Calculating Energy Rating of Domestic Refrigerators Through Laboratory Heat Transfer Measurements and Computer Simulations

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Abstract

Most domestic refrigerators are supplied with an energy rating label which consumers can use to assess the quality of their appliance. This rating can be determined by (a) testing the refrigerator under specified environmental conditions according to relevant standards and measuring its power consumption, or (b) analysing the refrigeration cycle thermodynamically, or (c) measuring the heat transfer through the walls of the refrigerator cabinet and its doors' gaskets. While refrigerator manufacturers use the first technique, most researchers take the second approach. However, this paper is based on our laboratory experimentations and computer simulations following the third methodology. The premise for this investigation is that under steady state conditions the power consumed by the compressor is equal to the rate of heat leakage through the refrigerator cabinet's walls and doors' gaskets as well as the heat input by auxiliary equipment such as light, fan, defroster, etc. This paper contains the details of this experimental study together with a computer simulation which is designed to remove the need for such laboratory experimentations.

After having validated the accuracy of the computer predictions by our own experimental data and those obtained from refrigerator manufacturers, the computer software can now be used especially by manufacturers (a) to find the energy rating of their refrigerators without performing laboratory experiments, and (b) to predict the energy rating of new types of refrigerators from their physical dimensions and characteristics, and environmental conditions before building a prototype. Also, the computer code can be used as a flexible design tool (a) to investigate the effect of varying a single or a multitude of variables on the total energy consumption of the appliance, and (b) to test various refrigerators according to standards from different countries to facilitate the comparison of their energy rating.

Notations

Roman Symbols

COP	Coefficient of performance
d	Gas cell diameter
F	Fraction of on-off time
h	Heat transfer coefficient
k	Thermal conductivity
L	Characteristic length
NDY	No of days/year
NHD	No of hours/day
P	Power
PD	Percentage drop
Pr	Prandtl number
Q	Heat transfer
Ra	Rayleigh number
T	Temperature

Greek Symbols

ε	Emissivity
σ	Stefan-Boltzmann constant
η	Efficiency

Subscripts

1,2	Conditions 1 and 2
aux	Auxiliary
c	Convection
cc	Convection in a channel
com	Compressor
i	Inside
m	Motor
o	Outside
p	Predicted
r	Radiation in cavity
si	Inner surface
so	Outer surface
w	Wall

1.0 Introduction

In most countries around the world, refrigerator manufacturers are required to determine the energy consumption of their products under some specified environmental conditions given in the relevant standards, and label the products accordingly so that the consumers can make an informed decision about their purchase. Australian manufacturers perform these measurements according to AS (Australian Standard) 2575.2 which involve physical laboratory testing of the appliance at an ambient temperature of 32 ^{o}C (as compared to 25 ^{o}C for the European standard ISO 7371-1985) with some specified tolerance. Although these tests are always time consuming and expensive, most companies comply with this requirement so that their products can be marketed. In addition, these tests are performed on finished goods, and therefore do not provide a flexible investigative tool to optimise the appliance's design before manufacture. In order to provide the industry with an inexpensive and fast-response alternative, the present study was undertaken. The research involved both in-house laboratory experimentation and computer simulation; the former was used to verify the predictions of the latter which is the tool that can be used to reduce the need for the laboratory testing of refrigerators required in many countries. In this paper, the technical details of these investigations are provided.

It has been reported that the Australian manufactured refrigerators are inferior to their European counterparts from an energy consumption point of view. For example, Choice magazine [1] reported that the European refrigerators consume 0.3-1.3 kWh/L which is at best over six times and at worst 1.5 times better than the approximately 2 kWh/L rating of the Australian units. Similarly, the Australian Consumers Association tests indicated that the Australian manufactured Kelvinator used 50% more energy when compared to the Japanese Fujitsu General refrigerators [2] of the same size. This disparity in performance may be attributed to the refrigerator design (ie, refrigeration cycle configuration, compressor type and efficiency, insulation type and thickness, and doors' gaskets and their design). Focusing on the thermal insulation as a design parameter, Christensen [3] through a numerical simulation found that by increasing the thickness of the wall insulation from 30 mm to 110 mm, the energy consumption was reduced by 31%. However, he added that any further increment of the insulation thickness did not produce any significant improvement of its energy efficiency. Although a thick-wall refrigerator may not appeal to the consumer, the need for increasing the wall thickness to improve thermal resistance is no longer an obstacle due to the advent of high performance insulation material especially advanced vacuum panels. For example, a 1 in (25.4 mm) thick commercially available VacuPanel [4] is reported to have the same thermal resistance as a 6 in (152.4 mm) thick polyurethane foam, the conventional insulation material for cabinet walls in the refrigeration industry. Similarly, the paper by De Vos et al. [5] contains data on vacuum-based insulation technology while the paper by Griffith and Arasteh [6] reports on their research to further improve the thermal resistance of advanced insulation panels using environmentally safe blowing agents. With specific reference to door insulation, Karlin and Bank [7] have reported the results of their work on the manufacture of a hollow all-plastic refrigerator door with a superior thermal resistance as compared to the conventional foam-insulated doors used in refrigerators.

The usual method of finding the energy efficiency of refrigerators is by studying their refrigeration cycle from a thermodynamics point of view. For example, Radermacher [8] has reported that a dual-loop system has the potential to reduce the energy consumption of these appliances by 20% when more efficient and small compressors are used. An alternative method which is used in the present investigation is to calculate this efficiency by determining the heat transfer characteristics of the refrigerator cabinet and its doors' gaskets because under steady-state conditions the rate of heat exchange between the inside and outside of the refrigerator is equal to the power consumed by the compressor and other auxiliary components such as the fan, the light, the defroster, etc. A number of other investigators have also used this procedure for the same purpose: Sand et al. [9] and Vineyard et al. [10] used both experimental and numerical techniques to estimate cabinet heat loss rates and closed-door energy consumption values from basic cabinet and refrigeration circuit inputs.

