Fridge Of The Future: Designing A One Kilowatt-Hour/Day Domestic Refrigerator-Freezer

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ABSTRACT

Several design options were investigated for improving the energy efficiency of a conventionally designed, domestic refrigerator-freezer. The options, such as cabinet and door insulation improvements and a high-efficiency compressor were incorporated into a prototype refrigerator-freezer cabinet and refrigeration system. Baseline energy consumption of the original 1996 production refrigerator-freezer, along with cabinet heat load and compressor calorimeter test results, were extensively documented to provide a firm basis for experimentally measured energy savings.

The goal for the project was to achieve an energy consumption that is 50% below the 1993 National Appliance Energy Conservation Act (NAECA) standard for 20 ft^3 (570 l) units. Based on discussions with manufacturers to determine the most promising energy-saving options, a laboratory prototype was fabricated and tested to experimentally verify the energy consumption of a unit with vacuum insulation around the freezer, increased door thicknesses, a high-efficiency compressor, a low-wattage condenser fan, a larger counterflow evaporator, and adaptive defrost control. The resulting energy consumption was 0.93 kWh/d, a substantial energy efficiency improvement of 45% compared to the 1996 model baseline unit (1.68 kWh/d) and 54% better than the 1993 NAECA standard for 20 ft^3 units (2.01 kWh/d). The cost for these improvements was estimated to be approximately $134 (manufacturer's cost). Since the high cost would probably prevent the design from being produced, a second more cost-efficient design was investigated. The second unit eliminated the vacuum panel insulation and larger counterflow evaporator. The cost-improved design resulted in an energy consumption of 1.16 kWh/d at a manufacturer's cost increase of $53. Assuming that there is a 100% markup from manufacturer's cost, the payback for this unit is approximately 6.6 years.

Introduction

Refrigerator-freezers are required to meet certain minimum energy-efficiency standards set up by the U. S. Congress under the National Appliance Energy Conservation Act (NAECA) and administered by the U.S. Department of Energy (DOE) (NAECA 1987). The initial standards went into effect January 1, 1990 and had one revision in 1993 which resulted in a cumulative 40% reduction in energy consumption. In the next revision, scheduled for implementation in July, 2001, the standards will require an additional 30% reduction in energy consumption (Appliance 1997).

Customer expectations and competitive pressures impose constraints on refrigerator-freezers produced in the United States. The excellent characteristics of CFC-12 and its use over a fifty-year period resulted in highly efficient and reliable refrigeration system components (UNEP 1991). Studies have shown that refrigerator-freezers give satisfactory performance for approximately 13 years on average (Appliance 1997). This high degree of reliability has caused consumers to expect long lifetimes and trouble-free operation from refrigerator-freezers. Additionally, refrigerator-freezers have become a relatively low cost commodity item. Therefore, increased costs associated with efficiency improvements must be justified on the basis of an improved environment and lower operating cost to the consumer.

Experimental Plan

An advisory group comprised of all the major refrigerator-freezer manufacturers was formed to give technical direction during the duration of the project. Based on feedback from the group, several options (Table 1) were considered for improving efficiency. The options fall into three main categories: 1) cabinet heat load reductions; 2) refrigeration system improvements; and 3) parasitic power reductions. Options 1 and 2, improvements to the cabinet/door insulation and door gasket, reduce the power requirement by lowering the heat gain to the refrigerated space. Options 3-6 deal primarily with improving the thermodynamic refrigeration cycle efficiency by using a high-efficiency compressor, improving heat exchanger effectiveness, and utilizing a different thermodynamic cycle. Options 7 and 8 reduce the parasitic power requirements by substituting electrically-commutated direct-current (DC) motors for those presently used in the evaporator and condenser and by using a long-term defrost control scheme to initiate defrost based on demand. In a previous study (Vineyard, et al. 1995) where most of the options in Table 1 were investigated both analytically and experimentally, the results showed that the largest energy savings came from cabinet insulation improvements, the high-efficiency compressor, and the low-wattage fan motors. Thus, the present effort concentrated on those same options along with adaptive defrost control and heat exchanger improvements. Advanced cycles had the lowest priority of all the options due to the increased complexity they would add to the system and would not be investigated unless the project goal could not be achieved otherwise.

