CONF-950624--1

IMPROVING THE ENERGY EFFICIENCY OF REFRIGERATORS IN INDIA

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Research sponsored by the U.S. Department of Energy (DOE) and the Asian Environmental Partnership Program through a grant provided by the Assisting Deployment of Energy Practices and Technologies (ADEPT) program and by the U.S. DOE Office of Building Technologies, under contract DE-AC05-84OR21400 with Martin Marietta Energy Systems Inc.

ABSTRACT

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Five state-of-the-art, production refrigerators from different manufacturers in India were subjected to a variety of appliance rating and performance evaluation test procedures in an engineering laboratory. Cabinet heat loss, compressor calorimeter, high-ambient pull-down, and closed-door energy consumption tests were performed on each unit to assess the current status of commercially available Indian refrigerators and refrigerator component efficiencies. Daily energy consumption tests were performed at nominal line voltages and at 85% and 115% of nominal voltage to assess the effect of grid voltage variations. These test results were also used to indicate opportunities for effective improvements in energy efficiency. A widely distributed "generic" computer model capable of simulating single-door refrigerators with a small interior freezer section was used to estimate cabinet heat loss rates and closed-door energy consumption values from basic cabinet and refrigeration circuit inputs. This work helped verify the model's accuracy and potential value as a tool for evaluating the energy impact of proposed design options.

Significant differences ranging from 30 to 90% were seen in the measured performance criterion for these "comparable" refrigerators suggesting opportunities for improvements in individual product designs. Modeled cabinet heat loadings differed from experimentally extrapolated values in a range from 2—29%, and daily energy consumption values estimated by the model differed from laboratory data by as little as 3% or as much as 25%, which indicates that refinement of the model may be needed for this single-door refrigerator type. Additional comparisons of experimentally measured performance criteria such as % compressor run times and compressor cycling rates to modeled results are given. The computer model is used to evaluate the energy saving impact of several modest changes to the basic Indian refrigerator design.

Keywords: Refrigerators, Refrigerator-Freezers, Energy Saving, Modeling, Rating

INTRODUCTION

As a signatory to the Montreal Protocol which ultimately bans the manufacture and use of Ozone Depleting Substances (ODS), India must develop a national strategy for changing over its emerging industrial base to non-ODS technologies. The refrigerator/refrigerator-freezer (RF) manufacturing sector will be an important component of this strategy because of its current dependence on CFC-chemicals, present size, rate of growth, and ultimate market potential. Early implementation of a workable path to the most energy-efficient, non-ODS RF is especially important in India to minimize the demand this growing market will place on a severely constrained electric power generating system.

It is projected that over 100 million RFs will be sold in India between now and 2010. This will increase market saturation from about 6—8% currently, to approximately 60% [Statt, 1992]. The energy conserving strategies and techniques that have resulted in dramatic improvements in U.S. RFs over the last twenty years and those which have been proposed to ease the transition to chlorine-free refrigerants and blowing agents should be considered in terms of what could suitably be applied to the common Indian RF.

The prevailing social, demographic, and economic conditions control the design of a refrigerator in India in a unique manner. High ambient temperatures, consumer shopping patterns, disposable income, availability of reliable repair services, the quality of electrical power, etc. all contribute to a refrigerator design that makes sense for the consumer. Indian units are generally in the 2.3 to 13.4 cubic foot (65 to 380 liter) range, with the 6.8 cubic foot

(165 liter) model accounting for 93% of 1990 sales. Compressors used on refrigerators are oversized to withstand the poor quality of electrical power which ranges from 125—270 VAC @ 50 Hz. Consequently, the average annual power consumption of 6.8 cubic foot Indian refrigerators is about 500 kilowatt-hours per year (kWh/y); about 25—30% higher than that for U.S. refrigerators of the same size [IIEC, 1991].

Government policies and practices are also important in establishing the type of refrigerator ultimately favored by the consumer. At present, India has an additional excise tax on refrigerators larger than 6.8 cubic feet (165 liters), and no incentives are provided for purchasing appliances with exceptional energy efficiency. The *voluntary* guideline for refrigerator energy consumption, 730 kWh/y for a 6.8 cubic foot (165 liter) cabinet [Indian Standard (IS), 1987], is not sufficiently rigorous to force innovation by manufacturers, unlike the energy use levels mandated by the National Appliance Energy Consumption Act (NAECA) in the United States [NAECA, 1987].

Three essential parts of the approach chosen for developing an efficient, environmentally acceptable refrigerator for India call for an evaluation and assessment of existing products by an impartial third party, establishing workable efficiency standards against which refrigerators can be rated and compared, and setting up a centralized refrigerator testing facility in India. Work described in this paper addresses the first two tasks.

PROJECT SCOPE AND TESTING PROTOCOLS

An experimental plan for U.S. testing of refrigerators manufactured in India and designed for the Indian market was formulated to encompass the four most common aspects of efficient and generally acceptable refrigerator operation:

- Compressor performance
- Cabinet heat loss rates
- Pull-down performance
- Daily energy consumption

Relevant aspects of American Society of Heating, Refrigerating, and Air-Conditioning Engineers Inc. (ASHRAE); Association of Home Appliance Manufacturers (AHAM); American National Standards Institute (ANSI); Indian Standard (IS); and several unpublished testing procedures were combined to give a laboratory procedure that fit the unique character of these products while offering some common ground for international comparisons [U.S. CFR, 1990] [Indian Standard, 1987].

Five representative refrigerators manufactured in India for the Indian market were obtained for laboratory testing. Four of these units were 6.2—6.8 cu. ft (150—165 liter), single-door cabinets with an internal static evaporator compartment, static condenser, and manual defrost (U.S. DOE Class 1). One refrigerator was a two-door, 5.0 cu. ft (142 liter),top-mount refrigerator-freezer with a fan forced evaporator, no-frost operation, and auto defrost (U.S. DOE Class 3).

Compressor Calorimetry. Calorimeter "maps" were generated for each of the compressors supplied with the refrigerators using a modification of the AHAM nine-point matrix of saturated evaporating and condensing temperatures usually prescribed for refrigerator compressors. Refrigeration capacities and compressor efficiencies measured in these procedures were needed to compare compressors from these products to those available elsewhere, to establish a range for the "best" to "worst" compressors in the sampling of units supplied to our laboratory, to further analyze why certain refrigerators performed better or worse than others, and to provide reliable inputs for the computer modeling work.

In the calorimeter procedure used for this work, compressor operating characteristics including refrigeration capacity (watts or Btu/hr) and energy efficiency ratios (COP or Btu hr^{-1} $watt^{-1}$) were determined at each point in a matrix of 120°F (48.9°C) and 130°F (54.4°C) condensing temperatures and -10°F (-23.3°C), 0°F (-17.8°C), and +10°F (-12.2° C) evaporating temperatures. Also specified in the test procedure are a 90°F (32.2°C) ambient temperature for the compressor, superheating of the compressor suction gas to 90°F (32.2°C), and subcooling of the liquid refrigerant line to 90°F (32.2°C) before a throttled expansion. The usual nine point map includes a series at 110° F (43.3° C) saturated condensing temperature and -20° F (-28.9° C) saturated evaporating temperature, but these tests were not performed because of the use of static condensers on the test units and due to the lower capacity limit of the calorimeter used for the measurements, respectively.

