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Effects of gasket heat gain and an alternative refrigerant on refrigerator/freezer performance

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Effects of gasket heat gain and an alternative refrigerant on refrigerator/freezer performance

by

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In the name God the gracious and the merciful.

Frist and foremost, I thank Almighty God *who is all knowing, all embracing.* Praise be to him for guiding me and giving me the strength and wisdom to accomplish this goal. He has filled my life with goodness, grace, joy, and mercy, the sum of which I cannot contain. May his name be forever parised.

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CHAPTER 1. INTRODUCTION

Background

Refrigerator/freezer production is a three billion dollar industry in the United States and is growing at about 2% annually [1]. Home refrigerators are the largest consumers of electricity among household appliances, consuming an estimated 7% of the total electricity used in the United States [2, 3].

The energy consumption of domestic refrigerator/freezers contributes to increased emissions of carbon dioxide by fossil fuel power plants. Carbon dioxide is suspected to be a major contributor to the green house effect or global warming phenomena. To mitigate fossil fuel demands and the environmental impacts, the manufacturers have been faced with progressively tougher energy-efficiency standards. The National Appliance Energy Conservation Act (NAECA)[4] was enacted in 1987 and went into effect January 1, 1990. The NAECA established energy efficiency standards for several consumer appliances including refrigerator/freezers. Also in 1990, the Department of Energy (DOE) established energy standards for the 1993 model year that require a 25% greater efficiency over 1990 levels [5].

Since 1978, several studies have been conducted to improve the efficiency of refrigerator/freezers. To achieve this goal, various designs were developed and tested. A variety of improvements were suggested, including compressor modification, better insulation, adaptive defrost, etc. Gasket improvements appeared in most of the studies as an option.

A portion of heat gain to a refrigerators/freezer occurs around the edges of the doors, through nearby portions of the cabinet surfàce, and through the door gaskets themselves. The gaskets in a refrigerator/freezer act as seals to contain the cold air and to thermally isolate the plastic liner from the outer steel structure. Figure 1.1 shows a typical refrigerator/freezer door gasket configuration. As shown, the gasket sits between the cabinet insulation and the door. The door gasket itself has trapped air *bubbles* which have low thermal conductivity and a piece of magnet used for door closing.

Recent studies show that the gasket area heat gain may account for as much as 21% of the total thermal load [6]. Some infiltration also occurs since the door seal cannot be perfect. There is not a unanimous agreement among manufacturers and literature sources regarding total gasket gain (gasket infiltration and heat gain) in a typical home refrigerator/freezer. This lead the present research to question the precise magnitude of the total gasket heat gain. Another source of energy consumption is the anti-sweat heaters placed near the gasket to eliminate condensation. Minimizing gasket heat gain in a refrigerator/freezer reduces the need for anti-sweat heaters and lowers energy consumption. In the near term, higher energy efficiency standards are providing considerable impetus to reduce gasket heat gain.

Another factor in reducing the energy consumption of refrigerator/freezer has been the use of chloroflurocarbon-12 (CFC-12) [7]. CFC-12 has been used in home refrigerator/freezers due to its favorable characteristics such as non-flammability, low toxicity and non aggressive behavior with other materials. A house hold refrigera-

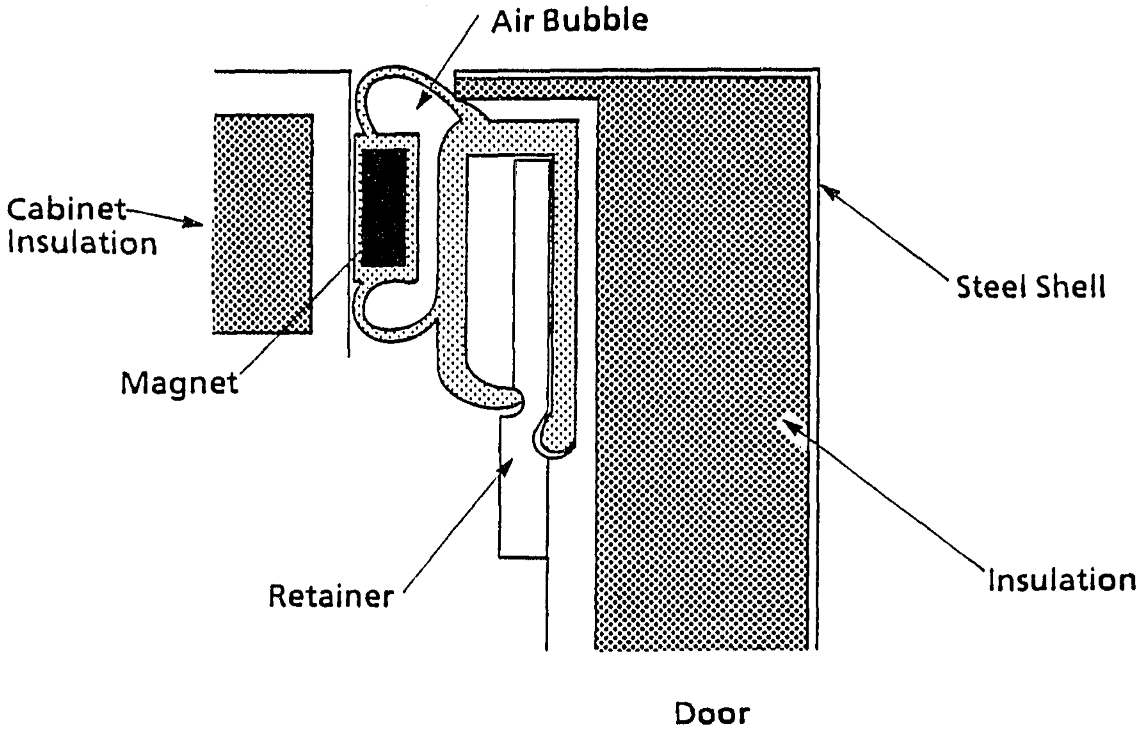


Figure 1.1: Typical door gasket configuration

tor/freezer unit typically uses 8 to 14 oz of CFC-12 in a vapor compression system [8].

It is well-known that CFCs slowly migrate into the stratosphere where they decompose by action of sunlight and split off free chlorine molecules that react with ozone in the upper atmosphere [9]. As a result, CFCs are suspected of playing a role as a major contributor to the ozone depletion that has occurred in the stratosphere. At the same time that the industries are confronted with ever tightening energy efficiency standards, they are also faced with CFC phase out.

In 1987 the Montreal Protocol was signed by 24 countries on substances that deplete the ozone layer [10]. The Montreal Protocol called for the CFC-producing nations to freeze their production of CFCs at their 1986 levels beginning in July of 1989, and then to reduce their production by 20% in 1993 and to 50% of 1986 levels in 1998.

Extensive search has been conducted during the past years to replace R-12 with a new refrigerant that does not contain CFCs. This search is being pursued to select an environmentally acceptable refrigerant that has minimal effect on energy consumption of refrigerator/freezers. Thus, researchers have come up with several different potential replacements for CFC-12 in refrigerator/freezers, such as HFC-134, HFC-134a, HCFC-22, HFC-152a. Several blends have also been considered.

At present HFC-134a is considered the most likely replacement for CFC-12 in house hold refrigerator/freezers. It has thermodynamic properties similar to CFC-12, and has the advantage that it contains no chlorine at all. Hence, it has a zero ozone depletion potential. One of the major drawbacks to using HFC-134a has been that it is immiscible with any of the commonly used refrigeration oils. Also, some studies indicate a slightly higher energy consumption for HFC-134a as a drop-in replacement for CFC-12. However, the question remains whether minor modifications to the refrigerator/freezer system using HFC-134a will improve their energy consumption to acceptable levels. As a result of the NAECA and Montreal Protocol, refrigerator/freezer manufacturers and researchers are presently searching for more efficient refrigerator/freezer designs which utilize non-CFC refrigerants.

Objective

The objective of this research was to investigate, both experimentally and theoretically, questions regarding total gasket gain (gasket infiltration and heat gain) and the effects of HFC-134a refrigerant as an alternative to CFC-12 on energy consumption of refrigerator/freezers.

First, the significance of gasket infiltration and heat gain in a 20 ft^3 home refrigerator/freezer was explored. Then, this research experimentally investigated the effect of HFC-134a as a drop-in replacement for CFC-12 in home refrigerator/freezer. The goal was to evaluate the performance of the unit charged with HFC-134a and its compatible lubricant while no system modifications were considered. In addition to using HFC-134a as a drop-in replacement, this research investigated the effect of different HFC-134a charges and capillary tube lengths on the refrigerator/freezer performance.

This research presents the results of an extensive literature review, interviews with refrigerator/freezer and gasket manufacturers, experimental and theoretical evaluation of gasket infiltration and heat gain. Also included is a description of the experimental evaluation of HFC-134a as a drop-in replacement for CFC-12 combined with different capillary tube lengths and different HFC-134a charges. A detailed discussion of each of these is presented in the following chapters.

CHAPTER 2. REVIEW OF LITERATURE

Review of Gasket Literature

A refrigerator/freezer cabinet consists of two or more compartments, with at least one compartment designed for the refrigerated storage of fresh foods at temperatures above 32 ^{o}F and with at least one compartment designed for the storage of frozen foods at 8 ^{o}F or below. As mentioned, a portion of the heat gain to a refrigerators/freezer occurs around the edges of the doors, through nearby portions of the cabinet surface, and through the door gaskets themselves. The gaskets in a refrigerator/freezer act as seals to contain the cold air and to thermally isolate the plastic liner from the outer steel structure (See Fig. 1.1).

Gasket improvement was part of several studies that have been conducted to reduce energy consumption of refrigerator/freezers. Kammerer and Maxwell [11] explored means for reducing energy use in existing refrigerator/freezer designs. They indicated that gasket heat gain might account for as much as 19% of the total heat load. However, they didn't include gasket improvements among their recommended design improvements. Hoskings and Hirst [12] calculated the gasket loads for 12 and 16 ft^3 refrigerator/freezers. Although they did not include gasket improvements in their list of suggested design changes, their computer model simulated the open door condition and calculated the gasket load. The calculated heat loads for a 12 and a 16 ft^3 refrigerator-freezer are 13.54 and 15.16 watts (13% and 12% of the total thermal load), respectively. Two major studies by Arthur D. Little Company (ADL) company explored gasket improvements as design options for higher efficiency refrigerator/freezers. In the first ADL study [13] , improvements were made to the door closure area to reduce infiltration of room air into the refrigerator.

The second study by ADL consisted of two phases. Phase I, reported in [14] , involved the design, construction, and laboratory testing of a 16 ft^3 high efficiency refrigerator/freezer prototype. ADL reported a 47% reduction in freezer heat flow by incorporating a vinyl type secondary gasket into the freezer compartment of the base line unit (see Fig. 2.1). However, this reduced the overall energy consumption by only 3%. The ADL study also showed that only the double door gasket in the freezer effectively reduced the energy consumption. As shown in the ADL model of door closure area (Fig. 2.1), the additional door seal is placed between the door shelf and the wedge and cabinet wall of the refrigerator/freezer. According to the ADL, incorporating a double door gasket in the freezer compartment caused heat flow reduction as shown in table 2.1.

Table 2.1: ADL gasket heat flow ($\frac{Btu}{hr}$)

	Base Line Value	Double Gasket
Evaporator fan on	62.5	41.9
Evaporator fan off	43.5	28.8

Phase II, reported in [15] , consisted of a field test that was carried out for an identical setup with the exception of the size. An eighteen cubic foot refrigerator/freezer was selected for the second phase. In the Phase II study, double door gaskets on freezer doors were not considered due to the limitation existing with dou-

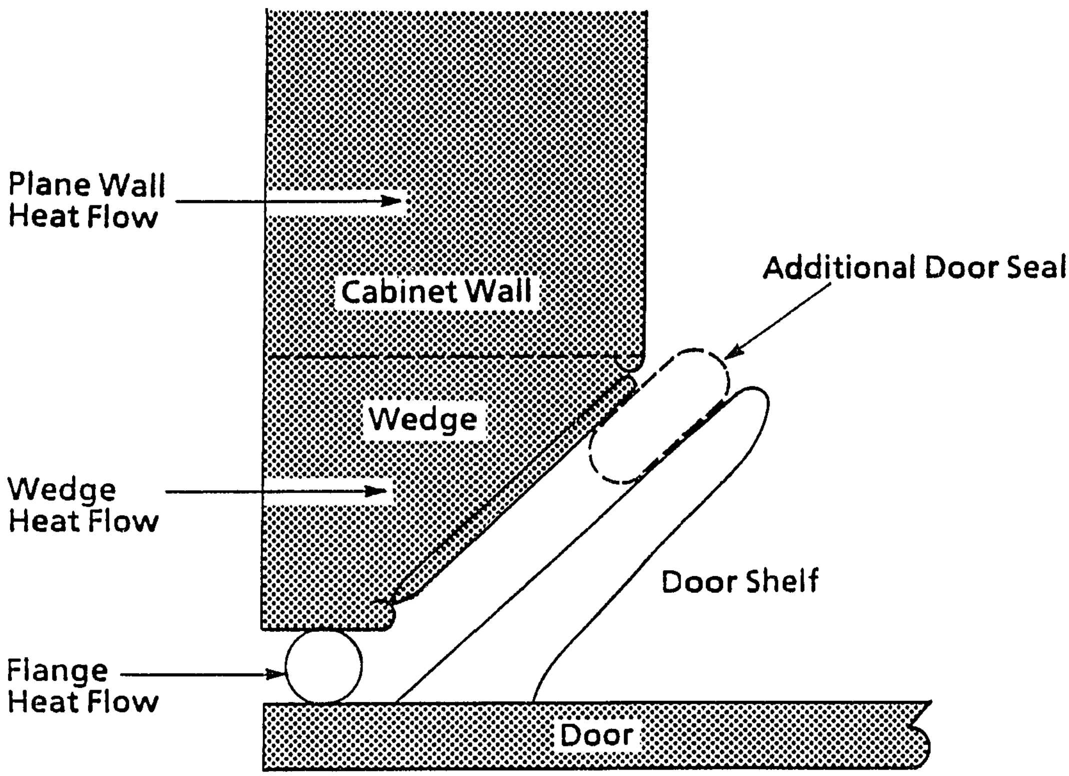


Figure 2.1: ADL model of door closure area

ble door gaskets associated with freezing of trapped moisture which can jam the door shut.

The results of the field tests as well as the data obtained from the Phase I study were published in four different reports. References [16] and [17] identified the results from Phase I, while references [18] and [19] highlighted the findings of the field test (Phase II) study.

Sterling [20] calculated the heat leakage through gaskets, using energy factor concepts. He determined increase in energy usage as the volume ratio (ratio of freezer volume to total volume) increases. Table 2.2 shows heat leakage through the gasket of a 15.6 ft^3 refrigerator/freezer as was calculated by Sterling.

Table 2.2: Sterling gasket heat flow

Volume Ratio	Freezer		Fresh Food		Total	
$\frac{freezer}{total}$	$\frac{Btu}{hr}$	Watts	$\frac{Btu}{hr}$	Watts	$\frac{Btu}{hr}$	Watts
.20	45.19	13.24	29.36	8.60	74.55	21.84
.30	56.50	16.55	28.12	8.24	84.62	24.79
1.0	121.40	35.57	-	-	121.40	35.59

Sterling's work confirms that heat leakage through the fresh food gasket area is significantly less than through the freezer gasket area.

Lawrence Berkeley Laboratory (LBL) also conducted research on home appliances in order to update the selection of design options [21]. The LBL study indicated that double door gaskets cause problems in the field due to freezing of trapped moisture. An improved single door gasket, which provided some of the double door gasket benefits without the indicated problems, was added to the list of new design options for higher efficiency refrigerator/freezers. However, the LBL study did not suggest any specific design improvements.

A study done by the Department of Energy (DOE) [22] did not include the double door gasket in the simulation analysis due to technical difficulties, but gasket improvement was among the design options suggested. These results were published in a paper by Turiel and Heydri [23]. The most recent study by Abrahamson, Turiel, and Heydari [6] indicated that about 21% of thermal load is due to gasket loss. They predicted that 5.9% of the fresh food load and 16.5% of the freezer load are due to gasket heat leakage.

In addition to the literature survey, the present study also involved contacting the major refrigerator/freezer manufacturers and gasket suppliers. Interviews with engineers indicated little agreement about the precise magnitude of gasket heat leakage. In addition, the definitions of the particular area associated with " gasket " heat leakage appeared to vary among manufacturers as well as among the other research studies discussed above. This may account for the apparent variation of between 5% and 30% of the energy consumption that different sources associate with gasket loads. Nevertheless, all manufacturers agree that improved gasket design to reduce heat leakage was a priority for helping to meet new energy standards, and as such was receiving considerable attention in their companies. In addition to gasket literature, extensive review of alternative refrigerant literature as conducted, and is presented in the following section.

Review of Literature on Alternative Refrigerants

The agreement of the Montreal Protocol to regulate the production and trade of CFC-12 has greatly influenced the refrigeration industries. At present, extensive research is being conducted at many institutions to replace the CFC-12. This search is being pursued to select a refrigerant that has minimal effect on energy consumption of refrigerator/freezers as well as on the environment. Thus researchers have come up with several different potential replacements for CFC-12 in refrigerator/freezers, such as HFC-134, HFC-134a, HCFC-22, HFC-152a. Additionally, several blends have been considered.

Fischer and Creswick [24] reported a quantitative assessment of the potential energy-use impacts of possible alternatives (mainly HCFC-123, HCFC-141b and HFC-134a) to CFCs for a variety of applications. The energy analysis was done for a domestic 18 ft^3 (.51 m^3) refrigerator/freezer, with 60% compressor and 80% motor efficiencies were assumed, respectively. In their study, the energy use of each alternative was evaluated on the basis of daily energy use per unit, and then compared with the energy use of the base. This study used the ADL model [14] to estimate daily energy use of the indicated unit.

