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Performance and Energy Impact of Installing Glass Doors on an Open Vertical Deli/Dairy Display Case

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ABSTRACT

In a typical supermarket, open, multi-deck, medium-temperature display cases could account for up to 50% of the total refrigerated display case line-ups (EPRI 1992). The major contributor to the total cooling load of this type of fixture is infiltration, which comprises approximately 70% to 80% of the total cooling load (Faramarzi 1999). The infiltration load of the display case refers to the entrainment of warm and moist air from the room, across the case air curtain, into the refrigerated space. This makes open multi-deck display cases vulnerable to indoor dry-bulb (DB) temperature and relative humidity (RH) variations. Installing glass doors on open vertical display cases can reduce the infiltration load, hence, the power consumption of the refrigeration system.

This paper presents the laboratory test results, which evaluated the performance and energy impact of installing conventional glass doors on an open five-deck refrigerated dairy/deli display case. Retrofitting the fixture with glass doors reduced the entrainment of warm and moist air from the room into the refrigerated space. This reduction caused the total cooling load of the case to decrease by 68%. Installing glass doors on the display case reduced the refrigerant mass flow rate by 71%, resulting in the reduction of compressor power demand by 87%.

INTRODUCTION

A Southern California utility conducted this test at its Refrigeration and Thermal Test Center (RTTC), located in Irwindale, California. The controlled environment chamber was maintained at a constant DB temperature of 75°F and a constant RH of 55% for all tests. The refrigeration system was charged with a hydrofluorocarbon (HFC) refrigerant (R-404A).

Throughout the test, the refrigeration systems' controller maintained a fixed saturated condensing temperature (SCT) of 95°F (±0.5°F). The test rack controller was programmed to run at the manufacturer's specified suction pressure of 59 psig, corresponding to saturated evaporator temperature of 23°F.

The performance of an open five-deck dairy/deli fixture, commonly found in supermarkets, was evaluated under fixed conditions to develop the baseline characteristics. The display case was then retrofitted with glass doors. Under the same indoor and operational conditions as the baseline, the performance of the retrofitted case was evaluated. The project then closely compared the key performance attributes of the fixture under both scenarios. The following table summarizes the two test scenarios:

| Test Scenarios | Test Description |
|-------------------------------------|--|
| Scenario 1 (base case) | The open display case operated at manufacturer's specification. |
| Scenario 2 (glass door retrofit) | The display case was retrofitted with three glass doors and anti-sweat heating system, which stayed on during entire test period. The case was operated according to manufacturer's specification. |

Glass doors used in this test were equipped with anti-sweat heaters (ASH) to prevent glass from possible fogging and the cold doorframe from forming condensation. A summary specification of the tested fixture and the glass doors are given in Tables 1 and 2. Figures 1a and 1b show images of the five-deck dairy/deli display case before and after retrofit.

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TABLE 1
Specification Summary of Tested Fixture

| Display Case Type | Application | Evaporator Temperature (°F) | Capacity (Btu/h-ft) | Length (ft) | Defrost |
|---------------------------|---------------------------|-----------------------------|---------------------|-------------|-----------------------|
| Open 5-Deck Front Loading | Deli/Dairy (medium temp.) | +23 | 1,492 | 8 | Off-Cycle (4 per day) |

TABLE 2
Specification Summary of Glass Doors

| Number of Doors | Door Width (in.) | Door Height (in.) | Door Panel | Glass Type | Emissivity, Outer Panel / Inner Panel | Glass Width / Height (in.) | Total Ohms Door / Frame, (nominal) | Total Amps, Frame / Heater (nominal) |
|-----------------|------------------|-------------------|------------|-----------------|---------------------------------------|----------------------------|------------------------------------|--------------------------------------|
| 3 | 31 1/16 | 53 5/16 | All Glass | 2PNHG- 60% REFL | 0.84 / 0.20 | 29 5/16 / 51 9/16 | 147.6 / 96.0 | 1.26 / 2.06 |

