Application of Best Industry Practices to the Design of Commercial Refrigerators

Development of a High Efficiency Reach-In Refrigerator

Cooperative Agreement No: DE-FC26-OONT41000
Recipient: Arthur D. Little, Inc.

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FINAL Report to
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Reference 72821

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Executive Summary

The substantial efficiency improvements which have been realized in residential refrigerators over the last twenty years due to implementation of the National Appliance Energy Conservation Act and changing consumer reactions to energy savings give an indication of the potential for improvement in the commercial sector, where few such efficiency improvements have been made to date. The purchase decision for commercial refrigerators is still focused primarily on first cost and product performance issues such as maximizing storage capacity, quick pulldown, durability, and reliability. The project applied techniques used extensively to reduce energy use in residential refrigeration to a commercial reach-in refrigerator. The results will also be applicable to other commercial refrigeration equipment, such as refrigerated vending machines, reach-in freezers, beverage merchandisers, etc.

The project described in this paper was a collaboration involving the Appliance and Building Technology Sector of TIAX, the Delfield Company, and the U. S. Department of Energy's Office of Building Technologies. Funding was provided by DOE through Cooperative Agreement No. DE-FC26-00NT41000. The program plan and schedule were structured to assure successful integration of the TIAX work on development of efficient design concepts into Delfield's simultaneous development of the Vantage product line.

The energy-saving design options evaluated as part of the development included brushless DC and PSC fan motors, high-efficiency compressors, variable-speed compressor technology, cabinet thermal improvement (particularly in the face frame area), increased insulation thickness, a trap for the condensate line, improved insulation, reduced-wattage antisweat heaters, non-electric antisweat heating, off-cycle defrost termination, rifled heat exchanger tubing, and system optimization (selection of heat exchangers, fans, and subcooling, superheat, and suction temperatures for efficient operation).

The project started with a thorough evaluation of the baseline Delfield Model 6051 two-door reach-in refrigerator. Performance testing was done to establish a performance baseline which, to meet end-users requirements, would have to be met or exceeded by the high-efficiency refrigerator design. Energy testing was done to establish the baseline energy use. Diagnostic testing such as reverse heat leak testing and insulation conductivity testing was done to evaluate factors contributing to the cabinet load and energy use.

Modeling was done to assess the energy savings potential of the energy saving design options. Discussion with vendors and cost modeling was done to assess the manufacturing cost impact of the options. Based on this work, the following group of design options was selected for incorporation in the final refrigerator design.

- Brushless DC evaporator fans.
- Improved face frame design

- Reduced antisweat heater wattage
- Condensate line trap
- Optimized refrigeration system

There was no net cost premium associated with these design changes, leading to a high-efficiency design requiring no payback of any initial additional investment.

Delfield incorporated these design options in the Vantage line design and built a first prototype, which was tested at TIAX. Additional design changes were implemented in the transition to manufacturing, based in part on results of initial prototype testing, and a pilot production unit was sent to TIAX for final testing. The energy use of the pilot production unit was 68% less than that of the baseline refrigerator when tested according to the ASHRAE 117 Energy Test Standard. The energy test results for the baseline refrigerator and the two new-design units is shown in Figure ES-1 below. The resulting energy consumption is well below Energy Star and proposed Canadian and California standards levels.

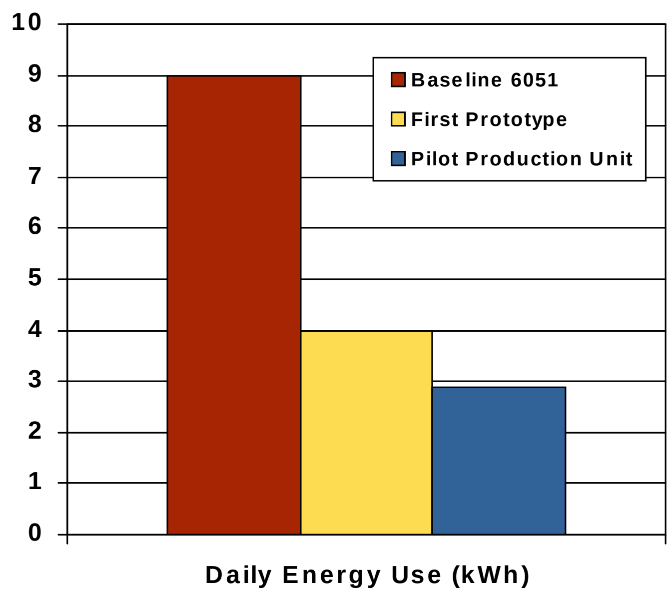


Figure ES-1: ASHRAE 117 Energy Test Results

Delfield has successfully transitioned the design to production and is manufacturing all configurations of the energy efficient reach-ins at a rate greater than 7,000 per year, with production quantities projected to double within a year.

1. Introduction

Reach-In refrigerators and freezers are used primarily in food-service establishments such as restaurants and cafeterias. They are used for temporary storage of food near food preparation and/or service areas. Food is placed in or taken out of reach-ins by opening the door and reaching in to the unit. A typical reach-in refrigerator is shown in Figure 1-1 below.

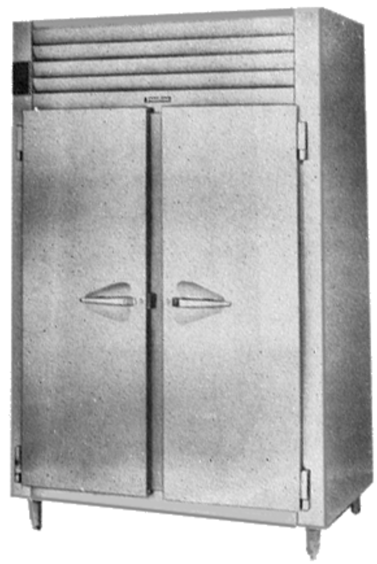


Figure 1-1: A Typical Reach-In Refrigerator

1.1 Current Energy Use

United States national annual energy use for commercial refrigeration totals roughly 91 billion kWh, or about 1 Quad of primary energy.[1] Energy use for reach-in refrigerators and freezers represents about 12% of this total [Reference 1].

For many years, little attention was paid to energy use of reach-in refrigerators. Typical energy consumption for a 2-door reach-in refrigerator was about 12 kWh/day in a 100°F ambient, or about 9 kWh/day when tested according to the ASHRAE Standard 117 Energy test procedure (see Section 1.2).

In recent years, more attention is being paid to energy use in reach-ins. An EPA Energy Star program has been developed for these products. Canada has developed draft proposals for energy regulations. California also has draft energy standards. Some efficient refrigerators are now available. However, prior to the commercialization of the refrigerator developed in this project, the majority of reach-in sales were of units with energy consumption no better than that of traditional units.

[1] Representation as primary energy includes consideration of generation, transmission, and distribution losses. The calculation assumes 11,000 Btu/kWh heat rate for conversion of site electricity use to primary energy use.

1.2 Regulatory and Standards Status

As mentioned above, more attention has been focussed recently on energy use of reach-in refrigerators. There has been discussion of U.S. government efficiency standards for reach-ins, and both Canada and California have developed draft standards. Also, an EPA Energy Star program has been established for the most popular solid-door reach-in product categories. All of these standards are based on the ASHRAE Standard 117 Test Procedure. The standards are summarized in Table 1-1 below.

Table 1-1: Energy Standards for Reach-In Refrigerators (kWh/day)

Equipment Category	Canada (Proposed)		California (Proposed)		EPA Energy Star (Voluntary)
	Standard	High-Efficiency	Tier 1	Tier 2	
Solid-Door Refrigerator	0.162V + 2.77	0.148V + 1.29	0.125V + 4.22	0.125V + 2.76	0.10V + 2.04
Glass-Door Refrigerator	0.323V + 5.53	0.293V + 2.58	0.172V + 5.78	0.172V + 4.77	
Solid-Door Freezer	0.471V + 2.55	0.427V – 3.48	0.398V + 2.83	0.398V + 2.28	0.40V + 1.38
Glass-Door Freezer	0.942V + 5.10	0.855V – 6.96	0.94V + 5.10	Same as Tier 1	
Solid-Door Combination	0.252AV + 5.21	0.230AV + 3.18	0.273AV + 2.63	0.273AV + 1.65	0.27AV – 0.71

Notes:
V: Internal Volume
AV: Adjusted Volume, equals 1.63 times freezer volume plus refrigerator volume
All standards are based on the ASHRAE Standard 117 Test Procedure

The standards for solid door reach-in refrigerators are compared in Figure 1-2 below. There are some differences in the standards. However, all of these efficiency levels are achievable without significant product cost impact, as is evident from the development described in this report.

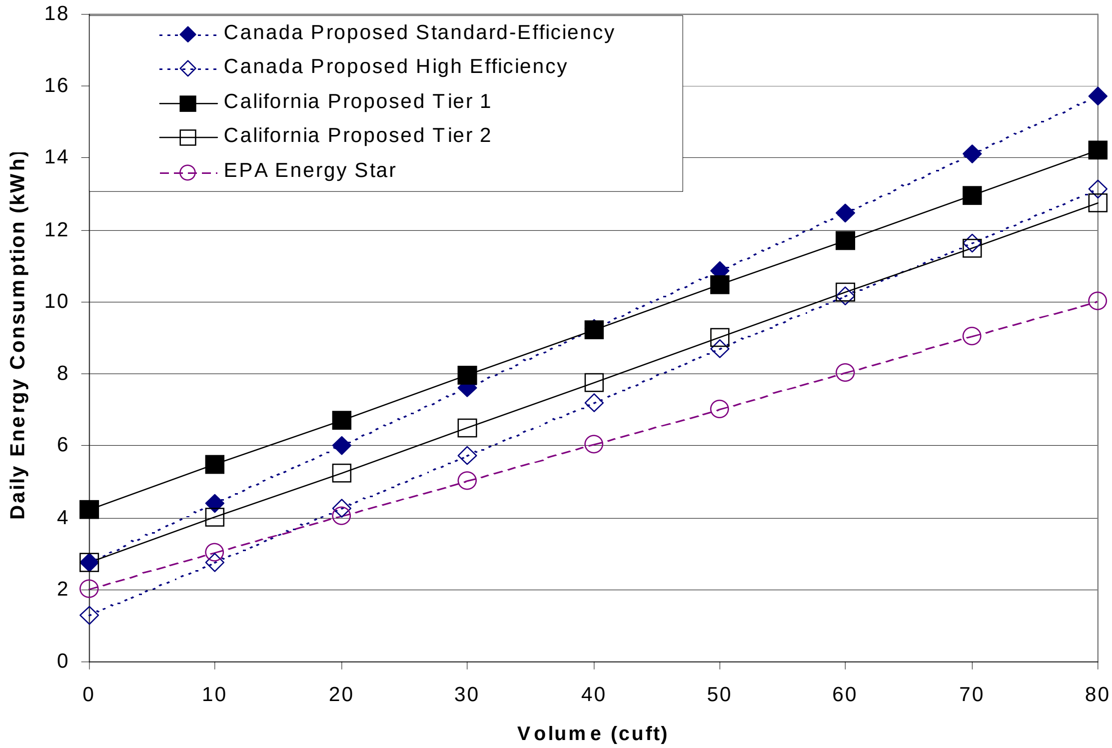


Figure 1-2: Comparison of Standards for Solid-Door Reach-In Refrigerators

Adoption of the ASHRAE 117 Test procedure by the proposed Canadian Standard set the precedent for its adoption by California and EPA Energy Star. The standard involves door-openings, control of ambient humidity, and simulated loads. While these requirements of the test potentially increase its ability to represent real-world conditions, the resulting complexities make it more difficult to carry out. The DOE residential refrigerator test standard, which is a closed-door test in a temperature-controlled but not humidity-controlled environment, is much more straightforward to carry out. The added complexity of the ASHRAE 117 test has to date not been seriously questioned, and it is likely that its adoption by Canada, California, and the EPA has created enough momentum for universal acceptance. The energy use measured with the test is comparable to energy which would be measured using a closed-door test in an 80^{o}F ambient (see Section 5-7).

1.3 Energy Savings Potential

Previous estimates of Reach-In Energy Use Reduction potential include the following sources.

- "Energy Savings Potential for Commercial Refrigeration Equipment", prepared by TIAX for the U.S. DOE Office of Building Technologies, June 1996 [Reference 1]
- "Cold Storage Temperature Stabilization Project Final Report", prepared by TIAX for the U.S.Army Soldier Systems Center, Natick, MA, July 12, 2000 [Reference 2]

- The proposal for this project provided an estimate for energy savings potential modified from the Reference 1 estimate.

The energy savings estimates for these sources are summarized in Table 1-2 below. These estimates showed great promise of significant energy use reduction at reasonable cost increase. During the course of this project, this potential for energy savings was demonstrated clearly: the energy use reduction realized for the two door refrigerator, the focus of the project, was 68% for the ASHRAE 117 Test procedure and 58% for closed-door testing in a 100^{o}F ambient. After cost engineering by Delfield, there was no cost increase for the high efficiency design.

Table 1-2: Previous Estimates of Reach-In Refrigerator Energy Savings Potential

Source	Design Changes	Energy Savings (%)	Cost Increase (%)
Reference 1	• Brushless DC Evaporator Fan • Brushless DC Condenser Fan • Hot Gas Antisweat • High Efficiency Compressor	44%	7%
Reference 2	• Face Frame/Gasket System Improvement • Hot Gas Antisweat • Variable Speed Compressor • Brushless DC Fan Motors • Off-Cycle Defrost Termination for Evaporator Fan • Improved Insulation	83%	Not evaluated
Proposal	• High Efficiency Compressor (20% Improvement) • Brushless DC Fans • Only One Evaporator Fan Runs during Compressor Off-Cycle • Improved Face Frame/Gasket Design • Hot Gas Antisweat • Improved Fan Blades • Refrigeration System Optimization • Eliminate Miscellaneous Thermal Shorts	51%	6%

1.4 DOE/TIAX/Delfield Development of High-Efficiency Reach-In Refrigerator

This project, funded by the U.S. DOE's Office of Building Technology and administered through the National Energy Technology Laboratory, was carried out in support of Delfield's development of the new Vantage line of reach-in refrigerators, freezers, and beverage merchandisers. The project organizational structure, showing staff involved in the project, is shown in Figure 1-3 below. Good timing was instrumental in the success of the project, as the need for energy use improvement, a good understanding of the potential for energy savings, the availability of key components, and Delfield's planned development of the new reach-in line came together at a time when funding was being provided by the DOE for this type of collaboration.

Delfield's willingness to adopt sensible energy-saving design options and the hard work of all project participants combined to create not just a design, but an entire line of reach-ins, whose energy performance is much better than the products which have been available previously, and at comparable cost.

