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EVALUATION OF DESIGN OPTIONS FOR IMPROVING THE ENERGY EFFICIENCY OF AN ENVIRONMENTALLY SAFE DOMESTIC REFRIGERATOR-FREEZER

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EVALUATION OF DESIGN OPTIONS FOR IMPROVING THE ENERGY EFFICIENCY OF AN ENVIRONMENTALLY SAFE DOMESTIC REFRIGERATOR-FREEZER

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ABSTRACT

In order to reduce greenhouse emissions from power plants and respond to regulatory actions arising from the National Appliance Energy Conservation Act (NAECA), several design options were investigated for improving the energy efficiency of a conventionally designed, domestic refrigerator-freezer. The options, such as improved cabinet insulation and high-efficiency compressor and fans, were incorporated into a prototype refrigerator-freezer cabinet and refrigeration system to produce a unit that is superior from an environmental viewpoint due to its lower energy consumption and the use of refrigerant HFC-134a as a replacement for CFC-12. Baseline energy performance of the original 1993 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.

A detailed refrigerator system computer model was used to evaluate the energy savings for several design modifications that, collectively, could achieve a targeted energy consumption of 1.00 kWh/d for a 20 ft^3 (570 l) top-mount, automatic-defrost, refrigerator-freezer. The energy consumption goal represents a 50% reduction in the 1993 NAECA standard for units of this size. Following the modeling simulation, laboratory prototypes were fabricated and tested to experimentally verify the analytical results and aid in improving the model in those areas where discrepancies occurred. While the 1.00 kWh/d goal was not achieved with the modifications, a substantial energy efficiency improvement of 22% (1.41 kWh/d) was demonstrated using near-term technologies. It is noted that each improvement exacts a penalty in terms of increased cost or system complexity/reliability. Further work on this project will analyze cost-effectiveness of the design changes and investigate alternative, more-elaborate, refrigeration system changes to further reduce energy consumption.

INTRODUCTION

Domestic refrigerator-freezers, used primarily for food preservation, are an important end user of electricity. Approximately 58 million new units are manufactured worldwide each year and hundreds of millions are currently in use (UNEP 1991). It is anticipated that the production of refrigerator-freezers will substantially increase in the near future as the result of an increased demand, especially in developing countries where growth is expected to be on the order of 10 to 15% per year for the next few years. It is imperative, in order to protect the environment, that industry incorporate environmentally safe refrigeration systems and cabinet designs as early as possible while reducing current levels of energy consumption. This task will be particularly challenging due to the fact that today's energy-efficient designs were made possible through the use of CFCs which are now being phased out.

Greenhouse gases and their damaging effects on the atmosphere have received increased attention following the release of scientific data by the United Nations Environment Programme and World Meteorological Organization that show carbon dioxide to be the main contributor to increased global warming (UNEP 1991). For domestic refrigerator-freezers operating on alternative refrigerants such as HFC-134a, the indirect contribution to global warming potential resulting from the energy consumption of the unit is approximately one hundred times greater than the direct contribution of the refrigerant alone. Therefore, in response to global concerns over greenhouse gases, efforts are being made to produce refrigerator-freezers with low energy consumption (Fischer et al. 1991).

In addition to the concerns of the global community over greenhouse emissions, refrigerator-freezers are also required to meet certain minimum energy-efficiency standards set up by the U. S. Congress and administered by the U.S. Department of Energy (DOE) (NAECA 1987). The initial standards went into effect January 1, 1990 and have had one revision, in 1993, that resulted in an average 25% reduction in energy consumption. By 1998, the standards are expected to require an additional 25% reduction in energy consumption. An historical chart showing actual and projected improvements in the electrical energy use of refrigerator-freezers is shown in Fig. 1. Also indicated in the figure is the stated goal for this project.

Customer expectations and competitive pressures impose an unwritten set of constraints on refrigerator-freezers produced in the United States. The majority of refrigerator-freezers employ the vapor compression refrigeration cycle with CFC-12 as the working fluid. The excellent characteristics of CFC-12 and its use over the past fifty years have led to highly efficient and reliable compressors and other refrigeration system components (UNEP 1991). Studies have shown that refrigerator-freezers give satisfactory performance for approximately 14 years on average (Appliance 1993). This high degree of reliability has caused consumers to expect long lifetimes and trouble-free operation from refrigerator-freezers and all appliances in general. Additionally, a low first cost is probably the most important consideration for an appliance that has become a commodity item. This mindset seems to conflict with the importance of a refrigerator-freezer in a household where the efficient storage and distribution of refrigerated food are essential to the existence of an industrialized society.

In an effort to design an environmentally safe, energy-efficient, cost-effective refrigerator-freezer, an industry/government Cooperative Research and Development Agreement (CRADA) was established to evaluate and test design concepts for an advanced domestic unit. The stated goal of the CRADA is to demonstrate, by 1995, advanced technologies which reduce the 1993 NAECA standard energy consumption for a 20 ft^3 (570 l) top-mount, automatic-defrost, refrigerator-freezer by 50% or more. For a unit this size, the goal implies an energy consumption of 1.00 kWh/d. The general objective of the research is to facilitate development of efficient appliances through establishing those design changes that can be conveniently incorporated into new products which are cost effective and reliable.