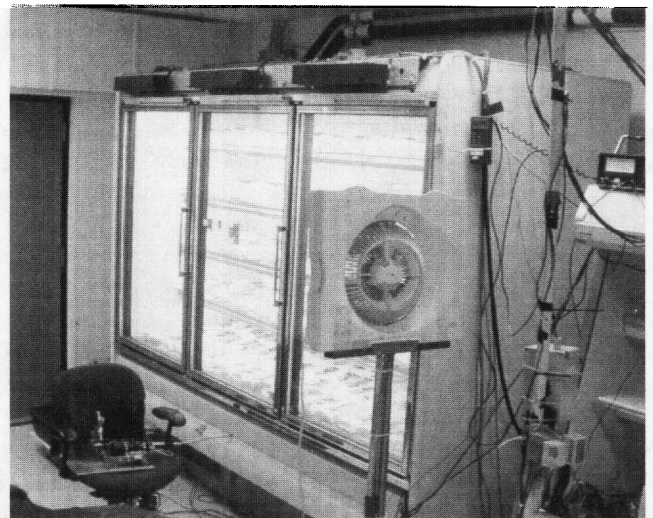
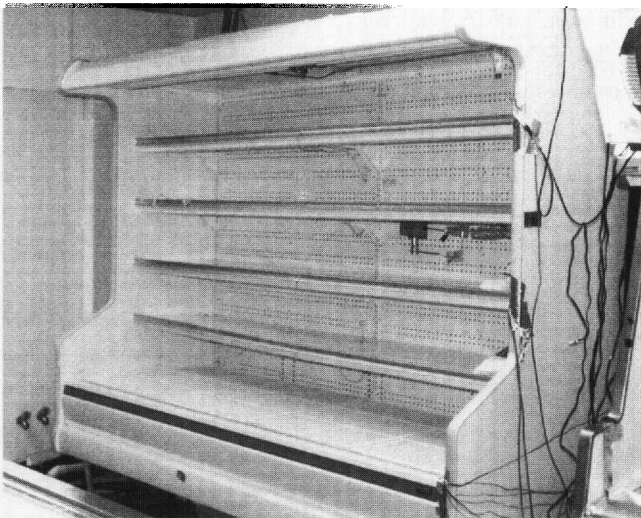


Figure 1 Photograph of the five-deck dairy/deli display case: (a) before retrofit and (b) after retrofit.

The paper specifically investigates the impact of retrofitting the fixture with glass doors on the following parameters:

- power consumption of main components
- mass of collected condensate
- product temperature at various locations
- display case cooling load and its components
- defrost duration

Test Design and Procedure

The test fundamentals followed *ANSI/ASHRAE Standard 72-1983* (ASHRAE 1983). Accordingly, both test scenarios were conducted under identical indoor conditions of 75°F DB and 55% RH over a 24-hour period. The discharge pressure was maintained at 220 psig, corresponding to SCT of 95°F, and the suction pressure was set at manufacturer's recommended pressure of 59 psig, corresponding to evaporator temperature of 23°F, for both test scenarios.

Under *ANSI/ASHRAE Standard 72-1983*, the "food product zones shall be filled with test packages and dummy prod-

ucts" to simulate the thermal mass of food in the case. According to section 7.2.1 of this standard, food products are composed of 80% to 90% water, fibrous materials, and salt. Accordingly, plastic containers were completely filled with sponge materials that were soaked in brine solution of salt and water to simulate the product mass during the tests. These test packages, or product simulators, were placed in locations where product temperature maintenance was most critical. The spaces in the test fixture where temperature measurement was not required were stocked with dummy products to add thermal mass and stabilize the temperature in the case. The product temperatures were monitored at the top shelf front right corner, top shelf rear center, bottom shelf front left corner, and bottom shelf front center.

To simulate the effects of shopper traffic, an automatic door opener on each of the three glass doors and a low-speed propeller type fan were utilized (Figure 1b). Each glass door was opened for 16 seconds every 12 minutes during the 24-hour period. In the base case test scenario, the fan, which was four feet away from the fixture, stayed on for one minute and

