters of potatoes in a commercial cold store. *Journal of Interacademicia* 7(2):186-189.

Chourasia, M. K., and Goswami, T. K. 2007. Product cooling load and moisture loss under different loading patterns and cooling rates of potatoes in cold storage. *Journal of Food Process Engineering* 31:339-353. Agricultural and Food Engineering Department, India Institute of Technology. West Bengal, India.

Christensen, K. G., and Bertilsen, P. 2004. Refrigeration systems in supermarkets with propane and CO2 - energy consumption and economy. *EcoLibrium*.

Chuangchid, P., and Krarti, M. 2001. Foundation heat loss from heated concrete slab-on-grade floors. *Building and Environment* 36(5):637-655.

CIGR. 1999. *CIGR Handbook of Agricultural Engineering*.

Cleland, A. C. 1983. Simulation of industrial refrigeration plants under variable load conditions. *International Journal of Refrigeration* 6(1):11-19.

Cleland, D. J., Boyd, N. S., and Cleland, A. C. 1982. A model for fish freezing and storage on board small New Zealand fishing vessels. *International Institute of Refrigeration*.

Cleland, D. J., Chen, P., Lovatt, S. J., and Bassett, M. R. 2004. A modified model to predict air infiltration into refrigerated facilities through doorways. *ASHRAE Transactions*:58-66.

Cleland, D. J., and O'Hagan, A. N. 2003. Performance of an air cooling coil under frosting conditions. *ASHRAE Transactions* 109:243-250.

Cole. 1987. Infiltration load calculation for refrigerated warehouses. *Heating, Piping, and Air Conditioning* 59(4):45-54.

Cole, R. A. Year. Infiltration: a load calculation guide. *Proceedings of the Sixth International Institute of Ammonia Refrigeration*.

Cole, R. A. 1989. Refrigeration loads in a freezer due to hot gas defrost and their associated costs. *ASHRAE Transactions* 95(2).

Congress, U. S. 2007. Energy Independence and Security Act of 2007 (42 USC 17001).

Cooper, C. E. B. 1973. Cold storage design and operation, some practical suggestions. *FAO Agricultural Services Bulletin No. 19*. Food and Agriculture Organization of the United Nations. Rome.

Devres, Y. O., and Bishop, C. F. H. 1995. Computer model for weight loss and energy conservation in a fresh produce refrigerated store. *Applied Energy* 50:97-117.

Diehl, H. G., and Berry, J. A. Year. The temperature factor in the freezing preservation of fruits and vegetables. *Proceedings of the American Society for Horticultural Science*, 29:287-289.

DOE. 2004. U.S. Energy Flow Trends - 2002, Net Primary Resource Consumption ~97 Quads. L. L. N. L. Energy and Environment Directorate, ed., U.S. Department of Energy.

DOE. 2006. 2004 commercial energy end-use splits, building energy data book. Office of Energy Efficient and Renewable Energy, U.S. Department of Energy.

DOE. 2008. 2006 commercial energy end-use splits, building energy data book. Office of Energy Efficiency and Renewable Energy, U.S. Department of Energy.

DOE. 2010a. DOE transcript - test procedures for walk-in coolers and freezers, notice of proposed rulemaking public meeting.

DOE. 2010b. Preliminary technical support document (TSD): energy conservation program for certain commercial and industrial equipment: walk-in coolers and walk-in freezers. Office of Energy Efficiency and Renewable Energy, U.S. Department of Energy, Washington, D.C.

DOE. 2010c. Test procedures for walk-in coolers and freezers notice of proposed rulemaking presentation. Building Technologies Program, U.S. Department of Energy, March 24.

DOE. 2011. Energy Conservation Program: Test Procedures for Walk-in Coolers and Walk-in Freezers; Final Rule. O. o. E. E. a. R. Energy, ed., U.S. Department of Energy, 21580-21612.

Downing, C. C., and Meffert, W. A. 1993. Effectiveness of cold-storage door infiltration protective devices. *ASHRAE Transactions* 99(2):356-366.

Edwards, B. 2010. California statewide codes and standards program - comments on walk-in test method. Re: NOPR for Test Procedures for Walk-in Coolers and Freezers, Docket No. EERE-2008-BT-TP-0014 and (RIN) 1904-AB85, U.S. Department of Energy.