MODELING ANALYSIS

A widely distributed computer model that combines a cabinet heat load model, refrigeration system model, and an "on/off" cycling algorithm was used to analytically evaluate the design options for improving energy efficiency (EPA 1993). The model, while simple to operate, is able to accommodate system hardware and refrigerant changes; a feature normally found in more empirically specific simulation models used by appliance manufacturers. The model also enables the user to assess the energy saving potential of options, such as improved door gaskets, that may not be presently available. A summary of the output information from the model includes the following: 1) cabinet heat loads in both the freezer and fresh food sections; 2) compressor run time; 3) power consumption for the compressor, fans, and heaters; and 4) total energy consumption. More detailed information, such as breakdowns for the cabinet heat loads and component efficiency information are also available.

Several options (Table 1) were considered for decreasing the energy consumption of a conventional refrigerator-freezer. The options were selected on the basis of previous studies and discussions with an advisory group comprised of all the major refrigerator-freezer manufacturers (Bohman 1987; Turiel and Heydari 1988). The options fall into four main categories: 1) refrigeration system improvements; 2) cabinet heat load reductions; 3) parasitic power reductions; and 4) cycling loss reductions. Options 1-5 deal primarily with improving the thermodynamic refrigeration cycle efficiency by using a high-efficiency compressor, improving heat exchanger effectiveness by adding more external heat transfer area, and using a refrigerant that is thermodynamically superior to the one that is presently used (CFC-12). Options 6 and 7, insulation and door gasket improvements, reduce the power requirement by lowering the heat gain to the refrigerated space. Option 8 reduces the parasitic fan power requirements for both the evaporator and condenser fans by substituting electrically-commutated direct-current (DC) motors for those presently used. Option 9, a liquid-line shut-off valve, has been used extensively with rotary compressors to prevent refrigerant migration to the evaporator during the compressor off-cycle. However, its energy-saving potential for reciprocating compressors has not been verified.

A 1993, 20 ft^3 (570 l) top-mount, automatic-defrost refrigerator-freezer was selected for the project based on its popularity and corresponding high market share. Since the unit was required to meet the new 1993 NAECA standards, the baseline energy consumption was quite low (1.80 kWh/d), thus making further reductions in energy consumption very challenging. Among the energy saving features incorporated into the design of the baseline unit were a high-efficiency compressor, increased insulation thicknesses, and liquid line flange heaters. Beginning with the baseline unit, design options were then added sequentially to the model so that the cumulative effects of each option could be accounted for. Manufacturer's specifications for cabinet dimensions and components were used as inputs to the model for the simulation calculations.

The results in Figure 2 show that the most significant energy savings come from a high-efficiency rated-speed compressor, better cabinet insulation, and a liquid-line shut-off valve.

The rated-speed compressor, an emerging technology, uses an electrically-commutated permanent magnet DC motor that is 10-15% more efficient than commonly used capacitor start/induction run or permanent split capacitor induction motors. For this application, the DC motor was operated at a single, optimal speed.

High-efficiency fan motors and increased heat exchanger areas also account for sizable energy savings. Since these changes resulted in a refrigerator-freezer whose predicted energy use was 1.07 kWh/d, further modifications were necessary to meet the goal even though smaller and smaller energy savings resulted from each consecutive change. A 1.00 kWh/d daily energy use was only achieved after additional changes were incorporated into the model for improved door gaskets, better suction-line heat transfer, and the use of a more efficient refrigerant.

It is recognized that analytical modeling studies of a conventional refrigerator-freezer design may not accurately reflect the corresponding realities for an actual unit. If it is determined through testing that the goal cannot be achieved with a conventional design, more exotic thermodynamic cycles, such as the Lorenz Cycle, are being considered for future phases:

- A single-evaporator, refrigerator-freezer operating with a low glide (10-20°F [5.6-11.1°C]) refrigerant mixture, counterflow heat exchangers, and a variable-speed compressor, and

- a two-evaporator, Lorenz-Meutzner refrigerator-freezer operating with a high glide (35°-50°F [20-30°C]) refrigerant mixture, counterflow heat exchangers, and a single speed compressor.

Both of these advanced cycle designs would involve increased system complexity and, perhaps, compromises in appliance reliability.

TEST PROCEDURE

All tests were performed on a 20 ft^3 (570 l) top-mount, automatic-defrost, refrigerator-freezer with a forced-air condenser and evaporator. A variety of different experimental tests were conducted for the project to verify energy performance, determine inputs for the model, and to analyze the effects of different design changes. The tests included reverse cabinet heat loss measurements, standard nine-point compressor calorimeter mappings, and the 90°F (32.2°C) closed-door, energy-consumption test as specified in section 8 of the Association of Home Appliance Manufacturers (AHAM) Standard for Household Refrigerators and Household Freezers (AHAM 1985).