Test Procedures

Several tests were conducted to quantify the effects on energy consumption of refrigeration system and cabinet design changes. All tests were performed on a 20 ft^3 (570 l) top-mount, automatic-defrost, refrigerator-freezer with a forced-air condenser and evaporator. The testing included reverse cabinet heat loss rate measurements, standard nine-point compressor calorimeter mappings, and 90°F (32.2°C) closed-door, energy-consumption tests as specified in section 8 of the Association of Home Appliance Manufacturers (AHAM) standard for Household Refrigerators and Household Freezers (AHAM 1985). The tests were performed in environmental chambers with airflows and temperature fluctuations within the specifications of the AHAM standard or according to manufacturers' recommendations for tests where no standard is specified, such as the reverse heat loss rate tests.

Reverse Cabinet Heat Loss Rate Measurements

Reverse cabinet heat loss rate measurements were made to assess the improvements in cabinet thermal performance from changes such as vacuum insulation or increased insulation thickness in the freezer section or doors. The procedure for measuring heat loss rate involves placing a cabinet in a cold chamber with controlled heat sources and small electrical chassis fans to maintain desired temperatures in both the freezer and fresh food compartments. The fans are run continuously during the test to prevent temperature stratification. Each fan draws approximately 6-7 watts of electricity and has an air circulation rate of 30 cfm (14 l/s). Temperature and watt measurements for both refrigerator-freezer compartments along with ambient temperature are recorded as the cabinet temperatures achieve desired levels. Once the cabinet temperatures achieve steady-state, data is compiled and averaged for a thirty-minute interval to determine overall heat loss rates for both compartments.

The heat loss rate is calculated in Btu/h (W) and plotted against the difference between temperatures inside each compartment and ambient air temperature. Tests were initially run with the temperatures in both compartments essentially equal. This ensured that there was no heat transfer across the mullion; thus allowing the freezer and fresh food compartment resistivities to be determined from dividing the power measurement by the temperature difference in each compartment. Once the compartment thermal resistivities were known, tests were performed with large temperature differences between the freezer and fresh food compartments to determine the mullion thermal resistivity. Plots were then generated from equations for heat transfer in each compartment based on the thermal resistivities of the freezer, fresh food compartment, and mullion.

Compressor Calorimeter Mappings

Reductions in the total cabinet heat load required corresponding changes in the capacity and design of the compressor. In order to determine the extent of these changes, the original and high-efficiency compressors were tested using a nine-point compressor calorimeter procedure to generate compressor maps. In this procedure, compressor operating characteristics, including refrigeration capacity and energy efficiency ratios (EERs) are determined at each point in a matrix of 110°F (43.3°C), 120°F (48.9°C), and 130°F (54.4°C) condensing temperatures and -20°F (-28.9°C), -10°F (-23.3°C), and 0°F (-17.8°C) evaporating temperatures. Also specified in the test procedure are a 90°F (32.2°C) ambient temperature for the compressor, superheating of the suction gas to 90°F (32.2°C), and subcooling of the liquid refrigerant line to 90°F (32.2°C) before throttled expansion. The nine-point maps generated from the tests are used to estimate changes in refrigerator-freezer energy consumption when using the high-efficiency compressor.

Energy-Consumption Tests

System performance for the baseline and enhanced cabinets was assessed using the standard 90°F (32.2°C) closed-door test procedure. In this procedure, the refrigerator-freezer is operated at two different control settings in a 90°F ± 1°F (32.2 ± 0.6°C) environmental chamber. Energy use and compartment temperatures are measured from the onset of one defrost cycle to the beginning of the next defrost. The test points are then used to calculate the energy consumption over a 24 hour period based upon a reference 5°F (-15.0°C) freezer temperature and 45°F (7.2°C) fresh food temperature. Other requirements of the test procedure are an outlet voltage level of 115 ± 1 volt AC to the refrigerator-freezer and an air circulation rate of less than 50 ft/min (15 m/min) in the environmental chamber. The high ambient temperature, 90°F (32.2°C) is used to simulate the contribution of door openings and food loadings. Comparisons of field performance to closed-door test ratings indicate the laboratory procedure is a valid indication of average energy use in field service (Meier and Jansky 1993). Previous refrigerator-freezer testing indicated that the test procedure with two different thermostat settings gives a broader indication of appliance performance at different ambients and internal operating conditions as opposed to a single-point test (Sand et al. 1993).

