EFFECTS OF THE FDA'S 1993 FOOD CODE ON THE PERFORMANCE AND POWER USE OF A REFRIGERATED DISPLAY CASE

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DISCLAIMER

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EXECUTIVE SUMMARY

The purpose of this project was to test and evaluate the impact of the new Food and Drug Administration's (FDA) 1993 Food Code on the power use and performance of a multi-deck display case, commonly used in supermarkets for storing and refrigerating dairy products. This display case was tested in a controlled environment maintained at 75 °F at 50% relative humidity, and served by a refrigeration system using a Hydroflurocarbon (HFC) refrigerant, R-404A. The new FDA Food Code requires that for additional protection against food-borne illnesses, food stores reduce the core temperature of hazardous products by 4 °F, from 45 °F (7.2 °C) to 41 °F (5 °C). This code is revised every two years. The fundamentals of the 1993 food code concerning product temperature, however, have been repeated for the 1995 and 1997 editions. Supermarkets which operate on a narrow profit margin, with refrigeration being their largest energy end-use, could be adversely impacted by the implementation of this code. As a result, testing and quantifying the impact of this code on the power use and performance of the refrigeration system could guide Edison customers to make intelligent decisions in complying with the FDA requirements.

Southern California Edison (SCE) conducted this test at its state-of-the-art Refrigeration Technology and Test Center (RTTC), located in Irwindale, CA. The RTTC's sophisticated instrumentation and data acquisition system provided detailed tracking of the refrigeration system's critical temperature and pressure points during the test period. These readings were then utilized to quantify various heat transfer and power related parameters within the refrigeration cycle. The results of SCE's test indicate that the FDA's temperature recommendation may increase the power consumption and cooling load of the dairy cases by about 31% and 15%, respectively. The summary table, below, depicts the impact of reducing the product temperature contained in a dairy display case, from 45 °F to 41 °F on various refrigeration parameters.

Summary Table

Scenario	Discharge Air Temperature (^oF)	Refrigeration Load of the Case* (MBtu/hr-ft)	Compressor Power (kW)	Rating* (kW/Ton)
45^{o}F	37.3	1.9	2.6	0.8
41^{o}F	32.1	2.2	3.4	0.9
% Δ	13.9%	-14.6%	-31.2%	-15.3%
% Δ/^{o}F	3.5%	-3.6%	-7.8%	-3.8%

* Calculated Values

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NOMENCLATURE

T_{db}	Room dry-bulb temperature (°F)
T_{dp}	Room dew point temperature (°F)
RH	Relative Humidity of the room (Taken from the Trane Psychrometric Chart based on the average dry bulb and dew point temperatures) (%)
DAT_1	Discharge air temperature @ location (1) (°F)
DAT_2	Discharge air temperature @ location (2) (°F)
DAT_3	Discharge air temperature @ location (3) (°F)
DAT_{avg}	Average discharge air temperature (°F)
MAT	Mid air curtain temperature (°F)
RAT	Return air temperature (°F)
ACTD	Temperature Drop across the Air Curtain (°F)
PT_{top}	Product temperature @ top shelf (°F)
PT_{bottom}	Product temperature @ bottom shelf (°F)
PT_{avg}	Average product temperature (°F)
$Tair_e$	Air temperature entering evaporator (°F)
$Tair_L$	Air temperature leaving evaporator (°F)
ΔT_{air}	Temperature difference of air across evaporator (°F)
$Tevap_1$	Evaporator temperature @ location (1) (°F)
$Tevap_2$	Evaporator temperature @ location (2) (°F)
$Tevap_3$	Evaporator temperature @ location (3) (°F)
$Tevap_{avg}$	Average evaporator temperature (°F)
TD_{approx}	Temperature Difference between entering air and evaporator temperature (°F)
MFR	Mass flow rate of refrigerant (lb/min)
SST_1	Saturated suction temperature of refrigerant @ location 1 (°F)
SST_2	Saturated suction temperature of refrigerant @ location 2 (°F)
SST_3	Saturated suction temperature of refrigerant @ location 3 (°F)
SST_{avg}	Average saturated suction temperature of refrigerant (°F)
SSP	Saturated suction pressure of refrigerant (psig)
SST_{theo}	Saturated suction temperature of refrigerant based on saturated properties tables + ΔTsuperheat (10^{o}F) (°F)
SDP	Saturated discharge pressure of refrigerant (psig)
CR	Compression ratio of compressor
$Tcond_{in}$	Temperature of superheated refrigerant vapor entering condenser (°F)
$Hcond_{in}$	Enthalpy of superheated refrigerant vapor entering condenser (Btu/lb)
$Tcond_{exit}$	Temperature of liquid refrigerant leaving condenser (°F)
$Hcond_{exit}$	Enthalpy of liquid refrigerant leaving condenser (Btu/lb)
$Tevap_{exit1}$	Temperature of superheated refrigerant vapor leaving evaporator @ location 1 (°F)
$Tevap_{exit2}$	Temperature of superheated refrigerant vapor leaving evaporator @ location 2 (°F)
$Tevap_{exit3}$	Temperature of superheated refrigerant vapor leaving evaporator @ location 3 (°F)
$Tevap_{exitavg}$	Average temperature of superheated refrigerant vapor leaving evaporator (°F)
$Hevap_{exit}$	Enthalpy of superheated refrigerant vapor leaving evaporator (Btu/lb)
$Hcomp_{in}$	Enthalpy of superheated refrigerant vapor entering compressor (Btu/lb)
$Tsubcool_1$	Subcooled temperature of liquid refrigerant entering expansion valve @ location 1 (°F)

$Tsubcool_2$	Subcooled temperature of liquid refrigerant entering expansion valve @ location 2 (°F)
$Tsubcool_3$	Subcooled temperature of liquid refrigerant entering expansion valve @ location 3 (°F)
$Tsubcool_{avg}$	Average subcooled temperature of liquid refrigerant entering expansion valve (°F)
Hsatliq	Calculated value of enthalpy at saturated temperature corresponding to saturated discharge pressure (Btu/lb)
$Tsat_{SDP}$	Temperature of saturated liquid @ discharge pressure (°F)
ΔTsubcool	Temperature difference between subcooled liquid entering the expansion valve and saturated liquid leaving the condenser (°F)
Cpsubcool	Specific heat of subcooled liquid refrigerant entering expansion valve (Btu/lb °F)
ΔHsubcool	Enthalpy change between subcooled liquid entering expansion valve and saturated liquid leaving condenser (Btu/lb)
Hsubcool	Calculated value of enthalpy of subcooled refrigerant entering expansion valve (Btu/lb)
Tpostexp	Temperature of refrigerant in the mixed phase leaving expansion valve (°F)
Wcomp	Actual work input of the compressor (kW)

INTRODUCTION

The purpose of this test was to evaluate the impact of the Food and Drug Administration's (FDA) 1993 Food Code on the power use and performance of a multi-deck display case typically used in supermarkets for storing dairy products. Southern California Edison (SCE) conducted this test at its state-of-the-art Refrigeration Technology and Test Center (RTTC), located in Irwindale, California. The RTTC's sophisticated instrumentation and data acquisition system provided detailed tracking of the refrigeration system's critical temperature and pressure points during the test. These readings were then utilized to quantify various heat transfer and power related parameters of the refrigeration cycle.

Supermarkets operate on a narrow profit margin. Energy costs play a crucial role in supermarket economics and competitiveness. In many cases the annual energy costs for a supermarket equals or exceeds the sales profit. A supermarket's annual energy costs depend heavily on the refrigeration systems' energy use. And, the energy systems' operations are sensitive to the impact of various food and energy codes.

The 1993 food code was established as a result of food-borne diseases causing 16,000 deaths and more than 25 million illnesses each year. Implementing strong measures to assure food safety is critically important. As a result, in 1993 the FDA's Food Code recommended a lowered storage temperature for certain refrigerated food products for further prevention of food-borne diseases. It required that the refrigeration temperature of meat, poultry, fish, dairy, deli, and cut produce be lowered from 45 °F (7.2 °C) to 41 °F (5 °C). The focus of the FDA's food code is preventing problems in store operation rather than detecting them in the finished product. The code is intended to assist the regulatory agencies at all levels of government in implementing food safety for food stores.

Figure 1 and Table 1 summarize the effects of the FDA code on various parameters of the refrigeration system. The results of SCE's test indicate the FDA's temperature recommendation may increase the power consumption and cooling load of the dairy cases by 31.2% and 14.6%, respectively. Retrofitting existing equipment to comply with the FDA code, provided that sufficient system capacity is available, will require the display case discharge air temperature to be lowered by about 5 °F (from 37.3 °F to 32.1 °F) and the saturated suction temperature of the system to be lowered by about 7 °F (from 28.9 °F to 21.7 °F), while the mass flow rate will be increased by 18.3%.

