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Energy Conservation and Economic Analysis of Residential Refrigerators

Robert A. Hoskins
University of Tennessee - Knoxville

To the Graduate Council:

I am submitting herewith a thesis written by Robert A. Hoskins entitled "Energy Conservation and Economic Analysis of Residential Refrigerators." I have examined the final electronic copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Mechanical Engineering.

William S. Johnson, Major Professor

We have read this thesis and recommend its acceptance:

A. F. Bailey, James A. Euler, Peter W. Cowling

Accepted for the Council:
Carolyn R. Hodges

Vice Provost and Dean of the Graduate School

(Original signatures are on file with official student records.)

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Vice Chancellor
Graduate Studies and Research

ENERGY CONSERVATION AND ECONOMIC ANALYSIS
OF RESIDENTIAL REFRIGERATORS

A Thesis

Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

Robert A. Hoskins

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ABSTRACT

A detailed computer model is developed to calculate energy flows and electricity use for residential refrigerators. Model equations are derived from applications of the first law of thermodynamics, analysis of manufacturers' literature, and related studies. The model is used to evaluate the energy (and associated initial cost) impacts of alternative designs to reduce refrigerator energy use.

Model results show that 56% of the total heat gain in a typical 0.45 m^3 (16 ft^3) top-freezer refrigerator is due to conduction through cabinet walls and doors. The remaining 44% is from door openings, heaters, fans, food, gasket area infiltration, and miscellaneous heat sources. Operation of the compressor to remove this heat and maintain the refrigerated spaces at constant temperatures accounts for 70% of the unit's electricity use. The remainder is for operation of heaters and fans.

Several energy-saving design changes are examined using the energy model. These changes are: increased insulation thickness, improved insulation conductivity, removal of fan from cooled area, use of anti-sweat heater switch, improved compressor efficiency, increased condenser and evaporator surface areas, and elimination of the frost-free feature. Application of all these changes would reduce refrigerator electricity use 71% and increase initial cost 5%. Implementing all these changes except for elimination of the frost-free feature would reduce electricity use 52% and increase initial cost 19%. These results show that there are large opportunities for reducing refrigerator electricity use with only slight initial cost increases.

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A = surface area of cabinet wall or door

$a, b, c, d, e, g, h,$
$c_1, c_2, c_3, c_4, \alpha, \beta$ = constants

COP = coefficient of performance

C_p = specific heat

D = depth of interior cabinet

E = electricity consumption per day

f = fraction of run time

H = height of interior cabinet

k = thermal conductivity

L = gasket length

N = number of door openings per day

n = number of defrost cycles per day

Q = thermal gain per day

p = rated power

T = temperature

t = run-time

V = volume

W = width of interior cabinet

$\dot{w}$ = mass per day

x = thickness of insulation

ρ = density

subscripts:

amb = ambient

comp = compressor

cond = condenser

def = defrost

drn = drain

evap = evaporator

f = freezer

htr = heater

misc = miscellaneous

r = fresh-food

tot = total

1. INTRODUCTION

The purposes of this report are to: (1) develop a computer model of energy flows and electricity uses in residential refrigerators, and (2) use this model to evaluate the energy and cost (both purchasing and operating) impacts of alternative energy-conserving designs. Outputs from these analyses are used as inputs to a detailed engineering-economic model of residential energy use developed at ORNL.[1] The energy use simulation model estimates the distribution of new residential equipment each year (from 1970 through 2000) as functions of fuel prices, consumer demand functions, and technological characteristics of each type of equipment. The present study provides the relationship between operating energy requirement and initial cost (technological characteristics) for refrigerators needed by the simulation model.

Table 1 shows the 1970 distribution of residential energy use* by fuel and end use.[1] Refrigerators account for 6% of total household fuel use and for 16% of residential electricity use. Thus, improvements in new refrigerators can have significant long-term energy conservation impacts.

Section 2 discusses historical trends in refrigerator sales, ownership, size, type, lifetime, and energy use. These data are used to define three typical refrigerators used with the energy model to evaluate energy use with various design changes.

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*Table 1 shows electricity in terms of its primary energy equivalent (29.7% efficiency). All subsequent electricity use figures are in terms of end-use energy.

Table 1. Household fuel use by fuel and end-use, 1970.

	Electricity[a]	Gas	Oil	Other[b]	Total
	10^{18} (Joules)				
Space heating	0.84	3.92	3.39	0.82	8.97 (56)[c]
Water heating	0.88	0.98	0.28	0.07	2.21 (14)
Refrigeration	0.91				0.91 (6)
Freezing	0.31				0.31 (2)
Cooking	0.39	0.32		0.03	0.74 (5)
Air conditioning	0.70				0.70 (4)
Other	1.62	0.45			2.07 (13)
Total	5.65	5.67	3.67	0.92	15.91
	(35)[c]	(36)	(23)	(6)	

[a]Electricity use figures are in terms of primary energy; that is, they include losses in generation, transmission, and distribution.

[b]Other fuels include coal and liquefied natural gases.

[c]Numbers in parentheses are percentages of the grand total, 15.9×10^{18} J.

Source: Ref. 1.

Section 3 develops the energy model. The first part of the model deals with thermal loads on the refrigerator. Heat gains to refrigerated spaces are due to conduction through walls, door openings, infiltration through gasket area, food, operation of heaters and fans, and operation of an ice maker in some units. The second portion of the energy model calculates electricity consumption. Most of the electricity is used to operate the compressor. Electricity is also used to power heaters and fans. Outputs from the refrigerator model include all thermal loads, all electricity uses, coefficient of performance (COP), and compressor run-time. Section 4 shows comparisons between model predictions and actual refrigerator energy use.

The model is used in Section 5 to evaluate the energy (and initial cost) impacts of several design changes to reduce electricity use. Initial cost impacts are obtained from discussions with manufacturers and reviews of related studies.

Section 6 summarizes the major features of the computer model and presents conclusions on alternative refrigerator designs, their cost-effectiveness and feasibility.

2. MARKET TRENDS

This section reviews data from the past several years on refrigerator ownership and purchases, initial price, types and sizes of units, lifetimes, and energy use. These data are synthesized to define three typical refrigerators that are analyzed later: 0.34 m^3 (12 ft^3) top-freezer refrigerator, 0.45 m^3 (16 ft^3) top-freezer model, 0.57 m^3 (20 ft^3) side-by-side unit.

Virtually 100% of American households own a refrigerator.[2] Saturation may actually exceed 100% because many families own two refrigerators (e.g., for use in a basement or in a second home). If incomes continue to rise, refrigerator ownership will probably continue to increase faster than household growth.

Figure 1 shows the number of refrigerators sold each year during the past 15 years and the percentage that were purchased to replace an existing unit. Because refrigerator saturation is already 100%, the replacement market dominates refrigerator sales. Sales during the past 15 years were generally rising (except for 1974 and 1975), with sales averaging 5 million/year during the past few years.[2]

Prices of new refrigerators declined steadily during recent years when measured in terms of "real" dollars.* In 1960, the average refrigerator cost $447 while in 1974, the price had dropped to $243 (1967-$), see Fig. 2.

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*Real or "constant" dollars correct for the impacts of inflation. For example, the price of a refrigerator in 1967-$ (constant-$) is equal to the nominal price divided by the Consumer Price Index.

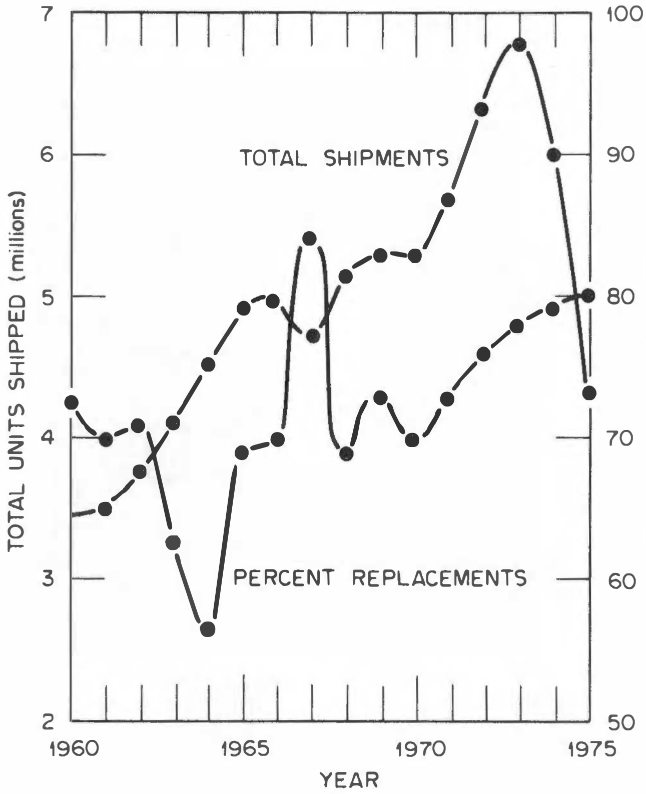

Figure 1. Refrigerator shipments and replacements, 1960-1975.

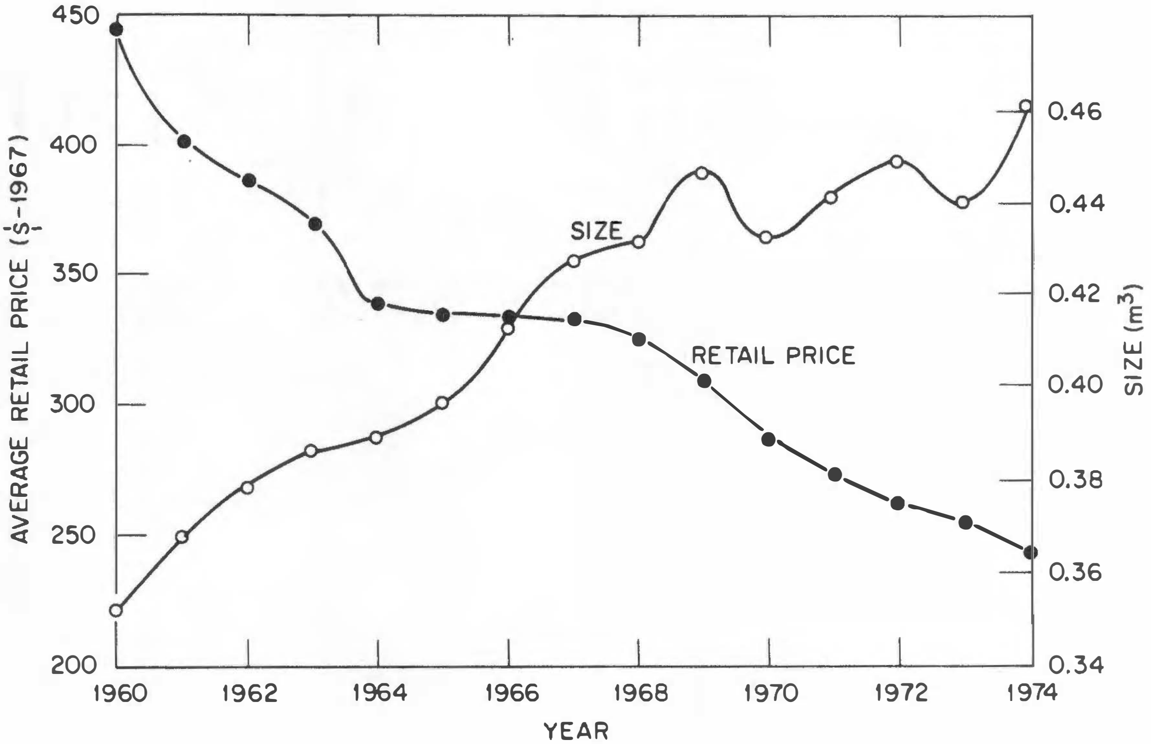

Figure 2. Average retail price and size for new refrigerators, 1960-1974.

Merchandising Week[2] lists manufacturers' "suggested retail prices" for refrigerators. However, the author found that these prices are much lower than those actually paid by consumers, based on an investigation of actual selling prices and results presented in *Consumer Reports*.[3] The author estimates that actual retail prices are almost 25% higher than those reported in ref. 2. A 1975 price of \$379 (1975-\$) for a typical 0.45 m^3 (16 ft^3) top-freezer unit is used as the basis for evaluating the design changes in Section 5.

