of the liquid prior to it entering the expansion device. By doing so, the capacity for refrigeration is increased.

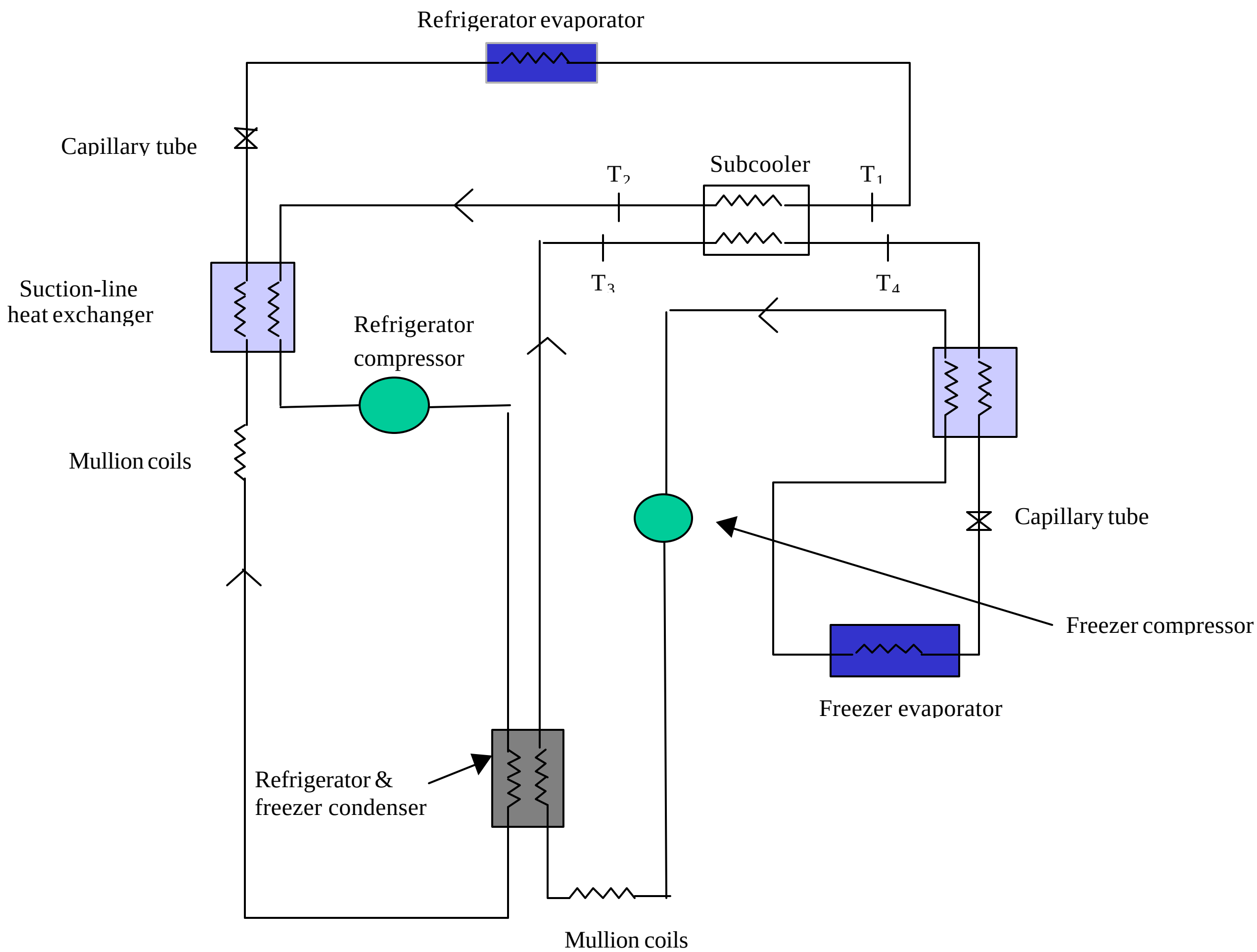

Figure 6.1 Diagram identifying one possible location for the subcooler.

Figure 6.1 shows a suitable location for the subcooler in the system. It may be placed before or after the suction-line heat exchanger in both the freezer and the refrigerator cycles. In other words, the location of the subcooler is flexible, so long as the ultimate goal of subcooling the freezer liquid is achieved. In addition, the state of the refrigerant at the inlet or the outlet of the subcooler is not fixed. The state of the refrigerant on the refrigerator side may be saturated either at the entrance or the exit of this device. Hence, there exist a variety of combinations that are possible and may each produce a different effect on the performance of the overall system (the best location for the subcooler is explained in the next section).

The construction of the subcooler may be similar to that of the suction-line heat exchanger (shown in Figure 5.4). The copper tubes that carry refrigerant from the freezer condenser and the refrigerator evaporator may be put in direct communication with other. The heat transfer rate depends, in part, on the surface area available for the fluids to come in contact. If need be, the subcooler can be made longer to increase the surface area, but this results in additional pressure drop. To prevent the refrigerator vapor from absorbing any heat from the environment, the tubes should also be well insulated.

However, the transfer of energy from the freezer liquid to the refrigerator vapor by this method may require that both cycles be run simultaneously. To avoid this constrain, the freezer liquid-line may be connected to the refrigerator suction-line by a set of fins. When the freezer cycles on, the energy from the freezer liquid is transferred onto the fin, which “stores” this energy and transfers it to the refrigerator vapor when the refrigerator cycles on.

The ability of the subcooler to transfer heat from the freezer to the refrigerator is described by its effectiveness. Since the product of mass and specific heat is larger for the refrigerant in the freezer cycle, the effectiveness of the subcooler is defined as

$$e = \frac{T_2 - T_1}{T_3 - T_1} \qquad \text{Eqn. 6.1}$$

where the subscripts correspond to the states in Figure 6.1. The effectiveness, then, is the parameter that affects the extent of subcooling that the freezer liquid experiences.

6.2 *Advantages and Disadvantages of Mechanical Subcooling*

In some ways, a subcooler behaves like a suction-line heat exchanger. The desired increase in capacity is achieved by means of subcooling the condenser liquid. Another advantage of subcooling is the removal of any residual liquid that leaves the evaporator, thus protecting the compressor. Furthermore, subcooling will also reduce the tendency toward the formation of flash gas at the entrance to the capillary tube by condensing any two-phase refrigerant that leaves the condenser. With a cycle run time of approximately 36%, the present compressor in the refrigerator cycle is well oversized. Concerns of the availability and the low efficiency have deterred the refrigerator manufacturer from seeking a smaller size compressor. Since the use of the subcooler effectively transfers some of the refrigeration load from the freezer to the refrigerator, it enables a more efficient use of the refrigerator compressor.

Aside from the similarities that they share, one major difference between a subcooler and a suction-line heat exchanger is the source of the liquid subcooling. Unlike suction-line heat exchangers, the recipient of this heat is not the suction vapor in the freezer cycle. In this configuration, the liquid refrigerant in the freezer cycle loses heat to the lower temperature vapor from the refrigerator evaporator. As such, the mass of the refrigerant in the freezer remains unaffected because the conditions of the suction gas do not change. Instead, the heat absorbed by the vapor leaving the refrigerator evaporator increases its specific volume, resulting in a reduction in refrigerant mass flow.

Table 6.1 summarizes the advantages and disadvantages that a mechanical subcooler offers.

Advantages	Disadvantages
Increases the cooling capacity of the freezer	May be redundant if a suction-line heat exchanger is in place
Distributes the loads more equally by transferring heat from the freezer to the refrigerator	An increase in suction gas temperature will lead to a decrease in mass flow for the refrigerator cycle
Ensures the removal of liquid on the suction-line of the refrigerator	Additional pressure drop on the low side of the refrigerator may eliminate the benefits of subcooling the freezer liquid
Prevents the formation of flash gas at the inlet of the freezer expansion device	

Table 6.1 The advantages and disadvantages of using a subcooler.

6.3 Effect of Subcooler on System COP

Subcoolers may or may not be used in conjunction with a suction-line heat exchanger. Since the two components share many similarities, in terms of their objectives, the use of both devices together may prove to be redundant. If the configuration in Figure 6.1 were used, the subcooler may "rob" the freezer suction-line heat exchanger of further potential to subcool the refrigerant. Should this be the case, the performance gains demonstrated by suction-line heat exchangers in Chapter 5 may be compromised or diminished by the presence of a subcooler.

Currently, a suction-line heat exchanger is installed in both the freezer and the refrigerator cycle. The benefits of using these heat exchangers were discussed in detail in Chapter 5 and they apply here. Figure 6.2 shows the benefits of installing a subcooler for a variety of cases. In all cases, the effectiveness of the suction-line heat exchangers were set at

90% while a low-side pressure drop of 0.5 psi (the estimated pressure drop across the suction-line heat exchanger is only 0.1 psi) across these heat exchangers and subcooler was assumed. A system configuration identical to that in Figure 6.1 was used in this analysis. This is because an analysis confirmed that the location of the subcooler as shown in Figure 6.1 had yielded the most potential for system enhancement. Nevertheless, the analysis also revealed that the difference between placing the subcooler before or after the suction-line heat exchanger was minor.

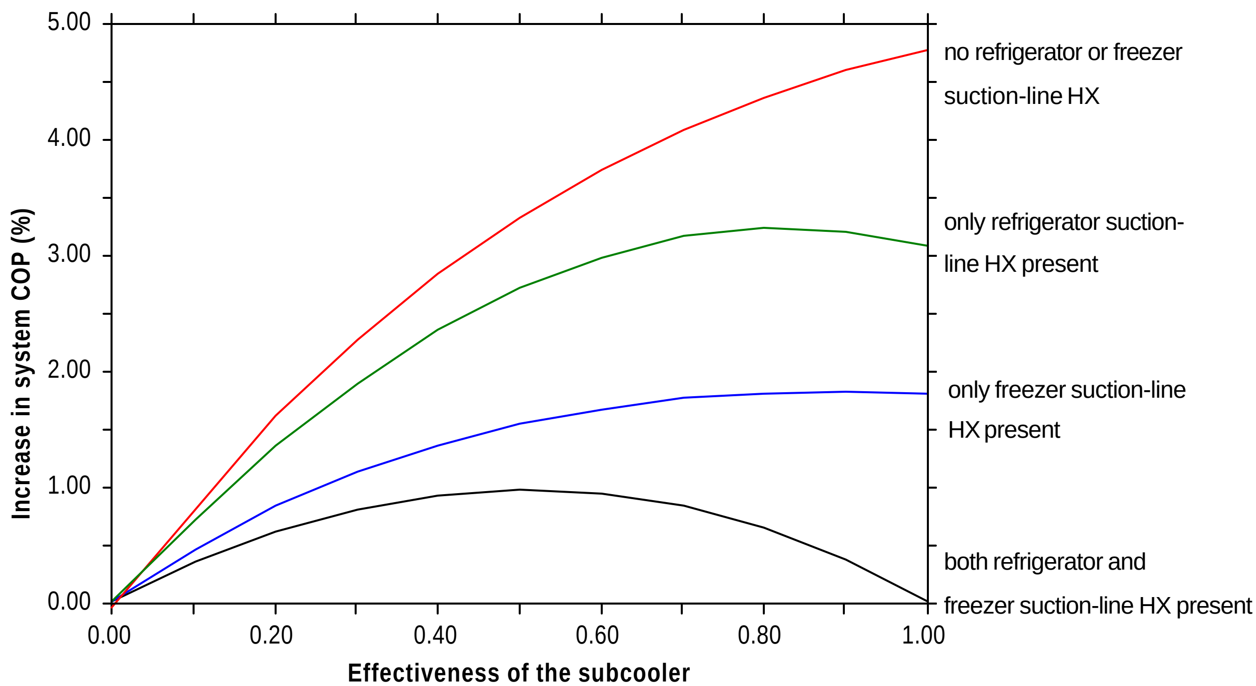

Figure 6.2 The increase in system COP as a function of the subcooler effectiveness. The suction-line heat exchangers all have a 90% effectiveness (The current suction-line heat exchangers in the refrigerator have an effectiveness of over 90%)

6.3.1 Subcooler with a Suction-Line Heat Exchanger in both the Freezer and Refrigerator Cycles

When there is already a suction-line heat exchanger in both cycles, the subcooler is shown to be a redundant feature which does not contribute toward any significant gains in the performance of the system. The maximum benefit that can be obtained is a mere 1% and corresponds to the situation when the subcooler has an effectiveness of approximately 0.5.

The redundancy of the subcooler can be better understood by referring to Figure 6.1. By placing the subcooler prior to the suction-line heat exchanger, the freezer liquid that enters the latter is at a lower temperature than it would have been had there not been a subcooler. As such, it is apparent that the potential for the freezer vapor to absorb heat from the liquid is diminished. Although the liquid will still enter the evaporator at a lower enthalpy than if the subcooler was not there, this difference is not that significant. One of the major advantages that the freezer cycle enjoys, however, is the fact that this extra cooling comes at no price. The only decrease in its performance, in terms of a reduction in mass flow, is due to the suction-line heat exchanger that it employs.

On the other hand, the presence of a subcooler will always reduce the performance of the refrigerator cycle. A subcooler with a high effectiveness will raise the temperature of the refrigerator vapor that enters the suction-line heat exchanger, thereby removing its capability to absorb any heat from the liquid that it comes into communication with. Hence, the refrigerator liquid realizes little or no change in its enthalpy and the capacity of the cycle is not enhanced. This result is compounded by the fact that the higher temperature vapor entering the compressor will reduce the mass flow in the cycle. With the mass flow rate reduced and only a small increase (or probably none-depending on the subcooler effectiveness) in the refrigeration effect, the overall result is a net decrease in the COP of the refrigerator cycle. The combined effect of the subcooler on the performance of the refrigerator and the freezer is shown in Figure 6.3.

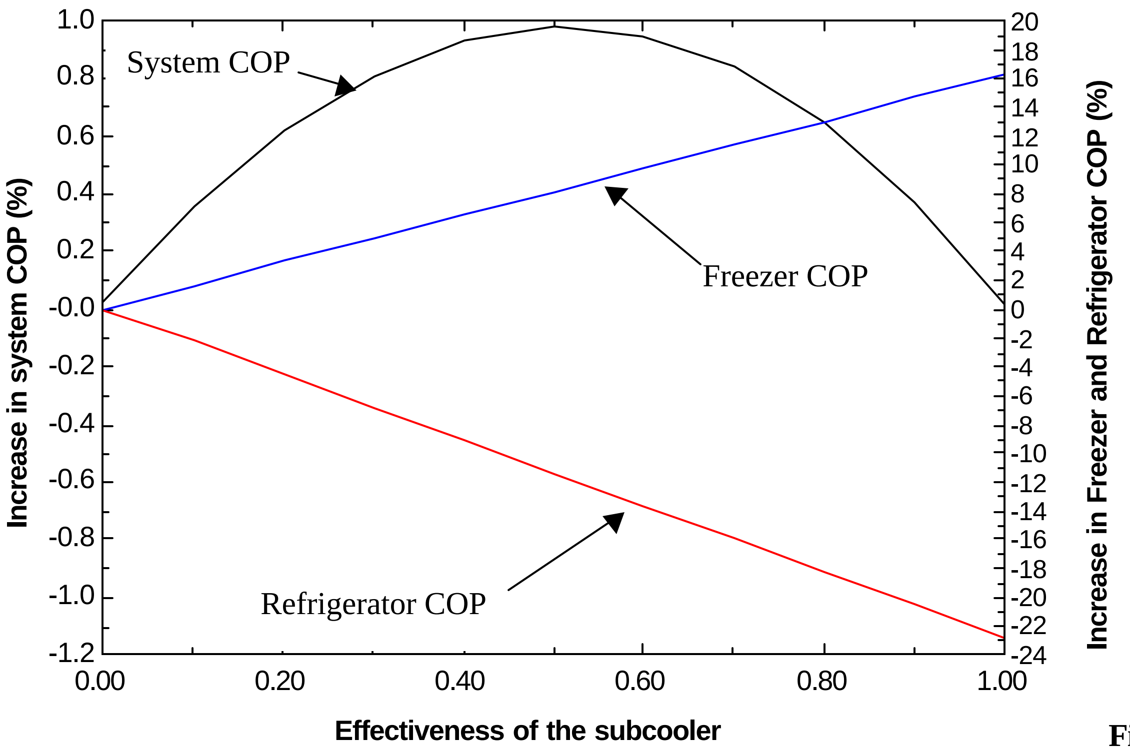

Figure 6.3 The change in the freezer, refrigerator and system COP with subcooler effectiveness.