In the experimental part of the investigation, a domestic refrigerator was instrumented with many thermocouples and heat flux meters to respectively measure the inside and outside air temperatures and the heat transfer through the walls of the unit. These measurements were taken under different environmental conditions as well as different walls' insulation thicknesses; the aim of the latter investigation was to quantify the drop in energy consumption when the refrigerator wall resistance against heat transfer is increased. The results of these measurements were then compared with the total energy used by the unit during the test period. As shown later in this paper, the results indicated that about 87% of the energy exchange in the refrigerator took place through the walls of the unit. The remaining heat transfer (ie, 13%) was assumed to have occurred mostly through the doors' gaskets. This figure compares well with the figure of 30% which is reported to be the fraction of the load being attributed to the "edge losses" by Boughton et al. [11] in a study of heat transfer in the region near the door and wall gaskets. Also, it was found in this study that by adding one layer of 30 mm polystyrene[1] to the walls of the refrigerator the energy consumption of the refrigerator operating under steady-state conditions and no load dropped by 11%; increasing the thickness of this additional insulation to 60 mm, reduced the energy consumption by 17%. As mentioned earlier, although increasing the wall insulation thickness might reduce sales, the development of new insulation material should provide a solution to this marketing obstacle.

In the numerical part of the study, a computer program using basic heat transfer principles was prepared to predict the energy rating of domestic refrigerators based on their physical characteristics and environmental conditions. The predictions of the computer code were tested against the experimental data gathered in-house (similar to the studies reported in [9-10]) as well as those obtained from a number of refrigerator manufacturers. The average error in estimating the power consumption of the domestic refrigerators considered here was found to be less than 10% for most units as illustrated later in the paper.

The additional advantage of the computer simulation package is that it can be used to investigate the effect of varying a single or a multitude of parameters on the total energy consumption of the product. For example, the thermal resistance of (or the insulation thickness in) the walls of the refrigerator can be increased to assess the improvements in its energy efficiency without performing actual laboratory experiments. Therefore, using this computer package, manufacturers not only can find the energy rating of their products but also design refrigerators with a specific energy consumption which can then be built for laboratory testing. The advantage of using this investigative tool is to reduce the laboratory work and prototype building which has also been the subject of a detailed study by Zalba et al. [12]. Furthermore, the computer code can be used to test various refrigerators according to standards from different countries to facilitate the comparison of their energy consumption, ie, a European refrigerator may not have as good an efficiency in Australia as it would in Europe where the ambient temperature is often lower than that in Australia. The results of this study would compliment those reported by Bansal and Kruger [13] and Bansal et al. [14] who in separate papers reported the results of their investigations of testing standards from a number of countries, the salient differences between them, and how to predict the performance of one refrigeration system according to different testing standards.

2.0 The Experimental Method

The refrigeration industry has been brought to the public's attention during the last few years by the politicians, researchers and scientists because of its role in the depletion of the protective ozone layer in the stratosphere. It is believed that the CFC-based refrigerants used in the domestic and industrial refrigeration installations have been one of the agents in this process. As a consequence, many researchers and manufacturers[2] have been looking for alternative fluids to replace the commonly used R-12. However,

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[1] The thermal resistance of 30 mm polystyrene is equivalent to that for 25 mm polyurethane foam.

[2] It was reported in 1993 [15] that Whirlpool in its search for environmentally safe refrigerators designed and tested a R134a-based refrigerator/freezer which is 29% more energy efficient than the prevailing government standards.

an equally important aspect of refrigeration which concerns the environment through the consumption of electric energy has received very little attention. The benefits of improving the energy efficiency of refrigerators are twofold: (1) the operating cost to the consumers will be minimised, and (2) the impact of the combustion gases on the environment as a result of the burning of fossil fuels to generate electricity will be reduced. Therefore, this study was initiated in order to address the latter problem of energy efficiency by looking at the heat transfer characteristics of the refrigerator cabinet and its insulation, and how they can be changed to reduce the heat exchange between the inside and the outside of the refrigerator.

The experiments reported herein were carried out on a fully-instrumented domestic refrigerator. It involved the measurement of the temperature gradient across, and heat flux through the walls of the cabinet, the temperature distribution inside and outside the refrigerator, and the power consumption during the experiments. The initial experiments were done on the unit without altering the wall insulations. However, in subsequent experiments, the insulation was gradually increased and the heat transfer characteristics were measured.

2.1 Experimental Set-up and Instrumentation

The experimental results reported in this paper are for a family size (370 L), two door, cyclic defrost, domestic refrigerator made in 1991 when the tests were conducted. The thermodynamic hardware of the unit is comprised of a compressor, two rollbond evaporators (one in the freezer and the other in the fresh-food compartment hereafter referred to as the cooler), a two-part condenser, a capillary tube and a thermostat which is located in the cooler and is activated by the temperature of the refrigerant in its evaporator. The outside of the refrigerator cabinet is made of a 0.6-mm thick enamel coated sheet steel while the inside of the unit has a 0.6-mm modified high impact polystyrene liner. The space between the liner and the outside shell is filled with expanded polyurethane foam using a mixture of carbon dioxide and R-11. Although the thermal resistance of the insulation was found to be different at different points (which could be due to the non-uniform expansion of the foam), the thickness of the insulation was about 60 mm in the freezer and around 40 mm in the cooler.