Table 1 - Effect of the FDA's 1993 Food Code on Refrigeration System Parameters

Scenario	Discharge Air Temperature (°F)	Saturated Suction Temperature (°F)	Mass Flow Rate of Refrigerant (lb/min)	Refrigeration Load of the Case* (MBtu/hr-ft)	Condenser Heat Rejection* (MBtu/hr)	Compressor Power (kW)	Rating* (kW/Ton)
45°F	37.3	28.9	10.5	1.9	43.9	2.6	0.8
41°F	32.1	21.7	12.5	2.2	53.3	3.4	0.9
% Δ	13.9%	24.9%	-18.3%	-14.6%	-21.4%	-31.2%	-15.3%
% Δ/°F	3.5%	6.2%	-4.6%	-3.6%	-5.4%	-7.8%	-3.8%

* Calculated Values

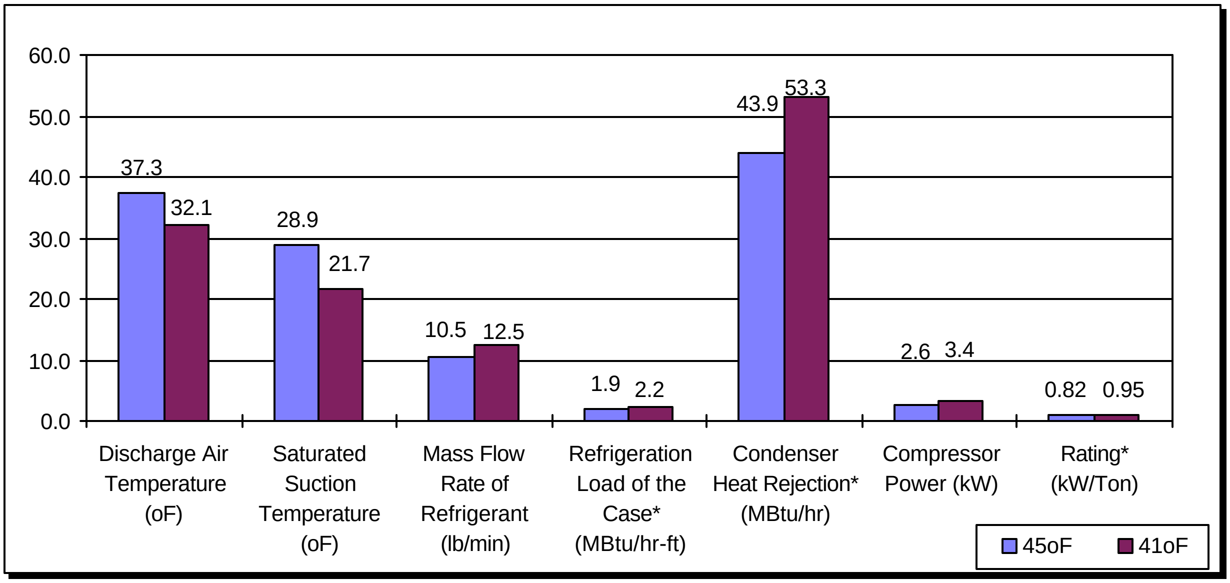

Figure 1 - Effect of the FDA's 1993 Food Code on
Refrigeration System Parameters

FDA's 1993 FOOD CODE OVERVIEW

In 1993, the FDA recommended a new code for additional prevention of food-borne diseases. This code consists of model requirements for safeguarding public health and assuring that food is unadulterated and honestly presented when offered to the customer. It combines elements of a number of previous code editions in use over a number of years. The 1993 code replaces the previous version that was issued in 1982. The code is not federal legislation or regulation, rather it is a recommendation. The backbone of the code is developed following the Hazard Analysis and Critical Control Point (HACCP) method, which provides a science-based, industry-managed process control approach to food safety.

Millions of people in the U.S. each year are made ill by consuming unsafe food. While the U.S. has some of the least contaminated food in the world, food-borne and waterborne illness cases due to pathogenic bacteria, viruses, parasites, and chemicals/toxins in food cause approximately 25 million illnesses and 16,000 deaths annually. This problem costs the food industry $7.7 to $23 billion dollars (per FDA) each year. Due to numerous deaths and illnesses attributed to food-borne diseases each year, this code addresses HACCP pre-control as a means of controlling food-borne illnesses. It intends to assist the regulatory agencies at all levels of government in implementing food safety for food stores. The focus of the FDA's food code is preventing problems in store operation rather than detecting them in the finished product.

The code represents the FDA's best advice for a uniform system of regulation to assure that food at the retail level is safe and properly protected. A portion of the Code addresses new temperature requirements for potentially hazardous food. The spoilage micro-organisms multiply five times faster at 45 °F (7.2 °C) than at 32 °F (0 °C), so it is critical to keep highly perishable and hazardous foods at sufficiently low temperatures. Hazardous foods, by definition, are those that have a potential to acquire a biological property that may cause an unacceptable consumer health risk. In a typical supermarket, hazardous food categories include:

- *Meat*
- *Dairy*
- *Deli*
- *Fish*
- *Poultry*
- *Cut Produce*

The code applies to the products' core temperature, not the air temperature within the case. It requires that the refrigeration temperature of the above products be lowered from 45 °F (7.2 °C) to 41 °F (5 °C) throughout packaging, shipping, receiving and loading. Some bacteria such as Listeria monocytogenes and Salmonella can grow at 45 °F (7.2 °C). Salmonella begins growth at 41 °F (5 °C). At a temperature of 45 °F (7.2 °C), food can be held up to 5 days. Proper refrigeration at 41 °F (5 °C), however, can ensure safe food for up to 10 days. The maximum product temperature of 41 °F (5 °C) is applicable during the entire refrigeration cycle including the defrost periods.

The code will not be enforced by the FDA. Portions, or all of the code will be enforced by the local health inspectors. If it is not adopted in total, only those portions that have been adopted by local governmental authorities will be enforced. The state of Rhode Island and the U.S. Army have already adopted the code. Most states are expected to adopt some or all of the proposed code in the near future. State and local regulatory agencies will implement the code in the following steps:

1. Enact the code into law by the State legislature
2. Pronounce the code as regulation
3. Adopt the code as an ordinance

TEST FACILITY

Edison's RTTC includes a refrigeration room, a controlled environment room and a computer room (Figure 2). The center is equipped with the state-of-the-art refrigeration test and data acquisition systems. Figure 3 illustrates the piping and major equipment used at the RTTC. This unique facility can test the performance of various refrigerants under actual supermarket indoor conditions. These tests can evaluate the performance of refrigerants with respect to numerous key parameters, including varying condensing and suction temperatures. Moreover, RTTC can assess the impact of numerous commercial refrigeration retrofit technologies on the performance of most defined systems.

The controlled environment room contains the low and medium temperature display cases, while the refrigeration room houses the sophisticated refrigeration and chiller test racks. The computer room houses the computer terminals loaded with sophisticated software which facilitates data recording and the programming of various modes of operation of the refrigeration rack.

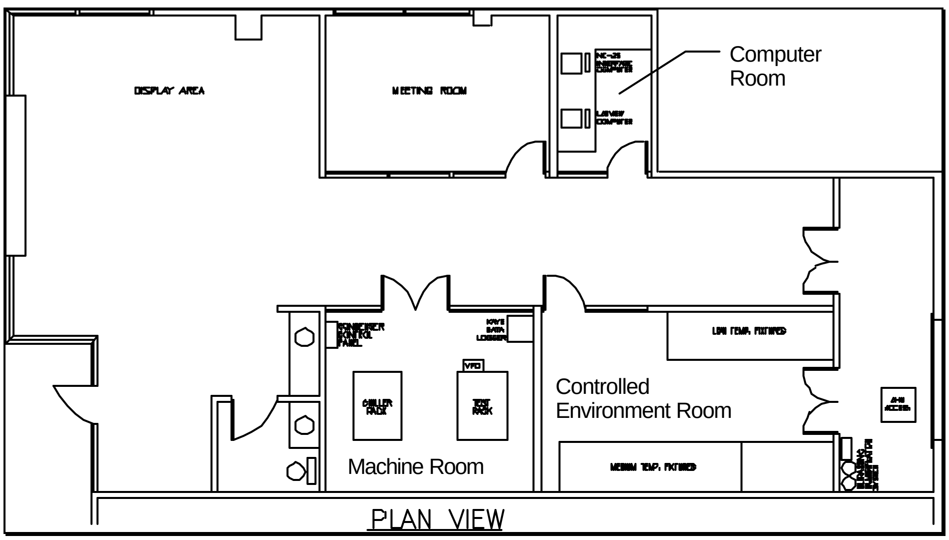

Figure 2 - RTTC Floor Plan

The controlled environment is an isolated thermal zone served by independent heating, cooling and humidification systems. The sensible cooling load representing people and other heat gain sources is provided by a constant volume direct expansion system reclaiming the waste refrigeration heat via a six row coil. While the air is conditioned to a desired thermostatic set point, an advanced ultrasonic humidification unit introduces precise moisture amounts to the air surrounding the display cases, representing the latent load due to outside air and people.

