CHAPTER 1 Introduction

1.1 Brief History

The fact that a volatile fluid chills the skin when it evaporates has long been known. The principle of refrigeration is built on this observation, and refrigeration cycles have been invented to exploit this effect. The knowledge of turning vapors or gases into liquids by compression followed by condensation was being gathered during the second half of the 18th century. J.F. Clouet and G. Monge liquefied sulfur dioxide in 1780, and ammonia was liquefied by van Marum and van Troostwijk in 1787 (Gosney, 1982). The idea of putting together the principles of refrigeration by evaporation and liquefaction by compression seems to have been first suggested by Oliver Evans of Philadelphia, but it remains unclear whether he had tried it or not. The first complete description of a refrigerating machine working in a cycle and its subsequent manufacture was given in a patent specification by Jacob Perkins, an American inventor working in London. This machine is the prototype of all subsequent vapor compression systems.

The Domelre, manufactured in Chicago in 1913, was the first functional household refrigerator. The Frigidaire trademark appeared one year later in 1919. Until 1916, the United States had a choice of a dozen models of costly and unreliable appliances. Most were composed of two quite distinct parts, which were an insulated casing cooled by brine coils and a chilling unit often placed in the basement. These units used sulfur oxide and oxygen as the refrigerant. An American by the name of Nathaniel Wales designed a device that was widely marketed under the name of Kelvinator starting in 1918. In 1931, the R12 machine was developed, but it did not enjoy immediate commercial success. In 1926, General Electric, an American company, manufactured a hermetically sealed household unit and, in 1939, it introduced the first dual-temperature refrigerator (Giscard d'Estaing, 1986). This allowed frozen food to be preserved in one of the unit's compartments.

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1.2 Environmental Concerns

Cooperation for the protection of the stratospheric ozone layer began on the global scale with the negotiation of the Vienna Convention for the Protection of the Ozone Layer, which concluded in 1985. The details of the international agreement were defined in the Montreal Protocol on Substances that Deplete the Ozone Layer. The Montreal Protocol was signed in September 1987 and became effective in 1989. It contains provisions for regular review of the adequacy of control measures that are based on assessments of evolving scientific, environmental, technical, and economic information.

At a meeting in London in 1990, the Parties to the Montreal Protocol agreed to a phase out of controlled substances. Another meeting of the Parties held in Copenhagen in 1992 accelerated the phase out schedules of the controlled substances like chlorofluorocarbons (CFC), halons, carbon tetrachloride, methyl chloroform, hydrochlorofluorocarbons (HCFC) and methyl bromide. In addition to the Montreal Protocol, other bodies such as the U.S. Environmental Protection

Agency and the European Community have imposed still more strict regulations and phase out schedules.

1.2.1 Safe Refrigerants

Until the early 1990's, the majority of refrigerators were using R12 (CCl_2F_2), a CFC, as the refrigerant. The problems with CFCs are not limited to their ozone-depleting capabilities, but also extend to their effects on the global climate. In fact, their contributions to global warming come in only second to those of CO_2, which accounts for 80% of greenhouse gas emissions in the United States. To overcome this problem, the use of HCFCs and hydrofluorocarbons (HFC) has been advocated as replacements for CFCs as refrigerants and foam blowing agents. Among the group of refrigerants that are recognized today as environmentally acceptable, the chemical compound tetrafluoroethane or R134a (CF_3CFH_2) seems to be one of the more promising candidates, and its use has since been widely adopted. Although HFCs such as R134a are less of a threat to the ozone layer, they have increasingly fallen into disfavor because they are greenhouse gases that have long atmospheric lifetimes. Should HFCs fall victim to environmental concerns, the refrigerant of the future would probably be hydrocarbons like propane or isobutane. Even though hydrocarbons are typically 4-5% more efficient than R12, their main advantages lie in the fact that they neither destroy the ozone layer nor contribute to global warming. At present, European refrigerator manufacturers have already switched to hydrocarbons but local manufacturers still remain hesitant, but for a good reason. Because hydrocarbons are flammable, many vendors are still concerned with potential lawsuits from refrigerator-related household fires. In addition, the increased risks associated with the storage and transportation of hydrocarbons have also compounded the problems of investing in them. To reduce the risk of fire to homeowners, refrigerants can be hermetically sealed. Although this reduces the chances of leakages, there is always a remote possibility of a fire arising from a leak into the defrost heaters.

1.2.2 Friendly Insulation

R11, an insulation-blowing agent that contains CFC, was once used to blow polyurethane foam into refrigerators. Now, the use of a HCFC, R141b, as a blowing agent has been widely adopted. However, the presence of chlorine in this chemical has also raised questions on its role as an environmental hazard. In fact, the use, manufacture or import of R141b would be illegal after the year 2003. At present, R245fa, a HFC, seems to be the most likely substitute based on its performance in laboratory tests. Since R245fa is still potentially dangerous to the environment, the future goal is to insulate refrigerators with non-ozone depleting materials. As a result, a considerable amount of time and funds have been poured into the research of vacuum insulation due to its tremendous promise. With so much focus drawn to this subject, the talk of insulating refrigerators with a combination of vacuum insulation and environment-friendly blowing agents is starting to lend some credibility. Although this combination is expected to be very efficient, issues on its cost and reliability will have to be resolved first before this innovative technology can be successfully implemented.

At Oak Ridge National Laboratory (ORNL), measurements have showed that the thermal resistivity of vacuum insulations exceeds that of the conventional by a factor of 3-7 times (ORNL, 1997). At present, the two types of vacuum insulation being actively tested at ORNL are the powder-evacuated panels (PEPs) and an insulation that contains fibrous glass. In vacuum insulations, the ceramic or metallic powder or fibrous glasses is sealed in evacuated envelopes. A cross section of the future refrigerator would reveal an outer steel skin, followed by a 1" vacuum insulation, then a1-2" ozone safe chemical foam and finally a plastic inner wall.

For all the benefits that vacuum insulation may offer, reliability remains the key issue. The lack of durability may lead to the development of holes, allowing air to leak in. Even in the absence of any crack, the destruction of the vacuum is also possible by the diffusion of air molecules across the plastic envelopes. Such is the case in some Japanese refrigerators, where the vacuum insulations have been known to lose their vacuum within the first year of service. Since refrigerators are typically expected to have a lifetime of 15-20 years, the problems faced here are of great concern.

1.2.3 Energy-Efficient Refrigerators

There are several reasons that persuaded the DOE to impose energy standards on refrigerators. The most obvious among them are the direct savings that consumers would realize from smaller energy bills. Besides that, these measures would not only be a partial solution to the problems of an extreme reliance on imported oil, but would also help utilities avoid risky capital investments in power plants to meet the escalating demands of energy users. However, the most compelling reason to curb the demand for electricity lies in its total impact on the environment. Fossil fuels used for power generation are large contributors of CO_2, the greenhouse gas responsible for global warming. Since energy use in buildings accounts for 36% of CO_2 production in the United States, the DOE has again put refrigerators on a strict energy diet due to their impact on the utilities (ORNL, 1997).

However, the widespread use of some CFC alternatives has also resulted in larger energy consumption, which indirectly increases the production of CO_2. Clearly, the introduction of these substitutes has both a direct advantage from an environmental standpoint but also an indirect disadvantage from its impact on energy consumption and emissions. This combined effect, which is called the total equivalent warming impact (TEWI), is the subject of hot debate among leading researchers in the field. As Sand (a researcher from ORNL) puts it, "The direct effect on global warming of a refrigerant leaking from refrigerators is less than the indirect effect on global warming of carbon dioxide from their energy use. For leaking automobile air conditioners, the direct effect of the leaks on global warming is larger than the indirect effect of burning gasoline. But, for refrigerators, the indirect effect of consuming electricity inefficiently from fossil fuel plants is much larger than the direct effect of refrigerant leaks. So, for environmental reasons, emphasis should be placed on improving the energy efficiency to reduce carbon dioxide releases." (ORNL, 1997) As this quote suggest, the net impact on global warming would be greater if the performance of refrigerators should suffer as a result of the change to alternative CFC chemicals for insulations and refrigerants. In order to achieve a positive outcome, the switch from CFCs to environment-friendly chemicals should necessarily be accompanied by an enhancement in the performance of refrigerators.

1.3 Motivation for Research Project - Energy Efficiency Standards

Refrigerators are a common feature in most modern homes. As an aggregate, residential refrigerators are large users of electrical energy, which the larger units consuming as much as 1,200 kWh of energy annually.