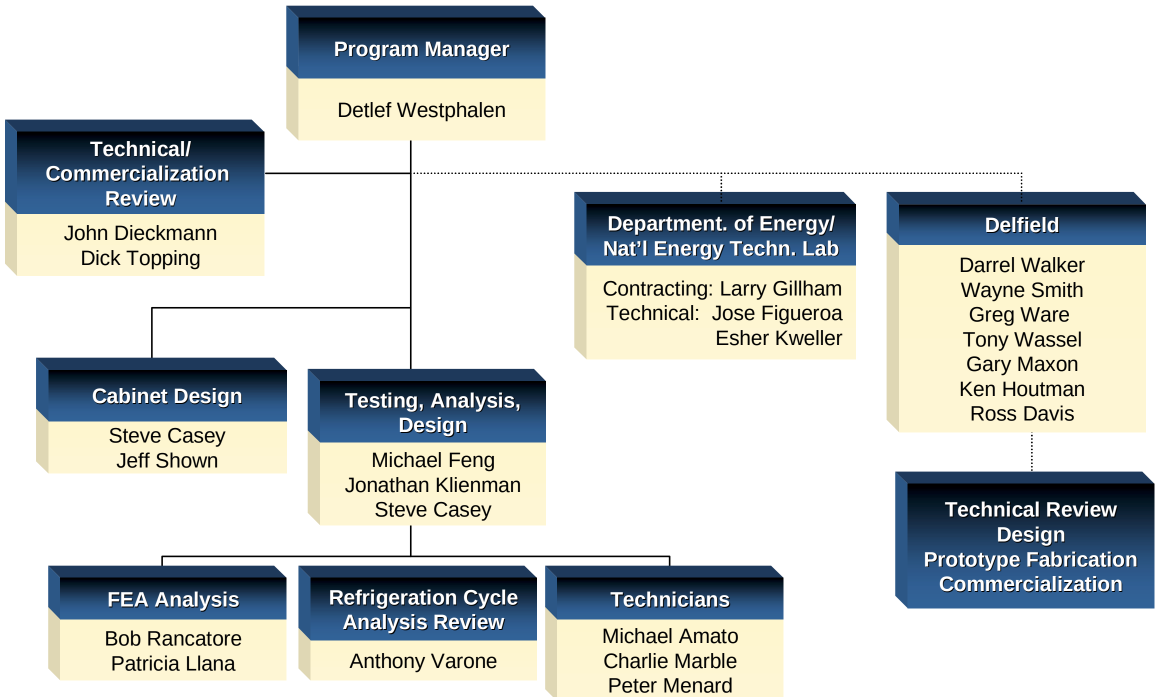


Figure 1-3: Project Organizational Chart

1.5 Project Objectives

The key objective of this project was to design a high-efficiency commercial reach-in refrigerator which would be commercialized. Introduction into the market was an important goal, since the product would do little to save energy if it were not purchased in the place of less efficient conventional refrigerators. While energy use reduction is relatively easy to measure, ability to be commercialized depends on cost, reliability, performance, and less tangible factors such as appearance.

Success criteria for the project and target values were chosen for the major quantifiable product attributes. These are listed in Table 1-3 below. The table lists both targets and success criteria. The latter are established as part of NETL project administration to evaluate projects' success. A further objective not specifically mentioned in the table is successful commercialization, which is essential for any of the project benefits to be realized in the marketplace.

Table 1-3: Project Success Criteria and Targets

Parameter	New Refrigerator Performance as Compared with Baseline 6051S Refrigerator	
	Target	Success Criteria
Energy Use Reduction	50%	33%
Manufacturing Cost Premium	$100	$150
Performance	Equal or better than Baseline Unit based on • NSF 7 Test • 100^{o}F/65%RH Door-Opening Test No visible sweat for 100^{o}F/65%RH Closed-Door conditions	
Reliability	Equal or better than Baseline Unit	
Aesthetics	Acceptable for typical food service environments	

1.6 Report Structure

The organization of this report is as follows.

Chapter 1 provides background information regarding energy use characteristics of traditional reach-in products, status of energy standards efforts, and previous estimates of energy savings potential. An introduction to the project is also provided.

Chapter 2 describes our thorough characterization of the baseline 2-door refrigerator which was replaced with the new design. This includes both modeling and testing.

Chapter 3 describes the energy savings options which were considered, and summarizes our analysis of savings potential and costs.

Chapter 4 describes the new Vantage-Series two-solid-door refrigerator design. Additional work done to support the design process is presented, as are results of testing of the initial prototype and final units.

Chapter 5 presents conclusions of this work

Note that performance test results are summarized in Section 2.2 for the baseline refrigerator and in Section 4.7 for the new design.

Appendix A provides summaries of preliminary refrigeration system design recommendations for different models of the Vantage line.

2. Baseline Refrigerator Characterization

Testing and Analysis were done to understand the energy use of the baseline refrigerator and the design and construction details which affect its cost.

2.1 Refrigerator Description

The baseline refrigerator is shown in Figure 2-1 below. It is a conventional two-door commercial reach-in refrigerator, with stainless steel outer skin as is typical for commercial refrigerators and freezers.

Figure 2-1: Baseline 6051 Refrigerator

The baseline refrigerator's entire refrigeration system is mounted behind the façade at the top of the unit. An insulated "evaporator box" covers the evaporator and the evaporator fans and isolates them from the ambient. Holes in the roof of the main cabinet allow cabinet air to circulate up, through the evaporator, and back down to the cabinet. A removable cover at the top of this box provides service access to the evaporator and fans. The interchanger and condensate drainage lines penetrate the evaporator box, and lead to the condensing unit, which mounts to the right of the evaporator box. The condensing unit is entirely exposed to ambient air behind the façade. The refrigeration system uses HCFC-22 refrigerant, and uses a capillary for refrigerant flow control. The key components of the refrigeration system are described in Table 2-1 below.

Table 2-1: Baseline 6051 Refrigerator Key Component Description

Component	Baseline Refrigerator (6051)
Compressor	Copeland JRS4-0050-IAA, HCFC-22
Condenser	Face 9" x 9" (229mm x 229mm) Tube rows 9 high x 3 deep Fins wavy, 6.5 fins per inch (3.9 mm fin spacing)
Condenser Fan	7.5-inch (191mm) diameter blade 6 W SP Motor, 35 W input
Evaporator	Face 21.5" x 7" (546 mm x 178 mm) Tube rows 7 high x 4 deep Fins wavy, 8 fins per inch (3.2 mm fin spacing)
Evaporator Fan	Two fans 6-inch (152 mm) diameter blade 6W SP Motor, 34W input

2.2 Performance Testing

A set of tests were developed to evaluate first the performance of the baseline refrigerator, and subsequently the performance of the advanced design unit. These tests were reviewed by Delfield prior to the start of the project to be sure that they would provide adequate assurance of equivalent refrigerating performance. Descriptions of the selected tests and the results for the baseline refrigerator are as follows.

1. National Sanitation Foundation (NSF) 7 Capacity Test: This is a test in 100ºF (37.8C) ambient used for commercial refrigerators to verify adequate capacity. Doors are closed and the refrigerator is empty during the test. The maximum compressor duty cycle for this test, required for a refrigerator to obtain NSF certification, is 70% [Reference 3], while keeping internal temperatures 40ºF (4.4C) or lower.
2. Closed Door Test in 80ºF (26.7C) ambient: This test is used as an indicator of energy use in more moderate ambient conditions. Cabinet temperature is controlled as for the NSF 7 test to be 40ºF (4.4C) or lower.
3. Door-Opening Test in 100ºF (37.8C) 65% RH ambient: This test is used to verify system performance in extreme conditions.
4. Energy Test (ASHRAE 117): This energy test has been adopted by Canada, California, and EPA Energy Star for commercial refrigerators. It is a complex test with door-openings, ambient humidity control, and internal salt-water test packages [Reference 4]. The ambient conditions of 75ºF (23.9C) and 55% RH are moderate for typical commercial kitchen environments. The daily energy use of the baseline unit was 9kWh.

The results of tests 1 through 3 are summarized in Table 2-2 below. For the NSF 7 test (100^oF Closed-door test), the percent compressor run time was 64%, representing comfortable margin as compared with the 70% requirement. The 24-hour energy use for this test condition was 12.1 kWh, which is consistent with published estimates of energy use (Reference 1 as well as Delfield product literature). The energy use and compressor run times for the 80^oF Closed-Door test are significantly lower than for the NSF 7 test, as is expected. For the two occasions when this test was done on the unit, the 24-hour energy use was 9.1 and 9.3 kWh. The results for the 100^oF 65% RH Door-Opening test showed that the baseline unit was not able to maintain temperature in the normal food storage range below 40^oF (4.4C). The temperature recovered between door openings only down to 53^oF (11.7C).

Table 2-2: Performance Test Results Summary for Baseline Refrigerator

NSF 7: 100^oF Closed-Door Test	
Date	12/11/00
Percent Run	64%
24-Hour Energy Use (kWh)	12.1
80^oF Closed-Door Test	
Date	12/14/00
Percent Run	46%
24-Hour Energy Use (kWh)	9.1
Date	12/19/00
Percent Run	47%
24-Hour Energy Use (kWh)	9.3
100^oF 65%RH Door-Opening Test	
Date	12/21/00
Typical Recovery Temperature[1]	53^oF

The significance of the recovery temperature for the door-opening test is demonstrated in Figure 2-2 below. The figure shows time series plots of cabinet temperatures and ambient temperature, and clearly shows the cabinet temperatures rising during door openings which occur every ten minutes. The cabinet temperatures recover to roughly 53^oF between door openings, and the compressor duty cycle is 100%.

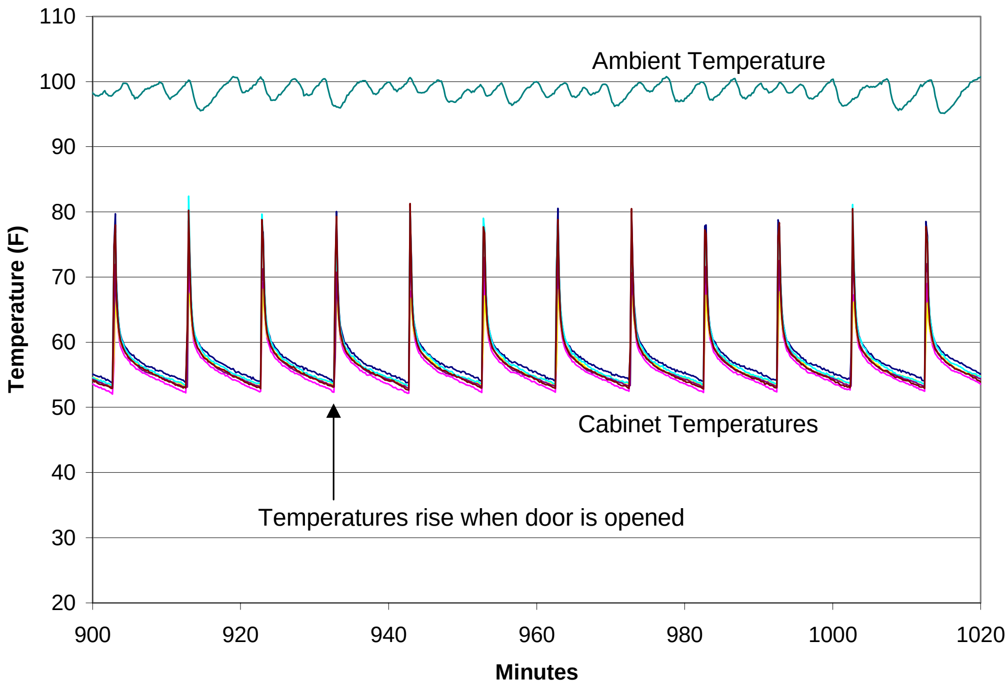


Figure 2-2: Baseline Refrigerator Door-Opening Test Data (100^{o}F 65% RH Ambient)

The 100^{o}F closed door test data was used as the basis for much of the modeling of refrigerator energy use. Typical refrigeration system operating parameters for this test are summarized in Table 2-3 below.

Table 2-3: Refrigeration System Operating Parameters for 100^{o}F Closed Door Test

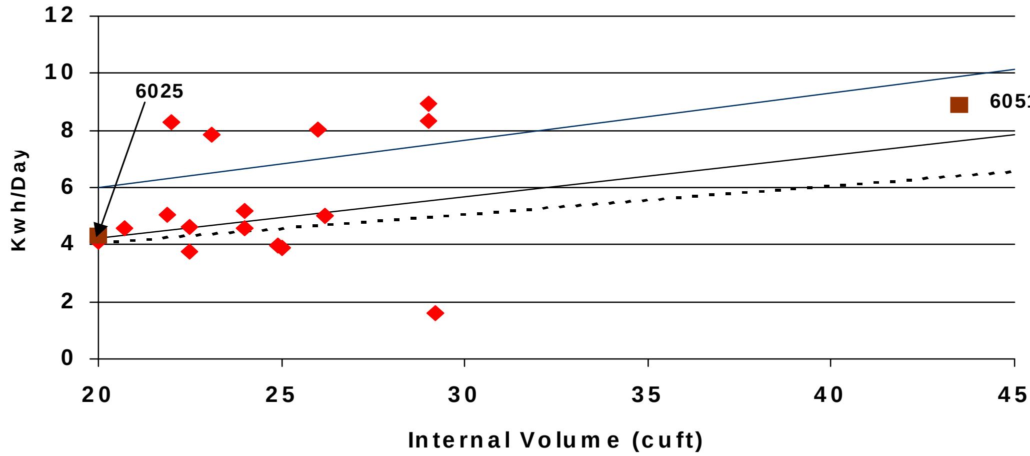
Parameter	Measurement	Notes
Discharge Pressure (psia)	293	125^{o}F saturated temperature
Suction Pressure (psia)	50	13^{o}F saturated temperature
Discharge Temperature (^{o}F)	223	
Condenser Exit Temperature (^{o}F)	123.5	1.5^{o}F subcooling
Evaporator Exit Temperature (^{o}F)	28	15^{o}F superheat
Suction Temperature (^{o}F)	65	37^{o}F temperature rise in interchanger
Evaporator Air Inlet Temperature(^{o}F)	35.4	
Evaporator Air Exit Temperature (^{o}F)	32.3	
Cycle Time (minutes)	10 to 30	Cycle time varied significantly for the handful of tests done. Differences between tests included charge quantity, compressor, and exact ambient conditions.

Note: Conditions are averages for the compressor on-cycle.

ASHRAE 117 Energy Test results for the baseline refrigerator are presented in Table 2-4 below. The energy use of about 9kWh is typical of commercial reach-in refrigerators, as seen in Figure 2-3 below. The data shown in the figure represents energy use reported to the California Energy Commission for solid-door reach-in refrigerators. The lines in the figure represent proposed California standards and the Energy Star standard. The 6025 unit indicated in the figure is a one-solid-door refrigerator which is, like the 6051, part of Delfield's 6000 line.

