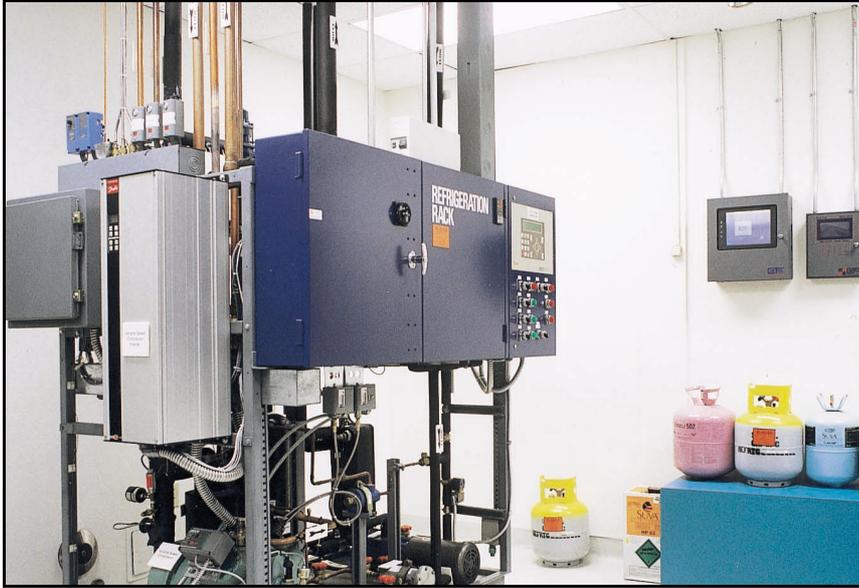


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The refrigeration rack (left) provides cooling to the display cases in the controlled environment. Water was used to represent milk during testing.

Colder Temperatures in Display Cases

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Meeting 1993 FDA Code Increases Energy Use And Refrigeration Load

Each year in the United States there are an estimated 10,000 deaths and 24 to 81 million illnesses attributed to micro-organisms in food.¹ Prompted by these statistics, the Food and Drug Administration's (FDA) new food code incorporates a framework for applying the Hazard Analysis Critical Control Point (HACCP), the best known available system for assuring food safety. In 1993 the FDA's food code recommended a lowered shipping, receiving, and storage temperature for potentially hazardous food (meat, dairy, deli, fish, poultry, and cut produce) for further prevention of food-borne diseases.

The food code is revised every two years, but the food temperature recommendations have remained unchanged in the 1995 and 1997 editions. According to the FDA's 1993 food code,² it is required that throughout receiving, slacking, and storing processes the temperature of potentially hazardous food be kept at

41°F (5°C). The 41°F (5°C) requirement is 4°F (2.2°C) lower than the previous version of the code. The objective of the FDA's food code is to implement preventative measures at the shipping, receiving, and storage stages rather than detect problems in the finished product.

Because 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. The energy systems' operations are sensitive to the impact of various food and energy codes.

The purpose of this project was to test and evaluate the impact of the FDA's 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 Edi-

son (SCE) conducted this test at its state-of-the-art Refrigeration Technology and Test Center (RTTC), located in Irwindale, Calif. 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. Based on the results of the RTTC's test, the new FDA food code could adversely affect the refrigeration load and power consumption of a dairy display case.

Test Procedure

A 6.5 hp (4.8 kW), semi-hermetic,

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Day of Test	Target Discharge Air Temperature	Target Suction Pressure
1	39°F	64.0 psig
2	38°F	62.5 psig
3	37°F	60.4 psig
4	36°F	58.4 psig
5	35°F	56.2 psig
6	34°F	54.8 psig
7	32°F	53.8 psig
8	31°F	52.5 psig
9	30°F	31.6 psig
10	29°F	50.4 psig
11	28°F	49.4 psig
12	27°F	48.2 psig

Table 1: Test target set points.

reciprocating compressor system served by a glycol condenser maintained a fixed saturated condensing temperature. This provided cooling to the medium temperature dairy display cases. The refrigeration system used R-404A, which is a hydrofluorocarbon (HFC) refrigerant, and polyol ester lubrication oil. The shelves of the display cases were stocked with plastic gallon containers of water in a typical arrangement found in supermarkets. Because of its similar specific heat capacity, water was used to represent milk. In accordance with the manufacturer's recommendation, 60-minute, time-initiated, time-terminated defrosts at six hour intervals were used to provide defrost for the cases. An oscillating fan simulated supermarket shopper traffic. The shelf lights remained on for the duration of the test.

