LAB E – REFRIGERATION

Group 4

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July 20th,2015

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Date: July 20th, 2015

Subject: CHG 3122, Refrigeration

The objective of the refrigeration lab is to measure refrigerant capacity and performance coefficient for the refrigeration unit provided. These factors are to be determined as a function of heat load and circulation of refrigerant. Subsequently, the refrigerant unit allow the comparison of its use in normal capillary mode with that of thermostatic expansion valve (TXV) mode. This report provides the answers to problem set 2

A Scott Air Condition and Refrigeration system was used with a working fluid of Freon-12 in this experiment. The system ran in two operating modes: normal capillary mode and TXV mode. The capillary mode ran at varying flow rates. Two trials were done in capillary mode with each at a different level of refrigerant, which were 9.4 and 11.5cm, in the system. The TXV ran at the higher liquid level of refrigerant while varying the fan speeds on the condenser and compressor. The level of refrigerant was adjusted by using an upstream valve. In the three trials, the temperature and pressure were taken at five specific points and consequently used to calculate entropy and enthalpy. The system was then compared to the ideal case to compare their efficiency.

When calculating vapour quality at point 4, the throttle process is assumed to be isenthalpic with no heat loss to the surroundings and kinetic and potential energy to be negligible. The vapour quality for the first and second capillary run ranged from 0.130 to 0.134 and 0.162 to 0.153 respectively. The vapour quality for the TXV mode ranged from 0.163 to 0.185. When modifying a fan speed, a decrease in the power to the condenser fan lead to $COP_{maximum}$, COP_{actual}, COP_{fluid} and compression cycle efficiency decrease while the compression efficiency and the compression ratio increased. On the other hand a decrease in the power to the evaporator fan decreased COP_{actual}, compression cycle efficiency, compression ratio while no specific trend was observed for $COP_{maximum}$, COP_{fluid}, compression efficiency and compression ratio. The overall efficiency of the TXV was shown to better. For the capillary mode, heat loss decreased from 42.7 to 35.0 Btu/min with flow rate at higher refrigerant levels and from 35.4 to 40.2 Btu/min lower levels. The capillary mode with more refrigerant in the system showed lower ratio of heat loss to the surrounding at 0.75 at flow rates of 1.7 lb_m/min. The cycle efficiency was never over 1. If it were over 1, it could be due to thermocouple 1 and 4 malfunctioning.

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NOMENCLATURE:

COP_{max}	Ideal Carnot Efficiency
COP_{actual}	Actual coefficient of performance
Q_C	Rate of energy absorbed in a cold reservoir per unit mass of refrigerant
Q_H	Rate of energy rejected I na hot reservoir per unit mass of refrigerant
W	Power input to the compressor
T_C	Temperature of the cold reservoir
T_H	Temperature of the hot reservoir
COP_{fluid}	Coefficient of performance of the fluid
h_n	Specific enthalpy of the working fluid at different points of the system (n)
RC	Refrigeration capacity
m_f	Circulation rate of the refrigerant
η_{comp}	Compression efficiency
η_{cycle}	Cycle efficiency
CR	Compression ratio

1. Equipment and Procedure:

In this experiment, a Scott Air Condition and Refrigeration system with Freon-12 is used, in the study of the refrigeration process, as a refrigerant. The experiment setup is shown in Figure 1. Where the components in blocks are listed below:

- Compressor (A)
- Condenser (B)
- Evaporator (C)
- Capillary expansion (D)
- Thermostatic expansion valve (E),
- Temperature sensor (F)
- Moisture and liquid sensor (G)
- Rotameter (H)
- Dryer (I)
- Liquid refrigerant tank (J)
- Oil and refrigerant accumulator tank (K)
- Oil storage tank (L)
- System valves (V).

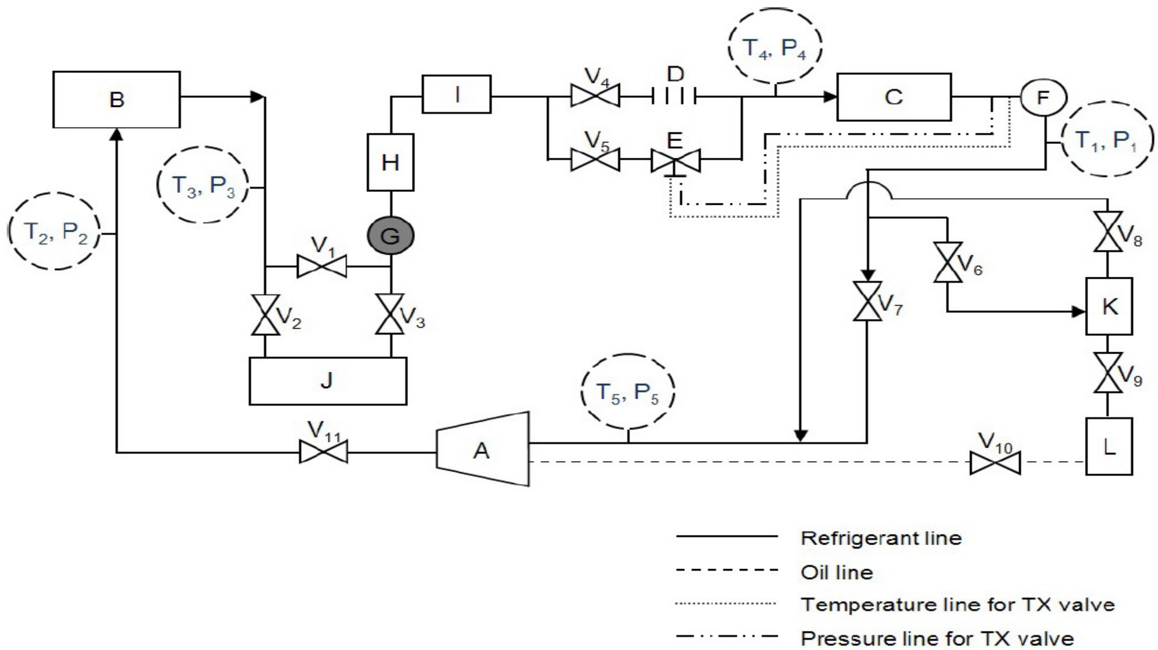
In addition to the equipment listed above, five temperature and pressure gauges, indicated by T and P on , were strategically placed in the system. To add additional cooling to the compressor and condenser, a three speed fan, with high, medium and low settings, were placed behind both components. A liquid level equipped to the liquid refrigeration tank (J) to monitor the amount of refrigerant in the system. Finally, there was a wattmeter, current meter, potentiometer and electrical switches to monitor and control the system.

The refrigeration process can operate in three modes:

1. Normal capillary mode
2. Thermostatic expansion (TX) mode
3. Reverse mode.

The system in the experiment operated in on the first two modes. Each mode receives liquid at high pressure. In the capillary tube, the frictional loses due to pressure drop causes the refrigerant liquid to evaporate. In this mode the capillary valve is extended further than in industry to assume complete evaporation. The thermostatic expansion sensor measures the temperature of the stream and consequently expands or contracts a temperature-sensitive bulb further downstream. The bulb contains Freon-12 which changes with temperature to alter the flow rate of the refrigerant passing through the system.

Figure 1: Diagram of components of the refrigeration unit



Before starting the system, the pressure and temperature gauges are compared and recorded to the ambient environment to ensure thermo-equilibrium. The fans are then turned on to high speed and will never fully be closed till the experiment is finished. The system was then turned on and the wattmeter, current meter and potentiometer value were recorded with just the fans on. A note when operating the system, when the pressure rises above 200psig the compressor must be turned off.

The first run involves setting the system to be in capillary mode by closing the valve to the TX valve. Both fans operated on the high setting. For this run the refrigerant was set to a level of 10.1 centimeters on the liquid meter and ran at 1.718 pound mass per minute. Note that when setting the level of refrigerant all value must be recorded before the refrigerant level increases to

0.4 above the initial value or the refrigerant level must be reset. The refrigerant may be adjusted by V_8 in figure 1 to increase the level since V_6 is broken. Pressure P_1 must be monitored to make sure it does not exceed the amount the supervisor has allowed. The temperature and pressure at all five gauge location and the power were recorded once the system is at steady-state. The trial is repeated at 4 more mass flow rates: 1.61, 1.502, 1.394 and 1.286 lb_m/min. The flow rates are monitored on a rotameter.

In the second run, the TXV valve was used by closing V_4 and opening V_5. When modifying your level of refrigerant, V_4 must be open and for this run it was set to 12.3 cm. The system was then run at four different fan combinations:

1. High condenser fan and high evaporator fan
2. High condenser fan and medium evaporator fan
3. Medium condenser fan and high evaporator fan
4. High condenser fan and low evaporator fan

The temperature and pressure at all five gauge location and the power were recorded once the system is at steady-state.

