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Application Of Best Industry Practices To The Design Of Efficient Commercial Refrigerators

D. Westphalen
Arthur D. Little

APPLICATION OF BEST INDUSTRY PRACTICES TO THE DESIGN OF EFFICIENT COMMERCIAL REFRIGERATORS

Detlef Westphalen, PhD, Senior Manager, Arthur D. Little, 20 Acorn Park,
Cambridge, MA 02140, USA; Tel.: 617-498-5821; Fax: 617-498-7213
E-Mail: Westphalen.D@ADLittle.com

ABSTRACT

A two-door commercial refrigerator was developed which consumes about one-third the energy of the model it replaced, while providing better refrigerating performance and without incurring a cost premium. The development was done in collaboration with the manufacturer, the Delfield Company. A baseline refrigerator was tested to measure energy use and performance attributes. A performance model was developed for evaluation of design options to reduce energy. A range of design options representing both cabinet and refrigeration system improvements were evaluated for both energy use and cost. An optimized design configuration was selected incorporating the most cost-effective of the options. The cabinet design serves as the platform for additional refrigerator and freezer models, with sizes from one-door to three-door with both solid and glass door options.

NOMENCLATURE

ABS	Acrylonitrile Butadiene-Styrene	NSF	National Sanitation Foundation
ASHRAE	American Society of Heating, Refrigeration	PSC	Permanent Split Capacitor
HCFC	Hydro-chloro-fluorocarbon	HFC	Hydro-fluorocarbon

INTRODUCTION

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 equipment.

The project described in this paper was a collaboration involving Arthur D. Little, 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 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).

BASELINE REFRIGERATOR INVESTIGATION

The baseline Model 6051 refrigerator was a typical two-door reach-in refrigerator of Delfield's 6000 series, shown in Figure 1 below. It has a stainless steel exterior and an ABS one-piece internal liner, with an insulating wall thickness of two inches (51 mm). The refrigeration system is close-coupled as a single unit, allowing fabrication on a benchtop with subsequent assembly on top of the unit. The evaporator side is enclosed with a separate insulated box, and the roof of the main cabinet has an aperture to allow evaporator air flow up to the evaporator and back down to the cabinet in the rear of the unit. A façade hides the refrigeration system, but otherwise the condensing unit is exposed to ambient air. The stainless steel exterior wraps around the entire face frame, which provides good support for the cabinet prior to foaming in the insulation, but represents a large heat leak path past the door gasket. The baseline unit's refrigerant system uses HCFC-22 refrigerant.

Baseline 6051

New Design

Figure 1: Baseline Two-Door Model 6051 Reach-In Refrigerator and New Design

The baseline unit was evaluated to understand its performance characteristics, design details, and cost. Testing of the baseline unit and the advanced design prototypes was done primarily in two environmental chambers at Arthur D. Little which allow for both ambient temperature and ambient humidity control. A computer-based data acquisition system running LabView™ and National Instruments data acquisition hardware was used for both measurement and for control of door-openings for some of the tests. Temperatures, including both air temperatures and surface temperatures, were measured with Type T thermocouples. Pressures were measured with a Heise electronic pressure gauge with data export capabilities as well as conventional manifold gauge sets. Ambient humidity measurements were made with an electronic temperature/humidity sensor as well as a digital hygrometer and a sling psychrometer. Testing of the baseline unit, done to evaluate performance, energy use, and cabinet load, and to support refrigerator modeling work, included the following.

1. National Sanitation Foundation (NSF) 7 Capacity Test: This is a test in 100F (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 1], while keeping internal temperatures 40F (4.4C) or lower. The baseline unit percent run time was 64%, and its 24-hour energy use for this test condition was 12.1 kWh.
2. Closed Door Test in 80F (26.7C) ambient: This test was used as an indicator of energy use in more moderate ambient conditions. Temperature was controlled as for the NSF 7 test to be 40F (4.4C) or lower. The 24-hour energy use for this test was 9.3 kWh.
3. Door-Opening Test in 100F (37.8C) 65% RH ambient: This test is used to verify system performance in extreme conditions. The baseline unit was not able to maintain temperature in the normal food storage range below 40F (4.4C). The temperature recovered between door openings to 53F (11.7C).
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 2]. The ambient conditions of 75F (23.9C) and 55% RH are moderate for typical commercial kitchen environments for reach-in refrigerators. The daily energy use of the baseline unit was 9kWh.
5. Reverse Heat Leak: This test was done with the environmental chamber at 35F (1.7C) and the cabinet interior heated with a measured wattage to determine cabinet heat transfer characteristics. Converted to heat leak for the NSF7 test conditions, cabinet load was 621 Btu/hr (182 W).
6. 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 baseline cabinet's infiltration rate was 1.0 cfm (28 lpm). Subsequent testing showed the impacts of making improvements in the unit air leakage.