Table 2-4: ASHRAE 117 Energy Test Result Summary for Baseline Refrigerator

Date Tested	1/29/01
Thermostat Setting	~1
Refrigerant Type	HCFC-22
Quantity (ounces)	13
Ambient Temperatures (°F)	
Dry Bulb	76
Wet Bulb	64
Test Package Temperatures (°F)	
Integrated Average (IAT)	36.2
Coldest Package Avg (CTPA)	32.9
Warmest Package Avg (WTPA)	39.5
Maximum Warmest Package	40.4
Avg Temperature Start	36.9
Avg Temperature End	37.2
Total Energy Input (24-hours, kWh)	8.98
Percent Compressor Run Time	
Overall	36.3%
During Door-Openings	56.6%



The 6051 Data Point is this project's measurement. Other data points represent energy use reported to the California Energy commission. Solid Lines show California proposed "Standard" and "High" efficiency standards. Dashed Line shows Energy Star Standard

Figure 2-3: Baseline Refrigerator ASHRAE 117 Energy Use Comparison with Other Reach-In Refrigerators

The baseline refrigerator testing reported above was performed in one of TIAX's environmental chambers which can be controlled for both temperature and relative humidity. Temperature data for the testing was recorded with a computer-based data acquisition system using National Instruments data acquisition hardware and LabView software. The data acquisition system was also used for some of the tests for monitoring of refrigerant pressures and ambient humidity levels, and for control of door-openings for the ASHRAE 117 test as well as the 100^{o}F 65% RH door-opening test. This approach for recording of test data was used also for testing of the prototype and pilot production unit of the new design refrigerator.

2.3 Diagnostics Testing

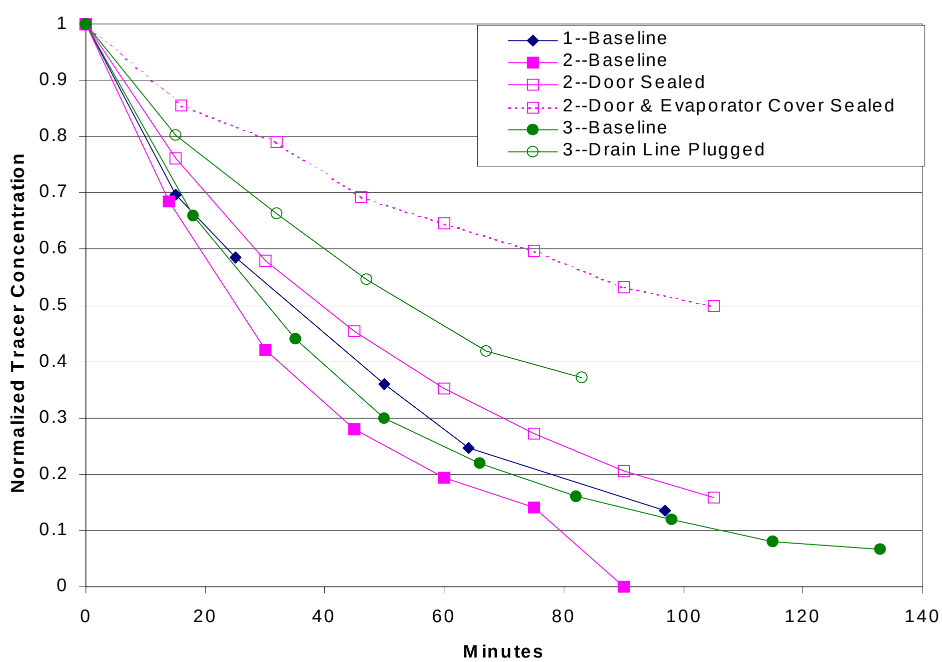
Additional testing was done with the baseline unit to improve the understanding of parameters affecting energy use. These additional tests include the following.

1. Reverse Heat Leak Test
2. Infiltration Measurement
3. Insulation Thermal Conductivity Measurement
4. Sweat Testing to Evaluate Reduction Potential for Antisweat Heater Wattage
5. Cabinet Load and Energy Use Impact of Antisweat Heater

These tests and their results are described in more detail below.

Reverse Heat Leak: This test was done with the environmental chamber at 35°F (1.7C) and the cabinet interior heated with a measured wattage to determine cabinet heat transfer characteristics. The internal wattage was controlled to maintain a 100°F interior temperature, thus subjecting the insulation to the same average temperature as for the NSF 7 test. Converted to heat leak for the NSF7 test conditions, cabinet load was 621 Btu/hr (182 W). Note that this load does not include the effect of the antisweat heater and that the internal load associated with the evaporator fan wattage would add to this to determine total refrigeration load.

Infiltration Measurement: A tracer gas test was done to determine cabinet infiltration level. Carbon monoxide tracer gas was injected into the unit until a suitable non-lethal initial concentration was established. The exponential decay in the concentration level, determined through occasional measurements, indicates the infiltration rate. The concentration profile results for these tests are shown in Figure 2-4 below. The infiltration rates calculated from the concentration profiles are presented in Table 2-5 below. The baseline cabinet's infiltration rate was roughly 1.0 cfm. The most promising options for reduction of infiltration are improvement of the door gasket seal and reduction of the infiltration associated with the condensate drain line. Note that the test involving sealing of the evaporator cover sealed the condensate drain line as well as the cabinet leakage in this area.



Note: Numbers 1, 2, and 3 represent different times that the testing was performed

Figure 2-4: Infiltration Testing--Tracer Concentration Decay Plots

Table 2-5: Infiltration Testing--Calculated Infiltration Rates

Test Series	Test Description	Infiltration Rate (cfm)
1	Baseline	0.9
2	Baseline	1.16
2	Door Sealed	0.76
2	Door and Evaporator Cover Sealed	0.28
3	Baseline	0.85
3	Condensate Drain Line Plugged	0.53

Thermal Conductivity Measurement: The conductivity of the baseline unit's insulation was measured using a Holometrix Lambda 2300V2 conductivity tester. The conductivity of the main cabinet insulation of the baseline unit was about 0.15 Btu-in/hr-sqft-F at 70^{o}F, roughly the average insulation temperature for the NSF7 test.. Test results for the Baseline 6051 are compared in Figure 2-5 below with measurements made of insulation of a residential refrigerator. The residential refrigerator insulation was measured both by its manufacturer and by TIAX to verify the accuracy of our measurements. The comparison of Delfield insulation with that of the residential refrigerator is of particular interest because both insulations were made using froth foam systems with HCFC-22 blowing agent. The Delfield insulation clearly was not as good as the residential refrigeration insulation. Furthermore, the purchased insulation panels used for the 6051 evaporator box top cover had even higher conductivity, although this was within its specification range of 0.14 to 0.19 Btu-in/hr-sqft-F at 70^{o}F average temperature.

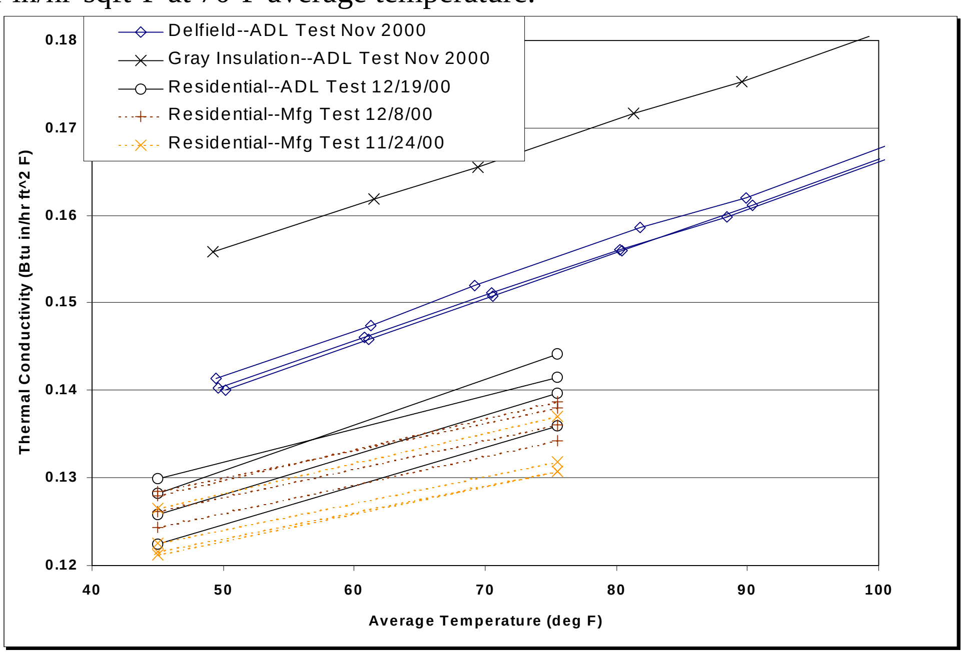


Figure 2-5: Insulation Thermal Conductivity Test Results

Sweat Testing to evaluate the potential to reduce antisweat heat: Testing was done to investigate possible reduction of the baseline refrigerator's antisweat heater wattages. This was a closed-door test in a 100^{o}F (37.8C) 65% RH ambient. The antisweat heater wattage was progressively reduced using a variac to adjust input voltage. The test indicated that the heater wattage could be reduced 30% before unacceptable condensation collected on the face frame surfaces.

Cabinet Load and Energy Use Impact of Antisweat Heater: An NSF 7 (100^{o}F Closed-Door) test was done to assess the cabinet load and energy use impact of the antisweat heaters. The percent run time of the compressor was reduced from 64% to 54% when the heaters were turned off, and 24-hour energy use reduced from 12.1 to 6.8 kWh.

2.4 Face Frame Modeling

Finite Element Analysis (FEA) was done to estimate face frame heat leakage, to assess the impact of design changes, and to estimate potential reductions in antisweat heater wattage for new face frame designs. The perimeter face frame was the focus of most of this work. The mullion area has slightly different thermal characteristics, especially for the new cabinet design, but an equally thorough investigation of the mullion was not considered necessary, since the perimeter face frame analysis was sufficient to point out trends and guide the design process. The FEA results for the baseline refrigerator design are discussed in this section.

A thermal finite element analysis was performed on the refrigerator door frame using ABAQUS 5.8-17 to help understand the heat loss of various gasket and door frame designs. Two-dimensional models of the door face frame were generated to analyze the heat transfer that takes place under various ambient and refrigerator temperatures and convection conditions, and for several gasket designs. The models included the antisweat heater and analyses were performed with and without the heater operating. Heat was input at the appropriate location to model the antisweat heater effects. Only a section of the face frame was modeled because it is believed that the door losses are constant far from those sections of the frame for a given door design. Metal, plastic and foam insulation material conductivities were included in the model to represent the various door frame components. Convection was included on the inside and outside surfaces of the frame.

Figure 2-6 below shows the cross section of the perimeter face frame of the baseline refrigerator. The outer liner wraps around the entire face and bends inward to meet the ABS interior liner. The gasket is attached to the door, while the magnet side of the gasket lands on the face frame to seal the door. The hollow spaces within the main cabinet and door are filled with insulation after assembly.

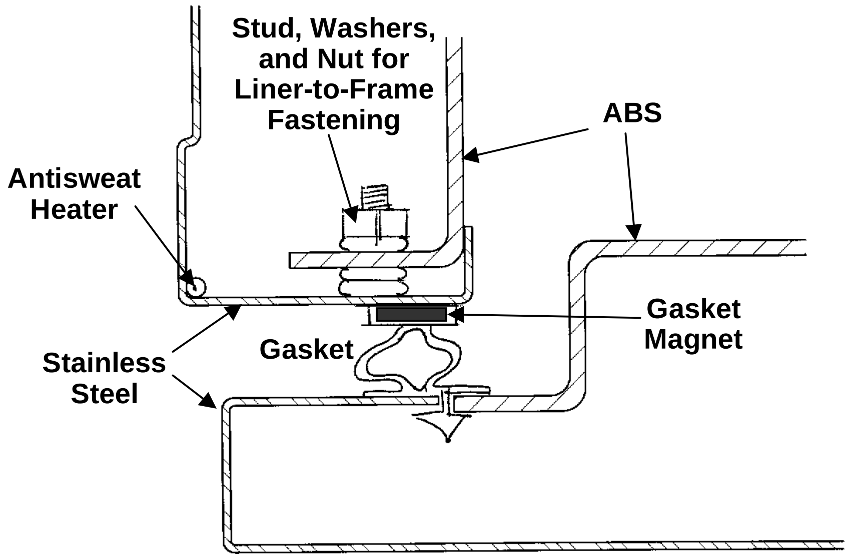


Figure 2-6: Baseline Refrigerator Face Frame Cross Section

The materials of construction of the face frame area and their thermal conductivities are well understood. However, the effective heat transfer between the interior and exterior surfaces and the air is somewhat less well understood, especially for the surfaces close to the gasket, which are not directly exposed to external air convection currents. A range of scenarios for the surface to air heat transfer were proposed and modeled in order to understand the sensitivity of the final result to the boundary condition assumptions. Surface temperature measurements were made to compare with the modeling to help assess which of the scenarios best reflects the actual system. Thermal loss per inch of face frame was determined for the face frame/gasket system and also for a solid corner representing the heat leak which would occur if no door were present. Analysis results are summarized in Table 2-6 below.

Table 2-6: Baseline Refrigerator Face Frame Results Summary

	Antisweat Heater Off	Antisweat Heater On
Cabinet Load with Face Frame (Btu/hr-in)	0.66	1.00
Solid-Corner Cabinet Load (Btu/hr-in)	0.28	0.28
Net Face Frame Load (Btu/hr-in)	0.38	0.72

2.5 Energy Use Modeling

The refrigerator cabinet load and energy use was modeled using analysis tools for refrigeration systems developed at TIAX. These analysis tools are a modification of the EPA Refrigerator Analysis (ERA) program.

The cabinet load summary for the baseline 6051 refrigerator operating with closed doors in a 100°F ambient (NSF7 Test) is presented in Table 2-7 below.

Table 2-7: Baseline Refrigerator Cabinet Load Summary

Component	Load (Btu/hr)
Cabinet and Door	422
Gasket and Face Frame System	137
Infiltration (1 scfm)	68
Antisweat Heaters	113
Off-Cycle Charge Migration Loss	100
Evaporator Fan Heat	232
TOTAL	**1,071**

The cabinet and door loads represent heat loss through the insulated box assuming no contribution from the face frame. This loss includes the load of the cabinet extension represented by the evaporator box on the top of the unit. The "Gasket and Face Frame System" load in the table, estimated based on FEA, does not include the contribution from the antisweat heater. Only sensible load was considered for the loss associated with infiltration. The subtotal of cabinet load for the first three items in Table 2-7 is 627 Btu/hr. This total compares well with the 621 Btu/hr reverse heat leak measurement, which measures the same three cabinet load components.

Additional contributions to cabinet load include the contribution to the face frame load represented by the antisweat heater, off-cycle charge migration loss, and the evaporator fan heat. The antisweat heater load adder was estimated based on FEA, and its cabinet load contribution represents 33% of the 100W (341 Btu/hr) heater power input. The charge migration loss was roughly estimated assuming that two-thirds of the charge starts out in the condenser as half liquid and half vapor, is transferred to the evaporator, and becomes 40ºF liquid prior to the next compressor on-cycle. This estimate assumed that average cycle time was 20 minutes. The 1,071 Btu/hr total cabinet load estimate is lower than but close to Delfield's cabinet load estimate of 1,207 Btu/hr.