The third run the capillary mode was run again as in run one but at a lower level refrigerant in the system by leaving the level at 12.3 cm. The mass flow rate used in this run were 1.502, 1.394, 1.286 lb_m/min. As per previous run the refrigerant level was checked at each trial to be in the tolerance of 0.4 cm and the temperature and pressure at all five gauge locations and the power were recorded.

2. Summary of Results:

Question Set 2:

1. The quality of vapour at point 4 is calculated using Equation 6.82a (J.M Smith, 2005) which is shown in the sample calculations. This equation assumes that when a system consists of saturated liquid and saturated vapour coexisting in equilibrium, the total value of the enthalpy of the two-phase system is the sum of the total properties of the phases (J.M Smith, 2005). Moreover, it is assumed that the throttling proceeds without any exchange of enthalpy nor specific enthalpy and the heat loss to the surroundings is negligible.

2. The effect of fan settings used in TXV mode on the performance of the refrigeration system was analysed by calculating various coefficients of performance. Decreasing the power to the evaporator while keeping the power to the condenser constant had the following effects.
   a) The $COP_{maximum}$ ranged from 8.26 to 10.12 and no specific trend was observed.
   b) The COP_{actual} decreased from 1.963 to 1.767.
   c) The COP_{fluid} ranged from 8.98 to 8.44 and no specific trend was observed.
   d) The compression efficiency ranged from 0.179 to 0.184 and no specific trend was observed.
   e) The compression cycle efficiency decreased from 0.238 to 0.182.
   f) The compression ratio decreased from 2.86 to 2.42.

Decreasing the power to the condenser fan while keeping the power to the evaporator fan constant had the following effects.

   g) The $COP_{maximum}$ decreased from 8.26 to 7.75.
   h) The COP_{actual} decreased from 1.963 to 1.828
   i) The COP_{fluid} decreased from 8.98 to 7.48.
   j) The compression efficiency increased from 0.179 to 0.193.
   k) The compression cycle efficiency decreased from 0.238 to 0.236.
   l) The compression ratio increased from 2.86 to 2.98.

3. For a flowrate of 1.750 lb_m/min and a similar level of refrigerant of 11.5, the $COP_{maximum}$, COP_{actual} , COP_{fluid}, compression efficiency, compression ratio and cycle efficiency were used to compare the performance of the system in the normal capillary (2nd trial) and the

TXV modes. The $COP_{maximum}$, COP_{actual} and COP_{fluid} values for the normal capillary mode were calculated to be 8.49, 1.945 and 9.43, while similar values for the TXV modes were 8.26, 1.963 and 8.98. The compression efficiency, cycle efficiency and compression ratio for the normal capillary mode were 0.152, 0.229, and 2.98 while similar values for the TXV modes were 0.179, 0.238 and 2.86. The TXV mode had a higher cycle efficiency, compression efficiency and COP_{actual} and as expected the system would perform better in the TXV mode. Please refer to Table 4 and Table 5 for the tabulated results.

4. A list of figures for the heat loss from the compressor is located in Table 6and Table 7. The trends of the system are located below:
    a. During the first run, the system operated at a lower level refrigerant and in normal capillary mode. The flow rates were increased from 1.308 to 1.696 lb_m/min and consequently the heat loss went from 39.0 to 35.0 Btu/min. The amount of heat loss compared to work done by the compressor varied from 0.91 to 0.75.
    b. During the second run, the system operated at a high level refrigerant and in TXV mode. The flow rate was kept constant at approximately 1.7 lb_m/min and the fan speeds were varied. When the condenser fan was set to high, the evaporator fan varied between high and low going 37.1 to 38.4 Btu/min and a heat loss to work required is 0.78 to 0.76. The fan was then kept at high for the evaporator and the fan on condenser was put from high to medium. The heat loss varied 37.1 to 37.4 Btu/min and the ratio was from 0.79 to 0.78.
    c. During the third run, the system operated at a high level of refrigerant and the flow rates were increased from 1.308 to 1.696 lb_m/min and consequently the heat loss went from 40.2 to 35.4 Btu/min. The amount of heat loss compared to work done by the compressor varied from 0.92 to 0.77.
5. For the capillary run with lower refrigerant the cycle efficiency varied from 0.20 to 0.23 with flow rate, shown on Table 3. For the TXV run, the condenser fan was kept at high and cycle efficiency ranged from 0.18 at low to 0.24 at high, shown on Table 4. When the evaporator fan was kept at high and the condenser fan varied, the cycle efficiency was kept constant at 0.24, shown on Table 5. In this experiment, the cycle efficiencies were never above one.

3. Discussion:

1.) The quality of vapour at point 4 is calculated using Equation 6.82a (J.M Smith, 2005). One of the key assumptions used in the calculations for vapour quality is that the system operates in isenthalpic conditions and no heat was lost to the surroundings. Furthermore, while calculating the vapour quality it was assumed that the kinetic and potential of the refrigerant as it flows through the valve is negligible. This thereby reduces the energy balance equation to;

$$\triangle H = Q + W \quad (1)$$

As the fluid passes through point 4 it is expanded and brought to or below its saturation pressure. Since no work is associated with expansion and the heat loss is negligible *Equation 1* reduces to

$$\triangle H = 0, \; H_1 = H_2$$

Finally, the vapour quality assumes that when a system consists of saturated liquid and saturated vapour coexisting in equilibrium, the total value of the enthalpy of the two phase system is the sum of the total properties of the phases (J.M Smith, 2005). Hence

$$x_v = \frac{H - H^l}{H^v - H^l} \quad (2)$$

H_V = Specific enthalpy of the saturated vapour

H_L = Enthalpy of the saturated liquid.

The point 4 represents the point in the refrigeration unit after the fluid has been expanded by the capillary or thermostatic feed valve. The expansion valve removes pressure from the liquid refrigerant to allow expansion or change of state from a liquid to a vapor in the evaporator. The orifice within the valve is used to reduce pressure and under a greatly reduced pressure, the liquid refrigerant is at its coldest when it leaves the expansion valve and enters the evaporator (Pette Hoffam, 2006). In Figure 3, the area under the curve represents the vapour-liquid flow and point 4 of the carnot cycle represents a two phase mixture of vapour and liquid.

For the first capillary run, as the flowrate decreased from 1.6964 lb_m/min to 1.3076 lb_m/min, the vapour quality remained fairly constant between 0.130 and 0.134. For the second capillary run, as the flowrate decreased from 1.6532 lb_m/min to 1.2968 lb_m/min, the vapour quality ranged from 0.162 to 0.153.

2.) The temperature of the refrigerant leaving is being controlled by the thermostatic expansion valve. The flow of the refrigerant is varied by 2 fans which are located behind the evaporator and the condenser. Different fan settings, for example, high-low, low – low, and low – high are used to vary the flow of the refrigerant. During the TXV mode, two phase flow was observed for fan combination high on the condenser and medium on the evaporator (HM) and high on the condenser, low on the evaporator (HL). The flowrates at these fan settings could not be obtained and a flowrate of 1.72 lb_m/min was assumed to carry on with the calculations. A flow rate of 1.175 lb_m/min was obtained for both high on the condenser and medium on the evaporator (HH) and medium on the condenser and high on the evaporator (MH).

When the power to the condenser fan was set on high while the power to the evaporator fan was gradually decreased, the COP_{max} ranged from 8.47 to 11.22 as the flowrate of the refrigerant was decreased. The COP_{max} is a function of the T_C and T_H. The temperature exiting the condenser, T_C, was observed to vary from 510.38°R to 516.38°R for similar fan settings above. This was not expected as the TXV system adjusts the flowrate and therefore should keep the outlet temperature of the condenser relatively constant. When the power to the evaporator fan was set on high while the power to the evaporator fan was gradually decreased, the COP_{max} decreased from 8.24 to 7.74 as the flowrate of the refrigerant was decreased. As the power to the condenser fan decreases, the rate of heat loss also decreases which implies that less heat is removed from the refrigerant and hence the value of T_H logically increases thereby decreasing COP_{max}..