Their analytical study showed a very small change in energy consumption of the refrigerator/freezer unit for HFC-134a. They project an increase of 0.08 quads/year in energy use nationwide as a result of the shift to HFC-134a.

Alternative refrigerants for CFC-12 in domestic refrigerator/freezer were studied by Boot [25]. In this study, Boot used a compressor calorimeter setup to evaluate the effect of alternative refrigerants (HFC-134a, HCFC-22/HCFC-142b, HCFC-22/HCFC124 and HCFC-22/HCFC-124/HFC-152a) on home refrigerator/freezers. In every case, the reciprocating compressors were tested with both the refrigerant alternative and CFC-12. His study showed that HFC-134a is less efficient than CFC-12, with about an 8% lower energy efficiency ratio (EER). His experiment also indicated about 4% lower EER using CFC-22/HCFC-142b and only about 1% reduction in

EER using HCFC-22/HCFC-124/HFC-152a.

Although in recent years many reports indicated increased energy consumption of refrigerator/freezer using HFC-134a, a study done by Hanson [26] indicates otherwise. He showed that there was no increase in energy consumption using HFC-134a in a domestic refrigerator/freezer if compressor and cooling circuits were modified. He used a standard CFC-12 compressor which was only modified to accommodate different torque requirements. The modification was made to produce the torque required for HFC-134a using lubricant PAG with a viscosity equal to that of mineral oil used in CFC-12 compressors at $104^{o}F$.

Household refrigerator/freezers were included in a study done by ADL which was summarized by Statt [27]. This study discussed the effects of alternative refrigerants on air conditioning, as well as refrigeration systems. For home refrigerator/freezers, this study considered different refrigerants (HCFC-22, HFC-134a, HFC-152a, and azeotropes blend) and showed 5 to 10% increase in energy consumption for HFC-134a, and HCFC-22 and 5-10% decrease in energy usage for the rest of them.

Vineyard [28] selected six different refrigerants, namely HFC-134a, HFC-134, HFC-152a, HFC-134a/HFC-152a, HCFC-22/HFC-152a/HCFC-124, and HFC-134a/HFC-152a/HCFC-124. These refrigerants were considered the most likely to replace CFC-12. He tested four pure refrigerants, including CFC-12, in an 18 ft^3 (.51 m^3) automatic-defrost top-mount refrigerator/freezer. A capillary tube manifold (a set of three capillaries) was installed on the base line unit in order to control the refrigerant flow as needed. Also, with each different refrigerant he used a new compressor that was sized to reflect changes in volumetric capacity of the corresponding refrigerant. The final results indicated that with HFC-134, HFC-134a, and HFC-152a, the energy consumption increased by 6.8%, 7.3%, and 7.3% respectively.

In earlier work, Vineyard, Sand, and Miller, [29] tested five different refrigerants (CFC-12, HFC-134a, HCFC-500, CFC-12/dimethylether(DME), and HCFC-22/HCFC-142b) for their energy consumption in an unmodified 18 ft^3, top-mount domestic refrigerator/freezer utilizing the identical lubricant. Their unmodified test unit consumed 7.8%, and 8.6% more energy for HFC-134a and HCFC-22/HCFC-142b compared to CFC-12 respectively. Also they showed 5.8% and 6.6% reduction in energy consumption for HCFC-500 and CFC-12/DME, respectively.

Jung and Radermacher [30] numerically investigated the effect of pure and mixed refrigerants as alternatives to CFC-12 in an unmodified domestic refrigerator/freezer. A computer simulation was performed on fifteen pure and twenty-one mixed refrigerants, and the results showed that none of the pure refrigerants could be used as drop-in substitutes for CFC-12. This was attributed to the change in volumetric capacity of the alternative refrigerants. Also, no significant increase in COP of the unit was reported for 21 mixed refrigerants by this simulation. The same finding was identified in reference [31].

CHAPTER 3. ANALYSIS OF GASKET HEAT GAIN

There are many design options available for producing improved and more efficient home refrigerator/freezers. Examples include compressor modifications, better insulation, adaptive defrost, and others some of the possible design improvements in existing home refrigerator/freezers are associated with reducing the gasket heat gain.

The current chapter presents the theoretical analysis of double and single door gaskets as well as possible design improvements. It also presents the analytical evaluation of gasket infiltration and heat leakage.

Double Door Gasket

According to literature and experts in the field, gasket heat gain appears to account for at least 10% of the thermal load of refrigerator/freezers. One concept for reducing the gasket loads is to insert an additional inner door gasket (see Fig. 2.1). This improves the insulating value of the gasket area and reduces energy consumption.

Despite the possible energy benefits, double door gaskets haven't been used by many manufacturers because of performance and cost. The limitations existing with double-door gaskets include the following:

1. Ice has a tendency to form between the freezer compartment gaskets due to trapped moisture. The ice greatly reduces the thermal effectiveness and can

freeze the door shut.

2. Inner seal problems exist due to requirements for special gasket materials. The materials developed must be highly compliant and yet durable to serve as a good inner seal held by the force of the magnetic outer gasket.
3. Double door gaskets tend to be visually unattractive.
4. Difficulties can exist with ease of door closing, which can detract from consumer acceptance.
5. Double door gaskets can make it more difficult to meet the minimum door opening force requirements of the Child Safety Act.

Single Door Gasket

Due to the limitations of double door gasket indicated earlier, refrigerator/freezer manufacturers and their gasket suppliers have focused their efforts on producing thermally improved single gaskets with higher insulating values and better sealing characteristics. The improvements to single door gaskets make the double door gasket concept of energy saving less important. According to [21] , heat gain by an improved single door gasket is 10% less than the 1982 ADL-proposed gasket [14].

The following discussions detail some of the considerations concerning materials, design evolution, and possible gasket design improvements.

Materials

Door gaskets are usually made of flexible plastic. The most common plastic materials used in gaskets are: thermoplastic elastometer (TPE) and polyvinylchloride (PVC). These materials range from the very soft and delicate to rigid and wear-resistant, and their suitability is mainly related to such considerations. The primary thermal barrier in the gasket is trapped air *bubbles*, which have low thermal conductivity. The materials themselves do not contribute significantly to the thermal resistance. The present study indicates that little improvement in thermal performance is possible in the area of gasket materials.

Design Evolution

Early designs for extruded single door gaskets depended upon mechanical compression provided by a latch mechanism to seal (Fig. 3.1a). While still suited to some applications, the compression design was improved dramatically by a development called *supported compression* (Fig. 3.1b). The next major design improvement was done by inserting magnetized extrusions of ferrite compounds for sealing (Fig. 3.1c). The magnets are used in place of latch and striker plate. This improvement resulted in consumer satisfaction and improved safety.

Remaining improvements in gasket design involved improving the thermal resistance. The next step was the *extended bubble magnetic* design (Fig. 3.1d). In addition to compression and magnetic attraction, this design introduced the *wand* which extended from the inner edge of the bubble. Currently the most efficient gasket is the *multiple bubble magnetic* design (Fig. 3.1e), where the extra bubble acts as an insulator, reducing heat leakage. As shown by Figure 3.1f the newest gasket design with additional air pockets incorporated in the gasket and retainer area. These refinements will assist in reducing the heat leakage, and therefore improve the energy efficiency by an amount yet to be identified. Due to the expected enhanced performance, in

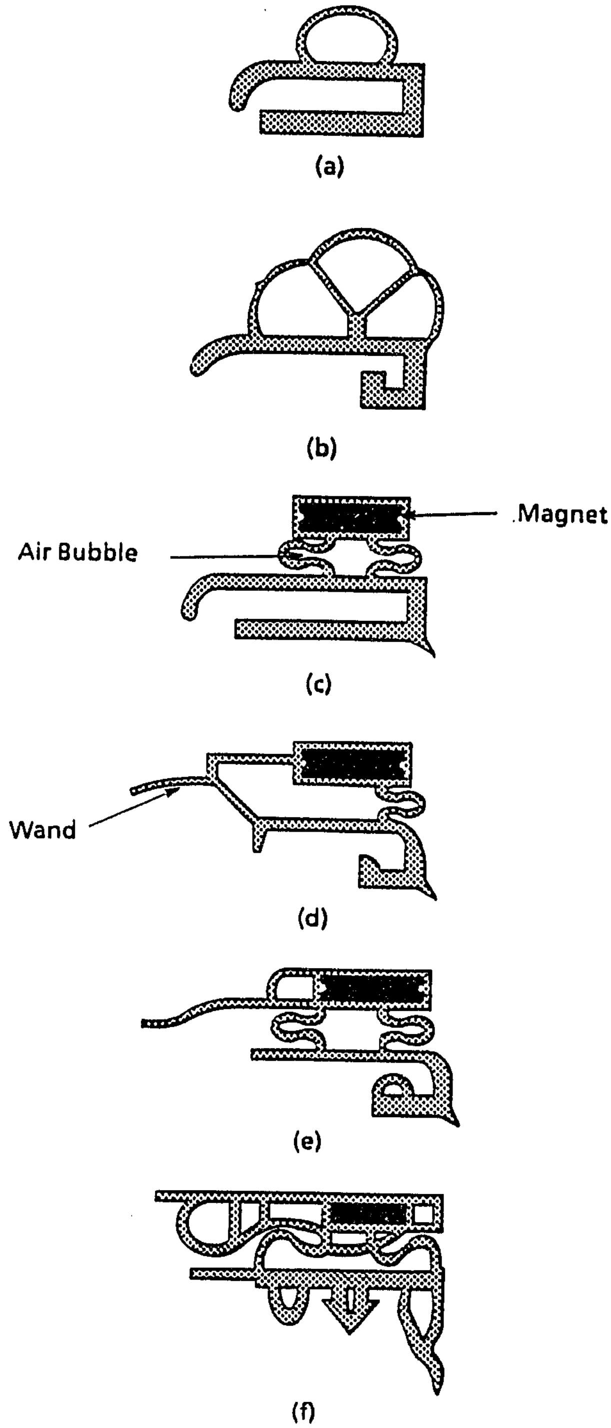


Figure 3.1: Gasket evolution

1993 the manufacturers of refrigerators/freezers will most likely standardize on this type of improved gasket.

Other Possible Design Improvements

Other possible areas of design improvement were identified in the course of the present study. These improvements can be divided into two separate categories:

- Reduction of the gap between the gasket and body
- Further increase in thermal resistance in the gasket area

It must be noted that some of the following concepts are already being incorporated and/or designed into existing products.

Possible areas of improvement include:

1. The use of a half-bellows design (Fig. 3.1d), which eliminates alignment problems and turn over on the hinge side. In general, the bellows design provides the ability to expand or collapse and influences stability for maximum sealing effectiveness. To achieve maximum effectiveness, one manufacturer recommends a bellows thickness of 0.17 inch.

2. The use of ribs as flow diverters within the compartments is quite common (Fig. 3.2). Fig. 3.2 shows a typical refrigerator/freezer with a flow diverter place near the freezer door to direct the cold air away from the gasket area. This helps to reduce heat gain by conduction through the gasket and to reduce infiltration. The primary design challenge is to adequately distribute air throughout the compartment while reducing the impingement of air directly on the gasket. No

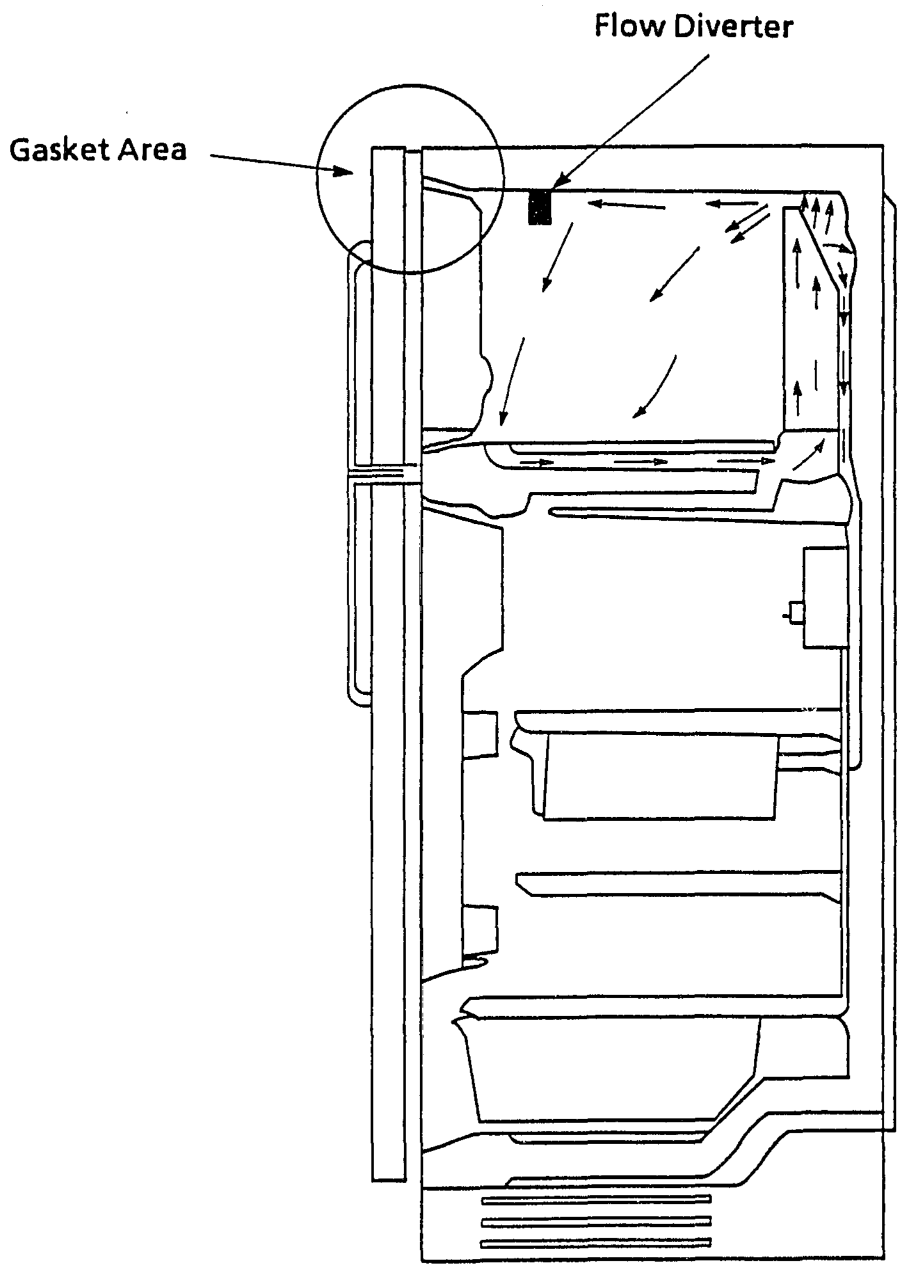


Figure 3.2: Schematic of flow diverter

quantitative estimate has been made of the impact of this practice on energy consumption.

3. The addition of one air pocket (different sizes and shapes) to each side of retainer (Fig. 3.1f) can also be beneficial. This provides better retaining of gasket in the body, and still allows for easy sealing. Additional air pockets increase the thermal resistance of the gasket and thereby reduce heat gain. However, the potential for this type of improvement is limited by the need for flexibility. Gaskets must typically collapse or expand from about 0.65 inch to about 1.0 inch.

4. Another suggested improvement is to fill some of the air pockets with insulating materials such as fiberglass or foam. However, adding these materials would reduce the flexibility of the gasket, and would therefore be unacceptable. Further, a trapped bubble of stagnant air is one of the best insulating mediums available, and it is doubtful that any improvement in thermal performance would be realized by filling the pockets with solid materials.

5. Mechanical door latching can provide better sealing than magnetic latches due to the increased pressure exerted by the door on the gasket. Potential improvements due to mechanical door latching are difficult to quantify. One effect would be to reduce infiltration, but the magnitude of this potential improvement is unknown. Another effect could be negative: to collapse the air bubbles that provide insulating value. Even if a mechanical latching door could be designed to reduce energy use and to meet the existing safety requirements (which are described later), this feature still may not be suitable. Present practice dic-

tates the availability of units with interchangeable right or left operation of the door, as desired by the customer. Thus, a universal reversible hinge design is commonly used to avoid having to market different models strictly due to door operation. The latching mechanism lacks this universality and would be used only as a last resort.

According to the Consumer Product Safety Commission (CPSC) [32] and Underwriters Laboratories Inc. (UL) [33], door-latching devices must follow standard rules and regulations. A door-latching device is a device that holds the door shut. A magnetic door gasket is considered a door-latching device for the purpose of these standards. Listed below are some of the requirements for latching devices; however, references [32] and [33] provide more detailed descriptions of the standards. Some of these requirements are:

(a) The door can be opened from a totally closed position from the interior.

(b) The opening device is accessible from anywhere in the interior.

(c) The device can be the application of an outward force from the interior.

(d) The applied force must not exceed 15 lb_f (66.7 N) directed perpendicularly to plane of the door anywhere along the latch edge of the inside of the closed door.

(e) A latch-release device must not depend on any electrical source for its operation.

(f) Latch-release device performance must be unaffected by spillage, cleaning, defrosting, and condensation.

(g) The device must satisfy wear and strength tests.

These regulations govern any changes that would be made to the door closure which are intended to improve energy conservation. It is unlikely that a mechanical door latching device will return to the market place, despite energy considerations.

6. Another potential area for reducing heat leakage due to door sealing is to design interior compartments that are separate from one another and that each have their own doors. This might reduce infiltration and provide greater thermal resistance between the coldest air and the outside of the cabinet. However, consumer acceptance and cost are likely to be barriers to the use of this concept. Also, the ADL study [14] indicated that the expected savings with internal doors would be comparable to the savings obtained using double door gaskets, which is a much simpler and less costly alternative. The concept of internal doors is not expected to be seriously pursued in the future.

