INVESTIGATION OF DESIGN OPTIONS FOR IMPROVING THE ENERGY EFFICIENCY OF CONVENTIONALLY DESIGNED REFRIGERATOR-FREEZERS

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ABSTRACT

Several design options for improving the energy efficiency of conventionally designed, domestic refrigerator-freezers (RFs) were incorporated into two 1990 production RF cabinets and refrigeration systems. The baseline performance of the original units and unit components was extensively documented to provide a firm basis for experimentally measured energy savings. A detailed refrigerator system computer model that could simulate cycling behavior was used to evaluate the daily energy use impacts for each modification, and modeled versus experimental results are compared. The model was shown to track measured RF performance improvement sufficiently well that it was used with some confidence to investigate additional options that could not be experimentally investigated.

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. For RF sizes typically sold in the United States (18 ft^3 to 22 ft^3 [510 L to 620 L]), more elaborate alternative refrigeration cycles may be required to achieve the program goal (1.00 kWh per day for a 20-ft^3 [570-L] top-mount RF).

INTRODUCTION

Recent reports indicating exacerbation of stratospheric ozone loss due to the release of long-lived chlorine- and bromine-containing compounds to the atmosphere and the resulting Montreal Protocol revisions have made it apparent that the world community will eventually prohibit the use of these chemicals as refrigerants or foam-blowing agents (Zurer 1993). For the refrigerator-freezer (RF) industry, these regulations will prohibit the use of dichlorodifluoromethane (R-12) as a refrigerant and trichlorofluoromethane (R-11) as a foam-blowing agent in RFs manufactured after 1995.

These useful appliances are also regulated under the 1987 National Appliance Energy Conservation Act (NAECA) in that they are required to meet certain minimum energy-efficiency standards set up and periodically revised by the U.S. Department of Energy (DOE) (NAECA 1987). A metric chart showing actual and projected improvements in the electrical energy use of RFs is shown in Figure 1. Also indicated in this figure is the stated goal for this collaborative industry/government research on RFs.

Additionally, customer expectations and competitive pressures impose an additional, "unwritten" set of constraints on RFs produced in the United States. Mass-produced kitchen cabinetry anticipates certain external RF dimensions; kitchen floors are built to support a limited weight density; a 15- to 20-year useful product lifetime is assumed; and plug-in-and-forget operation is the norm. Additionally, a low selling price is probably the most important feature for an appliance that has become a commodity item. Yet, there is no question that efficient and effective storage and distribution of refrigerated food are essential to the existence of our industrialized society.

Obviously, energy-efficient RFs that are manufactured with and use environmentally acceptable chemicals must be developed. Since there are approximately 115 million refrigerators in the United States and they consume about 2 quads (1 quad = 10^{15} Btu (2.9×10^{11} kWh) of primary energy, or 12% of the total residential energy budget annually, substantial improvements in domestic RF efficiency extended over the 15- to 20-year life of this appliance would significantly benefit national goals for energy conservation and environmental progress.

This paper describes results from the initial phases of an industry/government cooperative research and development agreement (CRADA) to evaluate and test design concepts for an advanced, energy-saving domestic refrigera-

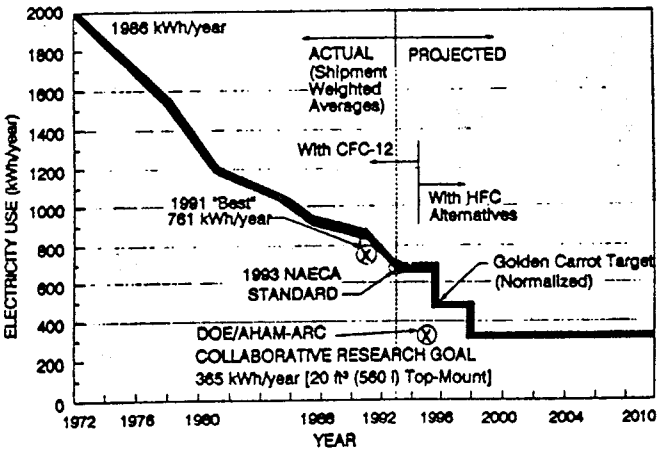

Figure 1 *Refrigerator-freezer energy use—history and program goals—top-mount weighed averages.*

tor. The stated goal of this CRADA is 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 works out to be equal to or less than 1.00 kWh per day. The general objective of the research is to facilitate development of efficient appliances by establishing which design changes that are cost-effective and reliable can conveniently be incorporated into the products.