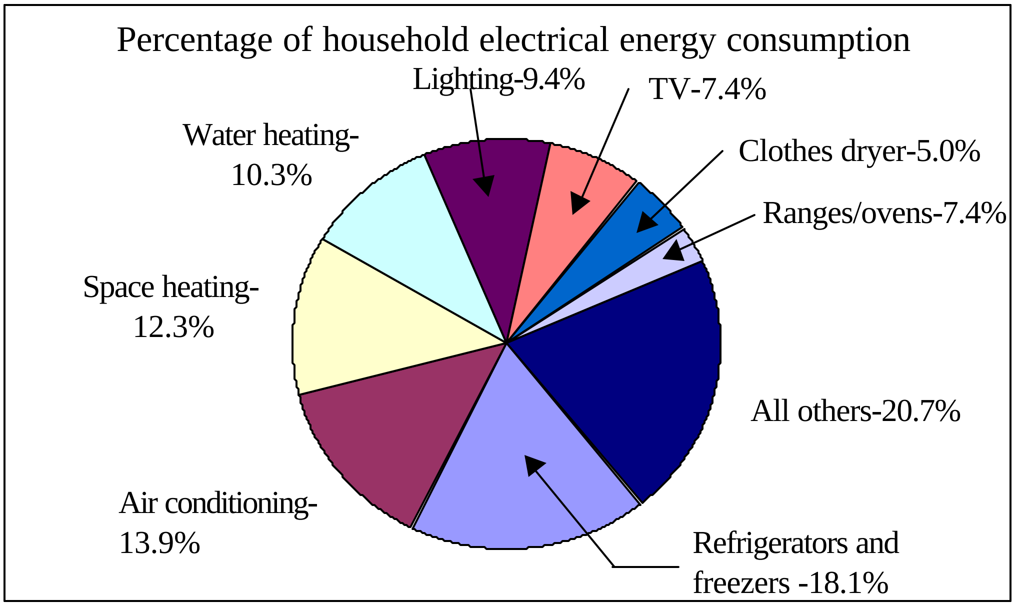


Figure 1.1 Distribution of household electrical energy consumption in a household (Reproduced from data accumulated by EIA, 1995).

As Figure 1.1 shows, refrigerators are typically among the largest consumer of electrical energy in an American home. In the United States alone, over 8.5 million refrigerators, which range from 1.7 cubic feet compact models to the 30.1 cubic feet side-by-side models, are sold annually (DOE, 1997). There are many of these appliances in use and they operate more or less continuously, so they have a significant impact on the utilities. In an effort to reduce their energy consumption, the Federal Government has enacted minimum performance standards administered by the US Department of Energy. These standards, first adopted in 1990, require progressive improvement in energy efficiency. The current standards went into effect in 1993.

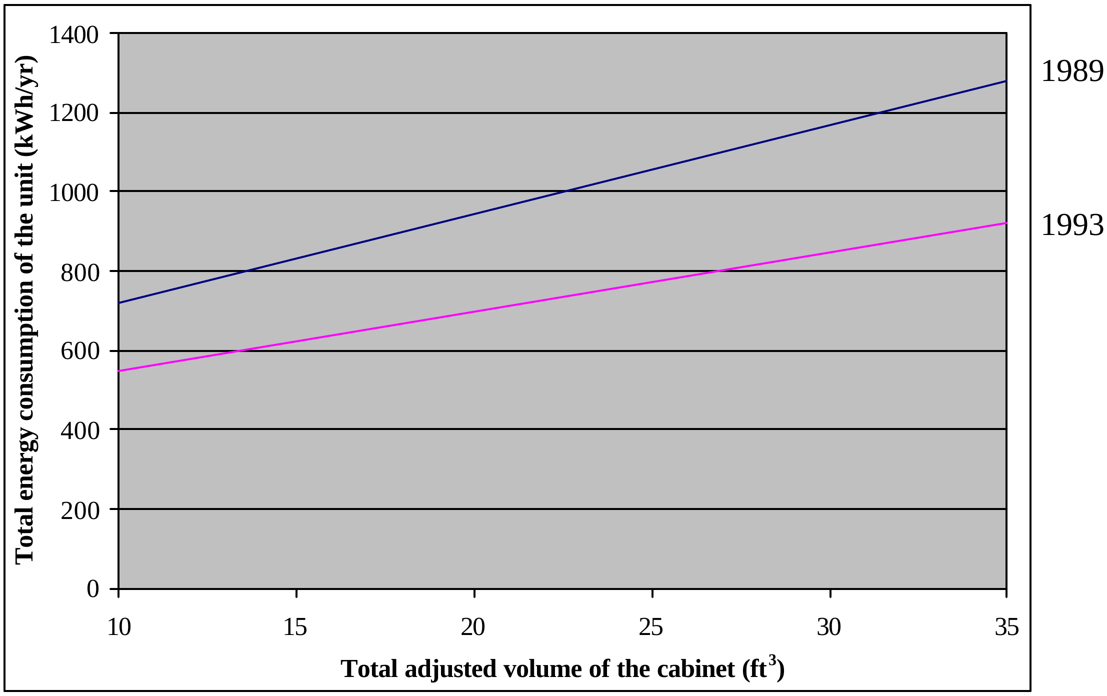


Figure 1.2 The average energy consumption of the refrigerator models in 1989 and 1993 (Reproduced from data accumulated by Lawrence Berkeley National Laboratory-LBNL, 1995).

As Figure 1.2 shows, there was quite a significant decrease in energy consumption between the models that were produced before the 1990 and those after the 1993 standards. In fact, refrigerators have made tremendous improvements over the last 25 years. In energy terms, an average new auto-defrost refrigerator with top mount freezer used about 2000 kWh/yr in 1972. A new unit in 1990 used about 900 kWh/yr, and in 1993 about 690. After 2001 (supposed to be 1998), a new unit will consume less than 500 kWh/yr (LBNL, 1995).

Newer standards which require a further 17-30% reduction in energy consumption relative to the 1993 values were supposed to come into force in 1998, but have been delayed for another 3 years. The 2001 standards do not call for an equal percentage reduction for all manufacturers over their current models, but employs a formula which uses the total adjusted volume of cabinet space to calculate the new common value that must be attained for each model, as shown below. As such, manufacturers which were able to produce models that far exceeded the 1993 standards were in a better position than their counterparts who had barely made the target.

Product class	**Energy standards equations (kWh/yr)**	
	1993	*1998 (postponed to 2001)*
Refrigerator-freezers -- automatic defrost with top-mounted freezer without through-the-door ice service	16.0AV + 355	9.80AV + 276.0
Refrigerator-freezers -- automatic defrost with side-mounted freezer without through-the-door ice service	11.8AV + 501	4.91AV + 507.5
Refrigerator-freezers -- automatic defrost with bottom-mounted freezer without through-the-door ice service	16.5AV + 367	4.60AV + 459.0
Upright freezers with manual defrost	10.3AV + 264	7.55AV + 258.3
Upright freezers with automatic defrost	14.9AV + 391	12.43AV + 326.1
Chest freezers and all other freezers except compact freezers	11.0AV + 160	9.88AV + 143.7
Compact refrigerators and refrigerator-freezers with manual defrost	13.5AV + 299	10.70AV + 299.0
Compact refrigerator-freezer – partial automatic defrost	10.4AV + 398	7.00AV + 398.0
AV = Total adjusted volume expressed in ft^3 = $Volume_{refg} + 1.63Volume_{frez}$		

Table 1.1 The energy standards that each model has to meet by the designated date (DOE, 1997).

The two largest classes, the top mount auto defrost refrigerator-freezer without through-the-door features and side-by-side refrigerator freezers with through-the-door features, require efficiency improvements of 29.6% and 29.3%, respectively. Together, these two classes account for 78% of the energy used by refrigerators and refrigerator-freezers and 57% of all refrigerator products including freezers (DOE, 1997). In addition to the models listed in Table 1.1, DOE has a comprehensive list of many other models that are subjected to a similar set of energy standards.

Aside from the different requirements for the various models, HCFC R141b-free refrigerators are also subjected to a different set of standards. In an effort to discourage the use of HCFCs as blowing agents, manufacturers which refrain from their use have been allowed an additional six years to meet the same target. To allow for the presumed energy penalty of replacements for R141b, DOE has proposed a 10 percent relaxation of the otherwise applicable standards for HCFC-free products for a period of six years after the effective date of the new standards. Already, a prospective candidate for this replacement is in sight. R245fa, a HFC, has fared well in laboratory test with results indicating a meager 0.9% reduction in performance as compared to R141b. While some uncertainties still remain, Allied Signal has announced that it will begin commercial production of R245fa in 1999. As of February 1997, it expects that appliance manufacturers will begin converting as early as 1999 and to complete their conversion a year later.

As a result of these standards, the DOE estimates that manufacturers will attach and additional $80 tag to the products they carry. Since the typical consumer is projected to save $20 annually from the more efficient refrigerators, the payback time for these products should be around 4 years. Over a 19-year period, consumers should experience a net saving of $300. More importantly, these standards is expected to reduce CO_2 and NO_x emissions by 513 million short tons and 1.5 million short tons respectively over the next 30 years (DOE, 1997).

1.4 Objective of Study

In the past, various methods have been used to design a more efficient refrigerator. Among them, one of the most effective methods is to reduce the cabinet load. This is accomplished by increasing the thickness of the insulation on the walls of the refrigerator. Even though this remains one of the most effective approaches, it decreases the available storage space. Other than the thickening of refrigerator walls, the installation of an anti-sweat switch (to allow the user to turn off the electric heaters when the relative humidity is low), a better sizing of heat exchangers, improvements in compressor technology and the use of delayed defrost have all contributed to system improvement to meet the 1993 targets.

Since the 2001 standards require yet another significant improvement, new measures have to be explored. They include the use of better door gaskets to reduce heat leaks, a further thickening of the walls, the use of DC fans and even dual-evaporator refrigerators and two-speed compressors if budget permits and reliability issues can be overcome.

This research project involves a collaboration of effort with a major manufacturer of high-end household refrigerators. The overall objective of this research is to propose and demonstrate one or more design changes to a current refrigerator unit that will cost-effectively meet the target.