While the freezer cycle is enhanced by the subcooler, Figure 6.3 shows that the opposite is true for the refrigerator. Although the decrease in performance is always greater for the refrigerator cycle, its impact on the overall system performance is less due to the way that the COP is weighted. The maximum COP is attained when the subcooler effectiveness is around 0.5. When the effectiveness of the subcooler increases beyond this value, the decrease in performance of the refrigerator begins to erode the improvements that were obtained from the freezer cycle. As the effectiveness of the subcooler increases further, the effect of subcooling is no longer able to offset the decrease in the performance of the refrigerator (due to the decrease in mass flow) and the performance system of the system "breaks-even" at an effectiveness of 1.

6.3.2 Subcooler with a Suction-Line Heat Exchanger in Freezer Cycle only

In this case, the subcooler is used in conjunction with only the suction-line heat exchanger in the freezer cycle (no suction-line heat exchanger in the refrigerator). The maximum improvement for this arrangement is about 1.8% and is obtained when the subcooler effectiveness is 0.9. With or without the subcooler, the refrigerator cycle experiences no gain in capacity because no suction-line heat exchanger was used. A subcooler, when applied in this case, reduces the mass flow rate in the refrigerator cycle due to the heating of the vapor by the freezer liquid. As a result, the capacity decreases accordingly and the refrigerator cycle experiences a negative impact when a subcooler is placed in the system.

Ultimately, the system COP increases slightly because of the gains made by the freezer cycle. Even with a suction-line heat exchanger already in place, the subcooler increases the opportunity for a greater amount of liquid subcooling to be performed. Again, the decrease in mass flow of the freezer cycle is only affected by its suction-line heat exchanger. In fact, a higher degree of heat transfer in the subcooler will actually promote the mass flow in the freezer cycle. When the subcooler has a higher effectiveness, the liquid leaves the subcooler and enters the freezer suction-line heat exchanger at a lower temperature, thus decreasing the amount of heating that the gas leaving the evaporator experiences. Since the refrigerator cycle benefits from a low subcooler effectiveness while the opposite is true for the freezer, the maximum increase in overall system performance occurs somewhere in between.

6.3.3 Subcooler with a Suction-Line Heat Exchanger in Refrigerator Cycle only

When the subcooler is employed with only a suction-line heat exchanger in the refrigerator cycle (no suction-line heat exchanger in the freezer), an effectiveness of 0.8 maximizes the system COP at 3.3%. For the freezer, a high subcooler effectiveness is more crucial in this case because it is its only source of subcooling. As such, the installation of the subcooler is seen to have more impact. With its mass flow rate remaining constant (because there is no suction-line heat exchanger), the freezer capacity will increase when more subcooling is performed.

The presence of the subcooler will always be a disadvantage to the refrigerator cycle. The ability for subcooling in the refrigerator cycle is reduced if the vapor leaves the subcooler at a high temperature. In any case, the mass flow will decrease while the amount of liquid subcooling that it experiences depends on the vapor temperature at the inlet of the suction-line heat exchanger.

6.3.4 Subcooler with No Suction-Line Heat Exchangers in either Cycle

The potential for improvement is greatest when the subcooler is the only device performing heat exchange. It serves a similar purpose as a suction-line heat exchanger, except that the cycle being subcooled does not experience a decrease in mass flow. While the freezer profits from an increase in capacity, the refrigerator loses with a lower circulation of mass flow in the cycle. The opposing effects created by this interaction results in a net enhancement in system COP. The fact that the maximum improvement occurs at an effectiveness of unity lends further support to the conclusion that the increase in freezer capacity overweighs the negative influence of a decreasing mass in the refrigerator cycle.

An improvement of 4.7% in this arrangement is less than the 7% (for effectiveness = 0.9, $\Delta P_{lowside}$ = 0.5 psi, T_{evap} = -10°F) gain that can be achieved by putting a suction-line heat exchanger in the freezer cycle only (Figure 5.8 - only HX, no subcooler). A smaller increase in capacity is undeniably the main reason for this observation. When comparing the two cases, two main reasons can be offered to explain the decrease in potential for liquid subcooling. Firstly, the refrigerator vapor enters the subcooler at a higher temperature in this case (only subcooler, no HX) because of the higher evaporating temperature of the refrigerator cycle. Therefore, the potential for heat exchange is smaller here. More importantly, the amount of subcooling depends on the mass flow rates in both cycles, as the following expression based on an energy balance on the subcooler shows.

$$\dot{m}_{frez}\left(h_{frez,in} - h_{frez,out}\right) = \dot{m}_{refg}\left(h_{refg,out} - h_{refg,in}\right) \qquad \text{Eqn. 6.2}$$

In Eqn. 6.2, $h_{refg,in}$ and $h_{frez,in}$ depend on the evaporating and condensing conditions while $h_{refg,out}$ is a function of the subcooler effectiveness. Hence, $h_{frez,out}$ is determined only by the mass flow rates in refrigerator and freezer cycles. Since the mass flow rate is smaller in the refrigerator cycle, the effect of subcooling is lower for the case that uses only the subcooler. The effect of the relative mass flow rates is examined in greater detail in the foregoing section.

6.4 Ratio of Mass Flow Rates on the Effect of Subcooling

In the previous analysis, the effect of subcooling was shown to be a function of the mass flow rates in the two cycles that communicate. A low rate of refrigerant mass flow on the refrigerator cycle was seen to limit the amount of heat transfer that took place. Because the main idea of installing a subcooler is to promote the transfer of loads from the freezer to the refrigerator cycle, these limitations would effectively defeat this purpose. Figure 6.4 shows the impact of subcooler effectiveness on the COP for systems with different mass flow rate ratios. The mass flow rate ratio, as used in the figure below, is defined as the ratio of the mass flow rate of the refrigerator cycle with a suction-line heat exchanger effectiveness of 0.9 to that of the freezer cycle which also has a suction-line heat exchanger with the same effectiveness, when a subcooler was not used in the system.

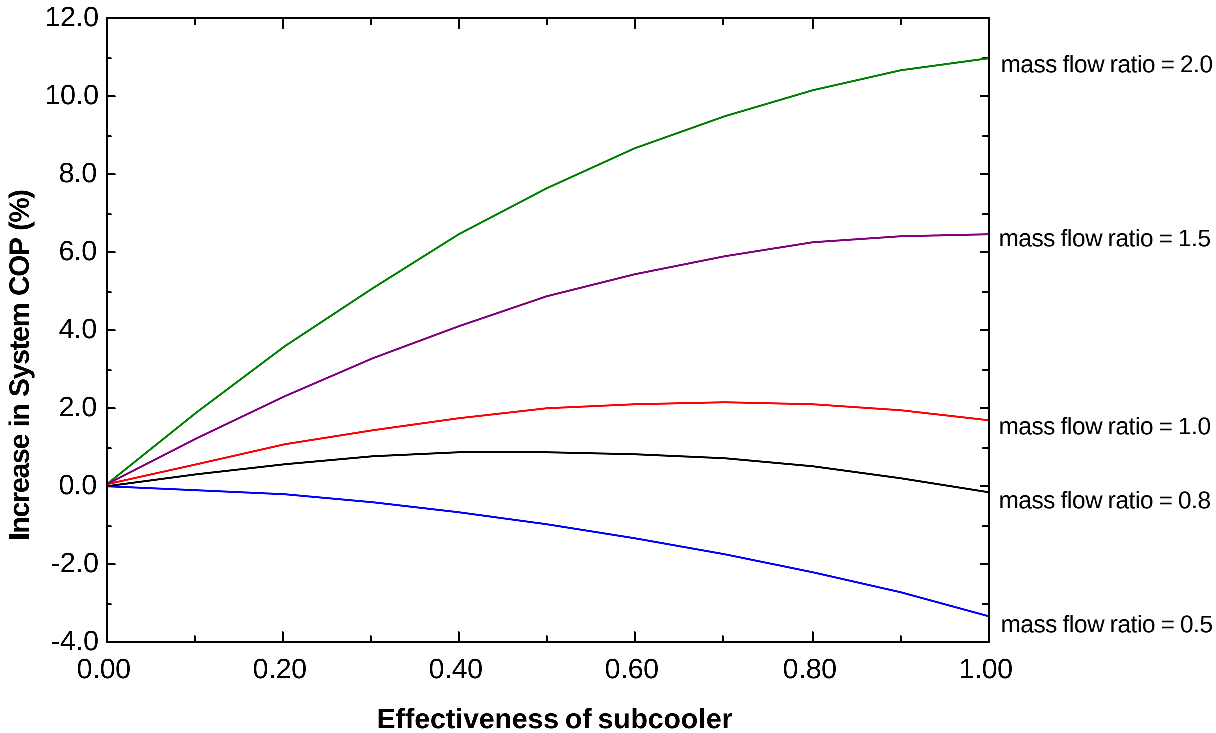

Figure 6.4 Effect of subcooler effectiveness on system COP for different mass flow rate ratios. The heat suction-line heat exchangers and subcooler all have a constant 0.5 psi low-side pressure drop.

Clearly, the mass flow rate ratio is a very important parameter that determines the amount of subcooling performed on the freezer liquid. Higher ratios were shown to yield larger gains in the system COP. The best part of this observation is the fact that this advantage comes at no extra cost. Unlike the earlier analysis, the increase in performance of the freezer is not achieved at the expense of the refrigerator cycle. Regardless of the mass flow rate ratio, the percentage of decrease in the refrigerator mass flow is only a function of the effectiveness of the subcooler and its suction-line heat exchanger. Therefore, the COP increase is greater for systems with a larger mass flow rate ratio because more subcooling is performed for the freezer liquid without further hurting the performance of the refrigerator.

In reality, the mass flow rate ratio is also a function of the subcooler effectiveness. Even though the refrigerator load increases with subcooler effectiveness, the mass flow rate does not because the load is transferred to the refrigerator vapor leaving the evaporator and not the refrigerator cabinet (the mass flow will only increase if the load from the freezer is transferred into the cabinet to warm the cabinet air). Instead, heat from the freezer liquid raises the temperature of the vapor in the refrigerator suction-line, which causes the mass flow rate to decrease.

At present, the benefit of using a subcooler in conjunction with a suction-line heat exchanger is poor because of its low mass flow rate ratio of 0.8. In systems that have a higher ratio, this arrangement can prove to be a very effective method in enhancing their performance.

6.5 Effect of Low-Side Pressure Drops

In the previous sections, the discussions were only limited to the case when the pressure drop on the low side of the suction-line heat exchanger and subcooler was set at 0.5 psi. Pressure losses may be the result of friction, a change in cross section of the tubes or due to an increase in potential energy if the fluid was elevated to a higher level. In Chapter 5, the consequences of a low side pressure drop on the performance of systems utilizing suction-line heat exchangers were studied in detail. The following section is dedicated to the study of these effects when a subcooler is used in conjunction with these heat exchangers.

Analysis in the previous section has shown that the room for improvement by installing a subcooler in a system which already have suction-line heat exchangers in both cycles is relatively small. Such may not be the case if a large pressure drop exist in the suction-line heat exchangers. Due to the effects of pressure losses, the performance of the system may be seriously compromised. It may then be more beneficial to remove these heat exchangers and install the subcooler instead. Just as it is impossible for a heat exchanger to have no pressure drop, so is the case with the subcooler. Since the construction of a subcooler resembles that of a suction-line heat exchanger, it may be assumed that both should have an equal pressure drop for the same length. Figure 6.5 examines the case when both suction-line heat exchangers having a certain pressure drop were removed and in return, a subcooler having the same pressure drop was installed. On the other hand, Figure 6.6 analyzes the effect of installing a subcooler with the same pressure drop as the suction-line heat exchangers without removing them.

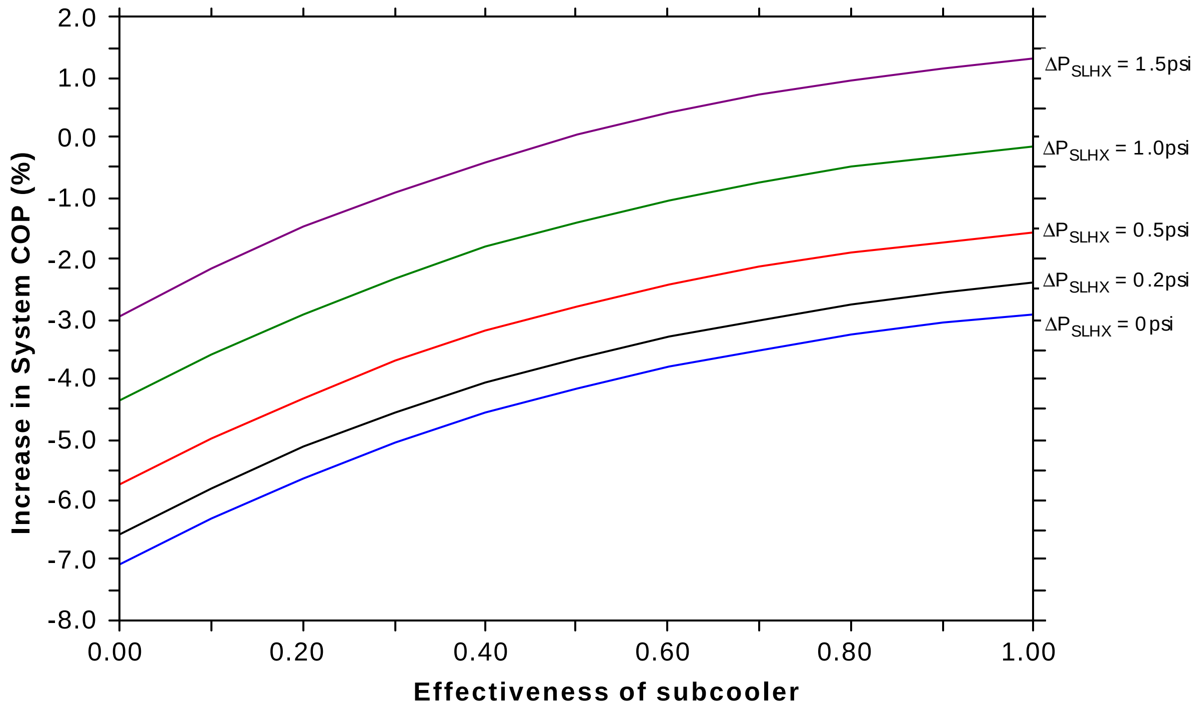

Figure 6.5 The change in system COP with subcooler effectiveness when the suction-line heat exchangers with a pressure drop was removed and a subcooler with the same pressure drop was installed.