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, which 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). This 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. This model allowed us to evaluate the energy-saving opportunities of options not generally available as hardware components in a manner that assesses each 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 commonly 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).

Since 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 the procedure, an RF cabinet is 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 that were 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, 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. These nine-point maps were input to the RF computer model for analytical evaluations. 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°C ±0.6°C) environmental chamber with the antisweat 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 resultant test points are used to calculate the RF energy consumption over a 24-hour (one-day) period based on 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 volt to the refrigerator and an air circulation rate of less than 50 fpm (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).

Since commercially manufactured refrigerators were used as laboratory test beds, a testing sequence of "as received" tests followed by tests of units with improved insulation and the original refrigeration circuit was established. 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.

EXPERIMENTAL PLAN

An experimental plan was formulated to help order and prioritize laboratory work. Two phases were anticipated. Phase I emphasizes changes in hardware and refrigerants that can 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 is 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. With regard to the phase I options, it was assumed that R-12 would be replaced with a non-ozone-depleting refrigerant in all of the experimental evaluations.

Phase I Options Options for improving conventional RF efficiency fall into four general areas: (1) improving the refrigeration cycle efficiency, (2) decreasing the cabinet heat load, (3) reducing parasitic electrical loads, and (4) 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 and 8) all affect the parasitic electrical load but also have an impact on the cabinet heat load. Options 7 and 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. Option 10, a liquid line shut-off valve, which has been extensively used with rotary compressors, prevents hot refrigerant migration 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.

MODELED ENERGY SAVINGS RESULTS

The modeling program was used to evaluate the energy-saving potential of several of these phase I design options. Table 2 lists the baseline and new model inputs used to obtain the results shown graphically in Figure 2. Since built-up units were used as experimental breadboards, some design changes *could not* be conveniently retrofitted into the cabinets 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 this figure. 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 these features. It is not meant to imply that these loads can be totally eliminated.

The other sector of energy savings shown in Figure 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/day) energy savings. Changing the compressor, improving cabinet insulation by 18% to 20%, using electrically commutated direct-current fan motors, and installation of a liquid line shut-off valve are all changes that could be experimentally implemented with relatively available components and a minimum of cabinet deterioration. Specifications of obtainable components were used as inputs to the model for the simulation calculations.

TABLE 1
Design Options to Improve the Energy Efficiency of a Conventionally Designed Refrigerator-Freezer

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

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 Watts	4.0 Watts
Condenser Fan Motor	13.5 Watts	8.0 Watts
Cabinet Load (@ 90°F) ($Q_{Freezer}/Q_{Total}$)	292.6 Btu/h (0.51)	250.8 Btu/h (0.52)
Liquid-line Shut-off	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

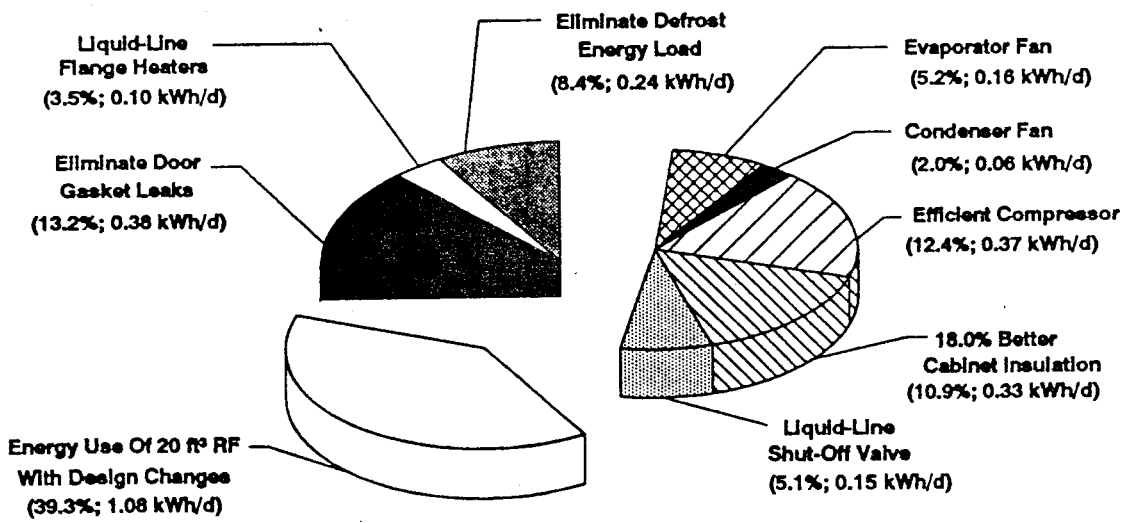

Figure 2 *Modeled energy savings from design changes—1990 base unit (2.87 kWh/day). Changes applied individually against base.*