Most of the life-cycle costs of owning and operating a refrigerator are due to electricity consumption: approximately 58% according to ref. 4. The relative importance of operating costs is due to both the long average lifetime of refrigerators (about 15 years[1,2,5]) and the high electricity consumption.

Refrigerators are classified as either single-door, top-freezer, bottom-freezer, or side-by-side. Bottom freezer models account for less than 1% of total sales.* Top-freezer models are the most popular because they take up less floor space per unit volume and cost less than side-by-side units. In 1975, about 80% of the refrigerators sold were top-freezers and 20% were side-side.[2]

The average size of new refrigerators increased steadily during the past several years (Fig. 2, p. 6). The average size of all refrigerators <u>in use</u> in 1970 was about 0.38 m^3 (13.5 ft^3). However, the average size

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*Bottom-freezers are not considered further in this study because they account for such a small fraction of the market and because their performance characteristics are nearly the same as those for top-freezer models.

of new units sold that year was 12% greater. The average size of new units in 1974 (0.46 m^3) was 30% greater than the average size in 1960.[2]

Refrigerators have one of three types of defrost systems—manual, partial, or automatic. Many refrigerators in use are of the manual defrost type; however, only small-sized refrigerators are now available with manual defrost. These refrigerators are cooled by gravity circulation of air, and defrosting is initiated manually by turning off the power or by setting a switch to forced-heat defrost. An excessive build-up of ice around the evaporator coil can increase energy use by as much as 25%, because the ice around the coil acts as insulation. Defrosting must be done two or three times a year for two-door refrigerators and more frequently for single-door units.

Most new refrigerators have automatic defrost. In 1965, 48% of new refrigerators had automatic defrost; this percentage rose steadily to 73% in 1975.[2] In automatic defrost units, a fan forces air over a single evaporator coil which is used to cool both the freezer and fresh-food compartments. A timer initiates the defrost cycle, and defrosting takes place through the use of electric heat or hot refrigerant gas.

Refrigerator energy use depends primarily on the design of the unit: size, defrost option, door configuration. Energy use is also influenced by operational characteristics such as room temperature, food load, and the number of door openings. Because of these design and operation factors, estimates of refrigerator electricity use vary considerably (Fig. 3).

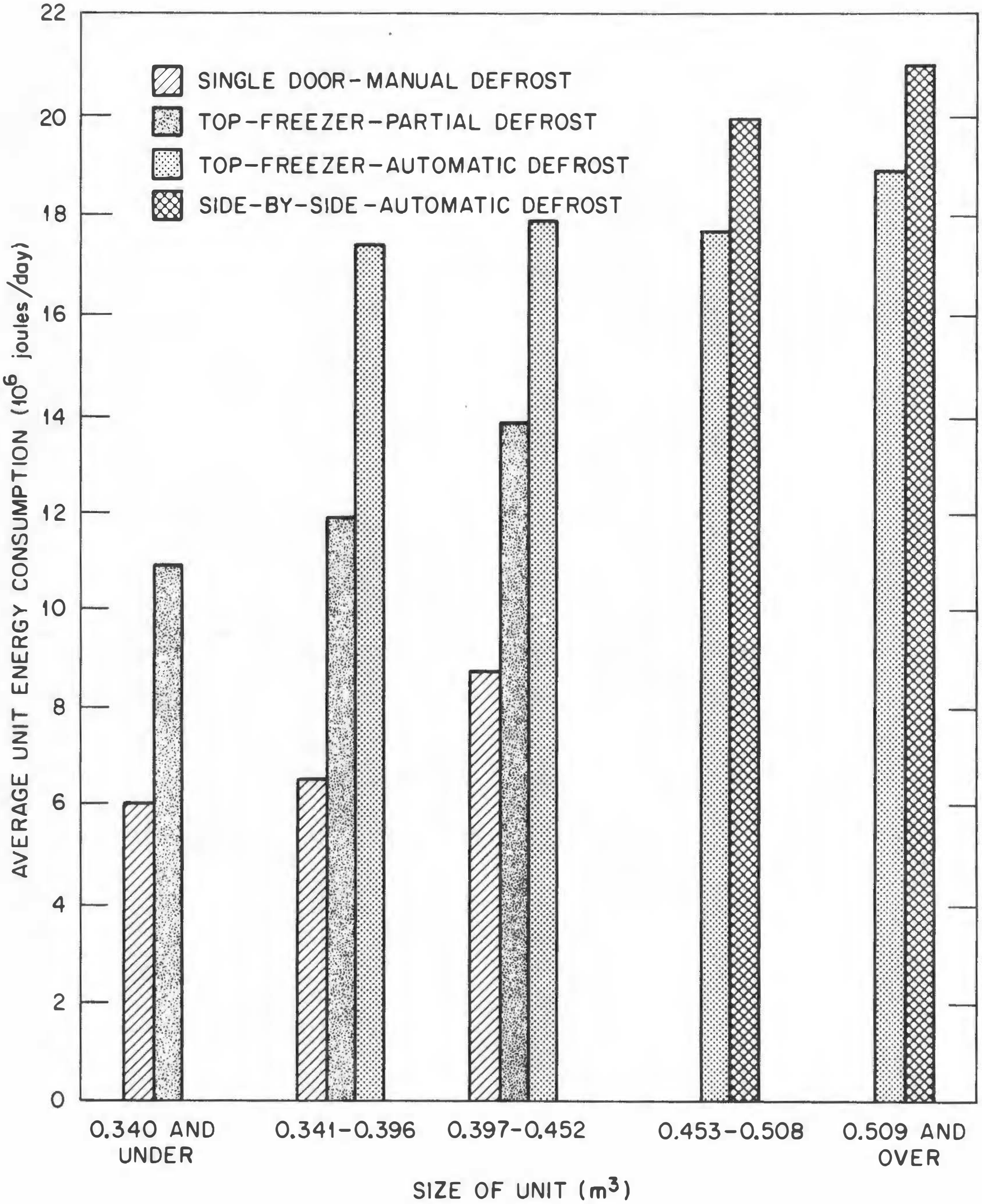

Figure 3. Refrigerator average unit energy consumption, 1975.

Tansil[6] estimated the average electricity use of a 0.40 m^3 (14 ft^3) frost-free refrigerator as 18×10^6 joules/day.* The Association of Home Appliance Manufacturers (AHAM)[7] reported electricity use estimates for refrigerators sold by 24 manufacturers (see Fig. 3, p. 9). These estimates suggest that a typical 0.45 m^3 (16 ft^3) top-freezer automatic defrost refrigerator will consume about 18×10^6 joules/day. Measurements reported in *Consumer Reports*[3] suggests an average consumption of 14×10^6 joules/day for a typical 0.45 m^3 unit.

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*Electricity is treated here at the point of end use, where 1 kwhr = 3.6×10^6 joules.

3. ENERGY MODEL

The energy model developed in this section is used in Section 4 to calculate the energy impacts of alternative refrigerator designs. The energy model performs three major functions. First, it evaluates heat gains to the refrigerator. Figure 4 is a schematic showing the major heat gains to a typical refrigerator. Second, the model determines electricity consumption based on the thermal load and operation of heaters and fans. Figure 5 is a schematic showing major electricity uses. Finally, the model is flexible enough so that energy conservation measures can be evaluated by changing values of various parameters in the model equations. A computer program was written to perform the

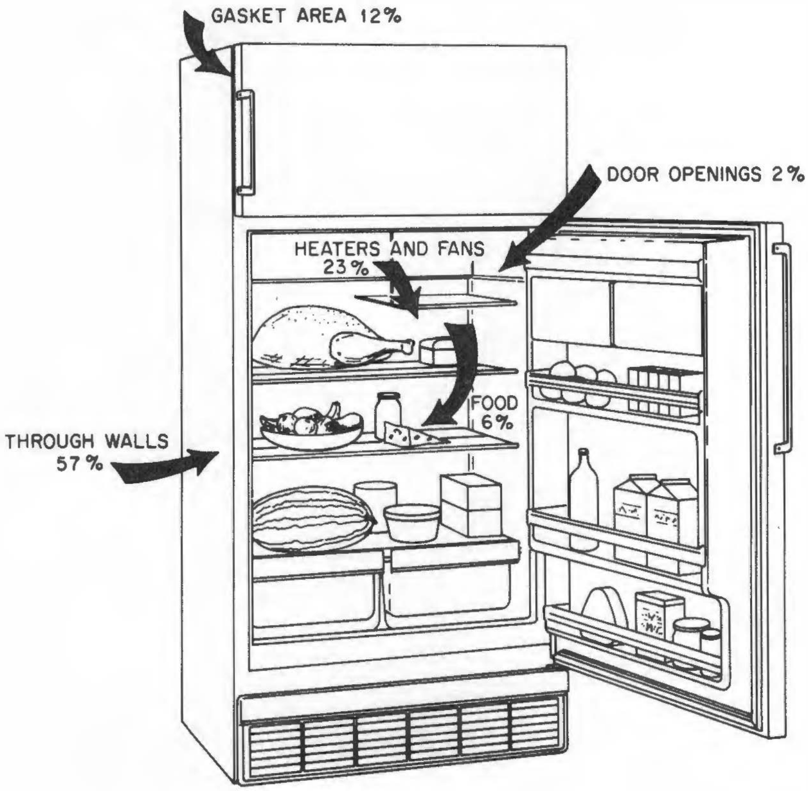

Figure 4. Refrigerator heat gains.

calculations required by the energy model for any size or type of refrigerator and for any given operating conditions (ambient temperature, storage temperatures, number of door openings, etc.).

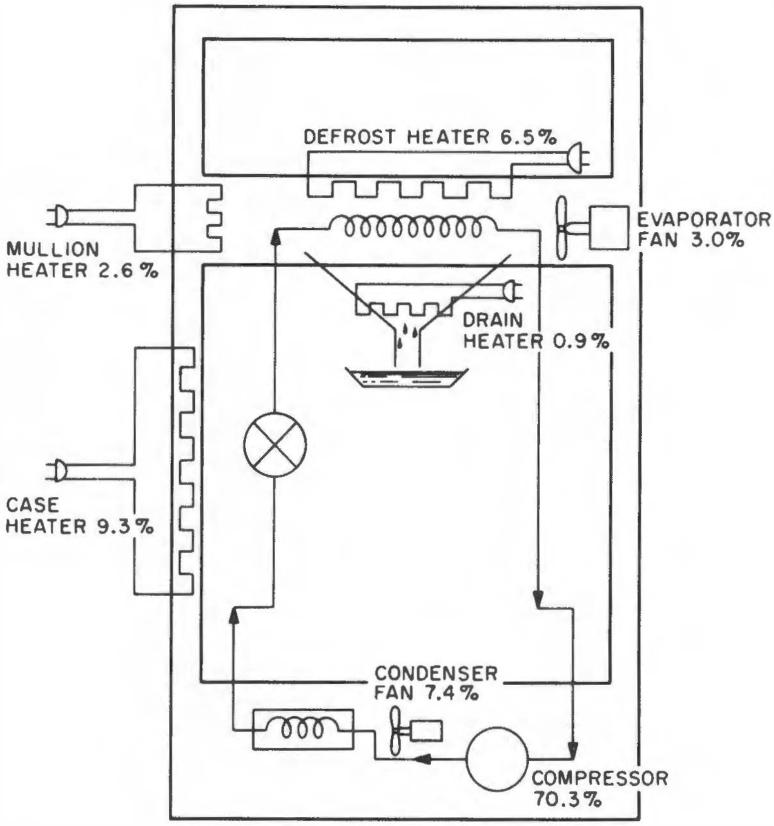

Figure 5. Refrigerator electricity consumption.

3.1 Thermal Load

Thermal load is the sum of individual heat gains to the refrigerator, which the refrigeration system must overcome to maintain constant temperatures in the fresh-food and freezer compartments. Calculation of thermal load enables one to determine compressor run-time and unit electricity use.