The energy use model for the baseline refrigerator is summarized in Table 2-8 below. The key energy-using components are the compressor, the condenser fan, the evaporator fans, and the antisweat heaters. The light bulb uses a significant amount of energy when it is on (40W), but its on-time is very low or zero for most testing and for most end-use applications.

Table 2-8: Baseline Refrigerator Energy Use Summary for 100°F Closed Door Test

Component	Input Power (W)	Duty Cycle (%)	Daily Consumption (kWh/day)
Compressor	484	64%	7.4
Condenser Fan	35	64%	0.5
Evaporator Fans	68	100%	1.6
Antisweat Heaters	100	100%	2.4
Lighting	40	0%	0
			12.0

The model of the baseline unit's refrigeration system was adjusted to match key measured operating parameters such as operating pressure, superheat, etc. Results of the model revealed that only about 50% of the baseline unit's evaporator was operating in the two-phase refrigerant zone. Subsequent testing with a surface thermocouple placed on an evaporator return bend about halfway through the evaporator's refrigerant circuit indicated that the refrigerant was superheated at this location, thus confirming the model result. This showed the potential for improvement through adjustment of the refrigerant charge and the capillary tube length.

3. Design Options for Energy Use Reductions

3.1 Summary

A number of options were evaluated for energy savings potential and manufacturing cost impact to assist in selection of the options to be incorporated in the new design refrigerator. This section provides a description of the technologies and summaries of the energy savings and cost impact analyses, both individually and for groups of technology options.

3.2 Design Option Description

3.2.1 High-Efficiency Compressors

Compressor vendors were contacted to obtain compressor performance information. Table 3-1 below presents information for both the baseline refrigerator compressor and the possible alternative compressors. Note that the alternative compressors include both HFC-134a compressors and R-404A compressors. One of Delfield’s goals in the development of the advanced reach-in line was to select a single refrigerant which could be used for all models including freezers and refrigerators. This goal would not be achievable with HFC-134a, because evaporating pressures would be below atmospheric for the low end of the commercial freezer operating temperature range, and this led eventually to the selection of an R-404A compressor. However, HFC-134a was investigated at the outset because the efficiency of these compressors is typically higher.

Some observations regarding the compressor options are as follows.

- HFC-134a compressors generally are more efficient than R-404A or HCFC-22 compressors.
- Low starting torque compressors are more efficient. However, capability for instant restart after compressor cutout is generally required in commercial refrigeration applications, which makes high starting torque necessary.
- Some of the smaller compressors were examined for possible use for a two-compressor refrigeration system.
- While the baseline compressor is more efficient than both the initially-selected Tecumseh compressor and the ultimately specified Copeland compressor, additional system redesign allowed the refrigeration system efficiency to be improved significantly. The improvements essentially involve optimization of the system to provide higher evaporator temperatures and lower condensing temperatures without significant system cost increase.

Table 3-1: Compressor Options

		Performance (see Note 1)				
Manufacturer	**Model**	**Capacity (Btu/hr)**	**Power (W)**	**EER (Btu/W-hr)**	**Refrigerant**	
Copeland	JRS4-0050-IAA	2,220	485	4.58	22	See Note 2
Copeland	JS25C1E	2,121	579	3.66	404A	
Copeland	ASE19C3E	1,783	440	4.05	404A	See Note 3
Copeland	ASE24C3E	2,089	570	3.66	404A	
Copeland	ASB12C3E	1,182	350	3.38	404A	
Danfoss	NF5.5CLX	1,627	431	3.77	404A	
Danfoss	TF4CLX	1,132	271	4.18	404A	
Embraco	NB6144GK	1,179	299	3.95	404A	
Embraco	NE6181GK	2,065	453	4.56	404A	
Americold	HP121-1-3087	2,734	528	5.18	404A	See Note 6
Americold	HP117-1-3085	2,389	467	5.11	404A	
Americold	HP110-1-3083	1,639	267	6.13	404A	
Americold	HP310-1	1,749	350	5.00	404A	
Americold	RSH120	873	184	4.75	134a	
Americold	GRH105-1	1,140	169	6.74	134a	
Americold	GRV-108	1,703	262	6.51	134a	See Note 5
Americold	GRH104-1	956	136	7.03	134a	
Tecumseh	AEA9415YXA	1,451	334	4.34	134a	
Tecumseh	AZA0395YXA	905	222	4.08	134a	
Tecumseh	AEA9422ZXA	2,019	560	3.61	404A	See Note 4
Tecumseh	AKA9427ZXA	2,064	589	3.50	404A	
Panasonic	DA57C84RCU6	1,175	200	5.88	134a	

Notes:

1. Performance estimated for 20°F evaporating temperature, 130°F condensing temperature, 65°F suction temperature, 0°F subcooling.
2. First Copeland compressor listed is used in the baseline refrigerator
3. Shaded Copeland compressor is used in the production version of the Vantage two-door refrigerator.
4. Shaded Tecumseh compressor was the initial selection and was used in development of refrigeration system design recommendations.
5. The Americold GRV-108 is a variable speed compressor.
6. The Americold HP100 series of compressors have low starting torque.

3.2.2 Efficient Fan Motors

Most fan motors used in conventional commercial reach-in refrigerators and freezers are shaded pole motors. Power input for commercial refrigerators is almost exclusively single-phase, either 115-volt or 230-volt. Single-phase motors require an auxiliary winding offset from the main motor winding to ensure proper motor rotation and to smooth out start-up torque. The least-expensive option for achieving this for the low-power output required for fans is the shaded pole design, which is inherently inefficient. The key motor types of interest for improved efficiency are permanent-split-capacitor (PSC) motors and Brushless DC motors.

PSC motors are similar to shaded pole motors in design, but they incorporate a capacitor in series with the offset winding to shift the current of this winding out of phase with the main winding. This improves both torque and steady-state efficiency. PSC motors have been available for many years, and they have been available in the same housings as shaded pole motors. Their use has been limited due to their increased cost, which is typically on the order of $10 for a motor with 6W shaft power output, which nearly doubles the $10 to $15 cost for a shaded pole 6W motor.

Brushless DC motors have been emerging as an alternative high-efficiency motor technology for a number of applications. While they are used extensively in premium applications requiring their improved performance, reduced weight, or reduced power input they have had limited market penetration due to higher costs. They have been used on a limited basis for applications such as AC unit indoor blower motors and residential refrigerator fan motors. Manufacturers have recently focussed more attention on this technology, however, and have introduced products which are more suitable for the commercial refrigeration industry. The brushless DC fan which was eventually selected for this project is shown in Figure 3-1 below. One of the unique features that this fan offers is that it can be purchased with two preselected operating speeds.

Blade Diameter 8.2 inches

Figure 3-1: GE Motors 58 Series Brushless DC Fan

Data for key fan motor options which were evaluated, as well as for the baseline refrigerator fans, are presented in Table 3-2 below. This list represents only the handful of fan motor options which were available in late 2000 when this phase of the work was ongoing. It is expected that a broader range of high-efficiency fan motors is now available.

Table 3-2: Fan and Fan Motor Options

				Performance (per fan)			Efficiency	
Manufacturer	Motor Type	Fan Type	Model #	Flow Rate [CFM]	Pressure [in wc]	Input Watts [W]	Electricity/ Air	Motor
Baseline	Shaded Pole	Axial	(Evaporator)	115	0.1	34	4%	
Baseline	Shaded Pole	Axial	(Condenser)	218	0.11	35	8%	
EBM		centrifugal	R2E133-BB72-13	175	0.1	30	7%	
EBM		centrifugal	R2S175-AB60-38	218	0.1	65	4%	
EBM		centrifugal	R2E220-AA44-23	520	0.1	100	6%	
Nidec		Axial	A30330	115	0.1	23	6%	
Nidec		Axial	A30318	240	0.1	32	9%	
GE Motors	ECM	Axial	58 Series	115	0.1	5	27%	70%
GE Motors	ECM	Axial	58 Series	220	0.11	9.7	29%	70%
GE Motors	ECM	Axial	58 Series	230	0.1	9.4	29%	70%
Morril	Brushless	Axial	SSC FV800CW20S38	115	0.1	9	15%	46%
Morril	Brushless	Axial	SSC FV800CW30S38	230	0.11	19	16%	57%
Morril	Brushless	Axial	SSC FV800CW30S38	230	0.1	19	14%	57%
Morril	PSC	Axial	PSC4BE6 - FT775CW18S63	126	0.1	18.75	8%	46%
Morril	PSC	Axial	PSC4BE6 - FV875CW20S38	238	0.1	19.75	14%	48%
Morril	PSC	Axial	PSC4BE6 - FV875CW20S39	238	0.1	19.75	14%	48%

Note: ECM is GE Motors' trade name for their brushless DC motors.

3.2.3 Heat Exchanger Options

The alternatives for heat exchanger design which were considered during this project are as follows.

- Increase in heat exchanger core dimensions.
- Increase in fin density was considered but not pursued due to concerns regarding fouling with dirt and dust for the condenser and the potential for frost for the evaporator.
- Use of rifled tubing rather than the smooth tubing which is used for the baseline unit.

Use of microchannel heat exchangers and/or more aggressive fin surface designs was not considered. Rifled tubing, microchannel heat exchangers, or alternative fin surfaces could result in more compact heat exchanger designs. However, heat exchanger size is not a cost driver for the two-door refrigerator as it would be for an air-conditioning unit or perhaps a freezer.

3.2.4 Improved Insulation

As shown in Section 2.3, the conductivity of Delfield's insulation was not as low as in other applications using similar materials. Defield will be using a froth foam insulation system using HCFC-22 as the blowing agent for many years to come, since the phaseout of this substance for use as an insulation blowing agent will not be phased out until January 1, 2010. Hence, the conductivity levels shown in Figure 2-5 achieved by the residential refrigerator manufacturer were assumed to be reasonable targets for Delfield. During the course of bringing the new design to manufacturing, Delfield transitioned to the same insulation vendor as is used by the residential manufacturer whose insulation we tested. The conductivity improvement assumed for analysis is 10%. This change reduces total refrigeration load by 4% and energy use by about 3% (for NSF 7 conditions). A nominal 20% cost premium over a baseline cost of $0.95/lb was assumed for comparison with other design options, but the actual cost difference depends more on vendor capabilities and negotiations. The total insulation quantity for the unit was 40 lb.

3.2.5 Improved Face Frame Design

One of the key differences in construction between residential refrigerators and conventional commercial refrigerators lies in the design of the face frame. As shown in Figure 2-6 the face frame metal wraps around the front of the face and penetrates into the refrigerator interior. This provides a path for heat leak. In residential refrigerators, the face frame metal is sealed from the interior by the gasket.

Four design concepts for the face frame were developed and analyzed for energy savings and cost impact. Two of these options are derivative of the baseline refrigerator face frame design. The other two involve a plastic extrusion face frame. See Figure 3-2 below.

Different variations on the face frame design concepts involve the following considerations.

- Repositioning of the antisweat heater closer to the coldest externally exposed edge of the stainless steel face frame should put the heat where it is most needed, reduce heat loss to the environment, and reduce total wattage.
- Repositioning of the gasket closer to the external corner allows the stainless steel face frame to be cut back further, thus moving this key path for heat leak further from the interior of the refrigerator.
- Use of a gasket with an inner pocket allows the gasket to seal the stainless steel face frame from being exposed to the interior of the cabinet.
- Antisweat heater wattage can be reduced if the face frame heat leak is reduced, because the externally-exposed surfaces are not cooled as much by this heat leak.

The cabinet load reduction and antisweat heater wattage reduction for some of the face frame design concepts were estimated using Finite Element Analysis. The cost premium estimates for the third and fourth design options were high enough to discourage further investigation of these options. The analysis results and cost premium estimates are presented in Table 3-3 below.

Table 3-3: Face Frame Design Concept Performance and Cost Premium Summary

Face Frame Design Concept	Cabinet Load Reduction (%)	Antisweat Heater Wattage Reduction (%)	Total Energy Use Reduction (%)	Total Cost Premium ($)
1	15%	46%	19%	$0
2	16%	78%	26%	$0
3				$20
4				$40

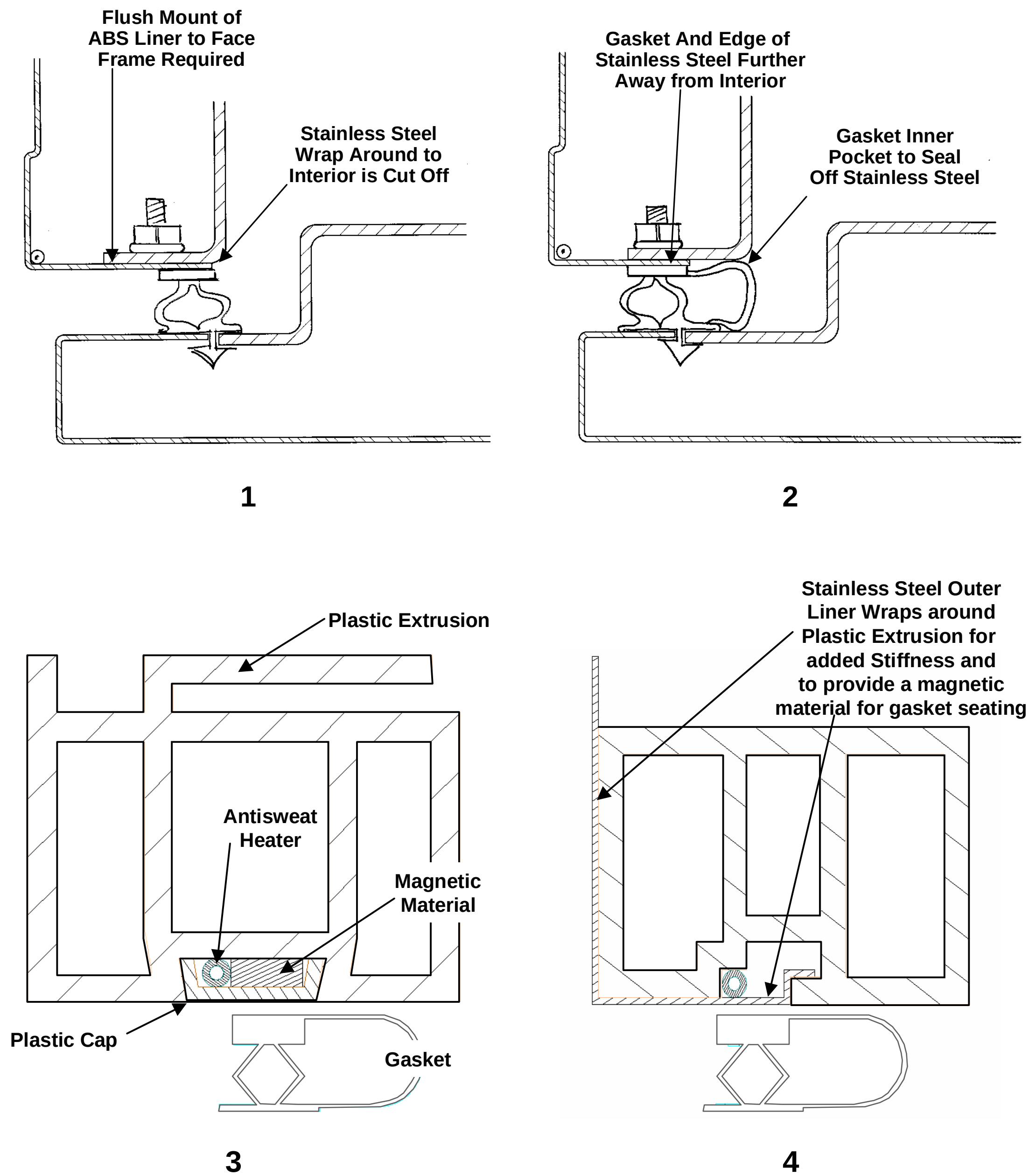


Figure 3-2: Face Frame Design Concepts

3.2.6 Off-Cycle Defrost Shutdown

Evaporating temperature for commercial refrigerators is generally below freezing, which makes some form of defrost necessary. In refrigerators, cabinet temperatures are high enough that the cabinet air can be used to defrost the evaporator. This requires that the evaporator fan run during the compressor off-cycle to assure good thermal contact between the air and the evaporator. As a result, evaporator fans generally run 100% of the time in commercial refrigerators. This is called off-cycle defrost. Energy savings can be achieved by running the evaporator fan only as long as is required to achieve defrost. Savings can be high, particularly for night time periods, when the door is not opened to admit water vapor, and when the low ambient temperature and low usage significantly reduces compressor duty cycle.