A constant volume air handler maintained space conditions in the controlled environment. 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 airstream. 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 $75 \pm 0.3^\circ\text{F}$ ($24 \pm 0.2^\circ\text{C}$) and 50% rh which are slightly different than ASHRAE-specified conditions. These selected indoor test conditions are believed to best represent the actual indoor operating conditions of major supermarkets in SCE's service territory.

The microprocessor controller managed the operation of the compressor, condenser, display case, and HVAC system. 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. All microprocessor control parameters could be modified and inspected using this interface. 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.

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 (3.8°C) and lowered one degree (0.55°C) per 24-hour period (see *Table 1*), to obtain a product temperature range of 45°F to 41°F (7°C to 5°C). Target set points were changed during the defrost cycle between 10 and 11 a.m. *Table 1* depicts the target set point settings during the 12 days of data collection. Throughout the test, the saturated condensing temperature was maintained at a fixed value of 89.5°F (32°C).

Data Collection/Reduction

Every 24 hours during the test, data was downloaded and checked for consistency and accuracy. Operating parameters were checked and verified to be within acceptable limits before the next run was started. 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 1* presents the profile of four parameters—average discharge air temperature, product temperatures at the top and bottom shelves, and saturated suction temperature—as singular data points for each one hour block throughout the duration of the test. A failure of the data acquisition system caused the gap in the graph during April 10 and 11. For that period, no data points were collected.

Between the start of the test on April 7 and its end on April 20, the average product temperature dropped 6°F (3°C), which corresponded to a 14°F (7.7°C) drop in the suction temperature (*Figure 1*). *Figure 2* shows the collected data points representing the compressor power and mass flow rate profiles within the entire test duration. The less dense line of data points in *Figure 2* depicts the set of data for the first hour after the defrost cycle termination. During defrost, the compressor did not provide cooling to display cases, and hence, the case load and product temperatures increased. This caused an increase in saturated suction temperatures and in mass flow

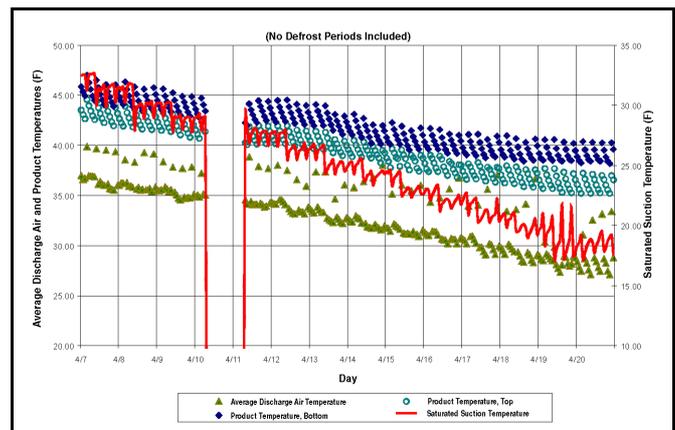


Fig. 1: Key parameters profile.

rate and compressor power during the first hour after defrost.

Power Consumption and Cooling Load Increased

Table 2 and Figure 3 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 (2.7°C) (from 37.3°F to 32.1°F [3°C to 0°C]) and the saturated suction temperature of the system to be lowered by about 7°F (3.8°C) (from 28.9°F to 21.7°F [−1.7°C to −5.7°C]), while the mass flow rate will be increased by 18.3%.

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 (5°C) (Figures 4 and 5).

Figures 4 and 5 were developed based on two representative test days (April 10 and 18). 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 10 and 18, product temperatures stayed reasonably close, yet did not exceed, 45°F and 41°F (7°C to 5°C).