Cabinet Heat Loss Testing. No known or published procedure was located for measuring the rate of heat permeation into insulated refrigerator cabinets -- cabinet-heat-loading. "Reverse heat loss" experiments were performed by placing the refrigerator in a cold chamber with a controllable, monitored source of heat in both the freezer and fresh food compartments. An attempt was made to obtain temperature differences across the cabinet walls that were comparable to those prescribed in the 32°C (89.6°F) closed-door energy consumption test [Indian Standard, 1987] where the refrigerator works to maintain cold internal temperatures in a hot room [Sand, et. al. 1994]. The main premise of the reverse heat loss test 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(s), and gaskets into the cold room. The thermal conductivity of CFC-11 blown, polyisocyanurate insulation is relatively constant over the applicable temperature range, but the direction of convective flow is reversed leading to minor inaccuracies.

A linear relationship between heat loss rate and temperature (equation 1) is assumed. These determinations were used to assess the effectiveness of cabinet insulation, to measure differences between the "best" and "worst" cabinets, and to provide experimental data that could be compared to cabinet-heat-loadings predicted by the computer model.

$$Q_{Cabinet} = UA_{Feezer}(\Delta T_{Feezer}) + UA_{Fresh\ Food}(\Delta T_{Fresh\ Food}) \tag{1}$$

Procedurally, The reverse heat loss experiments were performed with a single heat source (in the form of a string of small Christmas tree lights) distributed between the freezer and fresh food sections with a variable voltage control. Differences between the steady state compartment temperatures and the cold ambient temperature together with the electrical energy provided to the heat source were fitted to equation 1 to empirically determine the $UA_{Freezer}$ and $UA_{Fresh\ Food}$ values. At least four or five individual measurements were regressed to determine UA parameters for each cabinet. Several different heat input rates and distributions of heat (lights) between the freezer and fresh food sections were used to assure uniquely characteristic values for these empirical fits.

No-Load Pull-Down Tests. The 109.4° F (43° C) no-load pull down was also performed on all of the Indian refrigerators as specified in the IS [1987] with the thermostat (and defrost timer for refrigerator "E") defeated. Full temperature and wattage versus time curves were generated for each refrigerator so times and kilowatt-hours needed to achieve preselected temperatures were easily determined. Times and power consumption needed to reach a mean of 44.6° F (7° C) in the fresh food compartment were precisely measured. The tests were continued for 18—24 hours so that final/steady-state temperatures and compressor wattages could also be recorded. This pull-down test was performed at only the nominal Indian voltage condition, 230±1% VAC, 50Hz. This procedure is essentially the same as the 110° F (43.3° C) pull down test used by U.S. manufacturers. A summary of these test conditions are given in Table 1.

The no-load pull-down test can give the system design engineer insight as to the "balance" of the principal components in the refrigeration system. These are:

- The compressor -- pumping rate, motor torque, motor protection, etc.
- The condenser -- heat dissipation rate, internal volume, etc.
- The capillary tube -- restrictive characteristics
- The evaporator -- heat absorption rate, internal volume, etc.
- The refrigerant charge -- properly or improperly charged.

A well designed system that is properly balanced will display the following performance characteristics on the pull-down test.

1. The ability to achieve the desired internal temperatures within a reasonable time period after start-up. This is affected by compressor pumping rate, evaporator size, heat absorption rate and refrigerant charge.
2. The ability to pass the dynamic peak load point during the pull-down cycle without the compressor stalling or tripping the compressor motor protector. This performance characteristic is affected by compressor motor torque, the motor protector rating, capillary tube size, condenser size, and the heat dissipation rate.
3. The ability to achieve steady-state operation while exhibiting acceptable internal temperatures, system and component temperatures, and compressor power input.
4. The ability to start and continue operation under the conditions and "soak out" pressures developed with the refrigeration circuit at high ambient temperatures. This is affected by the condenser and evaporator internal volumes, total refrigerant charge, and compressor motor torque.

The no-load pull-down test in a 109.4° F (43° C) ambient is very rigorous and is an essential criteria to assess a well-balanced system design for household refrigeration.

Energy Consumption Testing. The 89.6° F (32° C) closed door energy consumption test as specified by the IS [1987] is reasonably comparable to the DOE/AHAM 90° F rating test used for U.S. products [U.S. CFR, 1990]. The test procedure used to obtain energy consumption results reported in this work was set up to achieve a 41° F (5° C) mean temperature in the fresh food compartment with a freezer temperature that does not exceed 23° F (-5° C). This differs from the DOE/AHAM test procedure which calls for a 15° F (-9.4° C) mean temperature in the freezer and 38° F (3.3° C) in the fresh food. Unfortunately, a recently reported correlation and comparison between various testing standards for household refrigerators and freezers does not include results obtained using the IS [Bansal, 1995].

When the thermostat could not be adjusted to obtain a mean compartment temperature within ±1.8° F (±1° C) of that specified in the procedure, two measurements were made at thermostat settings on either side of the ideal fresh food temperature and interpolation was used to determine energy-use, run times, compressor cycling frequencies, freezer temperatures, etc. at a fresh food temperature of 41° F (5° C). Thermocouple and external baffling arrangements were as specified in the IS procedure, and all the refrigerators were tested at 230±1 VAC, 190±1 VAC, 260±1 VAC, and 50 Hz which was obtained with an Elga 3001 power supply equipped with a model 401 variable oscillator.

Daily energy consumption results for the four manual defrost refrigerators were extrapolated from steady state running data accumulated over a minimum of six hours. Rather than adhering to a strict six hour time limit, an integral number of complete compressor cycles over this period were used for the extrapolation. Daily energy consumption for the one automatic defrost refrigerator was calculated using a complete defrost-to-defrost cycle and by an extrapolated steady-state method using compressor cycles between defrosts for comparison of these two procedures. This allowed estimates for the energies used by a defrost and the subsequent recovery after a defrost for the manual defrost models.

Table 1. Energy Consumption and No-Load Pull-Down Testing Conditions for Indian Refrigerators		
	Energy Consumption Test	No-Load Pull-Down Test
Ambient Temperature	89.6°F (32°C)±1.0°F	109.4°F (43°C)
Operating Power	230 VAC 50 Hz 190 VAC 50 Hz 260 VAC 50 Hz	230 VAC 50 Hz
Freezer Temperature	<23.0°F (<-5°C)	Steady-state
Fresh Food Temperature	41.0°F (5°C) (mean)	<44.6°F (<7°C) (mean)
Measured Results:	Kilowatt-hour/day (extrapolated and defrost-to-defrost)	Time to F.F. + 44.6°F (7°C)(mean) Complete pull-down curve

COMPUTER MODELING

Computer simulations of these refrigerators were made with a public domain, detailed refrigerator system model that could simulate compressor cycling behavior and single-door refrigerators with an enclosed freezer. These simulations were correlated with laboratory results and eventually used to assess the energy saving benefits and cost effectiveness of potential design changes. Specifications unique to each refrigerator were obtained from the various manufacturers and used as inputs to the model. Actual compartment temperatures obtained in the energy consumption testing and compressor efficiency and capacity data from the laboratory calorimeter testing were used in the simulations to make them correspond as closely as possible with actual testing conditions.