In order to achieve off-cycle defrost shutdown, a surface temperature thermostat must be mounted on the evaporator to assure that the fan continues to run until frost is melted. Figure 3-3 below shows a simple control circuit which would be used to implement this control. The OEM cost of the additional thermostat is about $10.

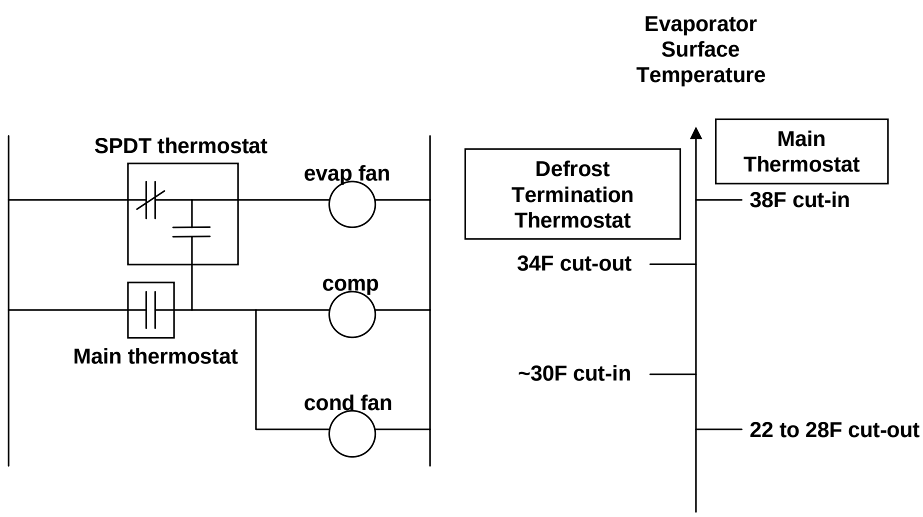


Figure 3-3: Control Circuit and Switching Logic for Off-Cycle Defrost Shutdown

3.2.7 Compressor Modulation

Compressor modulation is used in numerous air-conditioning and refrigeration applications to provide better performance and save energy. Examples are unequal parallel compressor racks for supermarkets and modulation for centrifugal chillers using inlet guide vanes or variable speed. The development of variable-speed refrigeration compressors for residential and small commercial refrigeration applications has not yet made much market impact. Nearly all compressor vendors have claimed to be working on development of variable speed compressors for residential applications at one time or another. Americold, Embraco, and Panasonic have developed compressors which have been used in residential refrigerators, most of them in Japan. These compressors all use permanent magnet motor technology to maximize efficiency. There is also work ongoing to apply variable speed compressors to commercial applications, but the focus in this application has been more on performance than energy savings, and these units are often induction motors operating with a variable speed drive. Besides variable-speed compressors, compressor staging with two small compressors can be used to improve commercial refrigerator efficiency.

Energy savings for compressor modulation would accrue due to

1) The reduction in mass flow during part load operation would allow the heat exchangers to operate more efficiently, thus reducing compressor pressure ratio.
2) The reduction in heat exchanger load would allow reduction of fan speed. A 50% reduction in fan speed during part load operation could result in a factor of 8 reduction in fan power, based on the fan law for power input.
3) The permanent magnet motors used in variable speed compressors are generally more efficient than the induction motors they replace.
4) Valve loss and friction losses in the compressor can be reduced.

The key drawbacks to use of modulating compressor systems are

1) High cost either for a variable-speed compressor system (which includes both a compressor and a separate motor drive) or for the second compressor of a dual-compressor system.
2) Either approach requires use of more sophisticated controls. This is especially true of a variable speed compressor, which requires a control system which can generate the required variable speed signal and also decide at what speed the compressor should operate. This generally implies that electronic controls are required. This is not common for most commercial refrigeration equipment. It still generally involves a manufacturing cost premium, most likely involves added development cost, and is more difficult for technicians to troubleshoot.
3) The added complexity of a modulating system could present more potential for system failure, and makes troubleshooting and repair more difficult for technicians.
4) Capillary tubes are generally used to control refrigerant flow in reach-in refrigerators and freezers. These devices are optimized for a given flow rate, and performance compromise would be made in sizing them for variable capacity systems.

Energy savings and costs were estimated for a variable speed compressor system. This analysis was based on the variable speed compressor listed in Table 3-1 above.

3.2.8 Hot Gas Antisweat Heating

Hot gas antisweat heating is used extensively in residential refrigerators to reduce energy use. In fact, there is very little use of electric resistive heating for antisweat purposes in residential units. The opposite is true for commercial units. Hot gas antisweat heating was considered for this project but not pursued, due to expectations that this design option would save limited energy, involve significant cost, and represent significant technical risk.

3.2.9 Condensate Line Trap

The infiltration testing discussed in Section 2.3 showed that blockage of the condensate line would significantly reduce infiltration, thus reducing cabinet load and energy use. A trap is a dip in a drain line, commonly used in building plumbing, which traps some of the draining water. The water seals the line, preventing the movement of air past the dip. Its cost is minimal.

3.3 Analysis Summary

Estimated energy savings and costs for the individual energy savings options discussed above are presented in Figure 3-4 below. Note that the energy savings is based on NS7 test conditions (100°F ambient closed-door operation). Variable-speed compressor results are off the chart with cost premium at $100.

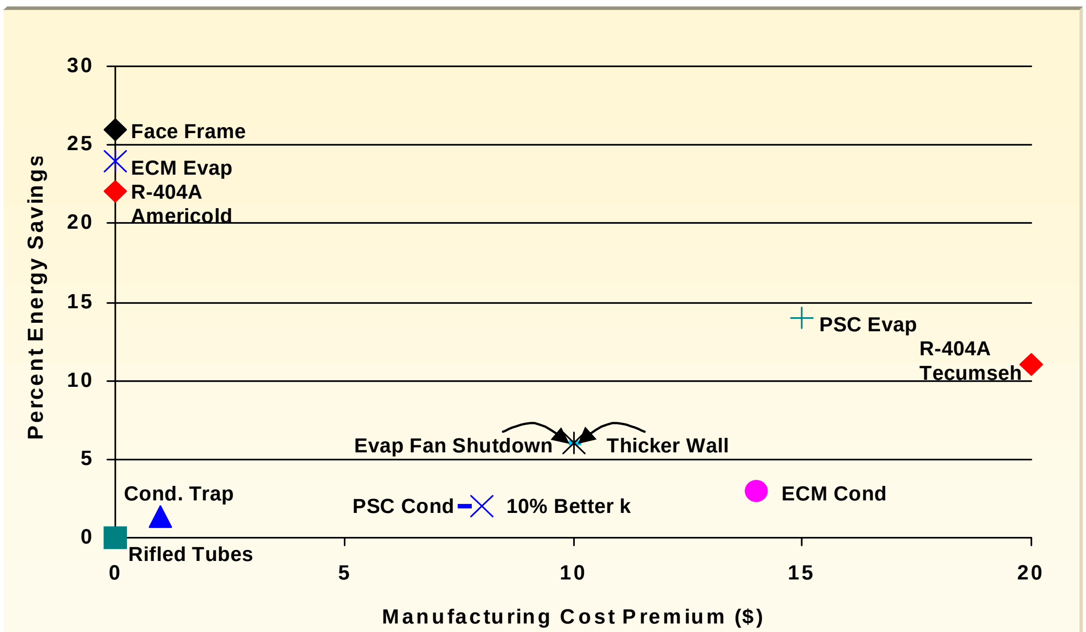


Figure 3-4: Energy Savings and Cost Estimates for Energy Saving Design Options

Based on the initial analysis involving individual options, groups of options were analyzed for cumulative savings. The groups of options are defined in Table 3-4 below. The summary of energy savings and costs for these groups of options is shown in Table 3-5. Again, the energy savings are based on operation with NSF7 test conditions.

Table 3-4: Analyzed Groups of Energy Saving Design Options

R-404A	◆ Single-speed R-404A compressor ◆ System Optimization ◆ Tecumseh and Americold Compressor Options
Thermally Improved Cabinet	◆ Thermally Improved Face Frame ◆ Thermally Improved Gasket ◆ Reduced Antisweat Heater Wattage ◆ Condensate Line Trap
Fan Improvements	◆ GE ECM Evaporator Fan ◆ PSC Condenser Fan ◆ Evap Fan Shutdown (after defrost)
R-404A, Cabinet, Fan Improvements	◆ **ALL THREE OF ABOVE IMPROVEMENTS**
Variable Speed System	◆ **CABINET AND FAN IMPROVEMENTS PLUS:** ◆ VS Refrigeration System (R-134a) ◆ Electronic Control ◆ Hot Gas Antisweat Loop

Table 3-5: Analysis Results for Groups of Energy Saving Design Options

	Refrigeration Load (Btu/hr)	Peak Amps	Daily Energy Use (kWh)	Manufacturing Cost Premium
Baseline 6051	1,070	8.7	12	(-)
R-404A - Americold Tecumseh	1,070 1,070	6.7	9.2 10.2	(-) $20 to $30
Cabinet	872	8.0	8.6	(-)
Fan	861	7.9	8.7	$18
R-404A, Cabinet, Fan - Americold Tecumseh	666 666	5.2	3.8 4.5	$18 $40 to $50
Variable Speed (R-134a)	664	4.5	2.2	$120

The R-404A Tecumseh option was based on use of the AEA9422ZXA compressor, while the Americold option was based on the HP310-1 compressor (see Table 3-2). Both of these compressors have high starting torque, which is desired for this application, but the Ameicold compressor has a significantly higher EER. Furthermore, the cost of the Tecumseh compressor is somewhat higher. However, one key product feature not available for Americold during the time when analysis was being done in early 2001, was widespread distribution and stocking which would allow immediate local purchase of a replacement compressor. This issue was important enough to make selection of Americold undesirable for Delfield.

3.4 Preliminary Energy Savings Verification Testing

Modifications were made to the 6051 unit which had been used for baseline refrigerator testing to provide preliminary verification of energy savings projections. This testing was done in two steps as follows.

1) The two existing evaporator fans were replaced with one GE 58 Series ECM fan, adjusted to provide the same evaporator air flow as for the baseline unit. Also, a trap was built in to the condensate drain line.

2) An attempt was made to remove the portion of the face frame which bends in toward the cabinet interior. This was done by using a small rotary saw. The face frame of the modified cabinet was similar to Configuration 1 of Figure 3-2 above. Unfortunately, the antisweat heaters were severed during the preparation. Hence, spare antisweat heaters were taped to the external surface of the face frame. The antisweat heater wattage was reduced from 100W to 86W to achieve comparable face frame surface temperature as for the baseline unit. Also, a first iteration was made in optimizing the capillary tube and refrigerant charge.

Both of the modified configurations were tested in NSF 7 conditions (100ºF ambient, closed door test). The results are presented in Table 3-6 below. As can be seen, both compressor duty cycle and energy use were significantly reduced. The discrepancy between predicted and measured savings for the final modification is attributed to (1) the attempted face frame modification was very rough, (2) external placement of the antisweat heaters would result in higher input wattage, (3) optimization of the system was partial—evaporator utilization for the test was still somewhat less than could be achieved. The reduction in antisweat heater wattage to only 86W indicates that the face frame thermal leakage was not reduced as much as it should have been. The results certainly presented good confirmation that the tested modifications were worth pursuing with a properly fabricated prototype.

Table 3-6: Preliminary Design Option Verification Test Results

Test Case	NSF 7 Test 24-hour Energy Use (kWh)	Compressor Duty Cycle (%)	Measured Savings (%)	Predicted Savings (%)
Baseline	12	64%	-	-
GE 58 Series ECM and Condensate Trap	9	54%	25%	26%
Above Modifications with Face Frame Configuration #1 Partial Capillary Optimization	7.5	47%	38%	49%

4. Delfield Vantage Series Design—Two-Solid-Door Refrigerator

4.1 Vantage Series Overview

While the focus of TIAX's work on this project was energy savings, Delfield had a number of objectives for the Vantage series of reach-in refrigerators and freezers. Two of the key objectives were reduction of manufacturing cost and design flexibility to address a range of market needs. The Vantage series provides an economical reach-in design to meet the needs of the traditional foodservice market. The series has been re-configured with a bottom-mount condensing unit, which allows it to be used also as a beverage merchandiser.

The Vantage series has multiple model configurations, including refrigerators, freezers, and beverage merchandisers. The initial offering as a beverage merchandiser uses the same refrigeration system as the refrigerator, but utilizes glass doors, different shelving, and has provision for sales graphics above the door. Future offerings may include a redesigned refrigeration system to meet more rapid temperature pulldown requirements specified by bottling companies.

Vantage series cabinet sizes range from single-door to three-door. The three-door unit consists of a two-door unit and a single-door unit combined in a single package. Door configurations include full-height and half-height doors, and both solid and glass doors are available. Glass doors for freezer units incorporate argon fill and low-emissivity glazing to reduce cabinet load.

Other noteworthy design features of the Vantage line include (1) use of a one-piece ABS interior liner, a popular feature which was also incorporated in the older 6000 series, (2) adjustable shelving system supported by shelf supports molded into the ABS liner, (3) evaporator and evaporator fan package (unit cooler) mounted inside the unit's insulated cabinet with piping penetration to the rear, (4) recess in rear of unit for routing of the interchanger and condensate drain line from the unit cooler to the condensing unit, (5) flexible connection of suction line and capillary to the condensing unit, to allow the condensing unit to be slid forward forward for service access.