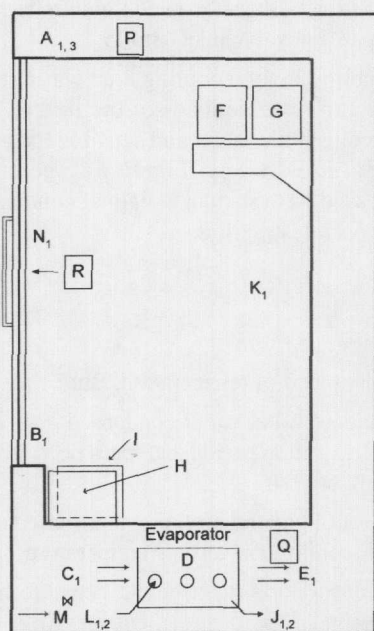


Figure 2 Location of sensors for dairy/deli display case.

| MAIN LEGEND | |
|------------------|--|
| A | DISPLAY CASE DISCHARGE AIR TEMPERATURE |
| B | DISPLAY CASE AIR TEMPERATURE AT RETURN GRILL |
| C | AIR TEMPERATURE ENTERING EVAPORATOR |
| D | COIL SURFACE TEMPERATURE (3 LOCATIONS) |
| E | AIR TEMPERATURE LEAVING EVAPORATOR |
| F | PRODUCT TEMPERATURE, FRONT RIGHT, UPPER SHELF |
| G | PRODUCT TEMPERATURE, REAR CENTER, UPPER SHELF |
| H | PRODUCT TEMPERATURE, FRONT CENTER, LOWER SHELF |
| I | PRODUCT TEMPERATURE, FRONT LEFT, LOWER SHELF |
| J | REFRIGERANT TEMPERATURE & PRESSURE AT COIL EXIT |
| K | AIR TEMPERATURE INSIDE CASE |
| L | REFRIGERANT TEMPERATURE & PRESSURE AT EXPANSION VALVE EXIT |
| M | REFRIGERANT TEMPERATURE AT EXPANSION VALVE INLET |
| N | INSIDE GLASS TEMPERATURE |
| P | CASE LIGHTS - POWER |
| Q | CASE FAN MOTORS - POWER |
| R | ANTI-SWEAT HEATERS - POWER |
| SUBSCRIPT LEGEND | |
| 1 | DRY BULB TEMPERATURE |
| 2 | PRESSURE |
| 3 | VELOCITY |

turned off for one minute and 50 seconds. However, it operated continuously in the post-retrofit test scenario.

Prior to the test, all temperature and pressure instruments were calibrated. Careful attention was paid to the design of the monitoring system to minimize instrument error and maintain a high level of repeatability and accuracy in the data. Figure 2 details the location of sensors within the test fixture.

A data scanner was used to log the test data. The data scanner was set up to process 128 data channels in ten-second intervals. The scanner was calibrated at the factory and is traceable to the National Institute of Standards and Technology's (NIST's) standards. Every ten seconds, the data acquisition system sampled the scanned data and created a time-stamped two-minute average. Initial data were reviewed on site to ensure that the key control parameters were within acceptable ranges. Once the data passed the initial screening process, they were downloaded remotely for further screening and processing.

The display case was tested in the controlled environment room of the RTTC. The controlled environment room is an isolated thermal zone served by independent cooling, heating, and humidification systems. This allows simulation of various indoor conditions of a supermarket. 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. Auxiliary electric heaters, located in the downstream of the heat reclaim coils, provide additional heating when required. While the air is conditioned to a desired thermostatic setpoint, an advanced ultrasonic humidification unit introduces precise amounts of moisture to the air surrounding the display cases, representing the latent load due to outside air and people.

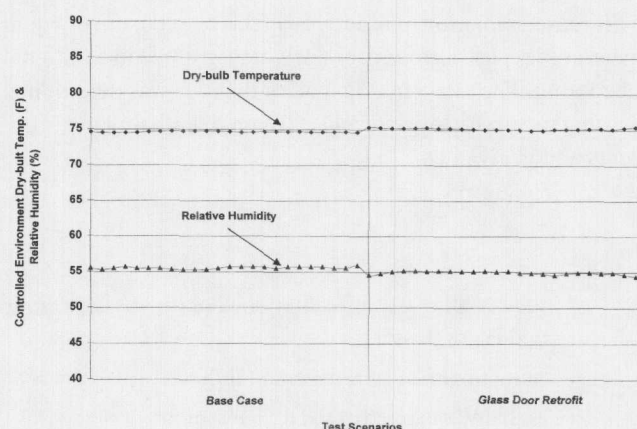


Figure 3 Hourly profile of the controlled environment room dry-bulb temperature and relative humidity for both scenarios.