Several energy efficiency measures will be investigated in detail to assist the manufacturer in identifying product designs that will meet the proposed minimum performance standards. The feasibility of proposals such as mechanical subcooling, two-stage cycles and a study on the optimum sizing of the heat exchangers was explored through the development of a computer model which aims to simulate the actual performance of a refrigerator. Since the model calls for the input of heat exchangers UAs, experiments were performed to determine the heat exchanger conductance values. In addition to the proposals mentioned above, the use of suction-line heat exchangers to increase the refrigeration capacity and mullion tubes to prevent sweating, which have already been implemented in this refrigerator, will also be studied. The most promising of the concepts identified will be submitted to the manufacturer and possibly fabricated and installed so that the performance of the modified refrigerator can be measured.

1.5 More About the Refrigerator Under Study

The refrigerator that is being studied is a side-by-side model. The refrigerator compartment has a storage capacity of 18.8 cubic feet while the size of the freezer compartment is 11.3 cubic feet. The unit stands 7 ft tall, and has a width of 3 ft and is 2 ft deep.

In most refrigerators, refrigeration is only performed at the freezer evaporating temperature to meet the cabinet loads. Unlike conventional designs, this refrigerator employs two separate cycles to provide cooling for the freezer and refrigerator compartments. Using this arrangement, refrigeration can be performed at a higher temperature for the refrigerator, which enhances the performance of the system. Based on the present cooling loads of the freezer (276 Btu/hr) and refrigerator (197 Btu/hr), the use of two cycles to meet the loads of the cabinets results in a 14% savings in energy over the conventional one-cycle approach (This calculation is based on a freezer and refrigerator evaporating temperature of –10°F and 20°F respectively, with the refrigerant condensing at 105°F. These values reflect the actual operating conditions in this refrigerator. The analysis does not consider the fact that smaller compressors which have lower EERs would be required in the two-cycle design to avoid excessive cycling losses). Both the freezer and refrigerator cycles use R134a as the refrigerant.

While a higher COP is desirable, the disadvantage of using two cycles lies in the fact that two compressors, evaporators, expansion devices and two suction-line heat exchangers are needed in this approach, one set for each cycle. However, the choice of using one or two condensers is purely optional. As the source of heat rejection, it is not essential to have a condenser for each cycle. The present design only employs one condenser, which is responsible for rejecting heat from both cycles to the surroundings (the refrigerants from both cycles do not mix in the condenser). Because both cycles are individually controlled to meet the respective cabinet setpoint temperatures, their operations are entirely independent. As such, both cycles have exclusive use of the condenser when they run separately and are forced to share this heat exchanger when they run simultaneously.

In the present design, the operation of the condenser fan is triggered when either or both of the compressors are on. A fan is also used in both the freezer and refrigerator evaporators, and is turned on when the compressor in that cycle operates.

CHAPTER 2 UA Measurement

2.1 Introduction

Many engineering processes rely on the ability of devices to transfer energy between two fluids at different temperatures. Devices used to implement these processes are called heat exchangers and specific applications may be found in air-conditioning, power generation, heat-recovery and in a variety of chemical processes. Heat exchangers form the backbone of a vapor-compression cycle, with the condenser and the evaporator being the devices that allow the exchange of heat between the refrigerant and the fluid that it communicates with.

The performance of a heat exchanger is often characterized by its effectiveness, which is a measure of its ability to transfer heat between the two contacting fluids. The overall heat transfer-area product, or UA, is a useful parameter that allows the computation of the amount of heat transfer that occurs when the temperatures of the fluid entering and leaving the heat exchanger are known. One of the goals of this research was to develop a computer program to simulate the current performance of the refrigerator to enable the evaluation of potential energy-saving proposal modifications. Among others, this program calls for the input of the condenser and evaporator UAs. Since the actual value of this parameter was not known, efforts were made to experimentally measure the UA of the present heat exchangers in the refrigerator. Ultimately, the goal is to determine the UA that optimizes the performance of the system (Chapter 4 is dedicated to this study).

2.2 Experimental Setup

Both the refrigerator and freezer evaporators as well as the shared condenser are counter-cross flow heat exchangers. While the refrigerator evaporator only has one row of tubes, the evaporator for the freezer cycle has twice that number to promote the amount of heat exchange that it performs. In contrast, two rows of tubes are dedicated to the refrigerator while the freezer employs only one row in the condenser. Figures 2.1 and Figure 2.2 are pictures taken of the freezer and refrigerator evaporators respectively.

To determine the UA, the temperature of the air and refrigerant were measured at the inlet and outlet of the heat exchanger. Thermocouples were attached to the copper tubes by tape to measure refrigerant temperature while ties were used to strengthen the joint. For every thermocouple located at the front face of the heat exchanger, there was another located on the same spot directly behind the heat exchanger so that the difference in the air temperature across the heat exchanger could be obtained. Initially, only 2 pairs of thermocouples were used, but this number was eventually increased to 8 for the freezer evaporator and condenser (Figure

2.3a) and 5 for the refrigerator evaporator (Figure 2.3b) when abnormal temperature distributions across the face of the heat exchangers were observed.

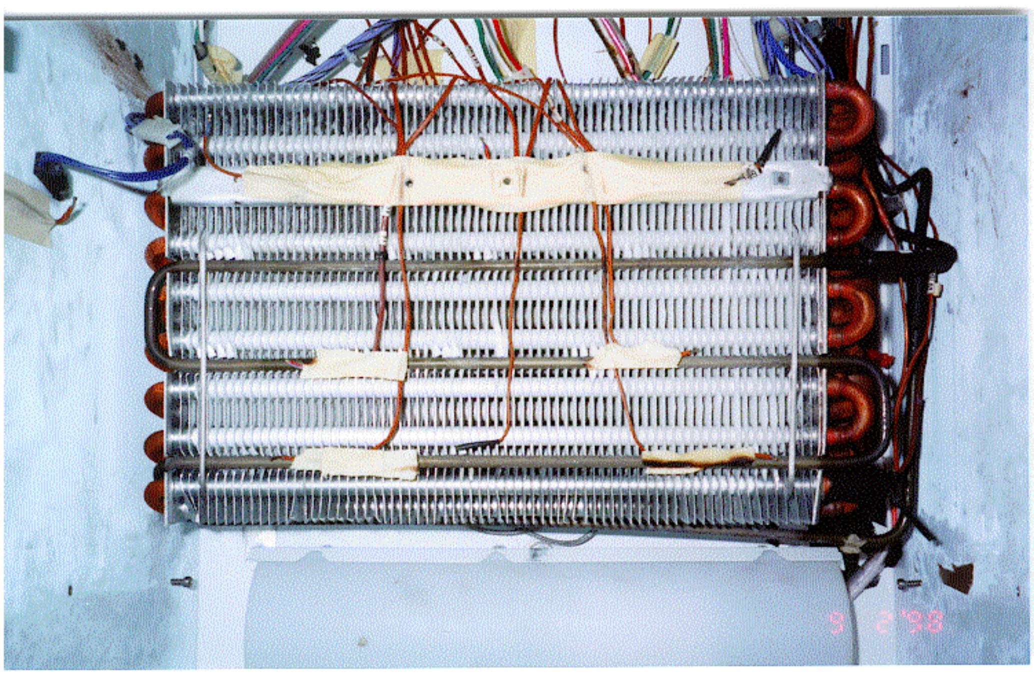


Figure 2.1 The freezer evaporator.

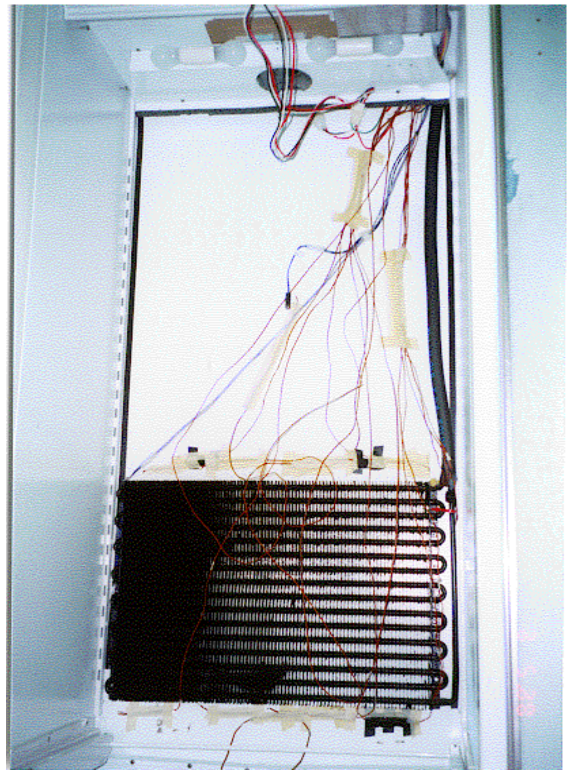
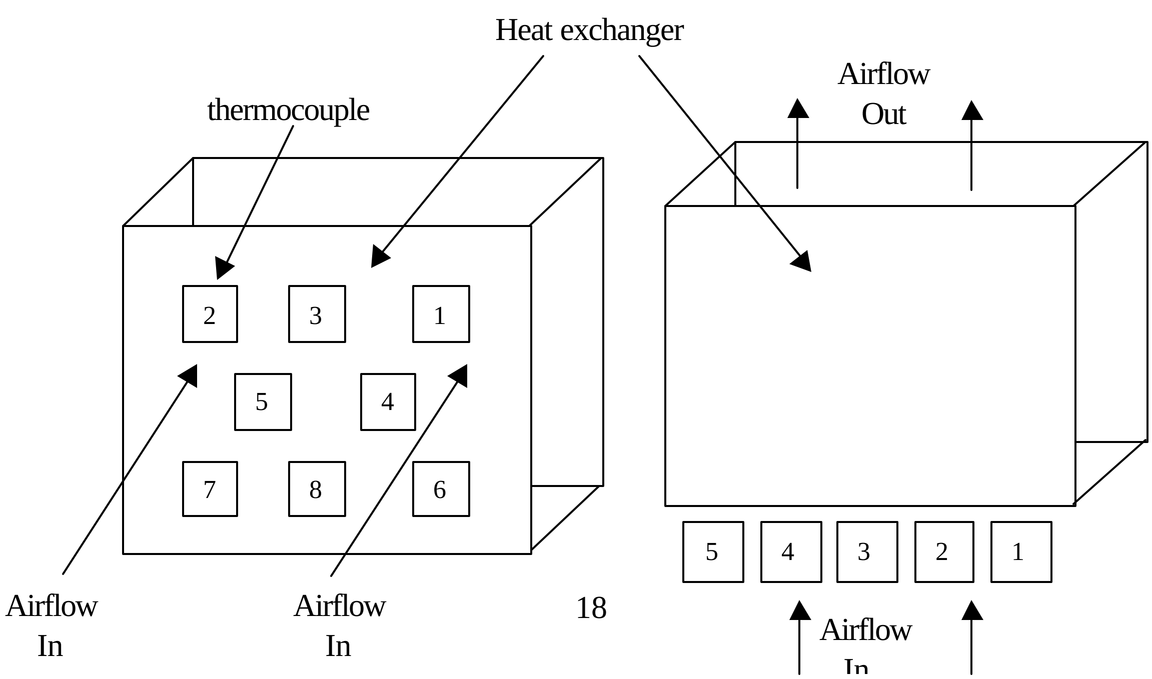