Gasket Infiltration

Some of the thermal load on refrigerator-freezers may be due to gasket infiltration. Infiltration is the uncontrolled leakage of air into the refrigerator-freezer through the door gasket. This is caused by a pressure difference across the boundary surface, and it accounts for some of the thermal load. After several conversations with different manufacturers and reviewing the literature, it is evident that there is no unanimity in the importance of infiltration. In fact, some literature contradicted the views of experts in the field. Infiltration of air was considered insignificant according to the ADL study (about 5 $\frac{Btu}{hr}$); however, some manufacturers indicated as much as $100\frac{Btu}{hr}$ heat gain due to the infiltration effect.

Because of the apparent uncertainty about the importance of infiltration, an attempt was made to model the heat gain due to infiltration. The following characteristic values for a typical $20ft^3$ top mount, refrigerator/freezer were used:

Room temperature, $T_o = 90\ ^oF$

Room humidity, $\omega_o = 0.031\ \frac{lb}{lb}$ (100% relative humidity)

Specific heat of air, $C_p = 0.24\ \frac{Btu}{lb.^oF}$

Specific volume of room air, $V_o = 13.986\ \frac{ft^3}{lb}$

Freezer compartment temperature, $T_{in} = 5\ ^oF$

Inner humidity, $\omega_{in} = 0.0004$ (10% relative humidity)

Enthalpy of vaporization, $i_{fg} = 1042.7\ \frac{Btu}{lb}$

A brief engineering analysis follows:

The sensible heat load due to infiltration, q_{sens}, can be expressed in terms of the infiltration rate, Q, as follows:

$$q_{sens} = \frac{Q.C_p.(T_o - T_{in})}{V_o} \tag{3.1}$$

Further, the latent load, q_{lat}, can be expressed as

$$q_{lat} = \frac{Q}{V_o}(\omega_o - \omega_{in})i_{fg} \tag{3.2}$$

Infiltration loads can be estimated using equations 3.1 and 3.2 for any given infiltration rate. The infiltration rate is dependent upon the pressure differential that exists between the inside and the outside of the refrigerator/freezer box. Because of frictional pressure drop through the internal ducting, slight negative and positive gauge pressures will exist between the suction and the discharge sides of the fan, respectively. The infiltration rate is also dependent upon the nature of the crack due to the gasket seal and any penetrations of the liner or duct work. An estimate of this relationship can be obtained using the following expression which is based upon data for a tight-fitting door in reference [34].

$$\frac{Q}{L} = K\Delta p^{0.64} \tag{3.3}$$

where L is the effective crack length (one half the total gasket length for both doors) in feet, Δp is the pressure differential in inches of water, infiltration rate, Q, is in cubic feet per minute, and K is a unit conversion constant.

Figure 3.3 shows the sensible, latent, and total loads as functions of pressure difference, for a total gasket length of 20.17 feet as used for a typical $20ft^3$ refrigerator/freezers. The curves show that the magnitude of the load due to infiltration may be substantial or may be negligible compared to other loads, depending on the pressure difference. Based upon discussion with manufacturers, 0.01 inches of water was selected as characteristic of the magnitude of pressure differential. With this value, the loads as determined from equations 3.1, 3.2 and 3.3 would be

$$q_{sens} = 43.3\frac{Btu}{hr}$$

$$q_{lat} = 72.4\frac{Btu}{hr}$$

$$q_{tot} = 115.8\frac{Btu}{hr}$$

From the calculations presented here and from the literature cited, there exists considerable uncertainty as to the magnitude of the infiltration effects. Although companies most likely have proprietary information, no actual data was found in the open literature.

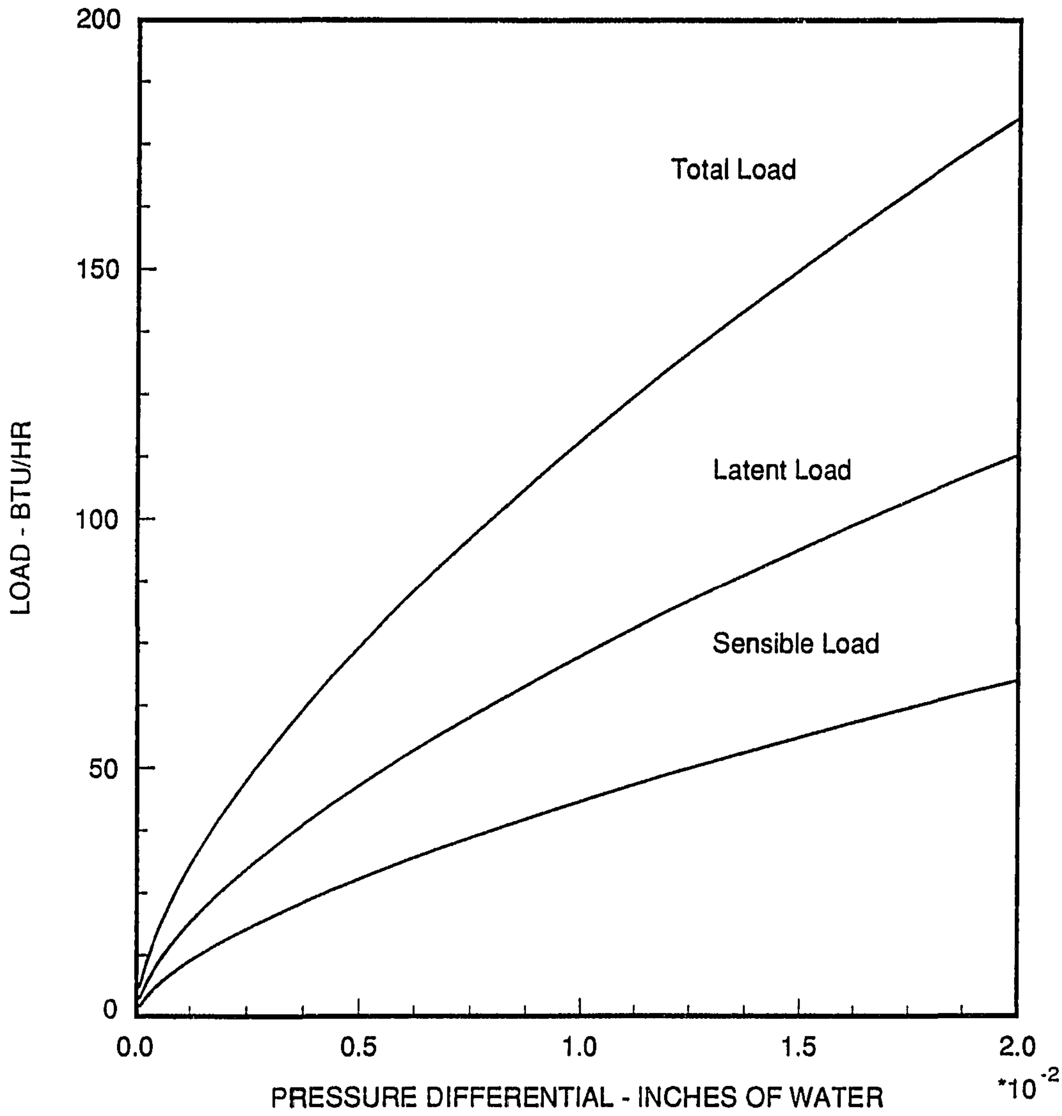


Figure 3.3: Load estimate due to infiltration

Gasket Heat Leakage

Gasket heat leakage, not including infiltration, is estimated analytically using two separate methods. Analytical calculation of the total gasket heat leakage is a combination of the following components (refer to Fig. 2.1):

- conduction along the flange
- heat leakage through the small gap between the gasket and wedges
- heat leakage through the gasket itself
- heat leakage between the gasket and door

Total heat load due to the gasket is calculated in various ways in the literature. Two methods of determining this load are:

METHOD 1 :

$$q_{gasket} = H_g l \Delta t \tag{3.4}$$

where

Δt = temperature difference between cabinet interior and ambient

l = total gasket length

H_g = gasket heat leak coefficient

Gasket heat leak coefficients can be found in reference [22], and are as follows:

$$Freezer - Fan\ on = 0.0069 \frac{Btu}{hr.in.^oF}$$

$$Freezer - Fan\ off = 0.00041 \frac{Btu}{hr.in.^oF}$$

$$Refrigerator = 0.00141 \frac{Btu}{hr.in.^oF}$$

METHOD 2 :

$$q_{gasket} = (L_r.\Delta T_r + L_f.\Delta T_f)(\alpha + \beta.f) \tag{3.5}$$

ΔT_r = temperature different between ambient and fresh food compartment (^oF)

ΔT_f = temperature different between ambient and freezer compartment (^oF)

L_r = length of fresh food gasket (door perimeter, ft)

L_f = length of freezer gasket (door perimeter,ft)

$\alpha = 0.05 \ \frac{Btu}{hr.ft.^oF}$ (static-fan off)

$\beta = 0.036 \ \frac{Btu}{hr.ft.^oF}$ (dynamic-fan on)

f = fan run time fraction

Using Equations 3.4 and 3.5 and using the data from the infiltration calculation presented earlier yields the following results:

METHOD 1: q = 70.67 $\frac{Btu}{hr}$ (fan on)

METHOD 2: q = 82.86 $\frac{Btu}{hr}$ (43% fan run time fraction)

From literature surveys, manufacturers input, and the engineering analysis mentioned earlier, there exists considerable uncertainty as to the magnitude of gasket infiltration and heat leakage. In light of these findings and the fact that little actual data are available in open literature concerning gasket heat gain, experimental measurements were conducted of infiltration and heat leakage through the door gasket. The details of the experimental investigation are presented in next chapter.

CHAPTER 4. EXPERIMENTAL EVALUATION OF GASKET HEAT GAIN

An experimental evaluation of gasket heat gain was conducted and is presented in this chapter. All experiments were conducted in a calorometic room in the Building Energy Utilization Laboratory (BEUL) of the Mechanical Engineering Department. This chapter includes a detailed description of the instruments used, a brief discussion of the experimental test procedures, and discussion of the results obtained.

Test Setup

Equipment and Instrumentation

The experimental setup for this research is illustrated in Fig 4.1. It consisted of a 20 ft^3 top mount, automatic defrost, refrigerator/freezer which is commercially available. The refrigerator-freezer was instrumented so that total energy consumption, average power usage, differential pressure between the inside and outside, pressure drop across the capillary tube, and the inside and outside temperature could be measured via a data acquisition system.

The following elaborates on the details of the instruments and equipment used in the experimental setup.

1. Thermocouples: The temperatures in the freezer and fresh food compart-

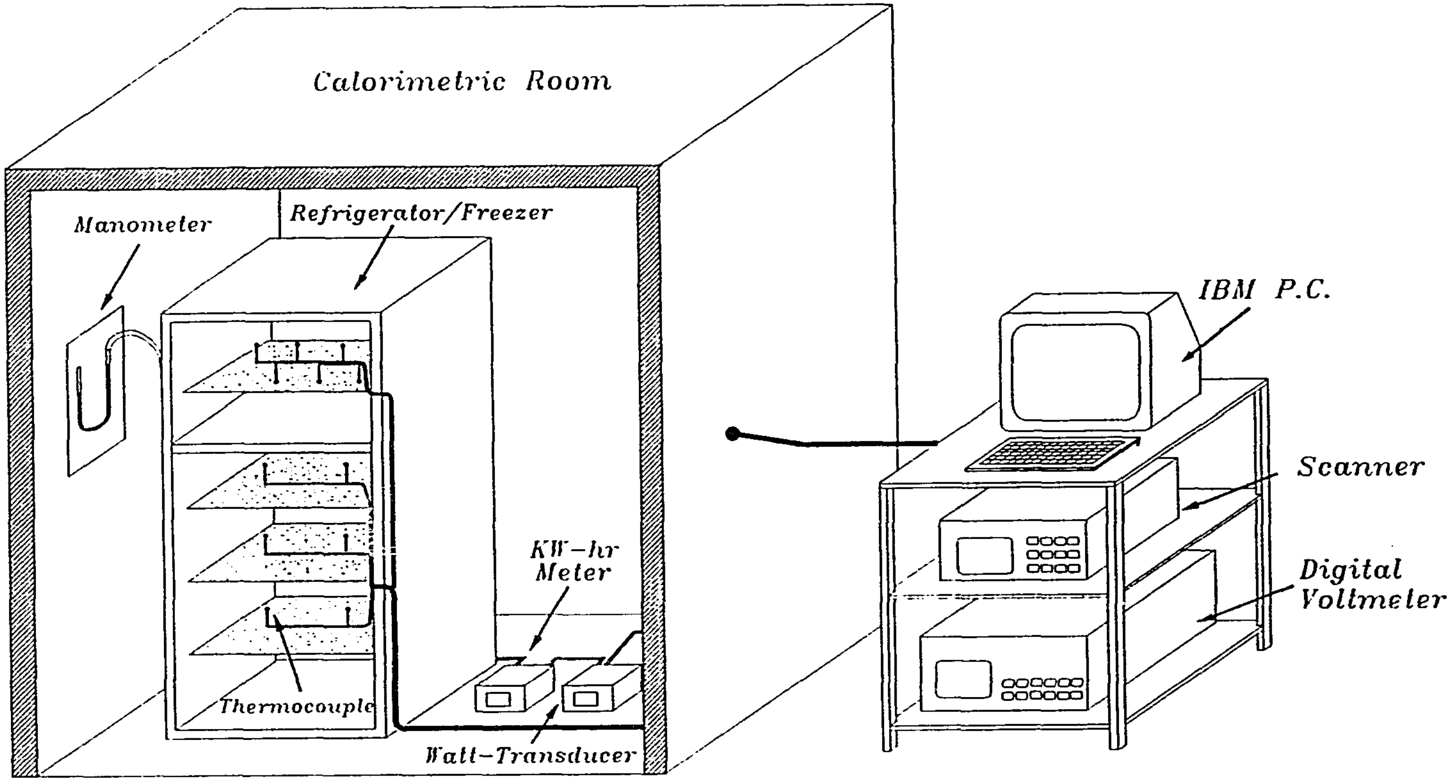


Figure 4.1: Experimental setup

ment of the unit were measured using a grid of 6 high quality T-type thermocouples (OMEGA, Catalog# PR-T-24, and Lot# T-0277) in each compartment. The thermocouples were calibrated using a hot water bath and a thermistor setup. These thermocouples were installed separately in each compartment in order to provide the average temperature for fresh food and freezer compartment during the test (see Fig. 4.1). The accuracy associated with the T-type thermocouples were estimated as $\pm .2^oF$.

2. Power transducer: The power into the compressor as well as the total energy consumption of the unit were measured using a watt transducer (Ohio Semitronics Inc., model# XL5C5A2, 1 phase, 2 wire, and 1 element) and a Kilowatt-Hour Meter (KW-HR Meter) (Westinghouse Type CS, 120 volts, and serial# 16434654). The watt transducer was calibrated by its manufacturer prior to its shipment and the accuracy was given as ±.2 % of full scale reading, while the error associated with a reading from the KW-HR Meter for a typical test was approximately ± 0.5 Watt.

3. Micro-manometer: To calculate the theoretical infiltration, a micro-manometer (Microtector, Dwyer Instruments Inc.) was utilized to manually measure the differential pressure across the refrigerator/freezer cabinet and its outside pressure.

4. Calorimetric room: In order to provide the desired outside temperature (T_O) and to maintain a constant outside temperature throughout the experiment, a calorimetric room was used. Outside temperature (T_O) was measured by averaging the four T-type thermocouples which were placed in four different locations inside the room. The baseline unit was placed inside the calorimetric room during the entire experiment so that where the surrounding environment could be carefully controlled.

Data Acquisition

All of the data taken from the experimental setup during a test run were recorded by a data acquisition system (see Fig 4.2). Figure 4.2 shows a schematic of the data acquisition system utilized in this research. This data acquisition system consisted of a computer controller (IBM PS/2) with a IEEE-488 bus, a 50 channel high speed scanner (Hewilt Packard, Model# 3488a), and a digital voltmeter (Hewilt Packard, Model# 3456A). A Quick Basic program was developed and utilized as the source code for the controller. This program scanned 16 channels (13 channels for temperature, 2 channels for pressure and one for power input into the compressor), read the voltage output and stored the data in a file. The controller, scanner, and the digital voltmeter communicated via an IEEE-488 bus. The data acquisition system automatically monitored the mentioned instrumentation (thermocouple, pressure transducer, etc) and collected data, respectively, upon achieving steady state conditions.

The voltage outputs of the thermocouples were converted to the corresponding temperatures using the vendor-supplied curve fit. This seven degree polynomial represents temperature as a function of voltage for T-type thermocouples. In addition, since the power transducer provided current output and the data acquisition system was only capable of accepting voltage input, a resistor (1 KΩ) was placed in series with the transducer, and the voltage drop across the resistor was recorded. The voltage output was then converted to power via the manufacturer supplied calibration information.