Two common test procedures are employed to determine actual thermal loads: the first uses a 32°C (90°F) ambient temperature with the doors always closed, and the second uses a 21°C (70°F) ambient temperature with the doors opened as in normal usage. The computer program developed here is capable of simulating either procedure. Because the second method better represents actual conditions, the open-door procedure is used in the following calculations. Table 2 lists the heat gains and contribution of each to total thermal load for three refrigerators; see

Table 2. Refrigerator thermal loads

	0.34 m³ (12 ft³) Top-freezer		0.45 m³ (16 ft³) Top-freezer		0.57 m³ (20 ft³) Side-by-side	
	10^6 J/day	%	10^6 J/day	%	10^6 J/day	%
Through walls[a]	4.85	54	6.01	56	6.51	49
Door openings[b]	0.15	2	0.20	2	0.24	2
Gasket area	1.17	13	1.31	12	2.19	16
Food[c]	0.38	4	0.60	5	1.02	8
Heaters and fans						
defrost heater[d]	1.08	12	1.08	10	1.08	8
drain heater[e]	0.13	1	0.13	1	0.13	1
mullion heater[f]	0.17	2	0.17	2	0.17	2
evaporator fan[g]	0.43	5	0.50	5	0.64	5
case heater[h]	0.62	7	0.62	6	0.62	5
Miscellaneous[i]	0.00	0	0.00	0	0.63	5
Total	8.98	100	10.62	100	13.23	100

[a]Fiberglass insulation (6.4 cm in sides, back and bottoms; 5.1 cm in top; 3.8 cm in doors).

[b]For 20 fresh-food and eight freezer door openings per day.

[c]Weight of food added per week proportional to total volume of refrigerator.

[d]500 watt, three cycles/day at 12 min/cycle.

[e]1.5 watt heater runs continuously.

[f]5.0 watt heater runs continuously.

[g]12.0 watt fan runs only when compressor is operating.

[h]18 watt heater runs continuously.

[i]Ice maker makes 0.9 kg (2.0 lb) ice/day.

also Fig. 4, p. 11. Parameter values used in these baseline calculations are given in Appendix D. The thermal load (Q, heat gain/day) is:

$$Q_{thermal} = Q_{\substack{through\\walls}} + Q_{\substack{door\\openings}} + Q_{\substack{gasket\ area\\(infiltration)}} + Q_{food} + Q_{\substack{heaters,\\fans}} + Q_{miscellaneous} .$$

Each of these heat gains is evaluated below.

Through cabinet: The largest heat gain in the refrigerator results from conduction of heat through the cabinet walls. This heat transfer is calculated as:

$$Q_{\substack{through\\walls}} = \sum_i \frac{kA_i \Delta T_i}{\Delta x}$$

Parameters are defined below.

(Thermal conductivity, k). Fiberglass and urethane foam are the insulation materials most commonly used in refrigerators. Thermal conductivities of these materials are 3.6 and 2.0 joule-cm/sec-m^2-^oC (0.25 and 0.14 Btu-in./hr-ft^2-^oF), respectively. The trend in refrigerator design in recent years has been toward increasing use of urethane foam.

(Surface area, A). To determine cabinet wall surface areas, a set of empirical equations is used to relate total unit volume to linear dimensions. Formulation of these equations is complicated because: (1) linear dimensions of refrigerators of equal volume vary significantly from manufacturer to manufacturer, and (2) the ratio of freezer volume to total volume increases as total volume increases.

A set of equations for dimensions was developed for each of the following units: side-by-side, top-freezer with volume ≥ 0.40 m^3 (14 ft^3), and top-freezer with volume < 0.40 m^3 (see Appendix C). These equations were derived from an analysis of linear dimensions for a large number of units and from relationships giving fresh-food and freezer volumes as functions of total volume:

$$V_f = c_1 V_{tot} e^{c_2 V_{tot}} \qquad\qquad V_r = V_{tot} - V_f$$

where

V_r, V_f, V_{tot} = fresh-food, freezer, and total volumes, respectively,

c_1, c_2 = constants.

Surface areas are then calculated from these linear dimensions.

(Temperature difference, ΔT). The difference between internal and external air temperature surrounding the cabinet walls depends on location of the surface area. Freezer temperatures are normally maintained between -18° and -13°C (0°-8°F) and fresh-food compartment temperatures range from 2° to 4°C (35°-40°F). Manual controls are located inside the unit to regulate these temperatures. The condenser, generally located on the back or bottom of top-freezer units and on the bottom of side-by-side units, contains hot refrigerant gas. The air temperature surrounding the coils is therefore higher than room temperature and is taken to be the same as the average condenser temperature. The compressor, located below the refrigerator, dissipates heat to the room causing warm air to flow across the bottom of the unit. A temperature of 60°C (140°F) for air surrounding the compressor was assumed for these calculations.

(Thickness, Δx). The thickness of insulation in refrigerators varies, with the most insulation normally located in the back or bottom, and the least in doors. Many manufacturers now use thinner walls with urethane foam (having a lower thermal conductivity than fiberglass). Typical insulation thicknesses for top-freezer and side-by-side models are shown in Table 3.

Table 3. Typical insulation thicknesses

	Thickness (cm)	
	Top-freezer	Side-by-side
Top	5.1 Fiberglass	4.1 Urethane foam
Bottom	6.4 "	5.7 "
Back	9.9 "	4.4 "
Sides	6.4 "	4.1 "
Door (freezer)	4.0 "	4.7 Fiberglass
Door (fresh-food)	3.4 "	4.1 "

Source: Ref. 8.

Door openings: The heat gain due to door openings is calculated from:

$$Q_{\substack{\text{door}\\ \text{opening}}} = (\rho \cdot C_p)_{air} (V_r \cdot N_r \cdot \Delta T_r + V_f \cdot N_f \cdot \Delta T_f)$$

where

Q = heat gain per day

ρ = density of air

V = volume of refrigerated space

C_p = specific heat (constant pressure) of air

N = number of door openings per day (fresh-food or freezer compartment)

ΔT = temperature difference between ambient and refrigerated air. This assumes that the average duration of a door opening (12 seconds) is just long enough to cause a 100% air change. An average of eight freezer door (N_f) and 20 refrigerator door openings (N_r) might be expected for a typical refrigerator.[8]

Gasket area (infiltration): Heat gain through the gasket area is calculated in various ways by different manufacturers. One method of determining this load is:

$$Q_{\substack{\text{gasket area}\\ \text{(infiltration)}}} = (L_r \cdot \Delta T_r + L_f \cdot \Delta T_f)\,(\alpha + \beta f_{\substack{\text{evap}\\ \text{fan}}})$$

where

α = 0.087 joule/sec-m-^{o}C (0.05 Btu/hr-ft-^{o}F) static--no fan

β = 0.062 joule/sec-m-^{o}C (0.036 Btu/hr-ft-^{o}F) dynamic--fan on[8]

$f_{\substack{\text{evap}\\ \text{fan}}}$ = fraction of run-time for evaporator fan (equal to compressor run-time).

The length of each gasket (L_r, L_f) is equal to the door perimeter.

Food: The heat gain due to food cool down is determined from the mass of food added to a typical refrigerator per week and a series of heat load calculations. The average food loads (in equivalent kilograms of water) for a 0.467 m^3 (16.5 ft^3) refrigerator are given in Table 4.[8] The heat gain in the fresh-food compartment is the sum of temperature reduction and food respiration loads. The heat gain in the freezer is the sum of heat gains from temperature reduction of the food to freezing temperature, latent heat of fusion, and temperature reduction to freezer temperature. Details of these heat gains are developed in Appendix B.

Table 4. Average food loads for a 0.467 m^3 ($16.5\ ft^3$) refrigerator (equivalent kg of water)

	Freezer	Fresh food
Light	0.5	1.4
Medium	9.1	13.6
Heavy	15.9	27.2

Source: Ref. 8.

Heaters and fans: Heaters and fans (evaporator fan, defrost heater, mullion heater, case heater, drain heater) located within the refrigerator are sources of heat gain. The condenser fan on forced-convection-cooled condenser units is located on the bottom exterior of the refrigerator and is, therefore, not a source of internal heat gain. The evaporator fan runs while the compressor is on and shuts off whenever the door is opened or the defrost heater is on. Thus:

$$Q_{\substack{evap\\fan}} = p_{\substack{evap\\fan}} \cdot f_{\substack{evap\\fan}}$$

where

p = rated power.

The defrost heater generally operates two or three times per day for 10-15 minutes per cycle:

$$Q_{\substack{def\\fan}} = p_{\substack{def\\fan}} \cdot t_{\substack{def\\fan}} \cdot n$$

where

$t_{\substack{def\\fan}}$ = duration of defrost cycle

n = number of defrost cycles per day.

In some models, a drain heater is located beneath the evaporator coil to prevent water from freezing in the drain. For drain heaters:

$$Q_{\substack{drain\\htr}} = P_{\substack{drain\\htr}} \cdot f_{\substack{drain\\htr}} .$$

In other models, the radiant heat from the defrost heater is great enough to prevent freezing of the water so there is no drain heater.

To prevent condensation from forming outside the cabinet, it is necessary to keep the exterior surfaces warmer than the dew point temperature. A mullion heater is located between the fresh-food and freezer compartments in the front of the unit to prevent condensation around the door area. Case heaters are located within the cabinet walls (on the exterior side of the insulation) to prevent sweating on the outside surface areas. These case and mullion heaters run 100% of the time unless connected to an anti-sweat heater switch provided with "dry/humid" settings that allow heater run times to be controlled by the owner. It is assumed that 40% of the heat generated by these heaters reaches the cooled area, with the remainder escaping to the outside.[5] Thus:

$$Q_{\substack{mullion\\htr}} = (0.40) \cdot P_{\substack{mullion\\fan}} \cdot f_{\substack{mullion\\fan}}$$

$$Q_{\substack{case\\htr}} = (0.40) \quad P_{\substack{case\\htr}} \cdot f_{\substack{case\\htr}}$$

The total heat gain due to heaters and fans is:

$$Q_{\substack{heaters,\\fans}} = Q_{\substack{evap\\fan}} + Q_{\substack{def\\htr}} + Q_{\substack{drain\\htr}} + Q_{\substack{mullion\\htr}} + Q_{\substack{case\\htr}}$$

Miscellaneous heat gains: Additional heat gains can occur from features such as automatic ice makers.* This heat gain is assumed proportional to the mass of ice produced. A typical ice maker consumes 6.3×10^5 joule/day (0.175 kwhr/day) to produce 0.9 kg (2 lb) of ice.

3.2 Electric Energy Consumption

Determination of refrigerator electricity use depends not only on the thermal load; but also on compressor efficiency, refrigeration capacity (amount of heat absorbed in evaporator when compressor runs 100% of time), condenser and evaporator characteristics, and accessories (heaters and fans). Figure 5, page 12, shows the major electricity uses in a typical refrigerator. Knowing thermal load and refrigeration capacity enables one to determine compressor run-time, coefficient of performance, and total electricity use.

Compressor and refrigeration capacity: Depending on the actual performance characteristics of the compressor, power requirements and refrigeration capacity can vary significantly between two different refrigerators with equal thermal loads. A series of curves giving compressor power and refrigerator capacity as functions of evaporator and condenser temperatures were provided by a leading compressor manufacturer from which the following empirical equations were obtained:

$$P_{comp} = a + b \cdot T_{cond} + (c + d \cdot T_{cond}) \cdot T_{evap}$$

$$Q_{evap} = e \cdot T_{evap} - g \cdot T_{cond} + h$$

*Heat gains from light bulbs and timers are negligible (timers are located outside the cabinet).

where

T_{cond} = average temperature of refrigerant gas in condenser

T_{evap} = average temperature of liquid refrigerant in evaporator

Q_{evap} = refrigeration capacity

a, b, c, d, e, g, h = constants

Refrigeration capacity represents the maximum heat rate of the evaporator, so this must be greater than the thermal load.

Compressor run-time: The fraction of time the compressor runs is found by dividing total thermal load by refrigeration capacity:

$$f_{comp} = \frac{Q_{thermal}}{Q_{evap}}$$

The design fraction is usually two-thirds to three-fourths (16-18 hrs/day) because compressors run more efficiently when they do not have to start and stop frequently; this also provides a safety factor.

