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LOCKHEED MARTIN

DOE/AHAM ADVANCED REFRIGERATOR TECHNOLOGY DEVELOPMENT PROJECT

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EXECUTIVE SUMMARY

In 1991, a cooperative research and development agreement was signed between Lockheed Martin Energy Research, Inc., under its U.S. Department of Energy contract and the Appliance Research Consortium to develop and test a new generation of refrigerator-freezers that are energy-efficient and that replace the R-12 and R-11 compounds traditionally used as refrigerants and foam-blowing agents with alternatives that are better for the environment. The goal of the project was to achieve an energy consumption level of 1.00 kWh/d, a 50% reduction compared with the 1993 National Appliance Energy Conservation Act (NAECA) standard for 20-ft^3 (570-L) units. The goal was established by a joint industry-government advisory committee in an effort to strengthen U. S. competitiveness by lowering the energy consumption of domestically manufactured refrigerator-freezers to meet 1998 energy and environmental targets.

The project consisted of three main phases: (1) an evaluation of energy-efficient design options using computer simulation models and experimental testing, (2) design and testing of an initial prototype unit, and (3) energy and economic analyses of a final prototype. The final prototype achieved an energy consumption level of 0.93 kWh/d, which is a 54% energy savings compared with the 1993 NAECA standard for 20-ft^3 (570-L) units. Assuming the manufacturer's cost ($134) would be doubled for the consumer, the cost analysis determined that it would require approximately 11.4 years to pay for the design changes. Since the payback was too long, a second, more cost-effective design was also tested. The less costly version used 1.16 kWh/d (a 42% savings) at a manufacturer's cost increase of $53. The payback for this unit would be 6.6 years.

PHASE 1: EVALUATION OF DESIGN OPTIONS

A detailed system computer model was used to evaluate the energy savings of several design modifications that collectively could achieve the project goal. The options investigated included a 5.2 energy efficiency ratio (EER) compressor; lower-wattage fan motors; cabinet heat load reductions; liquid-line shutoff valves; and elimination of the defrost energy, anti-sweat heaters, and door gasket leaks. The modeled results showed that the energy consumption would be reduced from 2.87 kWh/d for the baseline 1990 model to 1.08 kWh/d for the improved unit. Experimental tests were performed with a unit that had polystyrene foam panels attached to the cabinet exterior to simulate additional insulation thicknesses or vacuum panel insulation. The results from reverse heat loss tests indicate that the cabinet heat leak rate was reduced from 291.1 Btu/h to 236.9 Btu/h, a 19% improvement. The improvement in energy consumption for the additional insulation was determined to be 8% (3.00 to 2.77 kWh/d). Other design improvements, such as a 5.2 EER compressor, a 4-W evaporator and condenser fans, and a liquid-line shutoff valve, reduced the energy consumption to 2.07 kWh/d, a 31% improvement over the baseline unit.

PHASE 2: DESIGN AND TESTING OF INITIAL PROTOTYPE

In this phase of the project, cabinet and refrigeration system changes were made to a 1993 model refrigerator-freezer. Since the unit was required to meet the new 1993 NAECA standards, the energy consumption was quite low (1.80 kWh/d), making further reductions 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. Design options were modeled by adding them sequentially to the baseline configuration so that the cumulative effects could be accounted for.

Based on the results of the modeling study, insulation improvements, a high-efficiency compressor, lower-wattage fans, a larger evaporator, and a liquid line shutoff valve were added to the baseline unit. Using the reverse heat loss test procedure, the insulation improvements to the cabinet resulted in reduction of the heat loss rate from 246.0 to 224.1 Btu/h, a 9% reduction. Energy consumption tests showed that energy use was reduced from 1.80 to 1.65 kWh/d with the additional insulation. The HFC-134a high-efficiency compressor chosen for testing had an EER of 5.95 with a capacity of 621 Btu/h. This was a higher EER than that of the R-12 baseline compressor (5.42 EER) and a significantly lower capacity (792 Btu/h) to account for the lower cabinet heat loss rate. When the compressor and additional refrigeration system improvements were added to the cabinet with additional insulation, the energy consumption was further reduced from 1.65 to 1.41 kWh/d, a 22% reduction in energy consumption compared with the baseline unit and 30% below the NAECA standard for 20-ft^3 (570-L) units.

PHASE 3: ENERGY AND ECONOMIC ANALYSES OF FINAL PROTOTYPE

The third phase of the project concentrated on achieving the targeted goal of 1.00 kWh/day and determining the cost-effectiveness of the design changes. For this phase, a 1996 model refrigerator-freezer with an energy consumption of 1.68 kWh/d was selected as the baseline unit. Reverse heat loss tests were used to determine the most cost-effective door insulation improvements. The results indicated that thick (2-in.) doors were more cost-effective than doors with vacuum panels. Heat loss measurements were also determined for two cabinets, one fitted with vacuum panels around the freezer section and the other, a standard cabinet. The results showed that the cabinet with vacuum insulation achieved a 15% reduction in the cabinet heat loss rate.

Following the heat loss rate tests, the energy consumption was determined for the vacuum cabinet with a high-efficiency compressor, a low-wattage condenser fan, a larger evaporator, 2-in.-thick doors, and adaptive defrost control. Using the 90°F closed-door test procedure, the energy consumption was 0.93 kWh/d, a 45% reduction from the baseline unit and 54% lower than NAECA standards. A cost analysis of the unit, using an average cost for electricity of $0.0867/kWh, showed that the payback period would be 11.4 years, which is unacceptable.

A breakdown of the paybacks for the individual design changes showed that the vacuum panel insulation around the freezer and the larger evaporator had the highest payback periods of any of the changes. Based on that information, a second unit was assembled without vacuum insulation and the larger evaporator. The energy consumption of that unit was 1.16 kWh/d, 31% lower than the baseline and 42% below the NAECA standard. The payback for the less costly version would be 6.6 years. The payback for this unit could be reduced even more by using a compressor in the 5.2 to 5.3 EER range. The energy consumption would increase to 1.250 kWh/d, 38% lower than the NAECA standard, and would save 155 kWh/year for a cost savings of $13.44 annually. The payback would be less than 3 years.

ABSTRACT

As part of the effort to improve residential energy efficiency and reduce greenhouse emissions from power plants, several design options were investigated for improving the energy efficiency of a conventionally designed domestic refrigerator-freezer. The program goal was to reduce the energy consumption of a 20-ft^3 (570-L) top-mount refrigerator-freezer to 1.00 kWh/d, a 50% reduction from the 1993 National Appliance Energy Conservation Act (NAECA) standard.

The options—such as improved cabinet and door insulation, a high-efficiency compressor, a low-wattage fan, a large counterflow evaporator, and adaptive defrost control—were incorporated into prototype refrigerator-freezer cabinets and refrigeration systems. The refrigerant HFC-134a was used as a replacement for CFC-12. The baseline energy performance of the production refrigerator-freezers, along with cabinet heat load and compressor calorimeter test results, were extensively documented to provide a firm basis for experimentally measured energy savings.

The project consisted of three main phases: (1) an evaluation of energy-efficient design options using computer simulation models and experimental testing, (2) design and testing of an initial prototype unit, and (3) energy and economic analyses of a final prototype.

The final prototype achieved an energy consumption level of 0.93 kWh/d—an improvement of 45% over the baseline unit and 54% over the 1993 NAECA standard for 20-ft^3 (570-L) units. The manufacturer's cost for those improvements was estimated at $134; assuming that cost is doubled for the consumer, it would take about 11.4 years to pay for the design changes. Since the payback period was thought to be unfeasible, a second, more cost-effective design was also tested. Its energy consumption level was 1.16 kWh/d, a 42% energy savings, at a manufacturer's cost increase of $53. Again assuming a 100% markup, the payback for this unit would be 6.6 years.

Substantial improvements in RF efficiency were demonstrated with relatively minor changes in system components and refrigeration circuit design. However, each improvement exacts a penalty in terms of increased cost or system complexity/reliability. These factors must be taken into account by consumers when considering the benefits of improved efficiency and environmental gains from reducing greenhouse gases and global warming.

1. INTRODUCTION

Domestic refrigerator-freezers (RFs), 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). Increased demand is expected to drive up the production of RFs substantially in the near future, especially in developing countries where annual growth on the order of 10 to 15% is expected for the next few years. To protect the environment, it is imperative that industry incorporate environmentally safe refrigeration systems and cabinet designs while increasing energy efficiency. This task will be particularly challenging because today's energy-efficient designs were made possible through the use of chlorofluorocarbons (CFCs), which are now being phased out.

Reports indicating that long-lived CFCs and bromine-containing compounds released to the atmosphere exacerbate stratospheric ozone loss, and the resulting Montreal Protocol revisions, make it apparent that the world community will eventually prohibit the use of these chemicals as refrigerants or foam blowing agents (Zurer 1993). Such restrictions will prohibit use of R-12 as a refrigerant and R-11 as a foam blowing agent in RFs manufactured after 1995. At the same time, scientific data released by the United Nations Environment Programme and the World Meteorological Organization show carbon dioxide to be the main contributor to global warming (UNEP 1991). For domestic RFs using alternative refrigerants such as HFC-134a, the indirect carbon dioxide emissions resulting from power production contribute about 100 times more to the global warming potential than does the refrigerant alone (Fischer et al. 1991). Therefore, efforts are being made to produce RFs that use less energy.

In addition to these environmental concerns, RFs are regulated under the 1987 National Appliance Energy Conservation Act (NAECA) and are required to meet certain minimum energy-efficiency standards set up and periodically revised by the U.S. Department of Energy (NAECA 1987). In the next revision, originally scheduled for 1998 and delayed to 2001, the standards will require an additional 30% reduction in energy usage. Figure 1 shows actual and projected improvements in the electrical energy usage of RFs and indicates the stated goal for this collaborative industry/government research on RFs.

Customer expectations and competitive pressures impose an additional unwritten set of constraints on RFs produced in the United States. Makers of mass-produced kitchen cabinetry anticipate certain external RF dimensions; kitchen floors are built to support a limited weight density; a 15–20 year useful product lifetime is assumed; and plug-in-and-forget operation is the norm. Additionally, a low selling price is an important feature for an appliance that has become a commodity item, despite the importance of RFs in an industrialized society that must be able to refrigerate food efficiently and effectively. Therefore, unless consumers are motivated to spend more for efficiency, further improvements in efficiency will be hard for manufacturers to justify based on existing market conditions. External forces such as rebates, new selling techniques, or standards will be required to reduce RF energy consumption further below existing levels.

Obviously, energy-efficient RFs that are manufactured with and use environmentally acceptable chemicals must be developed. Because there are approximately 115 million refrigerators in the United States which annually consume about 2 quads[1] of primary energy, 12% of the total residential energy budget, substantial improvements in domestic RF efficiency extended over the 15-20 year life of the appliance would significantly benefit national goals for energy conservation and environmental progress.