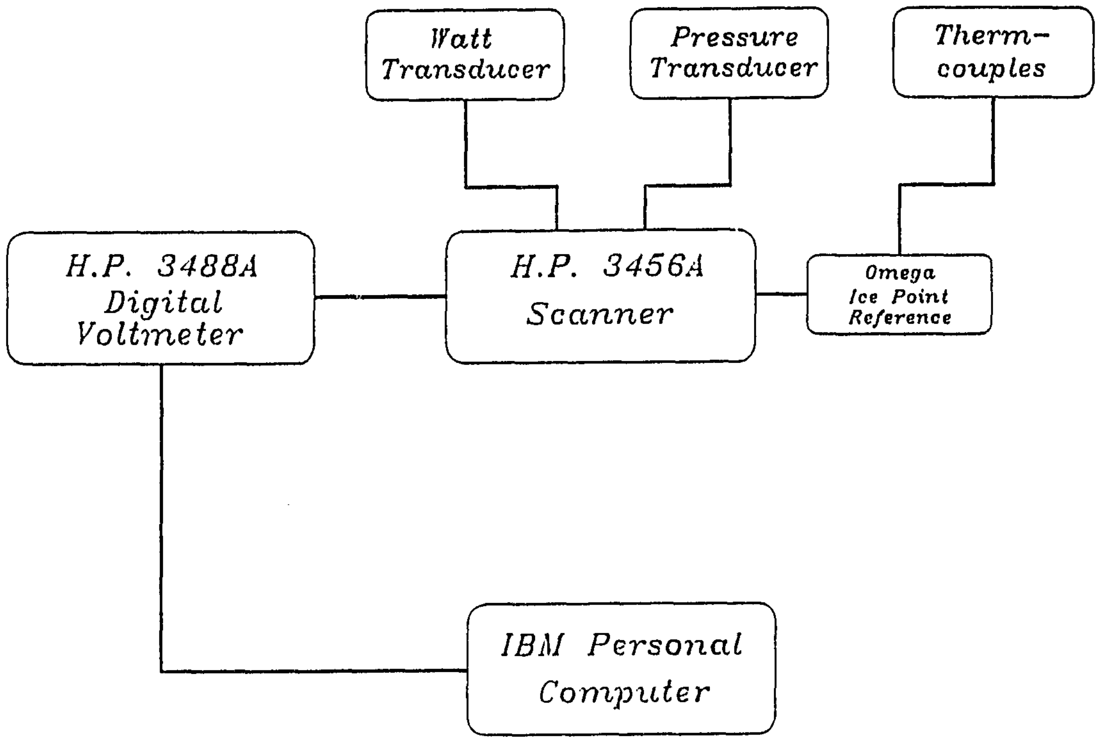


Figure 4.2: Data acquisition block diagram

Test Procedures

Three different test setups were utilized in order to measure infiltration and heat leakage through the gasket in a typical home refrigerator/freezer. These tests were:

Test A: baseline energy consumption

Test B: gasket infiltration effect

Test C: total gasket heat gain effect

The outside temperature variations effect the energy consumption of the refrigerator/freezer unit significantly. In order to determine the magnitude of this effect on energy consumption of the refrigerator/freezer, a series of tests were conducted and the results were plotted in Fig. 4.3. As shown by Fig. 4.3, the energy consumption increased with outside temperature. In all three tests, the outside temperature (T_O) was maintained at a constant level. In addition to a constant outside temperature, the thermostat of the refrigerator/freezer was set at 1/7 (maximum possible cooling for fresh food). This setting resulted in a measurement of $34.5 \pm .5^oF$ in the fresh food and 6.0 ± 0.2^oF in the freezer compartment throughout the experiments.

In order to obtain a reliable comparison between the individual tests (test A,B, and C), a 24 hour base for each test run was selected. Also, in order to insure accurate results, the same experiment was conducted 5 different times (sample size = 5) per each specific case (baseline, gasket infiltration, and total heat gain) and the results were averaged. Also, using Eq. 4.1 listed below, a statistical T-test about the mean ($\mu = \bar{X}$) was performed to determine the adequacy of a sample size of 5 tests per case.

$$\bar{X} \pm \frac{ts}{\sqrt{n}} \tag{4.1}$$

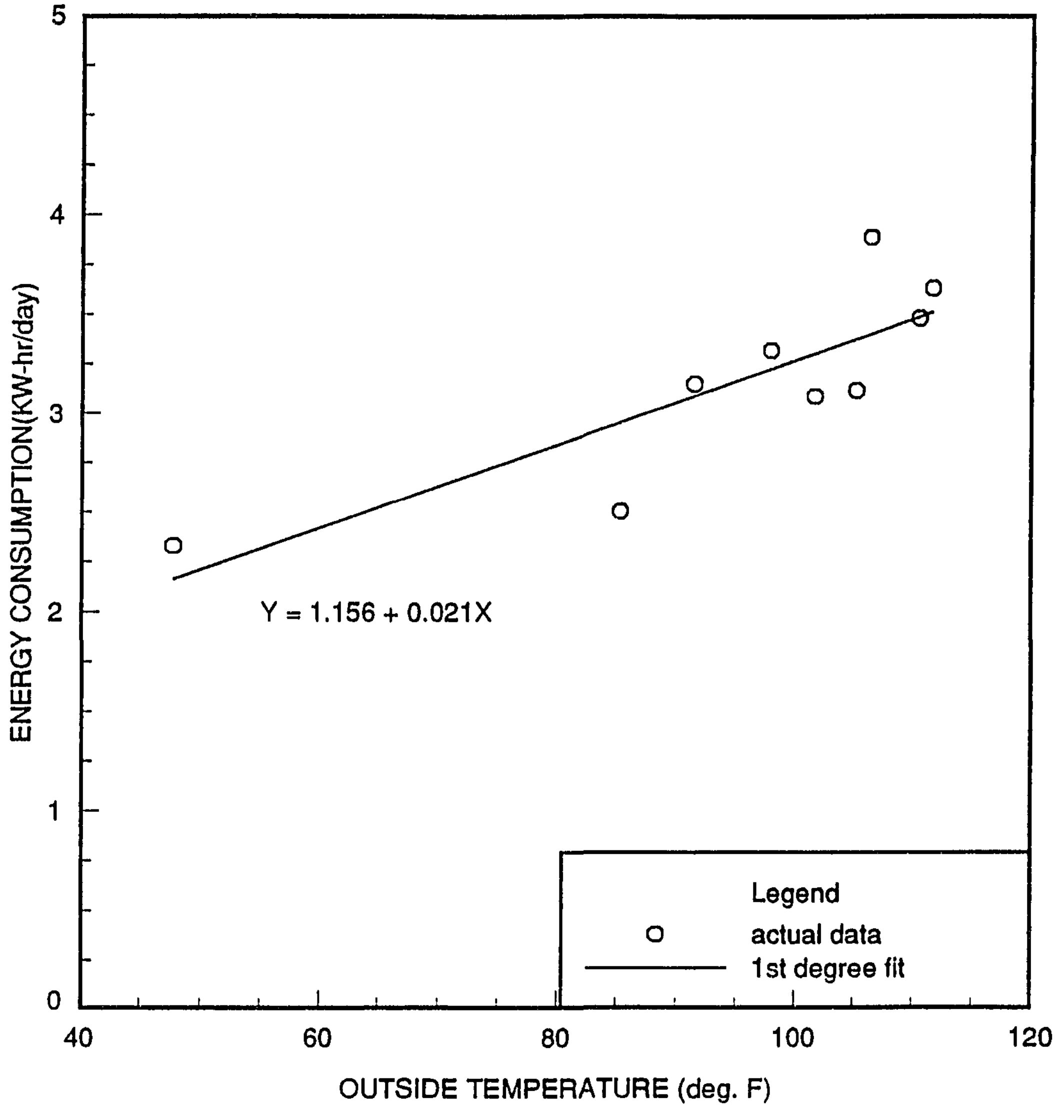


Figure 4.3: Effect of outside temperature on energy consumption

Where

$\bar{X}$ = average energy consumption

t = test statistics, t distribuition

s = sample standard deviation

n = sample size

The T-test (equation 4.1) was selected because the population standard deviation (σ) was unknown and the sample size was small ($n < 30$). In order to find the lower and upper limit of the mean with 99% confidence level, data from the baseline unit of test cycle one were utilized, as follows:

Average energy consumption, $\bar{X}$ = 2.502 KW

Sample standard deviation, s = 0.041

Confidence interval, α = 99%

Degrees of Freedom, ν = n-1 = 4

Critical value of t, $t_{4,.01}$ = 4.604

Substituting the above data into equation 4.1 gave the following lower and upper confidence limit

$$\text{lower limit} = \bar{X} - 0.084 \text{ KW}$$

$$\text{upper limit} = \bar{X} + 0.084 \text{ KW}$$

Therefore, the selected sample size (n=5) resulted in 99% confidence interval for the measured mean($\bar{X}$). Thus a sample size of 5 was selected for test cycles 1 and 2. This statistical method is presented in any of a number of statistics books, such as Statistical Design and Analysis of Engineering Experiments [35].

Gasket Infiltration

Experimental measurement of infiltration was done by comparing the total energy consumption of the baseline unit (test A) with the total energy consumption of the infiltration-free unit (baseline unit minus any infiltration effect, test B) both under the same conditions (constant outside temperature, 24 hour test cycle, etc.). Compressor run time was measured as well in order to provide the magnitude of the gasket infiltration effect on the refrigerator/freezer performance. As shown by equation 4.2, compressor run time is directly proportional to energy consumption.

First the total energy consumption and compressor run time of the baseline unit were measured. The energy consumption and compressor run time of the unit for one cycle and for an entire test were determined as follows:

for a data acquisition cycle i:

$$E_i = P_i \; RT_i \tag{4.2}$$

E_i = energy consumption of the unit per data acquisition cycle i (W-hr)

RT_i = compressor run time per data acquisition cycle i (hr)

P_i = average power per data acquisition cycle i (W)

and for a compressor cycle:

$$E_C = \sum_{i=1}^{N} E_i = \sum_{i=1}^{N} P_i \; RT_i \tag{4.3}$$

$$RT_C = \sum_{i=1}^{N} RT_i \tag{4.4}$$

where

E_C = Energy consumption of the unit per compressor cycle (W-hr)

RT_C = Compressor run time of the unit per compressor cycle (hr)

N = Number of compressor cycles in a test

Finally, the energy consumption of each compressor cycle was added in order to measure the total energy consumption of the unit during a test. The same procedure was utilized to measure the compressor run time of a test. Appendix A showes data from an entire three hour test consisting of several compressor cycles.

Next, to measure the effect of gasket infiltration on refrigerator/freezer unit energy consumption, the baseline unit gasket was taped tightly to its steel structure using a two sided tape. This eliminated all possible air infiltration through the gasket. Under the same condition as the baseline unit, the total energy consumption and compressor run time of the new setup were measured. The measurements were done in the same manner as for the baseline unit.

The measured energy consumption values were subtracted from the baseline energy consumption in order to provide the consumption due solely to the infiltration effect. Similarly, the baseline compressor run time was subtracted from the measured run time of the infiltration free unit to come up with the effect of infiltration on compressor run time. Also, the micro-manometer was utilized to measure the actual differential pressure between the inside of the refrigerator/freezer cabinet and the outside pressure. This was done to verify the previously estimated value ($\Delta p = 0.01$) that was utilized to calculate the infiltration. The micro-manometer showed $\Delta p =$ 0.012 inches of water which was very close to previously estimated value.

With the above experimental procedure and setup as shown in Fig. 4.1, the tests proceeded in the following manner:

1. Start-up of all the instruments and equipment.

2. Monitoring of the conditions by the computer to obtain the steady state con-

ditions. The steady state condition were obtained when compressor run time, temperature of fresh food and freezer compartment, and compressor off time reached steady state conditions.

3. Data taking (energy consumption, average power, temperatures, etc) for a specific period.

4. Repeat step 3 until the desired number of samples are obtained.

Gasket Heat Gain

To measure the total effect of gasket heat gain (conduction combined with infiltration, test C) on the refrigerator/freezer, a new modification was added to the infiltration-free unit. This modification involved adding insulation around the gasket area using cotton batting and duct tape. This reduced all possible gasket heat gain (conduction as well as infiltration) to a negligible level (see Fig. 4.4). Energy consumption and compressor run time of the refrigerator/freezer due to total gasket heat gain were measured under the same conditions as the base line unit. Again, the obtained energy consumption was subtracted from the baseline energy consumption to provide the consumption of the refrigerator/freezer due to total gasket heat gain (conduction combined with infiltration). Also, the energy consumption of the refrigerator/freezer unit due to gasket heat leakage alone was determined by subtracting the total energy consumption from the energy consumption due to infiltration alone. The same procedure was utilized to determine the effect of gasket heat gain (conduction combined with infiltration) on compressor run time.

An alternative method is sometimes used in industry to measure heat gain

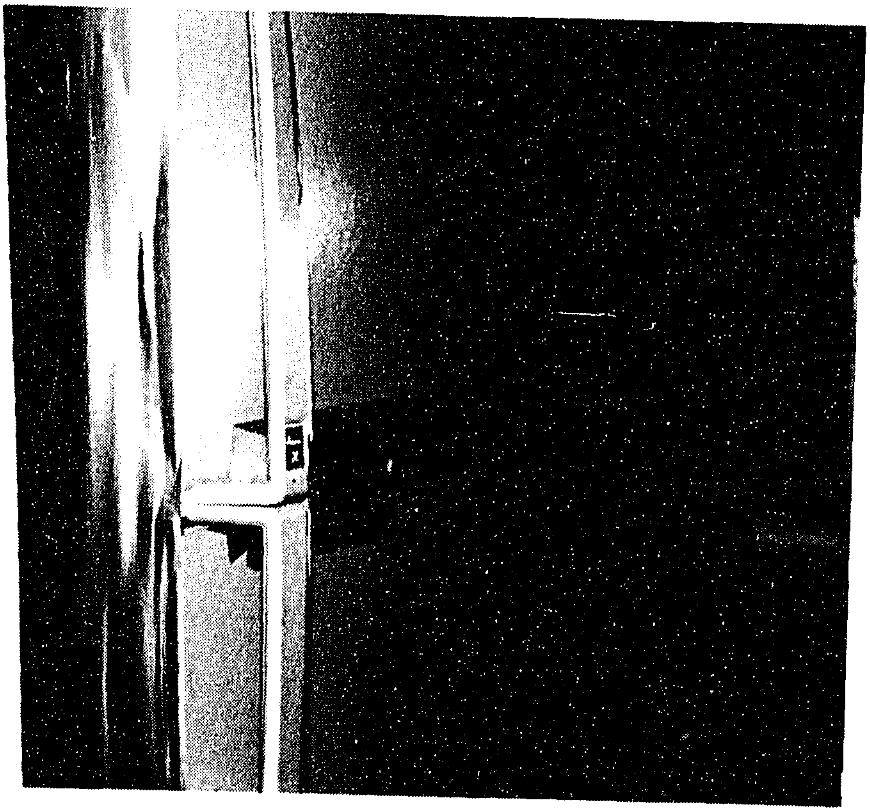


Figure 4.4: Photograph of the gasket heat gain setup

through the gasket as follows:

- Apply a constant heat source in the cabinet to measure the total heat loss of the unit. Also connect the same cabinet face to face with an identical cabinet, excluding the doors, and measure the heat loss through the walls of the two cabinets. The difference between the original test and one half the value measured in the second test is the total door loss. The loss through the door itself could then be analytically calculated. Finally, the gasket heat loss would be:

$$q_{gasket} = q_{total} - (q_{walls} + q_{door}) \tag{4.5}$$

This method leads to a plausible estimate of the heat gain through the gasket, but it relies on some speculation as to the door loss. However, the method that was utilized in this study is more direct and is felt to be more acuurate than the alternative one just mentioned.

Results and Discussion

In order to investigate the significance of gasket heat gain, two different test cycles (test A, B, and C per cycle) were performed and the results obtained are summarized in Tables 4.1 and 4.2, respectively. As mentioned before, in order to insure more accurate results, the reported results were obtained by conducting the same experiment on 5 different dates for each specific case (baseline unit test, infiltration free unit test, and gasket heat gain test).

Table 4.1 shows the results obtained during the first test cycle. As shown, the outside temperature stayed constant throughout this test cycle ($85 \pm 0.1^{o}F$) and the 1/7 thermostat setting (maximum possible cooling for fresh food compartment)

Table 4.1: The effect of gasket heat gain on refrigerator/freezer performance, test cycle 1

Charact-	Base Line (BL)				BL - Infiltration				BL - Total Gasket Losses			
eristic	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}
T_o [a]	85.3	.5	84.2	88	85.3	1.3	85.3	89.3	85.4	2.1	78	89.3
T_r [b]	34	.7	33	37.5	35	1.5	33.7	39.4	34.3	1	37.3	31.3
T_f [c]	5.9	.5	4.9	9	6	.3	5.5	6.7	6	.9	3.6	9.1
$\frac{cycles}{day}$	57	8	50	65	56	7	53	64	53	8	49	64
$\frac{kW-hr}{day}$	2.50	.04	2.44	2.53	2.47	.06	2.40	2.54	2.37	.06	2.30	2.43
RT [d]	40.6	6.6	32.8	46.1	40.0	3.4	35.4	44.6	38.6	3.3	35.2	42.3

[a] T_o= outside temperature, ^oF
[b] T_r=fresh food temperature, ^oF
[c] T_f=freezer temperature, ^oF
[d] RT = percent run time

resulted in measurements of $34.5 \pm 0.5^o F$ and $6 \pm .20^o F$ in the fresh food and freezer compartments, respectively. The results of test cycle 1 showed a total of 5% increase in energy consumption due to total gasket heat gain (heat leakage combined with infiltration effect). From this total of 5%, the increase in energy consumption due to the infiltration effect alone was about 1% and the rest (4%) was due to conduction heat leakage. Also, test cycle 1 showed similar results for the compressor run time and compressor cycles per day. As shown by Table 4.1, a 4.9% increase in compressor run time was measured due to total gasket heat gain (1.5% due to infiltration and 3.4% due to heat leakage). Also, compressor cycles increased by about 7% due to total gasket heat gain (2% increase due to infiltration and 5% due to conduction heat leakage)

Results obtained during the second test cycle are shown in Table 4.2. This test was conducted in order to verify the accuracy and repeatability of the first test cycle. Again, test cycle 2 was conducted while the outside temperature stayed constant throughout and the thermostat was set at 1/7. As shown by Table 4.2 similar results were obtained during the second test cycle compare to first test cycle. Energy consumption of the unit showed 5% increase due to total gasket heat gain (infiltration combined with heat leakage)(also see Table 4.4). From this total, 2% increase in energy consumption was due to the infiltration effect and the rest (3%) was due to heat leakage effect. Also a total of 5% of compressor run time is related to conduction heat leakage combined with infiltration. The small deviation between energy consumption of the unit during the first test cycle and second test cycle was partly due to T_O.