Reverse Heat Loss Measurements

Reverse heat loss measurements were made to assess baseline and improved cabinet thermal resistivity. The procedure for measuring heat loss involves placing a cabinet in a cold chamber with a controlled heat source, such as a shielded light bulb, and a small electrical chassis fan 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/sec), which is assumed to have negligible effects on the inside surface heat transfer of the refrigerator-freezer.

Reverse heat loss tests were conducted using temperature differences across the cabinet walls comparable to those prescribed in the 90°F (32.2°C) closed-door test procedure where the refrigerator-freezer works to maintain cold internal temperatures in a hot room. In order to accomplish the temperature differences, it was necessary to maintain the chamber at 10°F (-12.2°C). Since the thermal conductivity of insulating foam generally decreases with decreasing temperatures, this procedure may slightly underestimate actual cabinet heat loss rates (ASHRAE 1989).

Compressor Calorimeter Tests

Changes in the total cabinet heat load and substitution of a chlorine-free refrigerant for the CFC-12 necessitated corresponding changes in the capacity and design of the compressor and other refrigeration circuit hardware. Original and replacement compressors used for the modified unit 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 as inputs to the computer model to evaluate changes in energy consumption for different compressors.

System Tests

System performance, before and after modifications were made to the units, 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 with the anti-sweat heater switch in both the on and off positions. Energy use and compartment temperatures are measured from the onset of one defrost cycle to the beginning of the next defrost. The resultant test points are used to calculate the energy consumption over a 24 hour (one day) period based upon a reference 5°F (-15.0°C) freezer temperature and 38°F (3.3°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 quite valid indication of energy use in field service (Meier and Jansky 1993). Previous refrigerator-freezer testing indicated that the four-point 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

An experimental plan was formulated to help order and prioritize laboratory work. The plan emphasizes changes in hardware that can be incorporated into a conventional refrigerator-freezer design, defined as a unit with a single, fan-forced evaporator and condenser, single-speed compressor, and operating with a pure refrigerant or nearly azeotropic refrigerant blend (< 6°F [3°C] separation of dew and bubble points at evaporator pressures). Changes centering on a conventional design were considered to be more acceptable to manufacturers because they would require less retooling and have greater reliability. As part of the emphasis on designing a more environmentally acceptable refrigerator-freezer, the CFC-12 was replaced with a non-ozone-depleting refrigerant (HFC-134a) for the experimental evaluations.

Reverse Heat Loss Results

Steady state heat loss measurements were performed on the baseline refrigerator-freezer cabinet. Following the completion of the baseline tests, expanded polystyrene panels with R values of 5.3 to 5.6 h-ft^2-°F/Btu-in (2.4-2.5 m^2-°C/w-cm) were added to the cabinet exterior to simulate a reduced heat load cabinet design. Since it was beyond the scope of the project to develop an advanced insulation, a simulation of this nature was in order to properly design the refrigeration system for expected cabinet load reductions in the future. Data, similar to that in Figure 3, were obtained for the freezer and fresh food compartment heat loss rates under steady-state conditions. The results are measured in Btu/h and plotted against the difference between air temperatures inside the two compartments and ambient air temperatures (≈ 0-5 °F [-18 to -15°C]) maintained in an environmental chamber. A linear relationship between heat loss rate and temperature difference (equation 1) is assumed. Equations obtained through a least-squares fit of the data were used to calculate the cabinet heat load arising from heat permeation through walls, doors, and gaskets at operating and ambient temperatures. It is interesting to note that the extension of the lines showing cabinet heat loss as a function of temperature (Figure 3) do not pass through the origin as would be expected. The reason this occurs is that heat transfer takes place between the freezer and fresh food compartments through the mullion.

$$Q = UA \cdot \Delta T \qquad (1)$$

A 10% targeted improvement in the thermal resistivity of the cabinet was chosen as a reasonable design change that could be obtained through the implementation of evacuated panels or increased foam insulation thicknesses. The improvement in cabinet insulation was simulated by adding expanded polystyrene panels to the top and side exterior surfaces of the baseline refrigerator-freezer cabinet. No additional insulation was added on the bottom of the fresh food compartment due to the limited space around the condenser. Panels of different thicknesses were used on the freezer and fresh food compartment walls to maintain a similar freezer-to-total-cabinet-load (Q_{FRZ}/Q_{TOT}) ratio as the baseline cabinet. Experimental results based on data from the reverse heat loss experiments along with Q_{FRZ}/Q_{TOT} ratios for a refrigerator-freezer operating with a 5°F (-15°C) freezer and a 38°F (-3.3°C) fresh food compartment in a 90°F (32.2°C) ambient are summarized in Table 2. Also tabulated are the modeled results, obtained by entering cabinet dimensions, foam thicknesses, and thermal resistivities into the cabinet portion of the model. The experimental results using the reverse heat loss method indicate that the cabinet heat loss was reduced 6% (239.1 to 225.4 Btu/h) by adding 2 inches of insulation to the freezer section and 1 inch to the fresh food section. The modeled results, while showing good overall agreement (246.0 versus 239.1 Btu/h) for the base case, indicate that the additional insulation should have reduced the heat loss by 16% (246.0 to 205.7 Btu/h).