The refrigeration test rack, with the aid of a separate chiller system, can operate under various condensing temperatures. Additionally, the refrigeration system can be served by four individual heat rejection devices. The RTTC is capable of evaluating the performance of refrigeration systems under most defined loads, utilizing air-cooled, evaporatively-cooled, and water-cooled condensers.

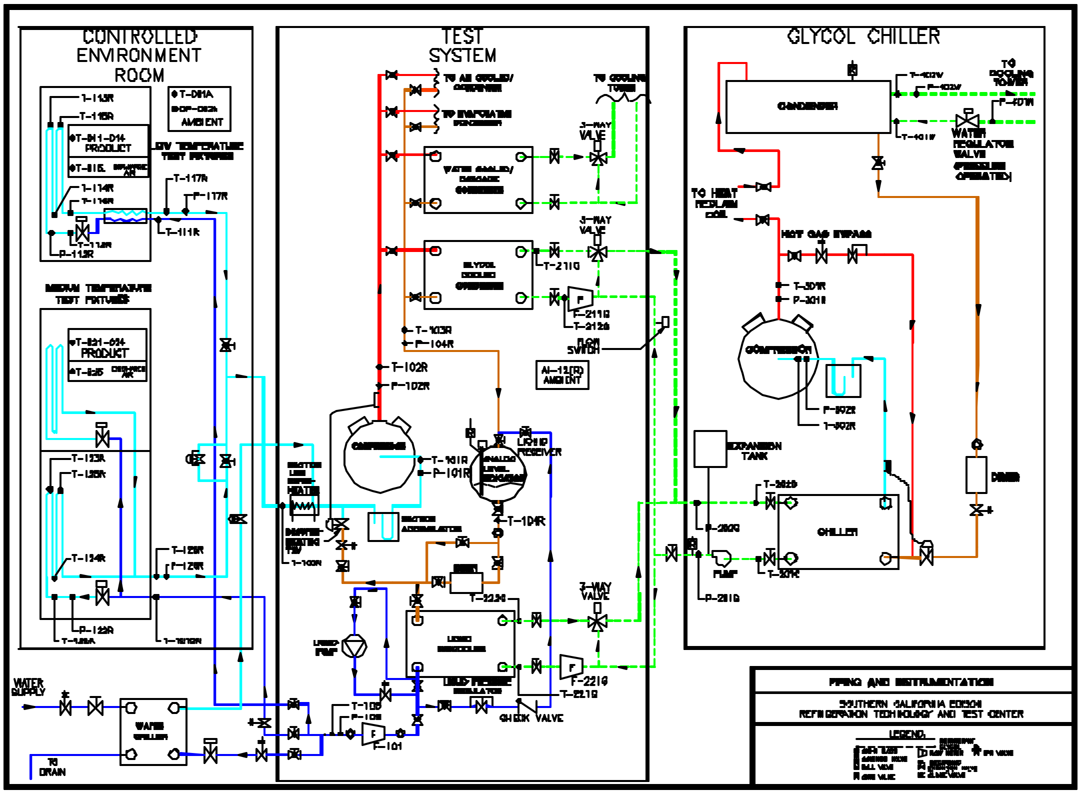

Figure 3 - RTTC Piping and Instrumentation

RTTC'S TEST EQUIPMENT

Refrigeration Test Rack

1. Major Components:

   a. Compressor:

      1 - Carlyle 6.5 HP, four cylinder, Model 06DR820 semi-hermetic compressor

   b. Condenser:

      1 - Alfa-Laval brazed plate-type condenser:

Heat Rate:	59,000 BTU/Hr
Flow Rate:	870 Lb./Hr. R-12; 13 GPM glycol

   c. Receiver Tank:

      Standard Refrigeration, P/N UV-70, 10" diameter X 28" high vertical receiver tank.

   d. Subcooler:

1 - Alfa-Laval plate-type subcooler:

Heat Rate: 12,400 BTU/Hr

Flow Rate: 870 Lb./Hr. R-12; 5 GPM glycol

2. Auxiliary Components:
   a. Suction Accumulator
   b. Suction Filter
   c. Crankcase Heater
   d. Oil Fail Switch, electronic type
   e. Suction De-superheating Expansion Valve
   f. Liquid Pump, Smith Precision Products Model GC-1, equipped with a Smith Precision Products Model WW-120 Liquid Bypass Differential Regulator.
   g. Danfoss YLT-3011 Variable Speed Drive for test compressor

Chiller Rack

1. Major Components:
   a. Compressor:

      1 - Carlyle 6.5 HP, four cylinder, Model 06DR820 semi-hermetic compressor.

   b. Condenser:

      1 - Standard Refrigeration shell and tube condenser, Model # SST755-8 Pass.

   c. Chiller:

      1 - Alfa-Laval brazed plate-type chiller:

      Heat Rate: 66,900 BTU/Hr

      Flow Rate: 977 Lb./Hr. R-22; 15 GPM glycol

Control System

The control system for the project's refrigeration equipment is as follows:

Danfoss NC-25 microprocessor-based control system, with analog input, analog output and digital I/O boards

Auxiliary Components

Three heating cables were provided for simulation of a long return suction line: Heating cables are tightly attached to the suction line with gear type clamps. The suction line and heaters are insulated with 2" fiberglass insulation with PVC jacket.

Medium Temperature Display Fixture

Cases, Tyler DDCM8 and DDCM12 - 4 rows of shelving, 20" wide.
Lights, 800 mA fluorescent, one row in the canopy.
Defrost Type: Ambient
This case is equipped with standard thermostatic expansion valves and LSHX

Low Temperature Display Fixture

Cases, Tyler D6F124 - 4 rows of shelving, 22" wide.
Lights, 430 mA fluorescent, two rows in the canopy and one row nose light.
Defrost Type: Electric
This case is equipped with thermostatic expansion valves and LSHX

HVAC Equipment for Controlled Environment Area

1. Air Handler, 1–Magic Aire Model 36 BH air handler (without coil)

   Motor: 3/4 HP, two speed

   Air Flow Rate @ External static Pressure: 1500 cfm @ 0.75 in-wg

2. Humidifier, 1-Type ENS Stulz Ultrasonic humidifier. The humidifier is installed inside the air stream of the distribution ductwork downstream of the heat reclaim coil.

3. Heat Reclaim Coil:

   Heat of Rejection: 89,950 BTU/Hr (available)

Major Data Acquisition System Components

1. Data Scanner, Kaye Instruments Digi-4 Model #X1520S. Kaye's Digi-4 has a special emphasis on temperature measurement, with excellent thermocouple signal processing. The scanner was calibrated at the factory, and is traceable to NIST standards.

   Analog inputs:

   a. Fifty-six special grade type T thermocouple inputs (+/- 0.03°C.)

   b. Fifteen precision 100 Ω Platinum RTD inputs (+/- 0.01°C.)

   c. Twenty-three analog inputs from pressure and other various transducers.

Outputs:

a. RS-232 Communication link, one report every 10 seconds. (A report includes instantaneous values of all data points)

2. Man/Machine Interface, IBM compatible 80486-based PC, running National Instruments' LabView 3.0 for Windows. A communications link was maintained with the data-scanning equipment at all times. The PC also provided the system with data storage and remote (modem) data access to the projects' test data.

   In general, the PC provided the following functions:

   a. Communicated continuously with the data scanner (via RS-232), acquiring one full data record (all scanned inputs) every 10 seconds. The data was then averaged over a 2-minute period, and the 2-minute averages written to storage (hard disk).

   b. Maintained modem communication capability (for contact by a remote computer) and when requested, transmitted user specified files to the remote system.

   c. Provided graphical representation of system performance data, both real-time and historical. Real-time graphic displays consisted of: 1) a schematic representation of the system, with the flows, temperatures, pressures, and other process data indicated on the screen, and 2) trend graph windows. The data to be displayed in the trend graph windows was user-configurable, with up to 18 variables in three separate graph windows.

