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Effects of Low-E Shields on the Performance and Power Use of a Refrigerated Display Case

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ABSTRACT

This paper discusses a test to evaluate the impact of low-emissivity aluminum shields on the power use and thermal performance of a multi-deck display case typically used in supermarkets for storing dairy products. The refrigerating system's critical temperature and pressure points were tracked during the test, and the readings were then utilized to quantify various heat transfer and power-related parameters of the refrigeration cycle. It was found that using shields for extended hours of operation provides the most reduction in refrigeration load and power use.

INTRODUCTION

The purpose of this test was to evaluate the impact of low-emissivity aluminum shields on the power use and thermal performance of a multi-deck display case typically used in supermarkets for storing dairy products. A California utility conducted this test at its Refrigeration Technology and Test Center (RTTC) located in Irwindale, California. The RTTC's instrumentation and data acquisition system provided detailed tracking of the refrigerating 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.

Thermal radiation and convection of warm air into the cold display case account for most of its refrigeration load. Low-emissivity shields can be utilized to cover the front opening of the display case and reduce the radiative and convective heat transfer into the case, thereby reducing power use while improving product temperature maintenance. Any reduction in refrigeration load could lower a supermarket's energy cost, improve its profit margin, and consequently, its competitiveness. As a result, testing and verifying the impact of these low-emissivity shields on the display cases could be beneficial to supermarket managers and operators.

The overall energy consumption of the refrigerated display cases could be reduced by enclosing the cases with reflective shields. This could be done during hours when the supermarkets are closed to the public. A large number of supermarkets in the utility's service territory are closed to the public for approximately six hours at night. Based on this, the test focused on three typical scenarios found in those supermarkets:

- Scenario 1 (base case, no shields)—Eighteen-hour operation with no shields utilized during closing hours (from midnight to 6:00 a.m.).
- Scenario 2 (shields applied)—Shields applied during closing hours (from midnight to 6:00 a.m.) and fully opened for the remaining 18 hours.
- Scenario 3 (holiday)—Shields applied for the full 24-hour period.

The three test scenarios were performed successively over a six-day period, beginning April 29, as shown in Table 1. The base case scenario was composed of data from May 3, the first full day after the shields were reopened. The holiday scenario utilized data from May 1, the first full 24-hour day with shields closed.

The selection of the data for scenario 2 was done to represent a typical 24-hour period for a supermarket that utilizes shields during its closing hours. Scenario 2 was developed based on data from the following test times: 6 hours of closed

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TABLE 1
Testing Conditions and Scenarios for the Shield Test

April 29	April 30	May 1	May 2	May 3	May 4
Shields open 100%. Established steady-state conditions.	Shields open until second defrost period at which time shields were closed 100%.	Shields closed 100%.	Shields closed until second defrost period at which time shields were opened 100%.	Shields open 100%.	Shields open 100%. Verification of steady-state conditions.

shield data followed by 18 hours of open display case data. Table 1 depicts the overall operation of the shields during the six days of the test.

The conditions within the test room were held constant at 75°F (23.9°C) and 50% relative humidity (RH). Throughout the test, the saturated condensing temperature was maintained at a fixed value of 90°F (32.2°C).

DISPLAY CASE HEAT TRANSFER MODES

Aside from internal heat-generating equipment, such as lights and fans, heat transfer components of the display case can be classified as infiltration (convection), transmission (conduction), and radiation. Heat transfer through the air curtains, also known as infiltration, functions as a convective load. The air curtain in open display cases acts as a primary barrier to reduce the infiltration load. The total performance of the air curtain and the quantity of heat transferred across it depends on several factors:

- Discharge air velocity and temperature
- Number of jets
- Air jet width
- Temperature and humidity ratio of the surroundings
- Rate of traffic adjacent to the air curtain
- Display case temperature and humidity ratio

An air curtain consists of a stream of air discharged from a series of small nozzles within a honeycombed configuration at the top of the display case. The air is discharged downward toward a return grille located approximately two feet above the floor on the front panel of the case. The air is drawn into a circulating fan where it picks up the fan motor heat and passes through the cooling coil (or the evaporator). By flowing across the evaporator, the air loses its sensible and latent heat. The chilled air is then supplied to the discharge grille. The discharged air travels downward between the still air in the store and in the case. The still air mixes with the discharged air, and this mixed stream develops new thermal characteristics. The mixing or entrainment of warm store air into the case takes place regardless of the store and display case temperatures. The temperature and moisture gradient between the cold display case and the warm surroundings within the mixing (or entrained) zone causes the sensible and latent heat from the warm side to transfer into the cold side.