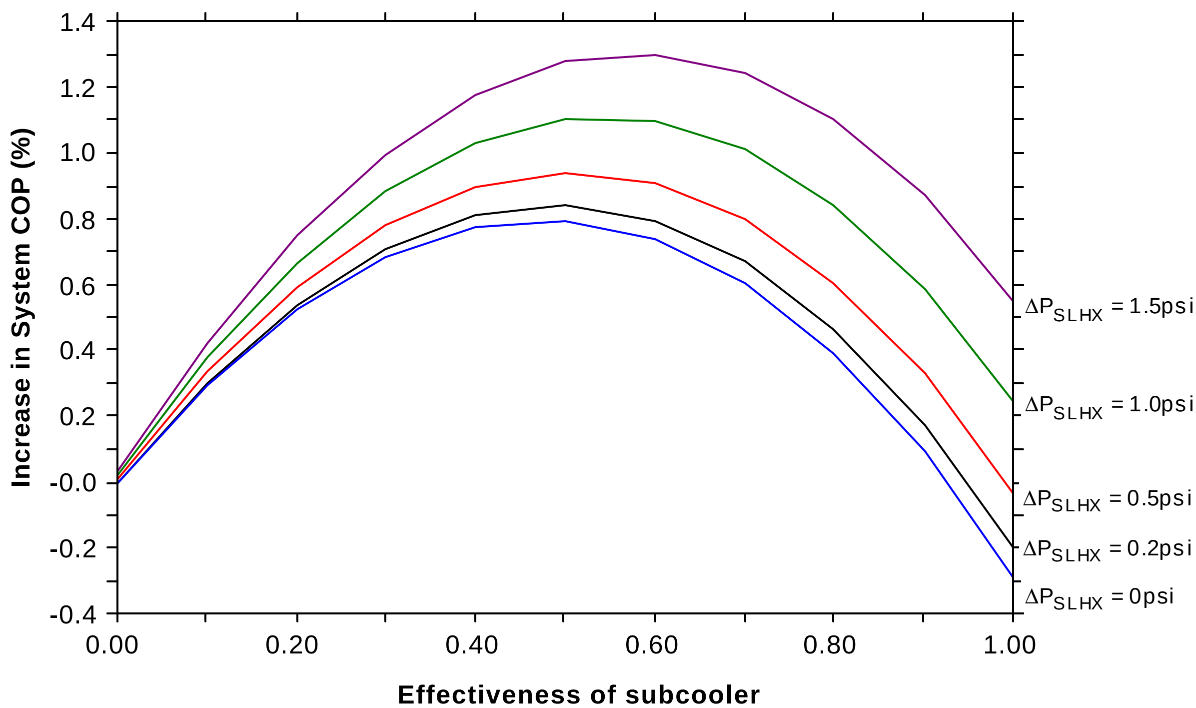

Figure 6.6 The change in system COP when a subcooler with an equal pressure drop was installed together with the existing suction-line heat exchangers.

In both cases, it is clear that the benefits were more pronounced when a large pressure drop had already existed in the suction-line heat exchangers. With the estimated pressure drop of 0.1 psi in the current heat exchangers, it can be seen that replacing them with a subcooler having an equal pressure drop is harmful to the performance of the system. In fact, the present heat exchangers would have to be experiencing a significant pressure loss (>1 psi) for this method to have no effect (0% change) on the COP. The alternative method seems to be a better approach. Comparing Figure 6.5 and Figure 6.6, it is evident that even with a high (1.5 psi) pressure drop, it is still better to retain the heat exchangers. Where the pressure drop is highest, the subcooler is seen to make the most impression. However, the gains demonstrated here are still in the vicinity of 1-2%, which is rather low. For fear of a larger pressure drop than expected or any uncertainties in implementation, the minor gains expected here may not justify the use of a subcooler.

6.6 Conclusions

The idea of transferring a portion of the cooling load from the freezer to the refrigerator cycle has been explored. The construction and location of a subcooler, a device which serves as a bridge that enables this heat transfer to occur, has been conceived and analyzed.

While the concept of subcooling has proven to be an effective method of increasing the capacity of a cycle, the benefits of installing a subcooler are not that promising. Among others, the enhancement in the performance of the system depends largely on whether suction-line heat exchangers are already installed. When used in conjunction with suction-line heat exchangers, the potential benefits arising from this subcooler was severely reduced. In this refrigerator, the maximum improvement of installing a subcooler was only 0.8% (based on a low-side pressure drop of 0.5 psi in the suction-line heat exchangers and subcooler).

A parameter, defined as the ratio of the mass flow rate of the refrigerator to that of the freezer cycle when a subcooler is not used, has been shown to exert a strong influence on the potential for subcooling. One of the major reasons for the poor benefits demonstrated here can be attributed to the low mass flow rate ratio in this refrigerator. For systems that have a higher mass flow rate ratio, a subcooler would be a more crucial component in promoting the COP of the system.

The possibility of replacing suction-line heat exchangers that have high pressure drops with a subcooler having an equivalent pressure drop has been shown to harm the performance of the system. Instead, subcoolers should be installed together with the current heat exchangers. Except for large pressure drops, the improvement in COP is unlikely to exceed 2% even with the installation of the subcooler.

CHAPTER 7 Mullion Tubes

7.1 Introduction

The cold surfaces on refrigerator cabinets provide the opportunity for condensation of water vapor from the environment. In refrigerators, this phenomenon is known as sweating. To prevent the occurrence of this undesirable event, the outer surface should be kept at a temperature higher than the dew point temperature of ambient air.

In older refrigerator designs, the heating of these surfaces is accomplished by installing electric anti-sweat heaters adjacent to the surface. An anti-sweat heater switch in some models allows the refrigerator user control of the operations of these electric heaters. Although this approach is capable of eliminating the possibility of sweating on exposed refrigerator surfaces, its main disadvantage is the high energy consumption of these heaters.

To avoid paying an extra penalty in energy consumption, the refrigerator under study employs mullion tubes that are installed around the outer frame of the freezer and refrigerator cabinets. Unlike electric heaters, this method utilizes “waste” heat from the discharge vapor or warm liquid refrigerant leaving the condenser to provide heating to the external surfaces of the cabinet. The location of these mullion tubes in the freezer and refrigerator cycles is shown in Figure 7.1. For the freezer cycle, the mullion tubes are positioned between the discharge of the compressor and the inlet to the condenser. Using this configuration, the mullion tubes utilize superheated vapor that would be otherwise rejected through the condenser. The source of heat for the mullion tubes in the refrigerator is the warm condenser liquid. When liquid refrigerant leaving the condenser is used to provide heat for this process, the returning liquid is subcooled.

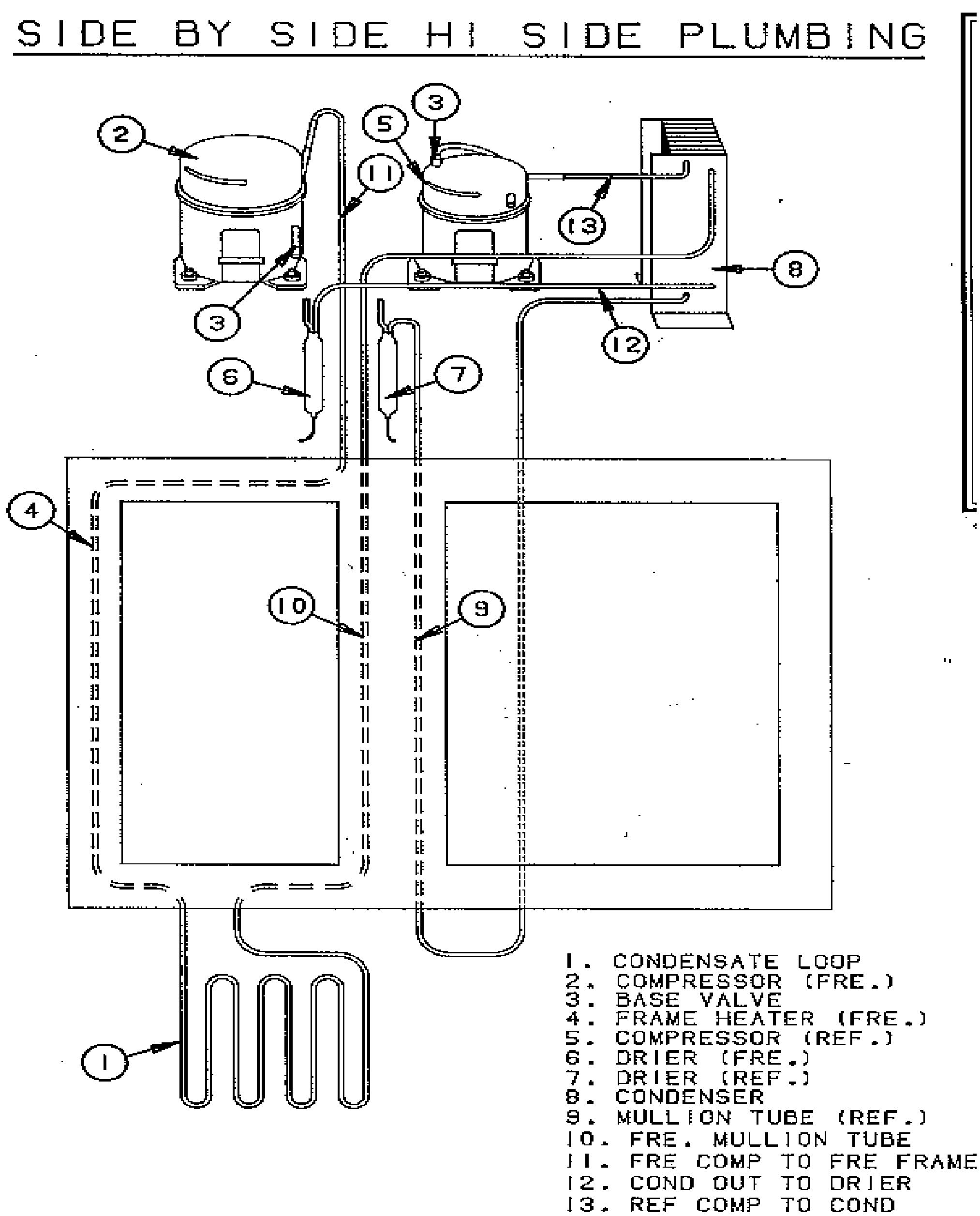

Figure 7.1 A diagram showing the location of the mullion tubes in the freezer and refrigerator cycles (Courtesy of Sub-Zero Freezer Company).

7.2 Advantages and Disadvantages of Using Mullion Tubes

The are a number of advantages and disadvantages associated with the use of mullion tubes. The foremost advantage, as described above, is its ability to prevent sweating without the use of additional electric energy. In fact, this is the main reason why the use of mullion tubes has been widely adopted in many residential refrigerators.

Another advantage of using mullion tubes is closely tied to the way heating is achieved in this method. By using heat from the discharge vapor in the freezer cycle, this method reduces the condenser load. Depending on the present size of the condenser, this may be important if the condenser is undersized as it can lower the condensing pressure of the cycle. Even with a different configuration, the refrigerator also benefits from the application of the mullion tubes in that it provides a certain degree of subcooling to the condenser liquid. Subcooling increases the refrigeration effect and enhances the performance of the cycle. Although a suction-line heat exchanger is available to perform subcooling, the lower temperature of the liquid entering this heat exchanger will reduce the extent of decrease in mass flow that results from the heating of suction gas leaving the evaporator.

When heat is rejected from the mullion tubes to maintain the temperature of the outer cabinet frame, a portion of this heat invariably escapes back into the refrigerated compartment. The increase in cooling load of the cabinet that results from this effect represents the main disadvantage of using mullion tubes.

Since mullion tubes traverse the entire cabinet, they are typically long and include many bends. The extra distance that the refrigerant has to travel induces added pressure drop. In anticipation of this pressure drop, the compressor is required to pump the refrigerant to a higher pressure, which increases its energy consumption.

The absence of any flexibility in controlling the operations of the mullion tubes is another disadvantage. Unlike electric heaters, which can be shut off when the relative humidity of the environment is low, the operation of mullion tubes are beyond the control of the refrigerator user.

The advantages and disadvantages of employing mullion tubes are summarized in Table 7.1.

Advantages	**Disadvantages**
Effectively prevents sweating without consuming extra electrical energy	Increases the cooling load when heat from the mullion tubes escape into the cabinet
Reduces the condenser load for the freezer cycle and subcools the condenser liquid in the refrigerator cycle	Induces a pressure drop which forces the compressor to operate over a larger pressure ratio
	The operation of the mullion tubes is beyond the control of the refrigerator user

Table 7.1 The advantages and disadvantages associated with the use of mullion tubes.

7.3 Objective of Study

The contribution of the mullion tubes to the total cabinet load was the main focus of the analysis in this study. In addition, the impact of the pressure drop on the compressor operation and system performance was also investigated. Specifically, the objective of this study was to determine the combined impact of mullion heat rejection and the high-side pressure on the cabinet load.

Like mullion tubes, the use of electric heaters also results in some heat rejection into the cabinets when the heaters are in operation. It is the intention of this study to evaluate the additional cooling load that results from the operation of these heaters so that a comparison between mullion tubes and electric heaters can be made to identify the more economical method of preventing sweating.

7.4 Heat Rejection by Mullion Tubes

The amount of heat that the mullion tubes reject into the cabinet was modeled using Finite Element Heat Transfer - FEHT (Klein *et al.*, 1998), which is capable of handling two-dimensional heat transfer analyses. Due to the different locations of the mullion tubes in the freezer and refrigerator cycle, a separate analysis was performed for each cycle.

7.4.1 Mullion Heat Rejection into the Freezer Cabinet

Referring to Figure 7.1, the mullion tube is located at the compressor discharge and runs along the side frame (4) before reaching the drain pan (1), which is responsible for collecting water that results from the condensation of vapor in the freezer compartment (the numbers in brackets correspond to the numbers in Figure 7.1). In this pan, heat from the warm refrigerant is used to evaporate any water that may have accumulated over the cycle.

From this evaporative loop, the refrigerant flows back to the condenser through a tube that lies on the middle frame (10) that separates the freezer and refrigerator compartments.

In this analysis, only the heat that was rejected by the mullion tubes from the side and middle frame that separates both cabinets were considered. To estimate the rate of heat conduction into the cabinet, the temperature of the tubes at every location mullion is required. However, only the temperature of the superheated vapor leaving the compressor and that entering the condenser was available from the experiments that were conducted to measure the UA of the heat exchangers (in Chapters 2 and 4). Therefore, these two temperatures represent the upper and lower bound of the mullion tube temperature. Since the amount of heat rejected into the cabinet depends on the temperature of the mullion tube, the analysis was performed for two situations. In the first situation, the temperature of the mullion tube was assumed to be equal to the temperature of the compressor discharge vapor. The analysis was then repeated by setting the temperature of the mullion tube to the temperature of the refrigerant entering the condenser. The actual amount of heat that is rejected into the cabinet should then fall within the range of values calculated by these analyses. From the UA experiments, the temperature of the discharge vapor was approximately 138°F while the refrigerant entered the condenser at around 102°F.