EXPERIMENTAL RESULTS

Compressor Calorimeter Results. A summary of measured and manufacturer supplied compressor performance values with CFC-12 at the -10° F saturated evaporating and +130° F saturated condensing rating point for each of the Indian refrigerator compressors is given in Table 2. The experimental results indicate that these compressors are, on the average, 25% less efficient than "state-of-the-art" compressors in the same capacity range [EPA, 1993a]. Some of this fall off in efficiency is attributable to the oversized motor windings built into the compressors to handle the variable line voltages in India. It is surprising that a 24% difference in COP is seen between the "worst" compressor in refrigerator "A" and the "best" compressor in refrigerator "C." This would imply that the energy consumption of refrigerator "A" could be significantly improved if the "best" compressor available in our limited sampling were used in this refrigerator.

Table 2 also indicates that some refrigerator manufacturers are using compressors that have measured COPs 12.9% less that their "catalog" ratings. This discrepancy between ratings published by compressor manufacturers for their products and actual, experimentally measured compressor performance is, apparently, not that unusual [Merriam, 1994] [EPA, 1993a]. The results imply that some allowances must be applied when published information from manufacturers is used as inputs in simulations models and the results compared to laboratory measurements.

Cabinet Heat Loss Results. A summary of experimental *heat loss* rates for the refrigerator cabinets and those calculated using individually fit empirical equations of the form shown in equation 1 is given in Table 3. The best, least squares, linear regression fit equations resulting from the experimental data are also shown. Four or five data sets contributed to each row in this table.

Experimentally controlled temperature differences between the heated freezer and fresh food compartments and the cold ambient maintained in an environmental chamber were adjusted in a narrow range to roughly parallel the ΔT's seen in the standard, closed-door, energy consumption test. This procedure specifies a 23° F (-5° C) freezer and a 41° F (+5° C) fresh food compartment in a 89.6° F (32° C) environmental chamber. For the *reverse* heat loss tests a 23° F (-5° C) external ambient chamber was used with a freezer heated to 89.6° F (32° C) and the fresh food section at 71.6° F (22° C). This is a 66.6° F (37° C) ΔT in the freezer and a 48.6° F (27° C) ΔT in the fresh food, similar, but opposite in direction to the temperature differences established in the closed-door energy consumption tests.

To avoid fitting experimental data with the additional constraint of a fixed ratio between the freezer and fresh food ΔT's (1.37:1.00 ideally), the internal light distribution (heat loading) between compartments was changed between heat loss experiments. This helped insure a more general applicability of the regression coefficients generated by the fitting program in that they were not dependent on the maintenance of this ratio for accurate simulation of cabinet heat loss rates. The apparent inconsistency in the relative magnitude of the freezer and fresh food ΔT coefficients for refrigerator "A" as compared to those for the other refrigerators could indicate that larger variations of the heat distribution may have been needed for measurements on this unit. Extrapolation of the empirical equation for refrigerator "A" with ΔT's outside normally encountered conditions would highlight this problem.

Starting with ideal temperatures for each of the food compartments (see above), it is possible to rearrange the terms in these equations to predict cabinet heat loading as a function of ambient temperatures. The plots shown in Figure 1 were generated in this manner. A 10-20% difference exists between the "best" and the "worst" cabinets tested. The cabinet on refrigerator "C" consistently showed the best performance at each ambient temperature and refrigerator "B" generally indicated the "poorest" (highest) heat loss rate.

Comparisons between cabinet heat loading values "predicted" from these experimentally-based empirical equations and those calculated from the cabinet simulation section of the generic computer model are given in a subsequent section of this paper.

Pull-Down Results. The transient and steady-state data resulting from laboratory pull-down testing of these refrigerators are summarized in tables 4 and 5, respectively. Data obtained from a typical pull-down test is shown in Figure 2. Table 4, emphasizes the time and power consumption needed to pull the fresh food temperature average down from 109.4° F (43° C) to 44.6° F (7° C) after the refrigerator was "soaked out" in a 109.4° F environmental chamber. Also listed in Table 4 is the freezer temperature at the time that 44.6° F was achieved in the fresh food compartment. Obviously, the time and energy needed to obtain a 44.6° F temperature in one compartment will be affected by the temperature the system maintains in the freezer.

The steady-state, 100% compressor run conditions established at the end of the 109.4° F (43° C) pull-down test which are listed in Table 5 illustrate more extensive internal baffling between the freezer and fresh food in refrigerator "A". The data clearly indicate the larger refrigeration capacity of the compressor and refrigeration circuit in refrigerator "C" (see Table 2). This combination of additional capacity and the most efficient compressor of those tested in this program give refrigerator "C" its rapid pull-down rate, ability to cope with higher ambient temperatures, and quickly cool off items added to the appliance for refrigeration.

A surprising variation is seen between the "best" and "worst" refrigerator performance. Refrigerator "A" took roughly twice as long as the others to reach the required fresh food temperature and consumed nearly twice as much energy when compared to similarly designed units. This can partially be attributed to the lower freezer temperature being maintained by this refrigerator, which, in this single-door/static-evaporator design, means that the freezer section is more enclosed, thereby impeding free convective circulation in the insulated cabinet. Naturally, this will lengthen the time (and the power) needed to cool down the fresh food compartment.

This augmented separation of freezer and fresh food compartments in refrigerator "A" provides a distinct advantage in the energy-consumption tests reported later in this paper because it makes the 23° F (-5° C) freezer/41° F (+5° C) fresh food temperature separation needed for the energy-consumption test easier to maintain. In refrigerators "B", "C", and "D", the ≤ 23° F temperature limiting condition for the freezer that is spelled out in the Indian test standard was exceeded before an *ideal* 23° F temperature could be obtained in the fresh food. See the Daily Energy Consumption Results section for additional explanation.

Refrigerator "E" is also uniquely different from the other units included in Tables 4 and 5 because it is a two-door, automatic defrost model with a freezer temperature control. This unit reaches and maintains the lowest freezer temperature in the pull-down tests despite adjustment of the freezer temperature control to its warmest setting. Quite obviously, the features and increased versatility of this two door design are going to result in additional energy use. This philosophy of additional features necessitating additional energy is apparent from the way different maximum energy consumption standards are calculated for different RF product classes in the National Appliance Energy Conservation Act (NAECA) regulations for U.S. products. The refrigerators with more features like through-the-door ice makers or side-by-side construction are not required to meet as stringent energy use standards.