The refrigerator was instrumented such that the temperature inside and outside the unit, the temperature gradient across and the heat flux through the walls, and the power consumption during the experiments could be measured simultaneously. There were 50 copper-constantan thermocouples (30 in the cooler and 20 in the freezer) for measuring the temperature distribution inside the refrigerator and at its internal walls. The temperature gradient across and the heat flux through the cabinet walls were measured at 15 points for the freezer walls and 25 points for the walls of the cooler; fewer heat flux meters as compared to the thermocouples were used in the experiments due to the nonavailability of sufficient number of meters with the same calibration constant. The heat flux meters were built in-house and calibrated against a standard commercial meter[16]. The resulting heat flux and temperature gradient data were used to determine the thermal resistance of the insulation at various points. The thermocouples were attached to the walls by using ordinary masking tape while the heat flux meters were mounted on the walls with a double-sided tape; specific experiments indicated that the resistance of the tape was negligible when compared with that of the insulation. For better identification, the walls of the refrigerator were labelled according to the diagram

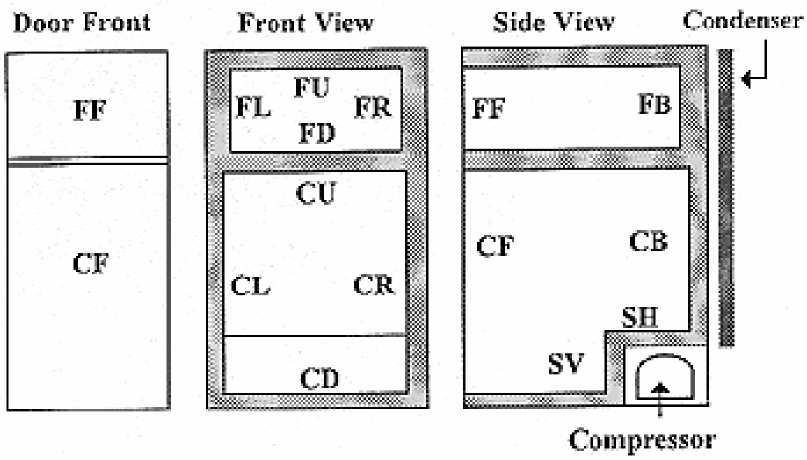

Figure 1: Identification of the walls of the refrigerator used in the experimental set-up.

shown in Figure 1. As can be seen in the diagram, the first letter (F or C) refers to the compartment (ie, freezer or cooler) and the second letter indicates the wall being U, B, L, F, R or D (for up, back, left, front, right or down, respectively). The only exception is the walls around the compressor which are identified by SV and SH (for step-vertical and step-horizontal, respectively).

Table 1: Heat flux (W/m^2) and thermal resistance data (m^2-K/W) given inside { } for two different walls as viewed from inside the unit shown in Figure 2.

Wall CL (cooler)		
41.2 {0.9}	27.6 {1.4}	22.3 {1.8}
41.2 {0.8}	29.7 {1.4}	26.8 {1.5}
43.3 {0.8}	37.1 {1.1}	28.0 {1.5}
39.2 {0.9}	31.8 {1.3}	28.0 {1.5}
32.0 {1.2}	23.9 {1.8}	16.9 {2.6}
Wall FL (freezer)		
27.7 {1.3}	14.8 {2.3}	9.6 {3.3}
30.2 {1.2}	16.1 {2.1}	11.0 {2.9}
19.0 {1.9}	12.8 {2.6}	9.2 {3.5}

The temperature and heat flux measurements were made by using Decipher which is a computer software written for data acquisition by a Datataker (DT100). This particular Datataker could handle up to 22 channels at a time. Therefore, when necessary, a make-shift approach was adopted to gather the data. The power consumption of the unit was measured by a type M3 kilowatt-hour meter.

2.2 Experimental Procedure and Results

The initial part of this investigation was directed towards the determination of the thermal resistance of the refrigerator insulation at various points. This was achieved by treating the refrigerator as a thermal box with an internal heat source so that the complications due to the switching of the compressor (which would cause internal temperature fluctuations) as well as the electric resistance heaters which are located inside the walls

Table 2: Heat transfer data for various walls with and without additional insulation.								
	Experimental Results					Computer Predictions		
Walls	Q_w [a] (W)	Q_w/Q (%)	Q_w [b] (W)	Q_w/Q (%)	PD [c] (%)	Q_p [d] (W)	Q_p [e] (W)	Q_p [f] (W)
FU	6.2	23	4.2	22	32	6.7	4.1	5.2
FB	5.1	19	3.1	17	39	4.7	3.3	4.1
FL	3.8	14	3.1	17	18	4.2	2.5	3.2
FF	4.6	17	3.6	19	22	4.7	2.9	3.7
FR	3.8	14	3.1	17	18	4.2	2.5	3.2
Subtotal	23.5	87	17.1	92	27	24.5	15.3	19.4
	Q = 27 W		Q = 19 W		30	-	-	-
Predicted Freezer Gasket Load						4.5	3.9	4.9
CB	20.2	25	11.8	24	42	16.9	6.5	9.8
CL	12.1	15	6.9	15	43	15.0	4.4	7.2
CF	18.6	23	9.5	20	49	16.9	4.9	8.1
CR	15.4	19	8.7	19	44	15.0	4.4	7.2
CD	4.1	5	3.4	7	17	7.2	2.8	4.3
Subtotal	70.4	87	40.3	85	43	71.0	23.0	36.6
	Q = 81 W		Q = 48 W		41	-	-	-
Predicted Cooler Gasket Load						10.9	4.5	7.2
Total	93.9[g]	-	57.4[g]	-	-	110.9	46.7	68.1

a Without additional insulation.