7.4.1.1 Mullion Tube on the Freezer Side Frame

Figure 7.2 is a representation of the top view of the side frame in FEHT. To provide better visual clarity, Figures 7.3 and 7.4 show the location where the view in Figure 7.2 was obtained.

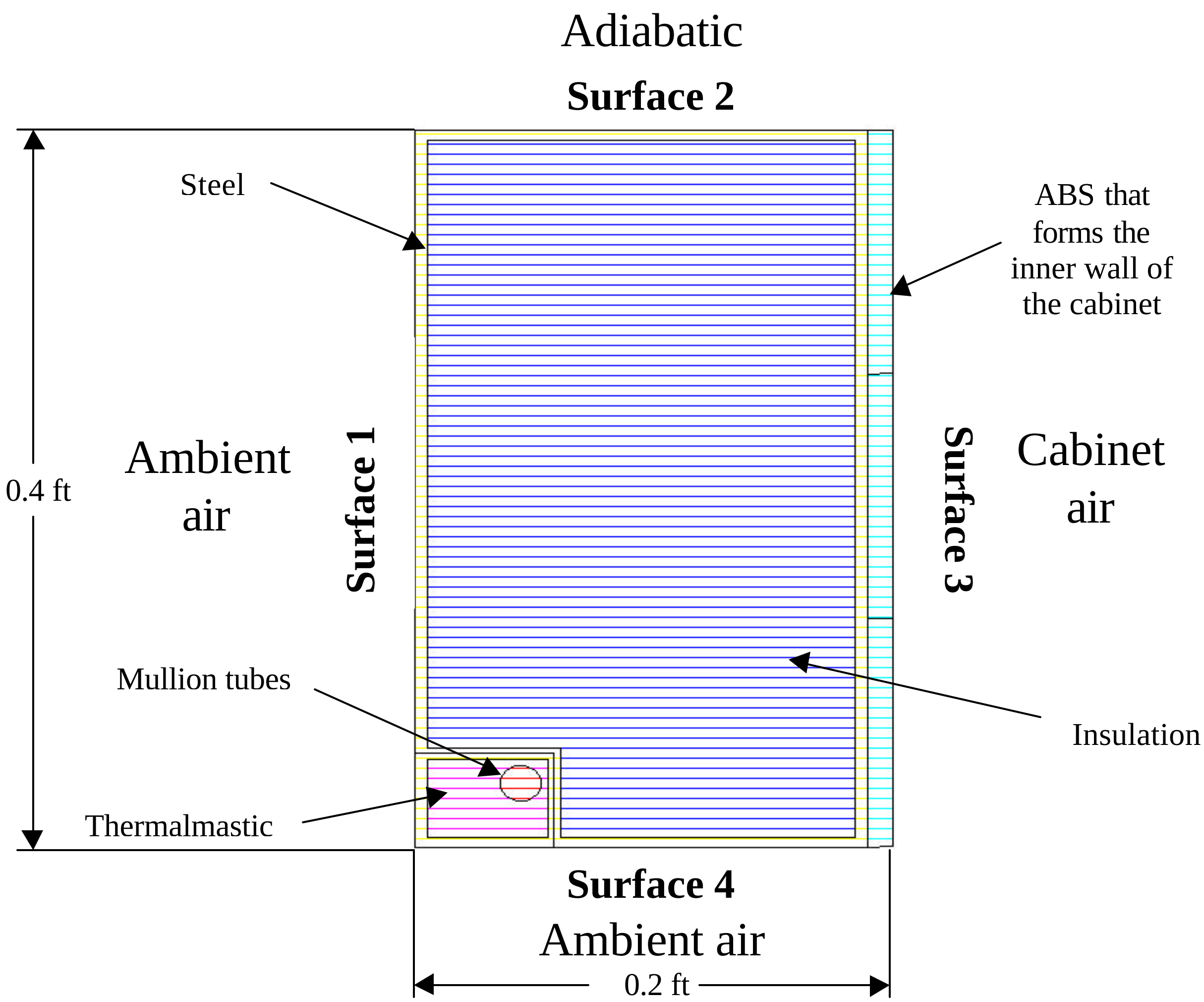

Figure 7.2 The view of the freezer side frame from the top of the refrigerator.

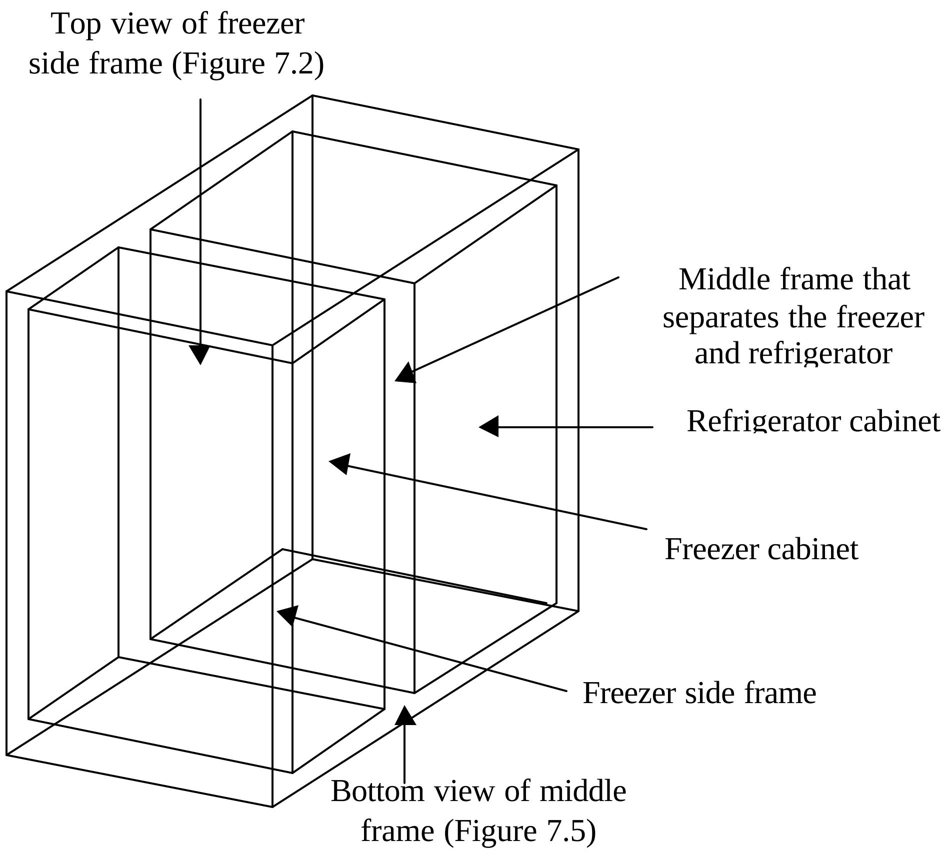

Figure 7.3 The location of the side frame and middle frame views.

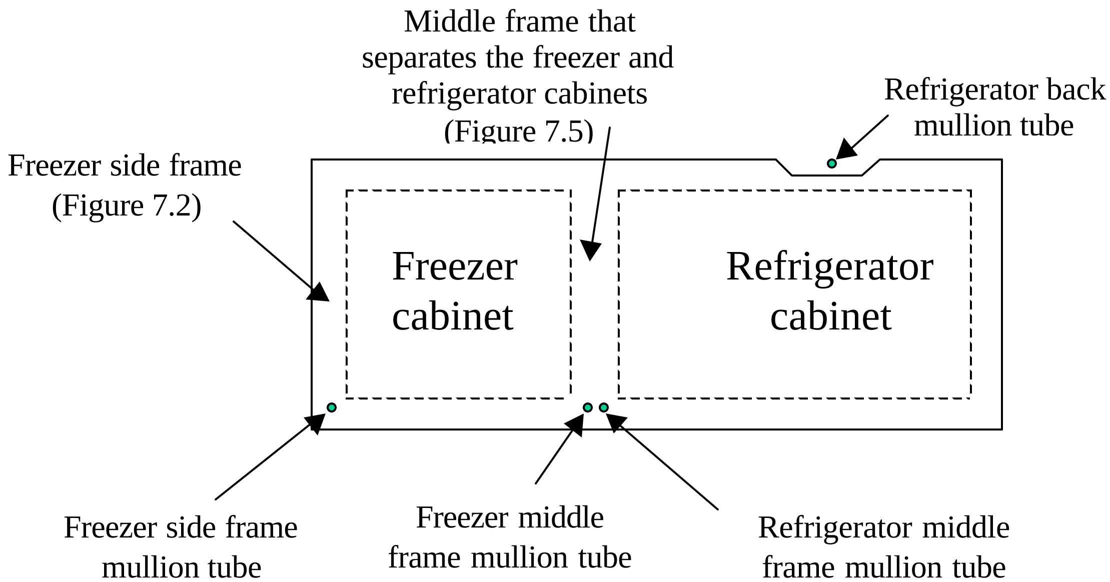

Figure 7.4 The top view of the refrigerator identifying the location of the back mullion tube.

The thermal conductivities of the materials used in this analysis are tabulated in Table 7.2. The value of the thermal conductivity for the insulation and ABS (that forms the inner wall of the cabinet) was provided by the refrigerator manufacturer while that of steel and copper were taken form Incropera and DeWitt (1996). A high value was assumed for thermalmastic because the main purpose of this material was to conduct heat from the tubes to the surface to prevent sweating.

Material	**Thermal conductivity (Btu/hr/ft/F)**
Copper (mullion tube)	223.000
Thermalmastic	5.000
Insulation	0.011
Steel	8.205
ABS	0.930

Table 7.2 Thermal conductivity of the materials used in Figures 7.2 and 7.5.

In reality, the length of the side frame (length of surface 1) in Figure 7.2 was 2 ft. However, a separate analysis (by treating the side frame as a frame that was adiabatic on one side) had shown that 95% of the heat rejected into the cabinet had occurred in the first 0.36 ft away from the mullion tubes. With the heat rejection through the remainder of the length insignificant, only the first 0.4 ft was used to represent the total length of the side frame.

As Figure 7.2 shows, heat convection occurs from surfaces 1, 3, and 4, but an adiabatic condition is assumed for surface 2. This assumption is justified because very little heat is conducted through the insulation (because only 0.4 ft of the 2 ft length of surface 1 was modeled). The heat transfer coefficients for air were determined using the relationship for heat transfer by natural convection shown in the following expression (Churchill and Chu, 1975).

$$\overline{Nu}_L = \left\{ 0.825 + \frac{0.387 Ra_L^{1/6}}{\left[1 + \left(\frac{0.492}{\mathrm{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 \qquad \text{Eqn 7.1}$$

with $$Ra_L = \frac{gb(T_s - T_\infty)L^3}{an} \qquad \text{Eqn. 7.2}$$

and $$\bar{h}_L = \frac{\overline{Nu}_L \cdot k}{L} \qquad \text{Eqn. 7.3}$$

where Pr = Prandtl number
g = gravitational acceleration
β = $1/(T_s + T_\infty)$
T_s = surface temperature
T_∞ = ambient or cabinet air temperature
L = length of the surface
α = thermal diffusivity
ν = kinematic viscosity
k = thermal conductivity
$\bar{h}_L$ = heat transfer coefficient

The convection coefficients calculated using the above equations range from 0.52 to 0.86, depending on the surface, ambient and cabinet temperatures. Since the temperature of the surface was not uniform, the surface temperature was obtained by taking an average of the nodal temperatures from the FEHT program. Owing to this approximate method of calculating the surface temperatures, a convection coefficient of 0.75 Btu/hr/ft^2/F was assumed for both the ambient and cabinet air instead of using a different convection coefficient for each surface (the sensitivity of the results to the convection coefficient is analyzed in Section 7.5).

Using a steady-state approach, the FEHT program was used to calculate the amount of heat that flowed into the cabinet for three cases. The temperature of the mullion tube was not prescribed in the first case, which is akin to the situation when mullion tubes are not used at all. As such, the cabinet load was only due to the difference between the ambient and cabinet air temperatures. For the second case, the temperature of the mullion tube was assumed to be the same as the temperature of the compressor discharge vapor. The analysis was repeated by setting the mullion tube temperature equal to the temperature of the refrigerant entering the condenser. Heat flowing into the cabinet in the latter two cases was then a combination of effects that stem from the temperature difference and the contribution of the mullion tubes. In all three cases, the ambient and freezer cabinet temperatures were set at 90°F and 5°F respectively.

Table 7.3 shows the amount of heat flowing across the boundaries of the four surfaces in Figure 7.2. A positive value represents heat flowing out from the frame.

Heat flow (Btu/hr/ft)			
	Case 1 (No tubes)	**Case 2 (T_{tube} = 102° F)**	**Case 3 (T_{tube} = 138° F)**
Surface 1	-5.99	-0.11	6.39
Surface 2	0.00	0.00	0.00
Surface 3	10.50	14.42	18.45
Surface 4	-4.51	-5.78	4.51
Total	0.00	8.53	29.35

Table 7.3 Heat flow across the four surfaces of the freezer side frame.

The results show that 10.50 Btu/hr (for every unit length in the direction of the freezer height) of heat flows into the freezer compartment as a result of the difference between the cabinet and ambient air temperatures. When a temperature of 102°F was prescribed for the mullion tube, the cooling load of the cabinet was increased to 14.42 Btu/hr. The difference between the two cases, which represents the sole contribution of the mullion tube, was 3.92 Btu/hr for every unit length along the height of the freezer. With a height of 7 ft, the total increase in load comes to 27 Btu/hr. However, the freezer only runs for approximately 41% of the time. Neglecting the effects of thermal capacitance, the mullion tube does not contribute to any additional load during the off-cycle. Hence, the mullion tube on the side frame is only responsible for an extra 11 Btu/hr of cooling load. An analysis using a mullion tube temperature of 138°F shows that the amount of heat rejection into the cabinet increases to 23 Btu/hr. Consistent with expectations, the amount of heat flowing into the cabinet increases when the temperature of the mullion tube is higher. The sensitivity of these results to the heat transfer coefficient will be discussed later.

7.4.1.2 Mullion Tube on the Freezer Middle Frame

A similar analysis was performed for the freezer mullion tube on the middle frame that separates the freezer and refrigerator compartments. Figure 7.5 illustrates how the mullion tube was represented in FEHT while Figures 7.3 and 7.4 identifies the location of this view.