Figure 2.2 The refrigerator cabinet with the evaporator located at the bottom.

Before the thermocouples were attached to their respective locations, they were calibrated twice, once in an ice bath (Figure 2.4) and another at the temperature of the 90°F environmental chamber.



(a) (b)

Figure 2.3 Locations of the thermocouples in (a) freezer evaporator and condenser (b) refrigerator evaporator.

In addition to the temperature measurements, the low and high-side pressures of the system were tapped by pressure transducers at the outlet of the heat exchangers. The thermocouples and the pressure transducers were all connected to a data acquisition system from National Instruments (Figure 2.5). Output from these measurements were digitally recorded by Labview and saved into a file on a PC. The power consumed by the compressor and fans were also measured in a similar fashion.

Figure 2.4 The calibration of the thermocouples in an ice bath.

Figure 2.5 The station that the thermocouples and pressure transducers were connected to.

The airflow rate through the heat exchangers was measured by the apparatus shown in Figure 2.6. This apparatus was equipped with a fan that supplied the system with air from the surroundings. The air that was drawn by the fan was separated from the pipe (that delivered the air to the heat exchanger) by an aperture. To regulate the amount of air that was allowed into the system, the size of this aperture can be controlled by turning a dial on this apparatus. The amount of air that flows into the pipe is measured by a venturi meter. Before any airflow measurements were performed, the static pressure of the system was equalized and the fluid level of the fluid in the manometer meter was marked. When the fan was turned on, the fluid level rose and the dial was turned to release air into the system until the level decreased to its original mark. The airflow rate supplied to the system was then read off a digital display on the apparatus. To ensure that air was properly channeled into the system, the cabinet was ducted as shown in Figure 2.7 so that the heat exchanger would be provided with only the ambient air from this apparatus.

The measurement of refrigerant mass flow was made by a mass flow meter from K-Flow (shown in Figure 2.8) installed at the discharge of the compressor. Although the calibration of this device was not performed here, it was previously calibrated using water and found to be accurate to within 1%. The tubes carrying the superheated vapor to and from this device were also well insulated to reduce the amount of heat loss to the environment. Figure 2.9 depicts the location of where the temperatures, pressures, airflow and mass flow measurements were made (also note that thermocouples were placed at the entrance and exit of the suction-line heat exchanger to determine its effectiveness).

Instead of the conventional AC fans, the freezer and refrigerator evaporators were equipped with DC fans. To control the speed, a variable voltage controller was placed between the electrical source and the fans so that the voltage may be regulated. The speed of the fan was adjusted to supply the freezer and refrigerator evaporators with 57 CFM and 25 CFM of air, respectively. The flow through the condenser was set at 124 CFM. These flow rates were chosen because they represent the actual conditions in the heat exchangers currently installed in the refrigerator.

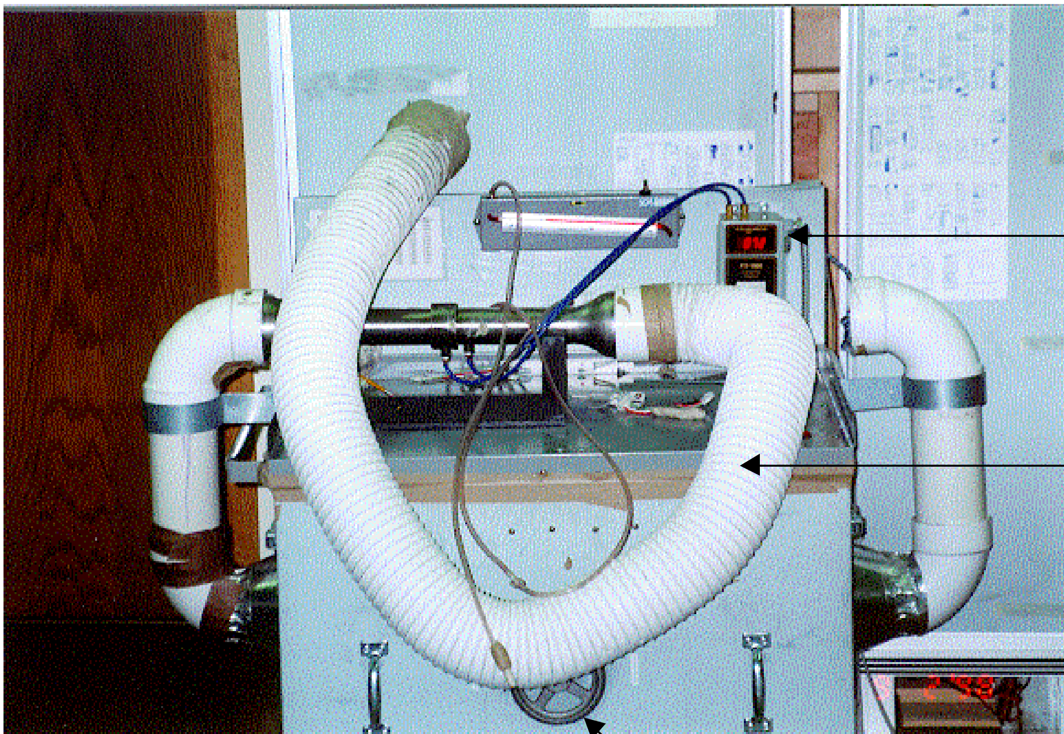


Figure 2.6 The apparatus that was used to measure the airflow rate.

Ducts to ensure that the heat exchanger was only supplied with the air from the apparatus

Figure 2.7 The air from the airflow rate apparatus is supplied to the evaporator through a duct.

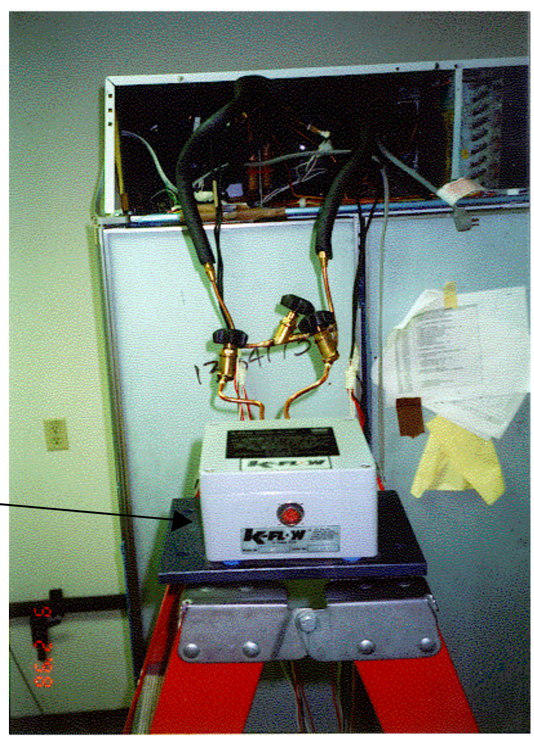


Figure 2.8 The mass flow meter connected to the system.