For high power to the condenser fan while decreasing the evaporator's fan power settings COP_{actual} decreased from 1.963 to 1.767 as the flowrate of the refrigerant was decreased. This decrease in COP_{actual} could be due to a decrease in heat supply to the evaporator which results from a decrease of the power to the evaporator fan as the power to the condenser fan is kept constant. However, COP_{actual} between fan settings HH and HM is observed to be fairly constant with an average value of 1.939. This is because in the TXV mode the lower temperatures were adjusted to match the higher ones by increasing the residence time resulting in fairly constant enthalpy measurements. Similarly, decreasing the power to the condenser fan while keeping the power to the evaporator fan constant decreased COP_{actual} from 1.963 to 1.828. From the formula in *equation 2* (Macchi, 2015), COP_{actual} is equally inversely proportional to the power input to the compressor. It is hypothesized that a decrease in the mass flowrate of the refrigerant will lead to a decrease in the power requirement to compress the gas. Since the mass flowrate of the refrigerant obtained during this step was constant, some other experimental factor such as expansion of the valve to compensate for temperatures could have caused the increase in power.

The COP_{fluid} decreased from 8.98 to 7.48 when decreasing the power to the condenser fan while keeping the power to the evaporator fan constant. This is due to the increase in enthalpy, which results from lower outlet temperatures of the condenser as a result of a decrease of heat in the system.

The COP_{fluid} ranged from 8.98 to 8.44 and no specific trend was observed when decreasing the power to the evaporator while keeping the power to the condenser constant. It is difficult to explain the reasoning behind this as the system operated in two phase mode and values were assumed to carry on with the calculation.

Decreasing the power to the condenser fan while keeping the power to the evaporator fan constant increased compression efficiency (which remained fairly constant between 0.17 and 0.19). Because of similar flowrates (1.7504 lbm/min) of the refrigerant through the system at the fan settings, the power requirement in the system remained fairly constant ranging from 47.48 BTU/min to 46.917 BTU/min thereby resulting in similar efficiencies. In decreasing the power to the evaporator while keeping the power to the condenser constant, the compression efficiency increased from 0.179 to 0.189 and then decreased to 0.184 resulting in no specific trend. Experimental errors could have occurred due to faulty readings, inaccurate pressure gauges, fluctuations in flow rates, overheating of the compressor, and possible oil in the refrigerant.

The cycle efficiency measures the performance refrigeration system during the experiment and is the ratio of COP_{max} and COP_{actual} . Decreasing the power to the evaporator while keeping the power to the condenser constant decreased the compression cycle efficiency and the compression ratio from 0.238 to 0.182 and 2.86 to 2.42 respectively. Decreasing the power to the condenser while keeping the power to the condenser constant decreased the compression cycle efficiency but increased the compression ratio from 0.238 to 0.236 and 2.86 to 2.98 respectively. Therefore at the highest fan setting, the TXV mode operates better.

3.) In order to compare the performance of the system in the capillary and the TXV modes, a similar level of refrigerant had to be chosen and flowrates had to be the same for both runs. For the capillary mode (second run), the high- high fan setting with a refrigerant level of 11.5 and a flowrate of 1.6532 (lbm/min) was chosen, while for the TXV mode, the high-high fan setting with a similar refrigerant level and a flowrate of 1.750 (lbm/min) was chosen. This values were the closest in terms of refrigerant level and flowrate.

The $COP_{maximum}$, COP_{actual} , cycle effeciency values for the normal capillary mode were calculated to be 8.49, 1.945 and 0.229, while similar values for the TXV modes were 8.26, 1.963 and 0.238. The differences between both values depend on T_C and the design of the evaporator. The T_C for the normal capillary mode was 507^{o}R while that for the TXV mode was 510.38 ^{o}R resulting in a higher $COP_{maximum}$ hence for similar refrigerant level and flowrate. The COP_{fluid} for the normal capillary mode was 9.43 while the TXV mode was 8.98. The compression efficiency, and compression ratio for the normal capillary mode were 0.152, and 2.98 while similar values for the TXV modes were 0.179 and 2.86. Please refer to Table 5 for the tabulated results.

The TXV mode is seen as a better model of the capillary valve as it actively controls refrigerant flow and has a 2-3% higher energy ratio efficiency (Kim B, 2005). Therefore one will expect the the $COP_{maximum}$, COP_{actual} , COP_{fluid}, compression efficiency, compression ratio and cycle efficiency to be higher for the TXV mode. Though the TXV mode was found to be better from experimental results, some values such as $COP_{maximum}$, COP_{fluid}, and the compression ratio were higher for the capillary mode. Sources of error such as fluctuation in flowrates, inaccurate readings or assuming that the expansion was isenthalpic might have led to skewed results.

4.) The compressor in the system was not completely insulated and lost heat to the surroundings. A fan was used to cool the compressor to not overheat. The compressor cycle efficiency is comparing the heat loss by the compressor to the ideal work required to operate the system. An energy balance to the compressor (between point 2 and 5 as shown in Figure 1) was used to calculate the heat loss to the system, as shown in the equation below.

$$\dot{Q}_{heat\ loss} = \dot{W}_{compressor} - \dot{m}_f \Delta H_{pt5 \rightarrow pt2} \quad (3)$$

Reading the value of the wattmeter and subtracting the power required to operate the fans determined the work of the compressor.

Both capillary runs showed similar trend. The first capillary run, at 9.4cm of refrigerant, loss more heat to the surroundings as the flow rates decrease going from 42.7 to 35.0 Btu/min. The ratio of heat loss to total work by the compressor increased from 0.75 to 0.89, once again as the flow rate decreased. The second capillary run, at 11.5cm of refrigerant, had the same trend, which went from a heat loss of 35.4 to 40.2 Btu/min and a ratio of 0.77 to 0.92. Both runs data are shown in Table 6. This trend can be attributed to the vapour absorbing more heat in the evaporator due to a higher residency time. The enthalpy at point 5 would then be higher and the compressor would have to do less work on the gas to increasing the pressure and temperature. If related back to the equation 3, the enthalpy at point 5 would increase at lower flow rates, at point 2 after the compressor it would remain constant and the compressor was shown not to decrease power relative to the flow rates change. The excess work would then be converted to heat and furthermore lost to the surrounding.

When looking at both capillary runs, the change in amount of refrigerant flowing through the system at the same flow rate does not affect the ratio of heat lost to power consumed. This is due to the fact that the compressor always gives the same enthalpy at the exit (point 2). So if power consumed and exit enthalpy are the relatively constant. The only factor affecting heat loss is the enthalpy of the gas entering. With more fluid in the system, the enthalpy gained through the evaporator will be decrease and consequently the rate of heat loss decreases. The ratio of heat loss to work provided, on the other hand, remains the same because the work increases ever so slightly with refrigerant level.

The TXV mode showed two trends relating to heat loss in the compressor as the fan speed decreases. When the evaporator fan was lowered the heat loss went from 37. 1 to 38.4 Btu/min showing an increase in heat loss, while a condenser fan speed decrease from 37.1 to 35.4 Btu/min. The ratio stayed relatively constant when the evaporator fan decrease from 0.78 to 0.79 but the condenser fan went from 0.78 to 0.76 decreasing. The evaporator fan had a relatively low effect on the compressor because the enthalpy leaving was regulated by the TXV to be a saturated vapour when exiting the thermostatic element constricted the flow. Thus the enthalpy would be relatively constant when entering the compressor.

When comparing to the capillary mode, the vapour leaving the evaporator would not be regulated and could have been superheated or in two phase region. Thus the efficiency of the compressor during the TXV mode is higher as shown in Tables 3, 4 and 5.

5.) Cycle efficiency is defined as the actual coefficient of performance minus that of the ideal Carnot one. In this experiment, no cycle efficiencies were above 1. All values are listed on table 3, 4 and 5. A major drawback of this compressor is that it operated at a relatively constant power output. To be more efficient, it should vary the power output according to the need. Also since a fan was used to cool the system, the cold side can be made well insulated, therefore once the cols side reaches the desired temperature the compressor can be turned off. This would increase the efficiency. Another way to increase efficiency would be to change the refrigerant in the system. This replacement refrigerant would be one with a high latent heat of evaporation and consequently also have a lower specific heat and volume. Thus COP_{actual} and efficiency would increase significantly.

The cycle efficiency could be over one for multiple reasons. A large reason could be instrumental error. If an error in the thermocouple reading at point 1 and 4 would result in a false refrigerant capacity. If an error at point 1 is made than the COP_{actual} is incorrect. At point 4, both the ideal efficiency and the actual coefficient of performance would not be correct. The refrigerant capacity is then miscalculated and furthermore the coefficient of performance would be over that of an ideal Carnot cycle.