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/day baseline starting point; thus, when all 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 Figure 2. For example, when the improved compressor, heat exchanger fans, insulation, and liquid line shut-off are combined into one RF, the net energy-use reduction is 31.2% (0.90 kWh/day) rather than the 35.6% (1.02 kWh/day) total suggested in Figure 2. It is apparent from this analysis that, even with the optimistic estimates calculated by comparing each improvement to baseline RF energy use and maximizing energy saving predicted from eliminating defrosts, gasket door leaks, and antisweat cabinet heaters, the program goal of 1.00 kWh/day or less was not met. Additional energy savings are needed from even more efficient components, better refrigerants, or an advanced design that utilizes a more efficient thermodynamic cycle to reach this ambitious target. Clearly, as evidenced by Figure 1, the domestic RF is rapidly reaching the point of diminishing returns as an opportunity to save significant amounts of national energy.

EXPERIMENTAL RESULTS

Reverse Heat Loss Tests

Steady-state heat loss measurements were performed on the baseline RF cabinets "as received" and cabinets with externally mounted expanded polystyrene panels to simulate reduced-load cabinet designs. Data similar to those plotted

Results Summary of Refrigerator Test Data
−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:	160.5	130.6	291.1	0.55
Modeled:	148.6	144.0	292.6	0.51
Cabinet w/Added Insulation:				
Experimental:	129.1	107.8	236.9	0.54
Modeled:	129.3	121.5	250.8	0.52