Electrical loads: Once the compressor power has been established from evaporator and condenser temperatures, and the compressor run-time determined; electricity consumption of the compressor may be determined from:

$$E_{comp} = f_{comp} \cdot P_{comp}$$

In addition to the compressor (which consumes 70% of the refrigerator electricity), there are several electricity-consuming devices in the refrigerator. These include: evaporator fan, condenser fan, case heater, defrost heater, drain heater, and mullion heater. The case and mullion heaters generally operate 100% of the time, unless an

anti-sweat heater switch is provided and used. The other devices listed above operate less than 100% of the time, as discussed earlier. Total electricity consumption in the refrigerator is then:

$$E_{tot} = E_{comp} + E_{evap\ fan} + E_{cond\ fan} + E_{case\ htr} + E_{def\ htr} + E_{drain\ htr} + E_{mullion\ htr} + E_{ice\ maker} .$$

Coefficient of performance (COP): The COP provides a useful measure of the effectiveness of a refrigeration system. The COP is determined in the energy model from the ratio of refrigeration capacity to total electricity consumed (see Fig. 6):

$$COP = \frac{Q_{evap}}{E_{comp}}$$

The energy model calculates a COP of 0.91 for the reference refrigerator, and ASHRAE[9] states that a typical vapor compression refrigerator COP is 0.88.

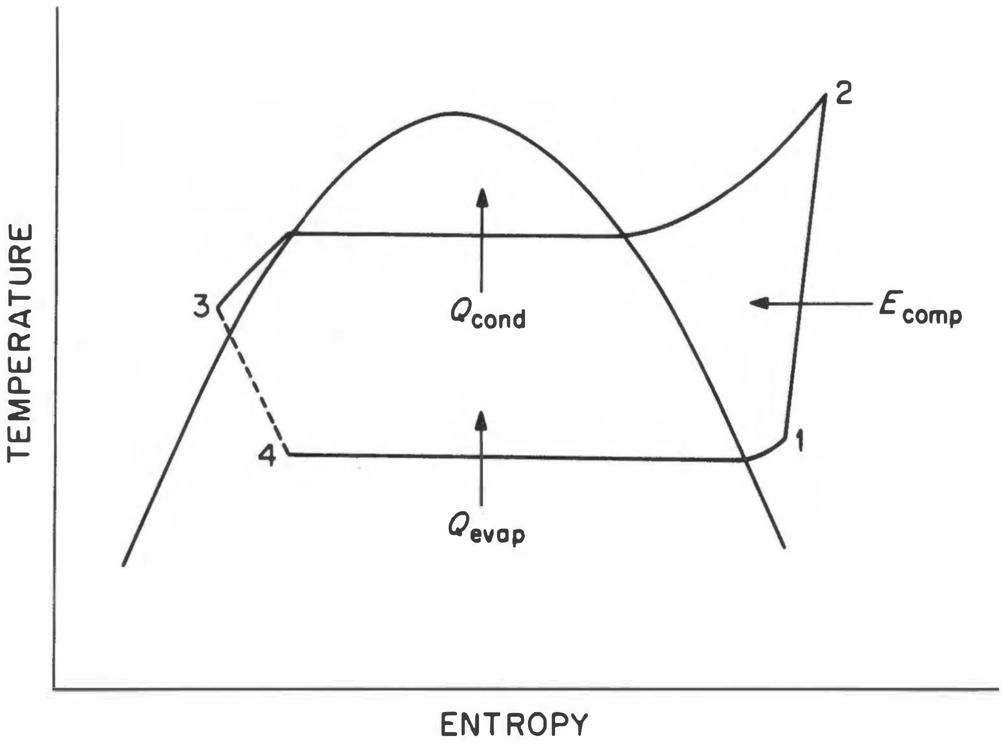

Figure 6. Temperature entropy diagram.

4. VALIDATION OF ENERGY MODEL

Predictions of refrigerator electricity use computed within the energy model agree well with data available from related studies, manufacturer experiments, the appliance industry trade association (AHAM), and Consumers Union. Figure 7 compares model estimates of electricity use per unit volume for <u>top-freezer</u> refrigerators with data from AHAM.[7,10] The energy model predicts a decline in energy use per unit volume as refrigerator capacity increases, in agreement with the AHAM data. The model results, both with and without the anti-sweat heater switch, fall roughly in the middle of the range of AHAM data.

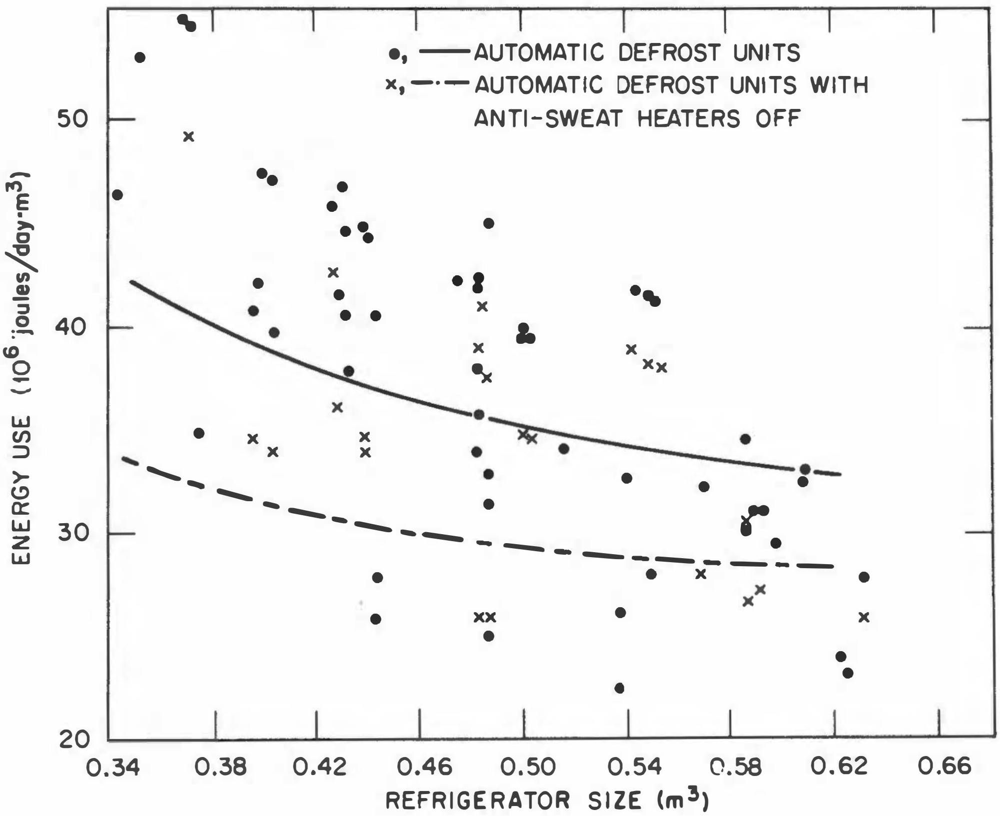

Figure 7. Comparison of energy model with AHAM data for top-freezer refrigerators.

Figure 8 shows that the energy model results are also in close agreement with AHAM[7,10] data for side-by-side units. Once again, the model predictions fall well within the range of values reported by AHAM for each size.

Figure 9 compares model estimates of energy use for top-freezer and side-by-side units with measurements reported in *Consumer Reports*[3,11,12,13] and Tansil's[6] estimate. The model predictions are higher than CR's and lower than Tansil's.

The author obtained detailed data from two manufacturers showing electricity use for the compressor, fans, and heaters for two different units.

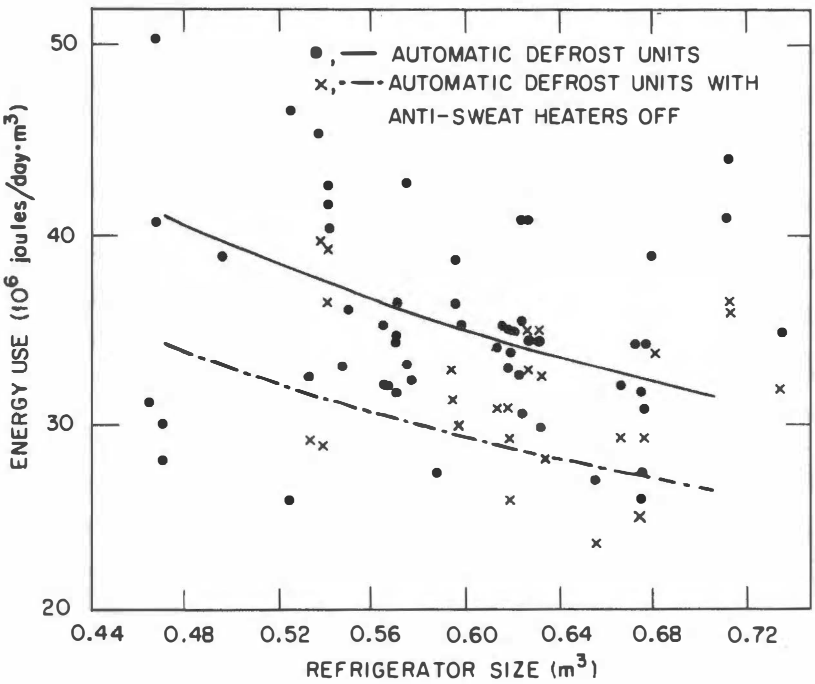

Figure 8. Comparison of energy model with AHAM data for side-by-side refrigerators.

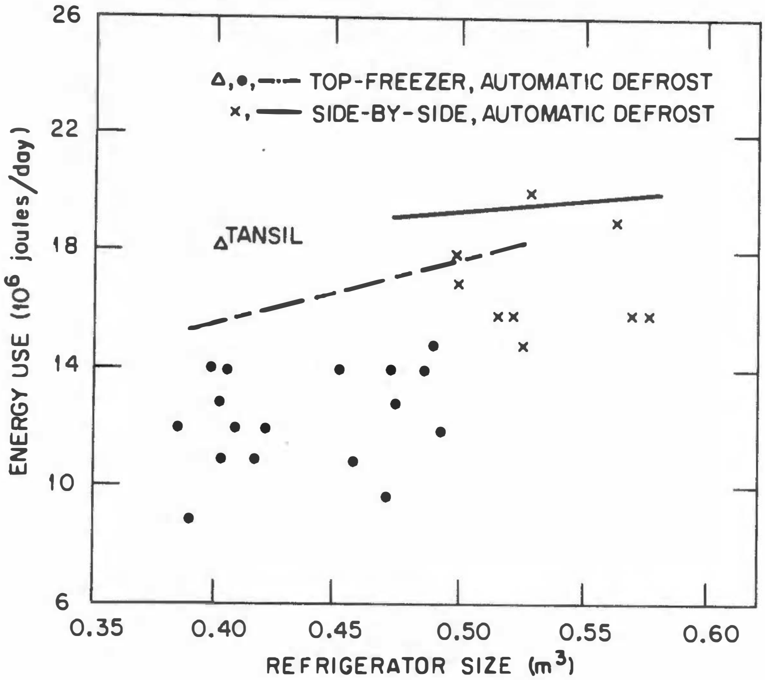

Figure 9. Comparison of energy model with data from *Consumer Reports* and Tansil.

Comparisons of the detailed and total electricity uses for these two units between the model and measurements are shown in Table 5. The model's predictions are roughly 14% lower than manufacturers' data.

Measurements and estimates of refrigerator energy use vary significantly among related studies and manufacturers, as shown in Figures 7-9 and Table 5. The energy model does a good job of predicting average refrigerator energy use. This lends confidence to our use of the model (in the next section) to evaluate alternative refrigerator designs that save energy.

Table 5. Comparison of energy model electricity consumption results with manufacturer data

Component	Unit 1[a] (10^6 J/day)		Unit 2[b] (10^6 J/day)	
	Model	Manufacturer	Model	Manufacturer
Compressor	11.1	12.8	11.4	12.9
Condenser fan	0.9	1.3	--	--
Evaporator fan	0.4	0.6	0.4	0.9
Mullion heater	0.6	0.6	0.4	0.9
Case heater	1.4	1.9	--	--
Drain heater	--	--	0.1	0.1
Defrost heater	1.1	1.2	1.1	0.4
Total	15.5	18.4	13.4	15.2

[a] 0.45 m^3 (16 ft^3) top-freezer, doors closed, freezer temperature -15°C, fresh-food temperature 4.4°C, evaporator temperature -22.2°C, condenser temperature 47.8°C, ambient temperature 32.2°C.