For further verification of these results, an analog KW-HR Meter was utilized

Table 4.2: The summarized effect of gasket heat gain on refrigerator/freezer performance, test cycle 2

Characteristic	Base Line (BL)				BL - Infiltration				BL - Total Gasket Losses			
	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}
T_o [a]	83.7	.9	81.7	87	83.8	1.5	81.9	91.7	83.8	2.1	77.5	86
T_r [b]	33.9	1.6	31.7	49	34	1.4	30.8	42.3	34	1.9	33.2	37
T_f [c]	5.5	.7	3.4	8.6	5.5	1.4	2.4	9.5	5.8	1.9	3.9	9
$\frac{cycles}{day}$	63	5	52	68	62	9	40	68	62	2	52	68
$\frac{kW-hr}{day}$	2.40	.26	2.21	3.19	2.36	.17	2.23	2.79	2.28	.04	2.24	2.31
RT [d]	43.3	5.0	40.0	54.8	42.5	6.7	25.1	54.0	41.1	4.9	40.7	42.3

[a] T_o= outside temperature, ^oF

[b] T_r=fresh food temperature, ^oF

[c] T_f=freezer temperature, ^oF

[d] RT = percent run time

Table 4.3: KW - HR Meter Read Out During Test Cycle 1 and 2

Charact-eristic	Test Number 1.					Test Number 2.				
	$\bar{x}$ [a]	σ	x_{min}	x_{max}	Δx [b]	$\bar{x}$	σ	x_{min}	x_{max}	Δx
Base Line	2.87	.17	2.65	3.07	-	2.81	.45	2.23	3.36	-
Inf. [c]	2.82	0.05	2.78	2.85	2	2.76	0.27	2.29	2.98	2
C & I [d]	2.72	0.1	2.65	2.78	5	2.66	0.04	2.61	2.69	5

[a] $\bar{x}$ = average energy consumption $(\frac{kW-hr}{day})$
[b] Δx = higher percent of energy consumption compare to base line
[c] Inf. = the infiltration effect on base line unit
[d] C & I = The conduction combined with infiltration effect on base line unit

Table 4.4: Comparison Between Different Test Cycle 1 and 2

Charact-eristic	Test Number 1.				Test Number 2.			
	$\bar{x}$ [a]	RT	Δx [b]	ΔRT [c]	$\bar{x}$	RT	Δx	ΔRT
Base Line	2.50	40.6	-	-	2.40	43.3	-	-
Inf. [d]	2.47	40.0	1.0	1.5	2.36	42.5	2.0	1.8
C & I [e]	2.37	38.6	5.0	4.9	2.28	41.1	5.0	5.0

[a] $\bar{x}$ = average energy consumption $(\frac{kW-hr}{day})$
[b] Δx = higher percent of energy consumption compare to base line
[c] ΔRT = higher percent of run time compare to base line
[d] Inf. = the infiltration effect on base line unit
[e] C & I = The conduction combined with infiltration effect on base line unit

during test cycle 1 and 2 and the values obtained from this meter are reported in Table 4.3. As indicated by Table 4.3, higher energy consumption were measured by this instrument compared with watt transducer and the data acquisition system (see Tables 4.1, 4.2, 4.3). This slight increase was partly caused by the compressor cooling fan energy consumption. In each compressor cycle, after the compressor shutoff, the compressor cooling fan continued to operate until the compressor cooled off. Thus, higher energy consumption was indicated. As shown in Table 4.3, a 5% higher energy consumption was measured by KW-HR Meter during test cycle 1 for total gasket heat gain (2% due to infiltration effect and 3% due to conduction effect). It also measured 5test cycle 2. From this total of 5% energy consumption, 2% was due to infiltration and the rest 3% was due to conduction.

The data taken during all tests show interesting trends as indicated in Table 4.4. In general, these tests indicate as much as 2% increase in energy consumption due to infiltration effects, which is not nearly as significant as has been claimed in the literature and by manufacturers. Also, the total energy use is 5% due to conduction heat leakage combined with infiltration effect. Again this is proven not to be as significant as has been claimed elsewhere. As mentioned before, according to some literature, up to a 21% increase in energy consumption of the refrigerator/freezer was obtained due to gasket heat gain.

Summary

In this chapter, the magnitude of gasket heat gain in a typical refrigerator/freezer was studied. Also, the significance of gasket infiltration and heat leakage was evaluated experimentally, and the primary findings were:

1. Experimental findings of the present chapter showed that only a small portion of total load was caused by infiltration. Even though there was little agreement among the manufacturers and researchers about the magnitude of infiltration, this study showed only 2% increase in energy consumption was due to gasket infiltration (see Table 4.4). This appears to contradict what some of the literature and manufacturers claim.

2. There is little certainty about the exact magnitude of gasket heat leakage among the literature and manufacturers, although most believe it is significant. This study showed as much as 3% increase in energy consumption due to gasket conduction which was insignificant compare to other claims (see Table 4.4).

3. Finally, the present study proved that the effect of total gasket heat gain (infiltration combined with conduction) on energy consumption of the refrigerator/freezer is far below what was suggested in the literature and by manufacturers. The present chapter showed that only 5% of total load is due to gasket heat gain while others claimed as much as 22.5% increase in total load due to gasket heat gain.

It is our hope that these data will help the industry and others place the proper perspective on reducing gasket heat gain as a potential for improving the efficiency of refrigerator/freezers.

CHAPTER 5. EXPERIMENTAL EVALUATION OF ALTERNATIVE REFRIGERANT

An experimental evaluation of alternative refrigerants was done and is presented in this chapter. All experiments were conducted in a calorometic room in the Building Energy Utilization Laboratory (BEUL) of the Mechanical Engineering Department. In this chapter, the details of the unit modification and its preparation are presented, followed by a brief discussion of experimental test procedures. This chapter also presents the results obtained during the tests, data analysis, and the conclusions drawn.

Test Setup

In order to conduct the desired tests on the previously mentioned refrigerator/freezer (baseline unit with CFC-12) with HFC-134a as a working fluid, the following modifications were made:

1. Evacuation of CFC-12 refrigerant.

2. Replacement of components for the refrigeration side (condenser, evaporator, compressor, and capillary tube).

3. Leak testing of the refrigerator/freezer unit.

4. Charging of the refrigerator/freezer unit with an ester based lubricant and HFC-134a refrigerant.

5. performance testing of the unit.

A brief description of each of these steps are as follows:

1. Line Evacuation

In order to replace HFC-134a with CFC-12 refrigerant, the CFC-12 refrigerant was evacuated from the baseline unit and was stored in a CFC-12 waste capsule. To extract CFC-12 from the unit, the temperature of the waste capsule was kept below the room temperature throughout the evacuation. This was done by placing the capsule in an ice bucket, hence causing CFC-12 to flow from the unit to the waste capsule. The lower temperature results in faster flow of the CFC-12 to the waste capsule.

2. Component Replacement

After CFC-12 refrigerant was evacuated from the baseline unit, all of the components on the refrigeration side (condenser, evaporator, compressor, and capillary tube) were replaced with new identical parts. The mentioned components were changed to avoid damage to the compressor and other components due to contact and/or interaction with even a small trace (one part per million) of CFC-12 with mineral oil and HFC-134a.

To insure that the replacement parts were identical with the previous ones, new components for the unit were obtained from its original manufacturer. In addition to these components (condenser, evaporator, compressor and capillary tube), two capillary tubes with different lengths were added in parallel to the original capillary tube (See Fig. 5.1). Figure 5.1 is a photograph of the capillary tube/needle valve manifold.

Based on the desired pressure drops across the capillary tubes, the following capillary tube lengths (CL) were selected:

1. CL = 125, D = 0.033 inch (original length, OL)

2. CL = 107, D = 0.033 inch (OL - 18 inch)

3. CL = 87, D = 0.033 inch (OL - 36 inch)

These capillary tubes built by the original manufacturer provide the necessary variation in the pressure drop. The capillary tube/needle valve manifold was utilized in order to identify the impact of pressure drop across capillary tube on the unit energy consumption using HFC-134a as a working fluid.

In addition to the previously mentioned instrumentation, a Setra (3-wire circuit, Model # 207, serial # 186213, range 0 to 500 psig, output 0.1 to 5.1 $VDC \pm 50$ mV, accuracy < ±0.13 % FS) and a Robinson-Halpern (model no. 174A, range 0 to 200 psig, output 0.1 to 5.1 VDC, accuracy < ±0.4 % FS at 200 psig) pressure transducer were installed at inlet and exit of the capillary tube, respectively, in order to measure the corresponding pressure at each station. These pressure transducers were powered externally by a 5-VDC power supply. Before installation, the pressure transducers were calibrated using a dead-weight tester.

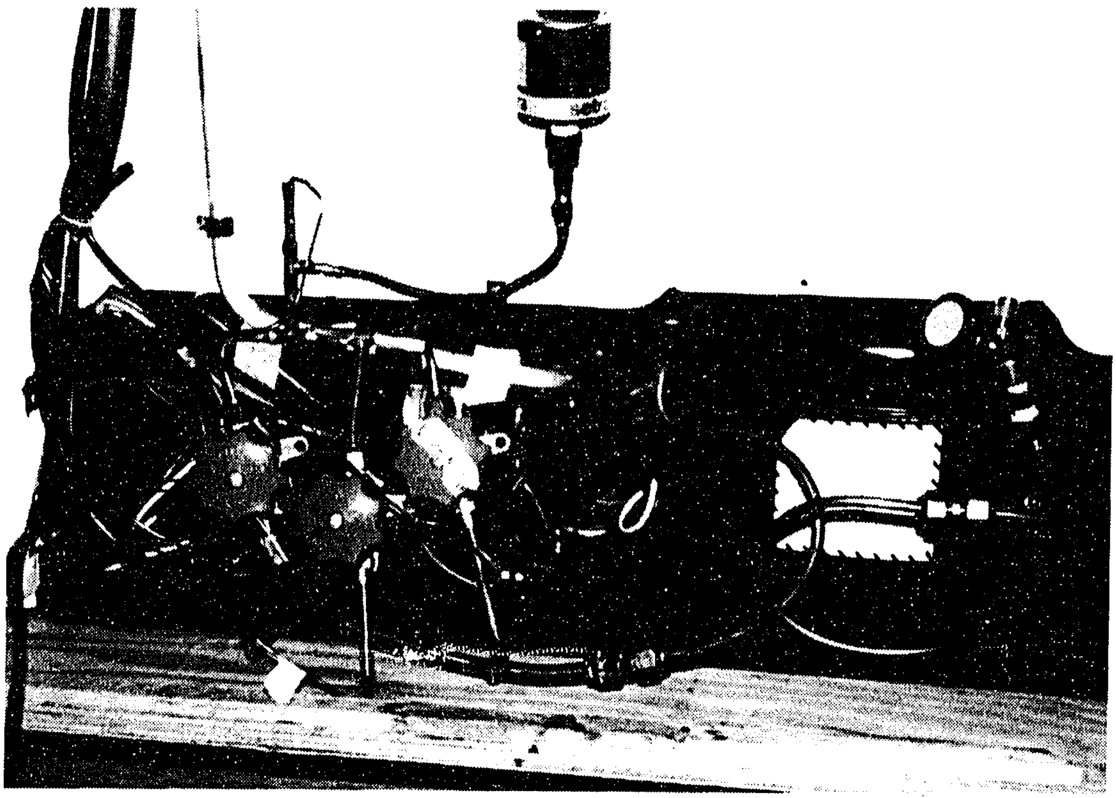


Figure 5.1: Photograph of the capillary tube setup

3. Leak Testing

Following the component change out, the baseline unit was tested for possible leaks. First, the unit was evacuated with a vacuum pump to a minimum of -30 in-Hg and left untouched for a period of time in order to detect the existence of any leaks in the unit. The rise in the system pressure (from -30 in-Hg to room pressure) was indicative of leaks in the unit. To pin point the location of each leaks, the unit was pressurized with HFC-134a vapor and a refrigerant electronic leak detector (General Electric halogen leak detector type H-10A) was used to detect the leaks. The tests were continued until all leaks were successfully detected and sealed.

4. System Charging

The final task involved the charging of the compressor and the system with a proper lubricant and HFC-134a. Since refrigerant HFC-134a is immiscible with mineral oil (the typical lubricant that used with CFC-12 refrigerant), it was necessary to use a different lubricant in the compressor. It is well known that the lubricant viscosity affects both the compressor and the overall system performance. Therefore, a thorough investigation was necessary prior to the selection of a specific lubricant for this purpose. The investigation led to the selection of an ester based lubricant (EMKARATE RL213B, polyol ester) which is miscible with HFC-134a. Additionally, the selected lubricant has viscosity close to that of mineral oil. The viscosity of 3GS mineral oil and the RL213b easter based lubricant is 29.5 and 21 CST (centistokes) at 104 ^oF (40 ^{o} C) and 4.3 and 4.5 CST at 212 ^oF (100 ^{o} C), respectively. Prior to charging the lubricant into the compressor, the lubricant was heated and placed under vacuum over night to ensure moisture removal. Using a precise laboratory balance, thirteen ounces of the lubricant were measured and charged into the compressor. This amount of lubricant was identical to the amount used with CFC-12 in the refrigerator/freezer, with a Tecumseh compressor. Again, the unit was placed under vacuum overnight to ensure that no moisture was picked up by the system during the charging.

After charging the compressor with an lubricant, the refrigerator/freezer unit was filled with HFC-134a refrigerant. A charge of 8.5 ounces of HFC-134a refrigerant was weighed using a precise electronic scale with ± 0.001 lb scale accuracy and introduced into the compressor. This amount of HFC-134a was identical to the amount used in a typical $20 ft^3$ refrigerator/freezer.

Test Procedure

In all tests that were conducted, the outside temperature (T_O) was maintained at a constant level and the thermostat of the unit was set at 1/7. Additionally, each specific test was conducted at least 5 different times to ensure accurate average results.

Two different test cycles were conducted in order to investigate the effects of capillary tube length as well as refrigerant charge on energy consumption of the refrigerator/freezer unit. These test cycles were:

1. Test cycle 1: effect of pressure drop variation across the capillary tube (different capillary tube length) on energy consumption of the refrigerator/freezer unit.

2. Test cycle 2: effect of HFC-134a charge variation on energy consumption of the refrigerator/freezer.

A brief description of each of these cycles follows:

1. Test Cycle 1: Pressure Drop Variation

In order to determine the effect of pressure drop across the capillary tube on energy consumption of the unit, a number of tests were conducted. First, during each test, a capillary tube length had to be selected. This was done by selecting one of the three fixed lengths or a combination of fixed length and needle valve adjustments. To obtain pressure drops other than those related to the three fixed capillary lengths (CL = 125, 108, and 87 inch), the needle valve of desired capillary tube was adjusted and monitored via the data acquisition system until the desired pressure drop was obtained. Following the selection of each capillary tube length (different pressure drop across the tube), the unit was allowed to run until steady state conditions were achieved. Then the required data were automatically read by the data acquisition system and stored in data files to be analyzed later.

2. Test Cycle 2: Charge Variation

In this test cycle, a total of 5 different charges were tested to determine their effect on energy consumption. In addition to these 5 different charges, throughout this test cycle the unit was operated with the capillary tube length that gave the lowest energy consumption during test cycle 1. To measure energy consumption of the unit utilizing different charges, first the system was evacuated overnight to ensure that no HFC-134a was presented in the compressor lubricant. Then, the refrigerant was charged in 1 ounce increments to the desired level for each test. Each charge was measured using a precise laboratory scale.

Results and Conclusion

Two different test cycles (test cycle 1 and 2) were conducted in order to investigate the significance of pressure drop across the capillary, as well as, the effect of different HFC-134a charges on the energy consumption of the refrigerator/freezer unit. As mentioned before, in order to ensure more accurate results, the reported average results were obtained by conducting the same experiment at least 5 different times for each of the cases within a cycle (different capillary lengths and different charges).

1. Test Cycle 1: Pressure Drop Variation

A total of eight different pressure drops across the capillary tubes were tested. First, testing was performed for each of the three available capillary lengths (CL = 125, 108, and 87 inch) while the respective needle valve was left fully open. The results are summarized in Table 5.1, while the corresponding energy consumptions, the calculated means and the pressure drops for the entire test cycle are presented in Fig. 5.2. As shown by Fig. 5.2, the pressure drop across the capillary is inversely related to the energy consumption of the unit and it shows a slight downward trend with increasing pressure drop. The lowest energy consumption was measured at $\Delta p = 141 psi$. This pressure drop ($\Delta p = 141 psi$) corresponds to the 125 inch capillary length which currently is standard with CFC-12 in the refrigerator/freezer. Also, for all three capillary lengths, a slightly lower energy consumption was measured relative to the CFC-12 baseline unit (see Table 5.1). While the obtained results were promising, they were not complete.