Experimental Results

The experimental approach emphasized hardware changes that can be incorporated into a conventional refrigerator-freezer design, which is defined as a unit with a single, fan-forced evaporator and condenser, single-speed compressor, and operating with a pure refrigerant. Changes centering on a conventional design were considered to be more acceptable to manufacturers because they would require less retooling and have greater reliability. In addition, a conventional design is more likely to be accepted by consumers since it would cost less to implement than a nonconventional design change, such as a dual evaporator system with nonazeotropic refrigerant mixtures.

Reverse Heat Loss Results

Steady state heat loss measurements were performed on two separate cabinets; a baseline refrigerator-freezer cabinet and an enhanced cabinet with vacuum insulation panels foamed around the freezer section. In addition to the standard doors, which were 1-inch (2.5 cm) thick, three sets of doors with varying degrees of insulation improvements were tested on the baseline cabinet. The door improvements consisted of the following: thick doors (2 inches) (5.1 cm), 1-inch (2.5 cm) thick vacuum insulation panels foamed into standard doors, and 1-inch (2.5 cm) thick vacuum insulation panels foamed into thick doors. For the tests with the enhanced cabinet, standard doors and thick doors with no vacuum insulation panels were investigated.

Cabinet heat loss rates for the baseline cabinet with the standard doors and door insulation improvements are shown in Figure 1. The heat loss rates are determined by using compartment and mullion UAs calculated from measurements made under steady-state conditions. The compartment heat loss rates are in Btu/h (W) and plotted for temperature differences between the ambient and compartment of 45°F (25°C) in the fresh food section and 85°F (47.2°C) in the freezer section. These temperature differences are representative of those for the freezer and fresh food compartments when using the 90°F (32.2°C) closed-door test procedure. Figure 2 shows the cabinet heat loss results for the enhanced cabinet with the standard and thick doors.

The cabinet heat loss rates are summarized in Table 2 along with Q_{FRZ}/Q_{TOT} ratios for a refrigerator-freezer. The experimental results indicate that the baseline cabinet heat loss rate was reduced 6.4% (195.2 to 182.7 Btu/h) (57.2 to 53.5 W) by replacing the standard doors with thick doors. Using 1-inch (2.5) thick vacuum panels and foaming them into standard doors resulted in the cabinet heat loss rate being reduced from 195.2 to 173.7 Btu/h (57.2 to 50.9 W), an 11.0% reduction. Finally, when 1-inch (2.5) thick vacuum panels were foamed into a thick door, the cabinet heat loss rate was reduced by 12.3%.

Examining the individual compartments, the additional insulation and vacuum panels appear to have the most benefit in the fresh food section, lowering the heat loss rate by as much as 20.7%. This is probably the result of the fresh food section initially having less overall insulation thickness than the freezer compartment. By contrast, the maximum improvement in the freezer section was less than half that amount (8.3%).

For the enhanced cabinet, vacuum panels foamed around the entire freezer section resulted in an overall cabinet heat loss rate of 165.9 Btu/h (48.6 W), or 15.0% lower than the baseline cabinet. Tests were also performed with thick doors on the enhanced cabinet resulting in a 20.4% reduction in the overall cabinet heat loss rate (195.2 versus 155.3 Btu/h) (57.2 versus 45.5 W). While the cabinet heat loss rate could have been reduced even further by using vacuum panel doors, the additional cost ($53.52) would have been prohibitive. Therefore, that configuration was not tested.

Compressor Calorimeter Results

Nine-point calorimeter tests were used to determine the performance over a range of operating temperatures for the baseline compressor used in the production refrigerator-freezer and the high-efficiency compressor used in the modified units. The high-efficiency compressor is a variable-speed model that can be run at speeds from 2200-3600 rpm with only minor variations in EER. For these tests, the compressor was run at the lowest speed (2200 rpm). The resulting compressor maps, shown graphically in Figure 3, are used as inputs for the modeling analyses. From the data in Figure 3, one can determine that, at the standard rating point for a -10°F (-23.3°C) evaporator and a 130°F (54.4°C) condenser, the EER for the baseline compressor is 4.28 while that of the high-efficiency compressor is 5.73, a 33.9% increase in EER.

The refrigeration capacity of the high-efficiency compressor was approximately 523 Btu/h (153.2 W) or 11% less than the baseline compressor (587 Btu/h) (172.0 W) it replaced. The high-efficiency compressor was run at the lowest speed possible in attempts to achieve reasonable run times once additional insulation was added to the cabinet doors and vacuum panel insulation was added to the freezer section. Using a compressor whose capacity is much greater than the load would have resulted in short, frequent compressor runs that increase system cycling losses.