Elsayed, M. M. 1998. Infiltration in cold rooms. *Heating, Ventilation, Air Conditioning and Refrigeration Research* 4(2):179-202.

Emerson. 2012. Emerson Climate Technologies Online Product Information. Emerson Climate Technologies, https://opi.emersonclimate.com/was.extension.opi.web/OPIInterfaceServlet. August 2012.

Faramarzi, R. 1999. Efficient display case refrigeration. *ASHRAE Journal* 41(11):46.

Faramarzi, R. T., Coburn, B. A., and Sarhadian, R. 2002. Performance and energy impact of installing glass doors on an open vertical deli/dairy display case. *ASHRAE Transactions* 108(1):673-679.

FDA. 2001. *Food Code 2001*. Washington, D.C.: U.S. Department of Health and Human Services,

Foster, A. M., Swain, M. J., Barrett, R., and James, S. J. 2003. Experimental verification of analytical and CFD predictions of infiltration through cold store entrances. *International Journal of Refrigeration-Revue Internationale du Froid* 26(8 December):918-925.

Fricke, B. A., and Becker, B. R. 2011. Comparison of vertical display cases: energy and productivity impacts of glass door versus open vertical display cases. *ASHRAE Transactions* 117(1).

Fritzsche, C., and Lilienblum, W. 1968. Neue messengun zur bestimmung der kalterluste an kuhlraumturen. *Kaltetechnik-Kilimatiserung* 20:279-286.

Goetzler, W., Goffri, S., Jasinski, S., Leggett, R., Lisle, H., Marantan, A., Millard, M., Pinault, D., Westphalen, D., and Zogg, R. 2009. Energy savings potential and research and development opportunities for commercial refrigeration, final report. to U.S. Department of Energy, Energy Efficient and Renewable Energy Building Technologies Program from Navigant Consulting, Inc.

Gortner, W. A., Fenton, F., Volz, F. E., and Gleim, E. 1948. Effect of fluctuating storage temperatures on quality of frozen foods. *Industrial and Engineering Chemistry Research* 40(8):1423-1426.

Gosney, W. B., and Olama, H. A. L. 1975. Heat and enthalpy gains through cold room doorways. *Proceedings of the Institute of Refrigeration*:31-41.

Granryd, E. 2003. Refrigerating engineering, part 2. Department of Energy Technology, Royal Institute of Technology, Stockholm, Sweden.
Hardenburg, R. E., Watada, A. E., and Wang, C. Y. 1986. *The Commercial Storage of Fruits, Vegetables, and Florist and Nursery Stocks*. Washington, D.C.: U.S. Department of Agriculture,
Hayes, F. C., and Stoecker, W. F. 1969. Heat transfer characteristics of air curtains and design data for air curtains. *ASHRAE Transactions* 75(2):153-180.
Heatcraft. 2008. *Engineering Manual: Commercial Refrigeration Cooling and Freezing Load Calculations and Reference Guide*. Stone Mountain, GA: Heatcraft Refrigeration Products, LLC,
Henderson, H. I. J., and Khattar, M. 1999. Measured impacts of supermarket humidity level on defrost peformance and refrigerating system energy use. *ASHRAE Transactions* 105:508-520.
Hendrix, W. A., Henderson, D. R., and Jackson, H. Z. 1989. Infiltration heat gains through cold-storage room doorways. *ASHRAE Transactions* 95(2):1155-1168.
Hirsch, J. J. 2006. eQuest-Refrigeration, version 3.61.
Huan, Z. 2008. Energy saving opportunities in food 'cold chain'. Vaal University of Technology, School of Mechanical Engineering, Vanderbijlpark.
Hustrulid, A., and Winters, J. D. 1943. The effect of fluctuating storage temperatures on frozen fruits and vegetables. *Agricultural Engineering* 24(2):416.
Hwang, Y., Jin, D.-H., and Radermacher, R. 2007. Comparison of R-290 and two HFC blends for walk-in refrigeration systems. *International Journal of Refrigeration* 30(4):633-641. Elsevier Ltd. Langford Lane, Kidlington, Oxford, OX5 1GB, United Kingdom.
IARW. 1999. *Energy Conservation Manual*. International Association of Refrigerated Warhouses,
IIR. 1986. *Recommendations for the Processing and Handling of Frozen Foods*. 3rd. International Institute of Refrigeration,
ISO. 1998. *ISO 13370, Thermal Performance of Building - Heat Transfer via the Ground Calculation Method*.
Jekel, T. B., and Reindl, D. T. 2000. Energy efficiency opportunities at SuperValu. University of Wisconsin, HVAC&R Center, Madison, WI.
Jones, B. W., Beck, B. T., and Steele, J. P. 1983. Latent loads in low humidity rooms due to moisture. *ASHRAE Transactions* 89:35-55.
Kassa, H., Keil, C., and Fent, K. W. 2004. A protocol for quantitative measurement of light intensity levels in food service operations. *Journal of Environmental Health* 67(2):9-14.
Kays, S. J. 1991. *Postharvest Physiology of Perishable Plant Products*. New York: Van Nostrand Reinhold,
Kimber, D. J. 1998. Final technical progress report. Nevada Energy Control Systems, Inc., Incline Village, NV.
Klein, S. A., and Alvarado, F. L. 1999. EES-Engineering Equation Solver. F-Chart Software, Middleton, WI.
Knudsen, H. J. H., and Pachai, A. C. 2004. Energy Comparisons Between CO2 Cascade Systems and State of the Art R404a Systems. Technical University of Denmark.