[b] 0.48 m^3 (17 ft^3) top-freezer, doors opened, freezer temperature -17.8°C, fresh-food temperature 2.2°C, evaporator temperature 28.9°C, condenser temperature 40.6°C, ambient temperature 21.1°C. Manufacturer data included only size and type of unit.

5. ENERGY-CONSERVING DESIGN CHANGES

Important energy conservation design changes for refrigerators include: (1) changes in insulation type and/or thickness, (2) removal of evaporator fan motor from refrigerated space, (3) use of anti-sweat heater switch, (4) elimination of frost-free feature, (5) improved compressor efficiency, and (6) increased condenser and/or evaporator heat transfer surface areas. Initial costs for these design changes were obtained from telephone communications with several manufacturers and engineers performing similar studies. These cost changes include manufacturer costs and profit plus wholesale and retail markups.

The energy and cost impacts of each design change are evaluated for a 0.45 m^3 (16 ft^3) top-freezer refrigerator that has a baseline electricity use of 16.6×10^6 J/day (4.62 kwhr/day) and an initial purchase price of $379 (1975-$). The energy impacts of these design changes are detailed in Appendix E; the impacts for this and two other refrigerators are summarized in Table 6. The relationship between changes in electricity use and purchase price for the reference refrigerator is presented graphically in Fig. 10.

Increase and improve insulation: Because 56% of the heat gain is due to conduction through insulated cabinet walls and doors, significant energy savings can be obtained by switching from fiberglass insulation to polyurethane foam (which has a much lower thermal conductivity). Many refrigerators sold today have both polyurethane foam and fiberglass insulation. Some refrigerators use urethane foam but decrease insulation thickness, so that the thermal heat gain is unchanged; then they can boast thinner walls and larger refrigerated volumes. Limitations

Table 6. Energy savings with various design changes

	$0.34\ m^3$ ($12\ ft^3$) Top-freezer[a]		$0.45\ m^3$ ($16\ ft^3$) Top-freezer		$0.57\ m^3$ ($20\ ft^3$) Side-by- side[b]	
	electricity use (10^6 J/day)	% savings	electricity use (10^6 J/day)	% savings	electricity use (10^6 J/day)	% savings
Baseline	14.5	-	16.6	-	20.6	-
1. Increase insulation thickness to 7.6 cm	12.6	13	14.1	14	17.5	13
2. Improve insulation thermal conductivity[c]	11.6	20	13.0	22	16.5	20
3. Remove fan motor from cooled area	14.1	3	16.1	4	19.8	4
4. Use anti-sweat heater switch[d]	13.0	11	15.1	9	19.0	8
5. Eliminate frost-free feature	10.1	31	11.8	29	14.5	23
6. Improve compressor efficiency 20%	12.6	13	14.4	13	18.0	12
7. Increase condenser surface area 20%	14.1	3	16.2	3	19.9	3
8. Increase evaporator surface area 20%	13.6	7	15.5	7	19.1	7
9. 1, 2	10.6	27	11.7	30	15.0	27
10. 7, 8	13.2	9	15.1	10	18.5	10
11. 1, 2, 3, 4, 6, 7, 8,	7.1	51	7.9	52	10.5	49

[a]Frost-free.

[b]Frost-free with automatic ice maker.

[c]Switch from fiberglass to polyurethane foam insulation.

[d]Anti-sweat heaters on 50% of time.

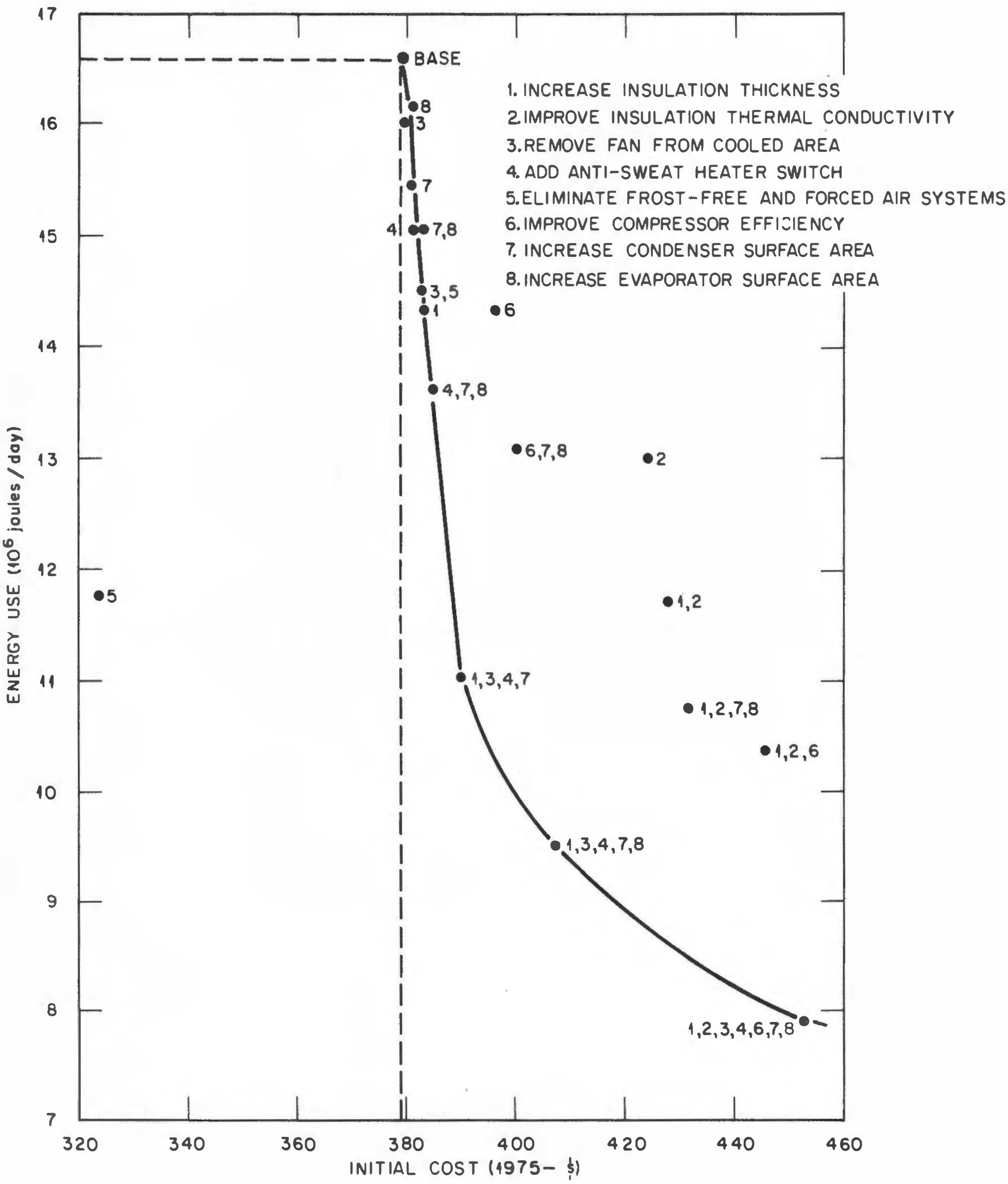

Figure 10. Energy use vs. retail price for various design changes for a 0.45 m^3 (16 ft^3) top-freezer refrigerator.

on the effectiveness of increased insulation thickness are diminishing returns from additional increments of insulation and decrease in food storage volume or increase in exterior dimensions.

Increasing thermal insulation thickness in new refrigerators to 7.6 cm (3 inches) would result in an energy savings of 14% (see Tables E.2-3). This added insulation would cost the consumer $4 more per unit and would save $7.3/year in operating costs (based on 1975 average electricity price of $\$8.88/10^9$ J [3.20¢/kwhr]). The simple payback period would be six months.

Conversion of all insulation material from fiberglass to urethane foam of equal thickness would save 22% (see Tables E.5-6). The foam insulation would cost the consumer $45 more per unit and would save $11.8/year in operating costs. The payback period would be four years.

"Foamed-in-place" urethane foam serves as a structural support for the interior cabinet liner; the steel structural supports otherwise needed can thus be eliminated. This will reduce heat gain, because steel supports allow more heat conduction into the refrigerator than does insulation.

The major constraints on using urethane foam are: (1) urethane is more expensive than fiberglass, (2) labor cost is increased because more labor off the assembly line is required to foam insulation in the cabinets, and (3) urethane foam provides less acoustic damping than fiberglass.

<u>Remove fan motor from cooled area</u>: The evaporator fan, which forces air past the evaporator coil to cool refrigerated air, is a heat gain because the fan motor gives off heat. If the shaft of the fan is

lengthened so that the fan motor is relocated outside the cold space, then the thermal load would only include the heat equivalent of evaporator fan work. Assuming a 15% motor efficiency,[8] the heat gain due to the evaporator fan is reduced 85%. Total thermal gain would be cut 4% (see Tables E.7-8). This design change would cost the consumer less than $1 per unit and would save $1.8/year in electricity. The payback period would be less than three months. Costs to the manufacturer for implementation would be minimal; however, the thermal insulation barrier would be broken, so a secondary sealant would be required (which could make the refrigerator slightly less reliable).

Add anti-sweat heater switch: The anti-sweat heater switch, which appears on many new refrigerators, can save 19% of energy consumption if used. This switch, located on the wall inside the cold storage compartment, has settings for dry and humid ambient air by which the owner is able to control operation of the anti-sweat heaters. Changes in thermal load and electricity consumption due to use of this switch depend on the fraction of time the switch is on. That is, mullion and case heater thermal gains and electricity consumption are controlled by the owner.* An average on-time of 50% is assumed, although studies indicate that people tend to turn the switch on when condensation appears and then leave the switch on during most of the refrigerator's life. An on-time of 50% can save 9% of energy consumption relative to 100% on-time (Tables E.9-10). The switch will increase purchase

*Increased insulation thickness and/or improved insulation thermal conductivity will reduce sweating on exterior cabinet surfaces. Anti-sweat heaters may be eliminated or used less frequently on models with lower thermal conductivity.

price \$3 per unit and will save \$5/year in operating costs. The payback period would be six months.

Eliminate frost-free feature: Elimination of the frost-free feature in refrigerators results in substantial energy savings, because most of the heaters and fans are eliminated. In a manual defrost refrigerator there is no evaporator fan, no defrost heater, no drain heater, and no condenser fan. Elimination of these heaters and fans not only reduces thermal gain, but also reduces the number of electricity-using devices. Total energy savings due to elimination of the frost-free feature (and forced-air system) is 29% (see Tables E.11-12). Manual defrost refrigerators are \$55 less expensive than frost-free refrigerators and consume \$15.7/year less electricity. The problem with this measure is that people purchase frost-free refrigerators for their convenience regardless of increased costs.

Improve compressor efficiency: The major energy consuming device in the refrigerator is the compressor, which uses 70% of total electricity. The overall efficiency (excluding motor losses) of compressors in refrigerators is about 50%.[8] Improving compressor efficiency cuts electricity use for operation of the refrigeration cycle. In addition, heat losses from the compressor are reduced and this lowers thermal gains to the refrigerator.

Increasing compressor efficiency to 60% would cut electricity use 13% (Tables E.13-14). The retail price of the refrigerator would increase \$20, and operating costs would be reduced \$7.3/year. The payback period is 2.5 years.

Increase condenser and evaporator heat transfer areas: An increase in heat transfer surface areas allows the same heat transfer in the condenser and evaporator with a smaller temperature difference. This raises the evaporator temperature and lowers the condenser temperature; which reduces compressor load and increases COP.

The increase of heat transfer surface area is limited, because enough space must be left for air to pass over the coils. A 20% increase in condenser area would reduce electricity use 3% (see Tables E.15-16). This would cut operating costs $1.5/year and increase retail price $2. The payback period would be 1.2 years. A 20% increase in evaporator surface area would cut energy consumption 7% (see Tables E.17-18). This would reduce operating costs $3.7/year and increase retail price $2. The payback period would be less than six months.