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[1] One quad = 10^{15} Btu (2.9×10^{11} kWh)

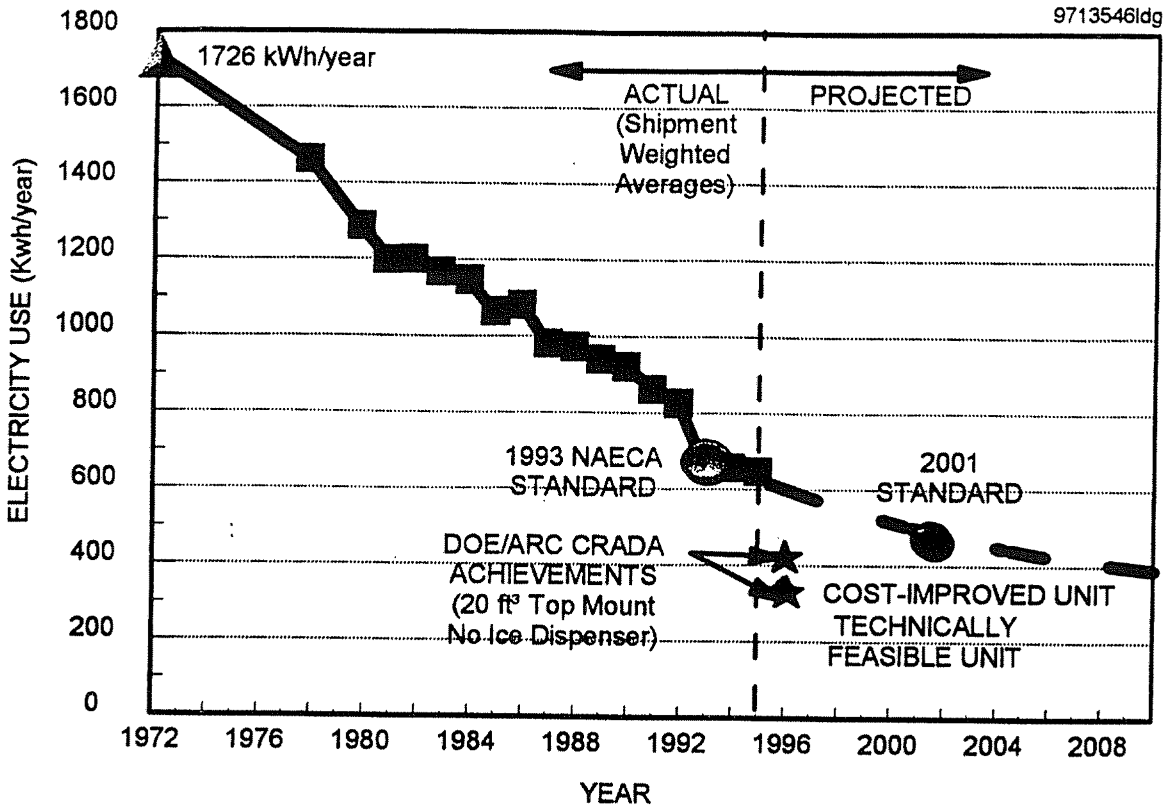

Fig. 1. Historical energy usage for refrigerator-freezers.

This report describes results from an industry/government cooperative research and development agreement (CRADA) to evaluate and test design concepts for an advanced, energy-saving domestic refrigerator. The stated goal of this CRADA was to demonstrate, by 1995, RF technology that requires 50% or less of the 1993 NAECA standard energy consumption. For a 20-ft^3 (570-L) refrigerator with a top-mounted freezer and no through-the-door features, this consumption level is equal to or less than 1.00 kWh/d. The general objective of the research is to facilitate development of efficient appliances through establishing which design changes can conveniently be incorporated into products that are cost-effective and reliable.

2. PHASE 1: EVALUATION OF DESIGN OPTIONS

A detailed system computer model was used to evaluate the energy savings of several design modifications that collectively could achieve the project goal. The options investigated included a 5.2 energy efficiency ratio (EER) compressor; lower-wattage fan motors; cabinet heat load reductions; liquid line shutoff valves; and elimination of the defrost energy, anti-sweat heaters, and door gasket leaks. The modeled results showed that the energy consumption would be reduced from 2.87 kWh/d for the baseline 1990 model to 1.08 kWh/d for the improved unit.

2.1 PROJECT SCOPE AND TESTING PROTOCOL

The research project was conceived to be primarily experimental with a firm basis in analytical appraisal. A widely distributed computer model that combines a cabinet heat load model, a refrigeration circuitry model, and an on/off cycling algorithm was used to analytically evaluate design changes (EPA 1993). The model was sufficiently versatile to accommodate system hardware and refrigerant changes in terms normally found in more empirically specific simulation models used by appliance manufacturers. It allowed us to evaluate the energy saving opportunities of options not generally available as hardware components, in a manner that assesses each one individually against the baseline performance. Commercially available production refrigerators were used as laboratory breadboards on which modifications were made. The top-mount configuration was chosen as the most typically purchased model. Experimental test procedures implemented in the program included reverse cabinet heat loss measurements, standard nine-point RF compressor calorimeter mappings, and the standard 90°F (32.2°C) closed-door energy-consumption test (AHAM 1985).

Because improved cabinet insulation was one of the energy saving options to be investigated, reverse heat loss measurements were made to assess baseline and improved cabinet thermal resistivity. In this procedure, an RF cabinet was placed in a cold chamber with a controllable, monitored source of heat in both the freezer and fresh food compartments. Shielded light bulbs and small electrical chassis cooling fans were used as individual heat sources. An attempt was made to obtain temperature differences across the cabinet walls comparable to those prescribed in the 90°F (32.2°C) closed-door test when the refrigerator works to maintain cold internal temperatures in a hot room. The main premise of this test procedure is that, at thermal equilibrium when steady state temperatures are obtained, the rate of heat addition into the cabinet is equal to the rate at which it is leaking out of the walls, door, and gaskets into the cold room.

Changing the total cabinet heat load and substituting a chlorine-free refrigerant for the R-12 necessitated corresponding changes in the capacity and possibly the design of the compressor and other refrigeration circuit hardware. Original and replacement compressors used for these laboratory breadboard units were tested using a nine-point compressor calorimeter procedure to generate RF compressor "maps." In this procedure, compressor operating characteristics, including refrigeration capacity and energy efficiency ratios, were 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. These nine-point maps were input to the RF computer model for analytical evaluation. The RF model will also accept compressor performance data as either a single-point calibration at the 130°F (54.4°C) condensing and -10°F (-23.3°C) evaporating calorimeter operating condition, or a physically based model that integrates the physical dimensions and operating parameters of a compressor to obtain an estimate of isentropic and volumetric efficiency.

Integrated system performance, before and after modifications were made to the laboratory breadboard units, was assessed using the standard 90°F (32.2°C) closed-door test. In this procedure the refrigerator 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 at each control setting. Energy use and compartment temperatures are measured from the onset of one RF defrost cycle to the beginning of the next defrost. The resulting test points are used to calculate the RF energy consumption over a 24-hour (1-day) period based upon an ideal 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 V to the refrigerator and an air circulation rate of less than 50 ft/min (15 m/min) in the environmental chamber. The high ambient test temperature, 90°F (32.2°C) is used to offset the contribution of door openings and food loading to the RF operating burden. Comparisons of field performance to closed-door test ratings indicate the laboratory procedure is a valid indication of energy use in field service (Meier and Jansky 1993).

Because commercially manufactured refrigerators were used as laboratory test beds, a sequence was established of "as-received" tests followed by tests of units with improved insulation and the original refrigeration circuit. Finally, improved cabinets with an optimized refrigeration circuit or cycle were tested. Baseline or starting-point performance was carefully documented to avoid any ambiguity about measured improvements resulting from design or hardware changes.

2.2 EXPERIMENTAL PLAN

An experimental plan was formulated to help order and prioritize laboratory work. Two phases were anticipated. Phase 1 emphasized changes in hardware and refrigerants that could be incorporated into the conventional RF design (a single fan-forced evaporator, single condenser, and single-speed compressor) operating with a pure refrigerant or nearly azeotropic refrigerant blend [< 6°F (3°C) separation of dew and bubble points at evaporator pressures]. Some cold air from the freezer was diverted to cool the fresh food compartment. Changes centered on this conventional design were considered to be more acceptable to manufacturers because they represented more conventional hardware and control algorithms, less rearranging of components, and greater reliability. A given about the Phase 1 options was that R-12 would be replaced with a non–ozone-depleting refrigerant in all of the experimental evaluations.

Phase 1 Options

Options for improving conventional RF efficiency fall into four general areas: improving the refrigeration cycle efficiency, decreasing the cabinet heat load, reducing parasitic electrical loads, and reducing on/off cycling losses (see Table 1) (Bohman 1987; Turiel and Heydari 1988). Options 1 through 4 in Table 1 deal primarily with improving the thermodynamic efficiency of the refrigeration cycle. Options 5 and 6 have a direct influence on reducing the cabinet heat load. Adaptive defrost, efficient fan motors, and liquid line flange heat (options 7–9) all affect the parasitic electrical load but also have an impact on the cabinet heat load. Options 7–9 have the potential for reducing incidental contributions to the refrigeration load, but the effect of option 9 may be to decrease the electrical load while increasing the net heat load that must be refrigerated.

Table 1. Phase 1 options

Phase I: Efficiency options	
Option 1	High-efficiency compressor substitution
Option 2	Evaporator/condenser size, surface enhancement, improved heat transfer, air flow
Option 3	Improved refrigerant
Option 4	Improved condenser/compressor cooling component location
Option 5	Enhanced cabinet and door insulation
Option 6	Reduced door gasket losses
Option 7	Adaptive defrost
Option 8	High-efficiency fan motor
Option 9	Liquid line flange heat for anti-sweat
Option 10	Liquid line off-cycle control

Option 10, a liquid line shutoff valve, which has been used extensively with rotary compressors, prevents migration of hot refrigerant to the evaporator during the compressor off-cycle, thereby reducing on/off cycling losses. Depending on their motivating force, these valves may contribute to the electrical load.

2.3 MODELED ENERGY SAVINGS RESULTS

The modeling program was used to evaluate the energy-saving potential of several of these Phase 1 design options. Table 2 lists the baseline and new model inputs used to obtain the results shown graphically in Fig. 2. Because built-up units were used as experimental breadboards, some design changes *could not* be retrofitted into the cabinets conveniently without compromising cabinet construction and original insulating integrity. Figure 2 indicates two sectors of modeled energy savings calculated for indicated hardware and operational changes on a 1990 vintage 20-ft^3 (570-L) RF. The results from "modeled only" changes are shown in the upper left sector of Fig. 2. As Table 2 indicates, the maximum energy impacts associated with reducing door gasket leaks, improving defrost control, and avoiding the use of electrical resistance heat for preventing condensate at the door gaskets were estimated by "zeroing out" the program inputs dealing with each option. This was done to assess the greatest energy savings attributable to improvements in those features. It is not meant to imply that those loads can be totally eliminated.

The other sector of energy savings shown in Fig. 2 shows modeled results calculated for reasonable improvements to components used in the original 1990 design. For example, replacing the original 4.43 EER compressor with one rated at 5.3 EER resulted in a 12.4% (0.37 kWh/d) energy savings. Changing the compressor, improving cabinet insulation by 18–20%, using electrically commutated direct-current fan motors, and installing a liquid line shutoff valve are all changes that could be implemented experimentally with available components and a minimum of cabinet deterioration. Specifications of obtainable components were used as inputs to the model for the simulation calculations.