Another complete set of tests were conducted in order to determine a pressure

Table 5.1: The effect of pressure drop across capillary tube on refrigerator/freezers performance for first test

Charact-eristic	Δp = 141 psi (L=125 in)				Δp = 136 psi (L=107in)				Δp = 128 psi (L=89 in)			
	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}
T_o [a]	85.2	.5	84.2	85.9	85.3	.7	83.7	86.3	85.3	.3	83.9	85.4
T_r [b]	36.5	.5	35.5	37.2	36.5	.6	35	37.3	36.1	.4	35.1	36.8
T_f [c]	7.4	.6	6.2	8.3	7.3	.7	5.7	8.2	6.7	.4	5.8	7.5
$\frac{cycles}{day}$	66	2	64	68	64	-	-	-	64	-	-	-
$\frac{kw-hr}{day}$	2.34	.02	2.36	2.32	2.38	.02	2.36	2.41	2.41	.02	2.39	2.43
RT [d]	45.1	.3	44.9	45.4	45.8	.8	45.3	46.4	46.4	.3	46.0	46.7

[a] T_o= outside temperature, ^oF

[b] T_r=fresh food temperature, ^oF

[c] T_f=freezer temperature, ^oF

[d] RT = percent run time

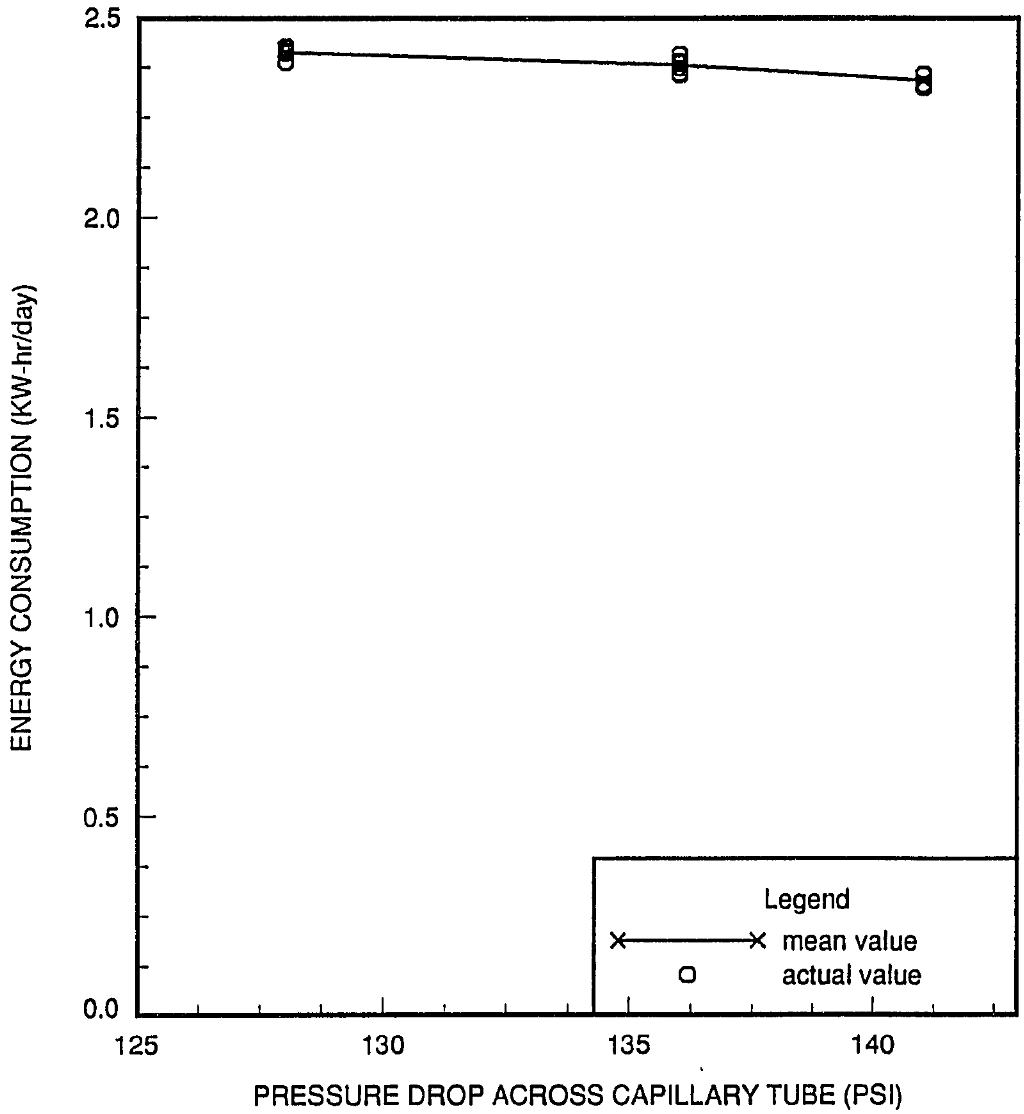


Figure 5.2: The effect of pressure drop across the capillary tube on refrigerator/freezer energy consumption for test one

drop across capillary tube that yield the lowest energy consumption. Utilizing the previously mentioned procedure, a total of 5 different pressure drops were selected and the obtained results are summarized in Table 5.2 while the measured energy consumptions, the calculated means and the corresponding pressure drops for the entire test cycle are presented in Fig. 5.3. As seen in Fig. 5.3, the energy consumption shows a slight minimum in the range of pressure tested, while $\Delta p = 137psi$ resulted in the lowest energy consumption during the second test. This pressure drop ($\Delta p = 137psi$) corresponded to the 125 inch capillary tube length. This result confirms the earlier finding regrading the capillary length that yield the lowest energy consumption. Also, as indicated earlier, the 125 inch capillary length is currently being used with CFC-12 in the refrigerator/freezer. Additionally, the second test shows similar results for the compressor run time (see Table 5.2).

To verify the repeatability of these results, another set of tests for the three original capillary tube lengths (L = 125, 108, and 87 inch) were conducted and the results are presented in Fig. 5.4. As shown by Fig. 5.4, the results obtained are similar to previous test. The results confirm that $\Delta p = 137psi$ yields the lowest energy consumption. Other pressure drops across the capillary were not tried due to difficulties associated with obtaining the very exact pressure drop using the installed needle valve setup.

The measured energy consumptions, the calculated means and the corresponding pressure drops for the entire cycle are presented in Fig. 5.5. Again, the energy consumption shows a slight minimum in the pressure range tested and the results confirm the previous findings. Also, it shows that the lowest energy consumption is obtained when the capillary tube length of 125 inch is utilized.

Table 5.2: The effect of pressure drop across the capillary tube on the refrigerator/freezer performance for second test

Characteristic	Δp = 144 psi				Δp = 140 psi				Δp = 137 psi (L=125 in)			
	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}
T_o [a]	85.7	.5	84.7	86.3	85.8	.3	84.8	86.3	85.5	.5	84.7	86.5
T_r [b]	36.5	.4	35.7	37.1	36.9	.3	36.2	37.3	36.7	.5	36	37.7
T_f [c]	7.5	.5	6.3	8.2	8.1	.3	7.2	8.7	7.7	.6	6.9	8.8
$\frac{cycles}{day}$	72	-	-	-	70	4	64	72	67	4	64	72
$\frac{kW-hr}{day}$	2.46	.04	2.40	2.52	2.36	.04	2.31	2.40	2.35	.02	2.33	2.38
RT [d]	48.4	.9	47.3	49.7	46.0	.7	45.0	46.9	45.9	.3	45.4	46.3

[a] T_o= outside temperature, ^oF
[b] T_r=fresh food temperature, ^oF
[c] T_f=freezer temperature, ^oF
[d] RT = percent run time

Table 5.2 (Continued)

Charact-eristic	Δp = 133 psi (L=107 in)				Δp = 127 psi (L=89 in)			
	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}
T_o	85.1	.5	84.6	86.7	84.9	.1	84.6	85.1
T_r	36.4	.5	35.9	37.9	36.2	.1	35.9	36.4
T_f	7.3	.6	6.7	9	7.1	.3	6.5	7.3
$\frac{cycles}{day}$	64	-	-	-	64	-	-	-
$\frac{kw-hr}{day}$	2.42	.04	2.37	2.48	2.44	.03	2.40	2.41
RT	47.1	.8	46.1	48.2	47.3	.5	47.4	47.6

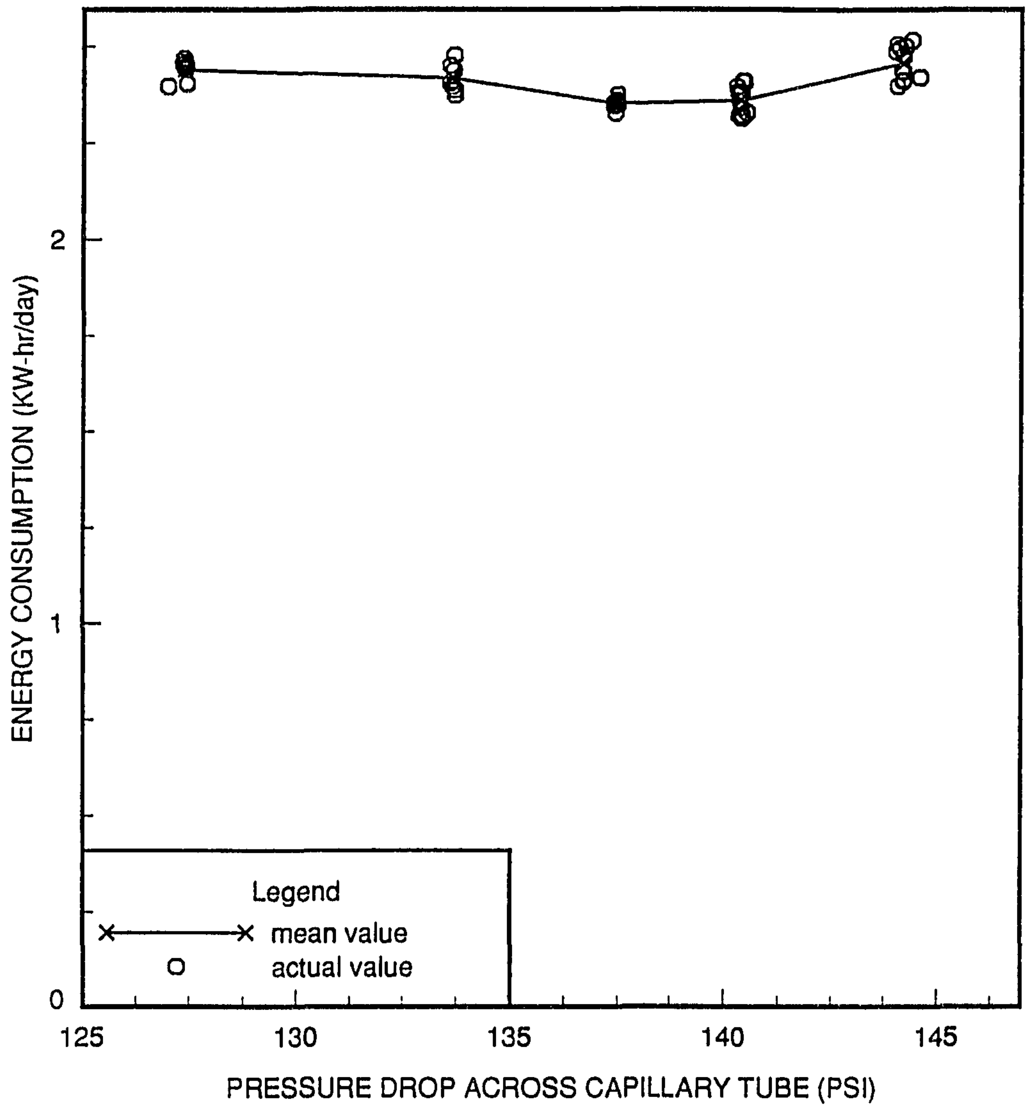


Figure 5.3: The effect of pressure drop across the capillary tube on refrigerator/freezer energy consumption, second test

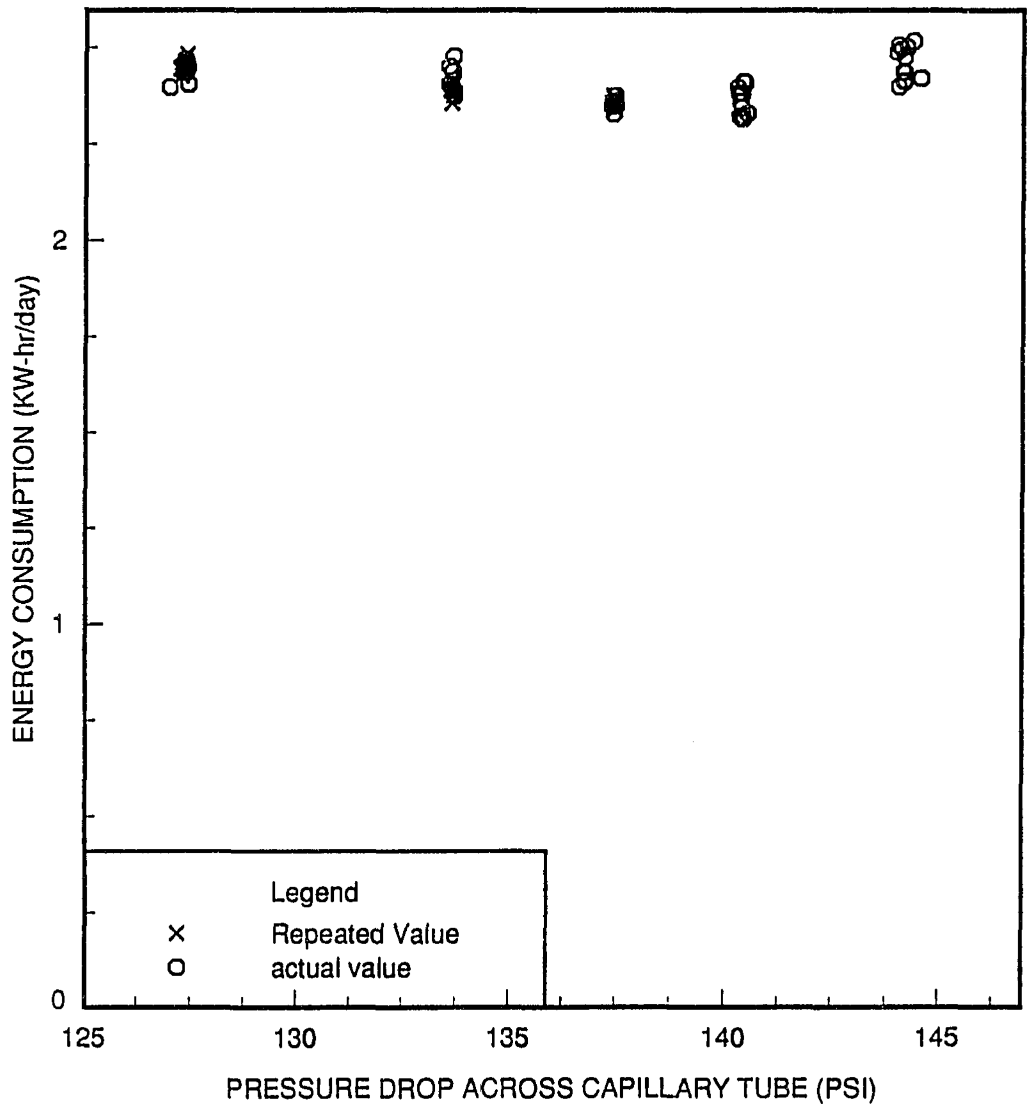


Figure 5.4: The effect of pressure drop across the capillary tube on refrigerator/freezer energy consumption, repeatability for second test

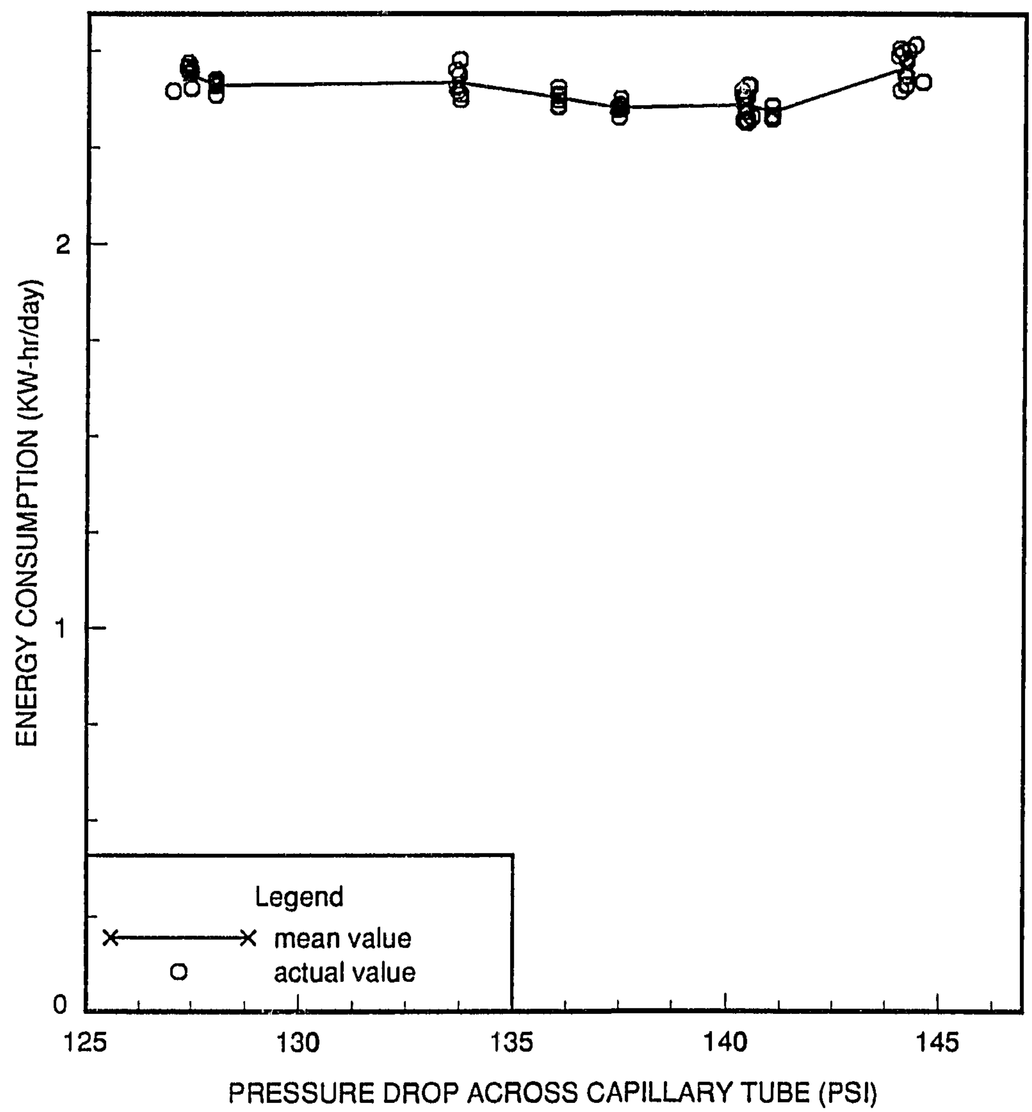


Figure 5.5: The effect of pressure drop across the capillary tube on refrigerator/freezer energy consumption, for test cycle 1

The results obtained from these tests indicate that the 125 inch capillary tube length yield the lowest energy consumption for HFC-134a refrigerator/freezer unit. This energy consumption is slightly lower than that of comparable CFC-12 unit, and the indicated capillary tube length (125 inch) was used to conduct the tests for HFC-134a charge variation.