Transmission load is a function of the case wall thermal conductivity, interior and exterior air film conductances, and the temperature difference between the interior and the exterior of the case. This component of the load is typically the smallest of all the load components.

The heat gain of the display case through radiation is a function of the inside conditions of the case, including wall temperature, wall emissivity, wall area, and view factor with respect to the surrounding (store) walls/objects, floor, and ceiling and their corresponding temperatures, emissivities, and areas.

SHIELD DESCRIPTION

Utilizing shields (Figure 1) is expected to reduce the radiation and infiltration loads of a display case. Additionally, it is expected to maintain lower product temperatures, thereby improving product shelf life.

The choice of aluminum as a shield material as opposed to other materials is due to its low emissivity. Materials with a low emissivity absorb very little radiated heat from the environment and reflect most of the heat back to their surroundings. The following equation (DeWitt and Incropera 1985), using a simplified case of radiation, expresses the relationship between the reflectivity and emissivity of an opaque surface:

$$\varepsilon = 1 - \rho \qquad (1)$$

where

ε = emissivity;

ρ = reflectivity.

In addition, all shields act as air infiltration barriers, reducing the convection of warm air into the display case. Consequently, these shields may reduce the sensible and latent loads of the evaporator. Shield material with high emissivities tends to absorb more radiant heat and eventually transfer that heat into the case via conduction. The reduced radiation into

Figure 1 *Display case with shields closed.*

the fixture, which was absorbed by the shield, is given by (DeWitt and Incropera 1985):

$$Q_{Rshield} = \sigma \cdot A \cdot \varepsilon \cdot F \cdot (T_{room}^4 - T_{shield}^4) \quad (2)$$

where

$Q_{Rshield}$ = radiation load on the shield, Btu/h (W);

σ = Stefan-Boltzmann constant, Btu/h·ft^2·°R^4 (W/m^2·K^4);

A = surface area of the shield, ft^2 (m^2);

ε = emissivity of the shield;

F = view factor between the shield and the room;

T_{room} = room temperature, °R (K);

T_{shield} = shield exterior surface temperature, °R (K).

This test utilized a woven aluminum fabric coated with a thin transparent film to eliminate oxidation and provide strength. A vertical rolling curtain arrangement, permanently attached to the top of the display case, allowed for easy storage of the shield when not in use (MGV 1997).

Differences in temperature and humidity between the inside of the display case and the ambient environment often cause condensation to form between the inner and outer fabric surfaces of the shields. Prevention of condensation on the aluminum fabric used in the test was provided via a precise pattern of tiny holes. The holes allowed the shields to breathe and condensed moisture to evaporate (MGV 1997).

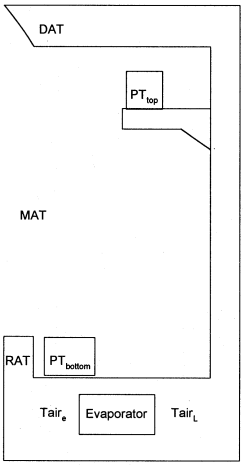
DATA COLLECTION/REDUCTION

The test facility is equipped with a data acquisition system that scans 94 sensors every 10 seconds and logs their outputs at two-minute intervals. Data were collected from each sensor and stored for six days. During each 24-hour test period, the data were downloaded and checked for consistency and accuracy. Operating parameters were checked to be within acceptable limits before the next run was started.

The collected data points for the two-minute intervals were averaged into one-hour blocks for each 24-hour period. After the hourly data were developed, the engineering calculations were performed.