b With additional 30-mm polystyrene insulation.

c Reduction in heat transfer due to additional insulation.

d Predictions of experimental data given in 2nd column.

e Predictions for standard operating conditions.

f Predictions for conditions specified in AS 2575.2.

g Excluding gasket load.

of the cabinet are eliminated. In order to show the thermal resistance characteristics of the cabinet walls, as an example, the resulting data for Walls CL and FL are given in Table 1. Although the measurements were taken at 25 points for the cooler and 15 points for the freezer, only 15 data for the former and nine for the latter are included in the table. This choice was made based on the results which showed that most of the variation took place over the front (approximately) 20% of the walls with the remainder of the wall having an almost constant thermal resistance.

For these experiments, a heat source of known magnitude was placed inside each of the two refrigerator sections, one in the cooler (85 W) and one in the freezer (25 W), with individual controls using separate variacs so that the temperatures could be adjusted as necessary. The average internal temperature was 66 ^{o}C for the former while for the latter it was measured to be 67 ^{o}C with a uniform ambient temperature of 22 ^{o}C. When steady-state was achieved, the heat flux through each wall, the heat input to each compartment and all relevant temperatures were measured. These experiments were performed twice, once all the measurements were taken with the refrigerator in its as manufactured condition, and then with an additional layer of a 30-mm polystyrene insulation added to all the walls. All the results are presented in Table 2. Note that the heat transfer through the wall between the freezer and the cooler is internal to the unit and therefore has not been measured.

The aforementioned experiments were carried out by treating the refrigerator cabinet as a thermal box with a controlled heat source in order to determine the heat transfer characteristics of the cabinet, namely, (1) the thermal resistance of its wall insulation and the effect of increasing this thermal resistance on the heat transfer through the walls, and (2) the sealing properties of the gaskets used in its freezer and coolers. Obviously, the refrigerator is used under different conditions in the field but they would not significantly affect the thermal resistance and the gasket sealing properties. Therefore, these data can be used for the case where the unit is operating as a refrigerator.

2.3 Discussion of the Experimental Results

The results reported in this paper were obtained under steady-state conditions which required a stabilisation time period ranging from 15 minutes to 2 hours for different runs. During the experiments, an attempt was made to ensure that the temperature distribution in the freezer and the cooler was as uniform as possible for the various experiments, and also that the temperature gradient across the freezer-cooler interface was almost zero. Furthermore, the heat source was placed in the bottom near the back side in the cooler so that the resulting circulation pattern is the exact opposite of that when the evaporator is running; this could not be readily created for the freezer which is cooled by the evaporator on all sides except for the front and the back.

The resulting internal temperature profiles showed an insignificant gradient in the horizontal direction, but the temperature change between the bottom and the top of the cooler was of the order of 3 ^{o}C; the vertical temperature gradient in the freezer was almost nil.

The data of Table 1 show the variation of the thermal resistance of the wall insulation as a function of space. The relatively lower values near the front edge can be partly explained by the tapered construction of the walls. Also, because of the face-down position of the cabinet during the foam injection process, it is possible that the polyurethane foam in this area has a higher density and therefore lower thermal resistance. The result is a relatively higher heat flux at the front edges as compared to the other parts. Furthermore, when the thermal resistance data for all the walls were examined, it became apparent that although the sides of the refrigerator showed a relatively good thermal insulation, the top, the front and the back walls of the unit needed more attention.

The heat transfer through the doors' gaskets could not be measured directly due to the uneven shape of the seals as well as the uncertainty associated with the heat flow through the seals and the edges of the doors. Therefore, this heat transfer rate was lumped together with other losses and estimated from the data provided in Table 2 by taking the difference between the power consumed by the heat sources and the total heat transfer through the walls that was measured by the heat flux meters. The final results are 14.1 W for the case with no additional insulation (column 2 of Table 2), and 9.6 W with the additional insulation (column 4 of Table 2). These data show that 13% of the power consumed by the heaters (when the refrigerator was treated as a thermal box) in the refrigerator cabinet with no additional insulation could not be measured by the network of heat flux meters used in the experiments. It is believed that a significant amount of this heat loss took place through the doors' gaskets, with the balance accounting for small sections of the cabinet which were not adequately insulated during manufacture (eg, area around the thermostat assembly) and sections where the polyurethane foam expansion was hindered[3]. Also, the data of Table 2 clearly show that the additional insulation reduced the power consumption of the heat sources by 30% in the freezer and by 41% in the cooler. These measurements were also taken when the refrigerator was operating. It was found that the power consumption of the refrigerator dropped by 11% and 17% respectively when 30 mm and 60 mm of polystyrene insulation was added to the walls of the refrigerator [18].

3.0 The Computer Simulation Software Using Basic Heat Transfer Principles

The premise upon which the simulation stands is that under steady state conditions the power used by the compressor is mainly for the purpose of removing the heat which has leaked into the refrigerator cabinet through the walls and the doors' gaskets and the heat input by the light, fan, defroster, etc. (hereafter referred to as the auxiliary components). Therefore, this energy consumption can be determined by considering the relevant heat exchanges which take place between the inside and the outside of the refrigerator and by accounting for the energy consumption of the auxiliary components.