2. Test Cycle 2: HFC-134a Charge Variation

A total of seven different charges were tested to determine the effect of charge on energy consumption. Results corresponding to the different charges are summarize in Table 5.3, and the measured energy consumptions, the calculated means, and the related pressure drops for the entire test cycle are presented in Fig. 5.6. As seen in Fig. 5.6, the energy consumption shows a slight changes with increasing HFC-134a refrigerant charge. As shown by Table 5.3 and Fig. 5.6, the 8.5 ounce charge of HFC-134a yields the lowest energy consumption by the unit. Again, this amount of charge (8.5 HFC-134a) is similar to that of CFC-12 which was used in the baseline refrigerator/freezer unit. Also, Table 5.3 shows similar results for the compressor run times. Figure 5.7 shows the energy consumption results obtained during the second test. This test was conducted to ensure the accuracy and repeatability of previous test. As shown by Fig 5.7, the obtained results confirm the previous findings identified during the second test.

The results obtained during both cycles (test cycle 1 and 2) were compared with the ones obtained from a bread-board type vapor compression system built by Technovate Inc. (see Fig. 5.8) [36]. This unit operates with CFC-12 as a working fluid and provides very accurate results during operations. In order to compare the

Table 5.3: The effect of HFC-134a refrigerant charge level on refrigerator/freezer performance for test 1

Charact-eristic	Charge = 6 (oz)				Charge = 7 (oz)				Charge = 8 (oz)				Charge = 8.5 (oz)			
	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}
T_o [a]	84.9	.2	84.5	85.3	84.7	.3	84.1	85.4	84.7	.3	84.2	85.4	85.5	.5	84.7	86.5
T_r [b]	31.7	.4	31.2	32.8	33.1	.3	32.6	33.6	34.7	.4	34.1	36.3	36.7	.5	36	37.7
T_f [c]	4.7	.3	4	5.2	5.3	.3	4.8	5.9	5.9	.3	5.2	6.9	7.7	.6	6.9	8.8
$\frac{cycles}{day}$	48	-	-	-	56	-	-	-	64	-	-	-	66.64	4.16	64	72
$\frac{kw-hr}{day}$	2.80	.02	2.78	2.83	2.54	.023	2.51	2.58	2.46	.032	2.40	2.51	2.35	.02	2.33	2.38
RT [d]	76.0	1.6	73.1	78.0	55.9	.5	55.3	57.0	51.3	.8	49.9	54.7	45.9	.3	45.4	46.3

[a] T_o= outside temperature, ^oF
[b] T_r=fresh food temperature, ^oF
[c] T_f=freezer temperature, ^oF
[d] RT = percent run time

Table 5.3 (Continued)

Charact-eristic	Charge = 9.0 (oz)				Charge = 10.0 (oz)				Charge = 11.0 (oz)			
	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}	$\bar{x}$	s	x_{min}	x_{max}
T_o	85.4	.3	84.1	85.2	85.6	.2	84.6	85.4	85.8	.4	84.7	86.5
T_r	35.4	.3	34.6	35.7	35.8	.3	35.3	37	35.4	.3	34.8	36
T_f	6.4	.3	5.5	6.8	6.8	.2	6.4	7.4	6.4	.6	5	7.1
$\frac{cycles}{day}$	64	-	-	-	67	4	64	72	64	-	-	-
$\frac{kw-hr}{day}$	2.37	.02	2.34	2.40	2.37	.013	2.35	2.39	2.58	.025	2.54	2.63
RT	47.3	.4	46.6	47.9	46.0	.6	45.1	46.8	48.9	.3	46.8	51.0

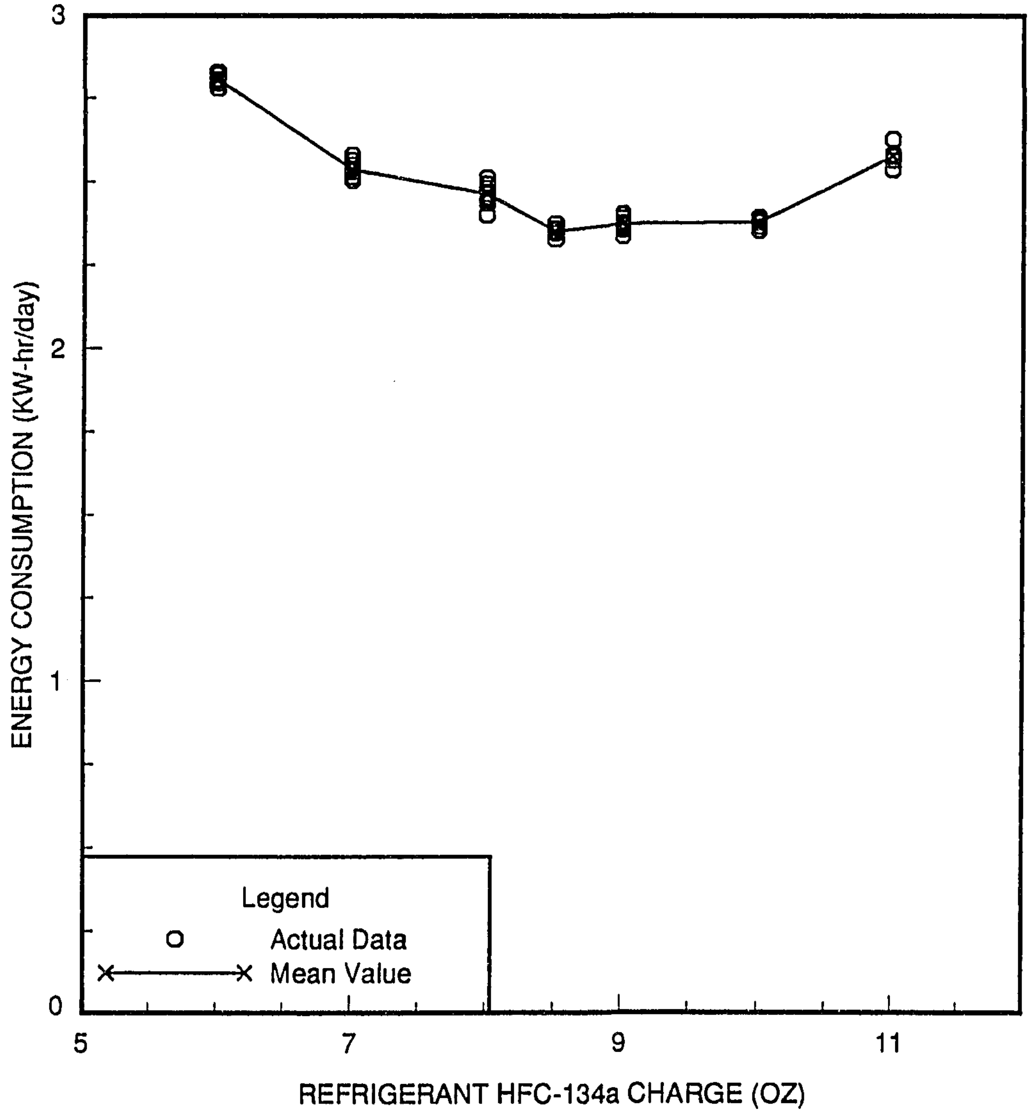


Figure 5.6: The effect of HFC-134a refrigerant charge level on refrigerator/freezer energy consumption for test 1

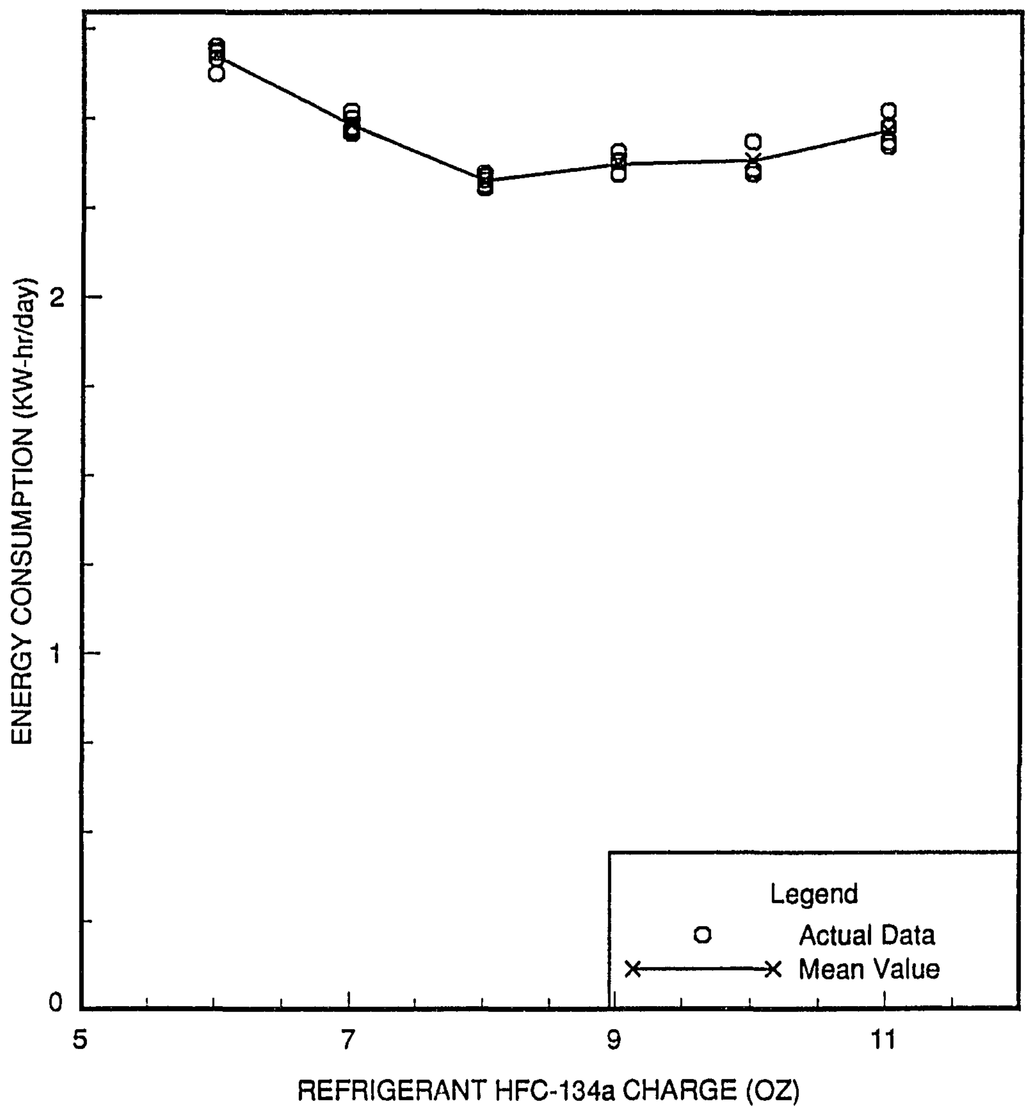
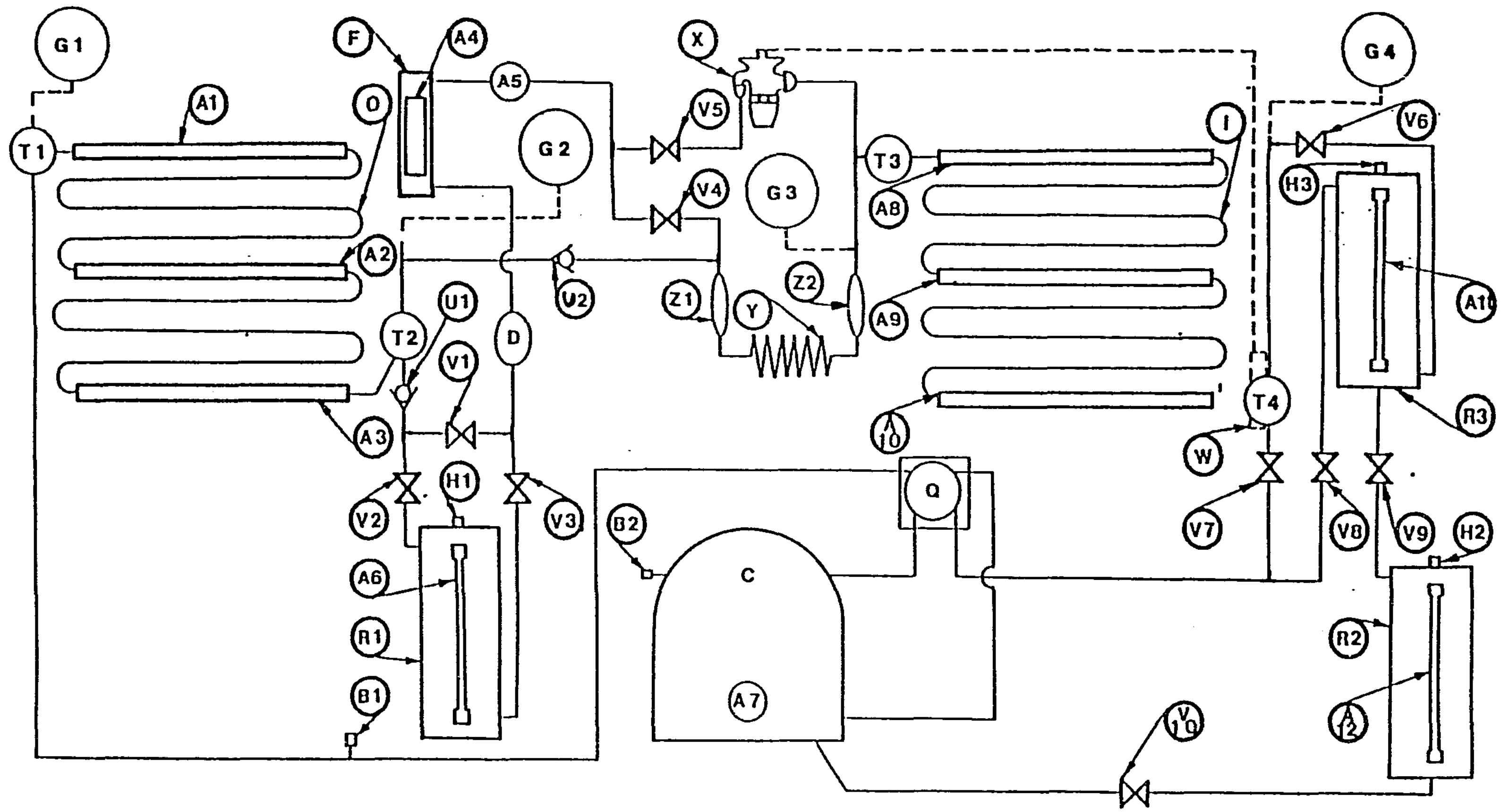


Figure 5.7: The effect of HFC-134a refrigerant charge level on refrigerator/freezer energy consumption for test two


G1
T1
A1
A2
A3
O
F
A4
A5
G2
T2
U1
U2
V1
V2
V3
D
H1
A6
R1
B1
X
V5
V4
G3
Z1
Z2
Y
T3
A8
A9
A10
I
B2
C
A7
Q
V10
G4
V6
H3
A11
R3
T4
W
V7
V8
V9
H2
R2
A12

A-1	Sight Tube, Outside Coil (inlet; cooling/outlet; heating)
A-2	Sight Tube, Outside Coil (center; cooling/heating)
A-3	Sight Tube, Outside Coil (outlet; cooling/inlet; heating)
A-4	Sight Tube, Flowmeter
A-5	Sight Glass, Liquid Line
A-6	Sight Tube, Refrigerant Receiver
A-7	Sight Glass, Compressor Sump
A-8	Sight Tube, Inside Coil (inlet; cooling/outlet; heating)
A-9	Sight Tube, Inside Coil (center; cooling/heating)
A-10	Sight Tube, Inside Coil (outlet; cooling/inlet heating)
A-11	Sight Tube, Refrigerant/Oil Seperator
A-12	Sight Tube, Oil Receiver
B-1	Access Fitting, High Side
B-2	Access Fitting, Low Side
C	Compressor
D	Drier
F	Flowmeter, Liquid Line
G-1	Gauge, Outside Coil (inlet; cooling/outlet heating)
G-2	Gauge, Outside Coil (outlet; cooling/inlet heating)
G-3	Gauge, Inside Coil (inlet; cooling/outlet heating)
G-4	Gauge, Inside Coil (outlet; cooling/inlet; heating)
H-1	Fusible Plug, Refrigerant Receiver
H-2	Fusible Plug, Oil Receiver
H-3	Fusible Plug, Seperator
I	Inside Coil
O	Outside Coil
Q	Reversing Valve
R-1	Receiver, Refrigerant
R-2	Receiver, Oil
R-3	Seperator, Oil/Refrigerant
T-1	Thermometer, Outside Coil (inlet; cooling/outlet; heating)
T-2	Thermometer, Outside Coil (outlet; cooling/inlet; heating)
T-3	Thermometer, Inside Coil (inlet; cooling/outlet; heating)
T-4	Thermometer, Inside Coil (outlet; cooling/inlet; heating)
U-1	Check Valve
U-2	Check Valve
V-1	Valve, Refrigerant Receiver By-Pass
V-2	Valve, Refrigerant Receiver Inlet
V-3	Valve, Refrigerant Receiver Outlet
V-4	Valve, Capillary Metering (Cooling Only)
V-5	Valve, Thermostatic Expansion (Cooling Only)
V-6	Valve, Seperator Inlet
V-7	Valve, Seperator By-Pass
V-8	Valve, Seperator Outlet
V-9	Valve, Oil Receiver Inlet
V-10	Valve, Oil Receiver Outlet
W	Remote Bulb (TXV)
X	Thermostatic Expansion Valve
Z-1	Strainer (Cooling Cycle)
Z-2	Strainer (Heating Cycle)