RESULTS

The location of the products whose temperatures were monitored during the test is shown in Figure 2a. With an open display case, entrainment of the warm air of the test room at 75°F (23.9°C) into the cold case can increase the temperature of products near the return air grille, as well as the return air temperature. While utilizing the shields, it was observed that products located in the front on the bottom shelf were approximately 2°F (1.1°C) cooler than those in the front on the top shelf (Figure 2b). However, without the shields, the products located on the top shelf in the front were slightly colder than those in the front on the bottom shelf. Also, the product in the front on the bottom shelf was about 5°F (2.8°C) cooler with the shields down than it was with the shields up. During the periods when the shields are closed, the cold air within the case settles at the bottom of the case due to the density gradients formed by the different air temperatures. This could explain colder temperatures at the bottom shelves.



DAT	DISCHARGE AIR TEMPERATURE
MAT	MID AIR-CURTAIN TEMPERATURE
RAT	RETURN AIR TEMPERATURE
Tair$_e$	AIR TEMPERATURE AT EVAPORATOR INLET
Tair$_L$	AIR TEMPERATURE AT EVAPORATOR EXIT
PT$_{top}$	PRODUCT TEMPERATURE AT TOP SHELF
PT$_{bottom}$	PRODUCT TEMPERATURE AT BOTTOM SHELF

Figure 2a *Location of products with temperature sensors within the display case.*

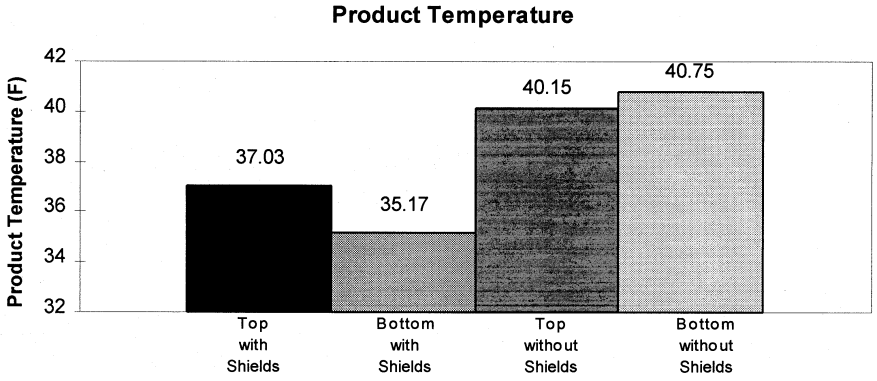


Figure 2b *Product temperature variation.*

During the 18-hour operation of the supermarket with the "shields applied" and holiday scenarios, the shield assembly caused the refrigeration load of the case to decrease by 8.5% and 42.4%, respectively (Figures 3a through 3c). Also, the compressor power decreased by 0.3 kW and 1.1 kW (0.4 hp and 1.5 hp), respectively, for the 18-hour period with the same scenarios. Clearly, utilizing shields for extended hours of operation provides the most reduction in refrigeration load and power use.

The compressor kW decreased as a result of the decreased case load. Therefore, the speed of the compressor decreased also, as it is controlled by a variable-frequency drive. Less refrigerant mass flow was required to satisfy the target saturated suction pressure and discharge air temperature control setting.

The ability of the display case to hold lower product temperature increased with the shields down. The difference in the average daily product temperatures between shields up and shields down was 3°F (1.7°C) for the upper and 5°F (2.8°C) for the lower products. The average discharge air temperature drop from the base case scenario was 1.2°F and 3.4°F (–17.1°C and –15.9°C) for the 18-hour supermarket operation with shields and holiday scenarios, respectively. The effect of the evaporator pressure regulator valve in maintaining a minimum saturated suction temperature caused the evaporator temperature to remain almost constant between approximately 22°F and 24°F (–5.6°C and –4.4°C) throughout the three test scenarios.