Daily Energy Consumption Results. Table 6 summarizes the measured daily energy use results of these Indian refrigerators using the 89.6° F (32° C) closed-door rating procedure outlined in the Indian Standard for refrigerator testing [IS, 1987]. As previously mentioned, these laboratory measurements were performed at operating voltages of 190 VAC, 230 VAC, and 260 VAC and 50 Hz to clearly indicate the effects of the variable voltage conditions in India. Voltages as low as 125 or 160 VAC are commonly experienced in some areas, but no testing was performed below 190 VAC to avoid the possibility of burning out or damaging compressor motors. Voltage stabilization transformer units are available in India where this is a recurring problem.

The data show quite clearly that lower line voltages resulted in more efficient refrigerator operation as measured by this test. One hundred ninety (190) VAC was not low enough to cause excess motor heating and overload trip-outs, although, the energy saving benefits of lower line voltages will, at some point, be counteracted by equipment failure problems.

The energy use results reported in Table 6 show a striking variability between units. A large portion of this variability in energy consumption is attributable to the inability of refrigerators "B", "C", and "D" to meet the minimum requirements of the Indian test standard as it is currently written. All three of these refrigerators were rated running at fresh food temperatures below the 41° F (+5° C) temperature specified in the standard because of the additional requirement in the procedure that the freezer temperatures remain at or below 23° F (-5° C) throughout the test. The internal partitions in refrigerators "B", "C", and "D" are not tight enough to establish the ideal 23° F to 41° F (-5° C to +5° C) temperature separation between the freezer and fresh food compartment at any thermostat setting. Therefore, the energy consumption results were determined at a thermostat setting that would result in a 23° F maximum for the freezer temperature, forcing the fresh food to values lower than the 41° F ideal. Maintaining these lower fresh food temperatures contributes significantly to the refrigerator's daily energy use.

Clearly, the rated energy use of unit "B" would be improved if internal baffling allowing disruption of the convective flow of cold air inside the cabinet and larger temperature differences could be established between freezer and fresh food compartments. Some company representatives viewing these results indicated that the defrost trays on some units have moveable flaps whose position can be adjusted to improve freezer segregation. Alternatively, if the present standard is a poor representation of refrigerator use in India, the standard should be changed to specify testing conditions indicative of field applications of this appliance.

Historically, DOE type 1 refrigerators of 6.8 ft^3 (165 liters) and larger (up to 12 ft^3) built by U.S. manufacturers operate most efficiently with a compressor cycling rate of 3—4 cycles per hour. Frost-free refrigerators in the U.S. typically operate at 24—40 cycles per day with a 50% run time in the DOE/AHAM 90° F, closed-door rating test. Single-door models "B" and "C" rated in this study averaged 120 to 180 cycles per day with 45% and 30% run times, respectively. This would indicate that the refrigeration capacity in these units is oversized or that the thermostat dead-band was too narrow, at least for this testing condition, and that some of the operating inefficiencies are due to cycling losses. One explanation for excess capacity may be that it is a carry-over from the use of fiber glass cabinet insulation which was not as effective as foams at preventing <u>all</u> forms of heat infiltration such as convection and radiation.

The apparent excess capacity in models "B" and "C" suggest that it was designed into these products for a reason. Oversized refrigeration capacity minimizes pull-down time, provides for routine operation at ambient conditions more demanding than the test conditions, supports the previously used testing conditions at a 109.4° F (43° C) ambient, and allows for staying with larger sizes of compressors that have better efficiency ratings. These frequent cycling and short on-time results may also be related to the unsatisfactory power situation in the country where the appliance is forced to operate at voltages well below 230 VAC. These lower, more typical voltages increase on-time and decrease the number of cycles in order to meet a fixed load.

The model used for analytical simulation of these refrigerators permitted specification of the compressor cycling frequency as one of the input parameters. When this parameter was changed from 180 cycles per day to 84 cycles per day (3.5 cycles per hour) for refrigerator "C" the predicted daily energy consumption decreased by 4.4% and the cycling COP of the compressor improved from 0.889 to 0.932. One way to achieve this reduction in cycling frequency is to increase the thermostat dead-band.

Refrigerator "E" is built to meet a dramatically different design and function, so its cycling and on-time performance are more typical of refrigerators available in the United States.

MODEL SIMULATION RESULTS

One of the more interesting aspects of the work and perhaps the most useful outcome involves comparisons between experimentally measured refrigeration performance and predicted performance using a generic refrigerator, RF model. The obvious advantage of a validated model is that it can quickly and easily be used to simulate the effects of changes which would require considerable time and effort to demonstrate and document experimentally.

There are two definitive areas where the modeling predictions can be compared to measurements made in the laboratory, cabinet heat loadings and daily energy consumption. Data in Table 7 compares cabinet heat loss rates predicted by the refrigerator model to those calculated from the empirical equations resulting from reverse heat loss experiments. Actual compartment and ambient temperatures measured during the daily energy consumption tests shown in Table 6 were used as inputs to generate these results. Additionally, internal and external cabinet dimensions, gasket heat transfer rates, and insulating foam characteristics obtained from the manufacturers were used as inputs to the cabinet model program for refrigerators "B" through "E". This data was not provided for refrigerator "A" so dimensional measurements were made on the cabinet in the laboratory and the foam and gasket values typical of the other refrigerators in this sampling were used to obtain modeled results for "A".

Very good correspondence ($\pm 4\%$) was obtained between modeled and measured cabinet loading rates for refrigerators "A" and "B". Significantly poorer, more pessimistic, (+16—24%) cabinet heat loss rates are predicted by the model for refrigerators "C", "D", and "E". Some refinement of the cabinet simulation portion of the refrigerator model is indicated.

Table 8 compares the modeled and experimental daily energy consumption results for the refrigerators. Energy consumption values in terms of kilowatt hours per day (kWh/d) are given for the modeling simulation and the laboratory tests. Heat exchanger dimensions, tubing sizes, and fin densities needed for model inputs were provided by the manufacturers for refrigerators "B", "C", "D", and "E". Actual compressor performance data and test temperatures were also used. Energy consumption predicted by the model agrees to within $\pm 9\%$ with that measured for the laboratory procedure for all but one case. An even distribution of higher and lower results were obtained from the model despite the consistently higher cabinet heat loss rates predicted by the model in Table 7. The large discrepancy between modeled and experimental results for refrigerator "B" indicates a problem such as an incorrect charge on this unit.