Test Critical Control Points

Figure 3 illustrates the hourly profile of controlled environment DB temperature and RH for both test scenarios. The controlled environment DB temperature stayed constant at 75°F, and the RH stayed unchanged at 55% throughout the entire test periods.

Figure 4 depicts the two-minute profiles of discharge and suction pressure for both test scenarios. In this test, the discharge and suction target pressures were maintained at 220 and 59 psig, respectively. The pressure fluctuations and drift from target setpoints slightly increased when doors were installed. The increase in pressure fluctuations was primarily

due to the hunting effect of the now oversized thermostatic expansion valve (TXV).

DISCUSSION OF RESULTS

Product Temperature Distribution

Reviewing the two-minute data indicates product temperatures in the open case stayed higher than in the retrofit case scenario (Figure 5). The highest product temperature in the base case test scenario was roughly 46°F, which was detected at the bottom shelf front center location. The highest post-retrofit product temperature was 42°F, which was observed at the top shelf front right location, while the top shelf rear center had the lowest temperature for both scenarios.

Cooling Load

Figure 6 compares the total cooling load of the fixture before and after installing glass doors. The total measured cooling load of the display case prior to installing the glass doors was 14,230 Btu/h. Retrofitting the display case with glass doors primarily decreased the infiltration and resulted in reducing the cooling load by 68%. This reduction was also due to decreased radiation load and partly due to exposure of liquid line in colder post-retrofit cabinet, which resulted in additional subcooling. The total cooling load of the case was determined by using the mass flow rate and refrigerant enthalpy measurements and is given by

$$Q_{total} = m_{ref} \times (h_{out} - h_{in})$$

where

Q_{total} = total cooling (refrigeration) load of the case, sensible and latent, Btu/h

m_{ref} = mass flow rate of refrigerant, lb/h

h_{out} = enthalpy of superheated refrigerant vapor at the evaporator outlet, Btu/lb

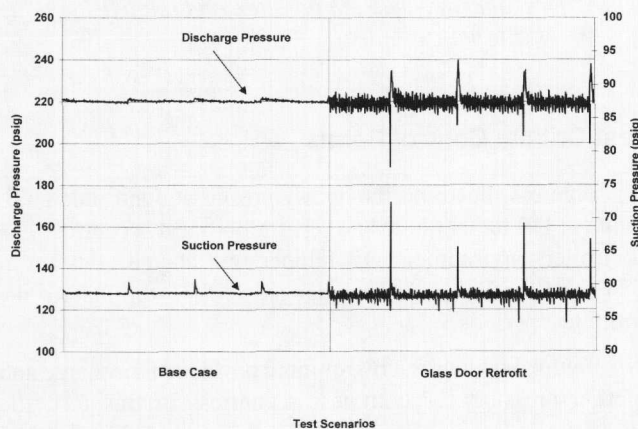


Figure 4 Two-minute profile of discharge and suction pressure for both scenarios.

h_{in} = enthalpy of the subcooled liquid refrigerant at the expansion valve inlet, Btu/lb

The reduction in total cooling load was a result of reduction in latent and sensible loads of the fixture. Installing the glass doors reduced the latent and sensible load of the case by 90% and 59%, respectively (Figure 6). The latent load was determined based on the actual weight of condensate collected at the end of each test period.

$$Q_{latent} = m_c \times h_{fg}$$

where

Q_{latent} = latent load of refrigeration, Btu/h

m_c = mass of water vapor condensed from air during refrigeration period and mass of melted frost during defrost, lb/h

h_{fg} = heat of vaporization of water (based on average evaporator coil surface temperature), Btu/lb

Once latent load is determined, sensible heat ($Q_{sensible}$) can be obtained:

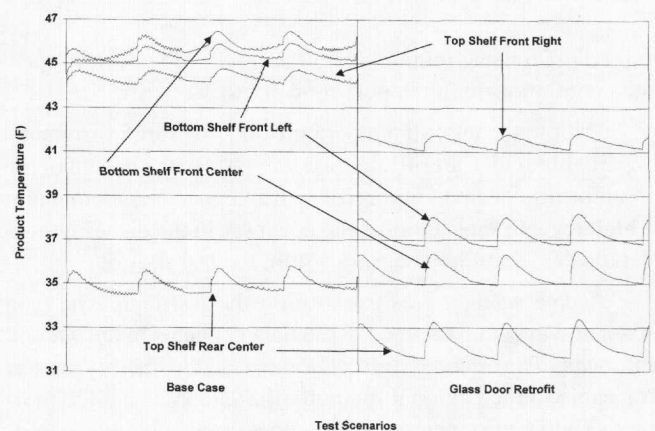


Figure 5 Two-minute profile of product temperatures for both scenarios.