4. Conclusion and Recommendations

Conclusion

- The underlying assumption used in the calculation of the vapour quality at point 4 was that the system was isenthalpic, no heat was lost to the surrounding, and the kinetic and potential energy were negligible. The vapour quality for the first and second capillary run ranged from 0.130 to 0.134 and 0.162 to 0.153 respectively. The vapour quality for the TXV mode ranged from 0.163 to 0.185.
- Decreasing the power to the condenser fan while keeping the power to the evaporator fan constant had the following effects decreased $COP_{maximum}$, COP_{actual}, COP_{fluid} and compression cycle efficiency while the compression efficiency and the compression ratio increased. Decreasing the power to the evaporator fan while keeping the power to the condenser fan constant decreased COP_{actual}, compression cycle effeciency, compression ratio while no specific trend was observed for $COP_{maximum}$, COP_{fluid}, compression efficiency and compression ratio. It was also observed that at the highest fan setting, the TXV mode operates better.
- The TXV mode was found to be better from experimental results, though some values such as $COP_{maximum}$, COP_{fluid}, and the compression ratio were higher for the capillary mode. Sources of error such as fluctuation in flowrates, inaccurate readings or assumption that the expansion was isenthalpic might have led to skewed results.
- The heat loss from the compressor represents a large percent of the work supplied to the compressor. A decrease in flow rate will generate higher heat loss to the compressor. When comparing refrigerant level, the less refrigerant the more the heat loss but the ration of heat loss to work provided remains constant. The TXV mode is more efficient and has less heat loss due to the regulating by the thermostatic element.
- The cycle efficiencyfor the two capillary runs and the TXV run were both between 0 and 1. Although if were to be over 1, a major source would be a thermocouple at point 1 or 4 malfunctioning.

Recommendations

- The two phase system was observed during the second and fourth trials in the TXV mode and hence made it practically impossible to obtain a rotameter reading accurately. It is recommended that an electronic rotameter to measure flow rate should be installed to reduce errors due to recording data.
- It is also recommended that more trials be carried out during this experiment for each mode and more flow rates as it was difficult to judge the trend of our data set with only 2-4 points.
- Proper insulation of all the apparatus will also improve the accuracy of our results. As heat loss during the experiment could have resulted in faulty values
- Valve 2 should be repaired to allow for a continuous flowing system and reduce the error the coolant slowly flowing back into the reservoir.

5. Work Cited

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3.) Kim, B. H., and D.L. O’Neal, “Effect of Refrigerant Flow Control on the Heating Performance of a Variable-Speed Heat Pump Operating at Low Outdoor Temperature,” Journal of Solar Energy Engineering. 127 (2): 277-286 (2005).

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6. Appendices:

6.11 Appendix A: Tables and Figures

List of Figures:

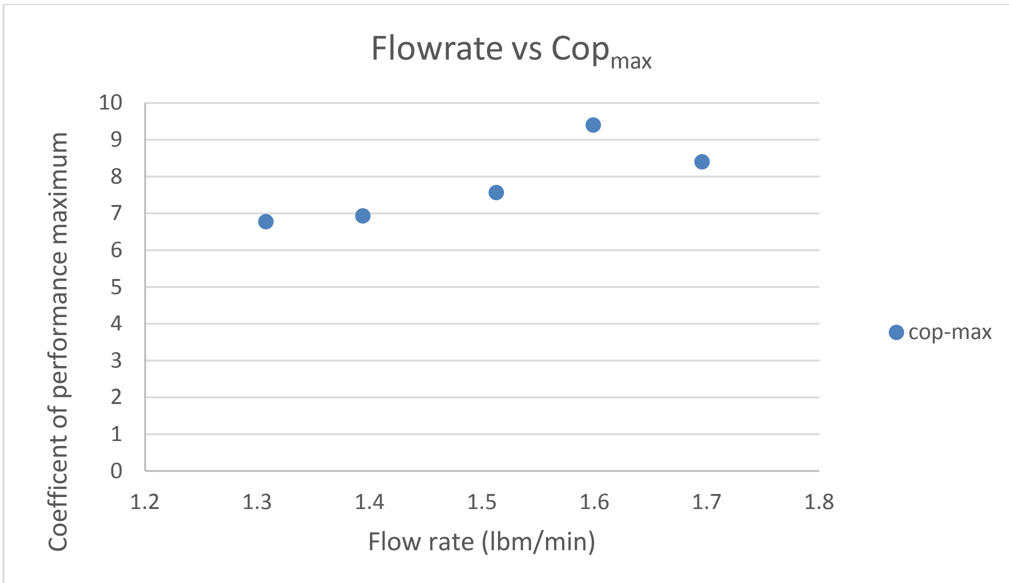

Figure 2 - Coefficient of performance fluid vs flow rate under Capillary mode (1st Run)

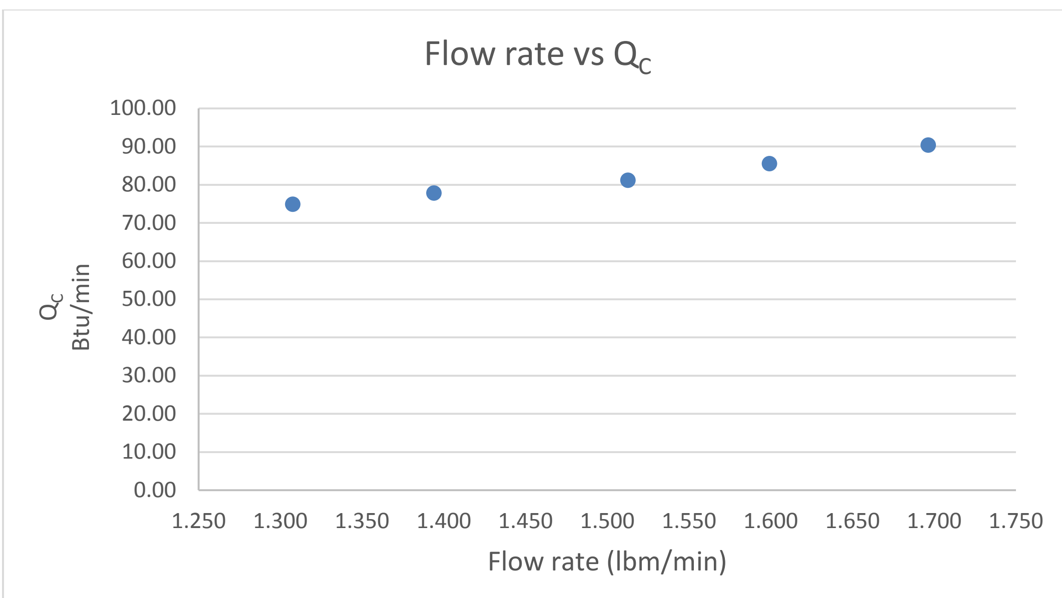

Figure 3- Qc vs flowrate for capillary mode (1st Run)

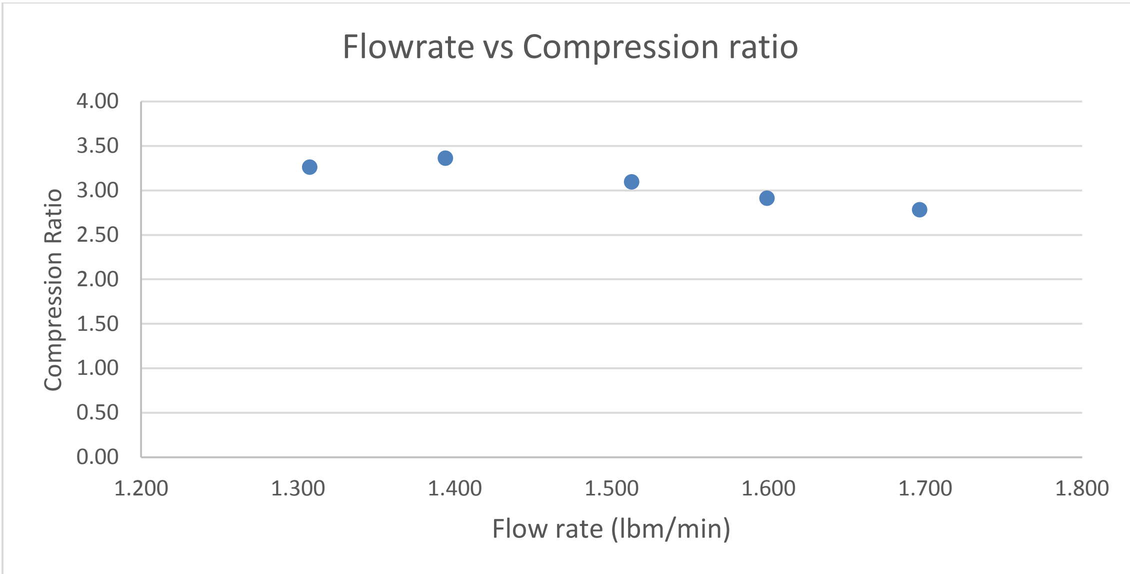

Figure 4-Flowrate vs Compression Ratio for capillary mode (1st run)