It is interesting that the best agreement between experimental and modeled results was obtained for refrigerator "A", the unit on which no manufacturer supplied modeling input data was received. In order to do any modeling work on this unit, the necessary cabinet dimensions, heat exchanger sizes and configuration, etc. measurements were made in the laboratory so the information could be used as inputs for the model. Some iterative adjustments of the more uncertain pieces of input data against observed operating conditions were also used to enhance the validity of the simulation. The questionnaire used to solicit modeling inputs from the manufacturers was constructed for a DOE class 3 product, so many of the questions and presumably the manufacturer's responses were poorly suited to this single-door, manual defrost design. This additional effort to precisely measure the parameters needed to simulate class 1 refrigerators and the further refinement of uncertain or judgmental input parameters against experimental data may improve the model's performance from barely adequate to acceptable.

Our observations indicate that the generic model used for this study may work well as a first-cut design and simulation model for Indian manufacturers who take the time to refine all the modeling inputs to best fit their product. These arguments also justify using the model to analyze and simulate the effects of energy saving design options for these appliances in the next section.

ANALYSIS OF ENERGY SAVING DESIGN OPTIONS

One conservative way of estimating *cost effective*, energy saving changes that could be built into the design of these refrigerators would be to combine all of the best features of each individual unit into a single "best-of-the-best" design. Obviously, this approach involves utilization of hardware that is currently available and in commercial production. Also, at least one manufacturer has determined that the additional expense of improved hardware is worth the value it adds to his product on the open market. Refrigerator "A" was chosen as the starting <u>baseline</u> for this analysis because it had the lowest initial energy consumption, the best match between modeled and laboratory results, the second lowest cabinet heat loading results, and because we had the highest degree of confidence in the modeling input data. Cabinet temperatures and ambients which correspond exactly to the IS procedure [IS, 1987] were used for this modeling. This explains why there is a difference between the <u>0.958 kWh/d</u> value listed in Table 8 for this unit and the <u>0.942 kWh/d</u> value used as a starting point for this analysis in Table 9.

A summary of this best available technology approach is presented in Table 9 and Figure 3. Case A establishes the ***baseline*** starting point. Case B shows the change in daily energy consumption and compressor run times if the compressor from refrigerator "C" (COP = 1.0; capacity = 405 Btu/h) is substituted for the one originally in "A" (COP = 0.76; capacity = 277 Btu/h). This change decreased the modeled baseline energy consumption of refrigerator "A" by 15% despite the fact that the compressor capacity becomes even more oversized for the application. When a 110 VAC, 60 Hz compressor with a COP of 1.0 and a 315 Btu/h capacity was experimentally substituted in "A" to get an indication of the actual potential for this change, only an 8.3% improvement in energy use was measured. However, it should be noted that the capacity of this 110 VAC compressor is less than that of compressor "C".

Current "state of the art" compressors in this range of sizes have COPs in the range of 1.0—1.2 [EPA, 1993a], and technology exists to improve the COP to 1.4.

A cost effective way to improve the efficiency of refrigerator compressors may be to change from resistance start/induction run (RSIR) to permanent split capacitor (PSC) compressor motors. The additional cost to the manufacturers has been estimated at $2.50 to $4.00 per compressor or 15—25% [EPA, 1993b]. The motor's power factor improves from 0.6—0.65 to 0.8—0.85 which translates into about a 10—15% improvement in motor efficiency. The low voltage starting capability of this motor would have to be considered in light of the variable supply voltage in India.

There are methods and design techniques to improve the mechanical efficiency of the compressor also, but these are more capital intensive than a motor change. Also, the results are more subtle and marginal than those obtained with a motor change.

Case C models the results obtained when the condenser on "A" with 4.6 ft^2 (0.43 m^3) air-side area is replaced with the 6.8 ft^2 (0.63 m^2) static heat exchanger from unit "C". No net improvement in performance was noted for this change, possibly because the condenser tubing size also changed from 3/16 in. (4.8 mm) diameter originally on "A" to 1/4 in (6.4

mm) diameter tubing on the larger condenser. The larger diameter tubing decreases the velocity of refrigerant flow and the model correspondingly calculates a lower, refrigerant-side, heat transfer coefficient because of this velocity difference. Additionally, the pressure drop across the condenser changed from 0.9 psi (6.23 kPa) for the 4.6 ft^3 condenser to 0.13 psi (0.93 kPa) for the larger 6.8 ft^3 heat exchanger.

When a larger (7.3 ft^2; 0.68 m^3), narrower tubed (5/16 in.; 7.9 mm), evaporator similar to the on found on unit "B" was substituted for the 3/8 in. (9.5 mm) tube component on refrigerator "A", the results (Case D in Table 9) showed an improved refrigerant-side heat transfer coefficient due to higher refrigerant velocities, and the daily energy consumption dropped to 0.776 kWh/d. Again, refrigerant pressure drop across the heat exchanger indicated this increase in velocity showing a pressure difference of 0.19 psi (1.32 kPa) for the 3/8 in. tubing and 0.48 psi (3.31 kPa) for the narrower 5/16 in. tubing.

Product and material costs should be slightly lower with smaller diameter tubing for heat exchangers, but substantial tooling expenditures and capital equipment costs would be necessary.

Widening the thermostat's dead-band to decrease compressor cycling frequency as discussed in a previous section should be a low- or zero-cost option. Potential problems with a larger dead-band would involve difficulties in getting the compressor to start in response to thermal loads added to the refrigerated space. Baffling the convective air flow from the evaporator area to the general storage area of these units would also beneficially effect cycling rates without substantially increasing the cost.

Two other logical design changes were modeled for these refrigerators. The effects of two incremental increases in refrigerator door insulation and two, step-wise improvements in the insulating ability of door gaskets were evaluated with the model. Results from these calculations are shown in matrix form in Table 10. Increased door insulation would be one of the least intrusive ways of decreasing cabinet heat loss since this component is not a part of the major cabinet tooling and fixturing.

Methods for reducing door gasketing heat losses are being thoroughly investigated as a method for helping U.S. manufacturers meet ever tightening NAECA Standards [Flynn et. al., 1992]. While a 20% reduction in gasket loss coefficient may be cost prohibitive, improved refrigerator "nose" designs and alternative door sealing mechanisms have certainly been able to achieve an energy savings of 10% or more. The results in Table 10 show how effective these two strategies would be if employed on these appliances.

Since gaskets provide the door sealing and latching functions for these refrigerators, there are limits on improvements that may be achieved. Some changes in positioning the door gasket could enhance its thermal performance, but a wider gasket with additional balloon sections could improve the thermal performance by reducing conductive heat flow into the cabinet. Whatever changes are incorporated, the gasket must still perform its primary function of sealing and latching the door. Costs to incorporate an improved gasket should be quite modest. Careful testing of prototypes would be needed to insure that modeled gains are realized.

Any increase in insulation thickness would result in a reduction of conductive heat flow into the cabinet and reduced energy use by the unit. Generally, increasing the thickness of the door is much simpler than similar changes in the cabinet. Costs are increased by increases in the door depth. In addition, more foam insulation is required to achieve the increased thickness. For refrigerators of the sizes that are popular in India, it is estimated that $1.00 to $2.00 additional material cost would be required in addition to the capital equipment changes and associated costs this change would incur.