The product temperature reduction due to the shields was largest in the holiday scenario and had only a modest impact in the 18-hour supermarket operation with shields case. The product temperatures in the base case and the holiday scenario consistently differed by approximately 4°F (2.2°C) throughout the 24-hour period (Figure 4). On the other hand, the product temperature variation between the base case and 18-hour supermarket operation with shields scenario started at approximately 4°F (2.2°C), the same as the holiday case, but after the shields were opened, the difference lessened until their

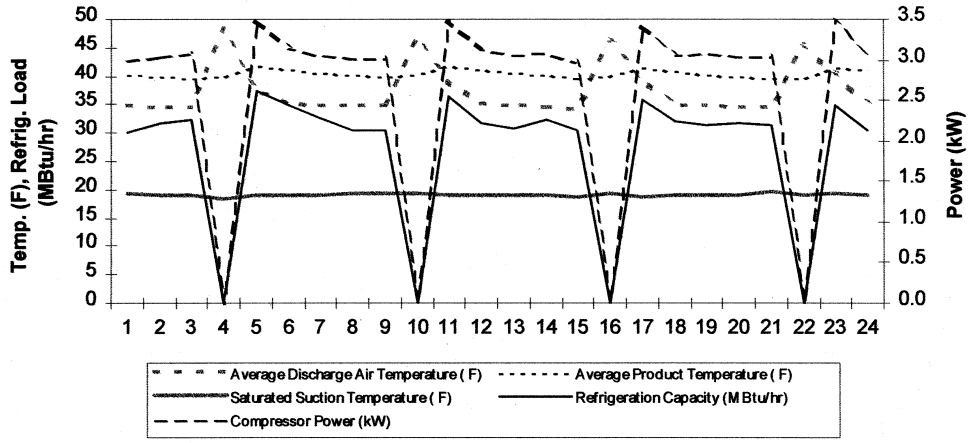


Figure 3a *Key parameters over time for base case (no shields).*

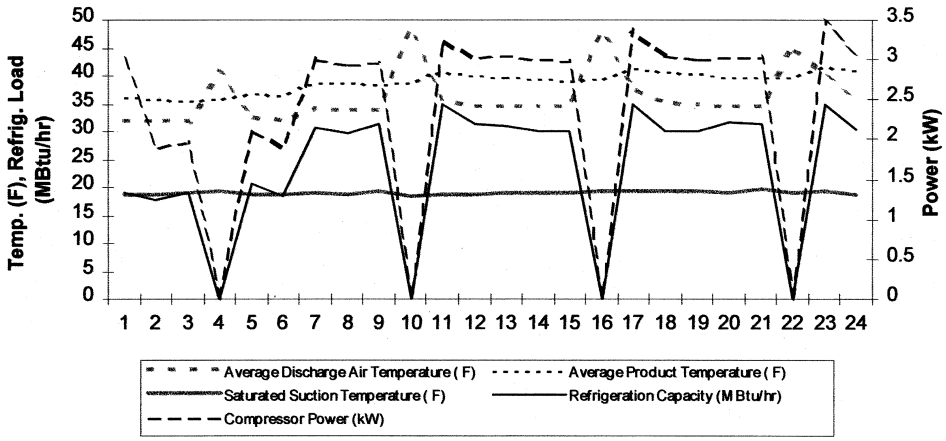


Figure 3b *Key parameters over time for 18-hour operation of the supermarket with "shields applied" scenario.*