6. CONCLUSIONS

The model developed here provides a detailed picture of energy flows and electricity consumption in residential refrigerators. The model is sufficiently flexible to handle different sizes, configurations (e.g., top-freezer), and operating environments (e.g., room temperature, food loads, number of door openings). The model's estimates of daily electricity consumption are in good agreement with measurements reported by manufacturers, AHAM, and Consumers Union.

The primary purpose in developing this model is to evaluate the energy (and related cost) impacts of alternative refrigerator designs. Eliminating the frost-free feature would yield the largest energy savings (29%) and would also reduce the retail price $55 (14%). Although this is the single most effective energy conservation measure, it is not likely to be adopted because consumers generally feel that the convenience offered by the frost-free feature more than compensates for the higher initial and operating costs.

Fortunately, several other design changes are feasible that do not involve lifestyle changes. The largest heat gain in the refrigerator is conduction through walls and doors (56%). Changing from fiberglass to urethane foam insulation reduces this heat gain and cuts energy consumption 22%. Savings in operating costs would pay back capital investments in four years. Increasing fiberglass insulation thickness would cut electricity consumption 14%, and would pay back capital investments in six months.

The major electricity consuming device in the refrigerator is the compressor (using 70% of electricity). Increasing compressor efficiency from 50% to 60% would cut electricity use 13%. The payback period would be 2.5 years.

Installation of an anti-sweat heater switch is also an effective conservation measure; electricity use can be reduced 9% (switch on 50% rather than 100% of time), and the payback period is six months. This feature is already included in most new refrigerators.

Removal of the evaporator fan motor from the cooled area can save 4% of total energy consumption. The payback period is three months.

Increasing the heat transfer surface areas of condenser and evaporator coils 20% will save 3% and 7%, respectively; the payback period for the condenser coil is 1.2 years and for the evaporator coil six months.

Total savings anticipated from all the options discussed in this report would be 71%. The retail price would increase (5%) but lifecycle cost to the consumer (including purchase and operation) would be less. These and the other results presented in Table 6 and Figure 10, p. 29, show the large opportunities for reducing electricity use in new refrigerators—with only slight increases in initial cost.

REFERENCES

REFERENCES

1. E. Hirst, W. Lin, and J. Cope, *An Engineering-Economic Model of Residential Energy Use*, Oak Ridge, Tennessee, Oak Ridge National Laboratory, ORNL/TM-5470, July 1976.

2. "1975 Statistical and Marketing Report," *Merchandising Week*, February 24, 1975 (and earlier annual statistical issues).

3. "No-Frost Top Freezer Refrigerators," *Consumer Reports*, October 1970.

4. G. C. Newton, Jr., "Energy and the Refrigerator," *Technology Review*, January 1976.

5. *The Productivity of Servicing Consumer Durable Products*, Cambridge, Massachusetts, Massachusetts Institute of Technology and Charles Stark Draper Laboratory, May 1974.

6. J. Tansil, *Residential Consumption of Electricity: 1950-1970*, Oak Ridge, Tennessee, Oak Ridge National Laboratory, ORNL/TM-5593, July 1973.

7. *1975 Directory of Certified Refrigerator/Freezers*, Chicago, Illinois, Association of Home Appliance Manufacturers, September 1975.

8. *Study to Produce More Energy Efficient Refrigerators and Water Heaters*, draft, Cambridge, Massachusetts, Arthur D. Little, Inc., January 1976.

9. *ASHRAE Handbook and Product Directory — 1975 Equipment*, New York, New York, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1975.

10. V. O. Haynes, personal communication, Oak Ridge, Tennessee, Oak Ridge National Laboratory, February 1976.

11. "Refrigerator-Freezers," *1970 Buying Guide Issue of Consumer Reports*, 33(12), Mount Vernon, New York, Consumers Union of U.S., Inc., 1969.

12. "Refrigerator-Freezers," *1970 Buying Guide Issue of Consumer Reports*, 34(12), Mount Vernon, New York, Consumers Union of U.S., Inc., 1970.

13. S. H. Dole, *Energy Use and Conservation in the Residential Sector: A Regional Analysis*, draft, Santa Monica, California, Rand Corporation, R-1641-NSF, June 1975.

14. R. Gelbard, personal communication, Louisville, Kentucky, General Electric, March 1977.

15. W. S. Chow, *Energy Conservation and Economic Considerations in Household Refrigeration*, Sc.M. Thesis, Department of Nuclear Engineering, Massachusetts Institute of Technology, Cambridge, Massachusetts, August 1973.

16. *ASHRAE Handbook of Fundamentals*, New York, New York, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1972.

APPENDICES

APPENDIX A

HEAT GAIN THROUGH WALLS

Heat gain through cabinet walls is calculated by the model assuming that the principal source of heat gain is conduction, and other heat transfer modes are negligible. This method is commonly used for refrigerators[15,16], however, more accurate results can be obtained if the combined effects of conduction and convection are used.

Heat gain through walls can be calculated by introducing the overall coefficient of heat transfer, U, such that,

$$q = U\ A\ \Delta T$$

where

$$U = \left(\frac{1}{h_i} + \frac{\Delta x}{k} + \frac{1}{h_o}\right)^{-1}$$

and

h_i = inside film or surface conductance

h_o = outside film or surface conductance

Values for h_i and h_o as functions of air velocity[14] are determined from

$$h_i = (3.85 \times 10^{-3})\overline{V} + 1.77 \qquad \text{(fresh-food)}$$

$$h_i = (3.75 \times 10^{-3})\overline{V} + 1.62 \qquad \text{(freezer)}$$

$$h_o = (3.65 \times 10^{-3})\overline{V} + 2.05$$

where

$\overline{V}$ = velocity of air, ft/min

and

h_i and h_o have units of Btu/hr-ft^2-^{o}F.*

Air flow through the evaporator fan is 40-55 ft^3/min with 10% of the flow directed through the fresh-food compartment and 90% through the freezer. Thus, h_i for freezer and fresh-food compartments are 1.83 and 2.10 Btu/hr-ft^2-^{o}F, respectively. Assuming still air outside the cabinet, h_o is 2.05 Btu/hr-ft^2-^{o}F (on the bottom of the refrigerator the condenser fan operates at 120 ft^3/min normal load, so h_o = 2.61 Btu/hr-ft^2-^{o}F).

Inclusion of the film or surface conduction terms yield values of wall heat gain 10% lower than values obtained by neglecting the air film. Thus, air film acts only as a minor source of heat resistance and is often neglected. Relative energy savings for various design changes calculated by the model are unaffected by omission of the film or surface conductance, since values of h_i and h_o remain unchanged.

---

*For conversion to Standard International units:

1 ft/min = 0.00508 m/sec

1 Btu/hr-ft^2-^{o}F = 5.677 J/sec-^{o}C-m^2

1 ft^3/min = 47.18 × 10^{-3} m^3/sec

APPENDIX B

DERIVATION OF HEAT GAIN DUE TO FOOD

Heat gain due to food in refrigerators is the sum of heat gains in the fresh-food and freezer compartments. The weight of food in each compartment is related to volume of cold storage space by:

$$\dot{w}_f = c_3 V_f \quad \text{and} \quad \dot{w}_r = c_4 V_r$$

where

$\dot{w}$ = weight of food added per week

V = volume

c_3, c_4 = constants

and subscripts f and r refer to freezer and fresh-food compartments, respectively.

Fresh-food: This heat gain is the sum of the temperature reduction and food respiration loads. For temperature reduction:

$$Q_{temp} = \dot{w}_r \, C_{p_{(water)}} \, \Delta T_1$$

where ΔT_1 is the temperature difference between the entering food and the refrigerated air and C_p is the specific heat of water.

The author assumes that the food enters at an average temperature mid-way between the ambient temperature and the fresh-food compartment temperature.

The respiration load is:

$$Q_{resp} = (\text{heat of respiration}) \, \dot{w}_r$$

However, not all food stored in the fresh food compartment requires respiration. The author assumes that one third of the food load requires respiration.

Freezer: Freezer food heat gain is the sum of heat gains from temperature reduction of the food to freezing temperature, latent heat of fusion, and temperature reduction to freezer temperature. The first is:

$$Q_{\substack{\text{temp}\\\text{red to}\\\text{freeze}}} = \dot{w}_f \, C_{p_{(\text{water})}} \, \Delta T_2$$

where ΔT_2 is the difference between the food temperature entering the freezer and the food freezing point and C_p is the specific heat of water. We assume that the temperature of the food entering the freezer is only 25% above the freezer temperature.

The latent heat of fusion is:

$$Q_{\text{latent heat}} = (\text{latent heat of fusion}) \quad \dot{w}_f \, .$$

The temperature reduction load from the food freezing temperature to the freezer temperature is

$$Q_{\substack{\text{temp}\\\text{red to } T_f}} = \dot{w}_f \, C_{p_{(\text{ice})}} \, \Delta T_3$$

where C_p is the specific heat of ice and ΔT_3 is the temperature difference between the freezing temperature of water and the temperature of the freezer (T_f).

Given these individual heat loads for the fresh-food and freezer compartments, the total heat load is:

$$
\begin{aligned}
Q_{food} &= Q_{fresh\text{-}food} + Q_{freezer} \\
&= \dot{w}_r \cdot [0.5\ C_{P_{(water)}}\ \Delta T_1 + 1/3\ (\text{heat of respiration})] + \\
&\quad \dot{w}_f \cdot [0.25\ C_{P_{(water)}}\ \Delta T_2 + (\text{latent heat of fusion}) + \\
&\quad C_{P_{(ice)}}\ \Delta T_3] \ .
\end{aligned}
$$

APPENDIX C

LINEAR DIMENSION EQUATIONS

The following equations for determining refrigerator linear dimensions are based on analysis of manufacturers' literature and actual measurements of a large number of units. These equations yield interior linear dimensions for an "average" refrigerator. These equations are expressed in English units (ft) as they appear in the computer model.*

TOP-FREEZER: Total Volume $\geq$ 14.0 ft^3

$D = 1.98$ $\qquad W_r = W_f = 2.14$

$H_f = 0.02195\ V_{tot} e^{0.06 V_{tot}}$ $\qquad H_r = 0.236\ V_{tot} - H_f$

TOP-FREEZER: Total Volume < 14.0 ft^3

$D = 0.0356\ V_{tot} + 1.177$ $\qquad W_r = W_f = 0.0933\ V_{tot} + 0.616$

$H_f = 0.093\ V_{tot} e^{0.06 V_{tot}} / (D \cdot W_f)$ $\qquad H_r = V_{tot} / (D \cdot W_r) - H_f$

SIDE-BY-SIDE

$D = 0.743\ V_{tot} + 0.3$ $\qquad W_f = 0.0075\ V_{tot} + 0.695$

$H_f = H_r = 0.0148\ V_{tot} + 4.394$ $\qquad W_r = V_{tot} / (D \cdot H_r) - W_f$

*For conversion to Standard International units: 1 ft = 0.305 m.

APPENDIX D

SAMPLE CALCULATIONS FOR A 0.45 m^3

TOP-FREEZER REFRIGERATOR

The following calculations are performed with English units as they appear in the computer model.* A 16 ft^3 top-freezer refrigerator was selected for the energy model sample calculations because this refrigerator accounts for the largest percentage of sales.

Input Values

$V_{tot} = 16.0\ ft^3$

$T_r = 36^oF$

$T_f = 0^oF$

$T_{amb} = 70^oF$

$T_{cond} = 105^oF$

$T_{evap} = -20^oF$

$T_{comp} = 140^oF$

$k = 0.25$ Btu-in./hr^{o}F ft^2

$x_{sides} = 2.5$ in.

$x_{back} = 2.5$ in.

$x_{bottom} = 2.5$ in.

$N_r = 20$ openings/day

$N_f = 8$ openings/day

$n = 3$ cycles/day

$t_{def\ htr} = 12$ min/cycle

$\dot{w}_{ice} = 0.0$ lb/day

$P_{evap\ fan} = 12.0$ Watts

$P_{case\ htr} = 18.0$ Watts

$P_{def\ htr} = 500.0$ Watts

$P_{drain\ htr} = 1.5$ Watts

*Conversion factors:

$T^oC = (T^oF - 32)/1.8$

1 inch = 0.0254 m

1 Btu-in./hr-^{o}F-ft^2 = 14.42 J-cm/sec-^{o}C-m^2

1 lb = 0.454 kg.