Krack. 1977. *Krack Refrigeration Load Estimating Manual RLE-278*. Krack Corporation,

Krack. 1992. *Krack Refrigeration Load Estimating Manual RLE-278*. Krack Corporation,

Krarti, M. 1999. *Ground-coupled Heat Transfer, a Chapter in: Advances in Solar Energy*. Boulder, CO: ASES,

Krarti, M., Chuangchid, P., and Ihm, P. 2004. Analysis of moisture and heat transfer beneath freezer foundations. part 1. *ASME Journal of Solar Energy Engineering* 126(2):716-725.

Krarti, M., Claridge, D., and Kreider, J. 1988. ITPE method applications to time-varying two-dimensional ground-coupling problems. *International Journal of Heat and Mass Transfer* 31(1):899-911.

Kun, X., Ben-zhe, X., Tian-ji, C., and Mao-quan, G. 2007. Analysis of Reclamation of Low Quality Exhausting Thermal Energy from Refrigerated Warehouse. College of Food Science, Shanghai Fisheries University, Shanghai, China.

Larkin. 2007. *Compak Unit Coolers - Technical Guide; LK-LOPTB*. Stone Mountain, GA: Larkin,

Longdill, G. R., Frazerhurst, L. F., and Wyborn, L. G. 1974. Air curtains. *MIRINZ Publ. No. 385*. MIRINZ. Hamilton, New Zealand.

Longdill, G. R., and Wyborn, L. G. 1978. Performance of air curtains in single-storey cold stores. *Proceedings of the 15th International Congress of Refrigeration* 4:77-88.

Love, R. J., and Cleland, D. J. 2007. A model for self-contained solid door refrigerated display cabinets. *ASHRAE Transactions* 113:253-262.

Lutz, J. M., Wright, R. C., and Caldwell, J. S. 1934. Preservation of young and lucretia varieties of dewberries. *Fruit Prod. Journal* 9:267-269.

Mago, P. J., and Sherif, S. A. 2002. Dynamics of coil-defrosting in supersaturated freezer air. *Journal of Mechanical Engineering Science* 216(C10):949-958.

Mago, P. J., and Sherif, S. A. 2005. Coil frosting and defrosting issues at low freezer temperatures near saturation conditions. *ASHRAE Transactions* 111:3-17.

Magoo, N. 2003. High-temperature refrigerated warehouse operation under real-time pricing of electricity, University of Wisconsin-Madison, Madison, WI.

Manske, K. A. 1999. Performance optimization of industrial refrigeration systems, University of Wisconsin - Madison, Madison, WI.

Marchese, J. 2002. Sizing walk-in coolers and freezers. *Air Conditioning, Heating and Refrigeration News* June 3, 2002. 216(5):27.

Marshall, S. A., and James, R. W. 1975. Dynamic analysis of an industrial refrigeration system to investigate capacity control. *Proceedings of the Institute of Mechanical Engineers* 189:437-444.

McHugh, J. 2010. California statewide IOU codes & standards program NOPR test procedures for walk-in coolers and freezers: introductory comments. McHugh Energy Consultants, Inc., Orangevale, CA.

Mitchell, J. W., and Braun, J. E. 1998. Design, analysis, and control of space conditioning equipment and systems. University of Wisconsin-Madison.