Measurement Point Configuration

1. Refrigerant and Water/Glycol Line Precision Temperature Points

   A single 100 Ω Platinum RTD is located in a thermo-well installed in the appropriate line. Two high-accuracy type-T thermocouples located on opposing sides of the line within one or two inches of the RTD thermo-well. The thermocouples provide a confidence check on the RTD measurement, and are firmly mounted to the outside of the bare refrigeration line using a heat transfer compound.

2. Case Discharge Air and Controlled Environment Area Ambient Temperatures

   A single 100 Ω Platinum RTD is located in free air. Two high-accuracy type-T thermocouples located within one or two inches of the RTD. The thermocouples provide a confidence check on the RTD measurement.

3. Pressure Measurements

   Pressure Transducers are fitted to 1/8" NPT female side connections located on dedicated transducer valves.

4. Refrigerant Mass Flow

   The refrigerant mass flow meter is a corriolis effect device, installed in the refrigerant liquid line.

5. Water/Glycol Flow meters

   The water/glycol flow meters are turbine-type sensors.

6. kW (Power)

   The measurements were taken by factory calibrated CT-transducer combinations, which provided analog outputs. The instantaneous power measurements were integrated with respect to time, yielding kWh, and the kW measurements were taken in five places:

   - Input of the variable speed drive device
   - Input to the compressor motor
   - Input to the condenser fan-motor
   - Input to the display-case fan-motor
   - Input to the display-case lighting

7. Dew Point Sensor

   The dew point sensor utilizes chilled-mirror technology.

8. Informational Temperature Points

   These points are located at various points in the refrigeration and water/glycol circuits. These consist of a single, high-accuracy type-T thermocouple, mounted to the bare copper line using a heat transfer compound.

   Informational temperatures are also located in the display fixtures. High-accuracy, type-T thermocouples are located at various locations in each display fixture, on the coils and in the product containers, and at the mid-point of the air curtain.

Table 2 - List of Sensors Used

Sensor Type	Make / Model	Accuracy [NIST Traceable]
Dew Point	EG&G DewTrak Model 200	+/- 1° F
Refrigerant Mass Flow	Micro Motion Model DS065S	+/- 0.2%
Power (kW1)	Ohio Semitronics Model PC5-062BX680	+/- 0.5% F.S. (.04 kW)
Power (kW2)	Ohio Semitronics Model P-143B	+/- 1.0% F.S. (.08 kW)
Power (kW3)	Ohio Semitronics Model PC5-062BX680	+/- 0.5% F.S. (.04 kW)
Power (kW4)	Ohio Semitronics Model PC5-062BX680	+/- 0.5% F.S. (.04 kW)
Power (kW5)	Ohio Semitronics Model PC5-062BX680	+/- 0.5% F.S. (.04 kW)
Water/Glycol Flow	EG&G Flow Technology Model FT8-8NENW-LEG-1	+/- 1%
Pressure	Setra Transducers Model 207 - 100 & 500 PSIG Pressure Ranges	+/- 0.13%
Temperature (RTD)	Hy-Cal Engineering Model RTS-37-A-100	+/- 0.01°C
Temperature (T/C)	Kaye Instruments T/W 50 through 80; Melt # 8032	+/- 0.1°C

DATA ACQUISITION PROCEDURE

With the objective of minimizing instrument error and maintaining a high level of repeatability and accuracy in the data, careful attention was paid to the system design, and steps were taken to:

1. Use sensors of the highest accuracy available.
2. Minimize sensor drift errors by use of redundant sensors.
3. Utilize calibration standard instruments of the highest accuracy.
4. Eliminate interference from power conductors and high frequency signals by double-shielding sensor leads.

A Kaye Instruments Digi-4 Model #X1520S Data Scanner was used to log the data. Kaye's Digi-4 has a special emphasis on temperature measurement, with excellent thermocouple signal processing. The data scanner processes 94 data channels. The scanner was calibrated at the factory, and is traceable to NIST standards. The system has fifty-six special grade type-T thermocouples accurate to ± 0.03°C, fifteen precision 100Ω platinum Resistance Temperature Device (RTD) inputs accurate to ± 0.01°C, and a combination of twenty-three analog inputs from pressure transducers, dew point sensors, flow meters, and CT-transducers. A RS-232 communication link sent one data report that included instantaneous values of all data points every 10 seconds. To ensure that the data collection was not compromised by the control sequence's priority over data acquisition, the data acquisition system for the project was designed to be completely independent of the supervisory control computer.

Every 10 seconds the data acquisition system sampled the entire 94 point data array and created time-stamped two minute averages. The two minute data was then saved to a file which was closed at the end of each 24-hour period. Every 24 hours, the data collected from the previous 24 hours was downloaded remotely for screening and processing.

The raw data, as well as environmental conditions in the controlled room, were analyzed daily for consistency and accuracy. In the event that any of the test parameters fell outside acceptable limits, the problem was flagged. In such cases, test runs were repeated upon correction of the problem. Tests were conducted over twelve twenty-four hour periods at various target discharge air temperatures (See Table 3). The target discharge air temperatures were reset between 10:00 am and 11:00 am daily (during the cycle defrost) to their new target values.

TEST PROCEDURE

The shelves of the Tyler medium temperature dairy case were stocked with plastic gallon containers of water in an arrangement typically found in supermarkets. Water was used to represent milk due to its reasonably close specific heat capacity. Sixty minute time initiated, time terminated defrosts at six hour intervals were used in accordance with Tyler's recommendations to provide defrost for the cases. Supermarket shopper traffic was simulated by the use of an intermittently oscillating fan.

Space conditions were maintained in the controlled environment by a constant volume air handler. A six row coil connected to the chiller rack reclaimed the heat of refrigeration and provided the sensible heating to the space. To maintain humidity in the space, an ultrasonic humidifier, installed in the supply air duct downstream of the heat reclaim coil, injected moisture into the air stream. Motorized bypass dampers adjusted the flow of air through the heat reclaim coil to obtain the precise supply air temperature required.

The conditions within the controlled environment room were held constant at 75 ± 0.3 °F and 50 percent relative humidity.

Temperature measurements considered to be critical to the process were recorded from a group of three sensors (one RTD and two thermocouples). These data points were extracted from the daily files, and the readings from the thermocouples were compared to each other, and to the reading from the associated RTD. The difference between sensor readings was compared to criteria established during the commissioning of the data acquisition system. Any data where the maximum difference fell outside the allowable standard deviation was flagged for further review.

The two-minute refrigerant flow rate data was examined for continuity, as occasional “flashing” in the liquid line could adversely affect the mass flow meter’s accuracy. In such cases, a filter was employed to remove the suspect readings. (Note: no flashing was observed during these tests).

The operation of the compressor, condenser, display case, and HVAC system were controlled by the microprocessor controller. The microprocessor controller was equipped with a stand-alone modem for remote access to the control parameters. An interface with the microprocessor controller was made through a serial connection to a PC located in the computer room. Through this interface all parameters of microprocessor control could be modified and inspected.

The Variable Frequency Drive (VFD) modulated the compressor speed (and thereby its capacity and the refrigerant mass flow rate) according to inputs from the microprocessor controller. The microprocessor controller changed the VFD output to the compressor according to *differences between actual and target case discharge air temperature and suction pressure.*

Heating control for the controlled environment room was accomplished with motorized dampers at the air handler that modulated to adjust the amount of airflow through the heat reclaim coil. Separately, the ultrasonic humidifier varied the amount of water vapor released into the air stream according to inputs from the microprocessor controller. The microprocessor controller adjusted the magnitude of change for the heating and humidification of the space according to differences between the target and actual dry bulb temperatures and relative humidity, respectively.

For the test run, the display case target discharge air temperature was set at an initial value of 39°F and lowered one degree per 24-hour period to obtain a product temperature range of 45 °F to 41 °F. Target set points were changed during the defrost cycle between 10:00 and 11:00 A.M. Table 3 depicts the target set point settings during the twelve days of data collection. Throughout the test, the saturated condition temperature was maintained at a fixed value of 89.5 °F.

Table 3 - Test Target Set Points

Day of Test	Target Discharge Air Temperature	Target Suction Pressure
1	39	64.0
2	38	62.5
3	37	60.4
4	36	58.4
5	35	56.2
6	34	54.8
7	32	53.8
8	31	52.5
9	30	51.6
10	29	50.4
11	28	49.4
12	27	48.2

ANALYSIS

Data Collection/Reduction

The test facility is equipped with a sophisticated data acquisition system that scans 94 sensors and logs their outputs on two-minute intervals. Data was collected and stored for each sensor for 12 days. Every 24 hours during the test, the data was downloaded and checked for consistency and accuracy. Operating parameters were checked and deemed to be within acceptable limits before the next run was started. Figure 4 depicts the locations of various sensors within the display case.