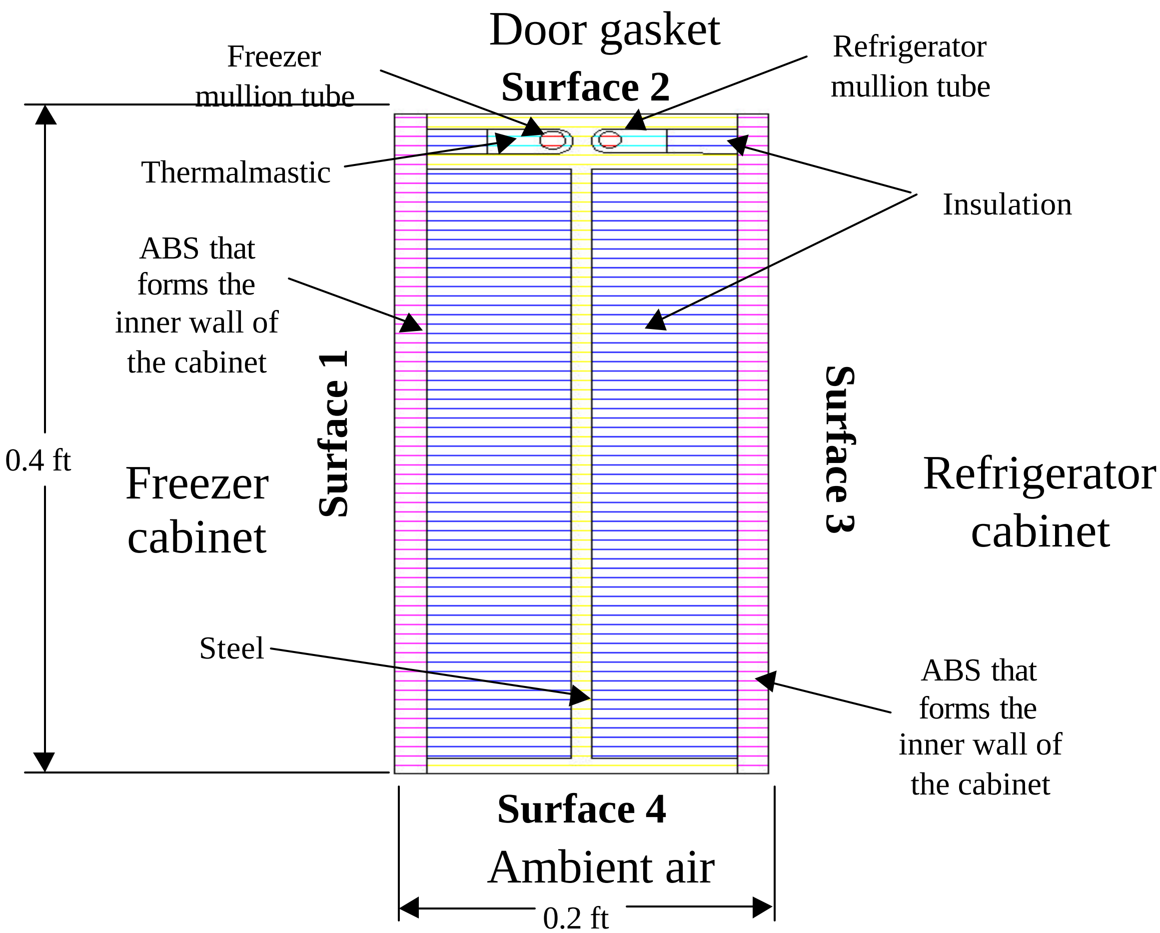

Figure 7.5 The view of the freezer middle frame from the bottom of the refrigerator.

The thermal conductivity of the materials used in this analysis were the same as those used for the freezer side frame. Similarly, the heat transfer coefficients for all the surfaces were assumed to be 0.75 Btu/hr/ft^2/F. While the ambient temperature was set to 90°F, the freezer and refrigerator cabinets were kept at 5°F and 38°F respectively. In the first situation, a temperature was not prescribed for the mullion tube but the temperature was set at 102°F and 138°F in the subsequent analyses. Aside from heat transfer by convection from surfaces 1 and 3, surfaces 2 and 4 were assumed to be adiabatic. The amount of heat that is lost through surface 2 is negligible because it comes into contact with the door gasket, which insulates it from the ambient air. On the other hand, surface 4 is adiabatic because very little heat is conducted to this surface due to the length of surface 1 (the same reason as why surface 2 in Figure 7.2 was adiabatic).

The results in Table 7.4 show that the amount of heat rejection into the freezer cabinet increased by 14.79 Btu/hr/ft when the temperature of the mullion tube was set at 102°F. This figure translates to 42 Btu/hr after taking into account the height and run time of the freezer cycle. When the mullion tube temperature was set at 138°F, 21.40 Btu/hr/ft, or 61 Btu/hr was added to the cooling load by this tube. Table 7.5 summarizes the individual contributions of the freezer side and middle frames to the additional cabinet load.

Heat flow (Btu/hr/ft)			
	Case 1 (No tubes)	**Case 2 (T_{tube} = 102° F)**	**Case 3 (T_{tube} = 138° F)**
Surface 1	3.81	18.60	25.21
Surface 2	0.00	0.00	0.00
Surface 3	-3.81	10.82	17.36
Surface 4	0.00	0.00	0.00
Total	0.00	29.42	42.57

Table 7.4 Heat flow across the four surfaces of the freezer middle frame.

Total heat rejection into freezer cabinet (Btu/hr)		
	T_{tube} = 102° F	**T_{tube} = 138° F**
Side frame	11	23
Middle frame	42	61
Total	53	84

Table 7.5 Total mullion heat rejection into the freezer cabinet.

The results show that the middle frame was responsible for the majority of heat rejected into the freezer cabinet. In reality, the refrigerant in the mullion tube is close to saturation by the time it enters the middle frame because of the large amount of heat that is required to evaporate water from the drain pan. Hence, the temperature of the tube should be much lower than 138°F and the amount of heat rejected into the cabinet should be close to the 42 Btu/hr estimate that corresponds to a tube temperature of 102°F.

The difference in the freezer cabinet load measured by the reverse heat leak test and the experiments that were performed to measure the heat exchanger UAs can be used to put the figures in Table 7.5 in perspective. The estimate of the cabinet load was 44 Btu/hr higher for the UA experiments since these experiments were capable of measuring the effect of mullion tubes and cycling losses. In fact, the difference in the estimated cooling load should be larger than this figure if the reverse heat leak test were performed using the same cabinet and ambient temperatures as the UA experiments (refer to Section 3.3.3 for a detail discussion on this subject). If cycling losses are negligible, the difference of 44 Btu/hr (or even higher) can be attributed to the additional load that results from the use of mullion tubes. Although the results in Table 7.5 seem high and account for somewhere between 19-30% of the cabinet load, they are quite reasonable in light of the comparison between the measurements of the cabinet load from the reverse heat leak test and the UA experiments.

7.4.2 Sensitivity of Results to Various Parameters

The primary sources of uncertainty in the estimate of the mullion heat rejection are the heat transfer coefficient and the thermal conductivity of the materials used in the analyses. Since the major resistance to heat transfer was from convection, the uncertainty in the estimation of the heat transfer coefficient contributes greatly to the errors in the analysis.

To study the sensitivity of the results to this parameter, the analyses were repeated by using a heat transfer coefficient of 0.5 Btu/hr/ft^2/F. The total (sum of the side and middle frames) amount of mullion heat rejection into the freezer cabinet was 42 and 67 Btu/hr for the cases when the temperature of the tube was set at 102°F and 138°F, respectively.

The conductivity of thermalmastic was found to have a smaller impact on the results. By decreasing the conductivity from 5 to 1 Btu/hr/ft/F, the total heat rejected into the freezer cabinet was only reduced by 2 Btu/hr.

At the same time, the additional cooling load may also be greater than the amount estimated in Table 7.5. It is worth noting that the amount of heat flowing into the *refrigerator* cabinet also increases when the freezer mullion tube is in operation. Likewise, it can be expected that the same situation would occur when the refrigerator cycles on while the freezer is in the off-cycle mode. Therefore, the more the cycles run separately, the more will the figures in Table 7.5 constitute as underestimates of the actual mullion heat rejected into the cabinet.

7.4.3 Mullion Heat Rejection into the Refrigerator Cabinet

For the refrigerator, the mullion tubes are located on the middle frame (Figure 7.5) and at the back of the refrigerator. The locations of the mullion tubes are illustrated in Figure 7.4, which shows the top view of the refrigerator. As Figure 7.4 shows, the mullion tube at the back of the refrigerator is not connected to the frame in any way, and as such, its contribution to the cabinet load is negligible. The heat rejection into the cabinet, then, is only significant for the middle frame.

The analysis for the refrigerator mullion tubes follows that of the analysis described in the previous sections, except for the temperature of the mullion tube. In the refrigerator, the mullion tubes are located at the exit of the condenser. By neglecting any effect of subcooling (the temperature of the refrigerant leaving the refrigerator mullion tube was not measured), a temperature of 105°F was assumed for the mullion tube, which corresponds to the condensing temperature of the refrigerant. Hence, the results obtained using this temperature represents the upper bound of the estimated mullion heat that is rejected into the refrigerator cabinet. To measure the cabinet load that resulted purely from the difference between the ambient, refrigerator and freezer cabinet air temperatures, a temperature was not prescribed for both the freezer and refrigerator mullion tubes in the first situation. In the second case, the temperature of the refrigerator mullion tube was set at 105°F, while a temperature was not assigned to the freezer mullion. The results from these analyses are tabulated below.

Heat Flow (Btu/hr/ft)		
	Case 1 (No tubes)	**Case 2 (T_{tube} = 105°F)**
Surface 1	3.81	18.95
Surface 2	0.00	0.00
Surface 3	-3.81	11.47
Surface 4	0.00	0.00
Total	0.00	30.42

Table 7.6 Heat flow across the four surfaces of the refrigerator middle frame.

Table 7.6 shows that the operation of the refrigerator mullion tube had increased the cabinet load by 15.28 Btu/hr/ft. With a 36% run time, the additional heat flowing into the refrigerator compartment was 39 Btu/hr, which represents 20% of the total refrigerator cabinet load.

The estimate for the additional cooling load to the refrigerator was also subjected to the same sources of uncertainty as those encountered in the freezer mullion calculations. By decreasing the convection coefficient from 0.75 to 0.5 Btu/hr/ft^2/F, the amount of mullion heat rejection was reduced to 30 Btu/hr. However, the thermal conductivity of thermalmastic was seen to have negligible impact on the results. Reducing the value of the thermal conductivity from 5 to 1 Btu/hr/ft/F, the amount of heat rejected into the cabinet was only reduced by 1 Btu/hr.

The amount of heat that is rejected into the refrigerator cabinet was less than that of the freezer because heat was only rejected through the middle frame. The estimated amount of heat flowing into the cabinet from the mullion tubes was less than the difference in the cabinet load measured by the reverse heat leak test and the UA experiments, which was 65 Btu/hr. Therefore, the results obtained in this analysis are reasonable.

7.5 High-Side Pressure Drop

The previous sections were dedicated to the study of mullion heat rejection into the freezer and refrigerator compartments. Aside from the negative impact of these additional loads, the cycles also suffer from the loss of pressure on the high side. To estimate the effect of this pressure drop on the system, the extent of pressure that is lost while overcoming the shear force in the mullion tubes must be evaluated.

To estimate the pressure drop, Eqn. 5.11 was evaluated using the following information. The inner diameter of the tubes was 0.125 in. The length of the freezer mullion tubes was approximately 33 ft while that of the refrigerator was 21ft. The number of 90° bends was estimated to be 20 (high due to the evaporative loop) for the freezer and 8 for the refrigerator, which correspond to a loss coefficient of 2.2 and 0.9, respectively. With a refrigerant mass flow rate of 8.6 lb_m/hr and 7.2 lb_m/hr in the freezer and refrigerator cycles respectively, the resulting flow was turbulent for the freezer cycle with a total pressure drop of 2.3 psi but laminar for the refrigerator with a 0.1 psi pressure drop.

To be on the conservative side, a pressure drop of 4 psi for the freezer and 2 psi for the refrigerator was assumed. The impact of this pressure drop was studied by using the computer program (described in Chapter 3) to predict the extra power consumed by the compressor. Everything else being equal, the compressor maps show that the power consumption increases by 5 Btu/hr for the freezer compressor and 4 Btu/hr for the refrigerator compressor as a result of these pressure drops. However, the compressors do not run 100% of the time. By accounting for the actual on-cycle time for these compressors, the increase in power consumption is only 2.0 Btu/hr for the freezer and 1.4 Btu/hr for the refrigerator. To express this increase in power as an equivalent cooling load (to enable a comparison with the impact of the mullion heat leak into the cabinet), the COP of the cycles is required. Using an average COP of 1.48 for the freezer and 2.30 for the refrigerator, the equivalent increase in cooling load is 3 Btu/hr for both the freezer and refrigerator cycles.

7.6 Comparison between Mullion Tubes and Electric Heater

The results show that both the freezer and refrigerator pay a heavy price for the use of mullion tubes to prevent sweating. The primary disadvantage lies in the additional cooling load that results from the flow of heat back into the cabinet. The estimated amount of heat that was rejected into the cabinet ranged from 53-84 Btu/hr for the freezer and 39 Btu/hr (upper bound of the estimate) for the refrigerator. Aside from the additional cabinet load, the high-side pressure drop had also added 3 Btu/hr to the freezer and refrigerator cooling loads, respectively. By adding the effects of these two disadvantages, the use of mullion tubes contributes to an equivalent of 56-87 Btu/hr in additional cooling load to the freezer and 42 Btu/hr to the refrigerator.

Alternatively, an electric heater could be employed to perform the functions of the mullion tubes. However, the use of electric heaters would increase the power consumption of the system and also add to the cooling load. Just as heat flows from the tubes into the cabinet, a certain degree of heat will leak into the cabinet from the warm heaters. By replacing the mullion tubes with electric heaters that were kept at a constant temperature of 75°F (to keep surface 2 of the middle frame in Figure 7.5 at 70°F), the amount of heat rejected by the heaters into the cabinet was 28 Btu/hr for the freezer and 25 Btu/hr for the refrigerator. In arriving at these figures, it was assumed that the heater was only turned on for the same percentage of time as the current mullion tubes, which was 41% for the freezer and 36% for the refrigerator. Furthermore, the surface area of the electric heaters, which would be much smaller, was assumed to be the same as that of the mullion tubes. This assumption, however, should not affect the results significantly as the majority of resistance to the heat transfer process is provided by convection. In this analysis, no heater was needed to warm surface 1 in Figure 7.2 of the freezer side frame as its current surface temperature of 67°F was sufficiently high to prevent sweating.

Therefore, the difference between the mullion tubes and the electric heaters, in terms of the additional cabinet load, is between 28-59 Btu/hr for the freezer and 17 Btu/hr for the refrigerator. To meet this additional load, the compressor has to run for a longer time, and consumes more energy in the process. The amount of extra power that it draws can be determined by dividing this additional load by the COP of the cycle. With a freezer COP of 1.48 and 2.30 for the refrigerator, the extra power consumption is between 19-40 Btu/hr for the freezer and 7 Btu/hr for the refrigerator. Using an electric heater to prevent sweating is then a better approach if it consumed less than 19 Btu/hr for the freezer and 7 Btu/hr for the refrigerator. By assuming that the heaters only operate over the same percentage of time as the mullion tubes, these figures correspond to a 14 W and 6 W heater for the freezer and refrigerator respectively. If the heaters were required to run continuously, it must have a lower power rating than this to emerge as the better choice.

7.7 Conclusions

The amount of heat that the freezer and refrigerator mullion tubes reject into the cabinet was found to be a significant portion of the total cabinet load. Based on the current cycle run times, the estimated heat leaking into the freezer cabinet was in the range of 53-84 Btu/hr while the corresponding estimate for the refrigerator was 39 Btu/hr. These rather high estimates account for a significant portion of the freezer cabinet load of 276 Btu/hr and refrigerator load of 197 Btu/hr. However, they can be rationalized by comparing the measurements of the cabinet load from the reverse heat leak test and the UA experiments.

In addition to the extra cooling load, the cycles also suffer from a high-side pressure drop. Although small, it contributes to the overall disadvantage of using mullion tubes to prevent the occurrence of sweating on the surfaces of the frame.

Like mullion tubes, electric heaters also reject some heat into the cabinet when they are in operation. The degree of heat leak is, however, smaller due to the lower temperature of the heater. Apart from the smaller amount of heat rejection into the cabinet, the electric heaters can also be controlled by the user to save energy when the relative humidity is low. A comparison between the use of electric heaters and mullion tubes shows that the former is only a better method if it consumes less than 14 W for the freezer and 6 W for the refrigerator, based on a similar run time as the mullion tubes.