The heat transfer through the walls of the refrigerator can be studied by considering the wall as a three-layer composite structure (with typical thicknesses) as shown in Figure 2. The outer wall (or shell) is generally made of steel, the middle layer which provides most of the resistance to heat transfer is formed by expanded polyurethane foam and the inner liner is made of commercially available polystyrene. Therefore, the heat transfer process through the inner and outer material can be adequately described by Fourier's diffusion equation. But the heat flow through the middle layer cannot readily be identified as conduction because of the gas-filled pores in the foam and therefore the existence of convection and radiation heat transfer. The relative magnitude of the various modes of heat transfer through the walls depends to a large extent on the foam density and therefore the size of the gas-cells in the foam as well as the orientation of these cells in relation to the direction of heat flow. As such, the heat transfer process is complex and difficult to model. However, a simplified model can be

[3] Mitani and Yokohama [17] reported that the expansion of polyurethane foam is dependent on the size of the cavity, the foam injection ports and the surface properties of the liner and the shell surfaces.

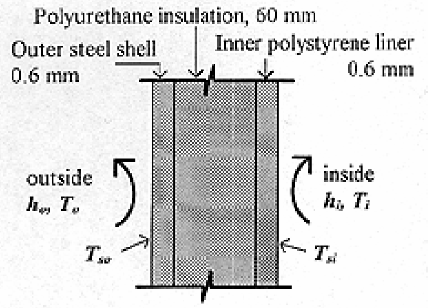

Figure 2: Schematic diagram of a typical refrigerator wall.

constructed by looking at the heat flow in the cavity of the wall assuming that the cavity is filled with the appropriate proportions of solid plastic matrix and the gas used as the blowing agent. Valenzuela and Glicksman [19] reported that 97% by volume of the expanded polyurethane foam is occupied by the blowing agent (which is usually 50% R12 and 50% CO_2). Therefore, a simple model can ignore the effect of the solid plastic matrix and compute the heat flow through a gas-filled cavity with a thermal conductivity of 0.011 W/m-K, an average value for a gas containing R12 and CO_2 on an equal basis [20]. The heat flow diagram for such a situation in the form of an electric circuit is given in Figure 3.

The convective heat transfer coefficients for the inside and outside surfaces of the wall can be computed from [21]

$$h_c = \frac{k}{L}\,(0.59\ Ra_L^{0.25})$$

while the radiation heat transfer coefficient can be found from the Stefan-Boltzmann

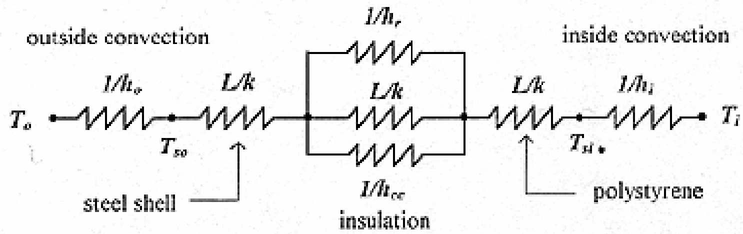

Figure 3: Heat flow circuit for the wall of Figure 2 .

law, ie,

$$h_r = \frac{\sigma\ \varepsilon\ (T_2^4 - T_1^4)}{T_2 - T_1}$$

where the surface emissivity is taken as 0.7 to account for the performance deviation of the surfaces from blackbody behaviour. The convective heat transfer coefficient for the cavity treated as an open channel can be found from [22]

$$h_{cc} = \frac{k}{L}\,(0.68 + 0.513\ Ra_L^{0.25})$$

The solution of these equations for a typical refrigerator wall of 1150 mm height and 600 mm depth and T_i=4 °C and T_o=22 °C show that the total heat transfer through the wall is 14.9 W. Intermediate results from the solution of these equations indicate that the percentages of heat transfer through the wall by conduction, radiation and convection are 3, 68 and 29, respectively; the latter figure disagrees with results available in the literature [19]. A slightly refined version of this simple model can include the effect of the 3% (by volume) solid polymer (k=0.25 W/m-K) [23] in the cavity but calculations indicate that it does not have a significant effect on the convection and radiation components of the heat transfer due to its small thermal resistance.

The problem with the above model is that the spacing between the two surfaces which enclose the gas is significantly larger than that for the surfaces which enclose the gas in the pores of the foam. Therefore, a better model can be constructed if the solid plastic component of the foam is considered to be consisted of thin plastic sheets which subdivide the gas-filled cavity into a number of small layers such that the width of each gas layer is the same as the gas-cell diameter present in real polyurethane foam. The cell diameter is generally accepted to be around 4 mm [19]. Therefore, considering the cavity to be composed of many 4-mm wide gas layers (***k***=0.011 W/m-K), each one followed by a thin plastic sheet, the total heat transfer through the wall drops to 3.2 W, majority of which (about 82%) is by radiation and about 14% by convection which are in good agreement with the results reported in [19]. In this analysis the cavity is treated as a rectangular enclosure with an aspect ratio between 1 and 2. Accordingly, the heat transfer coefficient was calculated from [24]

$$h_d = \frac{k}{d}\left(0.18\frac{\mathrm{Pr}}{0.2+\mathrm{Pr}}\,Ra_d\right)^{0.29}$$

Further reduction of the gas layer thickness to 2 mm does not change these percentages significantly indicating that below 4 mm the heat transfer mechanism remains unchanged.

A further simplification of the above model can be made by treating the polyurethane layer in Figure 2 as a solid with an effective thermal conductivity of 0.019 W/m-K [23]. Using the Fourier's conduction equation and the appropriate boundary conditions, the total heat transfer through the wall was found to be 3.07 W indicating that this simple model can produce accurate results similar to the model represented by the heat flow diagram of Figure 3. As explained in the next section, this heat flow rate also agrees well with experimental heat flux measurements of Hessami and Thesiera [18]. Therefore, this simple conduction model was used in the computer code to determine the energy consumption of refrigerators as described in the following section.