Table 2. Model inputs for Phase I changes

	Baseline unit (initial inputs)	**Improved unit (energy saving change)**
Improved compressor	4.43 EER (R-12)	5.3 EER (R-134a)
Evaporator fan motor	10.0 W	4.0 W
Condenser fan motor	13.5 W	8.0 W
Cabinet load (@ 90°F) ($Q_{Freezer}/Q_{Total}$)	292.6 Btu/h (0.51)	250.8 Btu/h (0.52)
Liquid line shutoff	No	Yes
Defrost energy (electrical and thermal load)	0.11 kWh/d	0.00 kWh/d
Anti-sweat heat	0.11 kWh/d	0.00 kWh/d
Door gasket heat transfer	0.04 Btu/ft °F (w/o fan) 0.07 Btu/ft °F (w/fan)	0.000 Btu/ft °F

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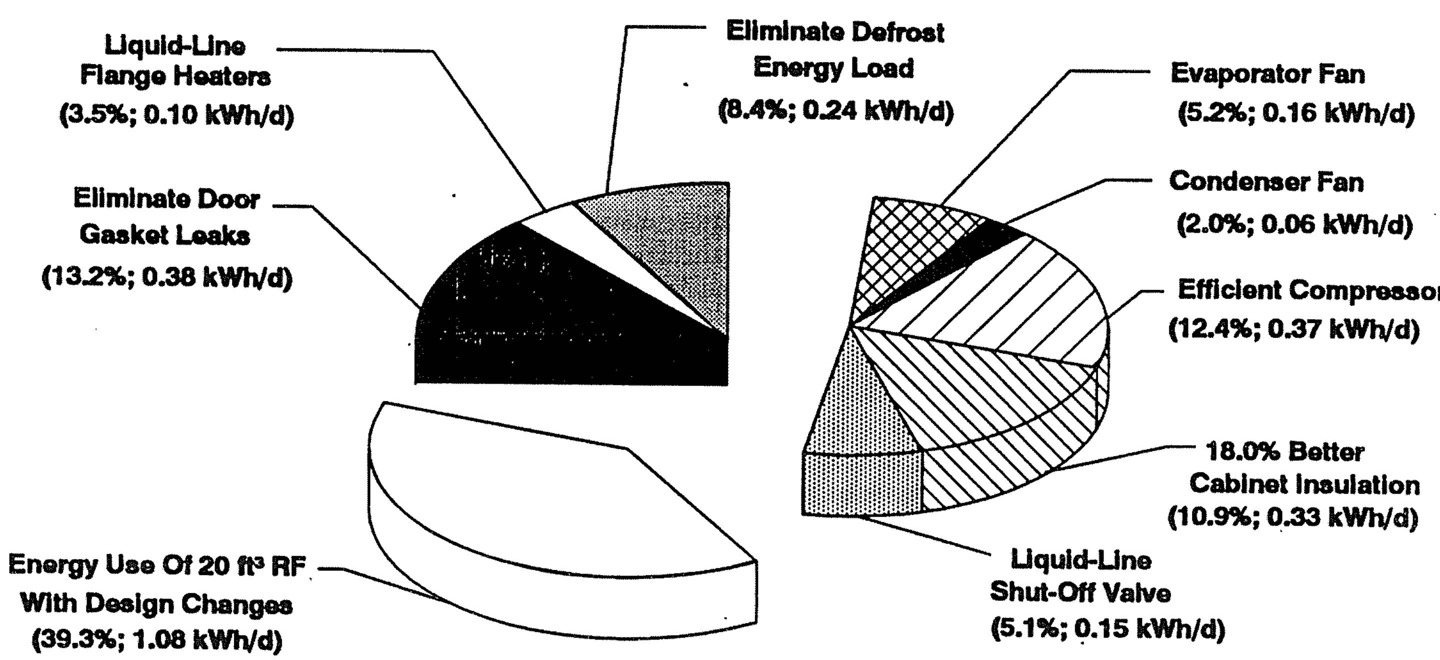

Fig. 2. Modeled energy savings from design changes—1990 base unit (2.87 kWh/d). Changes applied individually against base.

Individual vs Collective Energy Savings

Modeling allows each energy saving feature to be evaluated separately. Each potential improvement was determined individually against the 2.87 kWh/d baseline starting point; so when all of these changes are incorporated into a single system, the net improvement will necessarily be less than the total of all the separately determined improvements in Fig. 2. For example, when the improved compressor, heat exchanger fans, insulation, and liquid line shutoff are combined in one RF, the net energy use reduction is 31.2% (0.90 kWh/d) rather than the 35.6% (1.02 kWh/d) total suggested in Fig, 2. It is apparent from this analysis that—notwithstanding the optimistic estimates calculated by comparing each improvement with baseline RF energy use and maximizing energy saving predicted from eliminating defrosts, gasket door leaks, and anti-sweat cabinet heaters—the program goal of 1.00 kWh/d or less was not met. Additional energy savings are needed from even more efficient components, better refrigerants, or an advanced design that uses a more efficient thermodynamic cycle to reach this ambitious target. Clearly, as evidenced by Fig. 1, the domestic RF is rapidly reaching the point of diminishing returns as a vehicle for saving significant amounts of national energy.

2.4 EXPERIMENTAL RESULTS

2.4.1 Reverse Heat Loss Measurements

Steady state heat loss measurements were performed on the baseline RF cabinets as received and on cabinets with externally mounted expanded polystyrene panels to simulate reduced-load cabinet designs. Data similar to those plotted in Fig. 3 were obtained. In Fig. 3, heat loss rate under steady state conditions, measured in Btu/h (watts) is plotted against the difference between air temperatures inside the RF compartment and cold air temperatures [≈ 0 to 5°F (-18 to -15°C)] maintained by an environmental chamber. Separate heat sources in the form of shielded light bulbs and small fans controlled with a monitored variac are used in each compartment to obtain ΔTs similar, but opposite in direction, to those seen under normal RF operation. A linear relationship between heat loss rate and temperature difference (Eq. 1) is assumed. Equations obtained through a least-squares fit of the data were used to calculate the cabinet load arising from heat permeation through walls, doors, and gaskets at operating and ambient temperatures.

$$Q = UA \times T \qquad (1)$$

A 15–20% improvement in the thermal resistivity of the cabinet was chosen as a reasonable design improvement that could be obtained through implementing evacuated panels, increased foam insulation thickness, and/or improved door gaskets. Improved cabinet insulation was obtained by adding polystyrene panels to the exterior surfaces of the RF. No additional insulation was added on the bottom of the fresh food compartment or in the mullion section of this top-mount design because it would have decreased the internal RF volume. Panels of different thickness were used on freezer and fresh food compartment walls to maintain a similar freezer-to-total-cabinet load (Q_{FRZ} /Q_{TOT}) ratio. Theoretically, RF energy usage should decrease with decreasing Q_{FRZ}/Q_{TOT} because a slightly smaller thermal lift for the refrigeration is implied. However, modeling results with Q_{FRZ}/Q_{TOT} ratios over a range from 0.45 to 0.65 in the conventional, single-evaporator design result in only a 1% change in calculated daily energy use. Calculations based on data from these reverse heat loss experiments with Q_{FRZ}/Q_{TOT} ratios for a RF 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 3. Also tabulated are the analytical results obtained by entering cabinet

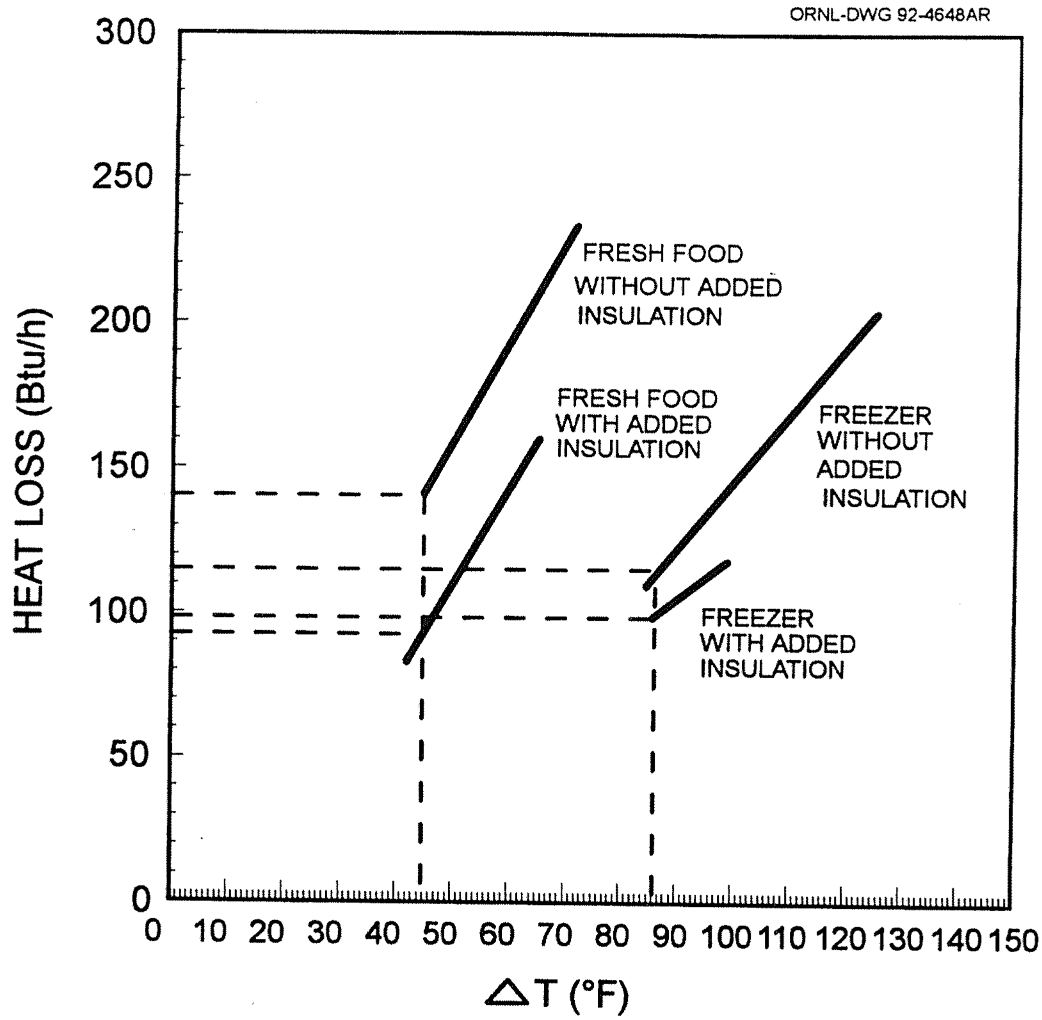

Fig. 3. Cabinet heat loss measurements for 1990 cabinet with and without added insulation.

Table 3. Results summary of reverse heat loss tests and modeling: 90°F ambient, 5°F freezer, 38°F fresh food compartment

	$Q_{Freezer}$ (Btu/h)	$Q_{Fresh\ Food}$ (Btu/h)	$Q_{Total\ Cabinet}$ (Btu/h)	$Q_{Freezer}/Q_{Total}$ (Btu/h)
Base cabinet				
Experimental	160.5	130.6	291.1	0.55
Modeled	148.6	144.0	292.6	0.51
Cabinet with added insulation				
Experimental	129.1	107.8	236.9	0.54
Modeled	129.3	121.5	250.8	0.52

dimensions, foam thicknesses, and thermal resistivities obtained from the RF manufacturer into the cabinet model portion of the model. Modeled values are also shown for the enhanced cabinets when measured insulation thicknesses and R values are plugged into the model for the added polystyrene panels. Results indicate that the cabinet model section of the overall RF model can estimate RF cabinet heat leakage rates to within ± 5 to 10% given the insulation dimensions and thermal conductivity. The model uses a one-dimensional heat flow algorithm with corrections for edge and corner effects; door gasket performance is not correlated with fan operation. It is readily acknowledged that an RF with an added layer of foam panels has different heat transfer characteristics from one with added insulation and a metal exterior surface, but the exterior panels were used as a concession to experimental expedience.

2.4.2 Compressor Calorimeter Results

Nine-point calorimeter tests were used to determine the performance of the R-12 compressor used in the baseline RFs and the R-134a compressor used in the modified units. Results from the original R-12 compressor and the R-134a compressor used to replace it are shown in Fig. 4. These data show that at the standard rating point of -10°F (-23.3°C) evaporating and 130°F (54.4°C) condensing for RF compressors, a 4.3 EER compressor was being replaced with one with an EER of 5.2. These compressor maps were used as inputs for the computer model.

It is also apparent that the refrigeration capacity of the R-134a compressor was approximately 10% less than that of the R-12 compressor it replaced. Less refrigeration capacity is required with the supplementary cabinet insulation. Using a compressor with a capacity much greater than the load would result in short, frequent compressor runs that increase system cycling losses. Load and refrigeration capacity should be balanced for an optimized refrigeration system.