CHAPTER 8 Two-Stage System

8.1 Introduction

In a vapor-compression cycle, the power drawn by the compressor is a function of the operating conditions. Specifically, the compressor power is a function of the refrigerant pressure at the compressor suction and discharge. When the compressor is forced to operate over a larger pressure ratio, the amount of work that it performs increases. Although the power consumption is more sensitive to the conditions on the low-side, the evaporating pressure of the refrigerant is dictated by the temperature at which refrigeration is provided and the size of the evaporator. On the other hand, the condensing pressure depends on the ambient temperature to which heat is rejected and the size of the condenser. Without the ability to control the ambient or cabinet setpoint temperatures, the pressure differences that the compressor operates over is essentially fixed.

However, if two compressors were used, the pressure ratio that each operates over may be reduced. By arranging them in series, the first compressor would only be responsible for pumping the refrigerant to an intermediate pressure, where the remaining work of raising the refrigerant to its final pressure is performed by a second compressor. From a thermodynamic standpoint, the use of two compressors to operate over a fixed pressure ratio is more efficient than using only one, regardless of the intermediate pressure.

As effective as this approach may be in reducing the power consumption, it invariably requires the use of two compressors. While this constitutes an additional cost to most refrigerators which only use one cycle (and compressor) to provide refrigeration, the refrigerator under investigation currently provides cooling through two separate cycles using independent compressors. Without having to contend with an additional compressor, this study is justified by the potential benefits that a two-stage cycle may offer.

8.2 Advantages and Disadvantages of a Two-Stage System

When the pressure ratio that the system operates over is split between two compressors arranged in series, a saving in compressor energy consumption can be expected. This is the essence of a two-stage cycle and is the main advantage of this approach. Due to its low evaporating temperature, the single-stage pressure ratio in the freezer cycle is relatively high. In contrast, the single-stage pressure ratio is lower in the refrigerator which has a higher cabinet setpoint temperature. By only requiring the freezer compressor to pump to an intermediate pressure, which would correspond to the evaporating pressure of the refrigerator in this case, the work of the freezer compressor is substantially reduced.

Figure 8.1 shows a possible configuration for a two-stage system. In this two-stage configuration, both compressors are required to run simultaneously so that the refrigerator cycle is able to provide saturated liquid at the intermediate pressure for the use of the freezer cycle. Because both cycles are individually controlled to meet the cabinet loads, the issues that surround the control of the cycles to run simultaneously are complex. To enable both cycles to run together, the freezer and refrigerator compressors would have to be correctly sized (based on their cabinet loads) so that their run times are matched. Not only is this a difficult procedure, but the resulting two-stage system would be vulnerable to any deviations from the design conditions (like a change in the ambient temperature).

In the configuration shown in Figure 8.1, a device known as the flash evaporator is used to store liquid refrigerant that is used by the freezer cycle. Due to its storage content, which is refrigerant at a low temperature, the flash evaporator cannot be placed outside the refrigerator. Therefore, additional space must be located in the refrigerator to house this device. However, this vessel can be designed to only hold a relatively small amount of refrigerant of about 5 lb since both cycles are be required to run simultaneously.

As a result of the reduced pressure ratio, the freezer cycle experiences an increase in performance. The opposite is true for the refrigerator. Although the pressure ratio over which the refrigerator compressor operates remains the same, it is required to pump more mass in this configuration. Because the energy consumption of the compressor is directly proportional to the amount of refrigerant that it pumps, the COP of the refrigerator cycle decreases accordingly. Ultimately, the increase in the freezer COP may not be sufficient in offsetting the decrease in the performance of the refrigerator, which result in a net decrease in system COP.

The advantages and disadvantages of using a two-stage system are summarized in below.

Advantages	**Disadvantages**
Effectively reduces the pressure ratio, which results in a decrease in compressor consumption for the freezer cycle	The decrease in refrigerator COP may outweigh the increase in freezer COP, resulting in a net decrease in system performance
Transfers load from the freezer compressor to the refrigerator compressor	Sizing of the compressors is important to allow both cycles to run simultaneously
	Additional space in the refrigerator is needed to locate the flash evaporator

Table 8.1 The advantages and disadvantages of a two-stage system.

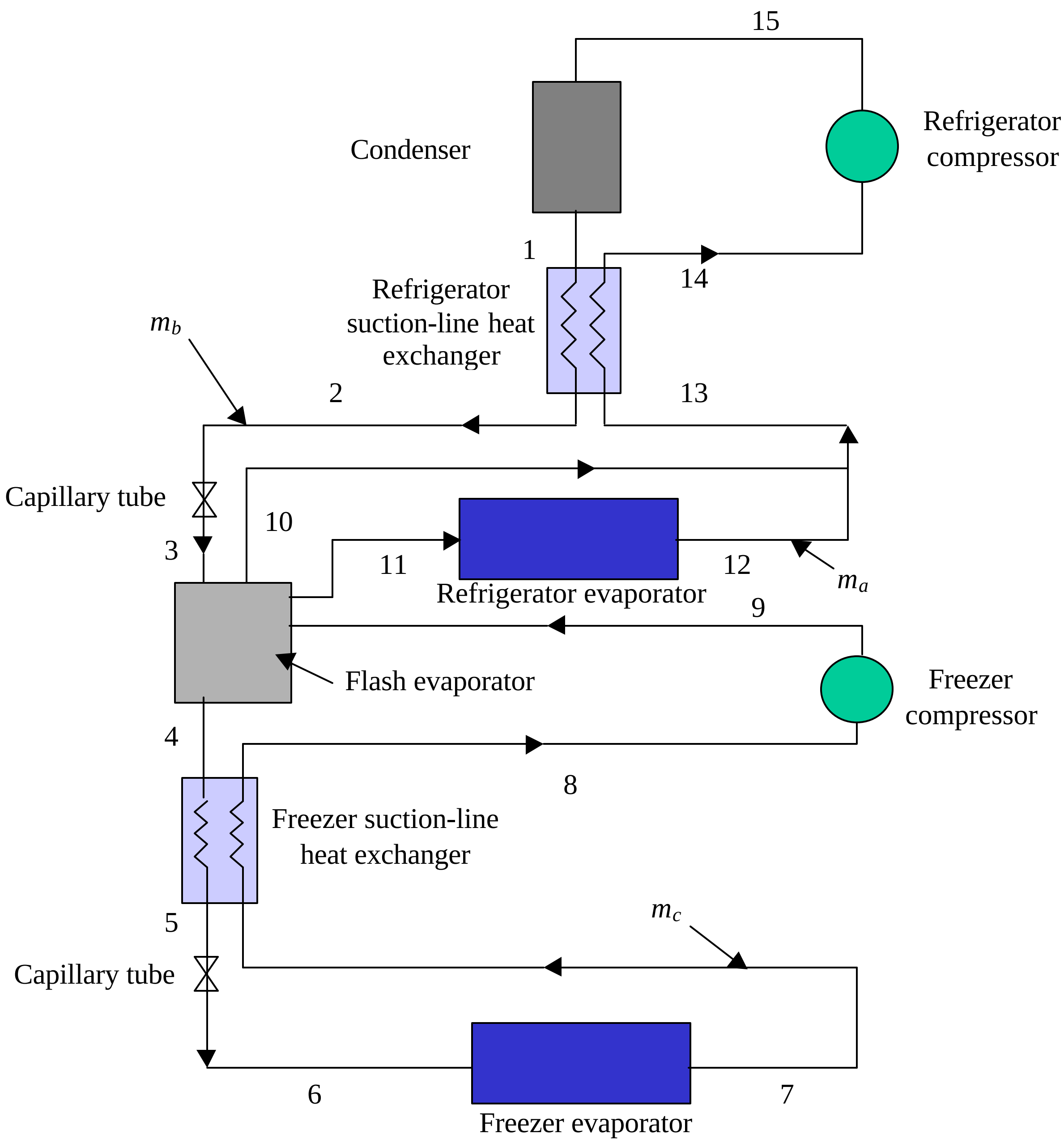

Figure 8.1 Schematic showing the configuration of the proposed two-stage system. The state of the refrigerant and the mass flow rate in the freezer and refrigerator cycles are also shown.

8.3 Analysis of the Two-Stage System

The main objective of this analysis was to compare the performance of a two-stage system to the current system which utilizes two single-stage cycles to independently provide cooling to the freezer and refrigerator cabinets. The configuration of the two-stage cycle that was studied is shown in Figure 8.1 while the two-cycle system uses two ordinary vapor-compression cycles for the freezer and the refrigerator. In this analysis, the performance of both the two-stage and the two-cycle systems were evaluated using a steady-state approach. The performance of the overall system is measured by the total energy consumption of the two compressors, which was expressed as follows.

$$TotalEnergy = \left(Power_{frez} + Power_{refg}\right) \times \%RunTime \times TotalTime \qquad \text{Eqn. 8.1}$$

where *TotalEnergy* = total energy consumed during a test period (Btu)
%RunTime = the percentage of time that the compressor is in operation
Power = power drawn by the compressor (Btu/hr)
TotalTime = the total time over a particular test (hr)

By evaluating the total energy consumption of the system, a direct comparison can be made between the performance of the two-stage and the basic two-cycle system. To ensure that the potential benefits of the two-stage system were only a result of its configuration, all the components used in the two-stage and two-cycle system were similar. Specifically, the compressors and the suction-line heat exchangers were identical. In addition, the same condensing and freezer evaporating temperature were used in both analyses (the intermediate pressure will be discussed shortly).

8.3.1 Assumptions made in the Analysis

Unlike the compressor maps that were used in the computer model, the performance of the compressors in this analysis was characterized by an isentropic efficiency because the intermediate pressure that the freezer compressor pumps to lies beyond the range of the maps. An isentropic efficiency of 0.5 was assumed for both compressors (the isentropic efficiency of the present compressors in this refrigerator is around 0.5).

The condition of the refrigerant at a number of locations in the cycle was also assumed. Referring to Figure 8.1, the refrigerant was assumed to be saturated liquid at location 1 while the assumption of saturated vapor conditions were made at locations 7 and 12. Although the results will vary depending on the condition of the refrigerant at the outlet of the evaporators and condenser, they are not sensitive to the assumptions made here. Figure 8.2 identifies the condition of the refrigerant in the two-stage system on a P-h diagram, where the numbers correspond to its location in Figure 8.1.

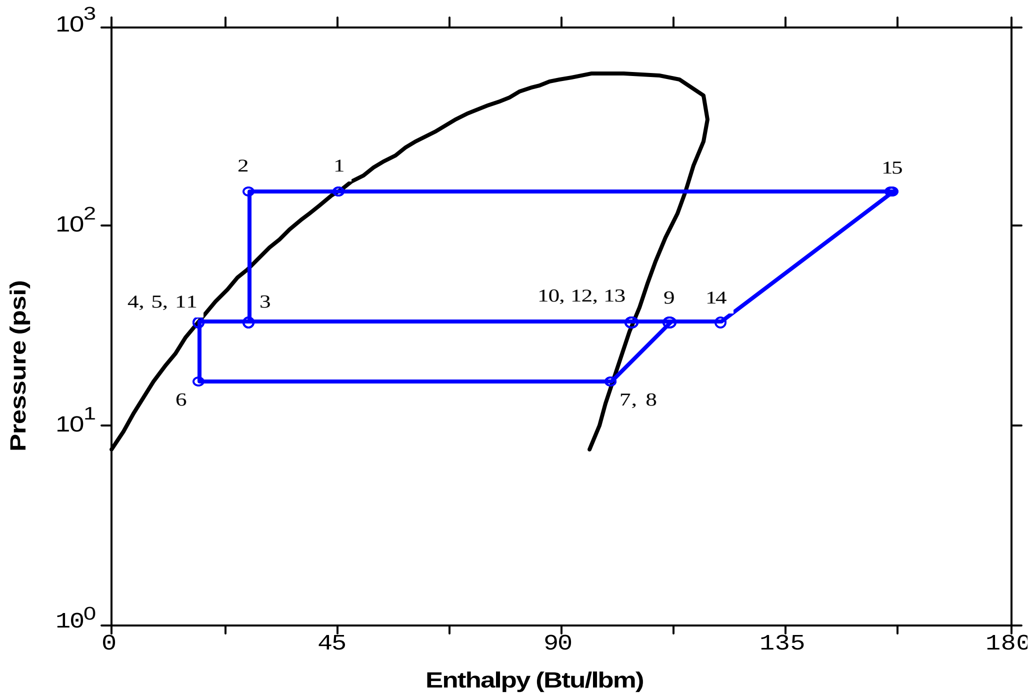

Figure 8.2 The state of the refrigerant on a P-h diagram. Note that the above plot represents the case when a suction-line heat exchanger was not used for the freezer (the reason for doing so will be discussed later).

8.3.2 Choice of the Intermediate Pressure

With a fixed pressure on the low and high-side of the system, the total power consumption of the two compressors is a function of the intermediate pressure. For an ideal gas, the optimum intermediate pressure that minimizes the work during a two-stage compression is given by Eqn. 8.2 (Threlkeld, 1962).

$$P_i = \sqrt{P_a P_b} \qquad \text{Eqn. 8.2}$$

where P_a = suction pressure of the freezer compressor
P_b = discharge pressure of the refrigerator compressor

For typical refrigerants, a small deviation from this prediction is expected. Figure 8.3 shows the impact of the intermediate pressure on the total energy consumption of the system.

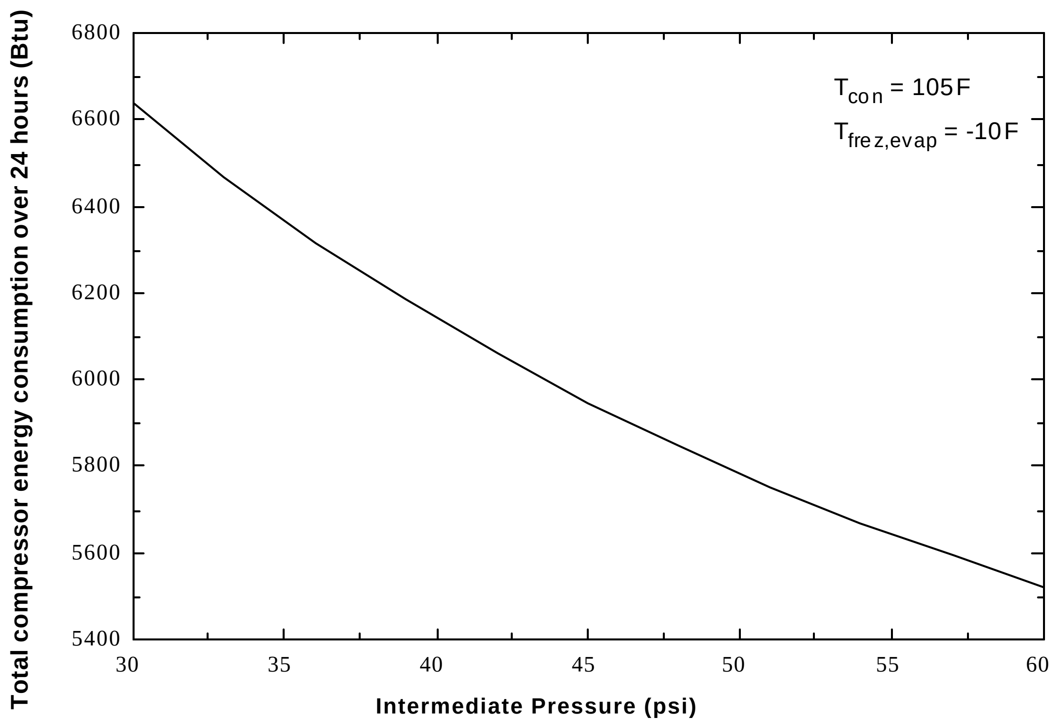

Figure 8.3 Total energy consumption of the system as a function of the intermediate pressure.