The collected data points from the two-minute intervals were averaged into one hour blocks for each 24-hour period. After the hourly data was developed, the calculation tables were created. Figure 5 graphically presents the profile of four parameters -- average discharge air temperature, product temperatures at the top and bottom sensors, and saturated suction temperature -- as singular data points for each one hour block throughout the duration of the test. The gap in the graph during April 10th and 11th is due to a failure of the data acquisition system. For that period, no data points were collected.

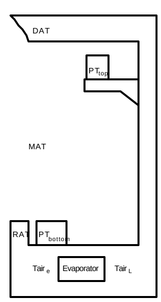

Figure 4 - Location of Sensors within Display Case

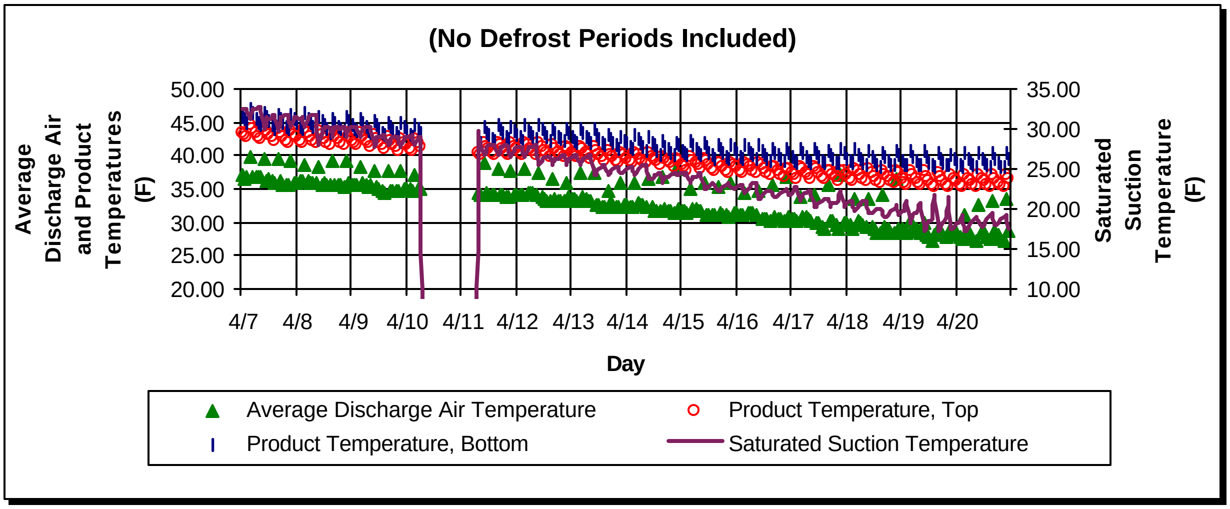

Figure 5 - Key Parameters Profile

Starting the test on April 7th and ending the test on April 20th achieved a 6 °F drop in average product temperature which corresponds to a 14 °F drop in the suction temperature (figure 5).

Figure 6 shows the collected data points representing the compressor power and mass flow rate profiles within the entire test duration. Figures 5 and 6 appear to have parallel data point lines. This less dense line of data points depicts the set of data points for the first hour after the defrost cycle termination. During defrost, the compressor did not operate, and hence, the system did not provide any cooling. Therefore, the case load and product temperatures increased, causing an increase in saturated suction temperatures, and thereby an increase in mass flow rate and compressor power during the first hour after defrost.

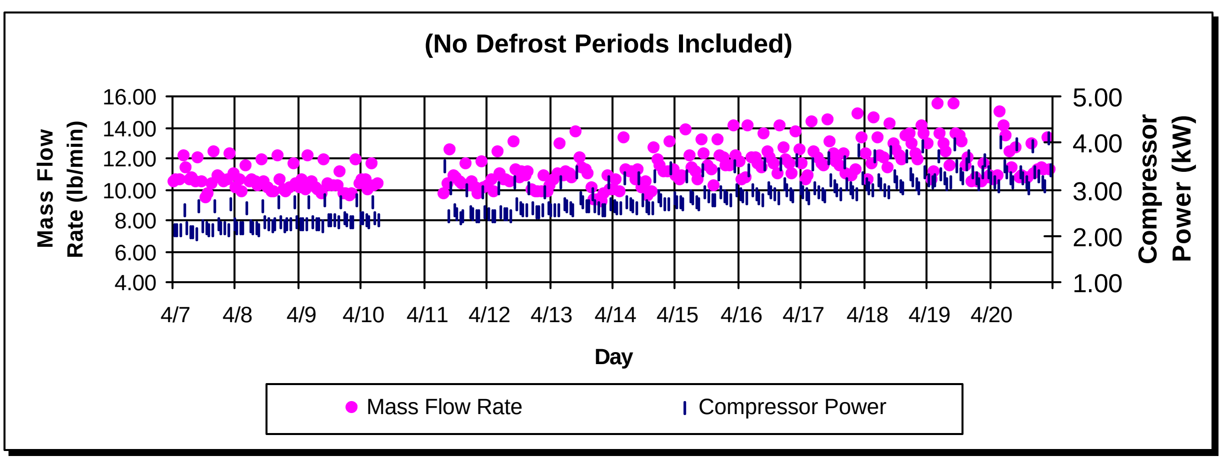

Figure 6 - Mass Flow Rate and Compressor Power Profiles

Screening Procedure

Once the data was compiled into hourly averages within the spreadsheet, tabular and graphical representations of various correlations and calculated parameters were produced. Over 20 graphs comprising three classes of information were created – two sets of sanity checks and one set of representational plots. The first set of sanity checks (Level I Sanity Checks) are located in Appendix B. Included in this group were the fundamental data points provided by the data acquisition system. These plots were used to determine the validity of each test.

After reviewing the first set of sanity checks and developing initial confidence in test results, more detailed screening of the data was performed (see Level II Sanity Checks in Appendix C). These plots related the saturated suction temperature to other data points, as well as to basic calculations. After careful examination of the two sets of sanity checks, the final group of plots were produced. This set provided relationships between the product temperature and points of interest to the Edison supermarket customers.

Calculations

A series of calculations were performed to obtain the key refrigeration parameters including refrigeration effect, refrigeration load of the case, heat rejection at the condenser, heat of compression, and compressor power per ton.

After the data was downloaded from the data logger and the data of interest was extracted, some preliminary calculations were performed. These calculations included averaging temperature data that were read by more than one sensor. Such data included discharge air temperature, product temperature, temperature at the evaporator inlet, saturated suction temperature, temperature at the evaporator exit, and temperature of refrigerant entering the expansion valve.

The refrigeration load of the case, heat rejection at the condenser, and compressor power per ton all depend on the refrigerant available heat or enthalpies at different locations of the refrigeration cycle. Enthalpies can be either obtained from the refrigerant manufacturer's data at various temperatures and pressures, or calculated with respect to specific heat capacities and temperatures.

Once the temperatures and pressures were determined, the enthalpies were obtained. DuPont's Suva Refrigerant Expert Program version 2.0 was used to determine the enthalpies of superheated vapor. The data logger provided all pressures in gage unit, and after conversion to absolute, the Refrigerant Expert was used to look up the enthalpies. The Refrigerant Expert program was used to obtain enthalpies at the evaporator exit, and the compressor and condenser inlets.

The enthalpies in the saturated phase were calculated using temperature dependent expressions provided by DuPont, as well as using basic thermodynamic relationships. Equation 1, provided by DuPont, determined the enthalpy in kJ/kg of refrigerant 404A for a temperature range of -20 °C to 40 °C. The temperatures of the saturated liquid were first converted to Celsius, then inserted into equation 1 to produce the corresponding saturated enthalpy. The enthalpy at the condenser exit and the enthalpy at saturated temperature and discharge pressure were found by use of the equations 1 and 2.

1) $H = A + BT + CT^2$

H = Enthalpy (kJ/kg)
A = 200
B = 1.438333
C = 0.003916667
T = Temperature (°C)

where A, B, and C were constants determined by DuPont from the relationship between saturated enthalpy and temperature. Next, equation 2 was used to convert the enthalpy in kJ/kg to Btu/lb. Because of a change in reference states from SI to English units, a reference conversion, H (ref), was included in Equation 2.