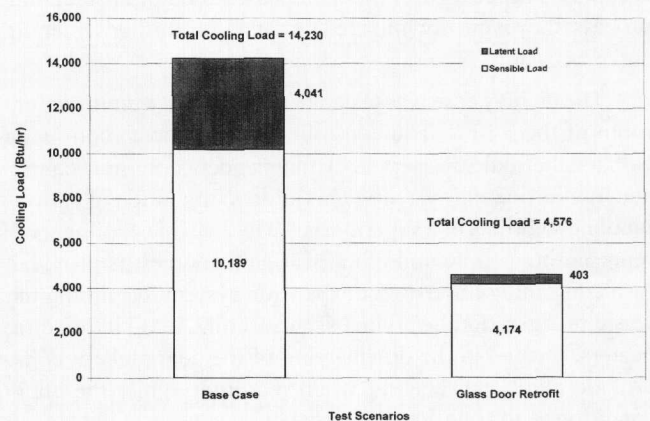


Figure 6 Comparison of latent and sensible component of the cooling load for both scenarios.

$$Q_{sensible} = Q_{total} - Q_{latent}$$

The constituents of total cooling load are incoming heat from the surrounding environment as well as internal sources. The incoming heat from the surrounding environment includes transmission (or conduction), infiltration, and radiation. The heat from internal sources includes lighting and evaporator fan motor(s) (Faramarzi 1999).

The first task in determining the conduction (transmission) load was to determine the overall coefficient of heat transfer of the case walls. This involved determination of all outside and inside air film convective coefficients, thermal conductivity of the outer and inner walls of the case, and thermal conductivity of the insulation between the inner and outer walls. Once the overall coefficient of heat transfer is determined, the conduction load can be given as (Faramarzi 1999):

$$Q_{cond} = U \times A \times (T_{room} - T_{case})$$

- Q_{cond} = transmission, or conduction, load of the case, Btu/h
 U = overall coefficient of heat transfer coefficient for the case walls, Btu/h·ft²·°F
 A = total surface area of case walls that are conducting heat, ft²
 T_{room} = dry-bulb temperature of the air in the room, °F
 T_{case} = temperature of the interior panel of the display case, °F

The case load due to radiant heat transfer for an open display case was determined by simply modeling the system as two gray surfaces—one surface representing the total surface area of the room (walls, floor, ceiling) and the other being an imaginary plane covering the opening of the display case. The imaginary plane at the case opening will exchange all of its radiation with the interior surfaces of the display case (Faramarzi 1999). For two surfaces, which exchange heat with each other and nothing else, Holman (1990) suggests that net radiant heat exchange can be given by:

$$Q_{rad} = \sigma (T_w^4 - T_c^4) / [1/(1 - \epsilon_w)/\epsilon_w A_w + 1/A_w F_{c-w} + (1 - \epsilon_c)/\epsilon_c A_c]$$

where

- σ = Stefan-Boltzmann constant, (0.1714 * 10⁻⁸ Btu/h·ft²·°R⁴)
 T_w = surface temperature of the room walls, °R
 T_c = surface temperature of the display case inner walls, °R
 ϵ_w = emissivity of the room walls
 A_w = adjusted area of room surfaces, ft²
 F_{c-w} = view factor from case to surfaces of the room
 ϵ_c = emissivity of the inside walls of the case
 A_c = total area of the inside walls of the case, ft²

The internal load for the display case was determined based on the power consumed by lighting and evaporator fan(s). The power consumed by these devices was recorded