Figure 5.8: Schematic of vapor compression system by Technovate

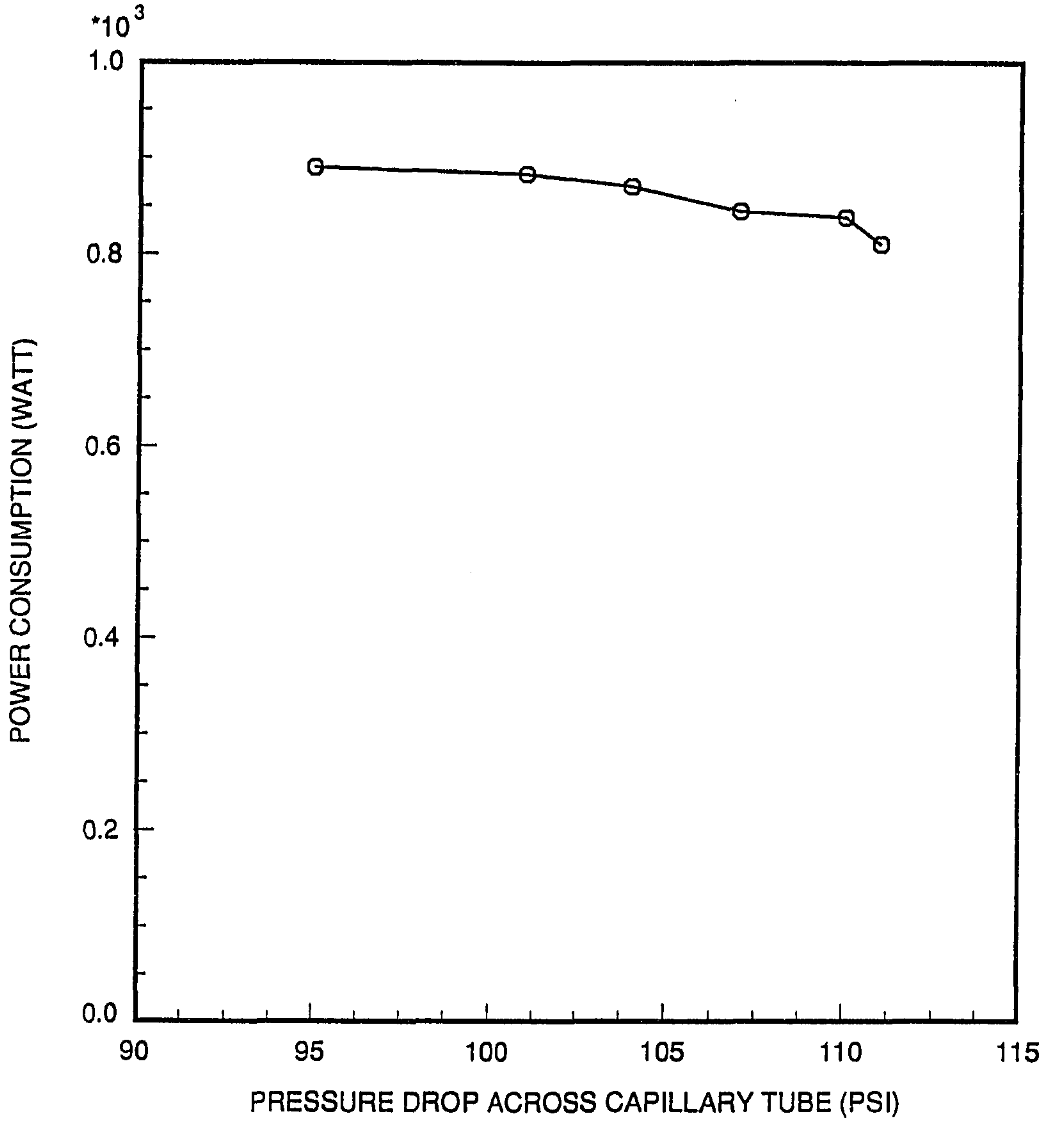


Figure 5.9: Effect of pressure drop across capillary tube on power consumption of vapor compression system, Technovate

Technovate results with test cycle 1 and 2, the trends instead of the actual values of the overall energy consumption were considered. The results of different pressure drops across the capillary tube in the Technovate unit are shown in Fig. 5.9. As shown by Fig. 5.9, the unit power consumption decreases as pressure drop increases. Several attempt were made to determine a pressure drop that provide the lowest power consumption. However, this was not possible due to compressor shut off.

Also, a series of similar tests were conducted to evaluate the effect of CFC-12 charge level on the Technovate system power consumption (see Fig. 5.10). As shown by Figure 5.10, the power consumption decreases as the charge inside the system increases. The results obtained for the Δp and the compressor charges confirm the trends indicated earlier.

Summary

In this chapter, the effect of pressure drop across capillary tube as well as HFC-134a charge level on energy consumption of a HFC-134a based refrigerator/freezer unit was evaluated and discussed. The main results are summerized as follows:

1. The lowest energy consumptions were measured when the typical capillary (CL = 125 inch) length was used.
2. A charge of 8.5 ounces of HFC-134a along with typical capillary tube length mentioned above resulted in lowest energy consumption.
3. For the HFC-134a baseline unit (8.5 ounce refrigerant and 125 inch capillary), lower energy consumptions were measured relative to the CFC-12 baseline unit.

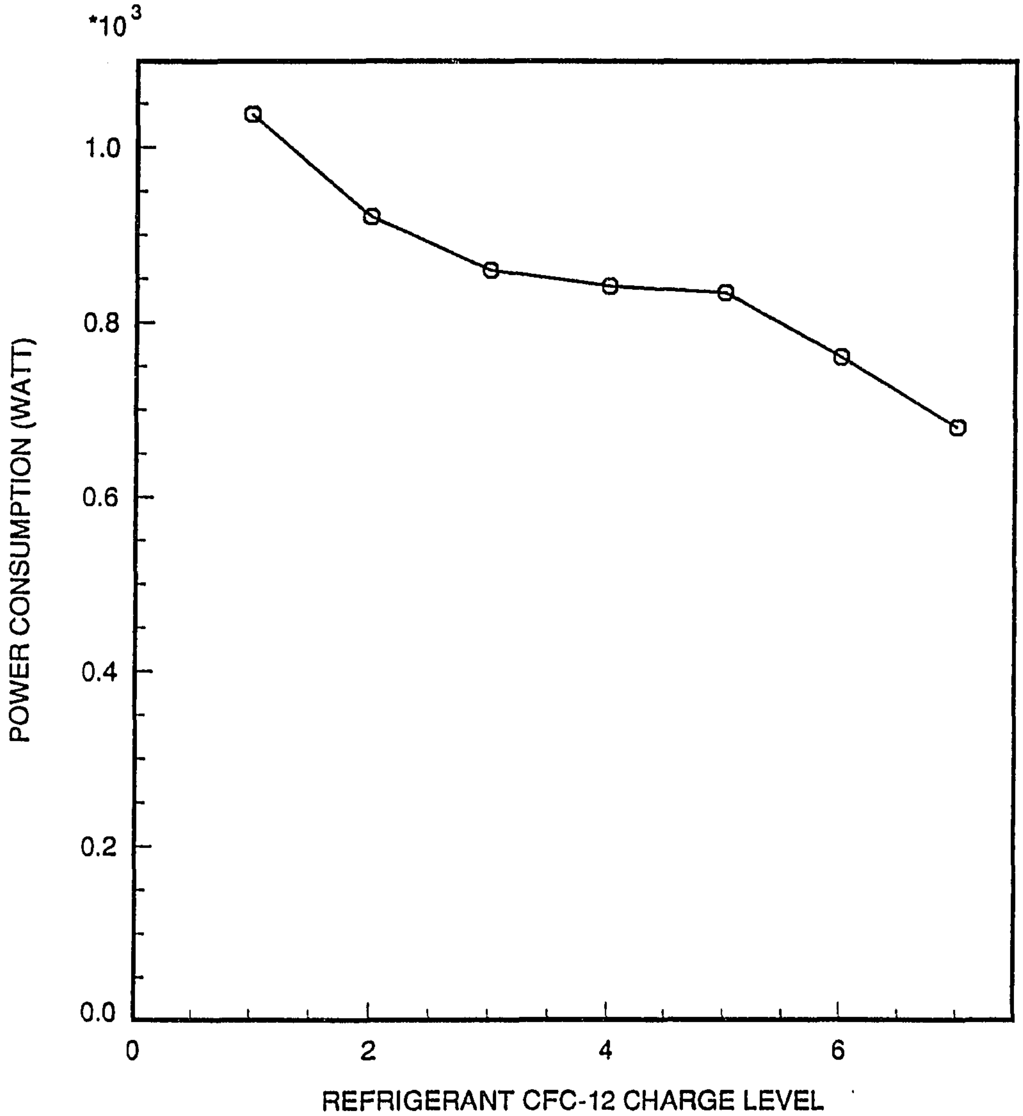


Figure 5.10: Effect of CFC-12 charge level on power consumption of vapor compression system, Technovate

The findings of this chapter apply only to residential refrigerator/freezers, and more specifically to the particular unit tested. In addition, the findings are based on the mentioned changes and modifications and may not be the same when the system operates under other conditions, such as various ambient temperatures. Also, the results could be affected by testing another manufacturer's products.

CHAPTER 6. SUMMARY AND CONCLUSIONS

This study was undertaken to investigate various experimental and theoretical, questions regarding total gasket gain (gasket infiltration and conduction heat gain) and the effects of HFC-134a on energy consumption of refrigerator/freezers.

The findings of this study apply only to residential refrigerator/freezers, and more specifically to the particular unit tested. In addition, the findings are based on the mentioned changes and modifications and may not be the same when the system operates under other conditions, such as various ambient temperatures. Also, the results could be affected by testing another manufacturer's products.

Gasket Study

This study presents the results of an extensive literature review, interviews with refrigerator/freezer and gasket manufacturers, and a theoretical calculation of gasket infiltration and gasket heat gain. Also, a $20 ft^3$ refrigerator/freezer was instrumented and three different test setups were utilized in order to experimentally measure gasket infiltration and gasket heat gain. This measurement was performed while the outside temperature was maintained constant throughout the tests. In addition to constant outside temperature, a sample size of five (N = 5, the same experiment five different times) and a 24 hour test cycle were selected. The primary findings of the gasket study were as follows:

1. The gasket area, including the gasket itself and the adjacent areas of the door and cabinet, has received considerable attention with respect to improvement of energy efficiency.

2. Based upon the literature findings and manufacturer's claims, there appears to be some uncertainty concerning the magnitudes of conduction heat leakage and infiltration effects, although most believe them to be significant.

3. The theoretical calculations done in the present study suggested that infiltration may be an important cause of heat gain.

4. However, experimental findings of the present study showed that only a small portion of total load was caused by infiltration. This study showed only a 2% increase in energy consumption is due to gasket infiltration.

5. This study showed as much as 3% increase in energy consumption due to gasket conduction, which was insignificant compare to other claims.

6. The present study demonstrated that the effect of total gasket heat gain (infiltration combined with conduction) on energy consumption of the refrigerator/freezer is far below what was suggested in the literature and by manufacturers. The present study showed that only 5% of total load is due to gasket heat gain while others claimed as much as 22.5% increase in total load due to gasket heat gain.

Experimental findings of this study showed that only a small portion of refrigerator/freezer thermal load (5%) is caused by gasket heat gain (infiltration as well as conduction) although most manufacturers and researchers believe it is significant. It is our hope that these findings will help the industry and others place the proper perspective on reducing gasket heat gain as a potential for improving the efficiency of refrigerator/freezers.

HFC-134a Study

In order to experimentally evaluate the effect of HFC-134a refrigerant on the previously mentioned refrigerator/freezer unit, the refrigeration side components were replaced with new identical parts. In addition to refrigerant side components (condenser, evaporator, compressor, and capillary tube), two capillary tubes with different lengths were added in parallel to the original capillary tube. The capillary tube and needle valve setup provided a wide range of pressure drops across the capillary tube.

A total of 8 different pressure drops and 7 different HFC-134a refrigerant charges along with the optimum capillary length were tested and the major findings of the conducted tests were as follows:

1. The lowest energy consumptions were measured when the typical capillary (CL = 125 inch) length was used.

2. A charge of 8.5 ounces of HFC-134a along with typical capillary tube length mentioned above resulted in lowest energy consumption.

3. For the HFC-134a baseline unit (8.5 ounce refrigerant and 125 inch capillary), lower energy consumptions were measured relative to the CFC-12 baseline unit.

This study showed that no dramatic changes in energy consumption occur when the same refrigerator/freezer is switched from CFC-12 to HFC-134a. However, with the mentioned minor modifications that were made to the refrigerator/freezer unit, the HFC-134a based unit showed slightly smaller energy consumption compared to that of CFC-12 unit for all cases. Also, the pressure drop and HFC-134a refrigerant charge findings provide important information for designers of refrigerator/freezers that utilize HFC-134a as a working fluid. It is note worthy that the replacement of CFC-12 with HFC-134a and a suitable lubricant resulted in only minimal variations of performance of the refrigerator/freezer tested. Further, the results were only slightly sensitive to effective capillary tube length and refrigerant charge. This suggests that the penalties paid in terms of performance may not be great in the short run and that future designs may show some gains as systems are re-optimized.

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APPENDIX SAMPLE DATA

This data is taken on 09-21-1992
****

TIME = 00:10:38
ITERATION #= 1.
TRi = 85.08
TFZi = 6.8
TFFi = 36.02
TRf = 85.14
TFZf = -3.27
TFff = 30.75
COMPRESSER RUNING TIME= 614.3499784469604
ENGERY CONSUMED IN THIS CYCLE(W-HR) 36.03557179340458
AVERAGE POWER FOR THIS CYCLE(W) = 211.1631203832727

TOTAL ENERGY CONSUMED (W-HR)= 36.03557179340458

****

TIME = 00:32:36
ITERATION #= 2
TRi = 84.99
TFZi = 6.69
TFFi = 35.96
TRf = 84.98
TFZf = -3.55
TFff = 30.53
COMPRESSER RUNING TIME= 614.06005859375
ENGERY CONSUMED IN THIS CYCLE(W-HR) 35.91301856324769
AVERAGE POWER FOR THIS CYCLE(W) = 210.5443352296348

TOTAL ENERGY CONSUMED (W-HR)= 71.94859035665223

****

TIME = 00:54:34
ITERATION #= 3
TRi = 84.81
TFZi = 6.53
TFFi = 35.83
TRf = 84.82
TFZf = -3.69
TFff = 30.3
COMPRESSER RUNING TIME= 612.090087890625
ENGERY CONSUMED IN THIS CYCLE(W-HR) 35.77082138346353
AVERAGE POWER FOR THIS CYCLE(W) = 210.3856270965781

TOTAL ENERGY CONSUMED (W-HR)= 107.7194117401157

****

TIME = 01:16:31

ITERATION #= 4
TRi = 84.74
TFZi = 6.36
TFFi = 35.7
TRf = 84.8
TFZf = -3.68
TFff = 30.36
COMPRESSER RUNING TIME= 612.909912109375
ENGERY CONSUMED IN THIS CYCLE(W-HR) 35.90985813582358
AVERAGE POWER FOR THIS CYCLE(W) = 210.9208657501617

TOTAL ENERGY CONSUMED (W-HR)= 143.6292698759393

****

TIME = 01:38:36
ITERATION #= 5
TRi = 84.74
TFZi = 6.33
TFFi = 35.66
TRf = 84.83
TFZf = -3.59
TFff = 30.36
COMPRESSER RUNING TIME= 618.7900390625
ENGERY CONSUMED IN THIS CYCLE(W-HR) 36.23682349149577
AVERAGE POWER FOR THIS CYCLE(W) = 210.8187855884483

TOTAL ENERGY CONSUMED (W-HR)= 179.8660933674351

****

TIME = 02:00:36
ITERATION #= 6
TRi = 84.72
TFZi = 6.44
TFFi = 35.75
TRf = 85.1
TFZf = -3.41
TFff = 30.52
COMPRESSER RUNING TIME= 616.2099609375
ENGERY CONSUMED IN THIS CYCLE(W-HR) 36.26959039954291
AVERAGE POWER FOR THIS CYCLE(W) = 211.8929159140901

TOTAL ENERGY CONSUMED (W-HR)= 216.1356837669782

****

TIME = 02:22:36
ITERATION #= 7
TRi = 84.97
TFZi = 6.55
TFFi = 35.83

TRf = 84.97
TFZf = -3.43
TFff = 30.49
COMPRESSER RUNING TIME= 616.76025390625
ENGERY CONSUMED IN THIS CYCLE(W-HR) 36.11123409374999
AVERAGE POWER FOR THIS CYCLE(W) = 210.7795402089262

TOTAL ENERGY CONSUMED (W-HR)= 252.2469178607282

****

TIME = 02:44:32
ITERATION #= 8
TRi = 84.89
TFZi = 6.53
TFFi = 35.79
TRf = 85.01
TFZf = -3.49
TFff = 30.36
COMPRESSER RUNING TIME= 612.140625
ENGERY CONSUMED IN THIS CYCLE(W-HR) 35.6510202436659
AVERAGE POWER FOR THIS CYCLE(W) = 209.6637073829192

TOTAL ENERGY CONSUMED (W-HR)= 287.8979381043944

F I N A lllllllll !!!!!!!!!!!!!!!!!!!!!!!!!!!!!

N O T E   I T   S T O P E D   W H I L E
C O M P R E S S O R   W A S   O F F

TIME =03:00:00
ITERATION NUMBER = 9
TOTAL ENERGY CONSUMED (W-HR)= 302.4839684510741