The energy consumption values listed in Tables 6, 9 and 10 certainly suggest that decreasing the maximum, no-load energy consumption value in the current Indian Standard from 2.0 kWh/d to 1.0 kWh/d should not tax the capabilities of current manufacturers.

CONCLUSIONS

Opportunities exist to save energy on Indian refrigerator/refrigerator-freezer designs. The laboratory results presented in this work indicate that either the refrigerators should be changed to meet the operating conditions specified in the standard, or the energy consumption standard should be changed to more accurately reflect the way the appliance is expected to perform in field applications. In three of four cases, the rated energy consumption of these single-door refrigerators was inflated because the freezer compartment was not adequately segregated from the rest of the refrigerated space.

An extensive field monitoring study similar to the one performed in the U.S. by Meier and Jansky would show if the current Indian Standard is an accurate predictor of field energy use and if conditions called for in the test are representative of field operating conditions [Meier and Jansky, 1993].

Excessive compressor cycling was observed for two of the single-door Indian refrigerators. This frequent cycling contributes to energy losses resulting from larger initial compressor power draw and the need to re-establish stable running conditions at the start of a compressor cycle (cycling losses). A better balance between refrigeration circuit capacity and cabinet heat loading would help avoid these losses.

The poorest energy consumption results for a class 1 refrigerator in this study was 1.24 kWh/d as compared to the current, voluntary, maximum standard in India of 2.0 kWh/d (730 kWh/y) [Indian Standard, 1987]. Based on the results presented in this study, the standard for refrigerator energy consumption could be reduced by at least 25% and possibly 50% without adversely impacting manufacturers' ability to produce these appliances.

The computer model predicted that combining the "best" features from each of the single-door units into a single refrigerator could result in an 18—20% reduction in energy consumption for the best rated unit, "A". The predicted "best-of-the-best" unit had a modeled energy consumption of 0.775 kWh/d, or roughly ⅓ of the current voluntary standard.

Additional, straight-forward changes such as increasing the thickness of door insulation, increasing the thermostat dead-band, and improving door gaskets can, cost-effectively boost energy savings by an additional 3—5% for this appliance.

ACKNOWLEDGEMENT

This work was performed under a grant provided by the Assisting Deployment of Energy Practices and Technologies (ADEPT) Program which is jointly funded by the U.S. Department of Energy and the Asian Environmental Partnership Program under the U.S. Agency for International Development. Oak Ridge National Laboratory is operated by the U.S. Department of Energy (DOE) under contract number DE-AC05-84OR21400 with Martin Marietta Energy Systems.

The authors would like to express the appreciation to Tom Wilbanks and Sujit Das of Oak Ridge National Laboratory's Developing Countries and Technology Economics groups who planned and managed this program. Appreciation must also be extended to Randy Linkous who personally supervised the instrumentation, set-up, test scheduling, data gathering, and data reduction involved in this work.

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Table 2.
Compressor Calorimeter Results
- Rating Performance with CFC-12 @ -10°F Evaporating/130°F Condensing Point -

	Experimentally Measured Performance		Manufacturer Reported Performance	
	COP	Capacity (Btu/h)	COP	Capacity (Btu/h)
Refrigerator "A"	0.76	277	*	*
Refrigerator "B"	0.89	296	0.88-1.02	355
Refrigerator "C"	1.00	405	1.00	375
Refrigerator "D"	0.83	302	0.78	299
Refrigerator "E"	0.88	424	1.01	446

* No data provided

Table 3.
Averaged Experimental versus Empirical Cabinet Heat Loss Results

	Average $\Delta T_{Freezer}$ (°F)	Average $\Delta T_{Fresh\ Food}$ (°F)	Heat $Loss_{Expt.}$ (Btu/h)	Heat $Loss_{Calc.}$ (Btu/h)	% Deviation	Empirical Equation
Refrigerator "A"	73.6	38.9	106.1	106.2	0.1	$Q_A{}^{1.} = 1.094\Delta T_{Frz.} + 0.661\Delta T_{F.F}$
Refrigerator "B"	69.2	46.4	107.1	107.2	0.1	$Q_B{}^{2.} = 0.697\Delta T_{Frz.} + 1.269\Delta T_{F.F}$
Refrigerator "C"	68.8	49.7	99.1	99.2	0.1	$Q_C{}^{2.} = 0.254\Delta T_{Frz.} + 1.643\Delta T_{F.F}$
Refrigerator "D"	64.4	48.4	103.1	102.6	0.5	$Q_D{}^{2.} = 0.534\Delta T_{Frz.} + 1.410\Delta T_{F.F}$
Refrigerator "E"	74.2	41.9	96.3	96.2	0.1	$Q_E{}^{1.} = 0.583\Delta T_{Frz.} + 1.264\Delta T_{F.F}$

1. Average of four experiments
2. Average of five experiments

Table 4.
109.4°F (43°C), No-Load, Pull-Down Tests
– 230 VAC, 50 HZ –

	Time to Reach 44.6°F (7°C) Fresh Food Average (min.)	Power Consumption (kWh)	Freezer Temperature (°F)
Refrigerator "A"	277	0.571	9.0 (-12.8°C)
Refrigerator "B"	146	0.218	20.4 (-6.4°C)
Refrigerator "C"	142	0.276	14.3 (-9.8°C)
Refrigerator "D"	180	0.377	17.6 (-8.0°C)
Refrigerator "E"	152	0.407	4.9 (-15.1°C)

Table 5.

109.4°F (43°C) No-Load Pull Down Tests
-- 230 VAC, 50 Hz, Final Steady State Conditions --

			Steady State	
	Time to 44.6°F (7°C) Fresh Food Ave. (Min.)	Watts	Freezer Temperature °F (°C)	Fresh Food Temperature °F (°C)
Refrigerator "A"	277	116.0	7.4 (-13.7)	41.9 (5.5)
Refrigerator "B"	146	104.5	12.4 (-10.9)	30.8 (-0.7)
Refrigerator "C"	142	130.3	4.0 (-15.6)	23.4 (-4.8)
Refrigerator "D"	180	113.7	15.0 (-9.5)	37.9 (3.3)
Refrigerator "E"	152	140.0	0.9 (-17.3)	34.5 (1.4)