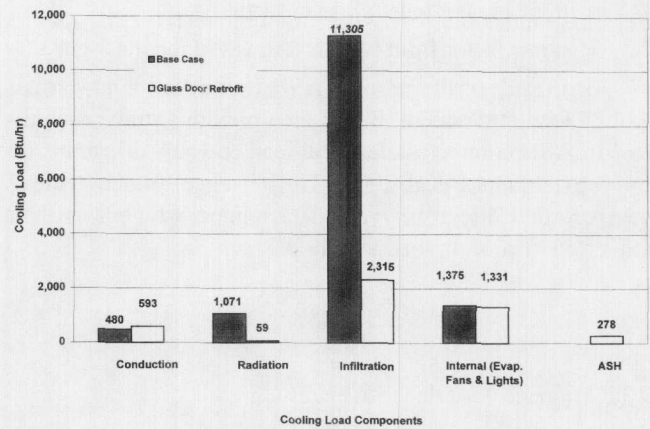


Figure 7 Comparison of cooling load components for both scenarios.

directly by the data logger, which was then converted to cooling load as follows (Faramarzi 1999):

$$Q_{evap-fans} = kW_{evap-fans} \times k$$

$$Q_{lights} = kW_{lights} \times k$$

where

- $Q_{evap-fans}$ = case load due to fan motors, Btu/h
 Q_{lights} = case load due to lighting, Btu/h
 $kW_{evap-fans}$ = power consumed by the fan motors, kW
 kW_{lights} = power consumed by the light fixtures in the case, kW
 k = conversion factor, (3413 Btu/h)/kW

Once the total case load along with all other components of the case load were determined, the infiltration load (Q_{inf} in Btu/h) can be obtained by (Faramarzi 1999):

$$Q_{inf} = Q_{total} - [Q_{evap-fans} + Q_{lights} + Q_{cond} + Q_{rad}]$$

Reviewing the cooling load components revealed that installing glass doors reduced the infiltration by roughly 80% (Figure 7). The cooling load of internal components (evaporator fans and lights) and conduction remained almost unchanged. The slight increase in conduction load was due to larger post-retrofit surface area since larger panels were used to fit the glass doors as well as colder post-retrofit cabinet temperature. The net radiation, however, was reduced by 94%, due to the effects of glass door on radiation heat. The effects of adding glass doors on radiation was calculated using:

$$Q_{rad} = \sigma (T_{ig}^4 - T_c^4) / [1/(1 - \epsilon_{ig})/\epsilon_{ig} A_{ig} + 1/A_{ig} F_{c-ig} + (1 - \epsilon_c)/\epsilon_c A_c]$$

where

- T_{ig} = surface temperature of the inner glass, °R
 ϵ_{ig} = emissivity of the inner glass

A_{ig} = total area of inner glass surfaces, ft^2

F_{c-ig} = view factor from case to surfaces of the inner glass

As a result of the retrofit, ASH introduced a new small cooling load component. Based on a previous study, roughly 35% of ASH connected electrical load ends up as heat inside the case (Faramarzi et al. 2001). The power consumed by ASH was recorded directly by the data logger, which was then converted to cooling load as follows:

$$Q_{ASH} = kW_{ASH} \times k \times k_F$$

where

Q_{ASH} = case load due to ASH, Btu/h

kW_{ASH} = power consumed by the ASH, kW

k_F = fraction of heat dissipated into the case, 35%

The cooling load analysis illustrated that infiltration makes up roughly 79% of the cooling load of this particular display case (Figure 8). The contribution of infiltration was reduced to only 50% when the glass doors were installed. The load due to ASH heat dissipation constituted approximately 6% of the total cooling load.

Defrost and Condensate Mass

The test display case was set up to have four off-cycle defrosts per day. Off-cycle defrost relies on the enthalpy of the ambient air, which is entrained into the case to melt the frost. Installing glass doors on the fixture reduced the infiltration of warm and moist air from the room into the refrigerated space. This reduction resulted in an increase of 48% in duration of the off-cycle defrost.

Reduction in air infiltration from the room into the refrigerated space was also evident in the mass of collected condensate (Figure 9). The total mass of condensate collected in 24 hours was roughly 80 pounds in base case test scenario. The glass doors caused a decrease in the frost mass on the coil and, consequently, the condensate weight was reduced by 88%.