Entrainment of the warm test room’s air (at 75°F [24°C]) 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. To maintain a product temperature of 45°F (7°C), products located on the top shelf had approximately 2°F (1.1°C) lower temperature than those on the bot-

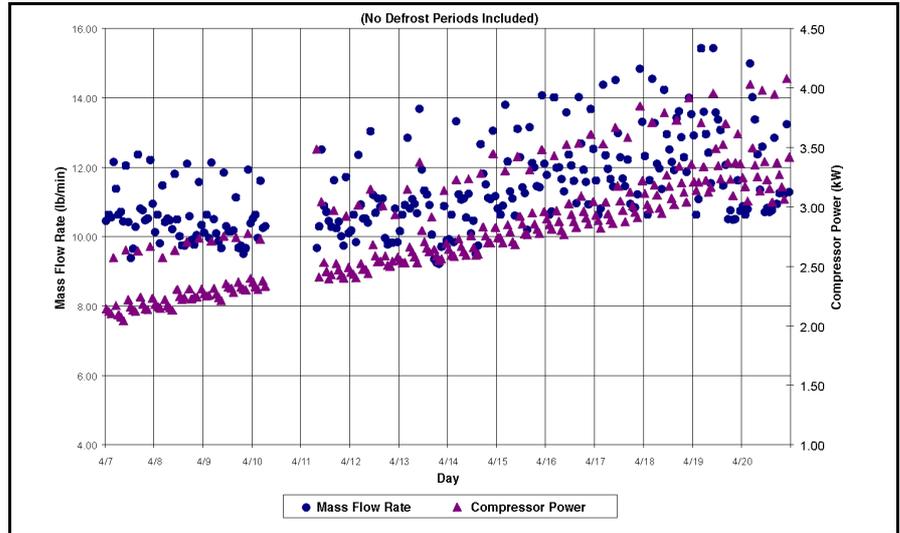


Fig. 2: Mass flow rate and compressor power profiles.

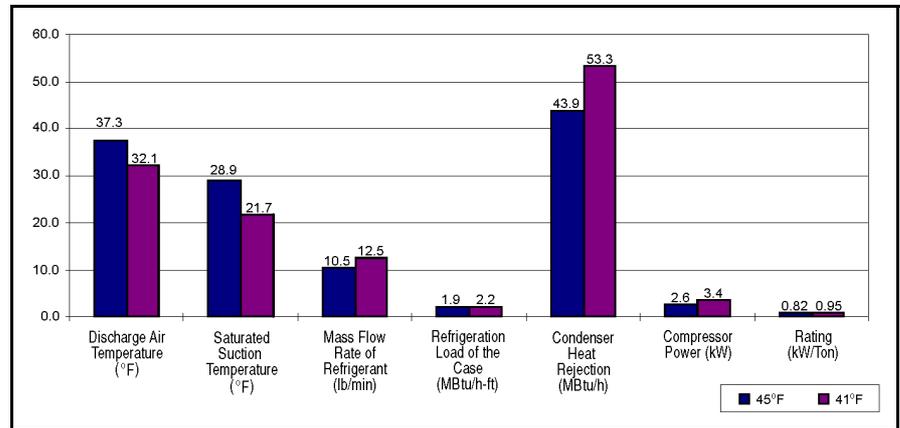


Fig. 3: Effect of the FDA code on refrigeration system parameters.

tom shelf, and under the 41°F (5°C) scenario, this difference was increased to 3°F (1.6°C). After analyzing the test data and conducting calculations, it was determined that during the first hour after defrost termination, the evaporator coil sensible heat ratio dropped by about 23%. This reduction in the sensible heat ratio indicates a large quantity of latent load that is imposed on the coil because of 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 increases because of radiation, infiltration, and conduction. The 4°F (2.2°C) reduction in product temperature caused the refrigeration load of the case to increase by 14.6% (Figure 6). Additionally, the refrigeration effect of the cycle decreased by 3.0% (Figure 6), indicating a loss in capacity of the case when operating at a lower suction temperature.

The saturated suction temperature dropped at a much faster rate than did the average discharge air temperature. Low-

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 (7°C)	37.3 (3°C)	28.9 (−1.7°C)	10.5	1.9	43.9	2.6	0.8
41°F (5°C)	32.1 (0°C)	21.7 (−5.7°C)	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%

Table 2: Effect of the FDA code on refrigeration system parameters.