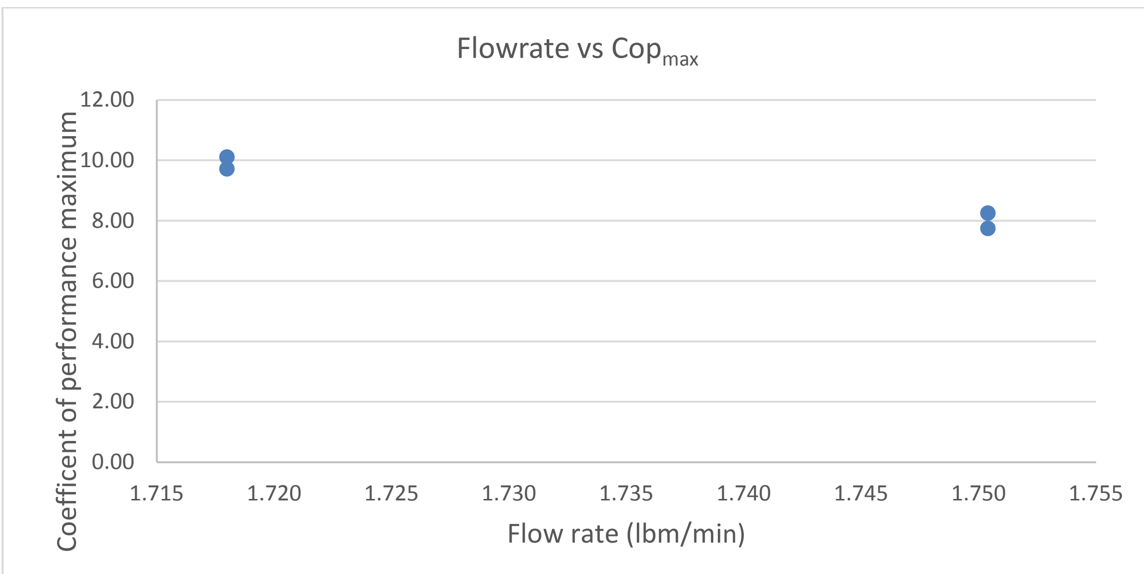

Figure 5 - Coefficient of performance fluid vs flow rate under TXV Mode

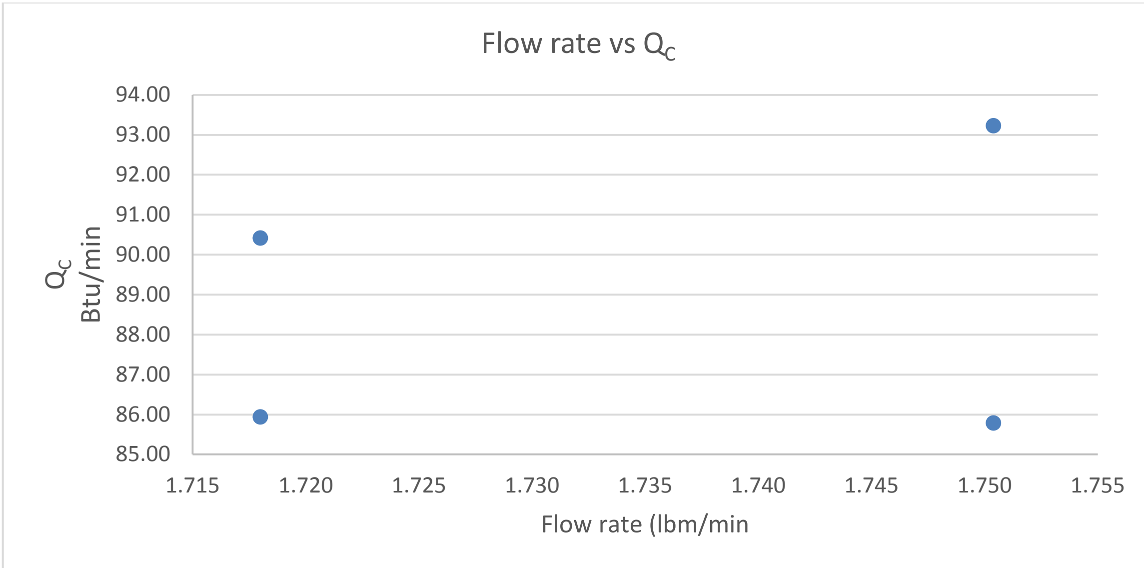

Figure 6-Qc vs flowrate for TXV mode

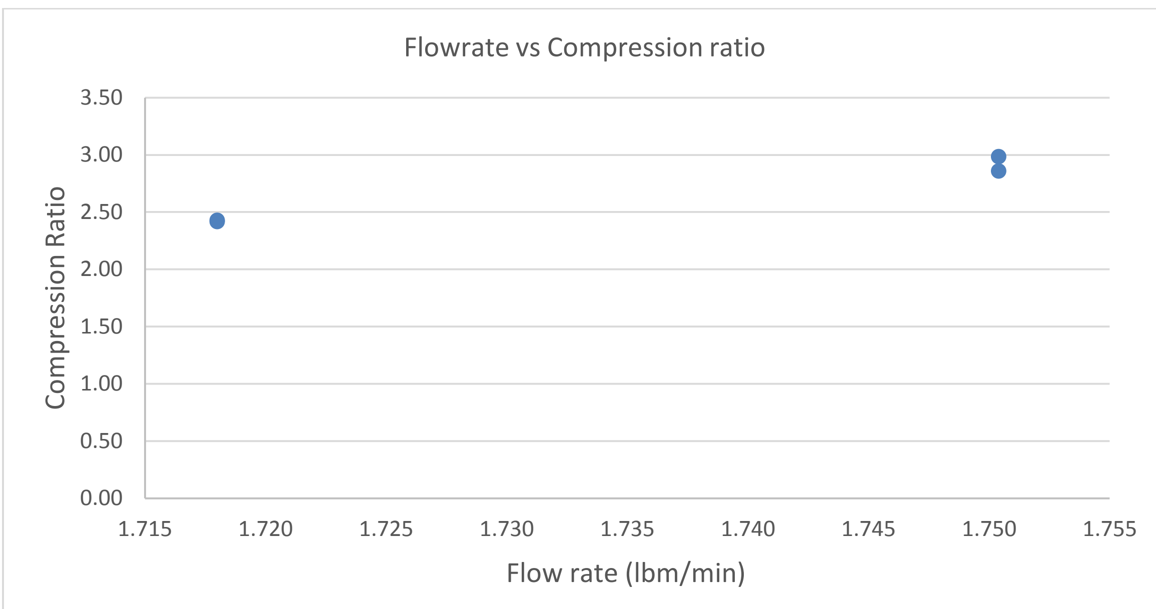

Figure 7-Flowrate vs Compression Ratio for TXV mode

6.12) List of Tables:

Table 1-Vapour quality at point 4 of refrigeration system in normal capillary mode.

Mass flow rate (lbm/min)	Vapour quality	
	Trial 1	Trial 2
1.718	0.130	0.162
1.610	0.129	0.132
1.502	0.129	0.157
1.394	0.131	0.151
1.2968	0.134	0.153

Table 2-Vapour quality at point 4 of refrigeration system in TXV mode.

Condenser fan setting	Evaporator Fan Setting	Vapour quality
High	High	0.163
High	Medium	0.134
Medium	High	0.231
High	Low	0.185

Table 3-Performance of refrigeration system in normal capillary mode (trial #1).

Mass flow rate (lbm/min)	1.718	1.61	1.502	1.394	1.2968
Refrigerant level	9.4	9.2	9.3	9.4	9.6
T_H(°R)	580.66	573.52	582.40	585.13	584.66
T_C(°R)	518.88	516.38	514.38	511.38	509.38
COP_{max}	8.40	9.04	7.56	6.93	6.77
$\dot{Q}_c$(Btu/min)	90.35	85.45	81.11	77.74	74.83
$W_{compressor}$(Btu/min)	46.92	47.20	48.34	48.34	47.77
COP_{actual}	1.926	1.810	1.678	1.608	1.566
COP_{fluid}	8.16	7.65	7.31	9.91	12.44
$\eta_{Compression}$	0.210	0.193	0.205	0.147	0.106
η_{Cycle}	0.229	0.200	0.222	0.232	0.231
CR	2.78	2.91	3.09	3.36	3.26

Table 4-Performance of refrigeration system in normal capillary mode (trial #2).