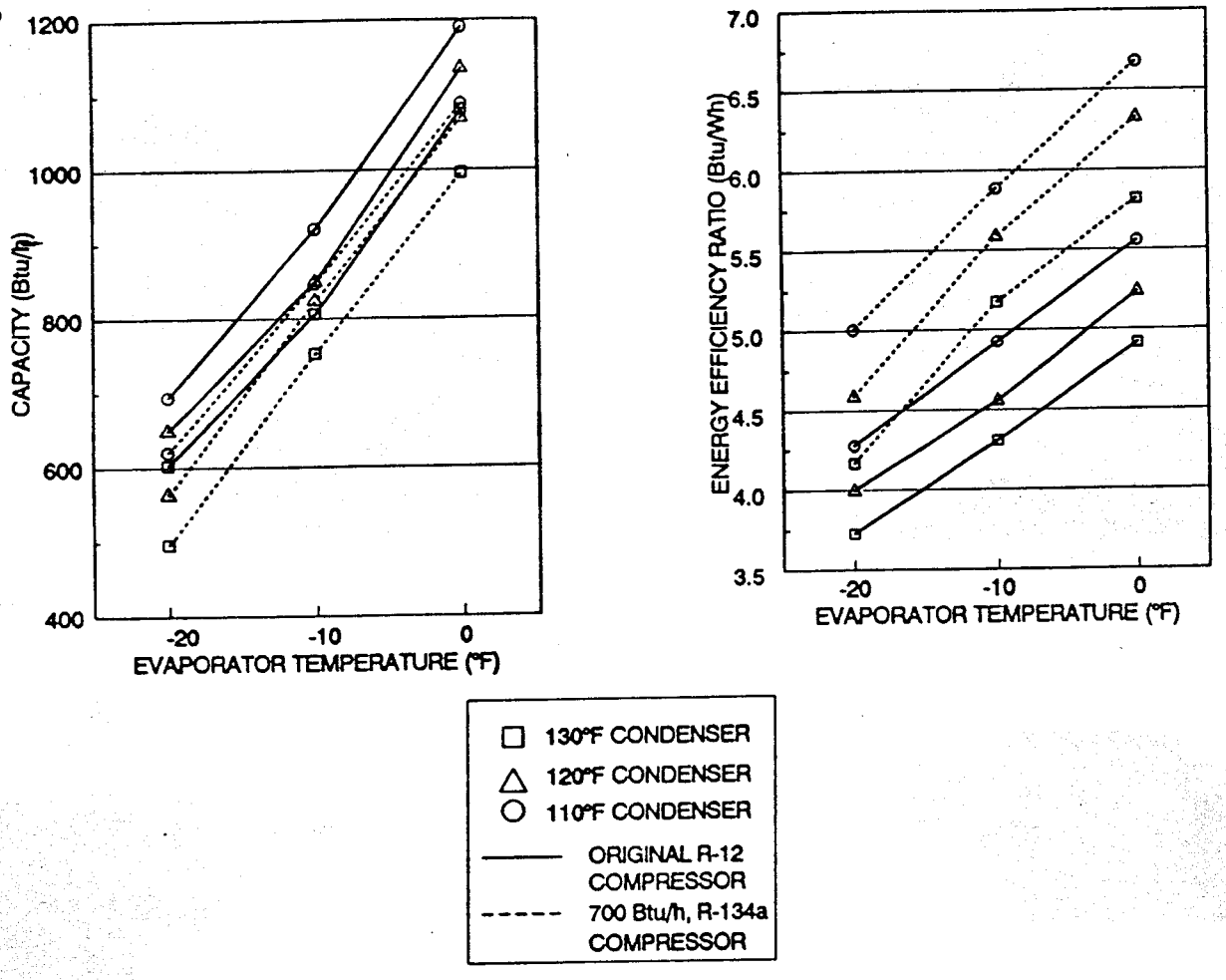

Figure 4 *Compressor calorimeter results.*

analytical results are listed for cases D and E because the RF model used does not have enough 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% to 5% discrepancy is seen between our experimentally measured, "as received" energy consumption 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,

in Figure 3 were obtained. In this figure, the heat loss rate under steady-state conditions, measured in Btu/h (W), is plotted against the difference between air temperatures inside the RF compartment and cold air temperatures (≈ 0°F to 5°F [−18°C 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 (Equation 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.

$$Q = UA \cdot \Delta T \tag{1}$$

A 15% to 20% improvement in the thermal resistivity of the cabinet was chosen as a reasonable design improvement that could be obtained through the implementation of 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

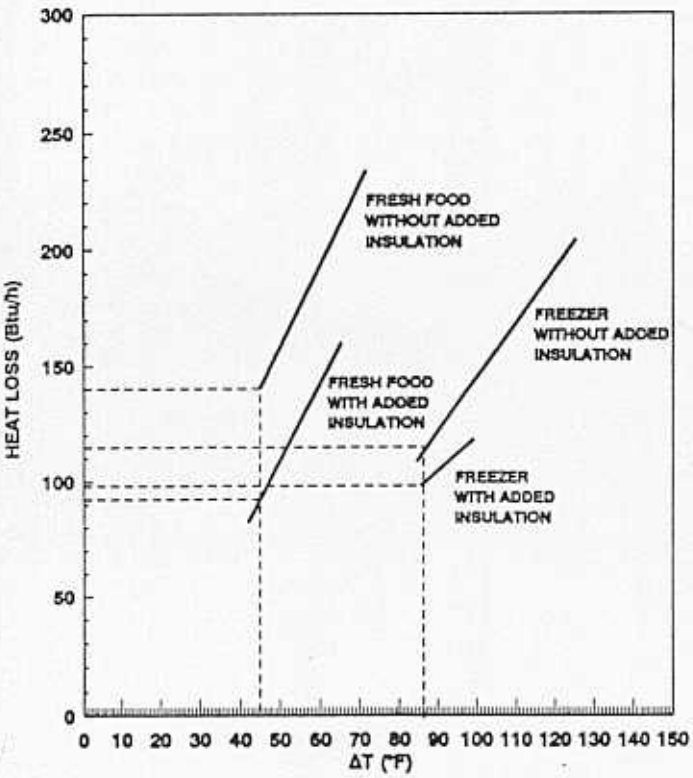

Figure 3 *Cabinet heat loss measurements for 1990 cabinet with and without added insulation.*

(Q_{FRZ}/Q_{TOT}) ratio. Theoretically, RF energy use 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 an 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 dimensions, foam thickness, 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% from 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 differing heat transfer characteristics than one with added insulation and a metal exterior surface, but the exterior panels were used as a concession to experimental expedience.