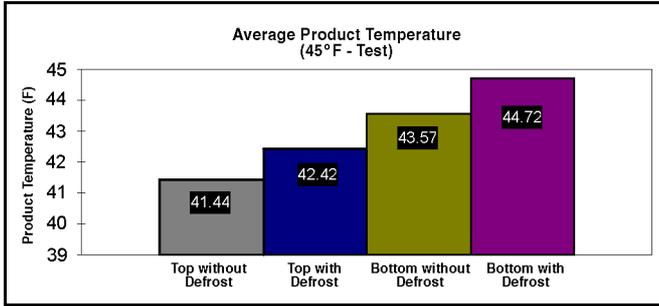


Fig. 4: Product temperature variation during the 45°F (7°C) test.

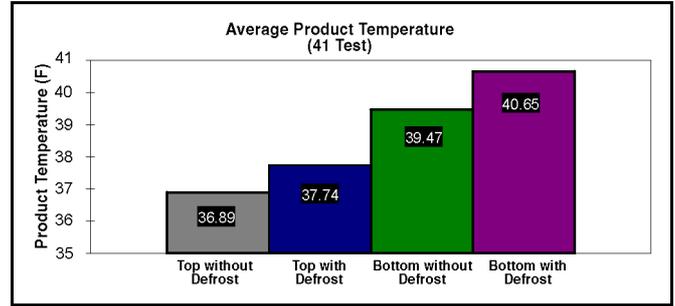


Fig. 5: Product temperature variation during the 41°F (5°C) test.

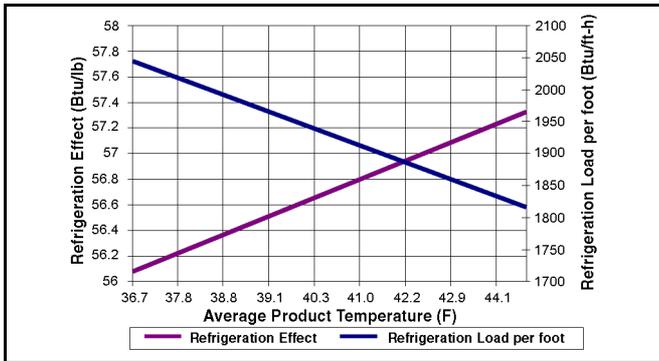


Fig. 6: Refrigeration effect and refrigeration load per foot versus average product temperature.

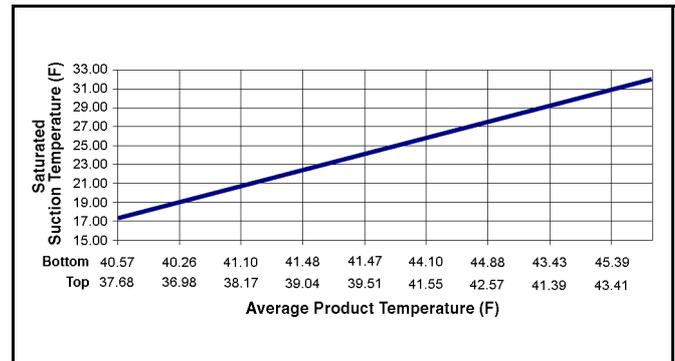


Fig. 7: Saturated suction temperature versus average product temperature.

ering the product temperature from 45°F to 41°F (7°C to 5°C) resulted in a 7.2°F (3.9°C) reduction in saturated suction temperature and a 5.2°F (2.8°C) decrease in discharge air temperature (Figures 7 and 8). Figure 7 shows the close linear relationship between saturated suction and product temperatures. The saturated suction temperature decreased from 28.9°F to 21.7°F (−1.7 to −5.7°C) as the product temperature was lowered from 45°F to 41°F (7°C to 5°C).

Figure 8 relates the average discharge and return air temperatures to the product temperature. The temperature difference between the discharge air temperature and the return air temperature stayed within 10°F (5.5°C) for most test runs, and slightly increased at lower product temperatures (Figure 8). The average discharge air temperature decreased from 37.3°F to 32.1°F (2.9°C to 0°C) and the return air temperature decreased from 47.9°F to 43.0°F (8.8°C to 6°C) as the product temperature was lowered from 45°F to 41°F (7°C to 5°C).