Figure 4-1 below shows a Vantage Line two-door refrigerator.

Figure 4-1: Vantage Line Two-Door Reach-In Refrigerator

4.2 Development Approach

As mentioned above, the key focus of the DOE-funded work supporting the Vantage series development was energy savings. This work was integrated into Delfield's development of the new reach-in line. This subsection provides a brief description of the integration.

The work described in Sections 2 (Baseline Unit Characterization) and 3 (Design Option Evaluation) were done prior to Delfield's design effort was fully ramped up. Meetings and further discussion were held with Delfield's key Vantage line development staff to present the options for energy use reduction and to understand and address key concerns regarding their adoption. A sensible group of design options was chosen to incorporate into the Vantage two-door refrigerator design, which was the key model examined in the TIAX work. The design options chosen include the following.

- GE ECM (brushless DC) evaporator fans.
- Improved face frame design, similar to Concept 2 in Figure 3-2
- Reduced antisweat heater wattage
- Condensate line trap
- Optimized R-404A refrigeration system

Collaboration and discussions continued through Delfield's design effort to assure that the technology transfer process was successfully managed. Key development steps in the process included design of cabinet details, specification of cabinet and door hardware and selection of vendors, design and specification of tooling required for cabinet fabrication, initial specification of refrigeration system key components and subsequent iteration on refrigeration system design, fabrication of prototypes, testing, and transition to pilot and full production. TIAX's involvement in this process included the following.

- Investigation of the energy impact of the bottom-mount condensing unit configuration.
- Development of design concepts for the face frame
- Mock-up testing of some face frame concepts and providing assistance to Delfield testing of face frame concepts
- Investigation of reliability of the GE ECM fans, including long-term durability testing of ten units
- Key component recommendations for the refrigeration system
- Antisweat heater wattage recommendations
- Testing of the first prototype
- Testing of one of the pilot production units, including the first ASHRAE 117 energy test

TIAX also provided preliminary refrigeration system recommendations for other Vantage series units, including the single-door and three-door refrigerator, one-, two-, and three-door freezer, and the one- and two-door beverage merchandiser.

4.3 Cabinet and Face Frame Area Design

The face frame design for the Vantage series cabinets is shown in Figure 4-2 below. This is essentially Design Concept 2 from Figure 3-2 above, with the following modifications.

- The gasket could not be moved outwards as much as was initially desired, due to interference between the gasket and the door hinge hardware. A reasonable compromise was made.
- The method for fastening the face frame to the liner was improved.

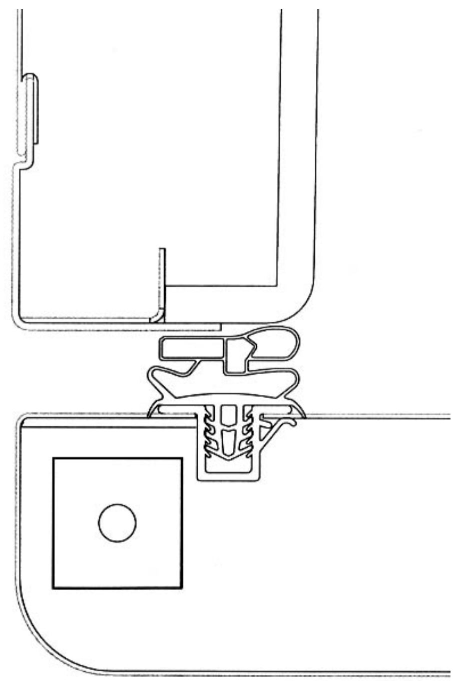


Figure 4-2: New Face Frame Design

The key challenges in finalizing the face frame design were as follows.

- Elimination of the additional bend of the face frame stainless steel towards the cabinet interior makes flatness of the face of the interior ABS liner important. The mold for the 6000 series ABS liner was not designed to provide tight tolerances in this area. The Vantage series liner mold design had to be adjusted to hold the required tolerance.
- Developing a lower-cost approach for fastening of the liner to the face frame was a key project goal. The elimination of the piece of the face frame which extends towards the cabinet interior provided an additional challenge, because this would have provided added structure prior to cabinet foaming. A number of design concept options were developed, and some of them were tested to determine feasibility. A clip approach is shown in Figure 4-3 as an example. The clip approach illustrated had some problems due to the tendency of the clip to push the ABS liner away from the sidewall, a problem which would not have been an issue with the old-style face frame design. A twist tab approach, shown in Figure 4-4, was used for the final design.
- The design of the face frame configuration for the mullion and for the overhang above the cabinet door, and transition between the different face frame regions provided additional design challenge.

- Development of the glass door units was complicated by the long lead time and additional cost which would have been required for development of custom extrusions for door frames. Selection of door frames was limited to the models available from the chosen vendor, Pike Machine Products. This resulted in differences in gasket alignment for solid and glass doors. The face frame was designed to be optimized for the solid door units, thus resulting in higher than desired thermal loss for the glass-door units. This problem may be addressed in future design changes.

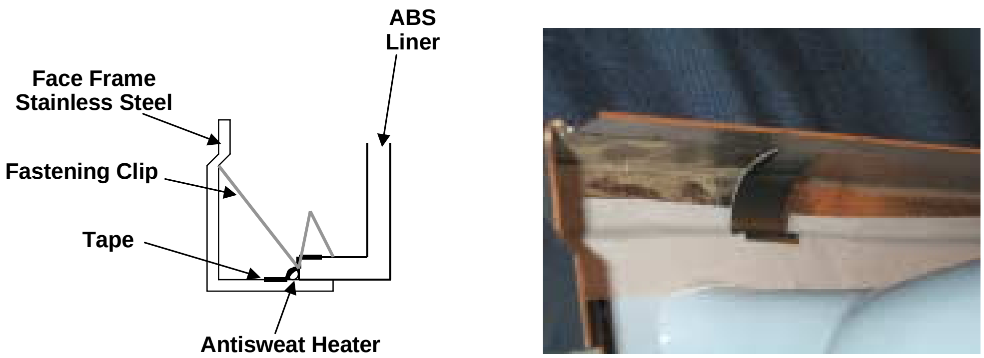


Figure 4-3: Clip Approach for Liner/Face Frame Fastening: Concept Sketch and Mockup Test

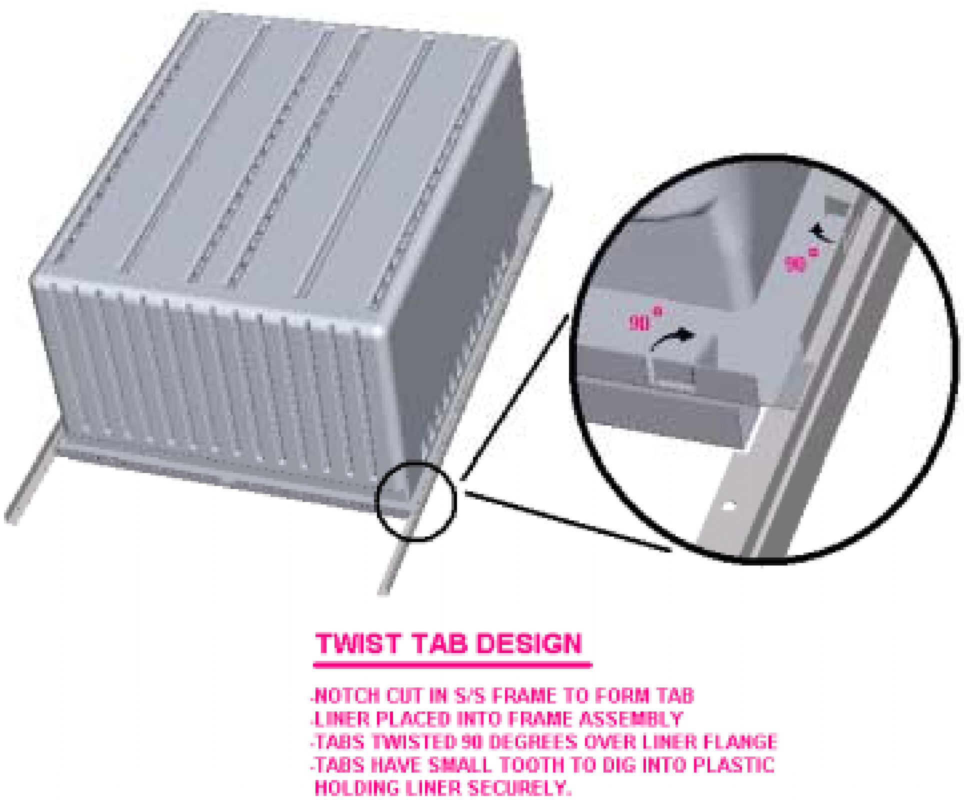


Figure 4-4: Twist Tab Approach for Liner/Face Frame Fastening

4.4 Investigation of Bottom-Mount Implications

Investigation including analysis and testing was done to assess the energy impact of the bottom-mount refrigeration system location for the Delfield Vantage series product design. A bottom-mount two-sliding-glass-door unit manufactured by a competitor was tested. The issues investigated and summaries of conclusions are as follows.

What is the added heat load of the warmer temperatures underneath the cabinet floor? Testing of the competitor's unit shows that the average temperature underneath the floor is roughly 20ºF higher than ambient. This would add 13 Btu/hr to the cabinet load of a two-door refrigerator and increase 100ºF closed-door energy use about 0.07 kWh, or 1.3% for a two-solid-door unit. While this is a contribution to energy use, it is not very large—certainly not large enough to abandon the bottom-mount configuration.

Does the warm condenser discharge air blanket the front door surfaces, thus increasing closed-door loads? Temperature measurements during testing with the doors closed shows no significant increase in air temperature just in front of the door and no significant increase in door surface temperature during the compressor on-cycle, indicating that there is no significant increase in door load resulting from warm condenser discharge air. Smoke flow visualization tests confirm that air flow from the grill moves more in the forward direction than upward. It is unlikely that the door load is elevated significantly due to the bottom-mount arrangement.

Does the warm condenser discharge air enter into the cabinet during door-opening, thus increasing load? Temperature measurements during door openings with the compressor both on and off show no difference in temperature excursions for air temperature just inside the door. Smoke flow visualization tests confirm that the cold air spilling from the cabinet pushes warm condenser air away from the unit. It is unlikely that condenser discharge air enters the cabinet to increase load.

Does warm condenser discharge air recirculate to the condenser air inlet, resulting in elevated condensing temperatures? Temperature measurements indicate that condenser air entering temperature is 10^{o}F higher than ambient for the initial bottom-mount unit configuration shown in Figure 4-5. This compares with negligible temperature elevation measured with the 6051, which has a top-mount condensing unit with an open top for good air circulation. A 10^{o}F rise in condenser air temperature will result in about 0.4 kWh or 7% increase in energy use for a two-solid-door refrigerator.

Are there differences in performance using a conventional louver grill as compared with the planned perforated plate grill? Delfield planned at one time to use a perforated plate grill for the Vantage series (0.09" holes separated 0.15" with ~33% open area). Testing with the competitive unit using its original louver grill and a perforated plate grill shows that there is no significant difference in condensing temperature elevation over ambient temperature between the two grill styles.

The main problem with the bottom-mount condensing unit arrangement is warmer average condenser inlet air, resulting from recirculation of condenser discharge air to the condenser air inlet. Additional testing was done to demonstrate that a baffle installed to prevent this recirculation would reduce condensing temperature and eliminate the problem. Figure 4-5 below shows the condensing unit layout for the original configuration and for an improved arrangement for which condenser discharge air is prevented by a baffle from increasing condenser inlet air temperature. The entire condensing unit was moved to the left side of the unit to avoid repiping of the unit which would have been required if the condensing unit had been rotated in its original location on the right side of the unit. The perforated plate grill was used to cover the entire front of the condensing unit opening for both of these configurations.

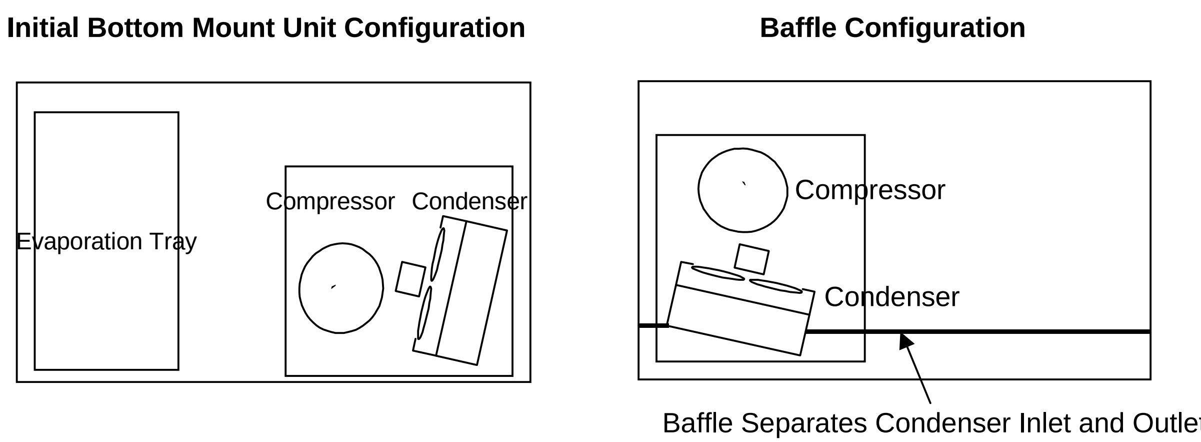


Figure 4-5: Bottom-Mount Condensing Unit Test Configurations

Results of the tests are presented in Table 4-1 below. The difference between condenser inlet air and ambient is reduced about 9 degrees for the test with the baffle, and the difference between condensing temperature and ambient is reduced about 5 degrees.

Table 4-1: Bottom-Mount Condensing Unit Test Data

Unit Configuration	Initial	With Baffle
Ambient	93.9	89.4
Condenser Inlet Air (Average[2])	103.5	90.4
Left Top	*100.5*	*89.9*
Right Top	*113.2*	*89.6*
Left Bottom	*107.3*	*92.9*
Right Bottom	*92.9*	*90.7*
Center		*89.3*
Condenser Mid[3]	115.8	106.8
Elevation above Ambient		
Condenser Air	9.6	1.0
Condenser Mid	22	17

Notes:
[1]Temperatures are averaged for the compressor on-cycle.
[2]Average for the first test includes only the four corner temperatures.
[3]Return bend temperature half-way through condenser circuit, representative of condensing temperature.

TIAX provided Delfield with a recommended approach to incorporating the condenser air baffle which would allow the condensing unit to be slid out for service. The baffle was not included in the units which have been tested to date. However, Delfield plans to implement a baffle with a future design change. The recommended and current Vantage series condensing unit design configurations are shown below in Figure 4-6. The bottom and rear of the condensing unit compartment are open, providing plenty of area for discharge air to exit, even if the unit is partially blocked by installation against a wall. Even though the unit does not have the recommended baffle, the condenser does face forward, and condenser air inlet temperature is only 2 to 3 ^{o}F above ambient temperature.