Table 6.
Energy Consumption Test Results
India Refrigerator-Freezers

	Temperature Fresh Food (°F)	Temperature Freezer (°F)	Energy Use kWh/d	% Run	Cycles/Day	Minutes/ Compressor Cycle
Refrigerator "A"						
230 VAC	40.8	15.8	0.929	28.8	86.7	16.61
190 VAC	40.8	15.6	0.875	29.5	83.1	17.33
260 VAC	40.9	15.5	1.069	29.7	84.0	17.14
Refrigerator "B"						
230 VAC	35.9	23.0*	1.238	45.0	118.7	12.13
190 VAC	35.8	23.0*	1.147	45.6	116.1	12.40
260 VAC	36.0	23.0*	1.386	44.4	122.3	11.77
Refrigerator "C"						
230 VAC	36.9	23.9	0.968	29.9	187.5	7.68
190 VAC	39.8	24.0	0.874	28.9	177.1	8.13
260 VAC	40.0	24.2	1.031	28.2	185.9	7.75
Refrigerator "D"						
230 VAC	40.1	23.42	0.963	32.7	81.0	17.78
190 VAC	39.9	23.4	0.837	31.4	81.3	17.71
260 VAC	40.2	23.7	1.034	30.8	82.5	17.45
Refrigerator "E"						
230 VAC						
D-D**	41.44	2.38	1.788	52.0	44.0	32.73
S-S***	41.36	2.23	1.682	51.9	42.3	34.04
190 VAC						
D-D**	42.90	3.55	1.525	53.4	25.1	57.31
S-S***	42.81	3.30	1.477	52.5	25.8	55.76
260 VAC						
D-D**	38.91	3.74	2.420	60.9	52.3	27.51
S-S***	38.76	3.53	2.354	59.8	53.3	27.0

* Interpolated from bracketing thermostat settings
** Defrost-to-Defrost
*** Steady State (6+ hours)

Table 7.
Cabinet Heat Loss Summary

--- Modeled versus Measured Test Results ---
(Experimental Energy Rating Temperatures Used in Model)

Unit	Cabinet Heat Loss Rates Modeled Btu/h	Measured Btu/h	Difference	%	Comment
"A"	109.7	113.6	-4.0	-3.6	[Model lower]
"B"	117.2	115.3	1.8	1.6	[Model higher]
"C"	134.0	104.0	29.9	22.3	[Model higher]
"D"	126.9	105.9	21.0	16.5	[Model higher]
"E"	138.8	112.4	26.4	23.5	[Model higher]

Table 8.
Daily Energy Use Summary, Indian Refrigerator

--- Modeled versus Measured Results ---
(Experimental Compartment Temperatures and Compressor Rating Used in Model)

Unit	Modeled kWh/day	Measured kWh/day	Difference (Meas.- Mod.)	%	Comment
"A"	0.958	0.929	-0.029	-3.1	[Model higher]
"B"	0.927	1.238	0.311	25.1	[Model lower]
"C"	1.050	0.968	-0.082	-8.5	[Model higher]
"D"	1.041	0.963	-0.078	-8.1	[Model higher]
"E"	1.628	1.788	0.160	8.9	[Model lower]

Table 9.
"Best-of-the-Best" Available Technology Analysis
for Indian Refrigerators

Case	Description	Modeled Energy Use (kWh/d)	Modeled Run-Times %
A	"Baseline" refrigerator "A." @ 32°C (89.6°F) no-load, closed-door Testing conditions	0.942	36.7
B	Case A + COP = 1.0 Compressor from refrigerator "C"	0.803	33.0
C	Case B + Larger condenser from unit "C"	0.803	33.0
D	Case C + Larger, smaller tube size evaporator from unit "B"	0.776	29.6

Table 10.
Effects of Additional Door Insulation and/or
Door Gasket Insulation on Indian Refrigerator Energy Consumption

Improved Door Gaskets	**Additional Door Insulation**		
	0%	+25% (1.05 cm increased thickness)*	+ 50% (2.10 cm increased thickness)*
0%	0.776 kWh/d	0.775 k/Wh/d	0.741 kWh/d
10% (8.0 - 7.2 watt/m 100°C)*	0.759 k/Wh/d	0.739 k/Wh/d	--------
20% (8.0 - 6.4 watt/m 100°C)*	0.742 kWh/d	--------	0.708 kWh/d