Mass flow rate (lbm/min)	1.653	1.61	1.502	1.394	1.297
Refrigerant level	11.5	11.8	11.7	11.7	11.8
T_H(°R)	567.27	572.55	575.03	566.63	565.13
T_C(°R)	507.38	516.38	513.38	517.88	518.88
COP_{max}	8.47	9.19	8.33	10.62	11.22
$\dot{Q}_c$(Btu/min)	89.58	85.94	78.71	77.63	74.19
$W_{compressor}$(Btu/min)	46.06	46.06	47.20	44.64	43.79
COP_{actual}	1.945	1.866	1.668	1.739	1.694
COP_{fluid}	9.43	8.57	8.24	13.23	16.42
$\eta_{Compression}$	0.152	0.192	0.181	0.107	0.075
η_{Cycle}	0.230	0.203	0.200	0.164	0.151
CR	2.98	2.63	2.87	2.94	2.83

Table 5 Performance of refrigeration system in TXV mode.

Fan Setting (Condenser, Evaporator)	(H,H)	(H,M)	(M,H)	(H,L)
Mass flow rate (lbm/min)	1.750	1.718*	1.750	1.718*
Refrigerant level	11.5	11.5	11.3	11.4
T_H(°R)	572.29	571.89	576.34	570.66
T_C(°R)	510.38	520.38	510.38	517.38
COP_{max}	8.24	10.10	7.74	9.71
$\dot{Q}_c$(Btu/min)	93.22	90.41	85.78	85.94
$W_{compressor}$(Btu/min)	47.49	47.20	46.92	48.62
COP_{actual}	1.963	1.915	1.828	1.767
COP_{fluid}	8.98	9.19	7.48	8.44
$\eta_{Compression}$	0.179	0.189	0.193	0.184
η_{Cycle}	0.238	0.190	0.236	0.182
CR	2.86	2.43	2.98	2.42

* Due to two phase flow in these settings a coolant flowrate for the energy balance of 1.718lbm/min is used.

Table 6 Heat loss from the compressor in the capillary mode at different levels of refrigerant and mass flow rates of coolant.

Mass Flowrate of Refrigerant (lbm/min)	Heat loss from Compressor (Btu/min)	Heat loss from Compressor (Btu/min)/ Total Compressor power, $\dot{W}$ (Btu/min)
First Run, Refrigerant Reservoir Level 9.4		
1.718	35.0	0.75
1.61	35.7	0.76
1.502	38.2	0.79
1.394	39.8	0.82
1.297	42.7	0.89
Second Run, Refrigerant Reservoir Level 11.5		
1.653	35.4	0.77
1.61	35.3	0.77
1.502	38.2	0.81
1.394	39.7	0.89
1.297	40.2	0.92

Table 7 Heat loss from the compressor during a system TXV mode, at different combinations of the fan speeds of condenser and evaporator.

Condenser Fan Speed	Evaporator Fan Speed	Heat loss from Compressor (Btu/min)	Heat loss from Compressor (Btu/min)/ Total Compressor power, $\dot{W}$ (Btu/min)
High	High	37.1	0.78
High	Medium	37.4*	0.79
Medium	High	35.4	0.76
High	Low	38.4*	0.79

* Due to two phase flow in these settings a coolant flowrate for the energy balance of 1.718lbm/min is used.

6.21) Appendix B: Sample Calculations

All sample calculations are taken from the capillary mode using a mass flow rate of 1.6532 lbm/min. Capillary (First Run)

1.0) Vapour Quality

$$H = (1 - x^v)H^l + x^v H^v$$
$$x^v = \frac{H - H^l}{H^v - H^l}$$
$$H^l(52.5\ °F) = 20.62\frac{Btu}{lb_m}$$
$$H^v(52.5\ °F) = 84.93\frac{Btu}{lb_m}$$
$$x^v = \frac{30.37\ \frac{Btu}{lb_m} - 21.63\ \frac{Btu}{lb_m}}{84.94\ \frac{Btu}{lb_m} - 21.63\ \frac{Btu}{lb_m}}$$
$$x^v = 0.138$$

2.0) COP_{max}

$$\mathrm{COP_{Max}} = \frac{\mathrm{T_C}}{\mathrm{T_H} - \mathrm{T_C}}$$

T_H = Temperature of the condenser

T_C = Temperature of the evaporator.

$$\mathrm{COP_{Max}} = \frac{518.88\ °\mathrm{R}}{580.66\ °\mathrm{R} - 518.88\ °\mathrm{R}}$$

$$\mathrm{COP_{Max}} = 8.39$$

3.0) Work supplied to the compressor

Because the evaporator and condenser fans are measured by the wattmeter, the work supplied to the compressor is adjusted.

$$\mathrm{W} = W_{total} - W_{fan}$$
$$\mathrm{W} = 935\ watts - 110\ watts$$
$$W = 825\ watts$$

4.0) Refrigeration Capacity, $\dot{Q}c$

Enthalpy balance across the refrigerator. It measures the amount of cooling a cycle could achieve

$$\text{Refrigeration Capacity} = \text{RC} = \dot{\text{Q}}_{\text{C}}$$

$$\dot{Q}_c = \dot{m}_f(h_1 - h_4)$$

$\dot{Q}_C$ = rate of energy absorbed in a cold reservoir per unit mass of refrigerant

h_1 = Enthalpy of fluid at point 1

h_4 = Enthalpy of fluid at point 4

$$\dot{Q}_c = 1.6964\ \frac{lbm}{min}(82.89 - 29.63)\frac{Btu}{lbm} = 90.35\ \frac{Btu}{min}$$

5.0) $\text{COP}_{\text{actual}}$

This represents the amount of heat absorbed compared to the amount of work which was put into the system

$$\text{COP}_{\text{Actual}} = \frac{\dot{\text{Q}}_{\text{C}}}{\text{W}}$$

$$\text{COP}_{\text{Actual}} = \frac{90.35\ \frac{Btu}{min}}{46.9176\ \frac{Btu}{min}}$$

$$\text{COP}_{\text{Actual}} = 1.93$$

6.0) $\text{COP}_{\text{fluid}}$

This measures of the potential for the fluid to transfer energy.

$$COP_{fluid} = \frac{h_1 - h_4}{h_2 - h_1}$$

Where: COP_{fluid} = the coefficient of performance of the fluid

h_1 = Enthalpy of fluid at point 1

h_2 = Enthalpy of fluid at point 2

h_4 = Enthalpy of fluid at point 4

$$\text{COP}_{\text{Fluid}} = \frac{82.89\frac{Btu}{lbm} - 29.93\frac{Btu}{lbm}}{89.41\ \frac{Btu}{lbm} - 82.89\ \frac{Btu}{lbm}}$$

$$\text{COP}_{\text{Fluid}} = 8.15$$

7.0) Calculation of the compression efficiency (η_{comp})

$$\eta_{comp} = \frac{\dot{m}_f(h_2^s - h_1)}{\mathrm{W}}$$

Where η_{comp} = the compression efficiency

$\dot{m}_f$ = the circulation rate of the refrigerant

h_2^s = Enthalpy of saturation of fluid at point 2

h_1 = Enthalpy of fluid at point 1

W = the power input to the compressor

$$\eta_{Comp} = \frac{1.6964\ \frac{lbm}{min}(88.68 - 82.89)\frac{Btu}{lbm}}{49.9176\ \frac{Btu}{min}}$$

$$\eta_{Comp} = 0.20951$$

8.0) Calculation of the cycle efficiency ($\boldsymbol{\eta_{cycle}}$)

$$\eta_{cycle} = \frac{COP_{actual}}{COP_{max}}$$

Where η_{cycle} = the cycle efficiency

COP_{max} = maximum coefficient of performance for an ideal cycle

COP_{actual} = Actual coefficient of performance

$$\eta_{cycle} = \frac{COP_{actual}}{COP_{max}} = \frac{1.93}{8.40} = 0.22$$

9.0) Calculation of the compression ratio

$$CR = \frac{P_2}{P_1}$$

Where: $CR = compression\ ratio$

$P_2 = pressure\ at\ point\ 2$

$P_1 = pressure\ at\ point\ 1$

$$CR = \frac{P_2}{P_1} = \frac{158.72\ psia}{61.22\ psia} = 2.593$$

10.0) Energy loss from compressor

$$\dot{Q}_{heat\ loss} = \dot{W}_{compressor} - \dot{m}_f \Delta H_{pt2 \rightarrow pt1}$$

$$\dot{Q}_{heat\ loss} = 49.9176\ \frac{Btu}{min} - 1.6964\ \frac{lbm}{min}(89.41 - 82.40)\frac{Btu}{lbm}$$

$$\dot{Q}_{heat\ loss} = 35.03\ \frac{Btu}{min}$$

7) Non-Essential- Raw Data

Table 8 Fan power consumption (Watts).