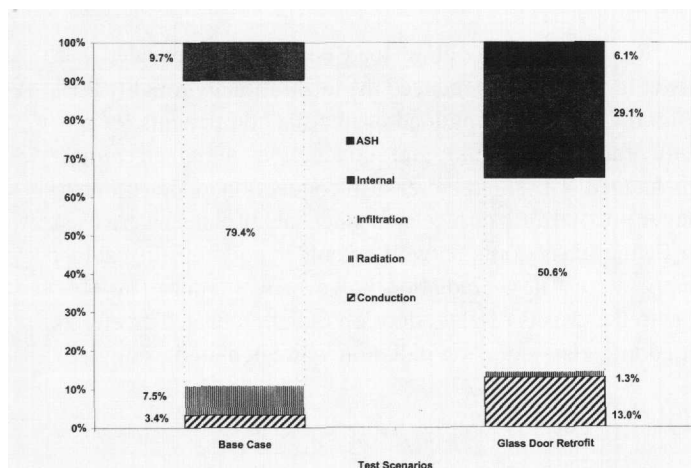


Figure 8 Comparison of refrigeration load percentage breakdown for both scenarios.

The horizontal lines in Figure 9 indicate the collected moisture during the refrigeration period. The vertical lines depict the mass of melted frost during each of the four defrost cycles.

Mass Flow Rate of Refrigerant

Figure 10 depicts a two-minute profile of the refrigerant mass flow rate for both test scenarios. For the base case scenario, the refrigerant mass flow rate reached its highest value when defrost was terminated. This increase was primarily due to post-defrost pull down load. The lowest flow rates were observed prior to initiation of defrosts. On the contrary, a fluctuating characteristic in refrigerant mass flow rate was observed in the post-retrofit scenario. This varying profile can be attributed to the reduction in cooling load of the case, which can cause the excessive hunting of the TXV in order to satisfy the coil superheat. Installing glass doors, which resulted in reduced cooling load, caused a reduction of 71% in refrigerant mass flow.

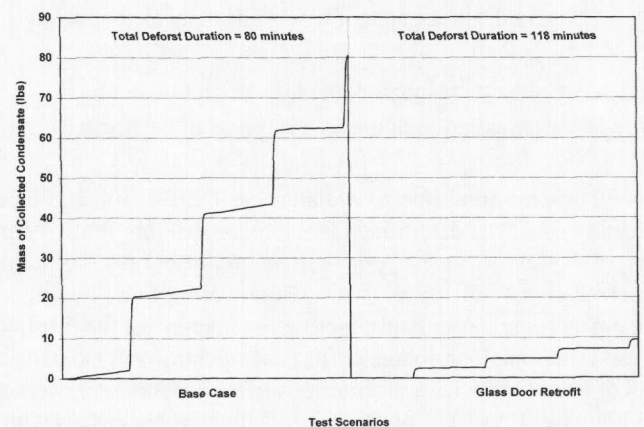


Figure 9 Two-minute profile of collected condensate mass for both scenarios.

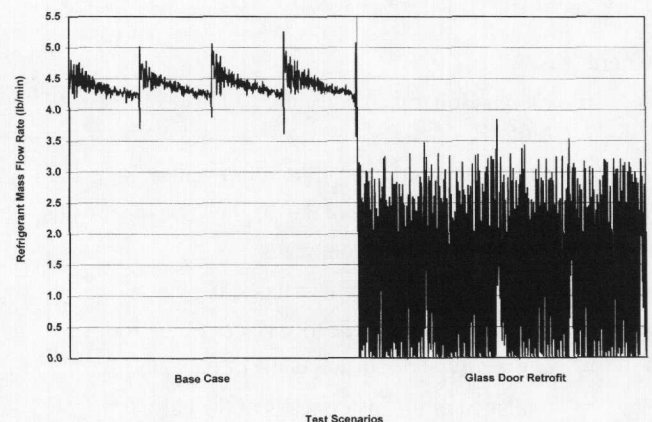


Figure 10 Two-minute profile of mass flow rate of refrigerant for both scenarios.