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 graphically in Figure 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 compressor with an EER of 4.3 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 the R-12 compressor it replaced. Less refrigeration capacity is required with the supplementary cabinet insulation. Using a compressor whose capacity is 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.

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 Figure 5. No

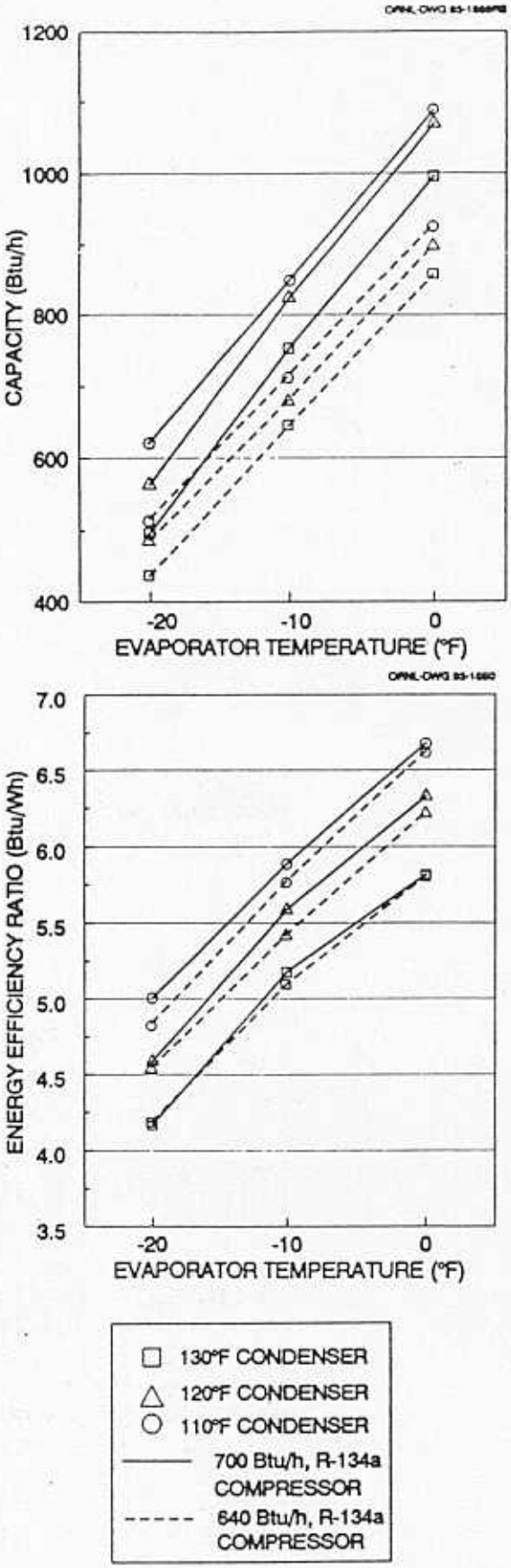

Figure 6 Compressor calorimeter results.

CONCLUSIONS

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 result in misleading information. Certainly these changes will reduce the thermal load and decrease 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 loading of refrigerators with perishable products that 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 of actual RF field energy use.

Also, better insulated compartments will require refrigeration systems with smaller and smaller capacity compressors to achieve a balance between thermal load and refrigerating capacity. The efficiency of RF compressors falls off sharply at lower capacities, as shown in Figure 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 shut-off valve, and redirected airflows 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 design changes have correlated 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.

ANALYSIS OF FUTURE DESIGN OPTIONS

Two 1993 20-ft^3 (570-L) cabinets have been 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/day. Starting with these RFs, the computer model was used to determine what design refinements would be needed to meet the program goal utilizing a *conventional design* (Figure 8). Design options were added to the model sequentially so that the cumulative effects of each option would be accounted for. The results in Figure 8 show that the most significant savings come from a high-efficiency, variable-capacity compressor; better cabinet insulation; and a liquid-line shut-off valve. More efficient fan motors and increased heat exchanger areas also account for sizable energy savings. Since these changes resulted in an RF whose predicted energy use was still 1.07 kWh/day, further modifications were simulated even though smaller and smaller energy savings resulted. Energy use of 1.00 kWh/day was only achieved after additional changes were

TABLE 4
90°F Closed-Door Cumulative Daily Energy Consumption/Results Run Time
—1990 Top-Mount Units—Experimental and Analytical Results/Design Options

Case	Description	Experimentally Measured	Analytically Calculated	Manufacturers 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% Increased 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 watt 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/hr Compressor and liquid line shut-off			
	Energy	2.07 kWh/d	—	—
	Run Time	46.6%	—	—

results shown in Figure 5 are obtained. These normalized results indicate that the model accurately reproduces energy consumption trends due to circuit modification and hardware changes.