In an attempt to verify which results more accurately reflected the effects of the additional insulation, energy consumption tests (Table 3) were conducted for the unit with and without additional insulation. The model was then used to calculate the cabinet heat load required to achieve the measured energy consumption. The results, given in Table 2, indicate that the cabinet heat load from the energy consumption test method (232.1 Btu/h) is closer to that using the reverse heat loss method (225.4 Btu/h) than the modeled cabinet heat load (205.7 Btu/h). Thus, simply adding panels to the exterior is an ineffective method for reducing the cabinet heat load since it results in less reduction than would be expected from the additional thermal resistivity of the expanded polystyrene panels. Apparently, the original metal exterior of the refrigerator-freezer, which cannot be covered with insulation at the flange where the freezer and fresh food doors seal, acts as a fin to conduct heat out of the cabinet and thereby partially defeats the additional insulation. This explanation is supported by temperature measurements made on the exterior before and after insulation was added.

A second attempt at achieving a 10% cabinet heat load reduction was made by adding 4 inches of insulation to the freezer section and 2 inches to the fresh food section. The cabinet heat load was verified in a similar manner as previously described. The results indicate that a 9% (246.0 to 224.1 Btu/h) heat loss reduction was achieved, which was considered close enough to the modeled conditions shown in Figure 2 to proceed with the experimental plan. It is noted that the insulation thicknesses are way beyond what would be allowable in a conventional design, however, the cabinet heat loss reductions are achievable with some forms of advanced insulation.

Compressor Calorimeter Results

Nine-point calorimeter tests were used to determine the performance over a range of operating temperatures for the CFC-12 compressor used in the baseline refrigerator-freezer and the HFC-134a rated-speed compressor used in the modified units. The resulting compressor maps, shown graphically in Figure 4, were used as inputs for the modeling analysis. From the data in Figure 4, 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 CFC-12 compressor is 5.42 while that of the HFC-134a rated-speed compressor is 5.95. Thus, while the HFC-134a rated-speed compressor achieved a 10% increase in efficiency, relative to the baseline CFC-12 compressor, it was less efficient than the 6.10 EER compressor used for the initial modeling simulation to determine the technology path.

The refrigeration capacity of the HFC-134a rated-speed compressor was approximately 628 Btu/h or 21% less than the CFC-12 compressor (792 Btu/h) it replaced. The reduced capacity was required to achieve reasonable run times once additional insulation was added to the cabinet exterior. Using a compressor whose capacity is much greater than the load would have resulted in short, frequent compressor runs that increase system cycling losses. In addition, since run times for the baseline unit were slightly lower than normal, an attempt was made to further reduce cycling losses by decreasing the capacity of the HFC-134a rated-speed compressor more than would have been required for the reduced cabinet heat load alone.

System Results

Of the nine options investigated for reducing the energy consumption of the refrigerator-freezer, only five were incorporated into the design of the modified unit. The five selected were liquid-line off-cycle control, improved cabinet insulation, high-efficiency compressor, high-efficiency fan motors, and enhanced evaporator heat transfer. The other modifications were not addressed due to the difficulties experienced from incorporating them into a commercially-manufactured cabinet without destroying the original cabinet integrity. One other modification, a more thermodynamically efficient refrigerant, is still under investigation to determine the best candidate for testing.

Following the addition of insulation to the cabinet exterior and assembly of the compressor, evaporator, fans, and liquid-line shut-off valve into the baseline unit, energy consumption tests were performed in accordance with section 8 of the AHAM Standard for Household Refrigerators and Household Freezers (AHAM 1985). The results, Table 3, show that the energy consumption was reduced from 1.80 kWh/d to 1.41 kWh/d, a 22% savings. In relation to the NAECA standard, the results represent a 32% improvement in energy consumption (2.06 to 1.41 kWh/d). Modeled results for the modified unit indicate that the energy consumption should have been reduced to 1.18 kWh/d. In a previous study, the model was shown to have good agreement with experimental results (Sand et al. 1994). Thus, the discrepancy of 0.23 kWh/d between the model and actual experimental results was larger than expected. Two explanations are offered for the discrepancy; overestimated liquid-line shut-off valve energy savings and suboptimal capillary tube length.

In separate tests that investigated the effects of a liquid-line shut-off valve, there was no change in energy consumption when a valve was used. In an effort to confirm the results, discussions with refrigerator-freezer manufacturers revealed that only minimal energy savings have been realized when liquid-line shut-off valves are used to prevent refrigerant migration to the evaporator during the off-cycle. If one assumes that the valve yields no savings, then half of the discrepancy would be accounted for by zeroing out the modeled energy savings of 0.12 kWh/d.