$x_{top} = 2.0$ in.

$x_{door} = 1.5$ in.

$f_{\substack{drain\\htr}} = f_{mullion} = f_{\substack{case\\htr}} = 1.0$

$p_{\substack{mullion\\htr}} = 5.0$ Watts

1) Linear Dimensions

$$D = 1.98 \text{ ft}$$

$$W_r = W_f = 2.14 \text{ ft}$$

$$H_f = 0.02195\ V_{tot} e^{0.06V_{tot}} = 0.02195\ (16.0)\ e^{0.06(16.0)} = 0.917 \text{ ft}$$

$$H_r = 0.236\ V_{tot} - H_f = 0.236\ (16.0) - 0.917 = 2.859 \text{ ft} .$$

2) Surface Areas

a) Freezer:

$$A_{top} = (2.14)\ (1.98) = 4.24 \text{ ft}^2$$

$$A_{sides} = 2\ (1.98)\ (0.917) = 3.63 \text{ ft}^2$$

$$A_{back} = (2.14)\ (0.917) = 1.96 \text{ ft}^2$$

$$A_{door} = (2.14)\ (0.917) = 1.96 \text{ ft}^2$$

b) Fresh-food

$$A_{sides} = 2(1.98)\ (2.859) = 11.32 \text{ ft}^2$$

$$A_{back} = (2.859)\ (2.14) = 6.12 \text{ ft}^2$$

$$A_{door} = (2.14)\ (2.859) = 6.12 \text{ ft}^2$$

$$A_{bottom} = (2.14)\ (1.98) = 4.24 \text{ ft}^2$$

3) Heat Gains Through Walls and Doors

a) Freezer:

$$Q_{\substack{facing\\amb\ air}} = Q_{top} + Q_{sides} + Q_{back} + Q_{door}$$

$$= \left(\frac{k_{top}\ A_{top}}{x_{top}} + \frac{k_{sides} A_{sides}}{x_{sides}} + \frac{k_{back} A_{back}}{x_{back}} + \frac{k_{door} A_{door}}{x_{door}}\right)\ (T_{amb} - T_f)$$

$$= \left[\frac{(0.25)\ (4.24)}{2.0} + \frac{(0.25)\ (3.63)}{2.5} + \frac{(0.25)\ (1.96)}{2.5} + \frac{(0.25)\ (1.96)}{1.5}\right]$$

$$(70 - 0)$$

$$= 99.10 \text{ Btu/hr}$$

b) Fresh-food

$$Q_{\substack{\text{facing}\\ \text{amb air}}} = Q_{sides} + Q_{back} + Q_{door}$$

$$= \left[\frac{(0.25)\ (11.32)}{2.5} + \frac{(0.25)\ (6.12)}{2.5} + \frac{(0.25)\ (6.12)}{1.5}\right] (70 - 36)$$

$$= 93.98 \text{ Btu/hr}$$

$$Q_{bottom} = \frac{k_{bottom} A_{bottom}}{x_{bottom}} (T_{comp} - T_r) = \frac{(0.25)\ (4.24)}{2.5} (140 - 36) = 44.09 \text{ Btu/hr}$$

c) Total:

$$Q_{\substack{\text{through}\\ \text{walls}}} = Q_{freezer} + Q_{fresh\text{-}food} = 99.10 + 93.98 + 44.09 = 237.17 \text{ Btu/hr}$$

4) Heat Gain Due to Door Openings

$$V_f = DW_f H_f = (1.98)\ (2.14)\ (0.917) = 3.89 \text{ ft}^3$$

$$V_r = V_{tot} - V_f = 16.0 - 3.89 = 12.11 \text{ ft}^3$$

$$Q_{\substack{\text{door}\\ \text{openings}}} = (\rho C_p)_{air} (V_r N_r \Delta T_r + V_f N_f \Delta T_f)$$

$$= (0.075)\ (0.24)\ [(12.11)\ (20)\ (70\text{-}36) + (3.89)\ (8)\ (70\text{-}0)]$$

$$(1 \text{ day}/24 \text{ hours}) = 7.81 \text{ Btu/hr}$$

5) Heat Gain Due to Heaters and Fans

$$Q_{\substack{\text{evap}\\ \text{fan}}} = P_{evap} f = \left(\frac{3.412 \text{ Btu}}{1 \text{ whr}}\right) (12.0)\ (0.485) = 19.86 \text{ Btu/hr}$$

NOTE: Value for f is obtained from an iterative process, shown in (11).

$$Q_{\substack{\text{def}\\ \text{htr}}} = P_{\substack{\text{def}\\ \text{htr}}} n t_{\substack{\text{def}\\ \text{htr}}} = \left(0.00237 \frac{\text{Btu-day}}{\text{whr-min}}\right) (500)\ (12)\ (3) = 42.66 \text{ Btu/hr}$$

$$Q_{\substack{\text{drain}\\ \text{htr}}} = P_{\substack{\text{drain}\\ \text{htr}}} f_{\substack{\text{drain}\\ \text{htr}}} = \left(\frac{3.412 \text{ Btu}}{\text{whr}}\right) (1.5)\ (1.0) = 5.12 \text{ Btu/hr}$$

$$Q_{\substack{\text{mullion}\\ \text{htr}}} = 0.4 P_{\substack{\text{mullion}\\ \text{htr}}} f_{\substack{\text{mullion}\\ \text{htr}}} = \left(\frac{3.412 \text{ Btu}}{\text{whr}}\right) (0.4)\ (5.0)\ (1.0) = 6.82 \text{ Btu/hr}$$

$$Q_{\substack{case\\htr}} = 0.4p_{\substack{case\\htr}} f_{\substack{case\\htr}} = (\frac{3.412\ Btu}{whr})\ (0.4)\ (18.0)\ (1.0) = 24.57\ Btu/hr$$

$$Q_{\substack{heaters,\\fans}} = Q_{\substack{evap\\htr}} + Q_{\substack{def\\htr}} + Q_{\substack{drain\\htr}} + Q_{\substack{mullion\\htr}} + Q_{\substack{case\\htr}}$$

$$= 19.86 + 42.66 + 5.12 + 6.82 + 24.57 = 99.03\ Btu/hr$$

6) Heat Gain Due to Food Load

$$\dot{w}_f = (4.84)\ (3.89) = 18.82\ lb$$

$$\dot{w}_r = (2.43)\ (12.11) = 29.43\ lb$$

$$Q_{food} = \dot{w}_r[0.5C_{p_{(water)}} \Delta T_1 + 1/3\ (\text{heat of respiration})] + \dot{w}_f[0.25C_{p_{(water)}} \Delta T_2 + (\text{latent heat of fusion}) + C_{p_{(ice)}} \Delta T_3]$$

$$= 29.43\ [(0.5)(1.0)(70-36)\ (\frac{0.00595\ week}{hr}) + 1/3\ (\frac{1\ day}{24\ hr}\ (4500\ \frac{Btu}{day\text{-}ton})\ \frac{1\ ton}{2000\ lb})] + 18.82\ (\frac{0.00595\ week}{hr})\ [0.25(1.0)(70-0) + 143.5 + (0.47)(32-0)] = 23.69\ Btu/hr$$

7) Heat Gain Through Gasket Area

$$L_r = 2(W_r + H_r) = 2(2.14 + 2.859) = 9.998\ ft$$

$$L_f = 2(W_f + H_f) = 2(2.14 + 0.917) = 6.114\ ft$$

$$Q_{\substack{gasket\\area}} = (L_rT_r + L_rT_r)\ (\alpha + \beta f_{\substack{evap\\fan}})$$

$$= [9.998(70 - 36) + 6.114(70 - 0)]\ [0.05 + 0.036(0.485)] = 51.80\ Btu/hr$$

8) Total Thermal Load

$$Q_{thermal} = Q_{\substack{through\\walls}} + Q_{\substack{door\\openings}} + Q_{\substack{heaters,\\fans}} + Q_{food} + Q_{\substack{gasket\\area}}$$

$$= 237.17 + 7.81 + 99.03 + 23.69 + 51.80 = 419.50\ Btu/hr$$

9) Compressor Rated Power and Refrigeration Capacity

$$p_{comp} = a + bT_{cond} + (c + dT_{cond})T_{evap}$$

$$= 290.67 + 0.566(105) + (-3.868 + 0.705(105))(-20) = 279.5 \text{ W}$$

$$Q_{evap} = eT_{evap} - gT_{cond} + h$$

$$= 32.0(-20) - 7.0(105) + 2240.0 =$$

$$865.0 \text{ Btu/hr}$$

10) Coefficient of Performance (COP)

$$COP = Q_{evap}/E_{comp} = \frac{Q_{evap}}{(\frac{3.412 \text{ Btu}}{\text{whr}})p_{comp}} \; 865.0/(3.412)(279.5) = 0.907$$

11) Fraction of Time Compressor is On

$$f = Q_{thermal}/Q_{evap} = 419.50/865.0 = 0.485$$

NOTE: An initial guess for f is used to solve for $Q_{thermal}$ in previous equations and that value is replaced with f calculated above continuously until two values are approximately equal.

12) Electrical Load

$$E_{comp} = fp_{comp} = 0.485(279.5) = 135.6 \text{ w} = 3.25 \text{ kwhr/day}$$

$$E_{evap} = fp_{evap} = 0.485(12.0) = 5.8 \text{ w} = 0.14 \text{ kwhr/day}$$

$$E_{\substack{cond\\fan}} = fp_{cond} = 0.485(29.0) = 14.1 \text{ w} = 0.34 \text{ kwhr/day}$$

$$E_{\substack{case\\htr}} = f_{\substack{case\\htr}}p_{\substack{case\\htr}} = (1.0)\ (18.0) = 18.0 \text{ w} = 0.43 \text{ kwhr/day}$$

$$E_{\substack{def\\htr}} = t_{\substack{def\\htr}}p_{\substack{def\\htr}}n = (12)\ (500)\ (3)\ (\frac{1 \text{ day}}{1440 \text{ min}}) = 12.5 \text{ w} = 0.30 \text{ kwhr/day}$$

$$E_{\substack{drain\\htr}} = f_{\substack{drain\\htr}}p_{\substack{drain\\htr}} = (1.0)\ (1.5) = 1.5 \text{ w} = 0.04 \text{ kwhr/day}$$

$$E_{\substack{\text{mullion}\\ \text{htr}}} = f_{\substack{\text{mullion}\\ \text{htr}}} P_{\substack{\text{mullion}\\ \text{htr}}} = (1.0)\ (5.0) = 5.0 \text{ w} = 0.12 \text{ kwhr/day}$$

$$E_{tot} = [E_{comp} + E_{\substack{\text{evap}\\ \text{fan}}} + E_{\substack{\text{cond}\\ \text{fan}}} + E_{\substack{\text{case}\\ \text{fan}}} + E_{\substack{\text{def}\\ \text{htr}}} + E_{\substack{\text{drain}\\ \text{htr}}} + E_{\substack{\text{mullion}\\ \text{htr}}}]$$

$$= [3.25 + 0.14 + 0.34 + 0.43 + 0.30 + 0.04 + 0.12] = 4.62 \text{ kwhr/day}$$

The Table below shows the computer program's outputs for the reference case calculations given above:

ELECTRIC ENERGY CONSUMPTION AND THERMAL LOADS OF A 16.0 CUBIC FOOT TOP-FREEZER REFRIG/FREEZER COMBINATION

		FRESH FOOD	FZR
DIMENSIONS:	HEIGHT	2.86	0.92
	WIDTH	2.14	2.14
	DEPTH	1.98	1.98

HEAT GAINS	BTU/HR
THROUGH WALLS	237.17
DOOR OPENINGS	7.81
GASKET AREA INFILTRATION	51.80
FOOD LOAD	23.69
HEATERS AND FANS	
A) DEFROST HEATER	42.66
B) DRAIN HEATER	5.12
C) MULLION HEATER	6.82
D) EVAPORATOR FAN	19.86
E) CASE HEATER	24.57
MISCELLANEOUS	0.00
TOTAL THERMAL LOAD	419.50

RATED POWER COMP 279.5 WATTS

QEVAP = 865.00

COP = 0.472

COMPRESSOR RUN TIME = 0.472

ELECTRICITY CONSUMPTION	KWHR/DAY
COMPRESSOR	3.25
CASE HEATER	0.43
CONDENSER FAN	0.34
DEFROST HEATER	0.30
EVAPORATOR FAN	0.14
MULLION HEATER	0.12
DRAIN HEATER	0.04
MISCELLANEOUS	0.00
TOTAL ELECTRIC LOAD	4.62

APPENDIX E

ELECTRICITY SAVINGS DUE TO DESIGN CHANGES

Electricity savings and heat gains from the computer model are expressed in English units. Insulation thicknesses are in inches and thermal conductivities in Btu-in/hr-ft^2-$^\circ$F. Values can be converted to Standard International Units from the relationships given in Appendix D.