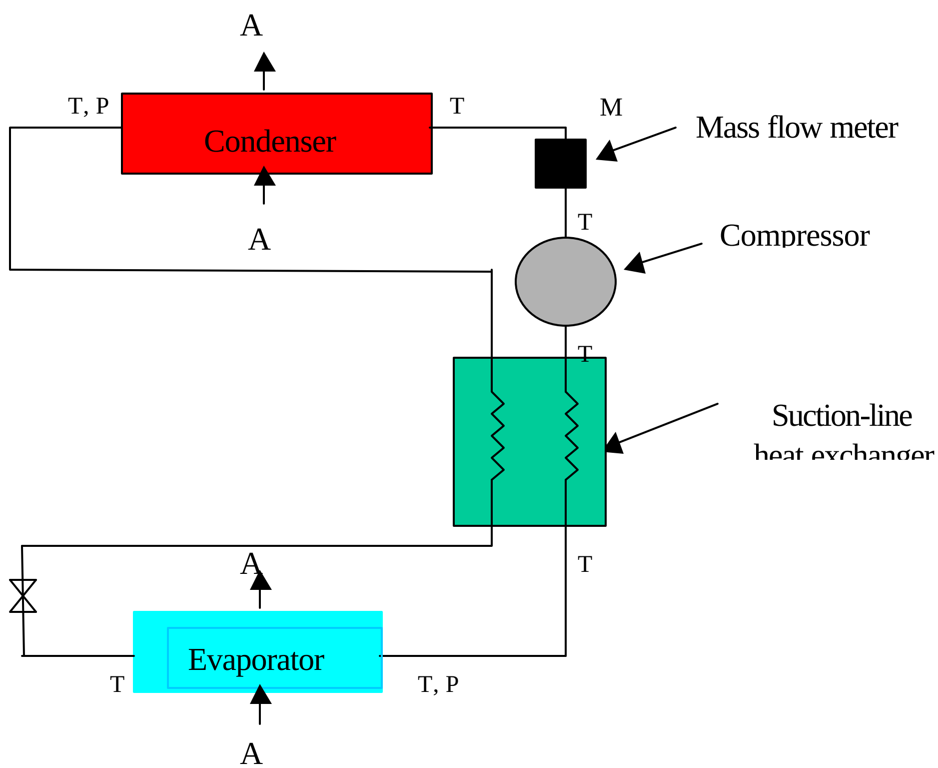


Figure 2.9 The placement of thermocouples (T), pressure transducers (P) and mass flow meter (M) in the system. Airflow measurements, designated as (A), were also made.

2.3 Experimental Procedure

To demonstrate that the refrigerator meets the energy standards, it should be tested according to the guidelines provided by AHAM (Association of Home Appliance Manufacturers). The test procedure calls for the unit to be subjected to two closed-door tests carried out in a 90°F test chamber. Most refrigerators are rarely placed in such severe environments, but these prescribed guidelines account for the effect of regular door openings that refrigerators experience when they are in use.

In the first test, the temperature controls in the freezer and refrigerator cabinets should be set to the middle position. The test is repeated by setting the temperature controls to the warm position. The energy reported to the DOE should be the higher of the two values that are obtained by a linear interpolation to an average freezer and refrigerator temperature of 5°F and 45°F, respectively. In conformance with these requirements, the measurements were made in a 90°F environmental chamber. Instead of measuring the freezer and refrigerator cabinet temperatures at both their middle and warm settings, the temperatures were set to 5°F and 38°F respectively. Since the doors of the cabinet were not open during the duration of the test, the humidity in the chamber had little effect on the results and was not controlled.

The readings from the mass flow meter had to be visually recorded during the test because this device was not connected to the data acquisition system. Although this was not possible during the night, the change in the mass flow rate as a function of time (during a cycle) was very consistent for every cycle during the day, hence giving no reason to believe that they would be any different at night.

To begin the test, the program connected to the Labview software was activated to enable the output scanned from that station to be recorded onto an MS Excel file. All the tests were performed over a 20-hour period. If any defrosting had taken place during this period, the test would be allowed to carry on for a longer period to enable the collection of more data. To terminate the test, the STOP button on the program was depressed so that the Excel file could be closed and saved by the application.

2.4 Analysis of Results and Discussion

2.4.1 Calibration of Thermocouples

Two calibration tests, one at 89°F (the environmental chamber was supposed to be kept at 90°F) and another in an ice bath, were performed before the thermocouples (TCs) were attached to their respective locations. The former was done by placing the TCs in the environmental chamber and leaving them over a 24 hour-period to ensure that it had come to equilibrium with the conditions of its surroundings. Owing to the greater rate of heat transfer, equilibrium was achieved in a much shorter period for the ice bath calibration. After only 1 hour, the TC readings were essentially constant.

With the exception of a few thermocouples, the observations had revealed that the reading for each TC was either consistently lower or higher than the reference temperature. In the analysis, the results obtained from the calibration were used to correct for the temperatures recorded by the data acquisition system. Using the results from the calibration tests, the measurements from the thermocouples were corrected to yield the actual air temperature as shown in Eqn 2.1.

$$T_{actual} = 32 - \frac{57\left(T_{TC,32} - T_{TC,T}\right)}{\left(T_{TC,89} - T_{TC,32}\right)} \qquad \text{Eqn. 2.1}$$

where $T_{TC,32}$ = Temperature of the thermocouple in the ice bath
$T_{TC,89}$ = Temperature of the thermocouple in the environmental chamber
$T_{TC,T}$ = Temperature of the air measured by the thermocouple
T_{actual} = Actual temperature of the air

2.4.2 Temperature Distribution Across Heat Exchanger

A temperature distribution was observed across the face of all the three heat exchangers studied. As a result, the number of TCs used for temperature measurements were increased to 8 pairs for the freezer evaporator and condenser while 5 pairs were deemed sufficient for the refrigerator evaporator. These distributions, shown in Figures 2.10-2.12, were very consistent for every cycle run during the tests. Note that the TC# on the legends correspond to the location of the TCs on Figure 2.3.

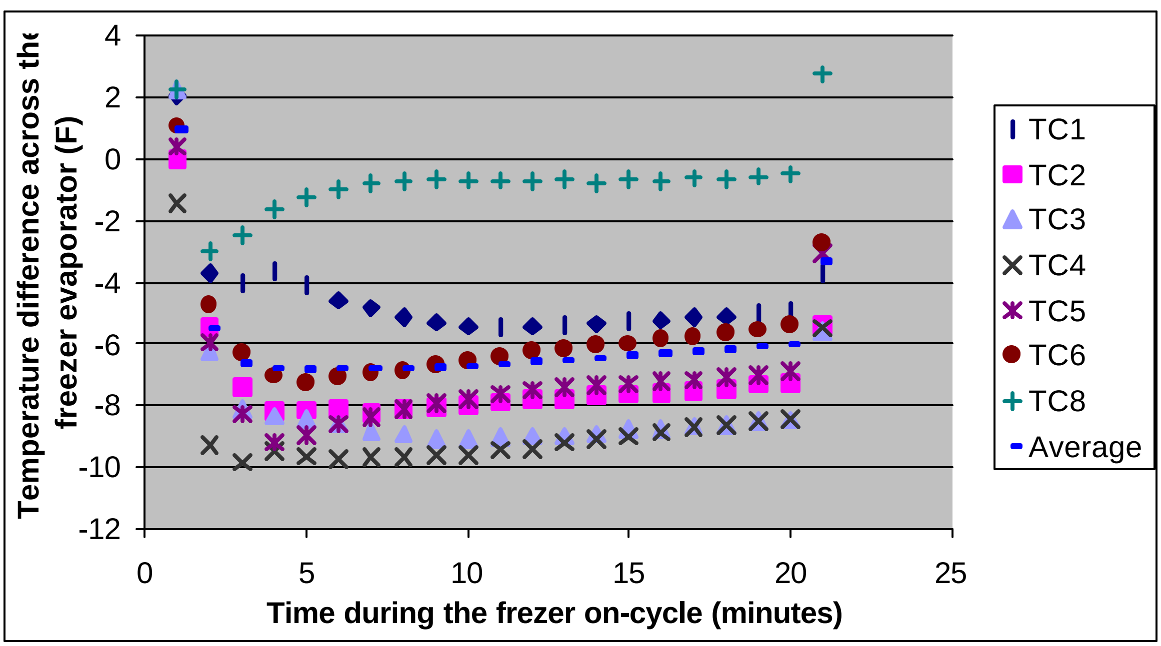


Figure 2.10 The difference between the air temperatures leaving and entering the freezer evaporator.

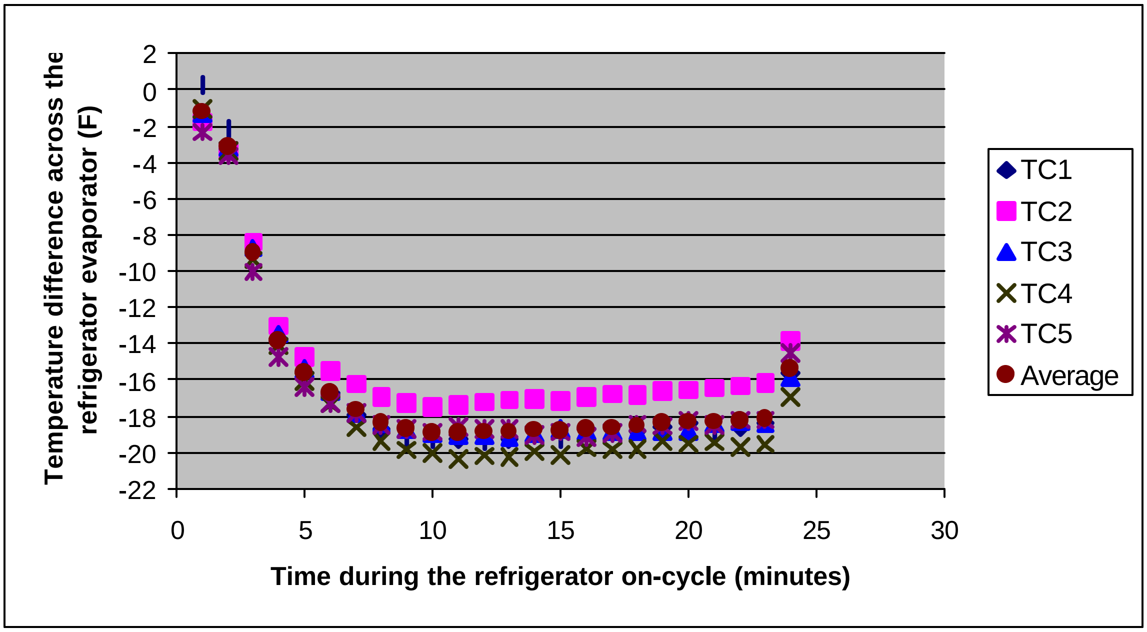


Figure 2.11 The difference between the air temperatures leaving and entering the refrigerator evaporator.