One of the main impacts of operating under lower suction temperatures is on the compressor power. Lower suction temperatures needed to maintain the 41°F (5°C) product temperature caused the compressor to work harder to raise the system pressure 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 an 18.3% increase in the refrigerant mass flow rate (Figure 9). Figure 9 depicts the change in mass flow rate as a function of product temperature. The mass flow rate of the refrigerant increased from 10.5 lb/min to 12.5 lb/min (80 g/s to 90 g/s) as the product temperature was lowered from 45°F to 41°F (7°C to 5°C). The combined effects of added heat of compression and increased refrigerant mass flow rate resulted in an increase of 31.2% in compressor power consumption (Figure 10).

Figure 10 presents the change in compressor power and compressor power per ton as the average product temperature was decreased from 45°F to 41°F (7°C to 5°C). 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 (7°C to 5°C). This implies for every degree Fahrenheit (0.55°C) decrease in product temperature, the system's kW/ton rating and power use deteriorated by 3.8% and 7.8%, respectively.

Conclusion

The FDA's 1993 temperature recommendation increases the 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.

Considerable product temperature fluctuation may potentially take place between the front and back of the case shelves, and between the opposite ends of the shelves. Tracking these temperature variations was beyond the scope of this test and can be considered for future tests. Product temperatures should be lowered below the FDA's requirement, so that by the end of the defrost cycle, the product does not exceed the allowable temperature limits. Some under-cooling or over-cooling of the products may occur, causing adverse impacts on the quality, safety and shelf life of the products.

COLD STORAGE

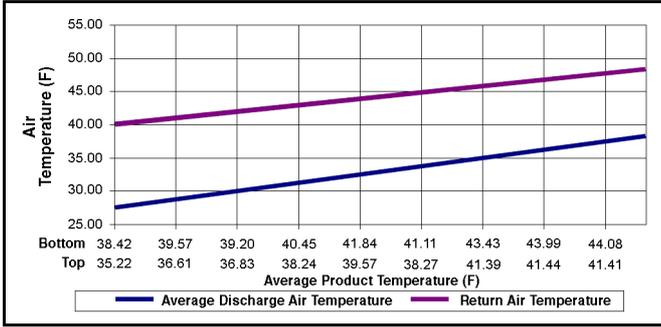


Fig. 8: Average discharge air and return air temperatures versus average product temperature.

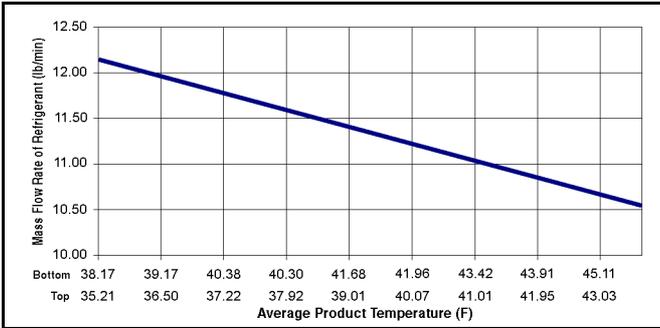


Fig. 9: Mass flow rate of refrigerant versus average product temperature.

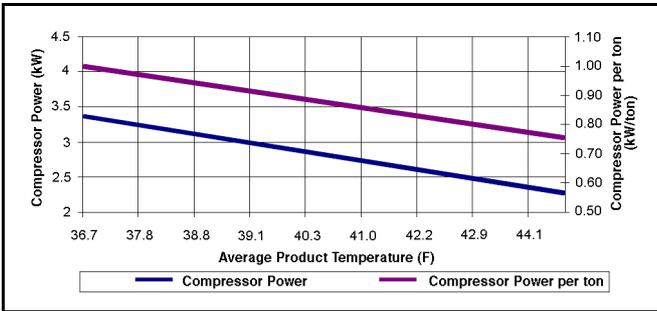


Fig. 10: Compressor power and compressor power per ton versus average product temperature.

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1. United States Public Health Service, Food and Drug Administration Food Code. 1995 Edition. U.S. Department of Commerce/Technology Administration. Springfield. 1995.

2. Ibid. ■

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