2) $H\ (\text{Btu/lb}) = [H\ (\text{kJ/kg}) - H\ (\text{ref})] \cdot 0.43021(\text{Btu/lb / kJ/kg})$

H (ref) = 145.6 kJ/kg for R404A

The Refrigerant Expert program does not provide subcooled enthalpies, therefore, in order to find the enthalpies for subcooled refrigerant, the thermodynamic relationship between enthalpy and temperature was incorporated. For this relationship, however, the correct liquid specific heat capacity was needed. Equation 3, provided by DuPont, calculated the liquid specific heat capacity of refrigerant 404A for a temperature range of -40 °F to 140 °F. The enthalpy at the entrance to the expansion valve was found using this equation and subcooled temperature.

3) $Cp = 0.306 + 4.083E\text{-}4\ T - 1.194E\text{-}6\ T^2 + 8.056E\text{-}8\ T^3$

Cp	=	Liquid Heat Capacity (Btu/lb °F).
T	=	Average temperature of the subcooled liquid (°F) for a range of -40 °F to 140 °F

The temperature difference between the subcooled liquid entering the expansion valve and the saturated liquid leaving the condenser was needed in order to find the corresponding enthalpy change due to subcooling.

4) $\Delta Tsubcool = Tsat - Tsubcool_{avg}$

ΔTsubcool	=	Temperature difference between subcooled liquid entering the expansion valve and saturated liquid leaving the condenser (°F).
$Tsubcool_{avg}$	=	Average pre-expansion valve temperatures (°F). This value was read directly using the data acquisition system.
Tsat	=	Temperature of Saturated Liquid at Saturated Discharge Pressure (°F). This value was determined by look-up using DuPont's Suva Refrigeration Expert Program and saturated discharge pressure data from the data acquisition system.

Next, the enthalpy change between the subcooled and saturated liquid was calculated by utilizing the following thermodynamic relationship.

$$5)\qquad \Delta Hsubcool = Cp \cdot \Delta Tsubcool$$

Finally, the enthalpy of the subcooled liquid was computed. In order to accomplish this, the enthalpy change between the subcooled and saturated liquid was subtracted from the enthalpy of saturated liquid.

$$6)\qquad Hsubcool = Hsatliq - \Delta Hsubcool$$

ΔHsubcool	=	Enthalpy change between subcooled liquid entering the expansion valve and saturated liquid leaving the condenser (Btu/lb).
Hsubcool	=	The subcooled liquid enthalpy entering the expansion valve (Btu/lb).
Hsatliq	=	Saturated liquid enthalpy (Btu/lb). This value was determined using equations 1 and 2.

After determination of all enthalpy values, calculations were made to determine parameters of interest such as refrigeration effect, refrigeration load of the case, heat of compression, heat rejection at the condenser, sensible load of the evaporator, compressor power per ton, and coefficient of performance.

Refrigeration effect is the quantity of heat that each unit mass of refrigerant absorbs to cool the refrigerated space. It simply represents the capacity of the evaporator per pound of refrigerant. It was calculated by subtracting the enthalpy of subcooled liquid entering the expansion valve from the enthalpy of superheated refrigerant at the evaporator exit (equation 7). Because refrigeration effect is directly proportional to saturation suction temperature, it is affected by changes in product temperature or discharge air temperature settings, and therefore saturated suction temperature.

$$7)\qquad RE = Hevap_{exit} - Hsubcool$$

RE	=	Refrigeration Effect (Btu/lb)
$Hevap_{exit}$	=	Enthalpy of superheated refrigerant vapor at the evaporator exit (Btu/lb).

The refrigeration load of the case is the amount of cooling or heat removal that takes place at the evaporator of the display case per hour (equation 8). This value is inversely proportional to saturation suction temperature, and is thus affected by changes in product temperature or discharge air temperature settings.

$$8)\qquad Qcase = mfr \cdot k \cdot RE$$

Qcase	=	Refrigeration Load of the case (Btu/hr)
mfr	=	Mass flow rate of refrigerant (lb/min)
k	=	Conversion Factor, 60 (min/hr)

Typically, the refrigeration load of the case is expressed on a per linear foot basis. Therefore, the refrigeration load per foot of the case was calculated by dividing the refrigeration load of the case by 20 feet, the length of the test case (equation 9).

9) $Qcase/ft = Qcase / 20ft$

Qcase/ft = Refrigeration Load per foot of the case (Btu/ft-hr)

It is also important to determine the refrigeration load of the case in tons. Thus, the refrigeration load of the case was divided by 12,000, a conversion factor from Btu/hr to tons (equation 10).

10) $Qcase\ (tons) = Qcase / 12000$

Qcase (tons) = Refrigeration Load of the case (tons)

Obtaining the heat of compression is also of interest because it can be used to calculate the amount of heat rejection at the condenser and the theoretical compressor power necessary to provide cooling for the system. Because the heat of compression is the difference between the enthalpies at the suction and discharge sides of the compressor, this value is also affected by variations in suction temperature (equation 11). The saturated condensing temperature remained unchanged at 89.5 °F throughout the test. Therefore, the superheated, high pressure enthalpy of refrigerant vapor at the condenser inlet was used instead of the refrigerant enthalpy at the compressor discharge to obtain the heat of compression.

11) $Qcomp = mfr \cdot k \cdot (Hcond_{in} - Hcomp_{in})$

Qcomp	=	Heat of Compression (Btu/hr)
k	=	Conversion Factor, 60 (min/hr)
$Hcond_{in}$	=	Enthalpy at the inlet to the condenser (value determined by look-up using DuPont's Suva Refrigeration Expert Program)
$Hcomp_{in}$	=	Enthalpy at the inlet to the compressor (value determined by look-up using DuPont's Suva Refrigeration Expert Program)

The heat rejection at the condenser was calculated based on the total heat of the system. The total heat of the system is a function of the sum of heat removal rate at the evaporator and heat of compression. The total heat rejected at the condenser can also be obtained by the product of mass flow rate and change in refrigerant enthalpies between the inlet and outlet of the condenser which includes de-superheating, latent (or phase change), and subcooling heat removals within the condenser.

The analysis, however, evaluated the heat rejection based on the enthalpies at the condenser inlet and saturated liquid (equation 12). This approach yielded results less than 1 percent difference versus total heat of the system (heat removal at the evaporator plus heat of compression).

12) $Qcond = mfr \cdot k \cdot (Hcondin - Hsatliq)$

Qcond	=	Heat Rejection at the Condenser (Btu/hr)
k	=	Conversion Factor, 60 (min/hr)

The sensible load of the evaporator is the amount of cooling the evaporator provides to reduce the air temperature without removing any of its moisture content. It was calculated by utilizing equation 13.

$$13)\qquad Qsenscoil = CFM \cdot \rho_{air} \cdot Cp_{air} \cdot k \cdot \Delta Tair$$

Qsenscoil	=	Sensible load of the evaporator (Btu/hr)
CFM	=	Volume flow rate of air (ft^3/min)
ρ_{air}	=	0.0749 lb/ft^3
Cp_{air}	=	0.24 Btu/lb °F
k	=	Conversion Factor, 60 (min/hr)
ΔTair	=	Air temperature drop across the evaporator (°F)

To calculate the volume flow rate of air across the evaporator, the cross sectional area of the discharge air grill was multiplied by the velocity of the air across the coil (equation 13a).

$$13a)\qquad CFM = V \cdot A$$

V = Velocity of air (ft/min). This value was assumed based on various manufacturer's data.
A = Cross Sectional Area (ft^2)

The cross sectional area of the air discharge grill is a product of the width of the air discharge grill and the length of the refrigeration case (equation 13b).

$$13b)\qquad A = L \cdot W$$

L	=	Length of Refrigeration Case (ft)
W	=	Width of air discharge grill (ft)

The coefficient of performance is a dimensionless ratio that determines the efficiency of the system. It is the relationship between how much cooling the system provides (refrigeration load of the case, Btu/hr) and how much power the system uses (compressor power, Btu/hr) (equation 14). Consequently, the larger the coefficient of performance, the more efficient the cycle performs.

$$14)\qquad COP = Qcase/Qcomp$$

COP = Coefficient of performance

The evaporator coil cools the air, and the cooled air in turn removes the heat gained by the display case. In cooling the air, the evaporator sensibly cools the air, and as a result, air temperature drops. It also provides latent cooling of the air which dehumidifies the air, and thereby lowers its moisture content. The sensible heat ratio is a characteristic index for the coil and is the ratio of sensible cooling to total cooling (sensible and latent) of the evaporator coil (equation 15). It indicates how well the coil removes air moisture.