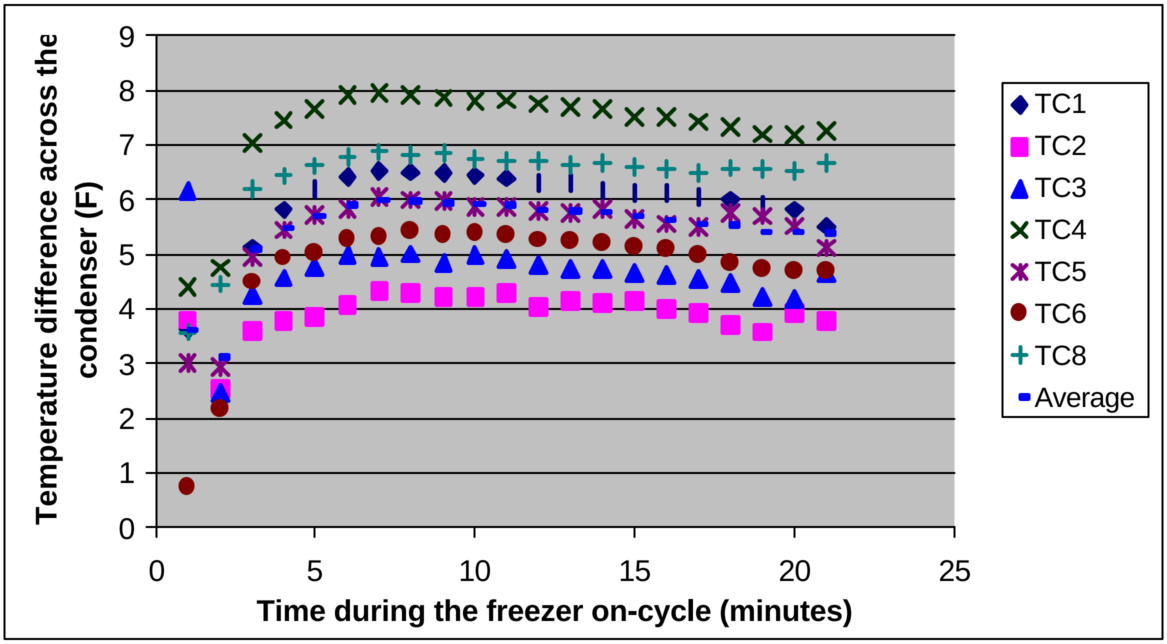


Figure 2.12 The difference between the air temperatures leaving and entering the condenser during the freezer on-cycle.

One of the 8 TC pairs used to measure the temperatures at the freezer evaporator was functioning improperly, and was therefore not included in Figure 2.10. Similarly, another pair of TC used for the condenser had returned erroneous results (The fact that pair no. 7 was faulty for both heat exchanger measurements was a mere coincidence. They were all different TCs). They were known to be faulty because the temperatures registered by their outputs were over 400°F, which was clearly impossible under the present conditions. The fact that the problem was caused by a faulty TC and not the recording channel in the station was confirmed when the latter was tested with a different TC and found to be in good working order. A dislocated wire is the most likely reason that may have caused the TC to become faulty.

Referring to Figure 2.10, the degree of cooling which the air experienced was a function of its location on the evaporator. While the average cooling was about 7°F, air passing through the lower middle part of the evaporator was hardly cooled. Located directly in front of the evaporator, the icemaker was probably the best explanation that accounts for the distribution of air temperatures. This is because it may have restricted the flow of air across certain parts of the evaporator, while forcing more air through other regions.

A smaller degree of dispersion was observed for the refrigerator evaporator and the condenser. Due to the absence of any object lying in its path, the flow of air is not hindered prior to its arrival at the refrigerator evaporator. For the condenser, the distribution may be explained by taking into account that the air has to make a 90° turn as it is drawn through the condenser, as shown in Figure 2.13. When the air is forced to execute this movement, the airflow across the face of the condenser is no longer uniform.

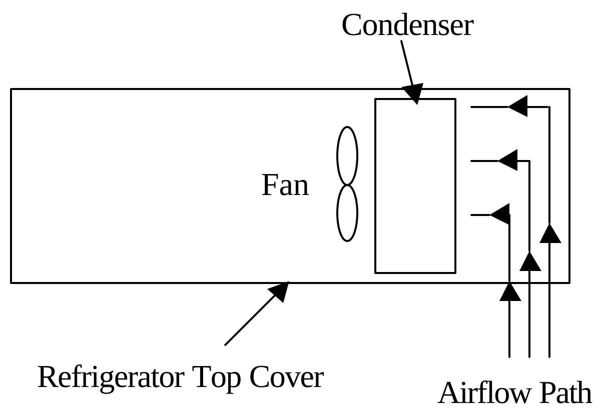


Figure 2.13 Top view showing the flow of air before it reaches the condenser.

Despite of the uneven flows and temperature distributions, the average temperature difference assumes a very smooth trend over time. In particular, the temperature difference decreases with time during the on-cycle, which is consistent with expectations that the capacity that the compressor delivers decreases as the evaporating temperature is lowered.

2.4.3 UA Calculations

The following discussion focuses on developing methods to experimentally determine the UA of the freezer evaporator only. To evaluate the UA, it is necessary to determine the rate of refrigerant flow in the cycle and the amount of heat transfer from the evaporator. For this purpose, an energy balance can be performed on the evaporator to determine the amount of heat transferred from the air to the refrigerant, as in Eqn. 2.2.

$$\dot{Q} = \dot{m}_{air}\, C_p \left(T_{air,in} - T_{air,out}\right) \quad \text{air-side energy balance} \qquad \text{Eqn. 2.2}$$

$$\dot{Q} = \dot{m}_{refg} \left(h_{refg,out} - h_{refg,in}\right) \quad \text{refrigerant-side energy balance} \quad \text{Eqn. 2.3}$$

With the temperatures of the air entering and leaving the evaporator, the airflow rate, and the temperature and pressure of the refrigerant entering the heat exchanger known, Eqn. 2.3, when used in conjunction with Eqn. 2.2, yields the rate of mass flow. However, the heat absorbed by the ambient air is also equal to that loss by the refrigerant in the condenser, which then provides another estimate of the refrigerant mass flow. Since the mass flow in the condenser and the evaporator should be equal, Eqn. 2.3 can be used to determine the amount of heat transfer in the evaporator. As a third alternative, the direct measurement of mass flow (hence the heat transfer rate) is also available from the meter located at the compressor discharge.

Clearly, three different methods could have been used to evaluate the rate of mass flow in the system, with each potentially predicting a different rate of cooling. Had the experiments been free of any uncertainties, the choice of method should be immaterial, and all three methods would yield the same results. The differences between the three methods, are, therefore, a result of experimental error. While Eqns. 2.2. and 2.3 apply for energy balance on both heat exchangers, the former is only accurate when all the cooling done in the evaporator is sensible. Sensible-only cooling is assumed after the initial cycles in the closed-door test as latent cooling is a consequence of air infiltration through the doors. When the rate of heat transfer is known by evaluating the equations above, the UA may be appropriately determined by the following equation.

$$\dot{Q} = UA \times \Delta T_{LM} \qquad \text{Eqn. 2.4}$$

where ΔT_{LM} refers to the log mean temperature difference, defined as

$$\Delta T_{LM} = \left[\frac{\left(T_{air,in} - T_{air,out}\right)}{\ln\left(\frac{\left(T_{air,in} - T_{refg,in}\right)}{\left(T_{air,out} - T_{refg,in}\right)} \right)} \right] \qquad \text{Eqn. 2.5}$$

Referring to the equation above, the temperature of the refrigerant entering and leaving the evaporator may be different due to superheating of the exit vapor. However, the use of the same refrigerant temperature for the inlet and outlet of the evaporator is appropriate since most of the heat transfer takes place at the evaporating saturation temperature.

The three methods discussed here were also employed to calculate the refrigerator and condenser UAs. The UA of the three heat exchangers are plotted in Figures 2.14-2.17 for a typical refrigerator or freezer cycle. Error bars (refer to Section 2.4.4) were included to indicate the amount of experimental uncertainties associated with each measurement.

In measuring the UA of the condenser, only the freezer and refrigerator cycles that ran separately were considered. While both Figure 2.16 and Figure 2.17 show the condenser UA measurements, the former was obtained using measurements recorded when only the freezer cycle was on, whereas the latter was constructed from data obtained when the refrigerator cycle was running alone.