The suboptimal capillary tube length is the result of unsuccessful attempts to lengthen the capillary tube to accommodate the change from refrigerant CFC-12 to HFC-134a. The existing capillary tube was embedded in the foam, therefore making it impossible to fabricate a capillary tube manifold and route it down through the foam in the rear of the cabinet. A second option, extending the length of the existing capillary tube, improved performance but failed to achieve an optimal balance with the volumetric capacity of the compressor, suction pressures, and the charge size in this unit. Superheat measurements at the evaporator exit near the end of a compressor on-cycle were used to adjust the refrigerant charge size. It is apparent that there is a mismatch between the capillary and the volumetric capacity of the compressor when an acceptable evaporator superheat and suction pressure cannot be obtained by adjusting the charge size.

CONCLUSIONS

Energy consumption tests which were performed to determine the cabinet heat load indicate that the power was reduced by 8% (1.80 to 1.65 kWh/d, Table 3) from the addition of insulation to the cabinet exterior. The results are believed to be indicative of what is technically achievable given new technologies, such as vacuum insulation, which are on the horizon. It is noted, however, that new insulations are unproven in terms of long term reliability and heat transfer degradation over time; two factors which must be addressed. In addition, the effects of door openings and food loadings, which are not a part of the closed door test, will become a more significant part of the load in the future as cabinet heat gain is reduced. Thus, the 90° F (32.2° C) closed door test procedure will most likely become less representative of actual refrigerator-freezer field energy use. Insulation improvements also require refrigeration systems with smaller capacity compressors to achieve a balance between thermal load and refrigerating capacity. Since the efficiency of refrigerator-freezer compressors tends to fall off sharply at lower capacities, there will eventually be a point at which insulation improvements will result in higher energy consumption.

Experimental implementation of some of the more easily tested design improvements, identified by the modeling analysis, resulted in a 22% improvement in energy savings over the baseline cabinet. While the results did not meet the project goal of 1.00 kWh/d, they were still impressive, especially when viewed in relation to the NAECA standard, where the 1.41 kWh/d energy consumption yielded a 32% improvement over the 1993 target for 20 ft^3 (570 l) automatic-defrost, top-mount refrigerator-freezers.

Future work on the project will concentrate on achieving the project goal with a second generation prototype. The prototype will use vacuum panel technology instead of exterior panels to achieve a reduced cabinet heat load. Refrigeration system changes will include an increased condenser area, compressor improvements to marginally improve EER, and a low-glide refrigerant mixture.

CLOSING REMARKS

American manufacturers of domestic refrigerator-freezers have established an enviable record of consistent improvements in the energy efficiency of their product. Wide-spread use of this appliance as a result of its efficiency, convenience, and reliable performance have made it a target for additional refinement, but, clearly, the margins for improving performance are reaching a point of diminishing returns. Switching to a design that performs well in standardized energy-consumption tests, but sacrifices many of the convenient and dependable features of this essential appliance could be a mistake for an established industry which supports a highly-skilled work force.

Clearly, there is a rationale for retaining many familiar aspects of a product design that has been refined and used for thirty years. However, some changes are needed to accommodate refrigerants and foam blowing agents that are more environmentally acceptable. Many of the design options that could have a significant effect on the energy use of a refrigerator-freezer have been clearly identified and are technologically available. In virtually every instance, however, substitution of components with improved efficiency is accompanied by increases in unit hardware cost. For example, replacing alternating current fan motors with electrically commutated DC motors results in a two to threefold improvement in component operating efficiency (and a corresponding drop in added load for the freezer). The corresponding cost increase, however, is more like of factor of three or four (EPA 1993). In addition, a proven product is being replaced with one that may result in additional service and warranty expenses.

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

OPTION NUMBER	EFFICIENCY OPTION
Option 1	High-efficiency compressor substitution
Option 2	Increased evaporator size, surface enhancement, improved heat transfer
Option 3	Improved refrigerant
Option 4	Increased condenser size, surface enhancement, improved heat transfer
Option 5	Improved suction line heat transfer
Option 6	Reduced door gasket losses
Option 7	Improved cabinet and door insulation
Option 8	High-efficiency fan motors
Option 9	Liquid-line off-cycle control

Table 2. Results Summary of Reverse Heat Loss Tests and Modeling -90°F ambient, 5°F Freezer, 38°F Fresh Food Compartment

	$Q_{Freezer}$ (Btu/hr)	$Q_{Fresh\ Food}$ (Btu/hr)	Q_{Total} Cabinet (Btu/hr)	Q_{FRZ}/Q_{Total}
Base Cabinet:				
Experimental:				
-Reverse Heat Loss	131.8	107.3	239.1	0.55
Modeled:	143.1	102.9	246.0	0.58
Cabinet w/2" Freezer + 1" Fresh Food Insulation:				
Experimental:				
- Reverse Heat Loss	115.8	109.6	225.4	0.51
- Energy Consumption	126.8	105.3	232.1	0.55
Modeled:	113.3	92.4	205.7	0.55
Cabinet w/4" Freezer + 2" Fresh Food Insulation:				
Experimental:				
-Energy Consumption	122.6	101.5	224.1	0.55