15) SHR = Qsenscoil/Qcase

SHR = Sensible Heat Ratio

Table 4 provides a summarized version (of calculated results) for a sample 24 hour test period. The blue rows indicate the defrost periods which were ignored in the analysis. During defrosts, to make heat available to the heat reclaim coil, a false load was imposed on the refrigeration system. During these periods, the liquid line was routed into a plate heat exchanger to remove the false load of city water passing through the heat exchanger. Hence, defrost values (shown in blue rows) are not relevant to the test.

Table 4 - Calculated Results (for a sample 24-hour test period)

hour	RE	Qcase	Qcase/ft	Qcase	Qcomp	Qcond	Qsenscoil	MFR/ton	COP	Effactual	SHR
	(Btu/lb)	(Btu/hr)	(Btu/hr ft)	(Tons)	(Btu/hr)	(Btu/hr)	(Btu/hr)	(lb/min-ton)		(kW/ton)	
1	56.7	35535	1777	2.96	7558	42737	34078	3.53	4.70	0.72	0.96
2	56.4	35954	1798	3.00	7644	43422	34739	3.55	4.70	0.71	0.97
3	56.5	35791	1790	2.98	7580	43158	35038	3.54	4.72	0.70	0.98
4	66.2	46720	2336	3.89	9289	49437	23024	3.02	5.03	0.93	0.49
5	57.8	42132	2107	3.51	9088	50260	31184	3.46	4.64	0.73	0.74
6	56.4	38504	1925	3.21	8191	46466	32655	3.55	4.70	0.68	0.85
7	56.5	36066	1803	3.01	7621	43438	32853	3.54	4.73	0.70	0.91
8	56.6	36388	1819	3.03	7661	43735	34041	3.53	4.75	0.68	0.94
9	56.6	35483	1774	2.96	7502	42616	34970	3.53	4.73	0.69	0.99
10	73.4	48602	2430	4.05	8850	46539	21907	2.72	5.49	0.80	0.45
11	59.2	42774	2139	3.56	9422	50189	30658	3.38	4.54	0.74	0.72
12	56.6	35382	1769	2.95	7818	42950	31811	3.54	4.53	0.75	0.90
13	57.0	32084	1604	2.67	7183	38710	32052	3.51	4.47	0.81	1.00
14	56.6	32796	1640	2.73	7355	39815	32783	3.53	4.46	0.78	1.00
15	56.6	34923	1746	2.91	7806	42408	32874	3.53	4.47	0.73	0.94
16	65.8	46755	2338	3.90	9508	50046	21231	3.04	4.92	0.93	0.45
17	58.0	43020	2151	3.59	9772	51688	30811	3.45	4.40	0.73	0.72
18	56.5	36650	1832	3.05	8107	44473	32059	3.54	4.52	0.73	0.87
19	56.4	36417	1821	3.03	7963	44183	32737	3.55	4.57	0.72	0.90
20	56.4	35467	1773	2.96	7744	42989	33715	3.54	4.58	0.72	0.95
21	56.4	35608	1780	2.97	7736	43144	34492	3.55	4.60	0.72	0.97
22	67.5	51323	2566	4.28	9907	53289	20992	2.96	5.18	0.86	0.41
23	58.2	42685	2134	3.56	9311	50708	30299	3.43	4.58	0.75	0.71
24	56.4	37064	1853	3.09	8038	44895	32110	3.54	4.61	0.72	0.87
Daily Avg.	**58.8**	**38922**	**1946**	**3.24**	**8277**	**45471**	**30963**	**3.42**	**4.69**	**0.75**	**0.82**
St. Dev.	**4.6**	**5246**	**262**	**0.44**	**856**	**3909**	**4408**	**0.23**	**0.25**	**0.07**	**0.19**

DISCUSSION OF RESULTS

Uneven product temperature distribution at the top and bottom shelves was one of the first observations in this test. While the products in the upper shelves held a lower temperature than the products in the bottom shelves during the entire test period, the temperature difference increased as the product temperature was lowered to 41 °F (figures 7a and 7b).

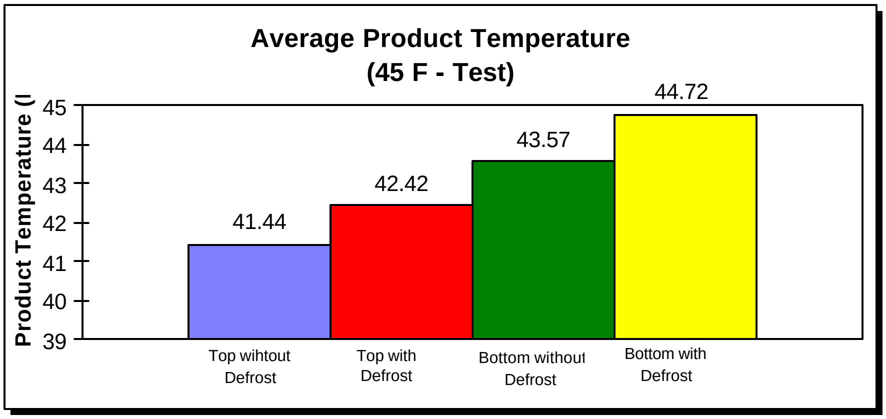

Figure 7a - Product Temperature Variation During the 45 ºF Test

Figures 7a and 7b were developed based on two representative test days (April 10th and 18th). They were chosen because the maximum average hourly product temperatures at the top and bottom (including defrost periods) did not exceed the set maximum temperatures. During April 10th and 18th, product temperatures stayed reasonably close to 45 °F and 41 °F, and yet did not exceeded them.

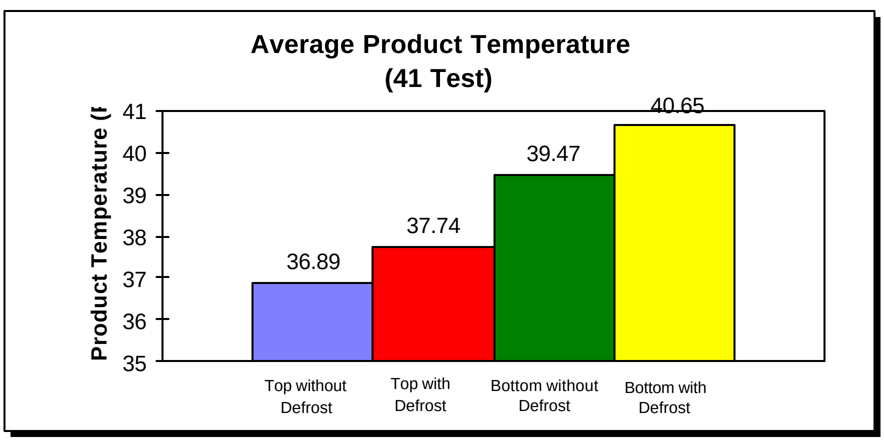

Figure 7b - Product Temperature Variation During the 41 ºF Test

Entrainment of the warm test room's air (at 75 °F) into the cold case and the mixing effect that takes place within the air curtain plane can increase the temperature of products near the return air grill, as well as the return air temperature. In maintaining a product temperature of 45 °F it was observed that products located on the top shelf had approximately 2 °F lower temperature than those on the bottom shelf, and under the 41 °F scenario, this difference was increased to 3 °F.

It was also observed that during the first hour after defrost termination, the evaporator coil sensible heat ratio dropped by about 23 percent. This reduction in the sensible heat ratio indicates a larger quantity of latent load that is imposed on the coil due to moisture migration from the room to the case when the system is in defrost.

By lowering the product temperature, the rate of heat gain to the case due to radiation, infiltration, and conduction increases. The 4 °F reduction in product temperature caused the refrigeration load of the case to increase by 14.6 percent (figure 8). Additionally, the refrigeration effect of the cycle decreased by 3.0 percent (figure 8), indicating a loss in capacity of the case when operating at a lower suction temperature.