7. Thermal Conductivity Measurement: The conductivity of the baseline unit's insulation was measured using a Holometrix conductivity tester. The conductivity of the main cabinet insulation of the baseline unit was about 0.15 Btu-in/hr-sqft-F (0.022 W/mK) at 70F (21C). Test results for the Baseline 6051 are compared in Figure 2 below with measurements made of insulation of a residential refrigerator. This comparison is of interest because both insulations were made with HCFC-22 as a 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.
8. Sweat Testing to evaluate reduced antisweat heat: Testing was done to determine overspecification of the baseline refrigerator's antisweat heater wattages. This was a closed-door test in a 100F (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.
9. Impact on NSF 7 Test of antisweat heater shutoff. A test was done to assess the cabinet load impact of the antisweat heaters. The percent run time of the compressor was reduced from 64% to 54% when the heaters were turned off.

Tests 1 through 4 were used to establish a performance baseline for the development work. Table 1 below shows results for Tests 1 through 3 for both the baseline unit and the final design, while Table 2 compares energy test results of the baseline unit, a first prototype and the final design.

Table 1: Test Results Summary for Baseline and New-Design Refrigerators

	Baseline Refrigerator (6051)	**New-Design Production Unit**
NSF 7: 100F (37.8C) Closed-Door Test		
Percent Run	64%	34%
24-Hour Energy Use (kWh)	12.1	5.1
80F (26.7C) Closed-Door Test		
Percent Run	47%	15.5%
24-Hour Energy Use (kWh)	9.31	2.77
100F 65%RH Door-Opening Test		
Typical Recovery Temperature[1]	53F (11.7C)	28F (-2C)

Table 2: ASHRAE 117 Energy Test Result Summary for Baseline and New-Design Refrigerators

	Baseline Refrigerator (6051)	**New-Design First Prototype**	**New-Design Production Unit**
Refrigerant Type	HCFC-22	R-404A	R-404A
Quantity—ounces (mg)	13 (369)		19.5 (554)
Ambient Temperatures °F(C)			
Dry Bulb	76 (24.4)	73.5 (23.1)	75 (23.9)
Wet Bulb	64 (17.8)	63 (17.2)	64 (17.8)
Test Package Temperatures (°F)			
Integrated Average (IAT)	36.2 (2.3)	36.7 (2.6)	38.7 (3.7)
Maximum Warmest Package	40.4 (4.7)	40.0 (4.4)	40.4 (4.7)
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

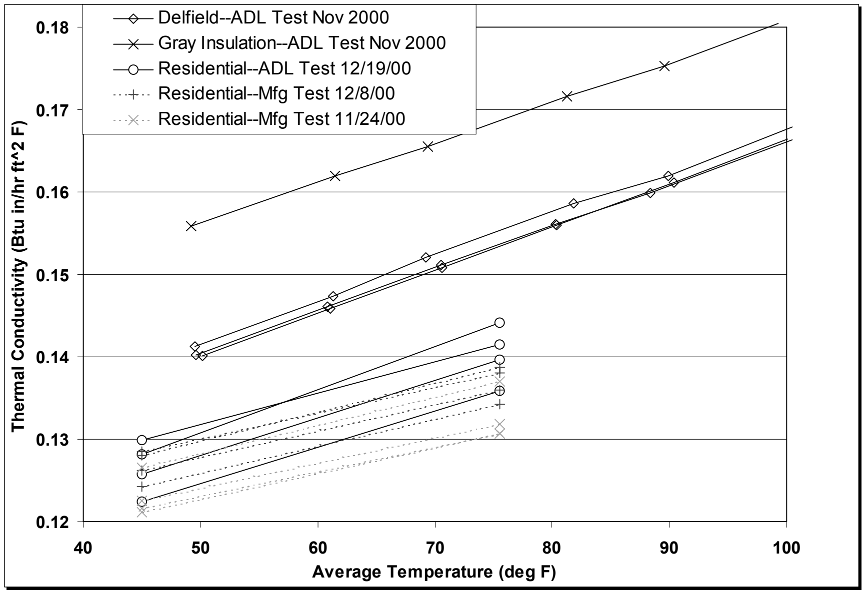

Figure 2: Thermal Conductivity Measurements

Notes:

1 “Delfield” Insulation was taken from the Baseline 6051
2 “Gray Insulation”, purchased by Delfield, is used for the 6051 evaporator housing.
3 “Residential” insulation was taken from cabinets of a residential refrigerator manufacturer.

A model of the baseline refrigerator’s cabinet loads and refrigeration system performance was developed to serve as the basis for projections of energy savings for design modifications. These models utilized ADL’s internally developed refrigerator modeling programs, which are adapted from the EPA Refrigerator Analysis (ERA) program, developed for residential refrigerator analysis for the EPA in the 1990’s. Finite Element Analysis (FEA) of heat leak through the face frame area was done to support cabinet load estimates and to assist development of alternative face frame designs. The cabinet load and energy use models for the baseline refrigerator are summarized in Table 3 below. While the largest contribution to cabinet load represents conduction through the walls and door, the other load contributions more than double the load.

Table 3: Baseline Refrigerator Cabinet Load and Energy Use Summary for NSF7 Test Conditions

<table>
<tr><th rowspan="8">Cabinet Load Summary</th><th>Component</th><th colspan="3">Load (Btu/hr)</th></tr>
<tr><td>Cabinet and Door</td><td colspan="3">422</td></tr>
<tr><td>Gasket and Face Frame System</td><td colspan="3">137</td></tr>
<tr><td>Infiltration (0.2 scfm)</td><td colspan="3">68</td></tr>
<tr><td>Antisweat Heaters</td><td colspan="3">113</td></tr>
<tr><td>Off-Cycle Charge Migration Loss</td><td colspan="3">100</td></tr>
<tr><td>Evaporator Fan Heat</td><td colspan="3">232</td></tr>
<tr><td>TOTAL</td><td colspan="3">1,071</td></tr>
<tr><th rowspan="6">Energy Use Summary</th><th>Component</th><th>Input Power (W)</th><th>Duty Cycle (%)</th><th>Daily Consumption (kWh/day)</th></tr>
<tr><td>Compressor</td><td>484</td><td>64%</td><td>7.4</td></tr>
<tr><td>Condenser Fan</td><td>35</td><td>64%</td><td>0.5</td></tr>
<tr><td>Evaporator Fans</td><td>68</td><td>100%</td><td>1.6</td></tr>
<tr><td>Antisweat Heaters</td><td>100</td><td>100%</td><td>2.4</td></tr>
<tr><td>Lighting</td><td>40</td><td>0%</td><td>0</td></tr>
</table>

				12.0

ENERGY-USE-REDUCING DESIGN OPTIONS

A number of options were considered for reduction of refrigerator energy use. These options addressed reduction of cabinet load, improved efficiency of components, and system control changes. Initial study of the energy saving potential and the manufacturing cost impact of these options for the two-door refrigerator are illustrated in Figure 3 below.