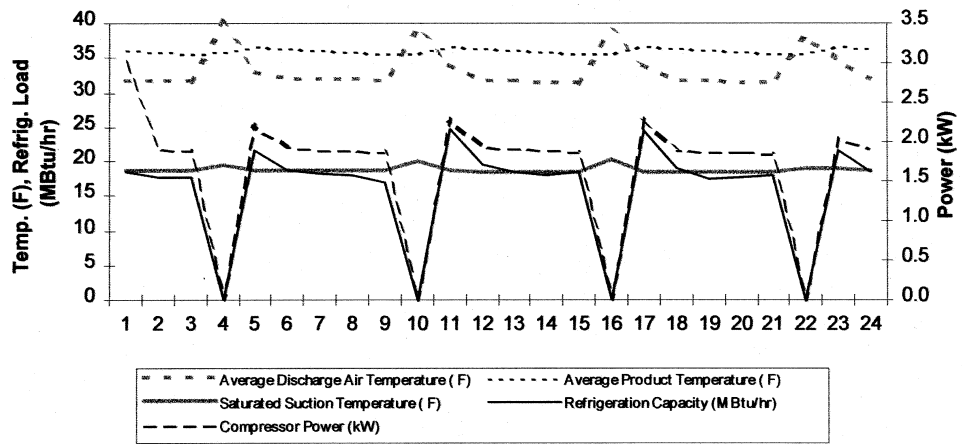


Figure 3c *Key parameters over time for holiday scenario.*

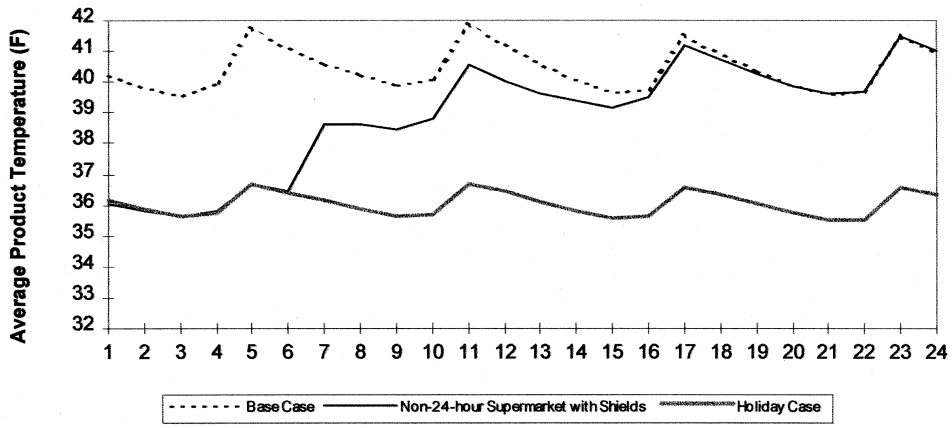


Figure 4 *Average product temperature comparison between the base case, 18-hour supermarket operation with shields case, and the holiday scenario.*

temperatures became equal (at approximately 9 p.m., see Figure 4). Utilizing the shields for six hours allowed the product temperature to remain below that of the base case for 15 hours after the shields were removed. This could result in less overall compressor energy required to maintain lower product temperatures.

Figures 5a and 5b present the transitional effects of shields on the key refrigeration parameters for two sample test days. On the first day (Figure 5a), the shields were open initially until the second defrost period (approximately 9 a.m.), then closed completely. Figure 5a shows a decrease in all product temperatures after the shields were closed. The decrease in product temperature at the bottom was greater than the decrease in product temperature at the top. This created a large difference between the two product temperatures. Conversely, the gap between the midair curtain temperature (MAT) and the return air temperature (RAT) decreased after the shields were closed.

On the second sample day (Figure 5b), the scenario was reversed. The shields were initially closed and were then opened during the second defrost period. As such, the trends in Figure 5a were reversed in Figure 5b.

The data points lying above the trend for refrigeration effect and refrigeration load in Figure 6 represent the data for the first hour after termination of the defrost cycle. During defrost, the compressor did not operate; therefore, the refrigeration load and product temperatures increased, causing an increase in mass flow rate during the first hour after defrost.

With the exception of some discontinuity during closing of the shields, the refrigeration effect remained fairly constant throughout the test. This relatively nonfluctuating trend was the result of the constant saturated condensing temperature and suction pressure setting. The refrigeration load of the case, on the other hand, fluctuated as the shields were opened and closed. The shields were closed during the second defrost period on April 30. Figure 6 shows a large drop in the refrig-