3.1 Refrigerator Energy Consumption Through Computer Simulation

Energy efficiency of refrigerators can be studied from a purely thermodynamics point of view [25-26] which can include, among other things, the effect of pure as well as mixed refrigerants on cycle performance [27-28]. Although such thermodynamic analyses provide useful data to design a better refrigeration cycle, they fail to provide any insight towards the design of better refrigerator cabinets which are equally important in improving energy efficiency. As mentioned before, because the refrigerator load under steady state conditions is mainly due to the heat exchange between the inside and the outside of the unit, it is desirable to know how this heat exchange can be minimised by changing the insulation properties of the cabinet walls and/or door gaskets. The conventional method of attack for such an investigation is to perform laboratory experiments which can be time consuming and expensive as described earlier. However, with the abundance of better computer facilities and the skills to write the necessary software, the same investigation can be carried out at a fraction of the cost and at a much faster speed using a personal computer. Hence, the motivation for preparing a computer code which can be used to study the energy efficiency of refrigerators from a heat transfer point of view as described below.

The computer code is written in TURBO PASCAL in a modular form. Modular program design has many advantages especially when developing a complex simulation. Components of the simulation can be modelled and developed independently, with minimal change to its other parts of the code. For example, the heat transfer through the cabinet walls may initially be assumed uniform throughout the cabinet but later modified to include the effects of variable insulation thickness and/or ambient temperature on the heat transfer; a modular program would allow this change to be made easily without altering the main code. Therefore, a rudimentary modular simulation using simplified equations can be refined and further developed at a later date by improving the accuracy of the existing modules and/or by adding new ones to the code.

The code which was developed during this research project and described herein contains 11 Pascal modules (each of which performs a specific task), one help file for on-line assistance, and two input data files. The program is menu-driven so that input-data alteration and program execution can be made by selecting the appropriate options from the menu bar which has the details shown in Figure 4. Once all the relevant data have been entered, the code calculates the load through each wall and the door gaskets as described in the flow-chart of Figure 5.

As shown in the flowchart, the inside and outside surface temperatures were found by iterations. Also, the simple conduction model described above is used to calculate the heat transfer through all the walls of the refrigerator. For convenience, the refrigerator

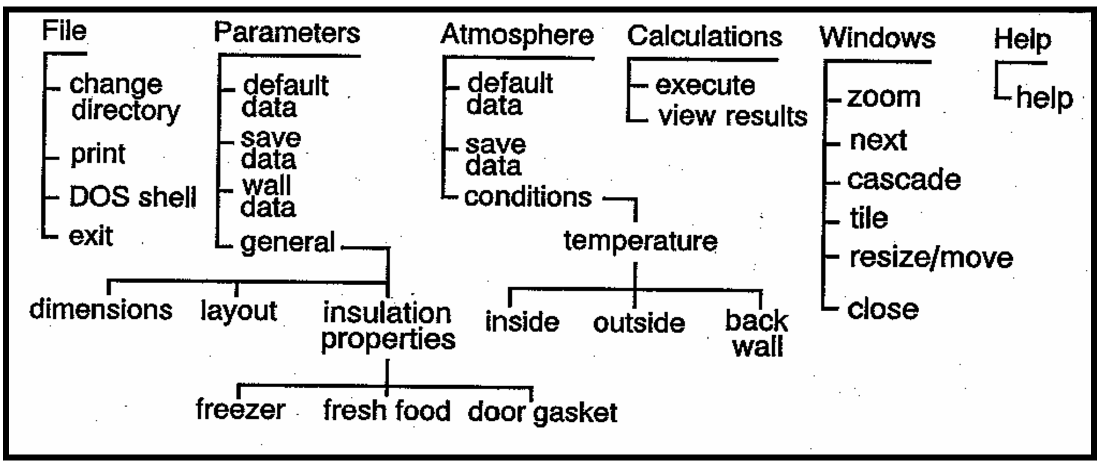

Figure 4: Details of the menu bar.

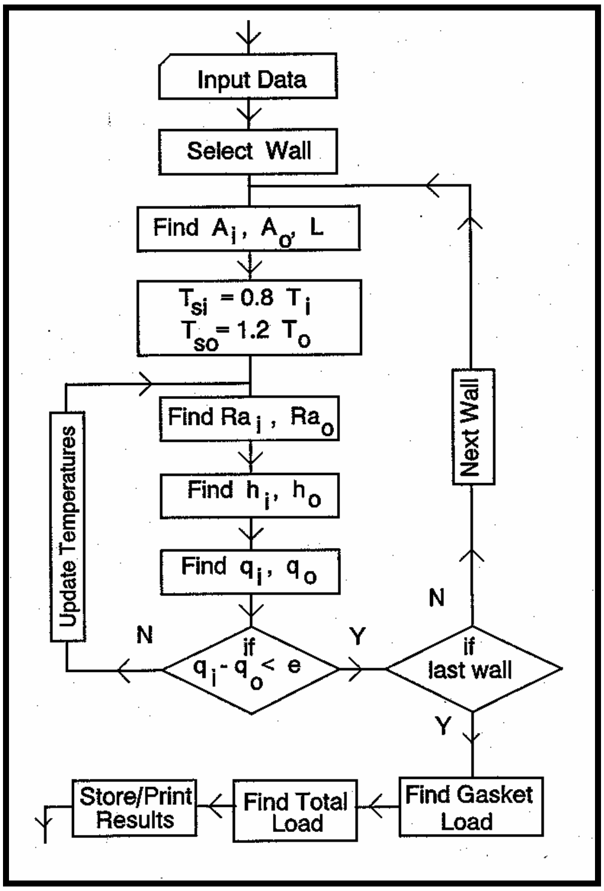

Figure 5: Computer code's flowchart.

cabinet was divided into its constituent panels and identified by a label as was shown in Figure 1.