2.4.3 System Test Results

A comparison of RF cumulative energy consumption results measured in the laboratory with those predicted by the computer model is given in Table 4 and Fig. 5. No analytical results are listed for cases D and E because the RF model has insufficient versatility to directly input parameters describing the enhanced evaporator being used experimentally. An iterative process of changing the evaporator *UA* (combined overall heat transfer coefficient and area) in the model and correlating model outputs with experimentally measured data such as heat exchanger temperatures and pressures would be required.

A 4–5% discrepancy is seen between the experimentally measured as-received energy consumption in our laboratory and that predicted by the model and measured by the manufacturer. Several attempts to resolve this difference in results were unsuccessful, so a decision was made to proceed with separate experimental and modeled baseline values and to compare energy savings relative to the appropriate starting point. When this approach is applied to the data in Table 4, the results shown in Fig. 5 are obtained. These "normalized" results indicate that the model reproduces energy consumption trends due to circuit modification and hardware changes quite accurately.

The improvement in RF performance as a result of increasing the air-side area of the evaporator by 33% (Case D) is quite disappointing if it is viewed only in terms of daily energy consumption. This increase in air-side surface area was obtained by switching from a plate-fin and tube heat exchanger design to one with lanced-fin stock wrapped in a spiral around the refrigerant tubing. A corresponding 16% increase in refrigerant-wetted surface area also resulted from this substitution. Changes in fin efficiency were not calculated or correlated with experimental results.

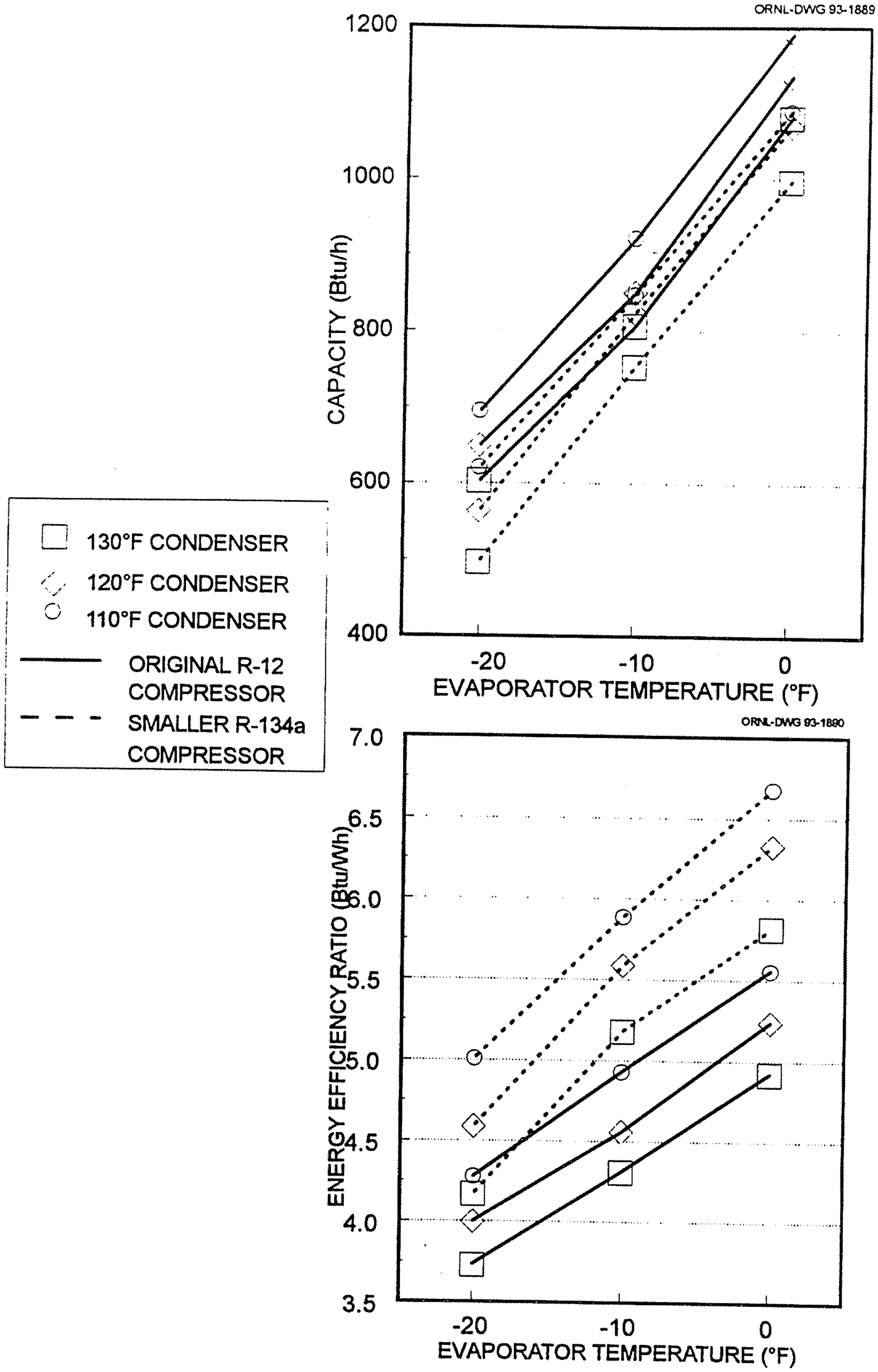

Fig. 4. Compressor calorimeter results.

Table 4. Closed-door cumulative daily energy consumption/results run time at 90°F: 1990 top-mount units—experimental and analytical results/design options

Case	Description	Experimentally measured	Analytically calculated	Manufacturer data
A	Baseline as-received units			
	Energy	3.00 kWh/d	2.87 kWh/d	2.88 kWh/d
	Run time	53.3%	52.0%	51.7%
B	Case A+ 18.6% increase in cabinet insulation			
	Energy	2.77 kWh/d	2.54 kWh/d	
	Run time	47.2%	45.0%	
C	Case B+ 5.2 EER, compressor, R-134a, 4-W evaporator/condenser fan motor			
	Energy	2.14 kWh/d	2.01 kWh/d	
	Run time	49.8%	50.0%	
D	Case C+ enhanced evaporator (33% air side, 16% ref. side surface increase)			
	Energy	2.10 kWh/d		
	Run time	41.3%		
E	Case D+ 5.1 EER; Btu/h compressor and liquid line shutoff			
	Energy	2.07 kWh/d		
	Run time	46.6%		

To fully appreciate the effect of this heat exchanger change, other system operating characteristics must be considered, such as compressor run times, compressor cycles per day, and air/refrigerant approach temperatures or the high- to low-side pressure ratios. In addition to reducing energy consumption by 1.9%, the enhanced evaporator reduced compressor run times from 49.8% to 41.3%, reduced the high- to low-side pressure ratio from approximately 9.9 to 8.9 at the end of the compressor "on" cycle, and reduced the number of compressor on cycles per day from 30–31 to 27–28. All of these results indicate that the new evaporator is much more effective or efficient at heat transfer, and that a lower capacity compressor could be used effectively with this enhanced heat exchanger.

The 5.2 EER compressor with a 700 Btu/h capacity was replaced with a less efficient 5.1 EER compressor having a 640 Btu/h refrigeration capacity (Case E, Table 4). A comparison of the capacity and EER results from calorimeter runs on these two R-134a compressors is shown in Fig. 6. RF system energy use decreased from 2.10 kWh/d to 2.07 kWh/d despite a less efficient compressor and longer run times. Improvement of efficiency is attributable to a better match between the refrigeration circuit and the cabinet heat load. Using measured run times and an assumption that 75% of the electrical energy used by a RF is due to compressor operation (Turiel and Heydari 1988), it is possible to estimate that the energy use of this Case E modification would have been 2.04 kWh/d (67–68% of the base) if a 5.2 EER compressor had been available in this capacity range.

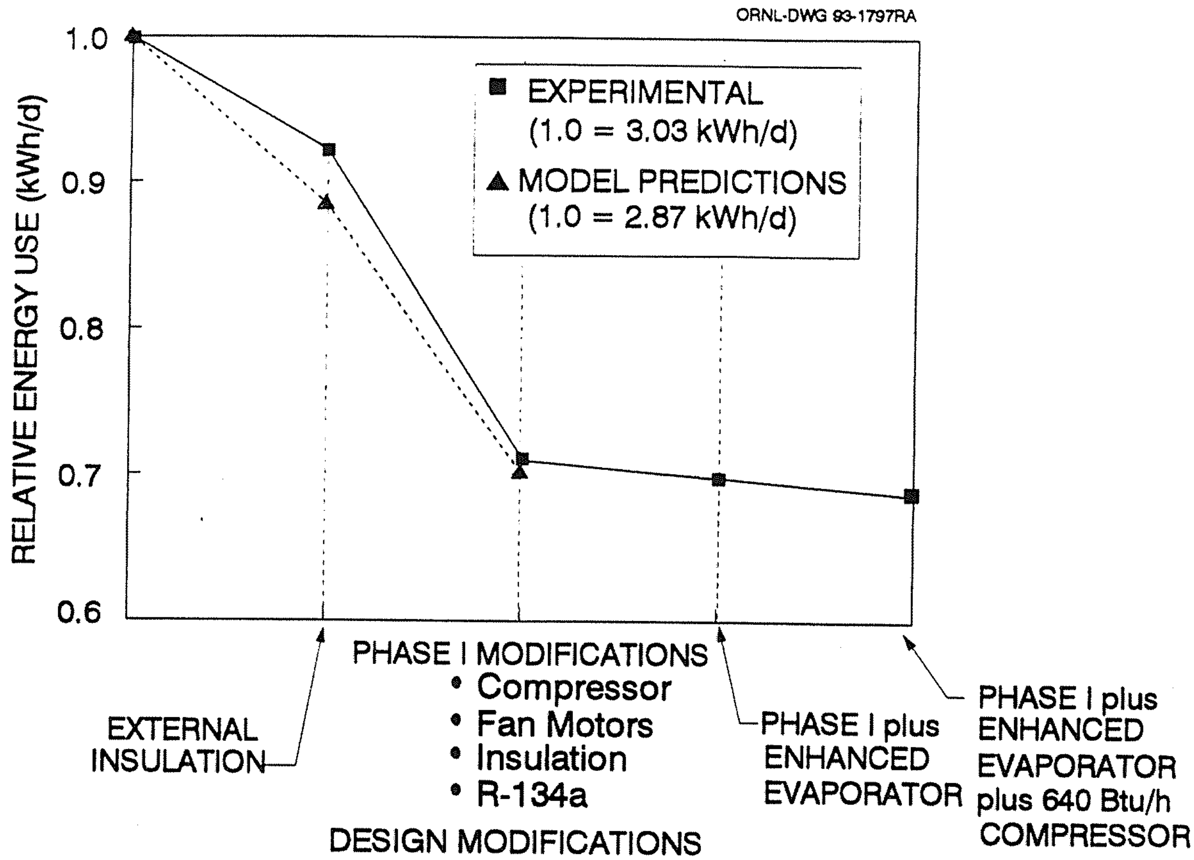

Fig. 5. Normalized improvements in refrigerator performance—experimental and modeled results.

2.5 CONCLUSIONS FROM PHASE 1 RESEARCH

Results from modeling runs performed using the 1990 RF units with substantial improvements to the RF cabinet insulation and/or the effectiveness of door gaskets may be misleading. Certainly these changes will reduce the thermal load and the energy consumed in the 90°F closed-door test, which is one of the main criteria used to rate the performance of RFs. But the effects of door openings and of loading of refrigerators with perishable products, which are not a part of the closed-door test, will become a more significant part of the load; and this rating method will probably become less representative in actual RF field use.

In addition, better-insulated compartments will require refrigeration systems with increasingly smaller capacity compressors to achieve a balance between thermal load and refrigerating capacity. The efficiency of RF compressors falls off sharply at lower capacities (Fig 7).