System Results

Of the eight options under consideration for reducing the energy consumption of the refrigerator-freezer, only five were required to achieve the goal of a 50% energy savings. Those five options were: 1) cabinet and door insulation enhancements; 2) a high-efficiency compressor; 3) a low-wattage condenser fan; 4) adaptive defrost control; and 5) a larger evaporator with a counterflow arrangement. Option number 5, a larger condenser with a counterflow arrangement, would have been the next design change to be introduced had it been necessary to achieve further savings. The other modifications, door gasket improvements and an advanced cycle design were low priority items due to their additional complexity and difficulties in incorporating them into a commercially-manufactured cabinet. However, they would have been addressed if the goal had not been achieved.

Energy consumption tests were initially performed on the baseline cabinet according to section 8 of the AHAM Standard for Household Refrigerators and Household Freezers (AHAM 1985). The results, Table 3, show that the energy consumption was 1.68 kWh/d. The NAECA standard for a unit of this type and size is 2.01 kWh/d. Thus, the baseline cabinet is 17% below the NAECA standard.

Following completion of the energy consumption tests on the baseline cabinet, tests were performed on an enhanced cabinet with vacuum panels foamed around the freezer section and a larger counterflow evaporator. The unit was further modified by exchanging the standard doors for ones that were 2 inches (5.1 cm) thick and by replacing the existing condenser fans and compressor with a low-wattage fan and a high-efficiency compressor (5.73 EER). In addition, a long-term defrost control algorithm was used to reduce the energy consumption associated with defrosts. The savings attributed to the defrost control algorithm was calculated using the method prescribed in the AHAM test procedure. The results for all the improvements (Enhanced Design l), Table 3, show that the energy consumption was reduced from 1.68 kWh/d to 0.93 kWh/d, a savings of 45%, compared to the baseline unit. Relative to NAECA standards, the results represent a 54% improvement (2.01 to 0.93 kWh/d).

An additional design configuration (Enhanced Design ll) was assembled by replacing the existing compressor and condenser fan on the baseline unit with the high-efficiency compressor and low-wattage condenser fan. In addition, the standard doors were replaced with the 2-inch (5.1 cm) thick doors and a long-term defrost control algorithm was utilized. Although the energy consumption for this configuration was expected to be moderately higher than for the Enhanced Design l unit, the changes were expected to be more cost-effective. The resulting energy consumption for the unit was 1.16 kWh/d, a 31% reduction from the baseline unit and 42% lower than NAECA standards.

Cost Analysis

In order to obtain a cost/benefit ratio of the energy-saving features, it was necessary to estimate the cost for each design change (Table 4). Most of the information was obtained from a study on the cost-efficiency of design options in support of the proposed 1998 NAECA standards (Hakim and Turiel, 1996). In that study, costs were collected from several refrigerator-freezer manufacturers and averaged to protect the confidentiality of the data. In addition to that information, manufacturer's costs were estimated by the suppliers for the high-efficiency compressor and vacuum panel insulation based on the added electronics and square footage of insulation added to the freezer section.

Using the information from Table 4, the estimated manufacturer's cost increase for the Enhanced Design l unit is $134.33. This estimate is based on using a high-efficiency condenser fan ($4.50), adaptive defrost control ($7.15), an increased evaporator area ($3.11), 2-inch (5.1 cm) thick doors ($6.73), a 5.73 EER high-efficiency compressor ($35.00), and vacuum panel insulation around the freezer section ($77.84). The energy savings from all these features is 273 kWh/yr, relative to the baseline unit (1.68 vs. 0.93 kWh/d). Based on an average cost for electricity of $0.0867/kWh, the annual savings is $23.67. Doubling the manufacturers cost to arrive at an estimated cost to the consumer gives a payback of 11.3 years ($268.66/$23.67 per year), which is considered too long for most consumers.