Although the performance of the system is enhanced when the intermediate pressure is increased, it should be remembered that refrigeration is provided to the refrigerator cycle at this intermediate pressure. To maintain a reasonable amount of heat transfer between the air and refrigerant, the intermediate temperature (and pressure) is limited by the setpoint temperature of the cabinet air. In the present refrigerator, the evaporating temperature of the refrigerant is approximately 20ºF. With this in mind, the intermediate temperature was set to 20°F for the two-stage system, which corresponds to a saturation pressure of 33 psi. Similarly, the evaporating temperature in the refrigerator cycle was assumed to be 20°F for the two-cycle system.

8.3.3 Flash Evaporator

Other than the typical components of a vapor-compression cycle, the flash evaporator is an important addition to this two-stage system. It is kept at the intermediate pressure and functions as a device that enables refrigerant to be separated into liquid and vapor components. Since both the freezer and refrigerator cycles run simultaneously, all the liquid that is used by the freezer is immediately replenished by the refrigerator; therefore, the sizing of the flash evaporator is not crucial to this design.

Refrigerant enters the flash evaporator at a two-phase condition after being throttled through the expansion device to the intermediate pressure. In the flash evaporator, the liquid portion of the refrigerant that enters at state 3 mixes with the liquid that is already in this device while the vapor portion returns to the refrigerator compressor. Refrigeration for the freezer cycle is performed by the saturated liquid that leaves the flash evaporator at state 4. This refrigerant enters the flash evaporator again after it leaves the discharge of the freezer compressor. The superheated vapor returning form the low-stage compressor discharge is cooled by the liquid in the flash evaporator so that it leaves this device saturated at state 10. At the same time, a portion of the liquid that is stored in this vessel is also bled off to supply the refrigerator evaporator with refrigerant. Due to the substantial amount of heat that is dumped into the flash evaporator by the refrigerant in the freezer cycle, a portion of the liquid in this device evaporates. The diminishing liquid is, however, replenished by the condenser liquid. Hence, the refrigerator compressor is required to supply the flash evaporator with liquid that has been evaporated by the freezer cycle. An energy balance on this device yields the following expression.

$$\dot{m}_c\left(h_9 - h_4\right) = \dot{m}_a\, h_{11} + \dot{m}_b\, h_{10} - \left(\dot{m}_a + \dot{m}_b\right)h_3 \qquad \text{Eqn. 8.2}$$

where $\dot{m}_a, \dot{m}_b$ and $\dot{m}_c$ represent the mass flow rates identified in Figure 8.1. The subscripts associated with the enthalpy also correspond to the states in Figure 8.1.

8.3.4 Suction-Line Heat Exchangers

In the proposed two-stage configuration, the option of using suction-line heat exchangers was available. While it was shown in Chapter 5 that suction-line heat exchangers are beneficial to systems that use R134a, the impact of these heat exchangers was investigated here to determine whether their application in a two-stage system would advance the performance of the system. The effect of using these heat exchangers is shown in Figures 8.4 and 8.5.

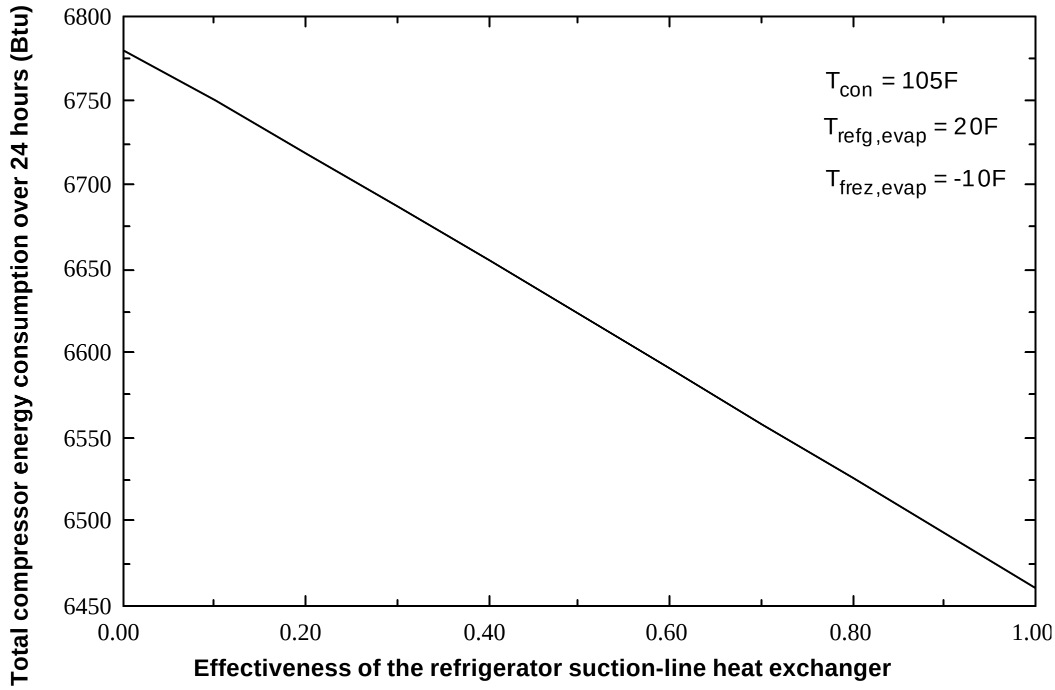

Figure 8.4 The impact of the refrigerator suction-line heat exchanger on the power consumption of the system.

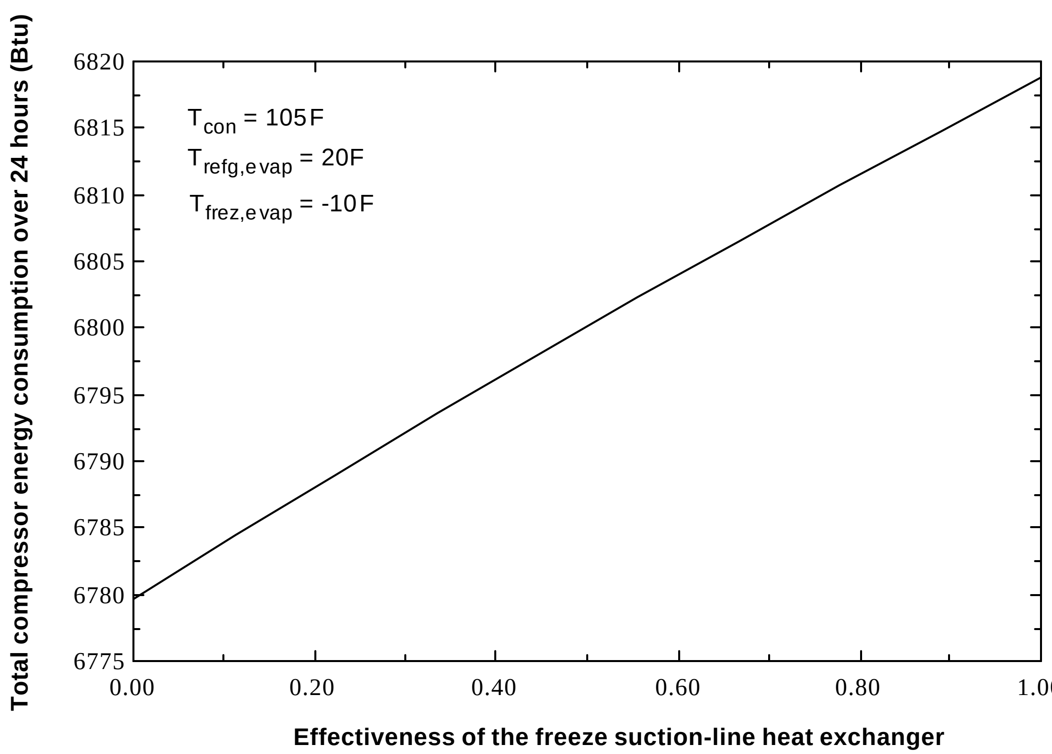

Figure 8.5 The impact of the freezer suction-line heat exchanger on the power consumption of the system.

As Figure 8.4 shows, the performance of the system (total energy consumption of the freezer and refrigerator compressors) is enhanced by the application of the refrigerator suction-line heat exchanger. Contrary to the results for the refrigerator, Figure 8.5 shows that using a suction-line heat exchanger on the freezer side hurts the performance of the system slightly. In Chapter 5, the benefits of using a suction-line heat exchanger were shown to be higher for cycles with greater temperature lifts. In this case, the decrease in performance for the freezer was, in part, due to the low intermediate pressure. Due to the small temperature lift in the freezer cycle for this two-stage system, the increase in refrigeration effect (per unit mass) was not adequate in offsetting the impact of the decrease in refrigerant mass flow (due to higher temperature vapor). The result of these opposing effects was a net decrease in the refrigeration capacity (or an increase in power due to a higher percentage of run time). Another reason that contributed to the increase in the compressor power consumption was the drop in low-side pressure. A pressure drop on the low side adversely affects the performance of the system because it decreases the refrigerant mass flow in the cycle. In view of these results, the effectiveness of the suction-line heat exchanger was set to zero and unity in the freezer and refrigerator cycles respectively. In the two-cycle system, the effectiveness of the suction-line heat exchangers was assumed to be 1 to maximize its COP, with a maximum pressure drop of 0.5 psi.

8.4 *Results from the Analysis*

The results in Table 8.2 show that the saving in energy consumption by using the two-stage system configuration over the two-cycle system was 2.6%. By reducing the pressure ratio of the freezer cycle from 9.0 to 2.0, the amount of energy that the compressor consumes decreases by 79%. As a result, the COP of the freezer increases from 1.47 when operating between temperatures of –10°F and 105°F to 7.02 when the condensing temperature decreases to 20°F. The high COP for the freezer cycle stems from the fact that the compressor consumes very little energy when operating over the small pressure ratio.

	Freezer cycle (Btu)	**Refrigerator cycle (Btu)**	**Total (Btu)**
Two-stage	943	5517	6460
Two-cycle	4493	2141	6634

Table 8.2 The energy consumption of the freezer and refrigerator cycles over a 24 hour period.

On the other hand, the energy consumption of the refrigerator compressor increases significantly. Although the temperature lift in this cycle remains the same, the amount of mass that the compressor delivers increases substantially. Not only was it responsible for circulating the refrigerant in the refrigerator cycle, but it was also required to provide condensed liquid for use of the freezer cycle. Therefore, the refrigeration capacity of the refrigerator cycle is unchanged because a large amount of the mass flow that the compressor delivers is used by the freezer cycle. Due to the increase in compressor consumption because of the larger mass flow of refrigerant, the COP of the refrigerator decreases from 2.24 to

0.87. It is this steep decrease in refrigerator COP that erodes most of the gains made by the freezer.

The rather small improvement in the overall performance of the system is due to the poor match between the low, intermediate and high-side temperatures in this two-stage system. To better illustrate this point, Figure 8.6 shows the improvement in system performance as a function of the freezer evaporating saturation temperature with a fixed condensing and evaporating temperature for the refrigerator cycle.

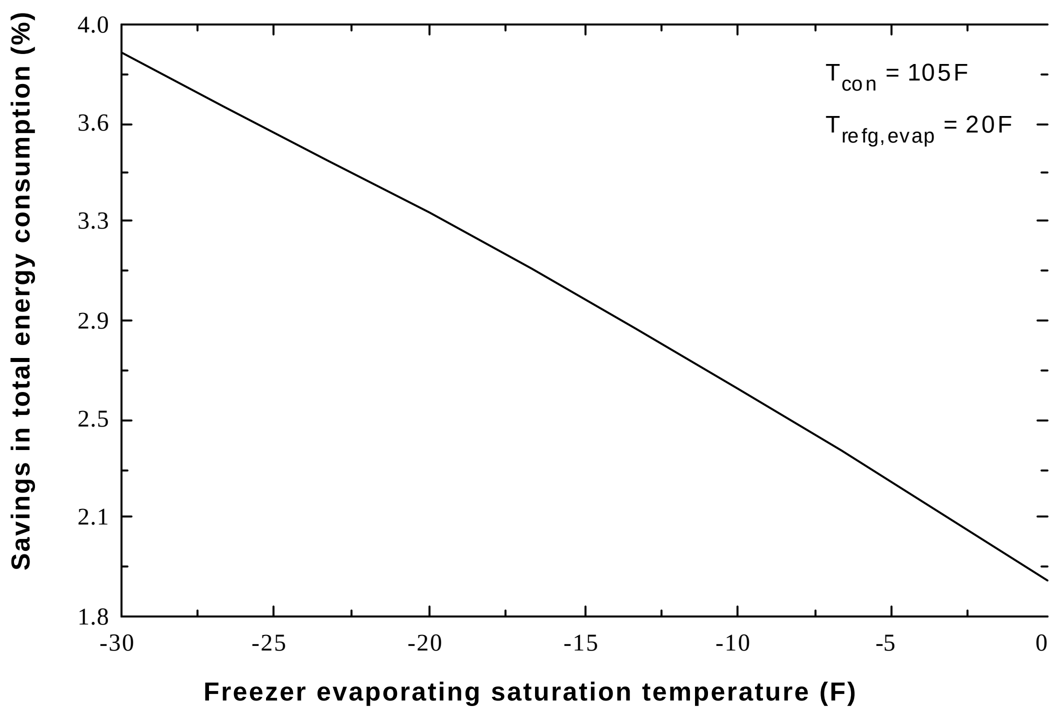

Figure 8.6 The benefit of using a two stage cycle as a function of the low-side evaporating temperature.

It is clear that the two-stage system offers a greater potential when the freezer evaporating temperature is low. In circumstances where the temperature lift is great, the arrangement of two compressors in series can significantly reduce the total energy consumption of the compressors. However, the rate of increase in benefits is seen to diminish as the evaporating temperature decreases further. The increase in benefits will continue to rise and eventually level out at an evaporating temperature lower than the –30°F shown in Figure 8.6. As a close approximation (because R134a is not an ideal gas) the evaporating temperature of the freezer at which the benefits of using a two-stage cycle reaches a maximum corresponds to the evaporating pressure which can be evaluated using Eqn. 8.2.

Another factor that influences the benefits of using a two-stage system is the assumption of the compressor isentropic efficiency in this analysis. Figure 8.7 shows the impact of the compressor efficiency on the percentage of savings in energy consumption of the two-stage system over the basic two-cycle system.