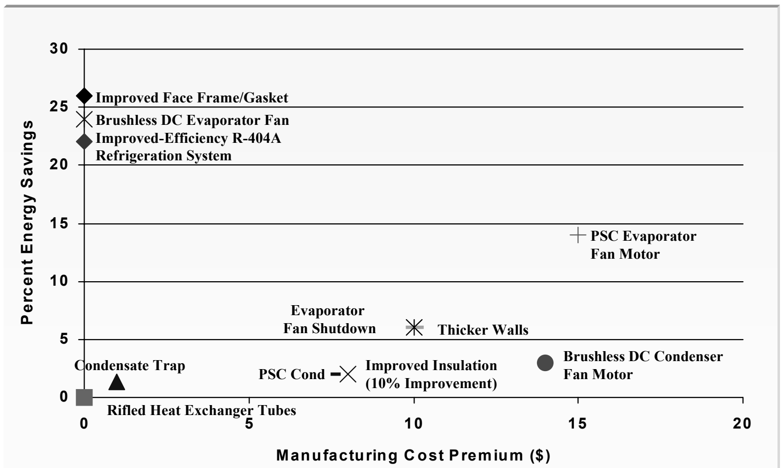

Figure 3: Design Option Energy Savings and Cost Impact

The energy savings of each of the options is presented in the chart as if it were the only design change—savings for multiple options would not be additive. The most cost-effective energy-saving design options are Improved Face Frame/Gasket, Brushless DC Evaporator Fan, and an Improved-Efficiency Refrigeration System. HCFC-22 refrigerant was not considered an option, due to the upcoming phaseout of HCFC-22 for new equipment. While HFC-134a refrigeration systems were considered for this project and would likely have been more efficient, Delfield's desire to use R-404A to simplify manufacturing in a multiple-product environment was considered more important. The cost impact of the improved face frame/gasket system was considered negligible since this represents little more than a design change. The energy savings of this option includes the effect of reduction in antisweat heater wattage allowed by the reduced face frame heat leak. The cost impact of the brushless DC evaporator fan was zero because the two shaded-pole-motor evaporator fans of the 6051 were replaced with a single brushless DC fan (this cost parity would not necessarily apply to other units, for instance single-door cabinets or freezers). The improved efficiency refrigeration system includes both a high-efficiency compressor and heat exchanger adjustments. The cost impact of this option was zero because of improvements in compressor technology and because adjustments to the rest of the system essentially represented very little increase in heat exchanger size or cost.

Many of the other design options, while saving some energy, are somewhat less cost effective. For the refrigerator design, rifled heat exchanger tubes don't improve efficiency for an equivalent-cost heat exchanger, and the potential size reduction is not needed. Infiltration of about 1/3 cfm was attributed to the large straight condensate drain line of the 6051. Incorporation of a condensate trap eliminates this infiltration. Some cost was attributed to this change in Figure 3, but the cost could actually be zero. An arbitrary cost increase was also assigned to the improved insulation, but this cost also may be negligible. The evaporator fan shutdown option involves shutdown of the evaporator fan during the off cycle when the evaporator temperature rises above freezing. The customary practice of 100% evaporator fan run for off-cycle defrost is not always necessary and adds significantly to energy use, especially when using inefficient shaded-pole motors. A strap-on evaporator thermostat is required to achieve this in a unit without electronic control.

IMPROVED REFRIGERATOR DESIGN

Based on the analysis of energy and cost impacts of the evaluated design options, a design concept was selected for the new refrigerator design which includes the following design options.

- Improved face frame and gasket design
- Reduced antisweat heater wattage
- Brushless DC evaporator fan
- Optimized R-404A refrigeration system
- Condensate Line Trap

The new refrigerator design also has a bottom-mounted condensing unit (compressor, condenser, and condenser fan). Analysis and testing of a two-glass-sliding-door beverage merchandiser were done to determine whether the bottom-mount arrangement would increase energy use. Concerns and observations were as follows.

- Warmer temperatures beneath the insulated refrigerator floor could significantly increase conduction load through the floor. However, analysis showed that this effect would add at most 10 Btu/hr (3W) to the cabinet load, and would not be a concern.
- Warm condenser discharge air could rise up the door surface or side or rear wall exterior surface, which could potentially elevate surface temperatures during compressor run time. Measurements of surface temperatures and air temperatures near the surfaces showed no noticeable temperature rise during compressor run time. Furthermore, smoke testing showed that throw of condenser discharge away from the walls and doors of the unit prevents this problem.
- Warm condenser air could potentially rise up and into the unit when the door is opened. Measurements of air temperatures near the doors showed that they drop rather than rise when the doors are opened. Smoke testing of air flows also showed that the cold air inside the unit spills out of the cabinet when the door is opened, thus pushing condenser discharge air away from the cabinet.
- The condenser discharge air could recirculate to the condenser air inlet. Testing of the beverage merchandiser showed that average condenser air inlet temperature was close to 10F (5.6C) above ambient temperature during compressor run times. Figure 4 below shows the condensing unit arrangement for the tested unit and for a recommended layout modification to reduce this recirculation using a baffle. The condensing unit could still slide out for service. For installations in which the refrigerator is located in a tight spot for which the air cannot escape out the rear or sides, the baffle would be half the width of the unit, thus allowing the condenser air to discharge towards the front away from the condenser air inlet.