Mitchell, J. W., Urban, R. E., Beckman, W. A., Dorgan, C. E., and Lindell, C. J. 1992. Analysis of supermarket dehumidification alternatives. EPRI TR-100352; Project 2891-03; Final Report, Electric Power Research Institute, Solar Energy Laboratory, University of Wisconsin-Madison.

Morris, K. 2012. Equipment Selections. Charles D. Jones Co., University of Missouri - Kansas City, Thermal Science Systems and Components Lecture 8, March 1. Kansas City, Missouri.
Nagaraju, J., Vikash, K., and Krishna Murthy, M. V. 2001. Photovoltaic-powered cold store and its performance. *International Journal of Energy Research* 25(5):389-396. John Wiley and Sons Ltd.
Navy. 1986. *Refrigeration Systems for Cold Storage*. Design Manual 3.04. Naval Facilities Engineering Command,
Nelson, B. I. 2011. Optimizing Hot Gas Defrost. Colmac Coil Manufacturing, Inc., Colville, WA.
Newell, T. 2011. Ground heat transfer. *ASHRAE Journal* 53(2):62-68.
O'Hagan, A., Cleland, D. J., and Cleland, A. C. 1993. Air cooling coil performance under frosting conditions. *IIF-IIR Commissions B1,B2,D1,D2/3* 3:345-362. Palmerston North (New Zealand).
Patel, R. F., Teagan, P. W., and Dieckmann, J. T. 1993. Characterization of commercial building appliances. ADL Reference No. 42520, Building Equipment Division, Office of Building Technologies, U.S. Department of Energy.
PG&E. 2004. Analysis of standards options for walk-in coolers (refrigerators) and freezers. Pacific Gas and Electric Company.
PG&E. 2007. Final report: refrigerated warehouses. Pacific Gas and Electric Company, San Francisco, CA.
Pham, Q. T., and Oliver, D. W. 1982. Air infiltration into cold stores. *Proceedings of the 22nd Meat Industry Res. Conf.*:106-112.
Pham, Q. T., and Oliver, D. W. 1983. Infiltration of air into cold stores. *Proceedings of 16th International congress of Refrigeration, IIR, Comm D1* 4:67-72. Paris, Frane.
Plagge, H. H. 1938. Suggestions for freezing and storing fruits in refrigerated locker plants. *Ice and Refrigeration* 95:31.
Rea, M. S., Ph.D., Editor-in Chief. 2000. *The Iesna Lighting Handbook Reference and Application*. Illuminating Engineering Society of North America,
Reindl, D. T., and Jekel, T. B. 2008. Infiltration rate determination for low temperature freezing systems. *ASHRAE Transactions*:264-272.
Roy, M. 2010. Green warehouse. *ASHRAE Journal*:64-70.
Royal, R. 2010. Heat recovery in retail refrigeration. *ASHRAE Journal*:14-22.
Sami, S. M., and Tulej, P. J. 1996. Drop-in-replacement blend HFC-23/HCFC-22/HFC-152A for air/refrigerant equipment. *International Journal of Energy Research* 20(9):787-796. John Wiley & Sons Ltd. Chichester, United Kingdom.
Sand, J. R., Fischer, S. K., and Baxter, V. D. 1997. Energy and global warming impacts of HFC refrigerants and emerging technologies. Oak Ridge National Laboratory.
SCE. 2008. Preliminary CASE report: analysis of standards option for walk-in refrigerated storage. Southern California Edison, Sacramento, CA.
SCE. 2009. Evaluation of an internet-based refrigeration monitoring and control system. Design and Engineering Services, Southern California Edison, Sacramento, CA.
Schulak, E. R., and Horvay, J. B. 2002. Energy transfer system for cold storage facilities.
Sekhar, S. J., Kumar, K. S., and Lal, D. M. 2004. Ozone friendly HFC134a/HC mixture compatible with mineral oil in refrigeration system improves energy efficiency of a

walk in cooler. *Energy Conversion and Management* 45(7-8):1175-1186. Elsevier Ltd.

Sekhar, S. J., and Lal, D. M. 2005. HFC134a/HC600a/HC290 mixture a retrofit for CFC12 systems. *International Journal of Refrigeration* 28(5):735-743. Elsevier Ltd.