		Evaporator fan setting		
		High	Medium	Low
Condenser fan settings	High	110	100	95
	Medium	100	85	80
	Low	95	80	75

Table 9 Ambient and initial condition of the system.

Ambient Temperature(°F)	78.71				
Ambient Pressure (mmHg)	769				
Point	1	2	3	4	5
Initial Temperature (°F)	73	72.5	70	72	73
Initial Pressure (Psig)	55	52.5	55	54	61

Table 10 Normal capillary mode with a level of refrigerant of 10.1, raw data of temperature, pressure, enthalpy, and entropy. These values correspond to run 1.

	Location	1	2	3	4	5
First Flow rate	Measured Temperature (°F)	48	120	86	52.5	44
1.718 lbm/min	Measured Pressure (psig)	50	160	160	57.5	54
	Enthalpy (Btu/lbm)	82.9	89.41	30.37	30.37	81.80
	Entropy (Btu/lbm R)	0.166	0.163	0.060	0.061	0.166
Second Flow Rate	Measured Temperature (°F)	44	120	83	50	42
1.610 lbm/min	Measured Pressure (psig)	48	160	130	55	52
	Enthalpy (Btu/lbm)	82.44	89.43	29.00	29.00	82.22
	Entropy (Btu/lbm R)	0.166	0.163	0.059	0.060	0.166
Third Flow Rate	Measured Temperature (°F)	42	122	82	48	46
1.502 lbm/min	Measured Pressure (psig)	45	164	164	52.5	49
	Enthalpy (Btu/lbm)	82.2	89.6	28.6	28.6	82.9
	Entropy (Btu/lbm R)	0.166	0.163	0.059	0.060	0.168
Fourth Flow Rate	Measured Temperature (°F)	53	123	80	45	49
1.394 lbm/min	Measured Pressure (psig)	43	171	171	50	47
	Enthalpy (Btu/lbm)	83.9	89.5	28.1	28.1	83.4
	Entropy (Btu/lbm R)	0.170	0.163	0.058	0.059	0.170
Fifth Flow Rate	Measured Temperature (°F)	62	124	79	43	66
1.297 lbm/min	Measured Pressure (psig)	42.5	168	168	46	45
	Enthalpy (Btu/lbm)	85.1	89.7	27.9	27.9	85.8
	Entropy (Btu/lbm R)	0.172	0.163	0.057	0.058	0.174

Table 11 TXV mode raw data for temperature, pressure, enthalpy, and entropy. This run has a level of refrigerant of 11.43.

	Location	1	2	3	4	5
Condenser Speed -high	Measured Temperature (°F)	48	114	87	44	44
Evaporator Speed - high	Measured Pressure (psig)	50	150	150	56	55
	Enthalpy (Btu/lbm)	83.2	89.1	29.9	29.9	82.7
	Entropy (Btu/lbm R)	0.169	0.164	0.061	0.062	0.168
Condenser Speed -high	Measured Temperature (°F)	48	110	88	54	44
Evaporator Speed - medium	Measured Pressure (psig)	50	140	139	58	55
	Enthalpy (Btu/lbm)	82.82	88.54	30.19	30.19	19.43
	Entropy (Btu/lbm R)	0.165	0.163	0.061	0.062	0.042
Condenser Speed -medium	Measured Temperature (°F)	48	120	104	44	45
Evaporator Speed - high	Measured Pressure (psig)	52	160	160	57.5	55
	Enthalpy (Btu/lbm)	83.2	89.7	34.2	34.2	82.9
	Entropy (Btu/lbm R)	0.168	0.165	0.068	0.071	0.169
Condenser Speed -high	Measured Temperature (°F)	47	110	98	51	46
Evaporator Speed - low	Measured Pressure (psig)	50	138	138	55	55
	Enthalpy (Btu/lbm)	82.7	88.6	32.7	32.7	82.6
	Entropy (Btu/lbm R)	0.165	0.164	0.066	0.067	0.165

Table 12 Normal capillary mode with a level of refrigerant of 11.7, raw data of temperature, pressure, enthalpy, and entropy. These values correspond to run 2.

	Location	1	2	3	4	5
First Flow rate	Measured Temperature (°F)	48	112	84	41	42
1.653 lbm/min	Measured Pressure (psig)	47.5	143	143	56	51
	Enthalpy (Btu/lbm)	83.4	89.2	29.2	29.2	82.7
	Entropy (Btu/lbm R)	0.171	0.165	0.060	0.061	0.170
Second Flow Rate	Measured Temperature (°F)	45	112	84	50	41
1.61 lbm/min	Measured Pressure (psig)	48	143	143	55	51
	Enthalpy (Btu/lbm)	82.58	88.81	29.20	29.20	82.12
	Entropy (Btu/lbm R)	0.166	0.164	0.060	0.061	0.166
Third Flow Rate	Measured Temperature (°F)	44	114	88	47	46
1.502 lbm/min	Measured Pressure (psig)	45	149	149	52.5	48
	Enthalpy (Btu/lbm)	82.6	88.9	30.2	30.2	83.0
	Entropy (Btu/lbm R)	0.167	0.163	0.061	0.063	0.169
Fourth and Third flow were obtained but not included in this table						

Refrigeration

Chemical Engineering Practice

CHG 3122

Ghareb Jassim Al-Mahmoud
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To: Dr. Kruczek
From: Ghareb Al-Mahmoud, Group 5
Date: March 8th, 2006
Subject: CHG 3122, Refrigeration Lab

Refrigeration is the process of removing heat from a place where it is not wanted, and transferring that heat to a place where it makes little or no difference.

The main objective of this experiment is to interpret the refrigeration capacity and performance coefficients for a refrigeration unit as a function of circulation rate of refrigerant and heat load. Another objective is comparing the performance of the refrigeration unit in two modes; The normal capillary mode and the thermostatic expansion valve (TVX) mode.

The experiment was set-up as described in Scott Air Conditioning and Refrigeration Education system with Freon-12 as a working fluid. The Temperatures' and pressures' readings were taken at strategic points that are important to do all the needed calculations to figure out the refrigeration unit performance.

The main problem faced the group was that the condenser gets hot really fast. Every time the condenser gets hot it shuts down. This problem forced the group to rush in taking the points to assure completing the run with the condenser being able to operate without shutting off. Obviously this could cause inaccuracy in the data especially if they were close to the saturation line which could lead to a big difference in the thermodynamics properties "between liquid and vapour".

The only obvious finding of this lab was that increasing the refrigerant over a critical amount will decrease the overall heat transferred. It was difficult to have more experimental findings or observations due to the bad results obtained as will be discuss furthermore in the report.

Equipment and procedure

The Experiment was performed using Scott Air Conditioning Education System with Feron-12 (R-12) as working fluid, see figure 1 in Appendix B. The system in this experiment is going to be run at two different modes: normal capillary and thermostatic expansion. The system consists of: compressor (A), condenser (B), evaporator (C), capillary (D), thermostatic expansion valve (E), temperature sensor (F), valves (V), liquid indicator (G), calibrated rotameter (H), drier (I), liquid refrigerant receiver tank (J), oil and refrigerant tank (K) and oil storage tank (L). Also there are temperature and pressure gauges located in all “strategic locations” and two variable speed fans one behind the condenser and the other one behind the evaporator. “Strategic locations” in this sense are defined as: point 1- Evaporator outlet, point 2- Condenser inlet, point 3- Condenser outlet, point 4- Evaporator inlet, and point 5- Compressor inlet.

The circulation rate of refrigerant in the system operating in normal capillary mode is controlled by valve 4. To switch to the TXV mode, valve V4 is closed and valve V5 should be opened.

Before starting the experiment, while the compressor is off, the system being in thermal equilibrium with surrounding was checked and readings of temperatures and pressures were recorded. Also recordings of power consumption for fans in nine different sittings were taken.

The experiment then started by setting the system to normal capillary mode (isenthalpic) with fans setting of high condenser and high evaporator, and so opening valve V4. Then compressor was turned on and circulation rate of fluid was set at a level corresponds to maximum reading of rotameter. Five measurements of temperatures and pressures recorded at five different circulation rates of the refrigerant.

After finishing the first capillary mode run, another one took place with adding more refrigerant to the system. The exact same procedure as described in the previous paragraph was carried on again in the second run.

The system was then switched to the TXV mode and the temperatures and pressures at the five strategic points described earlier were recorded at four different fans' sits. High-high, medium-medium, medium-high, and low-high.