ENERGY CONSUMPTION AND THERMAL LOADS DUE TO
* INCREASE IN THERMAL INSULATION THICKNESS

ELECTRICITY SAVINGS= 0.627KWHR/DAY

INSULATION	SIDES	BACK	BOTTOM	TOP	DOOR
THICKNESS	3.00	3.00	3.00	3.00	3.00
THERMAL CONDUCTIVITY	0.25	0.25	0.25	0.25	0.25

HEAT GAINS	BTU/HR
THROUGH WALLS	172.27
DOOR OPENINGS	7.81
GASKET AREA INFILTRATION	49.55
FOOD LOAD	23.69
HEATERS AND FANS	
A) DEFROST HEATER	42.66
B) DRAIN HEATER	5.12
C) MULLION HEATER	6.82
D) EVAPORATOR FAN	16.52
E) CASE HEATER	24.57
MISCELLANEOUS	0.00
TOTAL THERMAL LOAD	349.02

ELECTRICITY CONSUMPTION	KWHR/DAY
COMPRESSOR	2.71
CASE HEATER	0.43
CONDENSER FAN	0.28
DEFROST HEATER	0.30
EVAPORATOR FAN	0.12
MULLION HEATER	0.12
DRAIN HEATER	0.04
MISCELLANEOUS	0.00
TOTAL ELECTRIC LOAD	3.99

ENERGY CONSUMPTION AND THERMAL LOADS DUE TO
* CHANGE IN THERMAL INSULATION CONDUCTIVITY

ELECTRICITY SAVINGS= 1.008KWHR/DAY

INSULATION	SIDES	BACK	BOTTOM	TOP	DOOR
THICKNESS	2.50	2.50	2.50	2.00	1.50
THERMAL CONDUCTIVITY	0.14	0.14	0.14	0.14	0.24

HEAT GAINS	BTU/HR
THROUGH WALLS	132.81
DOOR OPENINGS	7.81
GASKET AREA INFILTRATION	48.18
FOOD LOAD	23.69
HEATERS AND FANS	
A) DEFROST HEATER	42.66
B) DRAIN HEATER	5.12
C) MULLION HEATER	6.82
D) EVAPORATOR FAN	14.49
E) CASE HEATER	24.57
MISCELLANEOUS	0.00
TOTAL THERMAL LOAD	306.16

ELECTRICITY CONSUMPTION	KWHR/DAY
COMPRESSOR	2.37
CASE HEATER	0.43
CONDENSER FAN	0.25
DEFROST HEATER	0.30
EVAPORATOR FAN	0.10
MULLION HEATER	0.12
DRAIN HEATER	0.04
MISCELLANEOUS	0.00
TOTAL ELECTRIC LOAD	3.61

ENERGY CONSUMPTION AND THERMAL LOADS DUE TO
* REMOVAL OF FAN MOTOR FROM COOLED AREA

ELECTRICITY SAVINGS= 0.156KWHR/DAY

HEAT GAINS	BTU/HR
THROUGH WALLS	237.17
DOOR OPENINGS	7.81
GASKET AREA INFILTRATION	51.24
FOOD LOAD	23.69
HEATERS AND FANS	
A) DEFROST HEATER	42.66
B) DRAIN HEATER	5.12
C) MULLION HEATER	6.82
D) EVAPORATOR FAN	2.85
E) CASE HEATER	24.57
MISCELLANEOUS	0,00
TOTAL THERMAL LOAD	401.94

ELECTRICITY CONSUMPTION	KWHR/DAY
COMPRESSOR	3.12
CASE HEATER	0.43
CONDENSER FAN	0.32
DEFROST HEATER	0.30
EVAPORATOR FAN	0.13
MULLION HEATER	0.12
DRAIN HEATER	0.04
MISCELLANEOUS	0,00
TOTAL ELECTRIC LOAD	4.46

ENERGY CONSUMPTION AND THERMAL LOADS DUE TO
* ANTI SWEAT SWITCH ON 0.50 OF TIME

ELECTRICITY SAVINGS= 0.428KWHR/DAY

HEAT GAINS	BTU/HR
THROUGH WALLS	237.17
DOOR OPENINGS	7.81
GASKET AREA INFILTRATION	51.26
FOOD LOAD	23.69
HEATERS AND FANS	
A) DEFROST HEATER	42.66
B) DRAIN HEATER	5.12
C) MULLION HEATER	3.41
D) EVAPORATOR FAN	19.05
E) CASE HEATER	12.28
MISCELLANEOUS	0.00
TOTAL THERMAL LOAD	402.45

ELECTRICITY CONSUMPTION	KWHR/DAY
COMPRESSOR	3.12
CASE HEATER	0.22
CONDENSER FAN	0.32
DEFROST HEATER	0.30
EVAPORATOR FAN	0.13
MULLION HEATER	0.06
DRAIN HEATER	0.04
MISCELLANEOUS	0.00
TOTAL ELECTRIC LOAD	4.19

ENERGY CONSUMPTION AND THERMAL LOADS DUE TO
* ELIMINATION OF FROST FREE FEATURE (AND FORCED AIR SYSTEMS)

ELECTRICITY SAVINGS= 1.344KWHR/DAY

HEAT GAINS	BTU/HR
THROUGH WALLS	238.49
DOOR OPENINGS	7.81
GASKET AREA INFILTRATION	49.61
FOOD LOAD	23.69
HEATERS AND FANS	
A) DEFROST HEATER	0.00
B) DRAIN HEATER	0.00
C) MULLION HEATER	6.82
D) EVAPORATOR FAN	0.00
E) CASE HEATER	24.57
MISCELLANEOUS	0.00
TOTAL THERMAL LOAD	351.00

ELECTRICITY CONSUMPTION	KWHR/DAY
COMPRESSOR	2.72
CASE HEATER	0.43
CONDENSER FAN	0.00
DEFROST HEATER	0.00
EVAPORATOR FAN	0.00
MULLION HEATER	0.12
DRAIN HEATER	0.00
MISCELLANEOUS	0.00
TOTAL ELECTRIC LOAD	3.27

ENERGY CONSUMPTION AND THERMAL LOADS DUE TO
* INCREASE IN COMPRESSOR OVERALL EFFICIENCY TO 0.60

ELECTRICITY SAVINGS= 0.624KWHR/DAY

HEAT GAINS	BTU/HR
THROUGH WALLS	227.28
DOOR OPENINGS	7.81
GASKET AREA INFILTRATION	51.46
FOOD LOAD	23.69
HEATERS AND FANS	
A) DEFROST HEATER	42.66
B) DRAIN HEATER	5.12
C) MULLION HEATER	6.82
D) EVAPORATOR FAN	19.35
E) CASE HEATER	24.57
MISCELLANEOUS	0.00
TOTAL THERMAL LOAD	408.76

ELECTRICITY CONSUMPTION	KWHR/DAY
COMPRESSOR	2.64
CASE HEATER	0.43
CONDENSER FAN	0.33
DEFROST HEATER	0.30
EVAPORATOR FAN	0.14
MULLION HEATER	0.12
DRAIN HEATER	0.04
MISCELLANEOUS	0.00
TOTAL ELECTRIC LOAD	3.99

ENERGY CONSUMPTION AND THERMAL LOADS DUE TO
* INCREASE IN CONDENSER SURFACE AREA TO 1.20 TIMES THE ORIGINAL AREA

ELECTRICITY SAVINGS= 0.127KWHR/DAY

HEAT GAINS	BTU/HR	ELECTRICITY CONSUMPTION	KWHR/DAY
THROUGH WALLS	237.17	COMPRESSOR	3.15
DOOR OPENINGS	7.81	CASE HEATER	0.43
GASKET AREA INFILTRATION	51.15	CONDENSER FAN	0.32
FOOD LOAD	23.69	DEFROST HEATER	0.30
HEATERS AND FANS		EVAPORATOR FAN	0.13
A) DEFROST HEATER	42.66	MULLION HEATER	0.12
B) DRAIN HEATER	5.12	DRAIN HEATER	0.04
C) MULLION HEATER	6.82	MISCELLANEOUS	0.00
D) EVAPORATOR FAN	18.89		
E) CASE HEATER	24.57	TOTAL ELECTRIC LOAD	4.49
MISCELLANEOUS	0.00		
TOTAL THERMAL LOAD	417.88		

ENERGY CONSUMPTION AND THERMAL LOADS DUE TO
* INCREASE IN EVAPORATOR SURFACE AREA TO 1.20 TIMES THE ORIGINAL AREA

ELECTRICITY SAVINGS= 0.320KWHR/DAY

HEAT GAINS	BTU/HR	ELECTRICITY CONSUMPTION	KWHR/DAY
THROUGH WALLS	237.17	COMPRESSOR	2.99
DOOR OPENINGS	7.81	CASE HEATER	0.43
GASKET AREA INFILTRATION	50.22	CONDENSER FAN	0.30
FOOD LOAD	23.69	DEFROST HEATER	0.30
HEATERS AND FANS		EVAPORATOR FAN	0.12
A) DEFROST HEATER	42.66	MULLION HEATER	0.12
B) DRAIN HEATER	5.12	DRAIN HEATER	0.04
C) MULLION HEATER	6.82	MISCELLANEOUS	0.00
D) EVAPORATOR FAN	17.51		
E) CASE HEATER	24.57	TOTAL ELECTRIC LOAD	4.30
MISCELLANEOUS	0.00		
TOTAL THERMAL LOAD	415.57		

ENERGY CONSUMPTION AND THERMAL LOADS DUE TO

* INCREASE IN THERMAL INSULATION THICKNESS
* CHANGE IN THERMAL INSULATION CONDUCTIVITY
* REMOVAL OF FAN MOTOR FROM COOLED AREA
* ANTI SWEAT SWITCH ON 0.50 OF TIME
* ELIMINATION OF FROST FREE FEATURE (AND FORCED AIR SYSTEMS)
* INCREASE IN COMPRESSOR OVERALL EFFICIENCY TO 0.60
* INCREASE IN EVAPORATOR SURFACE AREA TO 1.20 TIMES THE ORIGINAL AREA
* INCREASE IN CONDENSER SURFACE AREA TO 1.20 TIMES THE ORIGINAL AREA

ELECTRICITY SAVINGS= 3.264KWHR/DAY

INSULATION	SIDES	BACK	BOTTOM	TOP	DOOR
THICKNESS	3.00	3.90	3.00	3.00	3.00
THERMAL CONDUCTIVITY	0.14	0.14	0.14	0.14	0.14

HEAT GAINS	BTU/HR
THROUGH WALLS	94.54
DOOR OPENINGS	7.81
GASKET AREA INFILTRATION	43.45
FOOD LOAD	23.69
HEATERS AND FANS	
A) DEFROST HEATER	0.00
B) DRAIN HEATER	0.00
C) MULLION HEATER	3.41
D) EVAPORATOR FAN	0.00
E) CASE HEATER	12.28
MISCELLANEOUS	0.00
TOTAL THERMAL LOAD	185.20

ELECTRICITY CONSUMPTION	KWHR/DAY
COMPRESSOR	1.08
CASE HEATER	0.22
CONDENSER FAN	0.00
DEFROST HEATER	0.00
EVAPORATOR FAN	0.00
MULLION HEATER	0.06
DRAIN HEATER	0.00
MISCELLANEOUS	0.00
TOTAL ELECTRIC LOAD	1.35

VITA

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