Table 3. Energy Consumption Tests — Experimental and Analytical Results/Design Options

Case	Description	Experimentally Measured	Analytically Calculated	Manufacturers Data
A	Baseline Units Energy Run Time	 1.80 kWh/d 41.3%	 1.79 kWh/d 41.0%	 1.76 kWh/d 40.3%
B	Case A + 2" Freezer + 1" Fresh Food Insulation Energy Run Time	 1.70 kWh/d 38.6%	 1.52 kWh/d 35.1%	 — —
C	Case B + 4" Freezer + 2" Fresh Food Insulation Energy Run Time	 1.65 kWh/d 37.4%	 1.47 kWh/d 33.8%	 — —
D	Case C + 6.0 EER Compressor, High Efficiency Fans, Larger Evaporator, and Liquid-Line Shut-Off Valve Energy Run Time	 1.41 kWh/d 42.3%	 1.18 kWh/d 43.0%	 — —

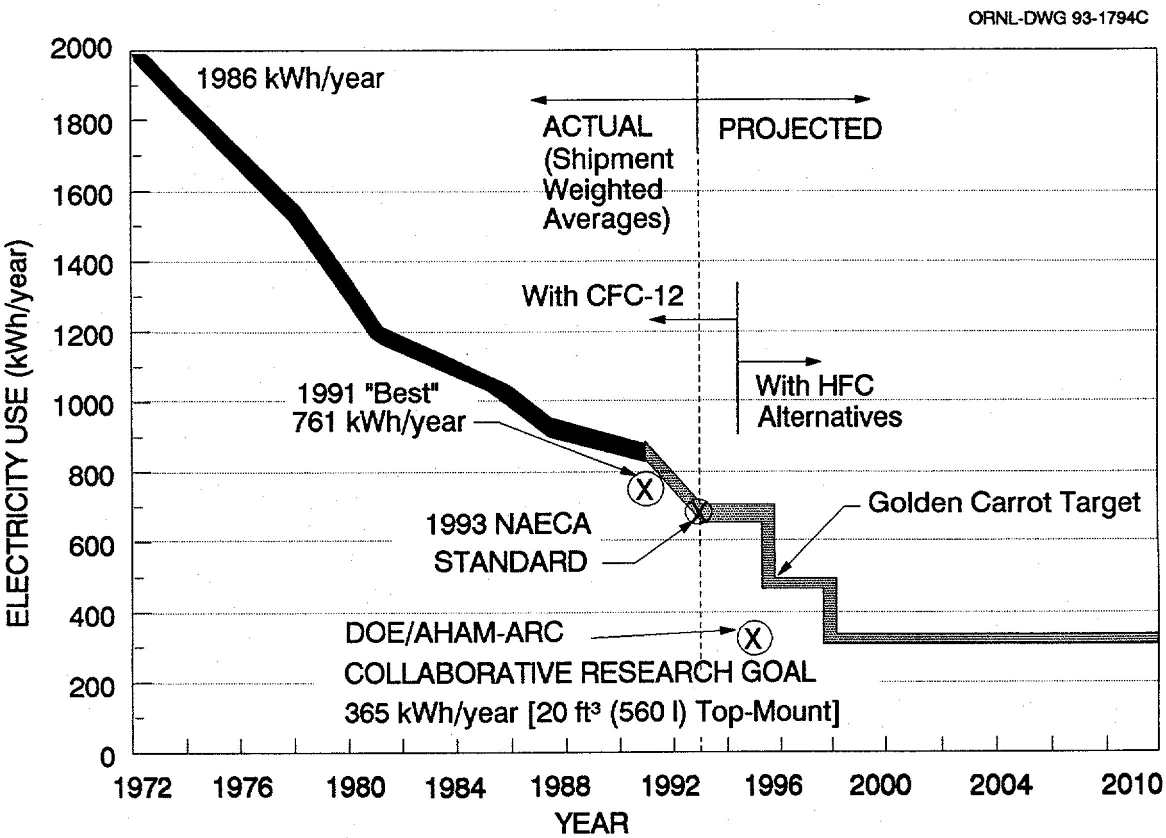

Fig. 1. Historical energy use for refrigerator-freezers.