SEL, Transsolar, CSTB, and TESS. 2006. Weather Data. TRNSYS 16: A Transient System Simulation Program, Solar Energy Laboratory, University of Wisconsin-Madison, TRANSSOLAR Energietechnik GmbH, Centre Scientifique et Technique du Batiment, Thermal Energy Systems Specialists.

Sezgen, O., and Koomey, J. G. 1995. Technology data characterizing refrigeration in commercial buildings: application to end-use forecasting with COMMEND 4.0. Lawrence Berkeley National Laboratory, Berkeley, CA.

Shaw, B. H. 1971. Heat and mass transfer by natural convection and combined natural convection and forced air flow through large rectangular openings in a vertical partition. *Symposium on Heat and Mass Transfer by Combined Forced and Natural Convection*:31-39.

Sherif, S. A., Mago, P., Mutawa, N. K., Theen, R. S., and Bilen, K. 2001. Psychrometrics in the supersaturated frost zone. *ASHRAE Transactions* 107(2).

Sherif, S. A., Mago, P. J., and Theen, R. S. 2002. A study to determine heat loads due to coil defrosting - phase 2. UFME/SEECL-200201, University of Florida, Atlanta, GA.

Smith, G. R. 1989a. Theoretical cooling coil calculations at freezer temperatures to avoid unfavorable coil frost. *ASHRAE Transactions* 95(2):1138-1148.

Smith, G. R. 1992. Latent heat, equipment-related load, and applied psychrometrics at freezer temperatures. *ASHRAE Transactions* 98(2):649-657.

Smith, W. W. 1989b. Computerizing refrigeration load calculations. *ASHRAE Journal* 31(11).

Somrani, R., Mun, J., and Krarti, M. 2008. Heat transfer beneath ice-rink floors. *Building and Environment* 43(10):1687-1698.

Stegmann, R. 2000. Low temperature refrigeration - practical guide. *ASHRAE Journal*.

Stoecker, W. F. 1998. *Industrial Refrigeration Handbook*. New York: McGraw Hill,

Stoeckle, R. 2000. Refrigerated warehouse operation under real-time pricing, University of Wisconsin - Madison, Madison.

Stoeckle, R., Reindl, D. T., and Mitchell, J. W. 2002. Strategies for operating refrigerated warehouse under real-time pricing. *ASHRAE Transactions* 108:145-154.

Sugar_Engineers. 2012. Psychrometric Calculations. http://www.sugartech.co.za/psychro/index.php. March 14, 2012.

Sugiartha, N., Tassou, S. A., Chaer, I., and Marriot, D. 2009. Trigeneration in food retail: an energetic, economic and environmental evaluation for supermarket application. *Applied Thermal Engineering* 29:2624-2632.

Sujau, M., Bronlund, J. E., Merts, I., and Cleland, D. J. 2006. Effect of defrost frequency on defrost efficiency, defrost heat load and coolstore performance. Innovative Equipment and Systems for Comfort and Food Preservation, Auckland, New Zealand, 525-532.

Takahashi, K., and Inoh, M. 1963. Some measurements on air curtain efficiency for cold rooms. *Proceedings of the International Congress of Refrigeration* 2:1035-1039.