The results of the computer simulations are given in the first column under Computer Predictions in Table 2 where the experimental data are listed. These computer results are intentionally generated verify the computer code's accuracy against the available hot-box experimental data in which $\boldsymbol{T_i}$=66 °C for the freezer and $\boldsymbol{T_i}$=67 °C for the cooler, and $\boldsymbol{T_o}$=22 °C. The insulation thicknesses in these calculations were specified to be 60 and 40 mm for the freezer and the cooler, respectively. As shown, the agreement between the experimental data and their predictions is reasonably good. Also included in the second column of Table 2 is the predicted heat transfer data for the various walls of the refrigerator for more realistic temperatures of $\boldsymbol{T_i}$=4 °C for the cooler and $\boldsymbol{T_i}$=-16 °C for the Freezer, and $\boldsymbol{T_o}$=22 °C. The heat transfer of 4.4 W for Wall CL (insulation thickness = 40 mm) is the computer code's result for the wall which was considered in the analysis of the previous section; these calculations for a 60-mm wall insulation gives a heat transfer of 3.2 W which is in excellent agreement with the results obtained for the models described in the previous section. The last column of Table 2 provides the heat transfer data for the refrigerator when operating under the test conditions specified in AS2575.2, ie, -15 °C, 3 °C, 32 °C and 38 °C respectively representing the freezer and cooler inside temperatures, the outside temperature and the temperature behind the refrigerator where the condenser and the compressor are mounted. As shown in the table, the total load on the refrigerator is 68.1 W. This is the major load on the compressor. However, in actual refrigerators, there is likely to be an electric fan, a defroster or anti-sweat heater (using either electricity or the relatively warm refrigerant), an internal light, etc. These auxiliary components affect the energy consumption of the refrigerator in two ways: (1) their heating contribution becomes an additional load on the compressor, and (2) they consume electricity for their operation. Therefore, the annual energy consumption (***AEC***) for the refrigerator which was tested in the laboratory, based on a total compressor power consumption ($\boldsymbol{P_{com}}$) of 68.1 W, an auxiliary load ($\boldsymbol{P_{aux}}$) of 22 W (the size of the electric defroster), a 50% on-off cycle designated by ***F***, a ***COP*** of 1.1 and an electric motor efficiency ($\boldsymbol{\eta_m}$) of 88%[4], becomes:

[4] In this analysis motor efficiency was assumed to vary from 0.73 to 0.94 depending on the size of the motor which in turn depends on the size of the refrigerator. The lowest efficiency of 73% was for a refrigerator having a refrigerated volume of

$$AEC = \left(\frac{P_{com} + P_{aux} \text{ x } F}{COP \text{ x } \eta_m} + \eta_m \text{ x } P_{aux} \text{ x } F \right) \text{x } NDY \text{ x } NHD$$

$$= \left(\frac{68.1 + 22 \text{ x } 0.5}{1.1 \text{ x } 0.88} + 22 \text{ x } 0.5 \right) \text{x } 365.25 \text{ x } 24 = 812.7 \frac{\text{kWh}}{\text{y}}$$

where ***NDY*** and ***NHD*** are the number of days per year and number of hours per day, respectively. This figure compares very well with the manufacturer's 810 kWh/y energy label of the refrigerator being studied.

It should be noted that there are a number of uncertainties in the computer model, namely, the treatment of the inside surfaces of the doors as flat surfaces, and the gaskets as 'black-boxes' having the thermal resistance deduced from experimental measurements; these aspects of the study are currently being investigated for further refinement. Despite these shortcomings, the agreement between the computer code's energy consumption rating of domestic refrigerators and their labels produced by manufacturers is reasonably good.

3.2 General Remarks on Computer Simulation's Results

The computer code which was described earlier is a useful tool which can be used to find the energy label of refrigerators without actual laboratory testing. Table 3 contains comparative data between the predictions using the specifications in AS 2575.2 and actual energy consumption labels provided by the manufacturers of eight different refrigerators. These data with 10% error bars on the 45^{o} line are plotted in Figure 6. Although the predictions seem to show an error of greater than 10% for three units, the remaining data fall within this error band which could be in the laboratory measurements or in the predictions of the present computer software. It should be noted that two of the three refrigerators with relatively large errors had their freezer located below the cooler unlike the other six units examined. It is possible that the computer code in its present form cannot accurately predict the performance of this type of refrigerators. Furthermore, the treatment of the door gaskets as a 'black-box' with a single value of thermal resistance for all refrigerators might have caused some error. In order to address this problem, a new experimental study has been initiated recently to study the heat transfer through door gaskets under various conditions. The results of this study will be incorporated into the computer code which is hoped to reduce the above discrepancy between the manufacturer's energy ratings and those found by the code. Despite these shortcomings, it can be concluded that the code in its present form can accurately predict the energy consumption of small refrigerators but needs further refinements to improve its accuracy for larger units.

The computer software is a flexible tool which can be used to design and manufacture more energy efficient refrigerators. For example, the aforementioned refrigerator with the energy consumption label of 810 kWh/y (with a predicted value of 813 kWh/y) would have an energy consumption rating of 740 kWh/y if the wall thickness in the freezer and cooler is increased by 10 mm. Additionally, if a better door gasket which

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100-150 L while the highest efficiency was for a refrigerator with a 450-500 L volume.