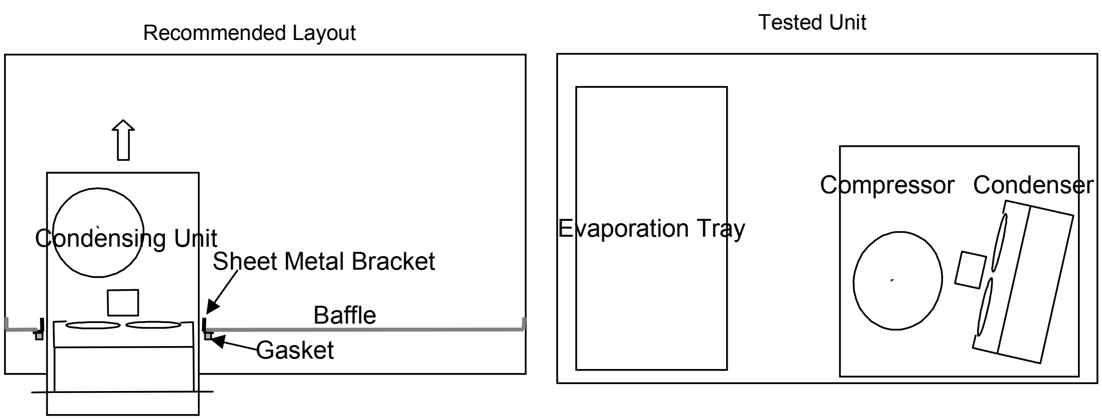

Figure 4: Condensing Unit Layout for Tested Beverage Merchandiser and Recommended Modification

Two energy benefits of the bottom-mount design are (1) the new design has the evaporator mounted entirely within the main refrigerator cabinet at the top, thus reducing the potential for heat loss paths associated with the separate top-mount evaporator box assembly, and (2) the high placement of the evaporator with respect to the condenser allows for a much longer condenser drain line, which reduces infiltration associated with the line.

Key system components of the baseline unit and the new design are compared in Table 4 below. The cost of the new refrigeration system is actually lower than that of the 6051, but is significantly more efficient.

Table 4: Refrigeration System Component Comparison

Component	Baseline Refrigerator (6051)	New 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 (3.9 mm fin spacing)	Face 9” x 9” (229mm x 229mm) Tube rows 9 high x 5 deep Fins wavy, 8 FPI (3.2 mm fin spacing)
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 (3.2 mm fin spacing)	Face 21.5” x 8” (546 mm x 203 mm) Tube rows 8 high x 6 deep Fins wavy, 8 FPI (3.2 fins per mm)
Evaporator Fan	Two fans 6-inch (152 mm) diameter blade 6W SP Motor, 34W input	GE 58 Series ECM, 1,950 rpm

Key aspects of the modifications to the face frame and gasket design for the new unit include the following.

- Prevent the face frame stainless steel sheet from penetrating into the interior of the unit. Allow the steel to extend only as far as required to provide a surface for the gasket magnet to contact.
- Wide gasket design with magnet at the exterior and a bubble on the interior which seals the face frame steel from the interior
- Reduced antisweat heater wattage, optimized for location (outer perimeter or mullion).

The baseline and new-design face frame designs are shown in Figure 5 below.