* SI units required by model

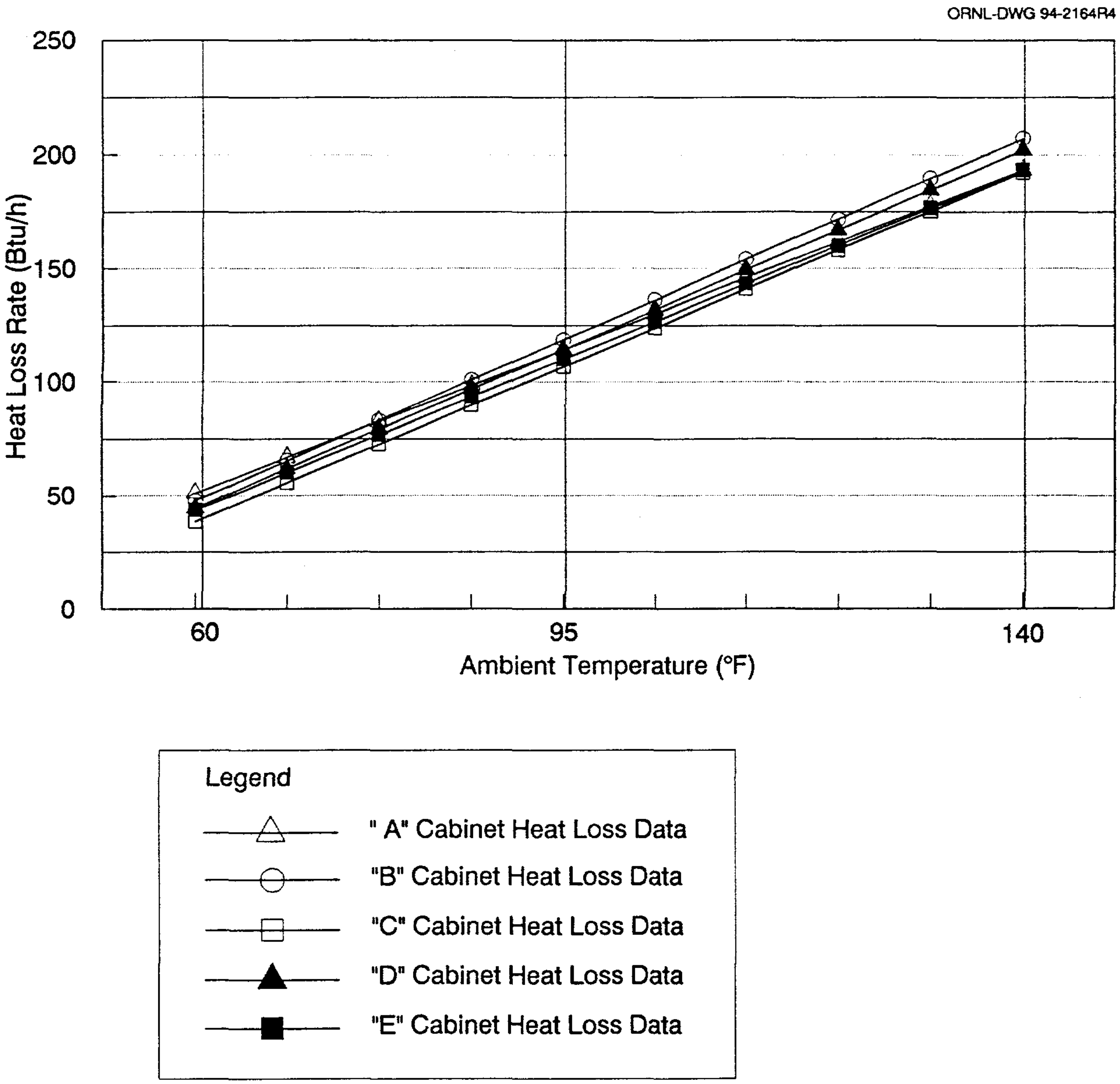

Fig. 1. Reverse heat loss results

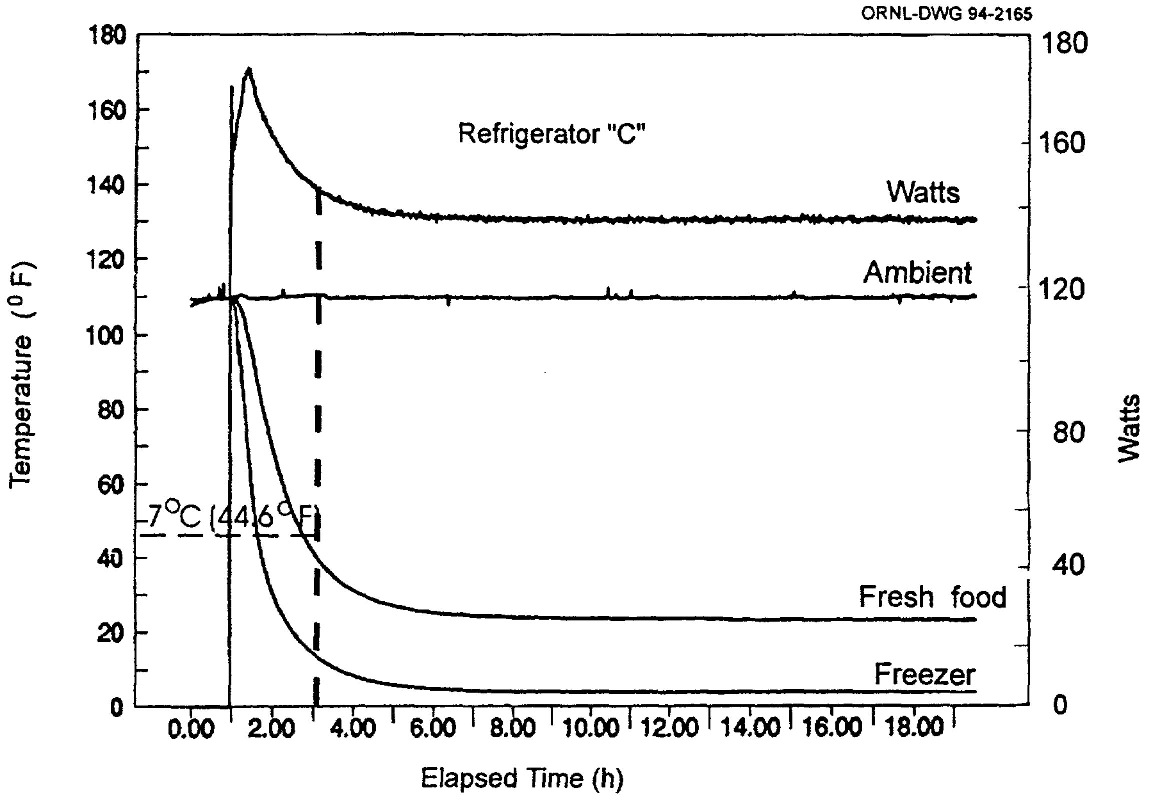

Fig. 2. Pulldown test results at 109.4 °F (43 °C)

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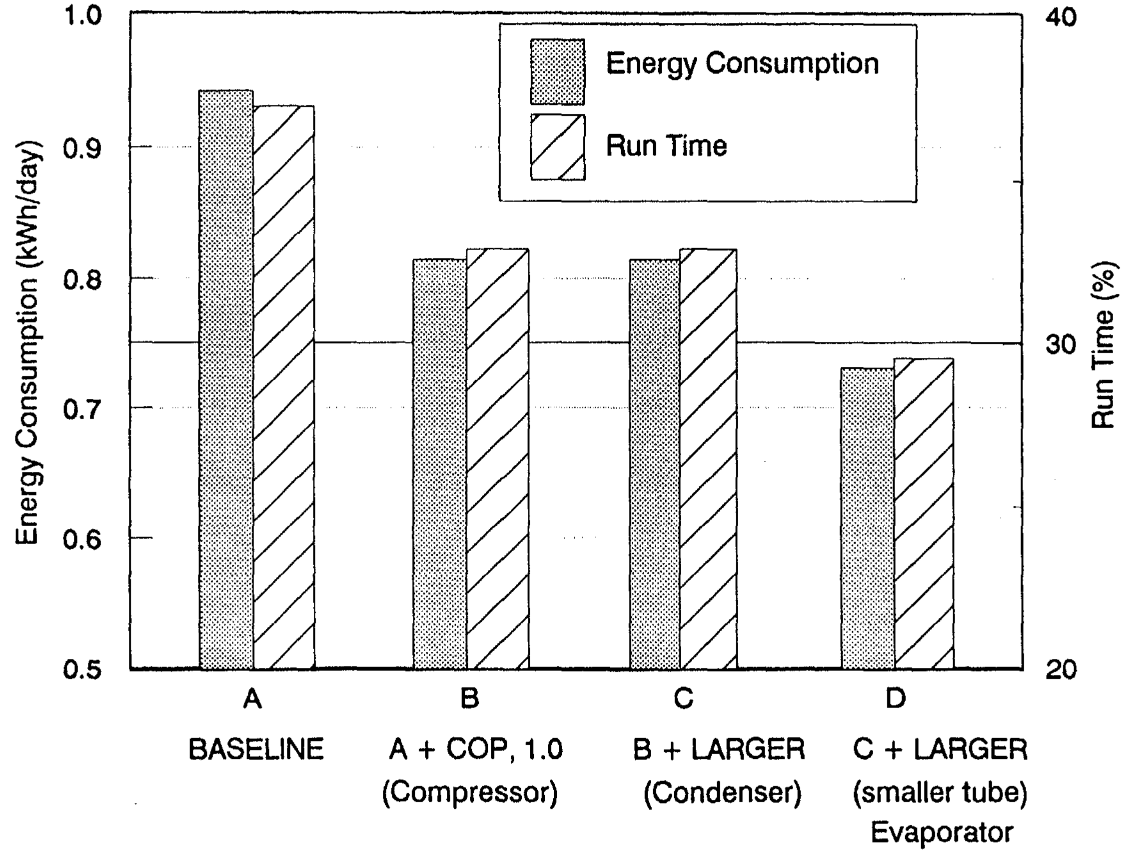

Fig. 3. Improved efficiency modeled from application of best components of each single-door unit tested.