Tamm, W. 1965. Air flow with air curtains to protect cold rooms. *Proceedings of the 11th International Congress of Refrigeration* 2:1025-1038.
Tamm, W. 1966. Kälteverluste durch kühlraumtüren. *Kältetechnik* 18:142-144.
Tamm, W. 1969. *Australian Institute of Refrigeration, Air Conditioning, and Heating* 23(12):41-43.
Thompson, J. F., and Spinoglio, M. 1996. Small-scale cold rooms for perishable commodities. University of California, Division of Agriculture and Natural Resources, Oakland, CA.
Tressler, D. K., and Evers, C. F. 1947. *The Freezing Preservation of Foods*. New York: Avi Publishing Co. Inc.,
Unattributed. 1969. *Food Engineering* November 1969.
USDA. 2010. Capacity of refrigerated warehouses. 2009 summary. United States Department of Agriculture.
Van Male, J. 1983. New concept for a vertical air curtain for cold storage doors. *Proceedings of the 16th International Congress of Refrigeration* 5:89-95.
Wade, N. L. 1984. Estimation of the refrigeration capacity required to cool horicultural produce. *International Journal of Refrigeration* 7(6):358-366.
Walker, D. H. 1992. Field testing of high-efficiency supermarket refrigeration. EPRI TF-100351, Project 2569-06, Final Report, Foster-Miller, Inc., Waltham, MA.
Walker, D. H. 2001. Development and demonstration of an advanced supermarket refrigeration/HVAC system. Oak Ridge National Lab.
Walker, D. H., and Baxter, V. D. 2002. Field testing of an advanced low-charge supermarket refrigeration system. IIR Commission D1/B1, International Institute of Refrigeration, Urbana, IL, 14p.
Walker, D. H., and Baxter, V. D. 2003. Analysis of advanced, low-charge refrigeration for supermarkets. *ASHRAE Transactions* 109 PART 1:285-292.
Walker, D. H., and Deming, G. I. 1989. Supermarket refrigeration modeling and field demonstration. EPRI Report No. CU-6268, Electric Power Research Institute, Palo Alto, CA.
Walker, D. H., Tsaros, T. L., Deming, G. I., and of Foster-Miller, I. 1990. Guide for selection of supermarket refrigeration systems. EPRI CU-6740; Project 2569-6; Final Report, Foster-Miller Inc., Waltham, MA.
Westphalen, D., Zogg, R. A., Varone, A. F., and Foran, M. A. 1996. Energy savings potential for commercial refrigeration equipment. Building Equipment Division, Office of Building Technologies, U.S. Department of Energy, Cambridge, Massachusetts.
Wichman, A., and Braun, J. E. 2009. Fault detection and diagnostics for commercial coolers and freezers. *HVAC&R Research* 15(1):77-99.
Wiegand, E. H. 1931. The frozen pack method of preserving berries. *Oregon Agricultural Experiment Station Bulletin 278*. Oregon State Agricultural College.
Winters, J. D. 1942. Quality of frozen fruits and vegetables. *Minnesota Agricultural Experiment Station Bulletin 362*.
Woodroof, J. G., and Shelor, E. 1947. Effect of freezing storage on strawberries, blackberries, raspberries, and peaches. *Journal of Food Freezing*:206-209.

Zhang, C., Ding, G., and Li, H. 1999. New three-zone model for performance prediction of adiabatic capillary tubes. *Shanghai Jiaotong Daxue Xuebao/Journal of Shanghai Jiaotong University* 33(8 Aug):934-938.

Zhang, J., and Groll, E. A. 2005. Survey of the design of refrigeration plants for public refrigerated warehouses. *ASHRAE Transactions* 111:327-332.

Zhang, M. 2006. Energy analysis of various supermarket refrigeration systems. *International Refrigeration and Air Conditioning Conference*.

VITA

Bryan Curtis Sartin was born on December 19, 1984, to John Curtis and Peggy Lee Sartin in Warrensburg, Missouri. He received elementary and secondary education in the Concordia R2 School District, graduating third in his class. He received a Trustee's Scholarship to the University of Missouri – Rolla (UMR) in 2003. Alongside receiving an education, Mr. Sartin held membership in Pi Tau Sigma, Tau Beta Pi, and Phi Eta Sigma. He was active in ASME serving various officerships, managed an intramural sports team, and spent one year as a residential assistant. Mr. Sartin had five internships at various companies, including Lexmark, Ford Motor Company, Caterpillar, Hunter Engineering, and Doe Run Resource Recycling. He also had six additional part-time positions, including work as an undergraduate assistant for laser-aided manufacturing development and a grader for the introductory Mechanical Engineering thermodynamics course at UMR. Mr. Sartin obtained his Engineer-in-Training Certification in March of 2008 and graduated summa cum laude in December of 2008 with a Bachelor of Science degree in Mechanical Engineering and a minor in Spanish.

After losing his job offer at graduation due to economic reasons, Mr. Sartin spent six month working as a laborer and searching for employment as an engineer. In mid-2009, he started working in the Research and Development group for SPX Cooling Technologies, Inc. Over the last three years, he has worked in four different departments and led projects ranging from product development to manufacturing. In the spring of 2010, he began work toward his M.S. in Mechanical Engineering at the University of Missouri – Kansas City. Upon completion of his degree requirements, Mr. Sartin plans to continue his work as a design and project engineer, while beginning to work towards obtaining his professional licensure as an engineer.

His work with "Walk-in Performance Modeling and Validation of Simplified Performance Rating Methods" has been published with his seniors, Dr. Bryan Becker and Dr. Brian Fricke, at the 2011 ASHRAE Conference in Montreal, Canada.