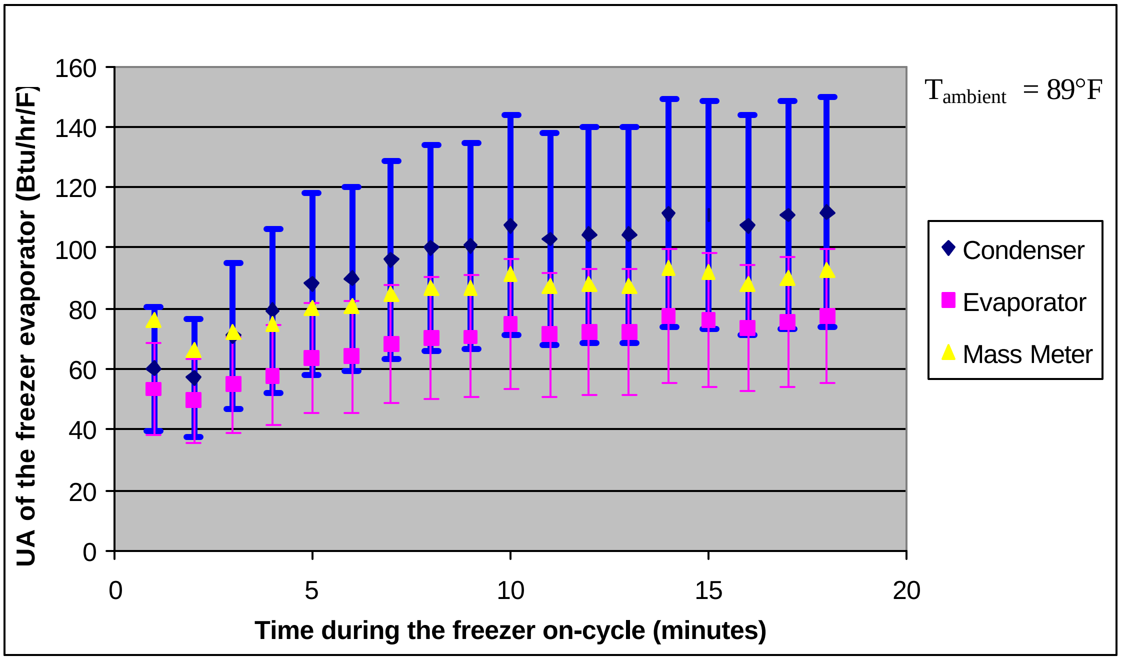


Figure 2.14 The UA of the freezer evaporator as a function of time calculated by three different methods.

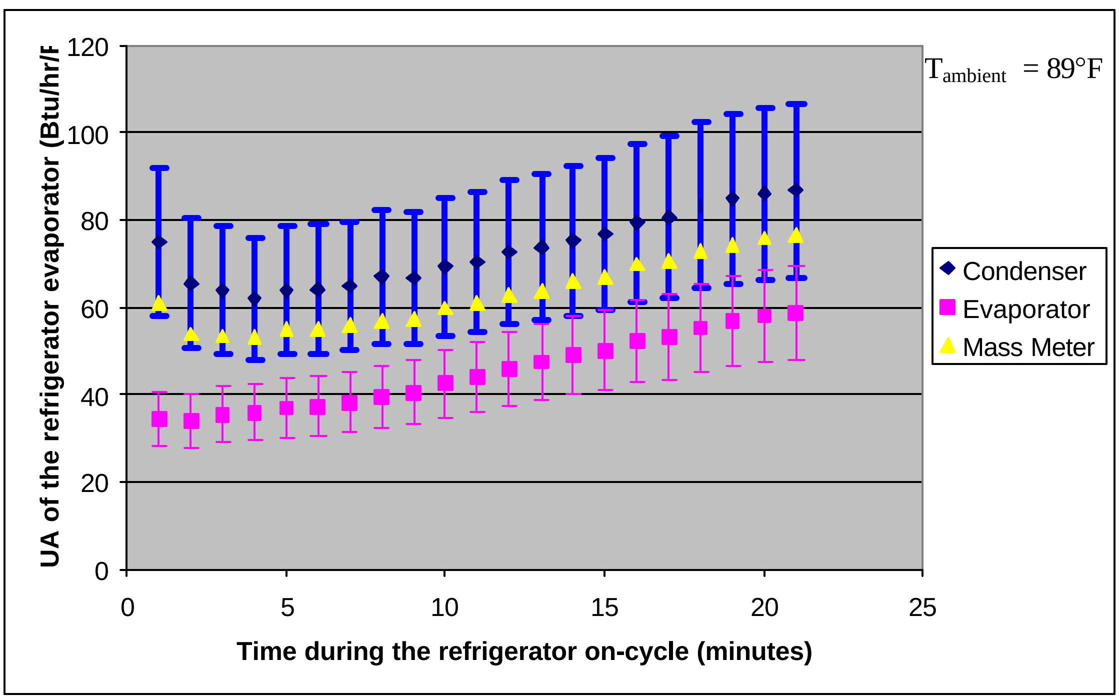


Figure 2.15 The UA of the refrigerator evaporator as a function of time calculated by three different methods.

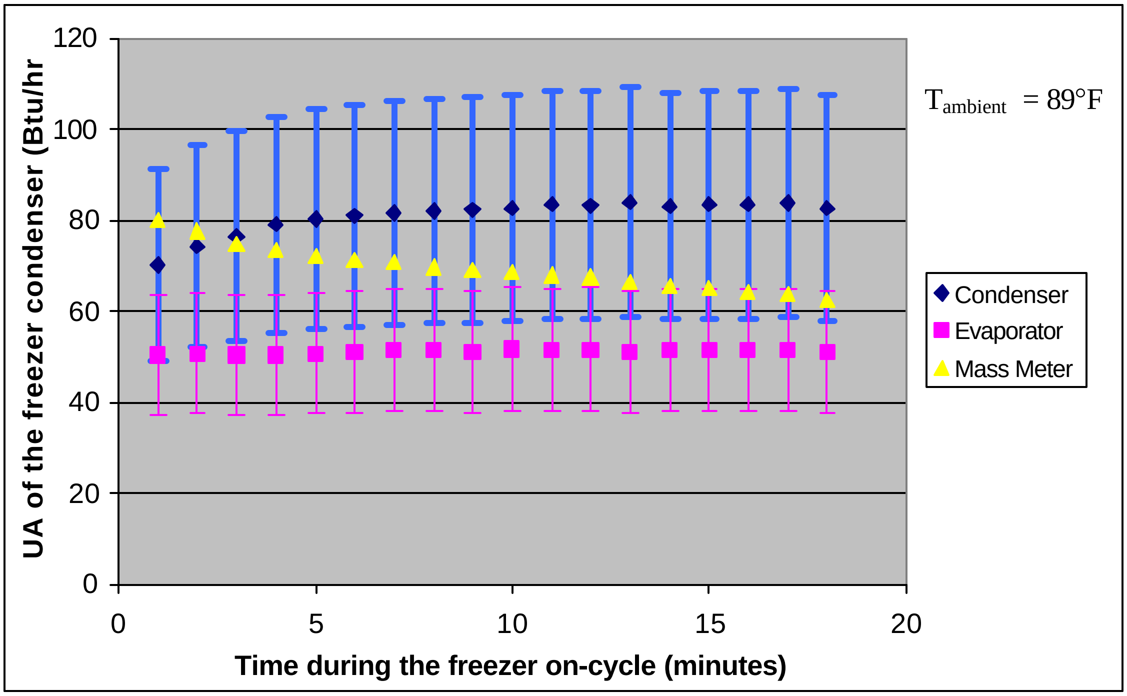


Figure 2.16 The UA of the condenser during the freezer on-cycle as a function of time calculated by three different methods.

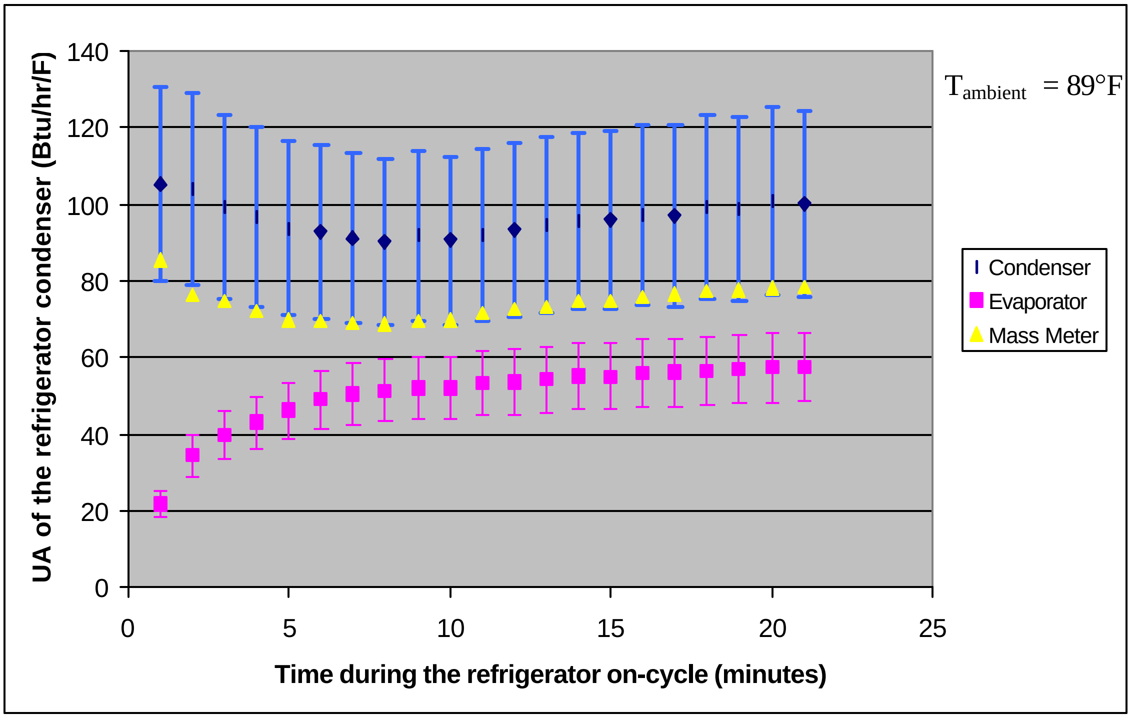


Figure 2.17 The UA of the condenser during the refrigerator on-cycle as a function of time calculated by three different methods.