The improvement in RF performance as a result of increasing the air-side area of the evaporator by 33% (case D) is disappointing if it is only viewed 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 spirally wrapped 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.

To fully appreciate the effect of this heat exchanger change, other system operating characteristics, such as compressor run times, compressor cycles per day, and air/refrigerant approach temperatures or the high-to-low-side pressure ratios, must be considered. In addition to the 1.9% decrease in energy consumption, the enhanced evaporator reduced compressor run times from 49.8% to 41.3%, decreased 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 to 31 to 27 to 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 effectively be used 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

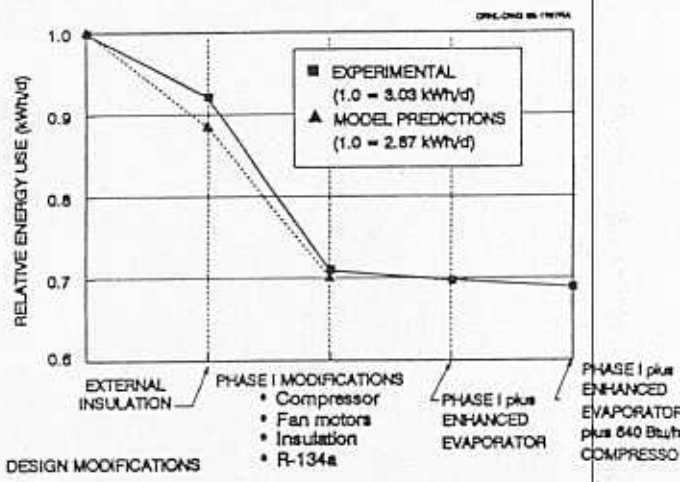

Figure 5 *Normalized improvements in refrigerator performance—Experimental and modeled results.*

calorimeter runs on these two R-134a compressors is shown in Figure 6. RF system energy use decreased from 2.10 kWh/day to 2.07 kWh/day despite a less efficient compressor and longer run times. Improvement of efficiency is attributable to a better match between the refrigeration circuit and cabinet heat load. Using measured run times and an assumption that 75% of the electrical energy used by an 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/day (67% to 68% of the base) if a 5.2-EER compressor were available in this capacity range.