Experimental implementation of some of the more easily tested design improvements—such as additional cabinet insulation, a more efficient compressor, efficient fan motors, larger heat exchangers, a liquid line shutoff valve, and redirected air flows on the 1990 units—resulted in a 30% improvement in energy savings over the as-received cabinet. Experimental energy savings resulting from specific hardware and

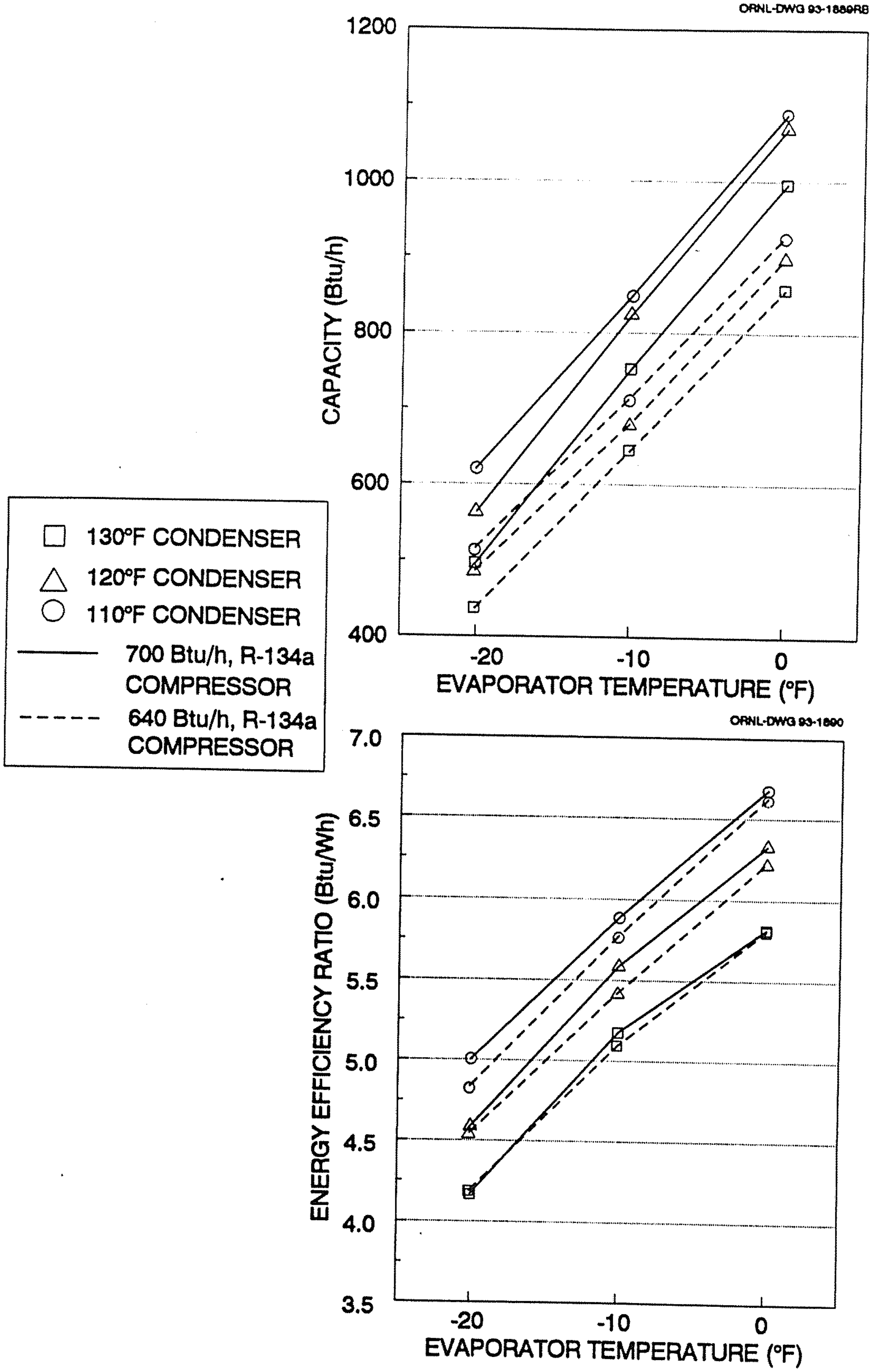

Fig. 6. Compressor calorimeter results.

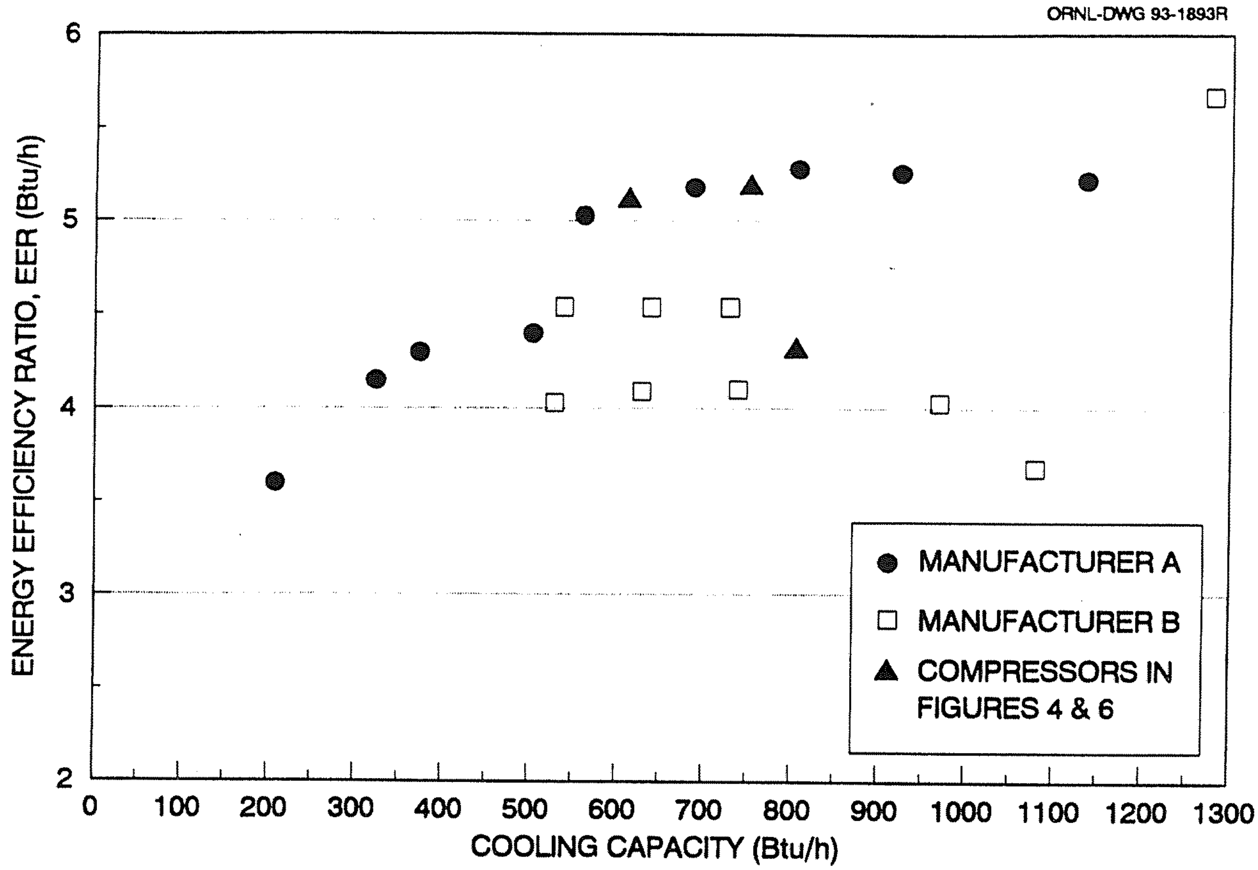

Fig. 7. Sampled compressor efficiency vs refrigeration capacity data—130°F (54.4°C) condensing temperature, -10°F (-23.3°C) evaporating temperature.

design changes have correlated quite well with those predicted by the detailed computer model. Some aspects of the model, such as its ability to describe and simulate advanced heat exchanger designs, are inadequate; however, it tracked the experimental results well enough that it was used with some confidence to determine an initial technology path to achieve the project goal.

2.6 ANALYSIS OF FUTURE DESIGN OPTIONS

Two 1993 20-ft^3 (570-L) cabinets were obtained that incorporate energy-saving features such as a 5.2 EER compressor, more efficient cabinet insulation, and liquid line-flange heat into the as-manufactured design. Baseline energy consumption for these units is less than 2.00 kWh/d. Starting with these RFs, the computer model was used to determine what design refinements would be needed to meet the program goal using a *conventional design* (Fig. 8). Design options were added to the model sequentially so that the cumulative effects of each option would be accounted for. The results in Fig. 8 show that the most significant savings come from a high-efficiency variable-capacity compressor, better cabinet insulation, and a liquid line shutoff valve. More efficient fan motors and increased heat exchanger areas also account for sizable energy savings. Because these changes resulted in an RF whose predicted energy use was still

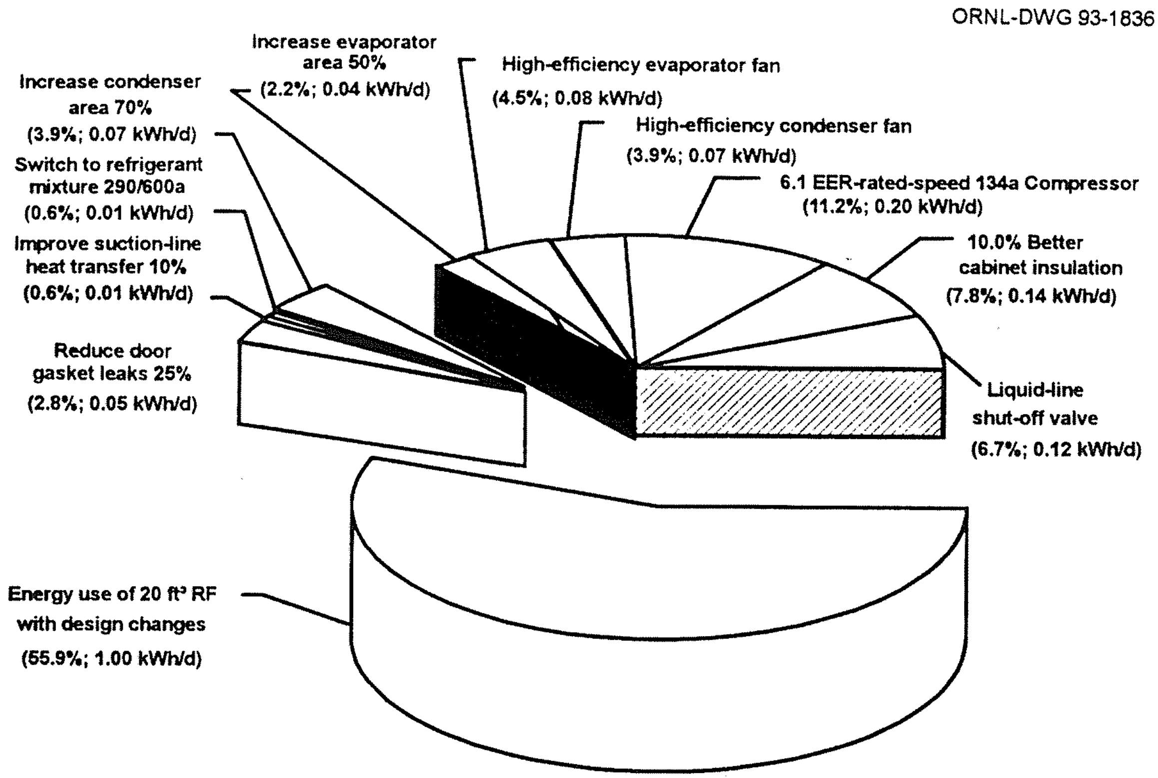

Fig. 8. Modeled energy savings from design changes—1993 baseline unit, 1.79 kWh/d.