It should be noted that the analysis for the initial two minutes of operation is typically discarded because of unrealistic log mean temperature differences (too small, causing the UA to be extremely large during the start-up transients). Under normal operating conditions, the freezer and refrigerator cycle on and off periodically. Due to this cycling nature, the cooling of cabinet air is a transient process. The fact that the UA had varied with time in Figure 2.14 and Figure 2.15 was just a reflection of the transient nature of the cooling process.

In both the freezer and refrigerator evaporators, the UA was seen to increase with time. To explain this observation, the change in the rate of heat transfer and ΔT_{LM} as functions of time must be clearly understood. When the compressor is turned on, refrigerant is pumped through the evaporator to initiate the cooling process. With the passing of time, the recirculated cabinet air returns at a lower temperature. To sustain the amount of heat transfer between the two mediums, the system responds by lowering the evaporating saturation temperature. As a result of the decrease in pressure, the vapor enters the compressor at a higher specific volume, and the mass flow rate is reduced. Since the capacity of refrigeration is directly proportional to the mass flow rate of refrigerant, it also decreases accordingly. Just as the capacity is a function of time, the change in air and refrigerant temperatures entering and leaving the evaporator with time will affect the ΔT_{LM}. With the temperature change exerting more influence on this parameter, the ΔT_{LM} decreases at a faster rate than the capacity. Because of the unbalanced change in the capacity and the ΔT_{LM} as a function of time, the UA experiences a net increase.

Signs of transient behavior were also evident for the condenser UA measurements, although this parameter was not a very strong function of time. In this case, however, it is not apparent that the UA increases with of time. Specifically, the calculations based on the mass meter measurements during the freezer on-cycle (Figure 2.16) had shown that the UA decreases with time.

2.4.4 Sources of Error

If the measurements were free of error, the exact method used for calculating the UA would not be an issue, and all the three curves in Figure 2.14 and Figure 2.15 would collapse to form only one for each evaporator. Similarly, the three curves in Figures 2.16 and 2.17 should also match if no experimental errors were encountered.

In a typical experiment, measurements often suffer from elements of uncertainties that stem from the inaccuracy of the measuring equipment or technique. Sources of error were encountered in every area, particularly in the temperature measurements. After performing calibrations at 32ºF and 90ºF and using Eqn 2.1 to correct for the temperature measurements, the majority of TCs were found to be only accurate to ±1ºF.

Besides the errors associated with the TC measurements, uncertainties were also present in the measurement of airflow and the relative humidity in the cabinet. The former may be the result of inaccuracy in the venturi meter and any air leaks that may have occurred in directing the airflow towards the heat exchanger. For systems with low flow rates, these leakages (on a percentage basis of total airflow) are significant, which is why measurements for the refrigerator evaporator are particularly vulnerable. In this analysis, the uncertainty of the airflow measurement was set at 10%.

The test was allowed to run for a minimum of 20 hours. To overcome the problem of measuring the relative humidity, only cycles that took place after the initial 2 hours of testing were considered in the analysis. Within that span of time, most of the moisture in the cabinet would have condensed and been drained out of the system after the first few cycles for a closed-door test, thus justifying the assumption of total sensible cooling.

Regardless of the condenser or evaporator UA measurement, the analysis had revealed that the approach using an energy balance on the condenser had yielded UA values that were consistently higher than those obtained by the mass meter or the evaporator. Apart from the uncertainty in temperature measurements, the leakage of air from the ducts may have partially accounted for the higher UA measurements when the condenser was used for analysis. When air leaks are present, the amount of air that actually flows through the heat exchanger is less than the value measured by the venturi meter. Because of this, the difference in the air temperature across the heat exchanger would be larger than if no leakages had occurred. With both the airflow and temperature difference recording higher measurements, the analysis would be prompted to overestimate the mass flow rate of refrigerant in the cycle. The UA, being directly proportional to the mass flow rate, would also increase accordingly.

The above explanation, however, goes against the observations of the lower UA measurements when the evaporator was the source of analysis. While the TC measurements remain questionable, the assumption of 100% sensible cooling in the evaporator is subject to contention. Latent loads, if present, would have increased the amount of cooling that was actually performed and hence the refrigerant mass flow too.

Since the measurements from the mass flow meter were assumed to be accurate, the extent of the differences between the three methods of analysis provides an indication of the degree of uncertainties in the measurements. As Figures 2.10-2.12 show, the difference between the entering and exiting air temperature was larger for the refrigerator evaporator than the condenser. It follows, then, that the uncertainties associated with the measurements are inherently larger for the condenser method because of the smaller temperature difference. When the difference in temperature across the heat exchangers are more equal, like the freezer evaporator and the condenser, the uncertainty of both methods is more even.

The uncertainties in this experiment were calculated using a propagation of error analysis in Engineering Equation Solver - EES (Klein and Alvarado, 1998), with the uncertainty in the temperature and airflow measurements set at $\pm 1°F$ and $\pm 10\%$, respectively. The error bars, which represent the uncertainties in the experimental measurements, confirm that the UA values predicted when the condenser was used as the source of analysis had fallen within the borders of experimental uncertainty. With the lack of better equipment to improve the measurements, the results obtained here should be quite satisfactory. In particular, all but one of the condenser curves were not more 17% higher than the mass meter measurements.

The same cannot be said for calculations that were based on evaporator measurements. Except for Figure 2.14, the experimental uncertainties alone were not able to explain the observed difference. As explained before, the assumption of total sensible cooling could have significantly contributed to this error. Neglecting to account for this uncertainty in the analysis may have been a cause of the inaccuracy. In addition, the measurement of air at low flow rates is also more vulnerable to errors. At low flow rate, air leaks from the system are significant and is a source of uncertainty in the measurement of airflow. The airflow rate for the refrigerator evaporator was on the order of 4 times less than that through the condenser. Hence, it should not be surprising if the errors were larger for the evaporator. Finally, the mass flow measurement may have been inaccurate by itself. This possibility was not considered in the analysis, and may have been sufficient in reconciling the differences in the calculations. With all the unknowns lumped together, the actual uncertainties in evaporator measurements may be higher than the condenser, although Figures 2.14-2.16 show otherwise (because no errors were associated with the assumption of sensible cooling).

In view of these uncertainties, the measurements from the mass meter should provide the most reliable method in estimating the evaporators and condenser UAs. Specifically, they are not subjected to the uncertainties that are associated temperature and airflow measurements, nor are they reliant on the assumption of sensible cooling-only by the evaporator.

Since the heat exchanger UAs determined in this experiment would serve as inputs in a computer model that employs a steady state model, it is more convenient to express them as stationary time-averaged quantities. The UA was estimated by taking its average over a cycle. Using this approach, the freezer and refrigerator evaporator UAs were 84 and 64 Btu/hr/F respectively. For the condenser, the UA for the freezer was 70 Btu/hr/F and 74 Btu/hr/F for the refrigerator. The results show that the condenser had a slightly higher UA value for the refrigerator than the freezer even though both cycles reject heat through the same source. This is most likely due to the difference between the arrangement of the refrigerator and the freezer tubes in the condenser and the fact that heat is rejected through two rows of tubes for the refrigerator as opposed to only one for the freezer.

2.5 Conclusions

The overall conductance of all three heat exchangers have been experimentally measured. To determine the UA, an extensive measurement of temperature, pressure, airflow rate and refrigerant mass flow was performed at various locations around the refrigerator and freezer cycles. Using these data, an energy balance was then performed to yield the UA.

The data collected had allowed three methods of analysis. While the direct measurement of mass flow constitutes one method, the remaining two result from energy balances on the evaporator and the condenser. In all cases, the measured UA was highest when the condenser was used as the source of analysis, followed by the mass meter and the evaporator method, in that order. The discrepancy between the results obtained by the three methods of analysis was the highest in the evaluation of the refrigerator condenser. In that estimate, the UA based on the evaporator method was 32% smaller than the mass flow meter measurements while the condenser method was 30% higher than the mass flow meter method.

The degree of experimental uncertainty was also greater for the condenser method than the evaporator, while the mass meter measurement was considered to be the standard for comparison. The primary sources of uncertainty in the experiment were attributed to the measurements of temperature and airflow. In addition, the assumption that only sensible cooling was performed by the evaporator had also contributed to the errors. Difficulties in performing measurements at low flow rates, coupled with the adverse effects of leakages, could also be used to explain the differences between the results from the each method.

As a result of these errors, the mass flow meter was the most reliable method of measuring the UA. The advantage of this method is apparent as it eliminates the uncertainties in temperature and airflow measurements and the need for any assumptions.

Owing to the transient nature of the cooling process, the UA was seen to change with time. To avoid the problems of expressing the UA as a time-dependent quantity, its value was approximated by taking the average over the cycle. Using this approach, the UA of the freezer and refrigerator evaporators were determined to be 84 Btu/hr/F and 64 Btu/hr/F respectively, while the condenser UA was estimated at 74 Btu/hr/F for the refrigerator and 70 Btu/hr/F for the freezer.