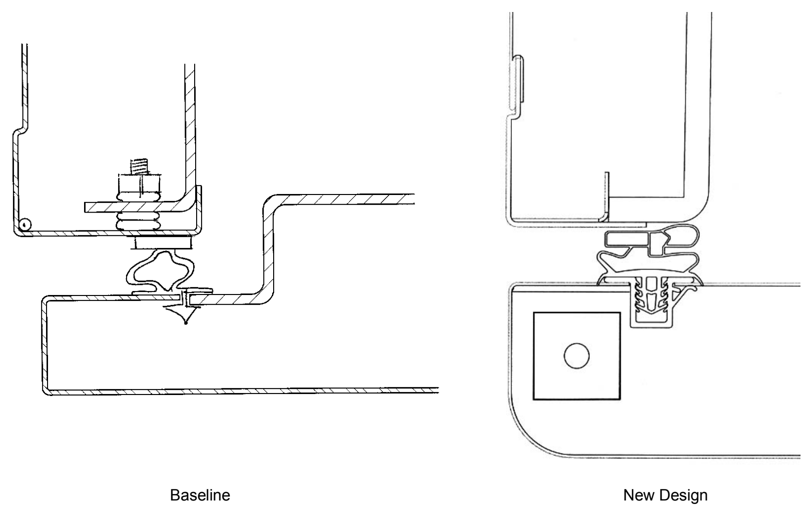

Figure 5: Baseline and New-Design Perimeter Face Frame Cross Sections

RESULTS AND DISCUSSION

Performance of a first prototype and a production version of the new refrigerator are compared with the baseline 6051 in Table 1 and Table 3 above. The energy use of the new refrigerator is significantly lower than that of the baseline unit, for all tests compared: the ASHRAE 117 energy test, and the 80F (26.7C) and 100F (37.8C) closed-door tests. Reduction for the 100F (37.8C) test is nearly 60%, and reduction for the moderate-ambient tests is about a factor of three. Performance of the new design is also significantly improved, as indicated by the temperature control achieved for the 100F (38.7C) 65% RH door-opening test. Recovery of cabinet temperature prior to the next door opening improved from 53F (11.7C) for the 6051 to 28 (-2C) for the new design.

The cabinet load and energy use breakdown for the new design for the NSF 7 test conditions are summarized in Table 5 below. As compared with the baseline refrigerator, there are significant reductions in the energy use associated with the evaporator fan and antisweat heaters. The compressor power is not much lower, in spite of the fact that compressor horsepower is significantly lower. This is explained by the fact that the system operates with a higher evaporating temperature--20F (-6.7C) compressor inlet temperature at compressor cutout as compared with 17F (-8.3C) for the baseline unit. While the power is higher, the efficiency is also higher. This combined with the reduced cabinet load, which greatly reduces run time, results in significant energy use reduction.

Table 5: New Design Refrigerator Cabinet Load and Energy Use for NSF 7 Test Conditions

<table>
<tr><td rowspan="8">Cabinet Load Summary</td><td>Component</td><td colspan="3">Load (Btu/hr)</td></tr>
<tr><td>Cabinet and Door</td><td colspan="3">359</td></tr>
<tr><td>Gasket/Face Frame</td><td colspan="3">70</td></tr>
<tr><td>Infiltration (0.2 scfm)</td><td colspan="3">14</td></tr>
<tr><td>Antisweat Heaters</td><td colspan="3">54</td></tr>
<tr><td>Off-Cycle Charge Migration Loss</td><td colspan="3">100</td></tr>
<tr><td>Evaporator Fan Heat</td><td colspan="3">41</td></tr>
<tr><td>TOTAL</td><td colspan="3">637</td></tr>
<tr><td rowspan="7">Energy Use Summary</td><td>Component</td><td>Input Power (W)</td><td>Duty Cycle (%)</td><td>Daily Consumption (kWh/day)</td></tr>
<tr><td>Compressor</td><td>457</td><td>34%</td><td>3.73</td></tr>
<tr><td>Condenser Fan</td><td>50</td><td>34%</td><td>0.41</td></tr>
<tr><td>Evaporator Fan</td><td>12</td><td>100%</td><td>0.29</td></tr>
<tr><td>Antisweat Heaters</td><td>28</td><td>100%</td><td>0.67</td></tr>
<tr><td>Lighting</td><td>40</td><td>0%</td><td>0</td></tr>
<tr><td></td><td></td><td></td><td>5.1</td></tr>
</table>

CONCLUSIONS

A high-efficiency commercial reach-in refrigerator has been developed 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 reduce 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 6000 series of Reach-Ins by The Delfield Company, who collaborated with us on the design effort. The cabinet design developed for the refrigerator 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 unit tested at ADL is shown in Figure 1 above.

ACKNOWLEDGEMENTS

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REFERENCES

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2 ANSI/ASHRAE Standard 117-1992, "Method of Testing Closed Refrigerators", American Society of Heating, Refrigeration, and Air-Conditioning Engineers, 1992