A breakdown of the energy savings from each design change is shown in Table 5. The magnitude of energy savings attributed to each measure is affected by the order in which improvements are made. The order shown in Table 5 is the order that changes were actually made to the baseline unit. Two of the entries, the condenser fan and adaptive defrost energy savings, were calculated rather than experimentally tested. The condenser fan savings was determined from multiplying the difference in the fan wattages of the production fan (11.6 W) and the low-wattage fan (2.7 W) by the number of hours of run time (44.2%). The savings for the adaptive defrost control was calculated from experimental data using the procedure outlined in the AHAM test procedure. The results show that the low-wattage condenser fan, thicker doors, and adaptive defrost control had paybacks in the range of 3.0 - 4.1 years. The high-efficiency compressor required 7.7 years to payback. The worst payback period was for the vacuum panel insulation/increased evaporator area combination which needed almost 36 years to payback, clearly an unacceptable alternative. For all the scenarios, it was assumed that the consumer cost was twice the manufacturer's cost.

Since the payback was determined to be too long for the Enhanced Design l to be economically feasible, a second unit (Enhanced Design ll) was assembled at a much lower cost. The estimated manufacturer's incremental cost for this unit is $53.38 based on using a high-efficiency condenser fan ($4.50), adaptive defrost control ($7.15), 2-inch (5.1 cm) thick doors ($6.73), and a 5.73 EER high-efficiency compressor ($35.00). The energy savings for this unit is 190 kWh/yr (1.68 vs. 1.16 kWh/d). Using a cost for electricity of $0.0867/kWh, the annual savings is $16.47. The payback, assuming the consumer cost is twice that of the manufacturer's cost, is 6.5 years.

Conclusions

Two significant accomplishments were realized from the project. First, it was shown to be technically feasible to build an extremely low energy-consuming 20 ft^3 refrigerator-freezer. It would have been possible to reduce the energy consumption even further had the vacuum panel doors been used. There were, however, two drawbacks to the Enhanced Design l unit; 1) the costs were prohibitively high and; 2) the compressor run time was too low, indicating that a much smaller compressor, probably in the 400 - 450 Btu/h (117-132 W) range, was required. Compressors in this capacity range traditionally have much lower EERs than those in the 700 - 800 Btu/h (205 -234 W) range. Thus, improving the efficiency of small capacity compressors would appear to be a high priority for reducing energy consumption in future refrigerator-freezers. This assumes that some form of cabinet improvement, such as vacuum insulation, thicker insulation, or door gasket improvements, will be used to significantly reduce the cabinet heat gain. At present, vacuum insulation, while an excellent technology, still appears too costly. In addition, vacuum panel insulations remain unproven in terms of long term reliability and heat transfer degradation over time;

two factors that must be addressed. Instead of being used to reduce the energy consumption, a more appropriate application for vacuum panel insulations in refrigerator-freezers appears to be in the area of gaining additional food storage volume by reducing the insulation volume in areas where it is thickest, such as the doors.

The second, and most promising accomplishment, was the Enhanced Design II unit, resulting in a 1.16 kWh/day energy consumption. Based on the cost analysis results (Table 5) indicating that the vacuum panel insulation and increased area evaporator were not cost-effective, the Enhanced Design II unit was assembled without these features. The new unit achieved a low energy consumption with a reasonable additional cost. The cost of the Enhanced Design II unit could be reduced even more by using a production compressor with a slightly lower EER than the high-efficiency compressor. Using a compressor with an EER in the 5.2 - 5.3 range would increase the energy consumption to approximately 1.25 kWh/d. The additional cost for the unit would be around $18 or $36 to the consumer. The unit would save 157 kWh/yr for a savings of $13.61 annually. The payback on a unit like this would be less than 3 years, which should be even more appealing to consumers than the 6.6 year payback for the Enhanced Design II version.

It is noted that the design changes were made to a top-mount, automatic defrost refrigerator-freezer which accounts for the largest share of the market in the U. S. The same changes could be applied to side by side units and achieve comparable energy savings.