Table 3: Comparison of actual and predicted refrigerator energy consumption (kWh/y)

Fridge ID	Size (L)	Exp.[a] (kWh/y)	Comp.[b] (kWh/y)	Error (%)	Comp.[c] (kWh/y)	% drop in energy rating
C170T	172	504	549.4	-9.0	479.7	12.6
RE221	219	620	634.03	-2.3	532.5	16.0
C2408[d]	246	642	816.6	-27.2	730.8	10.5
RE351	345	680	663.8	2.4	556.7	16.1
RE311	306	700	675.4	3.5	574.4	15.0
C420T	412	774	834.8	-7.9	706.6	15.4
C410B[d]	408	821	928.9	-13.1	820.2	11.7
RE441	436	960	736.8	23.3	630.5	14.4

a Manufacturer's data (experimental).

b Predictions under AS 2575.2 (computational).

c Predictions under ISO 7371-1985(computational).

d Freezer below cooler.

has a thermal resistance of 0.6 m^2-K/W (as compared to 0.43 m^2-K/W which has been used in the above calculations) is specified, the energy label would drop to 726 kWh/y. The first modification results in a predicted energy improvement of 9% when combined with the better door gasket would increase to 13%. Therefore, if a unit with a calculated energy label of 726 kWh/y is desired, by increasing the thermal resistance of the wall and the doors' gaskets in accordance with the above calculations, such a refrigerator can be manufactured without a great deal of laboratory experimentation.

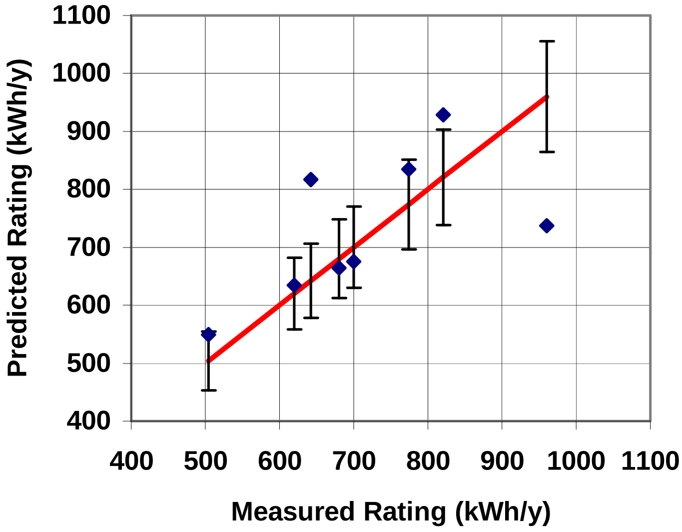

Figure 6: A comparison of energy consumption ratings between manufacturers data and those found by the computer simulation of this study.

Furthermore, the present computer code can be used to find the energy label for refrigerators which are manufactured for overseas markets, or for refrigerators manufactured overseas for domestic markets. In both cases, the savings in time and resources are significant. For example, the Australian refrigerators listed in Table 3 can be expected to have the energy ratings shown in the last column of this table if tested under the European standard ISO 7371-1985[5]. The reduction in the predicted energy consumption ranges from 10-16% which is a significant margin if Australian manufacturers sell their products to the European market. Of course, energy rating is one of many criteria to be considered if Australian manufacturers wish to introduce a new product in a non-traditional market.

4.0 Conclusions

Energy efficiency of refrigerators can be studied using basic heat transfer principles as described in this paper with reference to a PC-based computer simulation package. Using a simplified heat transfer model to calculate the heat flow through a three-layer composite wall, the code is shown to be a useful tool in predicting the energy consumption of domestic refrigerators with a reasonable degree of accuracy. Also, it is shown that the code can be used to design more energy efficient refrigerator cabinets at a high speed and a low cost.

The improvement in energy efficiency of domestic refrigerators when adopted globally can amount to a significant saving in the available energy sources, and also in protecting the environment by reducing the emissions of greenhouse gases. For example, in Australia, 99.6% of families own a refrigerator. Assuming that there are four persons

[5] The European standard specifications are: 25 ^{o}C , -18 ^{o}C (3 star freezer) and 5 ^{o}C for ambient, freezer and fresh-food compartment temperatures.

with one refrigerator per family, around 1.1 million domestic refrigerators are in use in Victoria (Australia) alone. Knowing that the energy rating of a family size refrigerator is about 840 kWh/y, the lower 11% energy saving in domestic refrigerators as a result of the additional 30-mm insulation would cause an energy saving of $102x10^6$ kWh/y in Victoria. Since the Victorian electricity is produced mostly from brown coal, around $133x10^6$ kg of coal will be saved annually and the same amount of CO_2 will not be emitted to the atmosphere. It is, therefore, easy to extrapolate that the energy saving in Australia will be of the order of $500x10^6$ kWh/y with a reduction in CO_2 emission of the same order of magnitude[6]. This total energy saving is equivalent to about half of the annual electric energy generated by Alcoa at Anglesea Power Station (Victoria, Australia) which has a capacity of $1{,}232x10^6$ kWh/y [29].

Although it may appear that a refrigerator with a 100-mm wall thickness will not be readily sold when offered in the market, it should be noted that this thickness is based on the conventional polyurethane-based insulating material. Upon the commercialisation of new and better insulations as a result of the research work of various institutions, the wall thickness will no longer be a limitation.

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[6] Calculations are based on the published data from SECV [29] indicating that 1.3 kg of brown coal generates 1 kWh of electricity and 1.3 kg of CO_2, and that Victoria's population is 4.4 million people as compared to 17.3 million for Australia.

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