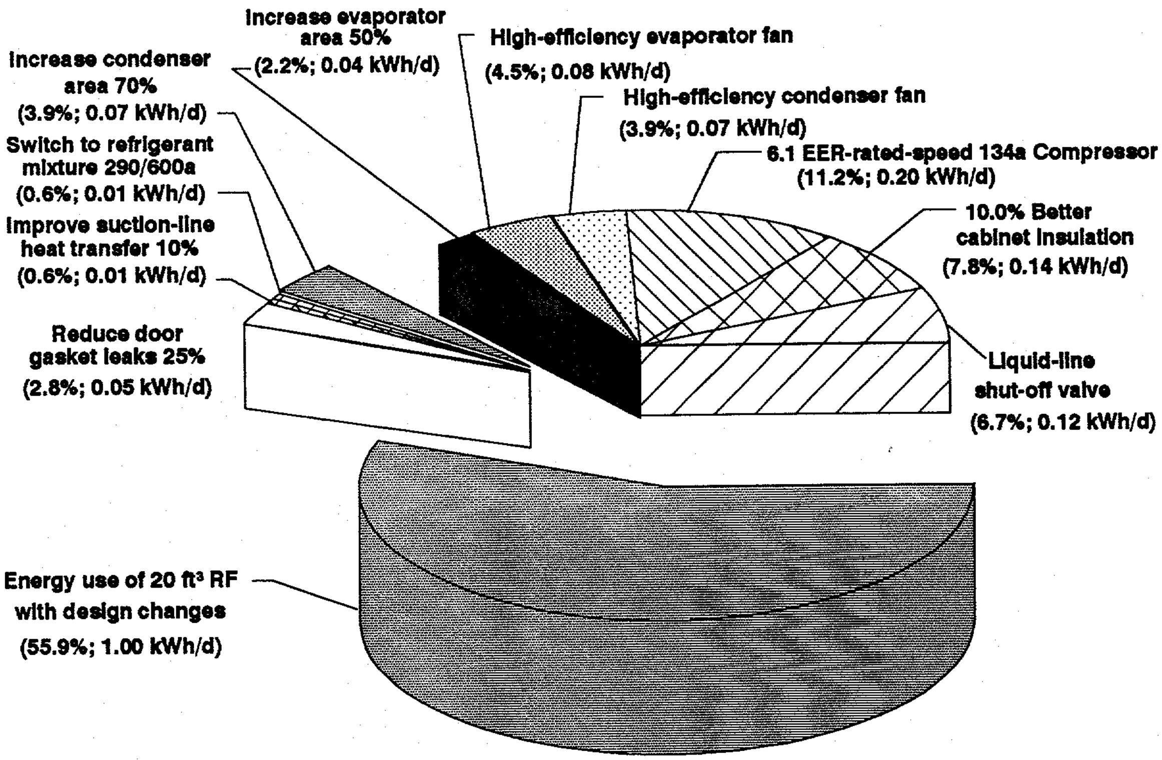

Fig. 2. Modeled energy savings from design changes - 1993 baseline unit, 1.79 kWh/d.

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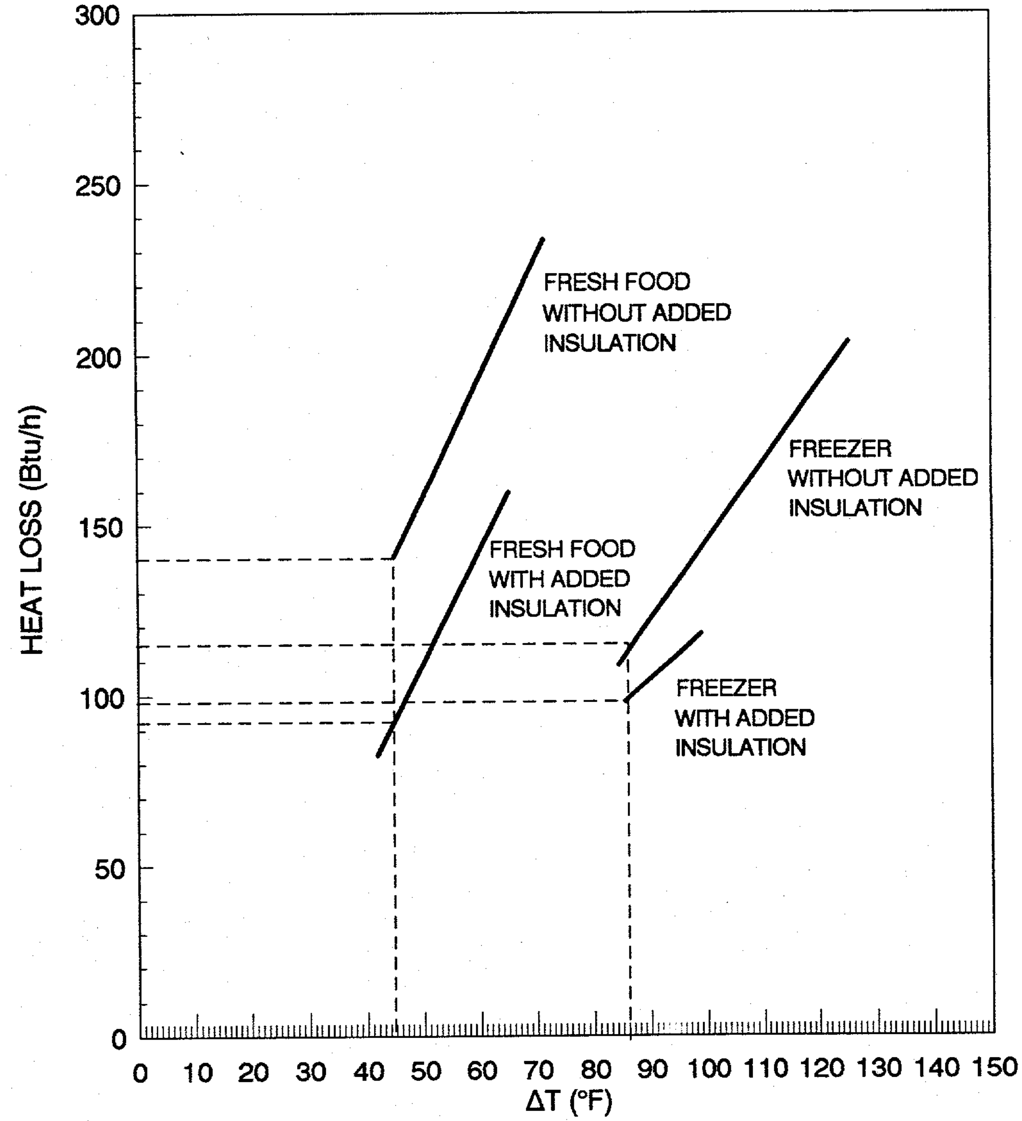