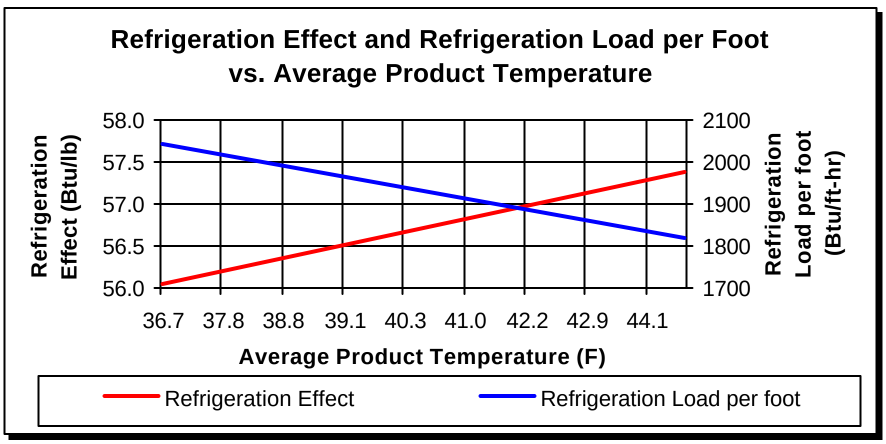

Figure 8 - Refrigeration Effect and Refrigeration Load per foot
versus Average Product Temperature

The saturated suction temperature drops at a much faster rate than did the average discharge air temperature. Lowering the product temperature from 45 °F to 41 °F resulted in a 7.2 °F reduction in saturated suction temperature and a 5.2 °F decrease in discharge air temperature (figures 9 and 10). Figure 9 shows the close linear relationship between saturated suction and product temperatures. The saturated suction temperature decreased from 28.9 °F to 21.7 °F as the product temperature was lowered from 45 °F to 41 °F.

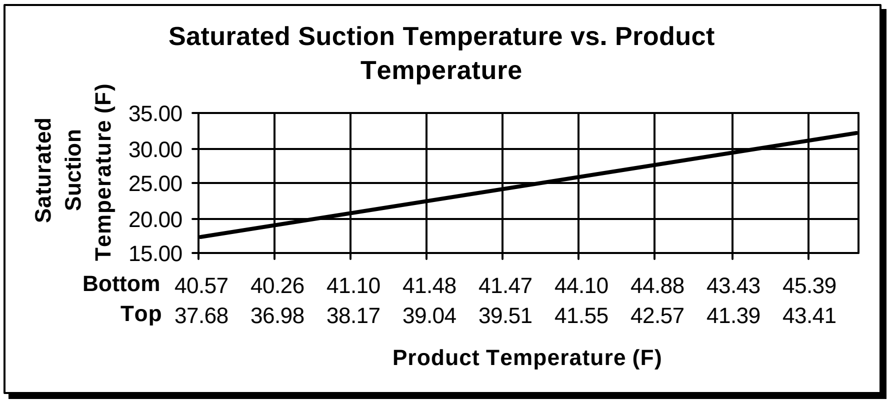

Figure 9 - Saturated Suction Temperature
versus Product Temperature

The temperature difference between the discharge air temperature and the return air temperature stayed within 10 °F for most test runs, and slightly increased at lower product temperatures (figure 10). The average discharge air temperature decreased from 37.3 °F to 32.1 °F and the return air temperature decreased from 47.9 °F to 43.0 °F as the product temperature was lowered from 45 °F to 41 °F.

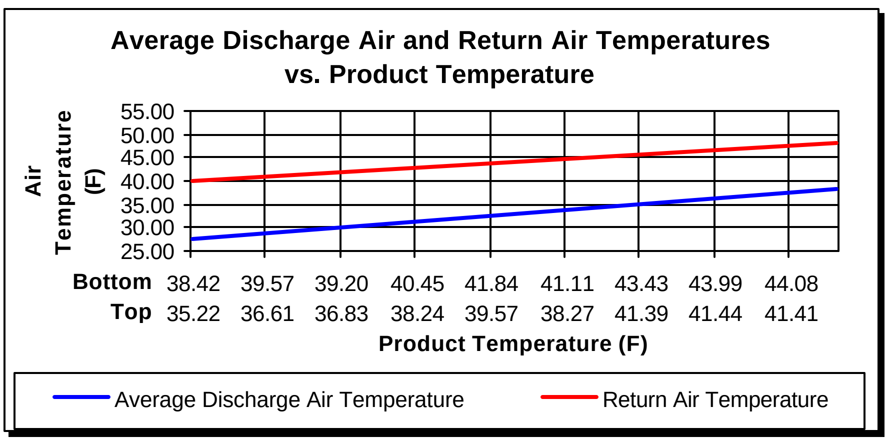

Figure 10 - Average Discharge Air and Return Air Temperatures
versus Product Temperature

One of the main impacts of operating under lower suction temperatures is on *compressor power*. Low suction temperatures caused the compressor to work harder to raise the system pressure from lower levels to a set saturated discharge pressure, thereby increasing the heat of compression. Additionally, at a lower product temperature, the case's heat removal rate increased causing a 18.3 percent increase in the refrigerant mass flow rate (figure 11). Figure 11 depicts the change in mass flow rate as a function of product temperature. The mass flow rate of refrigerant increased from 10.5 lb/min to 12.5 lb/min as the product temperature was lowered from 45 °F to 41 °F. The combined effects of added heat of compression and increased mass flow rate resulted in an increase of 31.2 percent in compressor power consumption (figure 12).

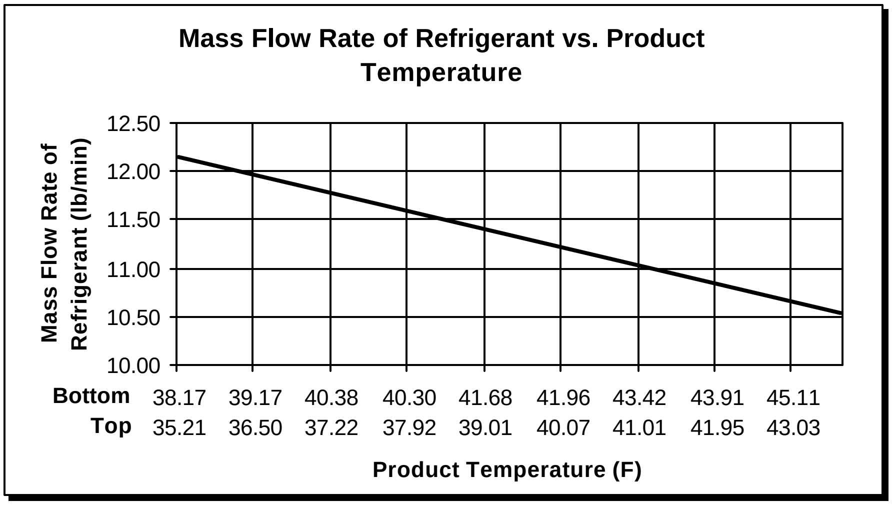

Figure 11 - Mass Flow Rate of Refrigerant versus Product Temperature

Figure 12 presents the change in compressor power and compressor power per ton as the average product temperature was decreased from 45 °F to 41 °F. The compressor power increased from 2.6 kW to 3.4 kW and the compressor power per ton increased from 0.82 kW/ton to 0.95 kW/ton as the product temperature was lowered from 45 °F to 41 °F. This implies for every degree Fahrenheit decrease in product temperature, the system kW/ton rating and power deteriorated by 3.8 percent and 7.8 percent, respectively.

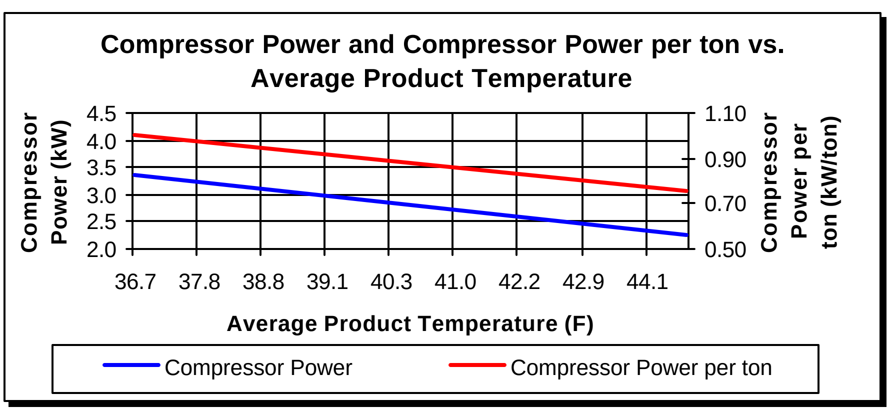

Figure 12 - Compressor Power and Compressor Power per ton versus Average Product Temperature

The FDA's 1993 temperature recommendations can noticeably increase refrigeration load and power consumption of dairy display cases. The variation of product temperature between the top and bottom shelves may cause a problem with targeting and maintaining a uniform product temperature within the case to comply with the FDA requirements. This may create undesirable cold temperatures for products located on top shelves when the temperature of the products on the bottom shelves is targeted to comply with the code. It must be noted that the product temperatures should be lowered below the FDA's requirement, so that by termination of the defrost cycle, the product does not exceed the allowable temperature limits. Under this condition, some under or over cooling of the products may occur, that could have adverse impacts on the quality, safety and shelf life of the products.