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Table 1. Design Options for Improving the Energy Efficiency of a Refrigerator-Freezer

Option Number	Design Change
Option 1	Improved cabinet and door insulation
Option 2	Reduced door gasket losses
Option 3	High-efficiency compressor substitution
Option 4	Increased evaporator size with counterflow arrangement
Option 5	Increased condenser size with counterflow arrangement
Option 6	Advanced cycle with zeotropic hydrocarbon mixture
Option 7	Low-wattage fan motors
Option 8	Adaptive defrost control

Table 2. Summary of Reverse Heat Loss Tests - 90° F ambient, 5° F Freezer, 45° F Fresh Food Compartment

Description	$Q_{freezer}$ (Btu/hr)	$Q_{fresh\ food}$ (Btu/hr)	Q_{total} (Btu/hr)	$Q_{freezer}/Q_{total}$	Percent Reduction
Base Cabinet:					
w/standard doors (1 inch)	103.4	91.8	195.2	0.53	----
w/thick doors (2 inches)	94.8	87.9	182.7	0.52	6.4
w/vacuum panels in standard doors (1 inch)	95.1	78.6	173.7	0.55	11.0
w/vacuum panels in thicker doors (2 inches)	98.4	72.8	171.2	0.57	12.3
Enhanced Cabinet:					
w/vacuum panels around freezer section	86.4	79.5	165.9	0.52	15.0
w/vacuum panels around freezer section + doors (2 inches)	80.3	75.0	155.3	0.52	20.4

Table 3. Energy Consumption and Cost Information

Description	Energy Consumption (kWh/d)	Percent Run Time	Manufacturer's Cost Increase (dollars)
Baseline Unit	1.68	44.2%	----
Enhanced Design I	0.93	36.5%	134.33
Enhanced Design II	1.16	47.6%	53.38

Table 4. Manufacturer's Cost Increase for Design Changes

Design Change	Manufacturer's Cost Increase (dollars)
Low-wattage condenser fan	$4.50
Increased evaporator area	$3.11
Vacuum panels around freezer section	$77.84
2-inch thick doors	$6.73
High-EER compressor	$35.00
Adaptive defrost control	$7.15

Table 5. Cost Analysis for Design Options for Enhanced Design 1

Case	Design Changes	Annual Energy Use (kWh/yr)	Annual Energy Savings (kWh/yr)	Cost Savings (dollars/yr)	Consumer Cost (dollars)	Payback (years)
A	Baseline Unit	613	---	---	---	---
B	A+ vacuum insulation around freezer, increased evaporator area	561	52	4.51	161.90	35.9
C	B+ low-wattage condenser fan	526	35	3.03	9.00	3.0
D	C+ 5.73 EER compressor	421	105	9.10	70.00	7.7
E	D+ 2-inch thick doors	380	41	3.55	13.46	3.8
F	E+ adaptive defrost control	340	40	3.47	14.30	4.1