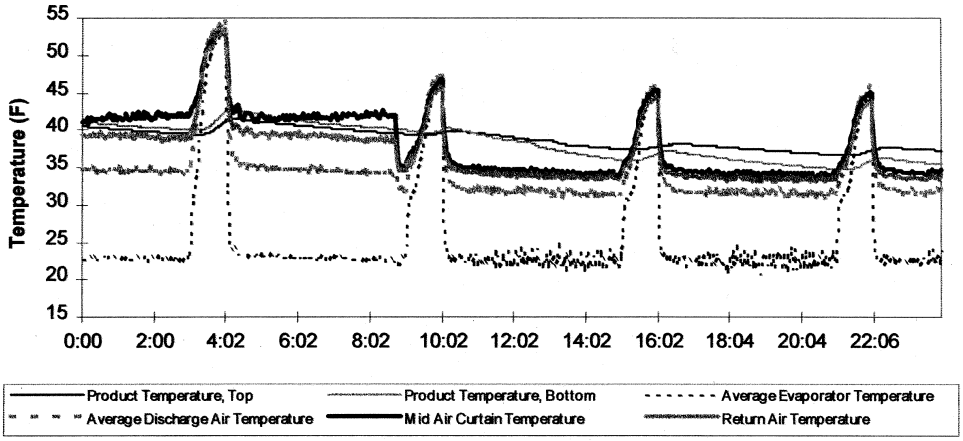


Figure 5a *Key parameter profile for a sample 24-hour period starting with shields open, then closing them during the second defrost period.*

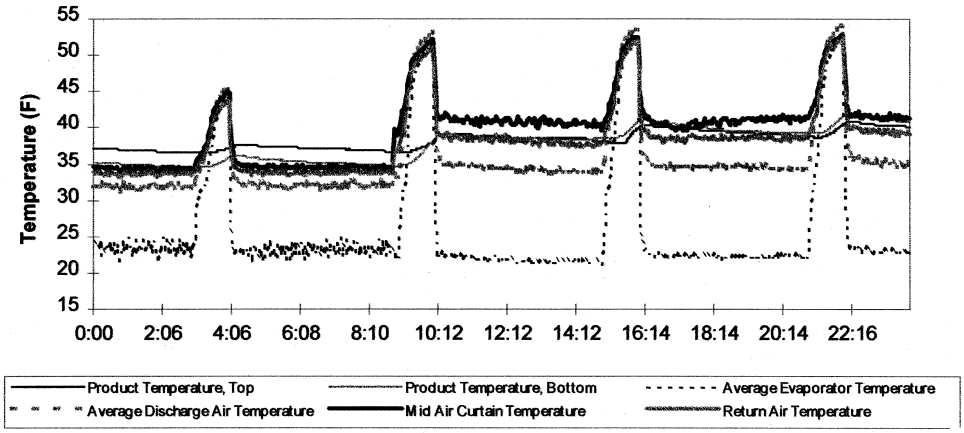


Figure 5b *Key parameter profile for a sample 24-hour period starting with shields closed, then opening them during the second defrost period.*

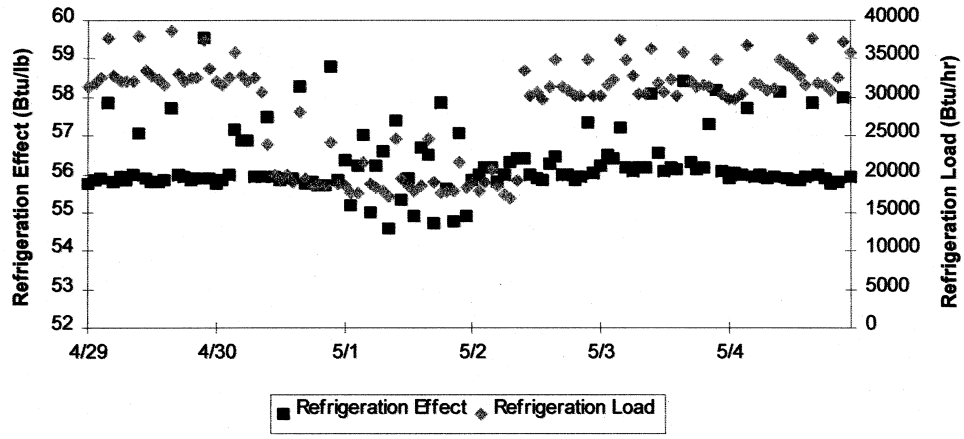


Figure 6 *Refrigeration effect and refrigeration load vs. time (no defrost data included).*

eration load at the same time. Likewise, the refrigeration load increased to its original value as the shields were reopened on May 2.