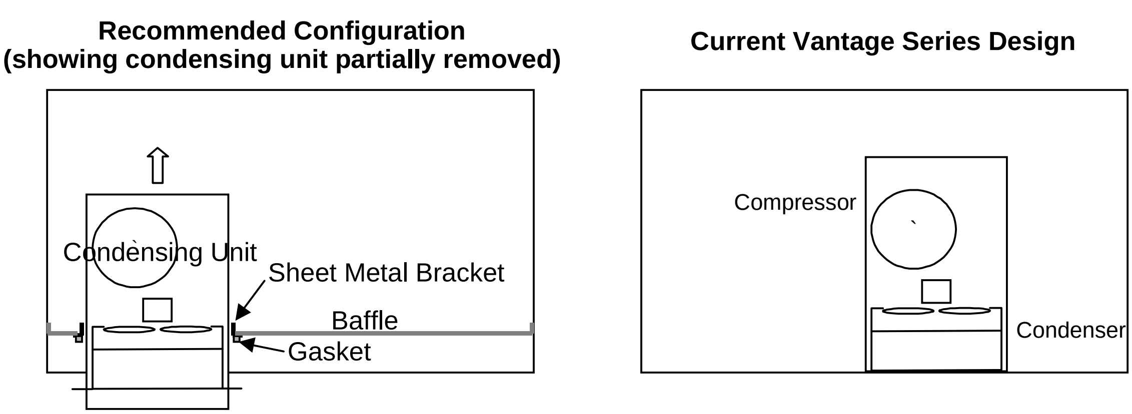


Figure 4-6: Recommended and Current Vantage Series Condensing Unit Design Configurations

4.5 Reliability Testing for GE ECM Fans

A group of ten GE 58 Series ECM fan samples was obtained for extended testing to help establish confidence in the fan's reliability. The samples represented three speeds spanning the fan's range from 950 to 1,900 rpm. The units were set up in the baseline 6051 refrigerator and set up to switch off for a minute once every hour. The fans ran for 6 months without a failure.

4.6 Refrigeration System Design

Recommendations for refrigeration system key components were prepared initially for the two-door refrigerator and also for most of the other models planned for the Vantage series. Recommendations for the other models is briefly summarized in Appendix A.

Key considerations in developing the design recommendations were as follows.

- A common refrigeration system design was desired for both solid-door and glass-door units, in spite of the large difference in cabinet loads of these units. This was desired to limit manufacturing costs associated with additional models and components, and to allow flexibility in product configuration in the distribution network.
- Based on preliminary analysis it was determined that a separate refrigeration system design would be required for a beverage merchandiser meeting typical bottlers' pulldown requirements. This is purely a pulldown performance consideration. However, the initial beverage merchandiser product offering did not use a different system, and does not meet all bottling companies' requirements.
- A balance of reliability and energy savings was applied in setting key design parameters such as superheat and suction temperature. Analysis was done to show the impact of selecting operating parameters more aggressive than Delfield was accustomed to. For instance, raising suction temperature 15^{o}F degrees by increasing the interchaner effectiveness would improve compressor efficiency by 2.6% but also increase discharge temperature 16^{o}F degrees, which would potentially lead to reduced compressor life. Also, decreasing evaporator exit superheat from 10^{o}F to 5^{o}F would improve efficiency 1% but may increase the chances of liquid slugging damaging the compressor in a worst-case compressor startup scenario. Final design was based on fairly conservative selection of these operating parameters.

The recommended key components for the two-door refrigerator are summarized in Table 4-2 below. Cost estimates for the components are compared with costs for equivalent components of the baseline 6051. Note that at this stage of the development, Delfield planned to fabricate condensing units rather than purchase completed condensing units. This approach allowed for flexibility in selection of the condenser and condenser fan which would be used with a given compressor. A wide cost range for condensing unit fabrication is presented, since this cost was not rigorously calculated and reviewed with Delfield. A preliminary estimate for this cost, based only on purchased component costs, was less than $50. The recommended refrigeration system was likely to be less expensive than that of the baseline 6051.

Table 4-2: Refrigeration System Design Recommendations for Two-Door Refrigerator

Component	Description	Cost	Cost for Equivalent Components of 6051
Compressor	Tecumseh AEA9422ZXA Medium Temperature Motor: 115V, 0.25hp, CSIR	~$70	Condensing Unit $202
Condenser	Face 13" by 10" inches 0.375" OD tubes, smooth Tube rows 10 high, 4 deep Fins Wavy, 6.5 FPI 289 cfm	$24	
Condenser Fan	Morrill SP-B9HUEM1 and FV875CW23538 6W output shaded pole 8.75-inch blade	$16.50	
Condensing Unit Fabrication	Sheet Metal, Fan Guard, Wiring, Piping, Labor	$50 to $100	
Evaporator	Face 24 by 7 inches 0.375"OD tubes, smooth Tube rows 7 high, 3 deep Fins Wavy, 8 FPI 275 cfm	$27	$31
Evaporator Fan	GE 58 Series ECM	$28	2 x ($13 + $1) 2 x (Motor + Blade/Bracket)
Total		**~$250**	**$261**

Performance predictions for the recommended refrigeration system are presented in Table 4-3 below. As can be seen, the maximum 70% run time which is required for the NSF 7 test is achieved for both solid- and glass-door units. The energy use for the two-solid-door unit was predicted to be reduced to 5.2 kWh, a 57% reduction, for NSF 7 test conditions.

Table 4-3: Performance Prediction for Recommend Two-Door-Refrigerator Refrigeration System

	Refrigeration Capacity (Btu/hr)	Cabinet Load (Btu/hr)	Percent Run Time (%)	24-Hour Energy Use (kWh)
Two-Solid-Door	2,478	637	26%	5.2
Four-Glass-Door	2,603	1,291	50%	9.7

Notes:
1. Performance is predicted for NSF 7 operating conditions (100°F ambient for solid door, 86°F ambient for glass door, doors closed)
2. Four-glass-door unit has four half-height doors. Estimates for this unit assume old-style face frame characteristics, since the glass door gasket will not seal the face frame from penetrating into the cabinet interior for initial production runs.

In addition to the refrigeration system components, a recommendation was also made for the antisweat heater wattage for the initial prototype. The initial recommendation was made conservatively for 50W total, exactly half of the baseline wattage. Sweat testing was done with the fabricated units to further optimize this wattage.

During the course of the project's transition to production, several changes were made in the refrigeration system key component selections. The key changes were as follows.

- A larger evaporator was used. This evaporator was also selected for use in Delfield's Meridian line of reach-ins, whose development preceded that of the Vantage line by a few months. Dual use of the evaporator was done to reduce costs across the lines of products.
- The evaporator fan was set up for operation at lowest speed, since the large evaporator made operation at higher speed unnecessary. This resulted in a very slight reduction of evaporator fan input wattage.
- A ¼-hp Copeland compressor was used for the initial prototype build, since availability of the recommended Tecumseh compressor was limited. Subsequently, a Copeland condensing unit including this new compressor was specified, since Copeland offered to build the condensing unit to meet the efficiency criteria for the project at a competitive cost.

The final refrigeration system key components are compared with those of the baseline 6051 in Table 4-4 below.

Table 4-4: Comparison of Final and Baseline Refrigeration System Components

Component	Baseline Refrigerator (6051)	Vantage Design
Compressor	Copeland JRS4-0050-IAA	Copeland ASE19-C3E-IAA
Condenser	Face 9" x 9" (229mm x 229mm) Tube rows 9 high x 3 deep Fins wavy, 6.5 FPI	Face 9" x 9" (229mm x 229mm) Tube rows 9 high x 5 deep Fins wavy, 8 FPI
Condenser Fan	7.5-inch (191mm) diameter blade 6 W SP Motor, 35 W input	7.5-inch (191mm) diameter blade 9W SP Motor, 50 W input
Evaporator	Face 21.5" x 7" (546 mm x 178 mm) Tube rows 7 high x 4 deep Fins wavy, 8 FPI	Face 21.5" x 8" (546 mm x 203 mm) Tube rows 8 high x 6 deep Fins wavy, 8 FPI
Evaporator Fan	Two fans 6-inch (152 mm) diameter blade 6W SP Motor, 34W input	GE 58 Series ECM, 950 rpm

4.7 Testing of Prototype and Production Unit

The following testing was done on the initial prototype.

- Sweat testing to set antisweat heater wattage. The wattage was set at 50W. Testing was done in a 100°F ambient and 65% RH, so this test also provided an indication of NSF 7 performance.

- ASHRAE 117 Energy Test.

The following testing was done on the pilot production unit.

- NSF 7 Testing to optimize the refrigeration system. Two adjustments recommended to increase evaporator temperature were increase of capillary bore and increase of refrigerant charge. These adjustments were implemented by Delfield.
- Sweat testing to optimize antisweat heater wattage.
- ASHRAE 117 energy test.
- 80ºF ambient closed-door test.
- 100ºF ambient 65% RH door-opening test.

Sweat testing done to optimize the antisweat heater wattage is summarized in Table 4-5 below. It was recognized quickly during this testing that the mullion needs more heat than the unit's perimeter. Delfield has decided to use a heater system which allows separate specification of the wattages for these locations. The testing at TIAX involved use of the existing internal door-loop heaters, with an additional heater placed externally on the mullion. While antisweat performance was not perfect, even with the final heater recommendation, the problem areas were due to alignment difficulties for the door and gasket. In many locations, the face frame metal is not properly sealed from exposure to the cabinet interior. Delfield chose to solve the alignment problems rather than overspecify antisweat heat. Delfield has also started to pursue a modification in the gasket profile to improve the inner bulge of the gasket to provide a better seal.

Table 4-5: Sweat Testing for Antisweat Heater Optimization

Antisweat Heater Wattages (W)			Observation
Doors[1]	Added on Mullion	Total on Mullion[2]	
25	0	16.7	No Face Sweat
20	0	13.3	No Face Sweat
15	0	10	Mist on Mullion
15	3.3	13.3	Sweat Lower 1/6 of Mullion
12	7	15	No Face Sweat
9	8.5	14.5	No Face Sweat
7	9	13.6	No Face Sweat
4	9	10.3	Some Mullion Sweat
0	14	14	Perimeter droplets top right
5	10	13.3	Some moisture on top right perimeter. No other evidence of face frame moisture.
5	12	15.3	Slight sweat on perimeter top right
7	11	15.7	Slight sweat on perimeter top right
9	9.7	15.7	Acceptable

[1]Two internal heaters. Each heater was set for the wattage listed. [2]Includes power in door heaters and external mullion heater

In order to reduce overall factory inventory requirements, Delfield has opted to use the same antisweat heater arrangement which is used on the freezer and cycle the antisweat heat with the compressor. This approach actually reduces average antisweat heater wattage for an energy test. The total antisweat heater wattage is 65W. For ASHRAE 117 conditions, this would give an average total wattage of only 11W. For NSF 7 test conditions, average wattage would be 22W.

The ASHRAE 117 Energy Tests for the Baseline 6051 Reach-In Refrigerator and for the two tested Delfield units are summarized in Table 4-6 below. Clearly the production unit achieved significant energy savings as compared with the baseline 6051. The cabinet construction of the first prototype was not up to production standards, which resulted in higher than desired antisweat heater wattage selection for this unit, and also higher energy.

Table 4-6: Energy Test Results

	6051	First Prototype	Final Unit
Date Tested	1/29/01	9/6/01	1/4/02
Thermostat Setting	1.5		4
Refrigerant Type	HCFC-22	R-404A	R-404A
Quantity (ounces)	13		19.5
Ambient Temperatures (°F)			
Dry Bulb	76	73.5	75
Wet Bulb	64	63	64
Test Package Temperatures (°F)			
Integrated Average (IAT)	36.2	36.7	38.7
Coldest Package Avg. (CTPA)	32.9	34.5	37.4
Warmest Package Avg. (WTPA)	39.5	39.2	39.9
Maximum Warmest Package	40.4	40.0	40.4
Avg. Temperature Start	36.9	37.6	39.3
Avg. Temperature End	37.2	37.3	39.3
Total Energy Input (24-hours, kWh)	8.98	4.03	2.86
Percent Savings (%)		55%	68%
Percent Compressor Run Time			
Overall	36.3%	22.5%	16.5%
During Door-Openings	56.6%	35.0%	25.9%
Antisweat Heater Wattages (W)[1]			
Perimeter	65	32.5	12.2
Mullion	35	17.5	15.6
Total	100	50	27.8

[1]Note that this wattage is included in the energy use.

The energy test results for the Vantage line one- and two-solid-door refrigerators are compared in Figure 4-7 with test results for competitive units. The test results for refrigerators meeting Energy Star standards as of February 2002 are included in the comparison. The one-door refrigerator energy use is among the best available. The two-door refrigerator energy use is the lowest. None of the refrigerators offered by

Delfield's key competitors (Traulsen, BeverageAir, True, TurboAir, etc.) have energy use matching that of either of these units.

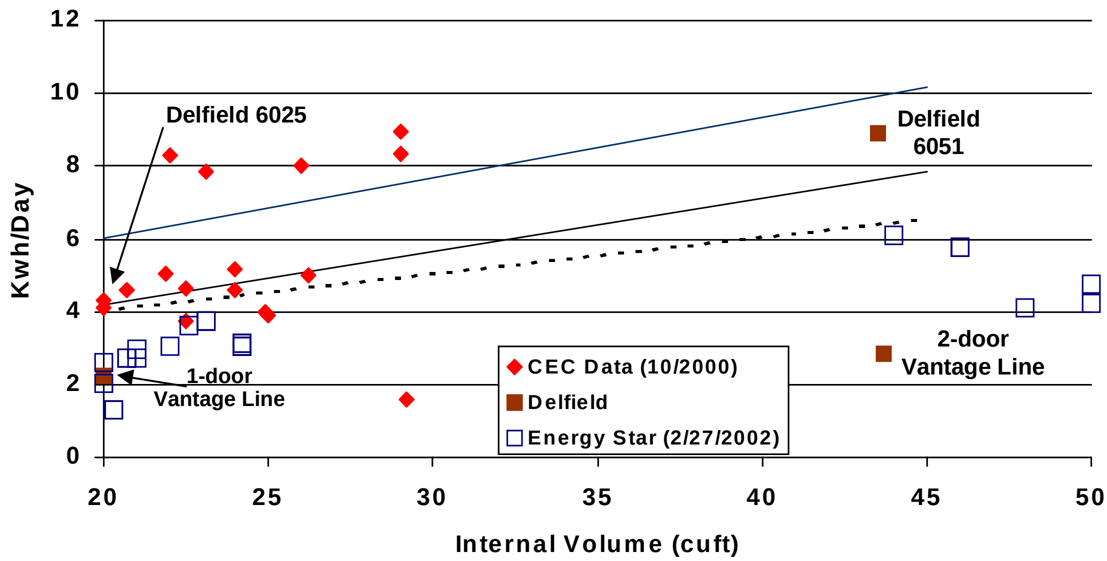


Figure 4-7: ASHRAE 117 Energy Test Comparison with Competitive Units

Performance test results are summarized in Table 4-7 below. The ASHRAE 117 Test results are also shown to allow comparison of run times and energy use for the different test conditions. The production unit had significantly lower run times and energy use than the baseline 6051. The energy use for the 100°F closed door test matched our 5.2 kWh prediction very well, even though compressor run time was higher. This was because the antisweat heater wattage for the tested unit was lower than the initial conservative prediction, which made up for additional compressor run time. In extreme-condition testing with door openings in a 100°F 65% ambient, the recovery temperature for the new unit is remarkably better than for the baseline unit. Finally, the results show that the 80°F closed-door test gives a reasonable prediction of ASHRAE 117 test energy.