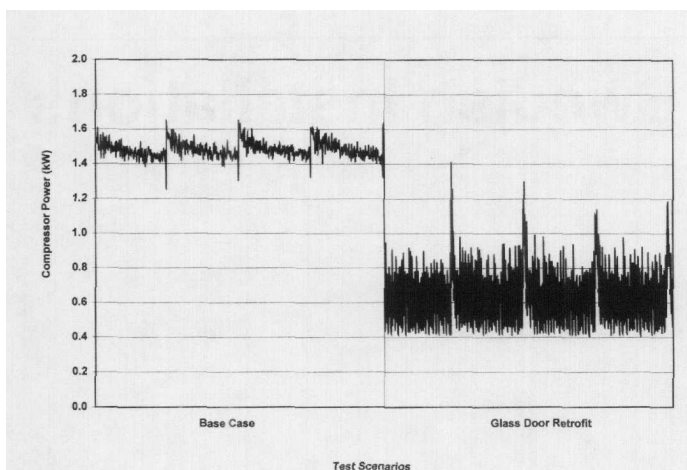


Figure 11 Two-minute profile of compressor actual power for both scenarios.

Compressor Power

Comparison of the two-minute refrigerant mass flow rate and compressor power profile revealed a close characteristic similarity between the two parameters (Figures 10 and 11). By maintaining fixed target suction and discharge pressures as well as constant indoor conditions, compressor power usage followed the same profile as refrigerant mass flow rate. Installing glass doors on the display case reduced the compressor power demand by roughly 87% due to the reduced refrigerant mass flow rate.

Power Use By Components

Figure 12 depicts the average component and compressor normalized power usage in both scenarios. The power consumed by evaporator fans and lighting system remained relatively unchanged for both scenarios. The ASH power consumption was measured at 233 watts in the post-retrofit case. The full-scale use of ASH in the post-retrofit case did not seem essential since the fog clearance time took place within few seconds.

CONCLUSIONS

The results of this investigation indicate that the thermal performance of an open vertical case is heavily influenced by the infiltration load. Retrofitting the fixture with glass doors reduced the total cooling load by 68% and the compressor

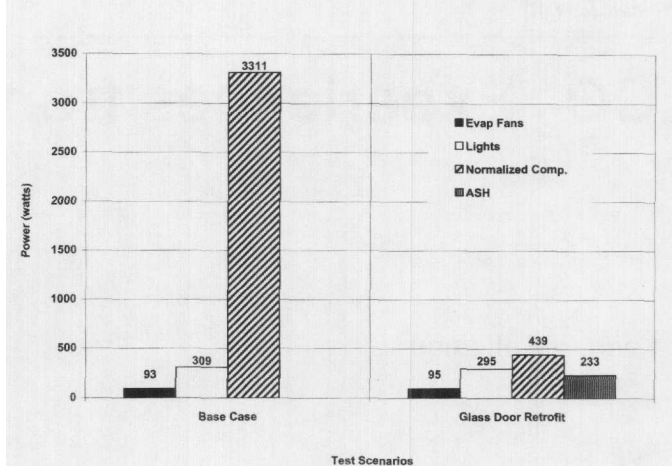


Figure 12 Average power consumed by each component for both scenarios.

power demand by 87%. It also resulted in 6°F lower average product temperatures.

In retrofitting open vertical cases with glass doors, close attention must be paid to downsizing of the TXV, resizing suction risers (for proper oil return), and possibly reducing the fixture airflow rate. Additionally, installing doors can cause a reversal pattern in product temperatures. As a result, products on the front upper shelf can become warmer than those on the front bottom shelf. This indicates a probable need for modification of air curtain system.

REFERENCES

- ASHRAE. 1983. *ANSI/ASHRAE Standard 72-1983, Method of testing open refrigerators for food stores*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.
- EPRI. 1992. *Field testing of high-efficiency supermarket refrigeration*, TR-100351, December 1992. Palo Alto: Electric Power Research Institute.
- Faramarzi, R. 1999. Efficient display case refrigeration. *ASHRAE Journal*, November, pp. 46-54.
- Faramarzi, R., B. Coburn, and R. Sarhadian. 2001. Anti-sweat heaters in refrigerated display cases. *ASHRAE Journal*, June, pp. 64-65.
- Holeman, J.P. 1990. *Heat transfer*, 7th ed., pp. 414. New York: McGraw-Hill, Inc.