Fig. 3. Cabinet heat loss measurements for cabinet with and without added insulation.

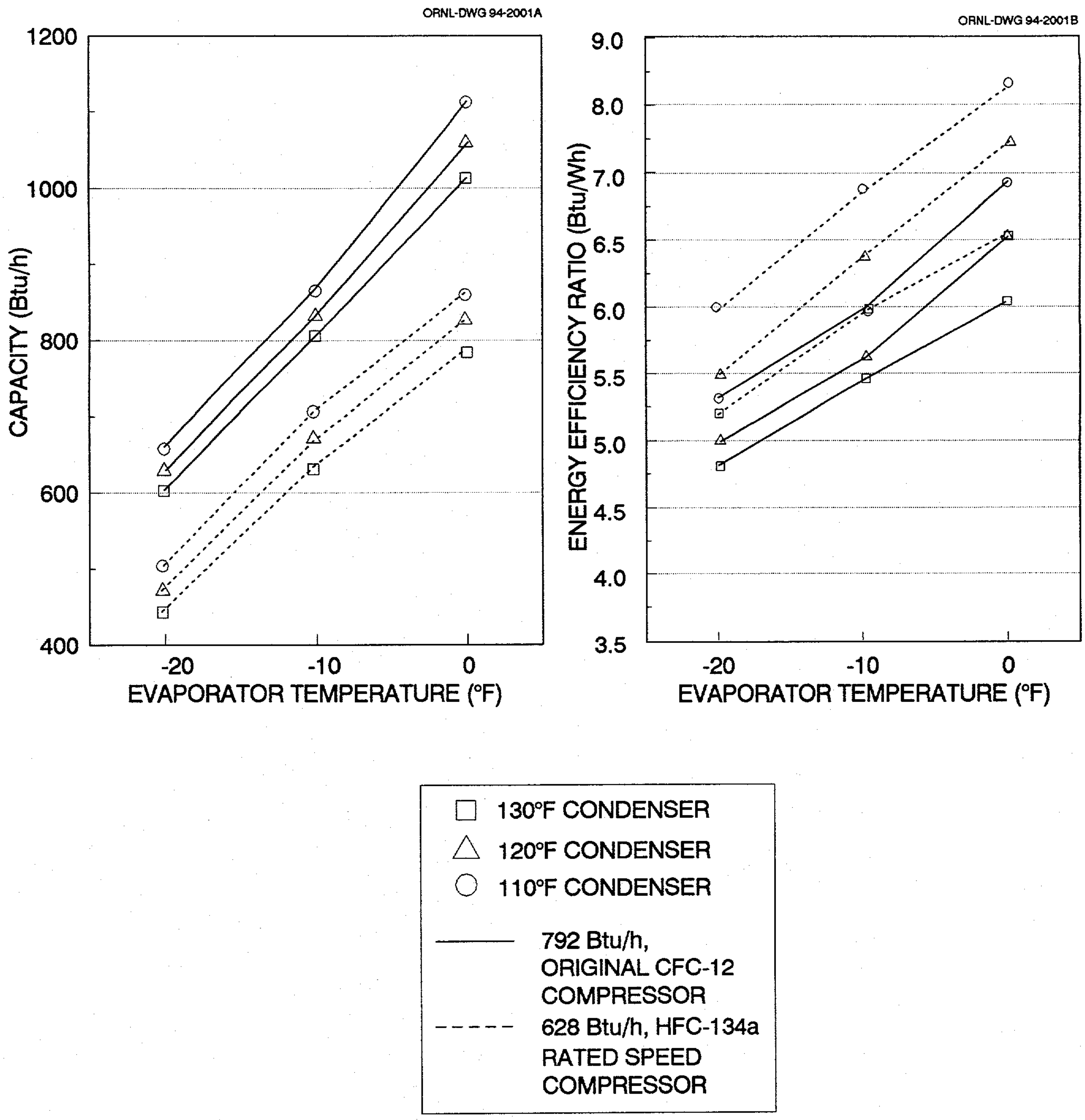

Fig. 4. Compressor calorimeter results