Table 4-7: Performance Test Results

	6051	Final Unit
100°F Closed-Door Test		
Date	12/11/00	12/18/01
Percent Run	64%	34%
24-Hour Energy Use (kWh)	12.1	5.1
80°F Closed-Door Test		
Date	12/14/00	1/7/02
Percent Run	46%	15.5%
24-Hour Energy Use (kWh)	9.1	2.77
Date	12/19/00	
Percent Run	47%	
24-Hour Energy Use (kWh)	9.31	
ASHRAE 117 Energy Tests		
Date	1/29/01	1/4/02
Percent Run	36.3%	16.5%
24-Hour Energy Use (kWh)	8.98	2.86
100°F 65%RH Door-Opening Test		
Date	12/21/00	1/11/02
Typical Recovery Temperature[1]	53°F	28°F

[1]Typical temperature to which cabinet thermocouples recover prior to the next door opening.

5. Conclusions

A high-efficiency commercial reach-in refrigerator has been developed and commercialized which uses significantly less energy than the unit which it replaces. The key design changes responsible for energy use reduction are a brushless DC evaporator fan, redesign of the face frame area for reduced heat loss, reduction of antisweat heater wattage, and use of an optimized refrigeration system utilizing R-404A refrigerant. The cost impact of the energy saving design features is negligible. The unit is currently in production as part of the Vantage series by The Delfield Company, who collaborated with us on the design effort. The cabinet design developed as part of this project serves as the platform for a full line of efficient reach-ins including freezers and beverage merchandisers which are available in one-, two-, and three-door sizes.

The key success criteria for the project, summarized in Table 1-3 in Section 1.5, were exceeded by a large margin. The success energy use reduction of 33% was doubled by the actual reduction of 68% in ASHRAE 117 testing. The success cost premium of $150 allowance was unnecessary since there was no cost increase. Refrigerating performance of the final design was also superior to that of the baseline unit.

A number of additional opportunities for energy savings besides those incorporated in the Delfield design have been identified during the course of this work. Some of these are more expensive than the options chosen, but others could be pursued cost-effectively. Some of these require additional work to fully understand their cost/benefit ratio. The list of options which could warrant additional investigation by manufacturers are:

- Higher-efficiency HFC-134a compressors.
- Variable-speed system operation.
- Use of the GE 58 Series ECM fan with two speeds, either with a variable speed compressor system or without it. This option could be used for performance enhancement (i.e. rapid pulldown capability) without sacrificing energy savings.
- High-efficiency condenser fan motors.
- Hot gas antisweat. This option starts to become feasible for the new cabinet design, for which the antisweat heating load is significantly reduced.
- Controlled termination of off-cycle defrost.
- Investigation of glass door options to assess cost/benefit ratio for different energy-saving options such as low-emissivity glass, argon fill, triple glazing, etc.

Additional work to address energy savings would also be appropriate for freezers. The energy use of freezers is significantly higher than that of refrigerators. While some analysis was done to support freezer system component selection for the Vantage line, the primary focus of this project was the two-door refrigerator. The key area offering additional energy savings potential in freezers is the defrost energy requirement.

6. References

1. “Energy Savings Potential for Commercial Refrigeration Equipment”, prepared by Arthur D. Little for the U.S.DOE Office of Building Technologies, June 1996

2. “Cold Storage Temperature Stabilization Project Final Report”, prepared by Arthur D. Little for the U.S.Army Soldier Systems Center, Natick, MA, July 12, 2000

3. ANSI/NSF Standard 7-99, “Commercial Refrigerators and Freezers”, National Sanitation Foundation International, 1999

4. ANSI/ASHRAE Standard 117-1992, “Method of Testing Closed Refrigerators”, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, 1992

Appendix A: Refrigeration System Design Recommendations Summaries

This appendix provides summaries of refrigeration system design recommendations for the following reach-in product configurations.

- 1-door refrigerator
- 2-door freezer
- 1-door freezer
- 3-door freezer
- 3-door refrigerator
- 1-door beverage merchandiser
- 2-door beverage merchandiser

	Two-Door Freezer			One-Door Freezer
	Option 1	Option 2	Option 3	
Compressor				
Model[1]	AJA2425ZXA	AJB2432ZXA	AJB2432ZXA	AJA2419ZXA
Capacity[2]	1917	2886	2886	1559
EER[2]	2.93	2.96	2.96	2.87
Motor hp	0.6hp	0.7hp	0.7hp	0.5hp
Type, Voltage	CSR, 115V	CSR, 115V	CSR, 115V	CSR, 115V
Condenser				
Core Size (in)	26x10x3.4	20x10x3.4	26x10x3.4	13x10x2.25
Tube Type	Smooth 3/8”	Smooth 3/8”	Smooth 3/8”	Smooth 3/8”
Tube Rows	10 x 4	10 x 4	10 x 4	10 x 3
Fins	Wavy, 6.5FPI	Wavy, 6.5FPI	Wavy, 6.5FPI	Wavy, 6.5FPI
CFM	350	400	315	250
Condenser Fan				
Shaft Power, Type	16W SP	16W SP	9W SP	6W SP
Evaporator				
Core Size (in)	24x7x4.3	24x7x3.4	24x7x3.4	17x7x2.6
Tube Type	Rifled 3/8”	Rifled 3/8”	Rifled 3/8”	Rifled 3/8”
Tube Rows	7 x 5	7 x 4	7 x 4	7 x 3
Fins	Wavy, 6FPI	Wavy, 6FPI	Wavy, 6FPI	Wavy, 6FPI
CFM	450	450	450	200
Evaporator Fan	2 GE 58 Series ECM	2 GE 58 Series ECM	2 GE 58 Series ECM	GE 58 Series ECM
Cost Estimate				
Compressor	$120	$120	$120	$120
Condenser	$36	$30	$36	$19
Cond Fan	$25	$25	$20	$16.50
Evaporator	$38	$32	$32	$22
Evap Fan	$56	$56	$56	$28
TOTAL	**$275**	**$263**	**$264**	**$206**
Solid Door Performance[3]				
Load (Btu/hr)	1000	1000		660
Capacity (Btu/hr)	2467	2874		1567
Run Time	41%	35%		42%
Comp/Cond EER[5]	3.3	2.95		2.9
Energy Use (kWh)	9.7	10.5		6.6
Glass Door Performance[4]				
Load (Btu/hr)	1800	1800	1800	1054
Capacity (Btu/hr)	2884	3025	2909	1791
Run Time	62%	60%	62%	59%
Comp/Cond EER[5]	3.85	3.3	3.3	3.3
Energy Use (kWh)	15.6	17.4	17.5	9.7

[1]Tecumseh model numbers.
[2]Compressor nominal capacity and EER based on 20°F Evaporating, 130°F Condensing, 65°F Suction, and 0°F Subcooling conditions for Refrigerators; -15°F Evaporating, 120°F Condensing, 40°F Suction, and 0°F Subcooling conditions for Freezers.
[3]Solid door performance is for a unit with full-height doors in a 100°F ambient.
[4]Glass door performance is for a unit with half-height doors in a 86°F ambient.
[5]Capacity divided by power input of compressor and condenser fan.

	One-Door Refrigerator	Three-Door Refrigerator	
		Two-Door Side	One-Door Side
Compressor			
Model[1]	AJA9415ZXA	AKA9427ZXA	
Capacity[2]		2566	
EER[2]		4.67	
Motor hp	0.2hp	0.33hp	
Type, Voltage	CSR, 115V	CSR, 115V	
Condenser			
Core Size (in)	13x10x2.25	20x10x3.4	
Tube Type	Smooth 3/8"	Smooth 3/8"	
Tube Rows	10 x 3	10 x 4	
Fins	Wavy, 6.5FPI	Wavy, 6.5FPI	
CFM	250	400	
Condenser Fan			
Shaft Power, Type	6W SP	6W SP	
Evaporator			
Core Size (in)	17x7x2.6	24x7x2.6	17x7x2.6
Tube Type	Smooth 3/8"	Smooth 3/8"	Smooth 3/8"
Tube Rows	7 x 3	7 x 3	7 x 3
Fins	Wavy, 8FPI	Wavy, 8FPI	Wavy, 8FPI
CFM	185	275	185
Evaporator Fan	1 GE 58 Series ECM	1 GE 58 Series ECM	1 GE 58 Series ECM
Cost Estimate			
Compressor	$60	$90	
Condenser	$19	$24	
Cond Fan	$16.50	$16.50	
Evaporator	$19	$27 + $19	
Evap Fan	$28	$56	
Additional		$55[6]	
TOTAL	**$143**	**$288**	
Solid Door Performance[3]			
Load (Btu/hr)	463	1071	
Capacity (Btu/hr)	1738	2519	
Run Time	27%	43%	
Comp/Cond EER[5]	4.2	4.3	
Energy Use (kWh)	3.0	8.7	
Glass Door Performance[4]			
Load (Btu/hr)	782	2050	
Capacity (Btu/hr)	1931	3097	
Run Time	40%	66%	
Comp/Cond EER[5]	5.0	5.1	
Energy Use (kWh)	5.8	15.6	

[1]Tecumseh model numbers.
[2]Compressor nominal capacity and EER based on 20°F Evaporating, 130°F Condensing, 65°F Suction, and 0°F Subcooling conditions.
[3]Solid door performance is for a unit with full-height doors in a 100°F ambient.
[4]Glass door performance is for a unit with half-height doors in a 86°F ambient.
[5]Capacity divided by power input of compressor and condenser fan.
[6]Two Solenoid Valves, Two Thermostatic Expansion Valves, and a Receiver

	Three-Door Freezer: System 1		Three-Door Freezer: System 2	
	Two-Door Side	Two-Door Side	One-Door Side	One-Door Side
Compressor				
Model[1]	AWA2450ZXD		AWA2460ZXD	
Capacity[2]	3642		5103	
EER[2]	3.10		3.20	
Motor hp	1.2hp		1.6hp	
Type, Voltage	CSR, 230V		CSR, 230V	
Condenser				
Core Size (in)	50x10x3.4		26x10x3.4	
Tube Type	Smooth 3/8”		Smooth 3/8”	
Tube Rows	10 x 4		10 x 4	
Fins	Wavy, 6.5FPI		Wavy, 6.5FPI	
CFM	800		400	
Condenser Fan				
Shaft Power, Type	2 16W SP		1 16W SP	
Evaporator				
Core Size (in)	24x7x3.4	17x7x2.6	24x7x3.4	17x7x2.6
Tube Type	Rifled 3/8”	Rifled 3/8”	Rifled 3/8”	Rifled 3/8”
Tube Rows	7 x 4	7 x 3	7 x 4	7 x 3
Fins	Wavy, 6FPI	Wavy, 6FPI	Wavy, 6FPI	Wavy, 6FPI
CFM	450	200	450	200
Evaporator Fan	2 GE 58 Series ECM	1 GE 58 Series ECM	2 GE 58 Series ECM	1 GE 58 Series ECM
Cost Estimate				
Compressor	$150		$170	
Condenser	$58		$36	
Cond Fan	$40		$20	
Evaporator	$32 + $22		$32 + $22	
Evap Fan	$84		$84	
Additional	$55[6]		$55[6]	
TOTAL	**$441**		**$419**	
Solid Door Performance[3]				
Load (Btu/hr)	1561		1561	
Capacity (Btu/hr)	3399		3534	
Run Time	46%		44%	
Comp/Cond EER[5]	2.98		2.75	
Energy Use (kWh)	16.1		17.0	
Glass Door Performance[4]				
Load (Btu/hr)	2763		2763	
Capacity (Btu/hr)	4107		4204	
Run Time	67%		66%	
Comp/Cond EER[5]	3.53		3.21	
Energy Use (kWh)	27.6		29.4	

[1]Tecumseh model numbers.
[2]Compressor nominal capacity and EER based on -15°F Evaporating, 120°F Condensing, 40°F Suction, and 0°F Subcooling conditions.
[3]Solid door performance is for a unit with full-height doors in a 100°F ambient.
[4]Glass door performance is for a unit with half-height doors in a 86°F ambient.
[5]Capacity divided by power input of compressor and condenser fan.
[6]Two Solenoid Valves, Two Thermostatic Expansion Valves, and a Receiver

	One-Door Beverage Merchandiser	Two-Door Beverage Merchandiser
Compressor		
Model[1]	AEA9422ZXA	AKA9438ZXA
Capacity[2]	2250	3750
EER[2]	4.13	4.69
Motor hp	0.25hp	0.5hp
Type, Voltage	CSIR, 115V	CSIR, 115V
Condenser		
Core Size (in)	13x10x2.25	20x10x3.4
Tube Type	Smooth 3/8"	Smooth 3/8"
Tube Rows	10 x 3	10 x 4
Fins	Wavy, 6.5FPI	Wavy, 6.5FPI
CFM	250	400
Condenser Fan		
Shaft Power, Type	6W SP	16W SP
Evaporator		
Core Size (in)	17x7x2.6	24x7x2.6
Tube Type	Smooth 3/8"	Smooth 3/8"
Tube Rows	7 x 3	7 x 3
Fins	Wavy, 8FPI	Wavy, 8FPI
CFM	200	450
Evaporator Fan	1 GE 58 Series ECM	2 GE 58 Series ECM
Cost Estimate		
Compressor	$70	$95
Condenser	$19	$30
Cond Fan	$16.50	$20
Evaporator	$19	$27
Evap Fan	$28	$56
Additional	$25[6]	$25[6]
TOTAL	**$178**	**$253**
Pulldown Time[3]		
Number of 12-oz cans	700	1,400
Hours	17	18
Steady State Performance[4]		
Load (Btu/hr)	841	1401
Capacity (Btu/hr)	2336	4122
Run Time	36%	34%
Energy Use (kWh)	6.6	10.5

[1]Tecumseh model numbers.
[2]Compressor nominal capacity and EER based on 20°F Evaporating, 130°F Condensing, 65°F Suction, and 0°F Subcooling.
[3]Pulldown from 90°F to 38°F in 90°F ambient.
[4]In a 90°F ambient.
[5]Capacity divided by power input of compressor and condenser fan.
[6]Thermostatic Expansion Valve and Receiver