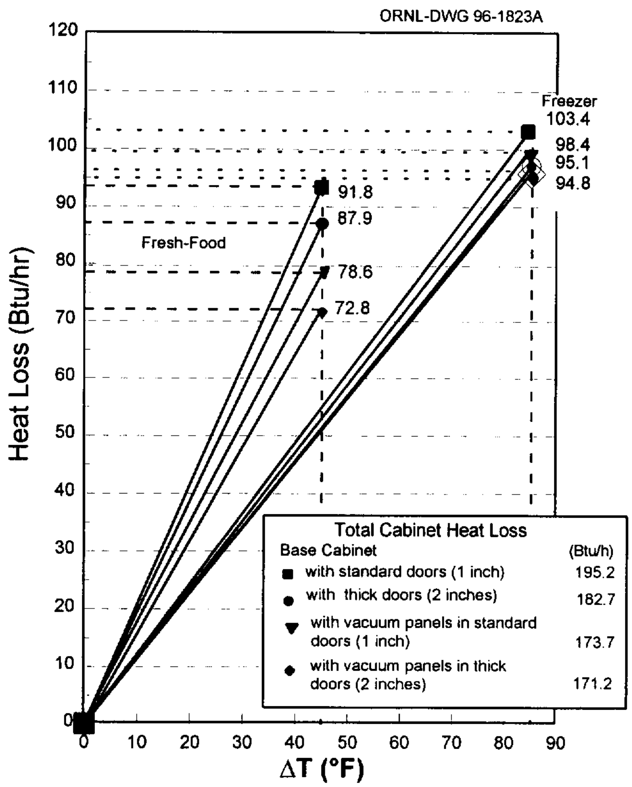


Figure 1. Reverse heat loss results for base cabinet

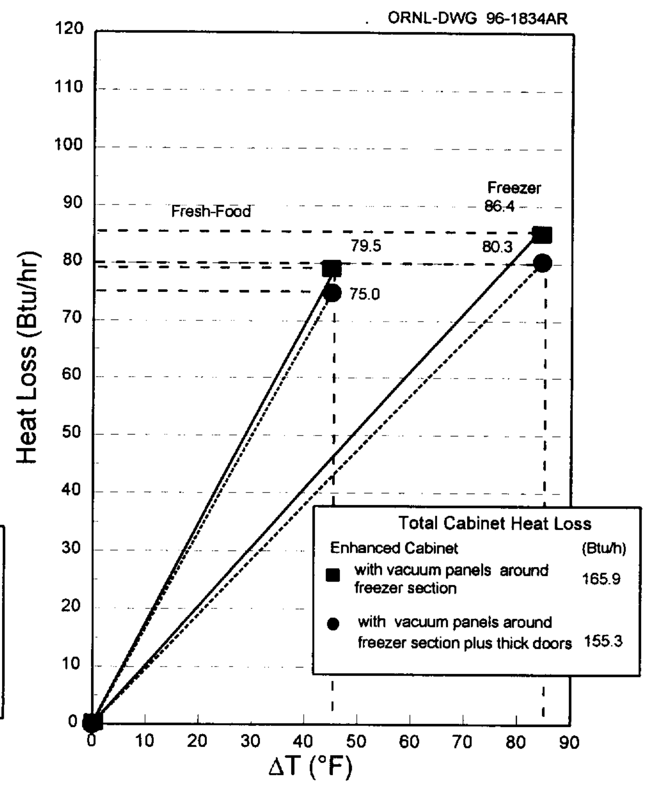
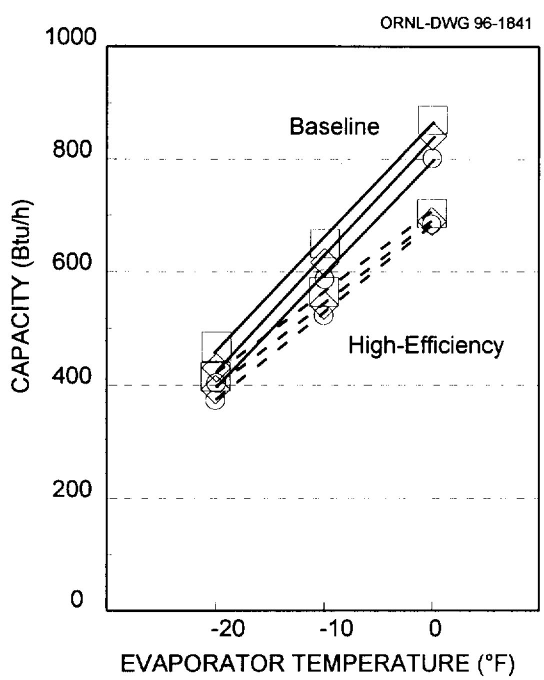


Figure 2. Reverse heat loss results for enhanced cabinet.



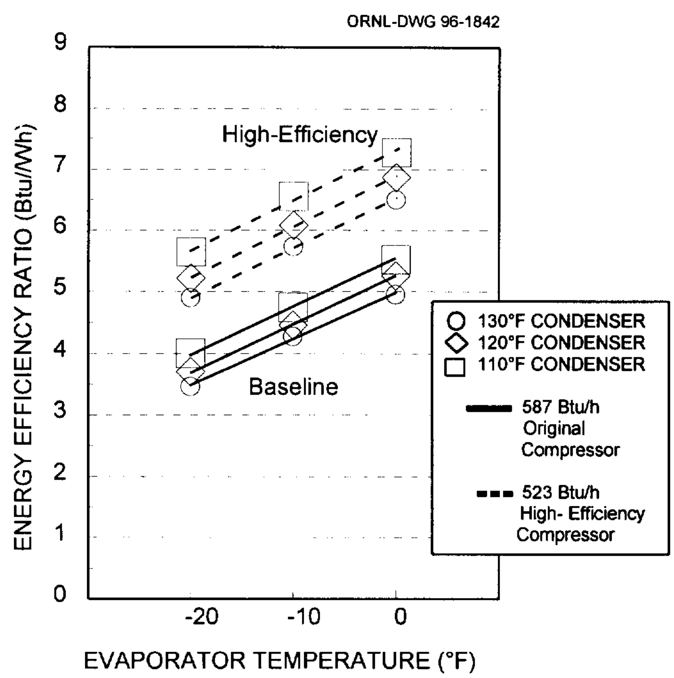


Figure 3. Compressor calorimeter results

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