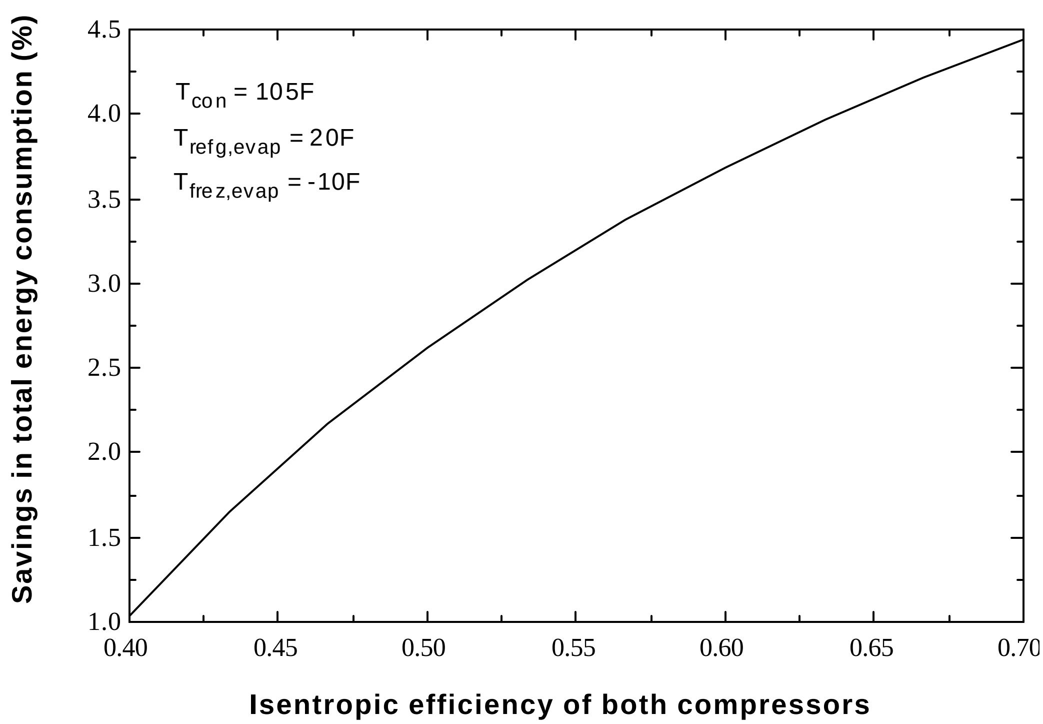

Figure 8.7 The percentage improvement of the two-stage system over the basic two-cycle system as a function of the compressor efficiency.

Figure 8.7 shows that the benefits of using a two-stage cycle increase with the efficiency of the compressors. Although the efficiency of the present compressors in this refrigerator is around 0.5, the performance of these compressors, particularly the freezer compressor, may be very different when it is only required to operate over a much smaller pressure ratio. If the compressors had an isentropic efficiency of 0.7, the improvement would be twice as much as that estimated in this study. Because the benefits of using this two-stage system over the two-cycle system is dependent on the performance of the compressor, compressor maps (if available) should be obtained to enable a more accurate prediction of the performance of the compressors at these low condensing temperatures.

8.5 Conclusions

The benefits of using a two-stage cycle over the basic two-cycle system was studied. Besides the normal components found in a vapor-compression cycle, the configuration of the proposed two-stage utilizes a flash evaporator. This device enables the storage of refrigerant at the intermediate pressure for the use of the freezer cycle.

Since refrigeration is provided to the freezer and refrigerator at the low and intermediate temperatures respectively, the choice of these temperatures is somewhat limited. Using a evaporating saturation temperature of –10°F for the freezer and 20°F for the refrigerator, with the ambient temperature set at 105°F, the total energy consumption of the compressors in the two-stage system was 2.6% less than that of the two-cycle system.

Two reasons can be offered to explain the minor gains demonstrated in this study. In the first, the isentropic efficiency of the compressor was assumed to be 0.5. Should the actual compressors be more efficient than this, the potential benefits would be higher than that calculated. The second, and perhaps the more important reason, is the poor match between the low, intermediate and high-side temperatures in this system. Because these temperatures are dictated by the operating conditions, they cannot be individually selected to optimize the performance of the two-stage cycle.

However, the behavior (isentropic efficiency) of a refrigerator compressor operating at such a low condensing temperature is not known since typical compressors do not operate at these conditions. The actual performance of the freezer compressor may be very different at low (33 psi in this analysis) discharge pressures. Instead of characterizing the performance of the compressor by a constant isentropic efficiency, compressor map data should also be obtained from compressor manufacturers to enable a more accurate prediction of the performance of this device at low discharge pressures. In view of the fact that no additional cost (except for the purchase of a flash evaporator) is required to convert the present two-cycle system to the proposed two-stage configuration, more efforts should be made to investigate the feasibility of this study.

CHAPTER 9 Conclusions and Recommendations

9.1 Summary of Results

This research project was aimed at developing a computer program to simulate the performance of a refrigerator and to propose and evaluate one or more methods that could enhance its performance. Because the overall heat exchanger coefficient-area product (UA) of the heat exchangers were inputs in this program, an experiment was carried out to measure this parameter.

In this experiment, the airflow rate through the heat exchanger, the temperature of the air entering and leaving the heat exchanger and the mass flow rate of refrigerant in the cycle were measured. Using these parameters, an energy balance was performed on the heat exchanger to yield the UA. Other than the uncertainties that stem from the measurements of temperature and airflow, the major source of error in the UA estimate was the assumption of 100% sensible cooling by the evaporator. Due to the uncertainties, the UA based on the mass flow meter measurements were deemed to be the most accurate.

By using these experimentally measured values and compressor maps supplied by the manufacturer, the computer program was able to predict the compressor power consumption and the on and off-cycle time of both the freezer and refrigerator cycles. To verify these predictions, the outputs from the program were compared to the compressor power and the cycle run times that were measured by the UA experiments. The results show that the predicted power consumption and off-cycle time was 6% and 2% higher than the measured values, respectively.

Besides measuring the UAs of the current heat exchangers, the impact of a larger coil UA on the performance of the system was studied by increasing the speed of the fans in these coils. The experiments confirm that the present condenser and refrigerator evaporator are accurately sized and no increase in performance is expected by increasing its size. On the other hand, the evaporator in the freezer was found to be undersized, and a 2% savings (including parasitic energy consumption) in the total energy consumption of the system can be realized by increasing the airflow rate by 40%. Apart from the sizing of heat exchangers, another objective of this experiment was to measure the performance of a set of DC fans. One such fan was used in the freezer evaporator and another in the refrigerator evaporator (a DC fan was not tested for the condenser). When compared to the present AC fans, the use of the freezer DC fan had saved 6.0% in energy consumption while the savings from the refrigerator DC fan was 2.8%.

In this refrigerator, suction-line heat exchangers have been installed in both cycles to increase the refrigeration effect. The impact of this device on the performance of a cycle was shown to be a function of the temperature lift and the extent of the low-side pressure drop across this heat exchanger. While cycles with higher temperature lifts had benefited more from the installation of this device, the performance of these cycles were also more vulnerable to a low-side pressure drop. Based on the estimated pressure drop of 0.2 psi across these heat exchangers, which have an effectiveness of 0.9, the use of these heat exchangers was shown to improve the performance of the freezer and refrigerator cycles by 8.3% and 5.4% respectively.

To prevent sweating on the outer surfaces of the cabinet, mullion tubes have been used to maintain the temperature of these surfaces above the dew point temperature of the environment. However, a portion of the heat that is rejected by these tubes invariably escapes back into the cabinet, thereby increasing the cooling load. In addition, the use of mullion tubes induces an additional pressure drop on the high-side of the system. Together, they contribute to an additional cooling load of between 56-87 Btu/hr (depending on the temperature of the mullion tubes) to the freezer and 42 Btu/hr to the refrigerator. The 42 Btu/hr for the refrigerator represents an upper bound to the estimate of the mullion heat rejection. Alternatively, an electric heater can be used to perform the functions of the mullion tubes. The study shows that electric heaters would only prove to be the more economical method of preventing the occurrence of sweating if it had a power rating of 19 Btu/hr or less for the freezer and 7 Btu/hr for the refrigerator (based on the same percentage of run time as the mullion tubes). Unlike mullion tubes, the operation of electric heaters can be controlled by the user. This is a particularly useful strategy in saving energy when the relative humidity is low.

Besides analyzing ideas that have already been implemented, a feasibility study was conducted on three proposals, which were mechanical subcooling, the simultaneous running of the freezer and refrigerator cycles, and the use of a two-stage system. Like a suction-line heat exchanger, a subcooler attempts to increase the refrigeration capacity of the cycle by subcooling the liquid that leaves the condenser. This is accomplished by transferring heat from the warm condenser liquid in the freezer to the suction gas in the refrigerator cycle. If suction-line heat exchangers were not employed in the freezer and refrigerator cycles, this technique was capable of increasing the system COP by 4.7%. Since both the freezer and refrigerator cycles already use a suction-line heat exchanger, the potential benefits of using this subcooler was only 1%.

Due to the large temperature lift in the freezer cycle, the potential benefit of using a two-stage system in place of the present two single-stage cycles was explored. Using the proposed two-stage system, a 2.6% savings in total energy consumption was demonstrated. The main reason for the small gain in performance is due to the poor match between the low, intermediate and high-side temperatures in this two-stage system. Because these temperatures are dictated by the cabinet and ambient temperatures, they cannot be randomly selected to maximize the benefits.

At present, the freezer and refrigerator cycles run independently. By forcing the cycles to run simultaneously, a portion of the energy that the condenser fan consumes can be saved. On the other hand, the condensing temperature of both cycles increases when they run together as they presently share the same condenser. A preliminary analysis has shown that the increase in compressor work due to the increase in condensing pressure exceeds the savings from the condenser fan. Based on the results from this analysis, the overall performance of the system should be enhanced when the freezer and refrigerator cycles run separately. The differences between the simultaneous and separate running of the cycles are, however, small and the issues that surround the control of these cycles to meet their individual cabinet loads are complex. Therefore, the small potential savings demonstrated in this analysis do not justify a further study into this matter. Table 9.1 ranks the proposals that were studied in this research (including those that have already been implemented) according to its potential benefits.

Proposal	Savings in Total Energy Consumption (%)
Freezer suction-line heat exchanger	8.3%
DC fan in the freezer evaporator	6.0%
Refrigerator suction-line heat exchanger	5.4%
DC fan in the refrigerator evaporator	2.8%
Two-stage cycle	2.6%
Increasing freezer evaporator UA	2.0%
Mechanical subcooling	1.0%
Mullion tubes	If freezer heater > 19 Btu/hr and refrigerator heater > 7 Btu/hr

Table 9.1 The estimated impact of each proposal on the performance of the system.

9.2 Recommendations for Future Work

The UAs of the evaporators and condenser were measured by three different methods (refer to Chapter 2 for detail explanation). In all three heat exchangers, the estimated UA was always the lowest when the method that performs an energy balance on the evaporator was used as the basis of analysis. The condensation of water in the evaporator was the best reason that could be offered to explain the observed differences between the UA estimates. To determine the validity of this explanation, the drain pan should be regularly checked for any accumulation of water. The existence of water would confirm the explanation that latent loads were indeed responsible for the lower estimate of the heat exchanger UAs based on the evaporator method.

Another source of uncertainty in the calculation of the coil UA was the error associated with the measurement of the air temperatures entering and leaving the heat exchangers. Based on calibration tests in an ice bath and at the ambient temperature (90°F), the thermocouples used in the measurements were only accurate to ±1°F. For the calculation of the condenser and freezer evaporator UA, this uncertainty is significant because the difference in the air temperatures across these two heat exchangers are small. To improve on the accuracy of the temperature measurements, a thermopile can be used. In a thermopile, two or more thermocouple are connected together to produce N junctions, with 1/2N junctions measuring the temperature of air at one location and the remainder placed at another location to measure its temperature. The voltage output from these measurements would then represent an average value of the difference in the temperature measured at the two locations.

In the study of suction-line heat exchangers, the performance of cycles using this device was found to be very sensitive to the pressure drop on the low-side. Although the estimated pressure drop across these heat exchangers was low, measurements of the loss in pressure on the low-side should be made to verify these estimates. This is because a significant pressure drop will reduce the benefits of the increase in refrigeration effect and the installation of these devices may ultimately hurt the performance of the system.

The impact of mullion heat rejection on the cabinet load was estimated to be significant for both cycles. To confirm this estimate, a test should be performed on the refrigerator, but with the mullion tubes removed. The amount of heat that results from the operation of these tubes can then be calculated by measuring the difference in the cabinet load from this test and the tests that have been already been performed (those that were used to measure the heat exchanger UAs).

A gain of 2.6% was demonstrated by converting the present two single-stage cycle system in the refrigerator to a two-stage system. In arriving at this conclusion, the isentropic efficiency of the compressor was assumed to be independent of the discharge pressure, which may not be the case in reality. In this regard, compressor maps (if available at low condensing saturation pressure) should be obtained to enable a more accurate prediction of the performance of the compressor at these low discharge pressures. Finally, a prototype of the two-stage system discussed in Chapter 8 could also be build and tested to verify the results from the analysis.

BIBLIOGRAPHY

AHAM, “American National Standard for Household Refrigerators/Household Freezers”, *Chicago : Association of Home Appliance Manufacturers*, 1985

Churchill, S.W. and H.H.S. Chu, Correlating Equations for Laminar and Turbulent Free Convection from a Vertical Plate”, *Int. J. Heat Mass Transfer*, *18, 1323*, 1975

Department of Energy, “Energy Conservation Standards for Refrigerators, Refrigerator-Freezers and Freezers”, 1997
http://www.eren.doe.gov/buildings/codes_standards/rules/refrule/viewer.htm

Department of Energy, “Refrigerator Energy Efficiency Standards”, (R-97-027)
http://www.osti.gov/html/doe/whatsnew/factshet/fs97027.html

Energy Information Administration, “Household Energy Consumption and Expenditures 1993”, 1995
http://www.eia.doe.gov/emeu/recs/recs2b.html

Giscard d’Estaing, V.A., “The Second World Almanac Book Of Inventions”, *Compagnie 12*, 1986

Gosney, W.B., “Principles of Refrigeration”, *Cambridge University Press*, *New York*, 1982

Haaland, S.E., “Simple and Explicit Formulas for the Friction Factor in Turbulent Pipe Flow”, *J. Fluids Eng.*, *pp 89-90*, 1983

Incropera, F. P. and DeWitt, D.P., “Introduction to Heat Transfer, 3rd ed.”, *John Wiley & Sons*, *New York*, 1996

Klein, S.A., *et al*., “FEHT : Finite Element Heat Transfer”, *University of Wisconsin-Madison, Version 7.023*, 1998

Klein, S.A and Alvarado, F.L., "EES : Engineering Equation Solver", *University of Wisconsin-Madison, Version 4.900*, 1998

Lawrence Berkeley National Laboratory, "Appliance Efficiency Standards, Part 2", 1995 http://eande.lbl.gov/CBS/NEWSLETTER/NL7/APS.html

Moran, M.J. and Shapiro, H.N., "Fundamentals of Engineering Thermodynamics", *John Wiley & Sons, New York*, 1993

Oak Ridge National Laboratory, "Fridge of the Future: ORNL's Refrigeration R&D", 1997 http://www.ornl.gov/ORNLReview/rev28_2/text/fri.htm

Sub-Zero Freezer Company, *4717 Hammersley Road, Madison, WI 53711*

Thornton, J.W. *et al*., "Dedicated Mechanical Subcooling Design Strategies for Supermarket Applications", *Int. J. Refrig., Volume 17, Number 8*, 1994

Threlkeld, J.L., "Thermal Environmental Engineering", *Prentice Hall, Englewood Cliffs, N.J.*, 1962

White, F.R., "Fluid Mechanics, 2nd ed.", *McGraw-Hill, New York*, 1986