1.07 kWh/d, further modifications were simulated even though smaller and smaller energy savings resulted. A 1.00 kWh/d daily energy use was achieved only after additional changes were incorporated into the model for improved door gaskets, better suction-line heat transfer, and use of a more efficient refrigerant. These new cabinets were used as laboratory test beds in Phase 2 to attempt to duplicate experimentally the path laid out to achieve the project goal.

Analytical modeling studies indicate that improvements to the standard RF design may not result in a unit that meets energy use goals for this project. To achieve energy savings of this magnitude, it may be necessary to design a system that uses a more thermodynamically efficient cycle such as the Lorenz Cycle, in which heat transfer irreversibilites are minimized. Two advanced cycle designs are being considered for experimental investigation in future phases of this project if required:

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

- A two-evaporator Lorenz-Meutzner RF 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 increases in system complexity and, perhaps, compromises in appliance reliability as well.

3. PHASE 2: DESIGN AND TESTING OF INITIAL PROTOTYPE

3.1 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 evaluate analytically 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 (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 data, is also available.

Several options were considered for decreasing the energy consumption of a conventional RF (Table 5). They were selected on the basis of previous studies and discussions with an advisory group comprising all the major RF 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 DC motors for those presently used. Option 9, a liquid line shutoff 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.

Table 5. 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

A 1993 20-ft^3 (570-L) top-mount, automatic-defrost RF was selected for the project based on its popularity and corresponding high market share. Because the unit was required to meet the new 1993 NAECA standards, the baseline energy consumption was quite low (1.80 kWh/d), 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 Fig. 8 show that the most significant energy savings come from a high-efficiency rated-speed compressor, better cabinet insulation, and a liquid line shutoff 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. Because these changes resulted in an RF 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 achieved only 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 RF 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 RF operating with a low glide (10–20°F [5.6–11.1°C]) refrigerant mixture, counterflow heat exchangers, and a variable- speed compressor

- A two-evaporator Lorenz-Meutzner RF 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.

3.2 TEST PROCEDURE

All tests were performed on a 20-ft^3 (570-L) top-mount, automatic-defrost RF 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 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 AHAM Standard for Household Refrigerators and Household Freezers (AHAM 1985).

3.2.1 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 W of electricity and has an air circulation rate of 30 cfm (14 L/s), which is assumed to have negligible effects on the inside surface heat transfer of the RF.

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 RF works to maintain cold internal temperatures in a hot room. 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).

3.2.2 Compressor Calorimeter Results

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 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.

3.2.3 System Test Results

System performance, before and after the units were modified, was assessed using the standard 90°F (32.2°C) closed-door test procedure. In this procedure, the RF 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 resulting test points are used to calculate the energy consumption over a 24-h (1-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 RF 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 RF 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).

3.3 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 RF design, defined as a unit with a single fan-forced evaporator and condenser and single-speed compressor, 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 RF, the CFC-12 was replaced with a non–ozone-depleting refrigerant (HFC-134a) for the experimental evaluations.

3.3.1 Reverse Heat Loss Measurements

Steady state heat loss measurements were performed on the baseline RF cabinet. Following the completion of the baseline tests, expanded polystyrene panels with R values of 5.3 to 5.6 $h \cdot ft^2 \cdot °F/Btu \cdot in$ (2.4–2.5 $m^2 \cdot °C/W \cdot cm$) were added to the cabinet exterior to simulate a reduced heat load cabinet design. Because it was beyond the scope of the project to develop an advanced insulation, a simulation of this nature was in order to design the refrigeration system properly for expected cabinet load reductions in the future. Data similar to those in Fig. 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 to 5°F (-18 to -15°C)] maintained in an environmental chamber. A linear relationship between heat loss rate and temperature difference (Eq. 2) 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 (Fig. 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.

$$Q = UA \times T \tag{2}$$

A 10% targeted improvement in cabinet thermal resistivity was chosen as a reasonable design change that could be obtained through implementing evacuated panels or increased insulation thicknesses. The improvement in cabinet insulation was simulated by adding expanded polystyrene panels to the top and side exterior surfaces of the baseline RF cabinet. No additional insulation was added on the bottom of the fresh food compartment because of 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 RF 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 6. 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 in. of insulation to the freezer section and 1 in. 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).

Table 6. 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 in. freezer + 1 in. 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 in. freezer + 2 in. fresh food insulation				
Experimental				
—Energy consumption	122.6	101.5	224.1	0.55

In an attempt to verify which results more accurately reflected the effects of the additional insulation, energy consumption tests were conducted for the unit with and without additional insulation (Table 7). The model was then used to calculate the cabinet heat load required to achieve the measured energy consumption. The results, given in Table 6, 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 because 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 RF, 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 in. of insulation to the freezer section and 2 in. 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 Fig. 8 to proceed with the experimental plan. It is noted that the insulation thicknesses are far beyond what would be allowable in a conventional design; however, the cabinet heat loss reductions are achievable with some forms of advanced insulation.

Table 7. Energy consumption tests—experimental and analytical results/design options

Case	Description	Experimentally measured	Analytically calculated	Manufacturer data
A	Baseline units			
	energy	1.80 kWh/d	1.79 kWh/d	1.76 kWh/d
	run time	41.3%	41.0%	40.3%
B	Case A+ 2 in. freezer + 1 in. fresh food insulation			
	Energy	1.70 kWh/d	1.52 kWh/d	
	Run time	38.6%	35.1%	
C	Case B + 4 in. freezer + 2in. fresh food insulation			
	Energy	1.65 kWh/d	1.47 kWh/d	
	Run time	37.4%	33.8%	
D	Case C + 6.0 EER compressor, high-efficiency fans, larger evaporator, and liquid line shutoff			
	Energy	1.41 kWh/d	1.18 kWh/d	
	Run time	42.3%	43.0%	

3.3.2 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 RF and the HFC-134a rated-speed compressor used in the modified units. The resulting compressor maps, shown graphically in Fig. 9, were used as inputs for the modeling analysis. From the data in Fig. 9, 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.

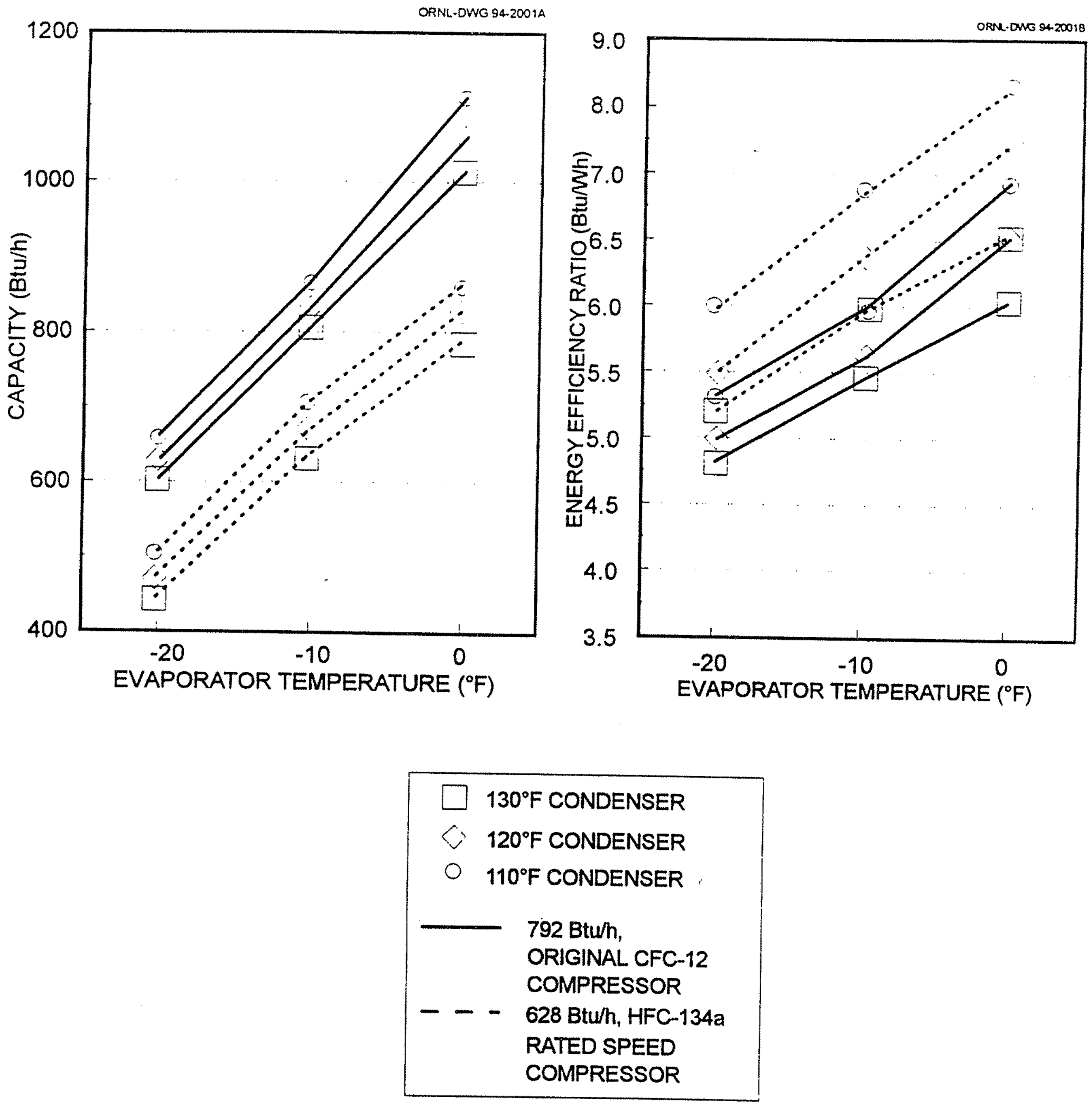

Fig. 9. Compressor calorimeter results.

3.3.3 System Results

Of the nine options investigated for reducing the energy consumption of the RF, 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 because of the difficulties experienced in 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 shutoff 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 7) 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 shutoff valve energy savings and suboptimal capillary tube length.

In separate tests that investigated the effects of a liquid line shutoff valve, there was no change in energy consumption when a valve was used. Subsequent discussions with RF manufacturers revealed that only minimal energy savings have been realized when liquid line shutoff valves have been 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 the refrigerant CFC-12 to HFC-134a. The existing capillary tube was embedded in the foam, 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.

3.4 CONCLUSIONS FROM PHASE 2 RESEARCH

Energy consumption tests performed to determine the cabinet heat load indicate that the power usage was reduced by 8% [1.80 to 1.65 kWh/d (Table 7)] from adding insulation to the cabinet exterior. The results are believed to indicate what is technically achievable given new technologies, such as vacuum insulation, which are on the horizon. Note, however, that new insulations are unproven in terms of long-term reliability and heat transfer degradation over time, two factors that 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 RF energy use in the field. Insulation improvements also require refrigeration systems with smaller-capacity compressors to achieve a balance between thermal load and refrigerating capacity. Since the efficiency of RF 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: The 1.41 kWh/d energy consumption yielded a 32% improvement over the NAECA 1993 target for 20-ft^3 (570-L) automatic-defrost, top-mount RFs.

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.

4. PHASE 3: ENERGY AND ECONOMIC ANALYSES OF FINAL PROTOTYPE

4.1 INTRODUCTION

A 1996 20-ft^3 (570-L) top-mount, automatic-defrost RF was selected as the baseline unit for testing. Because the unit was required to meet the new 1993 NAECA standards, the energy consumption was already quite low (1.68 kWh/d), making further reductions in energy consumption very challenging. Actual and projected improvements in the power consumption of RFs is shown in Fig. 10. Among the energy saving features incorporated into the design of the baseline unit were a low-wattage evaporator fan, increased insulation thicknesses, and liquid line flange heaters.