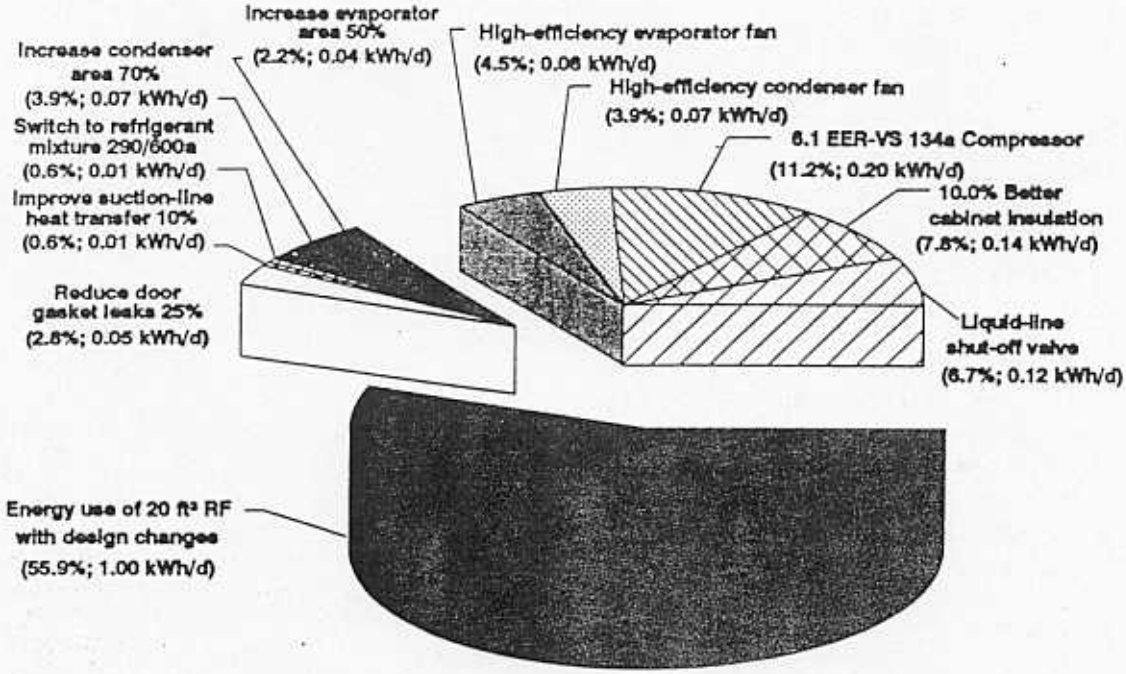

Figure 7 *Sampled compressor efficiency versus refrigeration capacity data—130°F (54.4°C) condensing, −10°F (−23.3°C) evaporating temperature.*

incorporated into the model for improved door gaskets, better suction-line heat transfer, and use of a more efficient refrigerant. These new cabinets will now be used as laboratory test beds to attempt to experimentally duplicate the path laid out to achieve the project goal.

Analytical modeling studies indicate that improvements to the standard refrigerator-freezer 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 utilizes a more thermodynamically efficient cycle such as the Lorenz cycle, where heat transfer irreversibilites are minimized. If required, two advanced cycle designs are being considered for experimental investigation in future phases of this project:

- a single-evaporator RF operating with a low glide (10°F to 20°F [5.6°C to 11.1°C]) refrigerant mixture and counterflow heat exchangers and a variable-capacity compressor, and
- a two-evaporator Lorenz-Meutzner RF operating with a high glide (35°F to 50°F [20°C to 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.

CONCLUSIONS

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

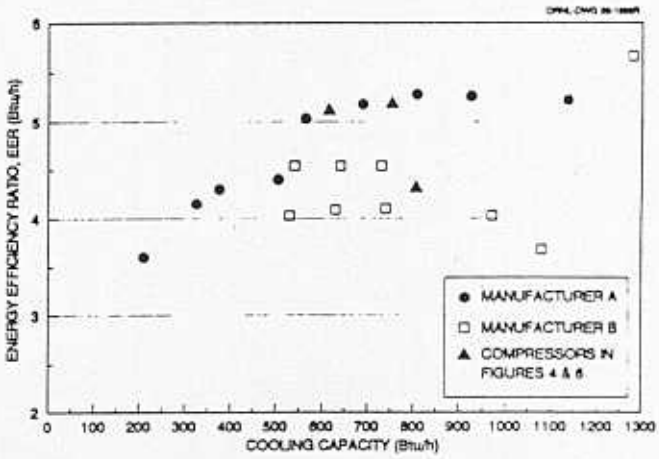

Figure 8 *Modeled energy savings from design changes—1993 base unit (1.79 kWh/day). Changes applied sequentially with liquid-line shut off.*

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 an 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 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.

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REFERENCES

AHAM. 1985. *AHAM standard for household refrigerators and household freezers*. Chicago: American Home Appliance Manufacturers.

Bohman, R.H. 1987. Enhancing the energy efficiency of refrigerator/freezers. First National Conference on Appliance Efficiency, Arlington, VA.

EPA. 1993. Multiple pathways to super-efficient refrigerators. EPA-430-R-93-008, June. Washington, DC: United States Environmental Protection Agency.

Meier, A., and R. Jansky. 1993. Field performance of residential refrigerators: A comparison with laboratory test. *ASHRAE Transactions* 99(1).

NAECA. 1987. Public Law 100-12, March 17 (as amended by National Conservation Policy Act, 9 Public Law 95-619), Part B—Energy conservation for consumer products other than automobiles.

Turiel, I., and A. Heydari. 1988. Analysis of design options to improve the efficiency of refrigerator-freezers and freezers. *ASHRAE Transactions* 94(2): 1699-1712.

Zurer, P.S. 1993. Ozone depletions recurring surprises challenge atmospheric chemists. *C&E News*, May 24, pp. 8-18.