Another observation from the test was the effect of the shields on the time necessary to defrost the ice on the evaporator coils. Figures 7a and 7b show the time needed to melt the coil ice with and without the shields. The medium-temperature display case used in this test utilized an off-cycle defrost to remove ice buildup from the case coil. During off-cycle defrost, the refrigeration to the case was shut down while the case fans continued to run using the ambient air to melt any ice that had built up on the coil.

During defrost, the coil temperature rose from approximately 22.5°F to 32°F (−5.3°C to 0°C) at which time ice started melting. Figures 7a and 7b each point to a region during the middle of the defrost period named the "ice melting stage." The evaporator temperature stayed at 32°F (0°C) throughout the phase change of ice from solid to liquid state. This ice melting stage was approximately twice as long with the shields closed than with the shields open (12 minutes vs. 6 minutes). With the shields closed, less warm ambient air from the room was available to aid melting of the ice on the evaporator coils. The average discharge air temperature was approximately 2°F (1.1°C) cooler with the shields closed than open. The cooler discharge air temperature and the lack of sufficient defrost heat (infiltrating from the room to the case) explains the longer melting time.

An important consideration when utilizing shields is the effect on evaporator and total system superheat. The lack of sufficient evaporator superheat may result in a flooded evaporator that can potentially feed liquid refrigerant into the compressor. Figure 8 shows the amount of evaporator superheat achieved for two extreme test scenarios, and it is clear that the use of shields reduces the evaporator superheat. Table 2 summarizes the average refrigerant superheat achieved at the evaporator outlet and compressor inlet for three scenarios.

Table 3 summarizes the effects of utilizing heat-reflecting shields on the key refrigerating system parameters. Under scenario 2 (18-hour supermarket operation with shields), the mass flow rate was lowered by 12% and the heat rejection to the condenser was reduced by 12%. For the holiday case, however, the refrigeration load dropped significantly (41%), resulting in an increase of kW/ton (9.0%).