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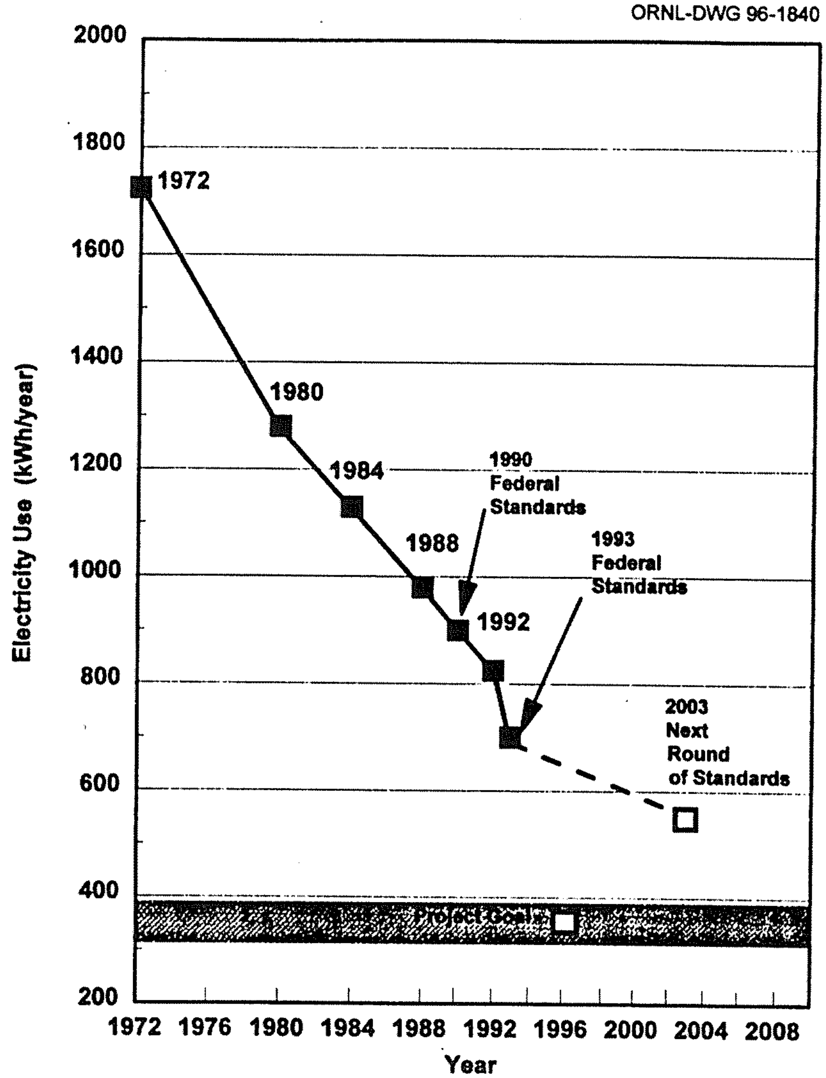

Fig. 10. Actual and projected refrigerator-freezer energy improvements from 1972 to 2003.

4.2 EXPERIMENTAL PLAN

In Phase 2 of this project, a prototype RF achieved an energy consumption of 1.41 kWh/d (Vineyard et al. 1995). The baseline unit for Phase 2 was a 1993 vintage model with an initial energy consumption of 1.80 kWh/d. Design changes incorporated into the unit to reduce energy consumption consisted of thicker insulation around the entire cabinet, a high-efficiency compressor, low-wattage evaporator and condenser fans, an enhanced evaporator, and a liquid line shutoff valve. The cabinet heat loss rate was determined to be 224.1 Btu/h (65.7 W), a 9% reduction from the baseline unit [246.0 Btu/h (72.1W)].

Following discussions with an advisory group comprising all the major RF manufacturers, several options (Table 8) were considered for the Phase 2 effort. They 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 using a different thermodynamic cycle such as the Lorenz-Meutzner. Options 7 and 8 reduce the parasitic power requirements by substituting electrically-commutated 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 the Phase 2 effort, most of these options were investigated both analytically and experimentally. The results showed that major improvements in energy savings came from cabinet insulation improvements, the high-efficiency compressor, and the low-wattage fan motors. Therefore, the priorities for this study were those same options, along with adaptive defrost control and heat exchanger improvements. Advanced cycles were the lowest priority and were not investigated experimentally because the project goal was achieved.

Table 8. 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

4.3 TEST PROCEDURE

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, RF 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 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.

4.3.1 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 W of electricity and has an air circulation rate of 30 cfm (14 L/s), which is assumed to have negligible effects on the inside-surface heat transfer of the RF. Temperature and wattage measurements for both RF compartments, along with ambient temperature, are recorded as the cabinet temperatures achieve desired levels. Once the cabinet temperatures achieve steady state, data are compiled and averaged for a 30-min 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. Heat loss rates for the freezer compartment were determined from Eq (3):

$$Q_{FRZ} = UA_{FRZ} \times (T_{FRZ} - T_{AMB}) + UA_{MUL} \times (T_{FRZ} - T_{FF}) \tag{3}$$

where
Q_{FRZ} is the heat loss rate for the freezer in Btu/h (W)
UA_{FRZ} is the overall freezer compartment thermal resistivity in Btu/h·°F(W/°C
$(T_{FRZ}-T_{AMB})$ is the temperature difference between the freezer and ambient in °F (°C)
UA_{MUL} is the thermal resistivity of the mullion in Btu/h (W)
$(T_{FRZ}-T_{FF})$ is the temperature difference between the freezer and fresh food compartments in °F(°C)

In a similar manner, the fresh food heat loss rate was determined from Eq. (4):

$$Q_{FF} = UA_{FF} X (T_{FF} - T_{AMB}) - UA_{MUL} X (T_{FRZ} - T_{FF}) \tag{4}$$

where
Q_{FF} is the heat loss rate for the fresh food compartment in Btu/h (W)
UA_{FF} is the overall fresh food compartment thermal resistivity in Btu/h·°F (W/°C)
$(T_{FRZ}-T_{AMB})$ is the temperature difference between the fresh food compartment and ambient in °F (°C)

Tests were initially run with the temperatures in both compartments essentially equal. This allowed the mullion heat transfer term to be dropped from both Eqs. (3) and (4) so that freezer and fresh food compartment resistivities could be determined from dividing the power measurement (Q) by the temperature difference in each compartment $(T_{FRZ}-T_{AMB})$ or $(T_{FF}-T_{AMB})$ Once the compartment thermal resistivities were known, tests were then performed with large temperature differences between the freezer and fresh food compartments to determine the mullion thermal resistivity. Plots were then generated using Eqs.(3) and (4) to represent the heat loss rates in both compartments for each cabinet and door configuration.

The tests were conducted using temperature differences across the cabinet walls comparable to those attained in the 90°F (32.2°C) closed-door test procedure where the RF works to maintain cold internal temperatures in a warm room. To achieve the temperature differences, it was necessary to maintain the chamber at 0°F (-17.8°C). Since the thermal conductivity of insulating foam generally decreases with decreasing temperatures, this procedure could slightly underestimate actual cabinet heat loss rates (ASHRAE 1989).

4.3.2 Compressor Calorimeter Results

Reductions in the total cabinet heat load along with efficiency improvements required corresponding changes in the capacity and design of the compressor. 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 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 RF energy consumption when using the high-efficiency compressor.

4.3.3 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 RF 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-h 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 RF 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 energy use in field service (Meier and Jansky 1993). Previous RF 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).

4.4 EXPERIMENTAL RESULTS

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

4.4.1 Reverse Heat Loss Measurements

Steady state heat loss measurements were performed on two separate units, a baseline RF cabinet and an enhanced cabinet with vacuum insulation panels foamed in around the freezer section. Three sets of doors with varying degrees of insulation improvements were tested on the baseline unit. The three sets of doors consisted of the following improvements: thick doors (2 in.)(5.1 cm) versus standard (1 in.)(2.5 cm) doors, 1-in. (2.5-cm) -thick vacuum insulation panels foamed into standard doors, and 1-in. (2.5-cm) -thick vacuum insulation panels foamed into thick doors. Only one set of doors, the thick doors with no vacuum insulation panels, was tested on the enhanced cabinet.

Cabinet heat loss rates for the baseline unit with the standard doors and door insulation improvements are shown in Fig. 11. The heat loss rates are determined from Eqs. (3) and (4) 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 12 shows the cabinet heat loss results for the enhanced cabinet with standard and thick doors.

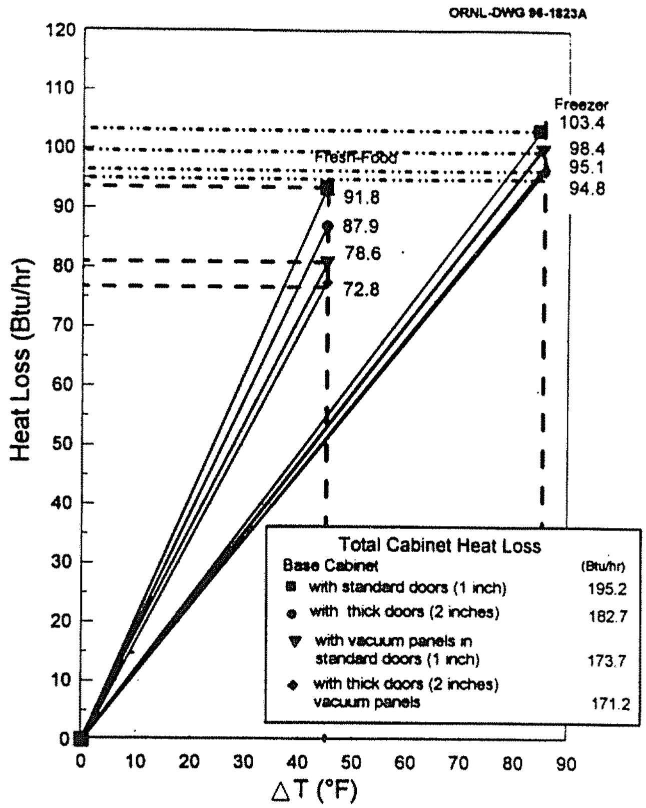

Fig. 11. Reverse heat loss results in base cabinet.

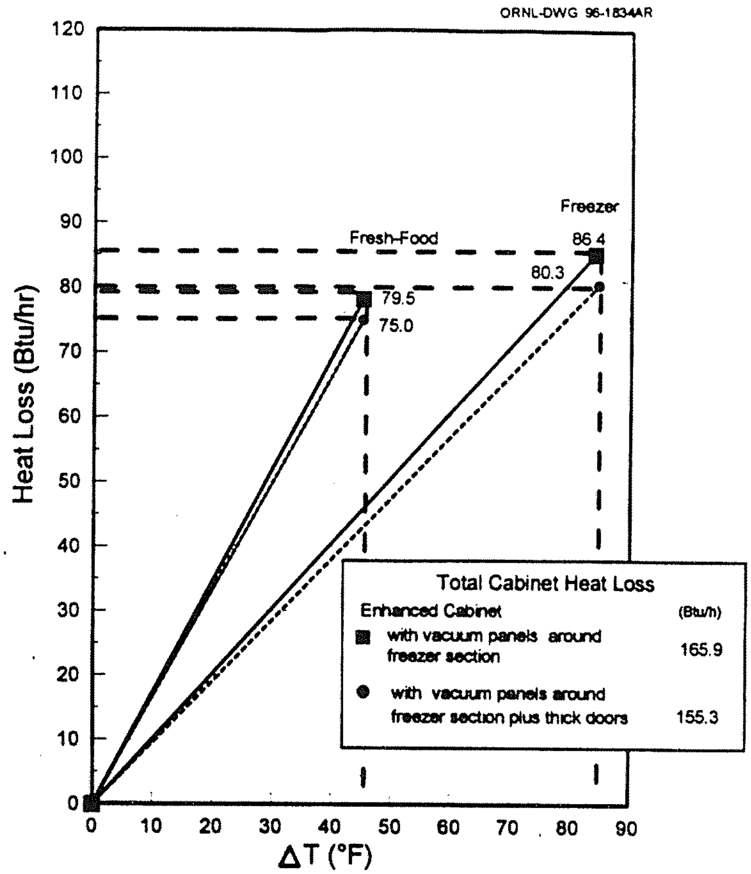

Fig. 12. Reverse heat loss results in enhanced cabinet.