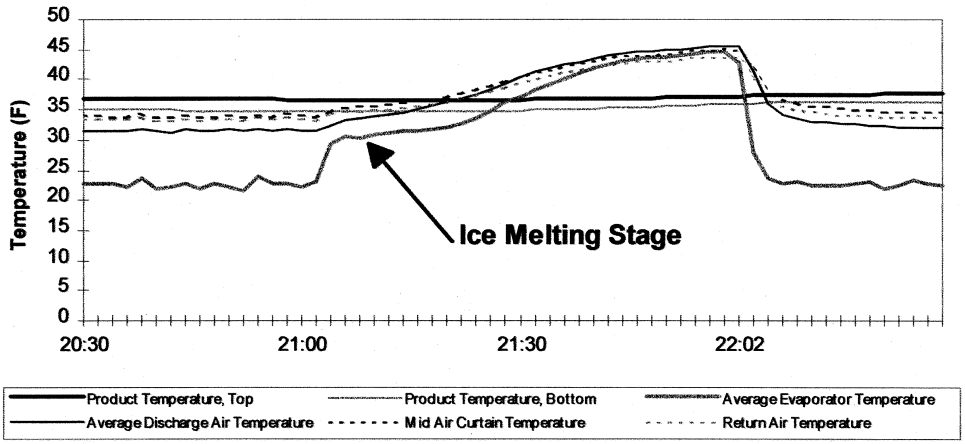


Figure 7a *Last defrost period of the day after shields were closed.*

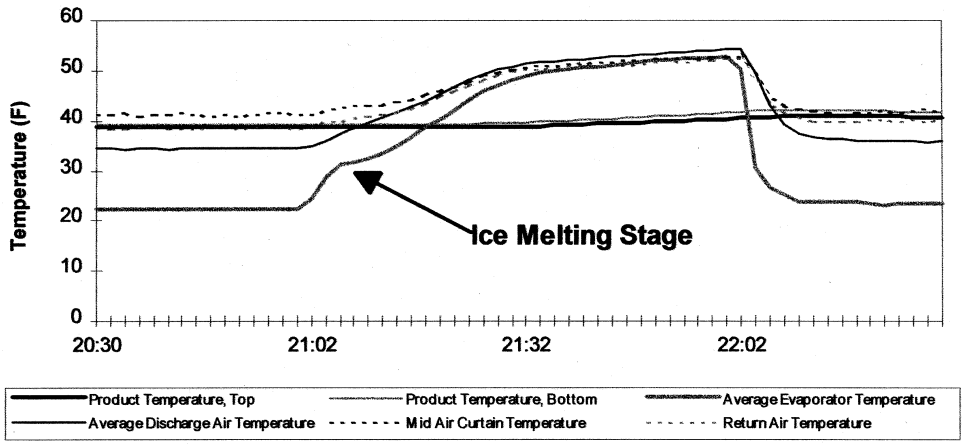


Figure 7b *Last defrost period of the day after shields were opened.*

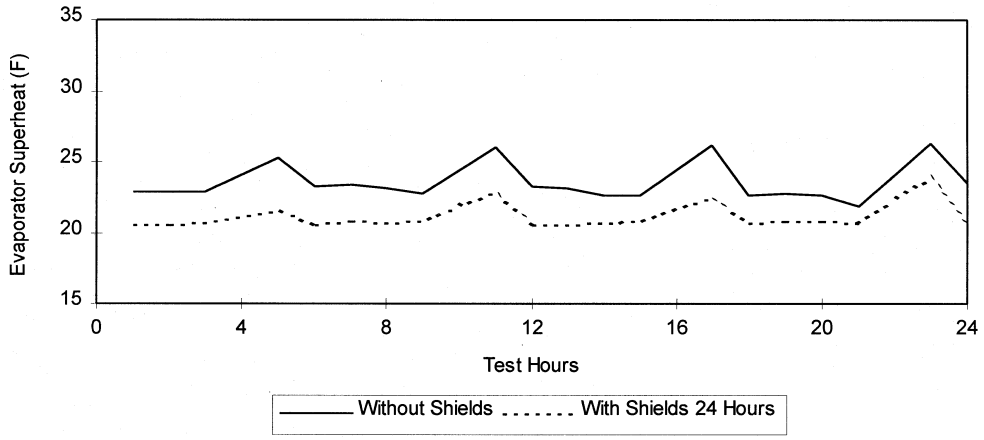


Figure 8 *Evaporator superheats for two scenarios.*

TABLE 2
Effects of Shields on Evaporator and Total Superheat

	Base Case (Scenario 1)	18-Hour Supermarket with Shields (Scenario 2)	Holiday Case (Scenario 3)
Average Total Superheat, °F	32.11	32.68	35.70
Average Evaporator Superheat, °F	24.36	23.66	22.25

TABLE 3
Effects of Utilizing Heat-Reflecting Shields on Refrigerating System Parameters: 18-Hour Supermarket with Shields and Holiday Case vs. Base Case

Scenario	Discharge Air Temp. (°F)	Average Saturated Suction Temp. (°F)	Average Product Temp. (°F)	Mass Flow Rate of Refrigerant (lb/min.)	Refrigeration Load of the Case* (MBtu/h·ft)	Condenser Heat Rejection* (MBtu/h)	Compressor Power (kW)	Rating* (kW/ton)
(1) Base Case	36	19.0	40.6	9.5	1.6	41.9	3.1	1.16
(2) 18-Hour Supermarket with Shields	34	19.1	38.9	8.4	1.4	36.9	2.9	1.23
%Δ	3.3%	0.0%	4.0%	12.1%	12.6%	11.9%	9.0%	–5.6%
(3) Holiday Case	32	18.6	36.1	5.7	1.0	25.7	2.0	1.27
%Δ	9.7%	2.4%	11.0%	40.3%	41.0%	38.7%	36.0%	–9.0%

* Calculated values.

The increase in kW/ton with longer shield applications can be attributed to inefficient operation of the compressor and the variable-speed drive at lower loads. Under reduced or partial refrigeration load, the variable-speed drive controller works harder and less efficiently to provide a matched compressor capacity. As a result, compressor kW dropped by 9% and 36% (for scenarios 2 and 3, respectively). The overall efficiency of the compressor, however, is reduced because it is operating under partial load and is, therefore, operating at a point further away from the design point of maximum efficiency.

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