The cabinet heat loss rates are summarized in Table 9 along with Q_{FRZ}/Q_{TOT} ratios for a RF. 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-in. (2.5-cm) -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-in. (2.5-cm) -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%. 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 vs 155.3 Btu/h) (57.2 vs 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.

Table 9. Summary of reverse heat loss tests 90°F ambient, 5°F freezer, 45°F fresh food compartment

Description	$Q_{Freezer}$ (Btu/h)	$Q_{Fresh\ Food}$ (Btu/h)	Q_{Total} (Btu/h)	Q_{Feezer}/Q_{Total}	Percentage reduction
Base cabinet:	103.4	91.8	195.2	0.53	
w/standard doors (1 in.)	94.8	87.9	182.7	0.52	6.4
w/thick doors (2 in.)	95.1	78.6	173.7	0.55	11.0
w/vacuum panels in standard doors (1 in.)	98.4	72.8	171.2	0.57	12.3
w/vacuum panels in thicker doors (2 in.)					
Enhanced cabinet:					
w/vacuum panels around freezer section	86.4	79.5	165.9	0.52	15.0
w/vacuum panels around freezer section and doors (2 in.)	80.3	75.0	155.3	0.52	20.4

4.4.2 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 RF 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 Fig. 13, are used as inputs for modeling analyses. From the data in Fig. 13, 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 10.9% 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.

4.4.3 System Test Results

Of the eight options under consideration for reducing the energy consumption of the RF, only five were required to achieve the goal of a 50% energy savings. The five 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 5, a larger condenser with a counterflow arrangement, would have been the next design change introduced had it been necessary to achieve further savings. The other modifications, door gasket improvements and an advanced cycle design, were low-priority items because of their additional complexity and difficulties in incorporating them into a commercially manufactured cabinet. However, they would have been addressed had the goal not have 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

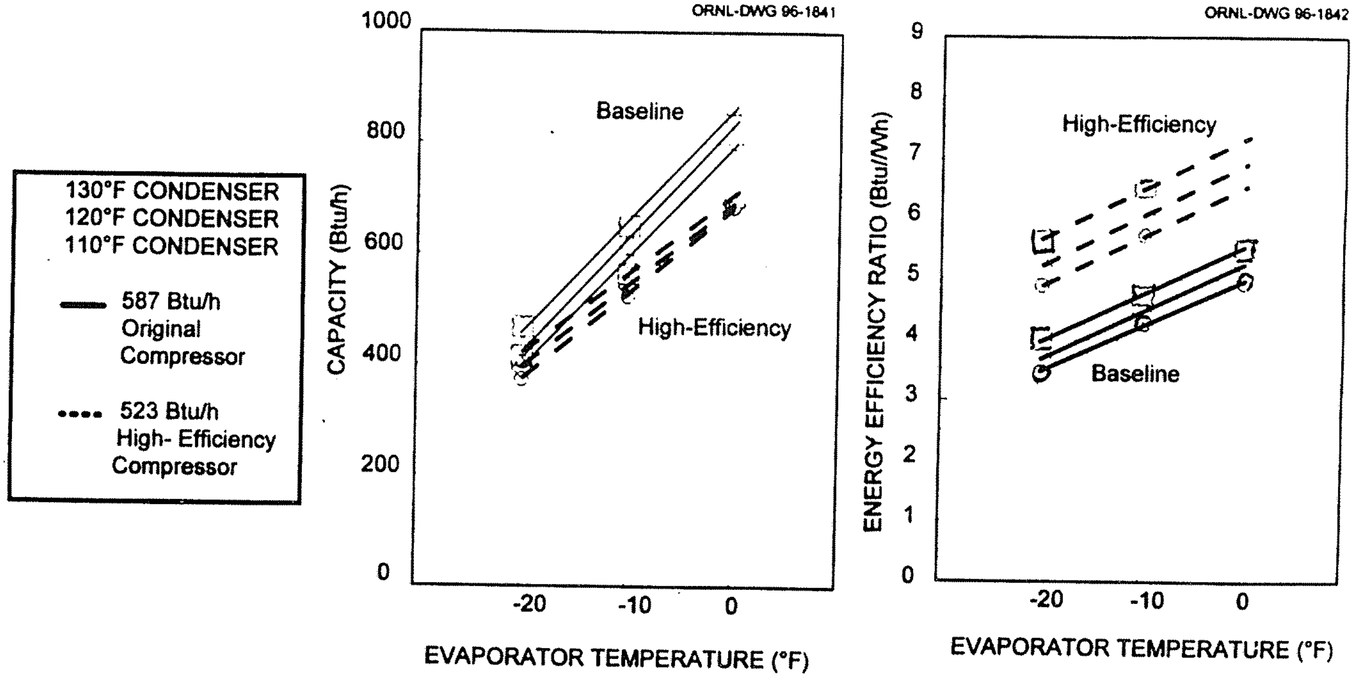

Fig. 13. Compressor calorimeter results.

show that the energy consumption was 1.68 kWh/d (Table 10). The DOE standard for a unit of this type and size is 2.01 kWh/d. Thus the baseline cabinet is 16.5% below the DOE standard.

Next, an enhanced cabinet with vacuum insulation panels foamed around the freezer section was tested. In addition to the vacuum insulation, the unit also was assembled with a larger counterflow evaporator. The daily energy consumption for that unit was 1.53 kWh/d, an 8.5% reduction from the baseline unit and 23.6% lower than the DOE standard.

Following completion of the energy consumption tests on the enhanced cabinet, the unit was modified by exchanging the standard doors for ones that were 2 in. (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 further reduce the energy consumption. The results for all the improvements (Table 10) show that the energy consumption was reduced from 1.53 kWh/d to 0.93 kWh/d, a savings of 39.5%. Relative to the baseline unit and NAECA standards, the results represent a 44.6% improvement (1.68 to 0.93 kWh/d) and 53.8% improvement (2.01 to 0.93 kWh/d), respectively.

An additional design configuration 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-in. (5.1-cm) -thick doors, and a long-term defrost control algorithm was used. Although the energy consumption for this configuration was expected to be

Table 10. Energy consumption and cost information

Description	Energy consumption (kWh/d)	Percent run time	Manufacturer cost increase ($)
Baseline unit	1.68	44.2%	----
Baseline unit with 2-in.-thick doors, 5.73 EER compressor, low-wattage condenser fan, and adaptive defrost	1.16	47.6%	53.38
Enhanced cabinet (vacuum panels around freezer section) with 2-in.-thick doors, 5.73 EER compressor, low-wattage condenser fan, larger evaporator, and adaptive defrost control	0.93	36.5%	134.33

moderately higher than for the enhanced cabinet model, the design changes were expected to be more cost-effective. The resulting energy consumption for the unit was 1.16 kWh/d, a 30.5% reduction from the baseline unit and 42.0% lower than NAECA standards.

4.4.4 Cost Analysis

To obtain a cost/benefit ratio of the energy-saving features, it was necessary to estimate the cost for each design change (Table 11). Most of this 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 RF 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.

Based on the information in Table 11, the estimated manufacturer's cost increase for the 0.93 kWh/day design is $134.33. This estimate is based on using a high-efficiency condenser fan ($4.50), an adaptive defrost control ($7.15), an increased evaporator area ($3.11), 2-in. (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/year 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 manufacturer's cost to arrive at an estimated cost to the consumer gives a payback of 11.4 years ($268.66÷$23.676 per year), an unacceptable alternative.

A breakdown of the energy savings from each design change is shown in Table 12. 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 Section 8 of the AHAM Standard for Household Refrigerators and Household Freezers (1985). The results show that the low-wattage condenser fan, thicker doors, and adaptive defrost control had paybacks in the range of 3.0 to 4.1 years. The high-efficiency compressor required 7.7 years for payback. The worst

Table 11. Manufacturer's cost increase for design changes

Design change	Manufacturer's cost increase ($)
Low-wattage condenser fan	4.50
Increased evaporator area	3.11
Vacuum panels around freezer section	77.84
2-in.-thick doors	6.73
High-EER compressor	35.00
Adaptive defrost control	7.15
Total	134.33

Table 12. Cost analysis for design options

Case	Design changes	Annual energy use (kWh/year)	Annual energy savings ($/year)	Cost savings ($/year)	Consumer cost ($)	Payback (years)
A	Baseline unit	612				
B	A+ vacuum insulation around freezer, increased evaporator area	560	52	4.51	161.90	35.9
C	B+ low-wattage condenser fan	525	35	3.03	9.00	3.0
D	C+ 5.73 EER compressor	420	105	9.10	70.00	7.7
E	D+ 2-inch thick doors	379	41	3.55	13.46	3.8
F	E+ adaptive defrost control	339	40	3.47	14.30	4.1

payback period was for the vacuum panel insulation/increased evaporator area combination, which was almost 36 years, 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 unit to be economically feasible, a second unit was assembled at a much lower cost. The estimated manufacturer's cost for this unit is $53.38 based on using a high-efficiency condenser fan ($4.50), adaptive defrost control ($7.15), 2-in. (5.1-cm) -thick doors ($6.73), and a 5.73 EER high-efficiency compressor ($35.00). The energy savings for this unit is 187 kWh/year (1.68 vs 1.16 kWh/d). Using a cost for electricity of $0.0867/kWh, the annual savings is $16.22. The payback, assuming the consumer cost is twice the manufacturer's cost, is 6.6 years.

4.5 CONCLUSIONS

In exceeding the goal of a 1.00 kWh/d RF, two significant accomplishments were realized. First, it was shown that it is technically feasible to build an extremely low energy 20-ft^3 (570-L) RF. 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 unit: (1) the costs were prohibitively high and (2) the compressor run time was too low, indicating that we needed a much smaller compressor, probably in the 400 to 450 Btu/h range. Because compressors in this capacity range have much lower EERs, it appears that the cabinet technology has exceeded the compressor technology. Thus an excellent research topic would be to address improving the efficiency of small-capacity compressors. This assumes that some form of cabinet improvement—such as vacuum insulation, thicker insulation, or door gasket improvements—will be used in future RF applications to reduce the heat gain significantly. At present, vacuum insulation, while an excellent technology, is still too costly. In addition, vacuum insulations are still 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 insulations in RFs appears to be for increasing volume by reducing the insulation volume in areas where it is thickest, such as the doors.

The second and most promising accomplishment was the cost-improved RF, resulting in a 1.164 kWh/day energy consumption. Based on the results from the low-energy RF indicating that the vacuum panel insulation and increased area evaporator were not cost-effective (Table 12), a second unit was assembled without these features. The new unit achieved low energy consumption with a reasonable additional cost. The cost of this unit could be reduced even more by using a production compressor that has a slightly lower EER than the high-efficiency compressor and no cost premium. Using a compressor with an EER in the 5.2 to 5.3 range would increase the energy consumption only to approximately 1.25 kWh/d. The additional cost for the unit would be only around $18 or $36 to the consumer. The unit would save 155 kWh/year for a savings of $13.44 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 1.164 kWh/day version.

5. CLOSING REMARKS

Clearly, there is a rationale for retaining many familiar aspects of a product design that has been refined and used for 30 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 RF 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 AC 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), but the corresponding cost increase is more like a 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.

American manufacturers of domestic RFs have established an enviable record of consistent improvements in the energy efficiency of their product. Widespread use of this appliance as a result of its efficiency, convenience, and reliable performance have made it a target for additional refinement; but it appears that the margins for improving performance are reaching a point of diminishing returns. However, many of the design options that have a significant impact on the energy use of an RF have been identified and are technologically available.

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