Summary of Results

- In the first run of the Capillary mode:
  - Tc = 50 F.
  - Th = 85 F.
  - COP_{max} = 1.428.

  "Note: looking at figure 4 in appendix B; it can be said that it is difficult to have Tc and Th for each cycle. So, Tc & Th were averaged from figure 4 and the same thing have been done for the other two runs".
- In the second run of the Capillary mode:
  - Tc = 48.2 F.
  - Th = 85 F.
  - COP_{max} = 1.3099.
- In the run of the TXV mode:
  - Tc = 52.2 F.
  - Th = 97.7 F.
  - COP_{max} =1.147.
- In the two runs of the Capillary mode, no experimental obvious relation could be drawn down here as a result.
- The TXV mode run plots weren't any better than those of the Capillary mode. "more about those results is in the Discussion"

Discussion

In general, the experimental results were not good at all. The obtained results were not as expected as will be explain in the next paragraphs.

The different coefficients of performance calculated neglecting the negative ones “Impossible” experienced an expected scenario were the fluid COP was always higher than the actual one since it neglects heat losses in the system while the actual one doesn’t.

Although it was difficult to observe the effect of the refrigerant circulation rate; it should have a major effect on the refrigeration process and its performance. As it is known from Bernoulli’s equation, increasing the velocity of a fluid in a section leads to decreasing its pressure, if all other parameters been unchanged. That would lead to a higher heat absorbed from the cold reservoir and less heat rejected from the hot reservoir because the lower pressure will cause the refrigerant to evaporate quicker and will the vapour will be superheated. Superheated vapour doesn’t corresponds to pressure/temperature relationship; And since there will be no liquid remain to boil off to vapour, no more vapour pressure can be generated when heat is added. The vapour will take on sensible heat when heated and the temperature will rise. This eventually will increase the coefficient of performance of the refrigeration unit and all other performance parameters.

R-12 was the first halo-carbon refrigerant to go into general use. Its immiscibility with oil is one of its best characteristics. Nonetheless-having it in excess will tend to reduce the heat transfer rate and as a result will lead to a poor refrigeration unit performance. In the second run of Capillary mode when extra refrigerant was used, the maximum coefficient of performance was lowered than the first one which was expected.

One can ask: what could be the ambient temperature affect on the refrigeration unit? To answer this question a simple schematic was drown in figure 28 and 29 in appendix B to illustrate the change in the T-S and P-H diagrams if the ambient air temperature was to be raised by few degrees, it is explained in the next three paragraphs.

Assuming the change in the ambient air temperature will have no effect on the compressor performance. This illustration will focus on the effects that change will have on the other two major components in the refrigeration system, the **Evaporator** and the **Condenser.**

The evaporator function is very simple. It absorbs heat into the system from the surroundings. This could lead to the simple conclusion that increasing the surrounding temperature will drive this process forward and will result a super saturated vapour with higher sensible energy “h1”. Higher enthalpy means higher Entropy as well.

The condenser rejects heat into the surroundings. So, as concluded in the previous paragraph. Increasing the surroundings temperature will decrease the rate of this rejection and eventually increase “h3” and S3.

Conclusions and Recommendations

- Increasing the refrigerant circulation rate leads to a better refrigeration unit performance.
- COP_{fluid} is higher than the actual one.
- Increasing the amount of refrigerant over a critical value will decrease the refrigeration unit performance quality.
- The ambient air conditions have different affects on the Evaporator and Condenser in the refrigeration unit.
- Problems and Recommendations:
    - The main source of error in this experiment results was the compressor incapability to bare the job. It turned off a lot of times which made the group to rush in taking the readings which may cause some calculation error especially if the reading was at the saturation point or clause to it which will cause a huge deference in the thermodynamic properties. One recommendation for that problem could be placing the compressor in an isolated place from the whole system for being able to keep the fan running all the time. Or the fan will effect the readings otherwise.
    - The piping wasn’t insulated which caused losing the heat in the piping. This is obvious in the T-S diagrams. In all of them it can be seen that point 4 has lost some energy which placed it really close or almost at the saturation curve. So another recommendation is using a thermal insulator on the pipes.

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Appendix A

Tabulated Results

Appendix B

Figures

Appendix C

Sample Calculations

Appendix C

Raw Data

CHG 3122
Chemical Engineering Practice

Refrigeration

Yusuf Fadairo

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Shidan Cummings

5204398

To: Dr. Arturo Macchi

From: Yusuf Fadairo and Shidan Cummings of Group 2

Date: July 16, 2011

Subject: CHG 3122, Refrigeration

Refrigeration is one of the many important concepts of heat exchange that can be found in all modern homes. Refrigeration or the temperature reduction occurs by the aid of a refrigeration cycle. Generally, the refrigeration cycle uses a gas that evaporates in at low temperature, usually ammonia gas with a boiling point of negative 32°C, which is heated up or pressurized by a compressor. The heated gas is then sent through a coil which reduces the temperature of the gas. The resulting gas is then expanded to high pressure by an expansion valve. Expanding the gas will the cause the gas to boil and vaporizes to about negative 32°C, this low temperature can then be used in a fridge. The cold gas is pulled by the compressor and the refrigeration cycle stats again (Figure 1, Appendix A).

In this experiment, the refrigeration cycle using Freon-12 gas as the working fluid is studied at various points in the cycle to determine the optimum operation of the refrigerator. The main objective of the experiment is to study cycle and determine the refrigeration capacity and performance coefficients as a function of circulation rate of refrigerant and heat load. Another objective of the experiment is to observe the refrigeration in its normal capillary mode and its thermostatic expansion valve mode and compare the performance of both modes.

Other side objective of the experiment pertains to the correction of the temperatures and pressures measured in the experiment, and discussing the circulation rate and the amount of the refrigerant in the normal capillary mode. The last side objective of the experiment is to discuss the observation from one of the runs using the Thermostatic expansion valve mode.

Equipment and Procedure

This refrigeration experiment utilized the Scott Air Conditioning and Refrigeration Education system with Freon -12 as the refrigerant. The refrigeration system can be run in normal capillary mode, thermostatic expansion mode, and reverse mode, but only the first two modes were used in this experiment. The components and schematic of the system is given in Figure 3 of the Laboratory Handout. For the normal capillary expansion mode, the expansion step of the refrigeration cycle takes place in a capillary feed tube. The circulation flow rate of refrigerant in the cycle can be controlled using the V4 valve in the system. For the other mode, the thermostatic expansion mode (TVX), the expansion step takes place in the thermostatic expansion valve. The normal capillary mode is changed by turning off the V4 valve, and the thermostatic expansion mode is activated by opening the V5 valve.

The provided flow chart for the system is first studied for better understanding of the unit and the electrical switches in the unit is set up for the normal capillary mode according to the provided settings table. The system is then left to reach thermal equilibrium with the environment with ambient temperature of 76.5°F and ambient pressure of 757.1 mmHg. The compressor is then shut off and the wattage of the fans at the two location of the system in recorded at all positions. In the capillary mode operation, both fans are then set on high the compressor is turned on. The level of refrigerant is set to a given maximum rotameter reading. The system is left to stabilize and three readings are taken of all the gauges showing the pressure and temperatures at all points in the cycle. The circulation rate of refrigerant is reduced and another reading it taken as the system reaches steady state. This process is performed at four different circulation rates of the refrigerant. The system is changed to the TVX mode, with the fans still on high, and the system is left to reach steady state before more temperature and pressure readings as well as rotameter and power consumption readings are taken. The process is repeated again but for three different fan speeds of High-Medium, Medium-High, and High-Low. The system is then switched back to the normal capillary mode with the fans both on High speed and more reading are done as the system reached steady state for the same refrigerant circulation rates that were performed before the TVX mode.

Summary of Results

Discussion

Conclusion and Recommendation

1. This is

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Appendix A

Tables and Figures

Tables

Table 1: Temperature and Pressure readings for the first normal capillary mode run

	Readings	Point 1	Point 2	Point 3	Point 4	Point 5
	Initial Temperature (°F)	76	76	69	74	75
	Initial Pressure (Psig)	75	72.5	74	74	82
First Flow Rate	Temperature Reading (°F)	49	120	94	54	45
	Pressure Reading (psig)	51.5	157.5	157	59	54
	Rotameter reading	15.1	Watts Reading	980	Refrigerant Level	8.5
Second Flow Rate	Temperature Reading (°F)	47	120.5	92	51	52.5
	Pressure Reading (psig)	49	160	158	57.5	45.8
	Rotameter reading	14.3	Watts Reading	980	Refrigerant Level	8.5
Third Flow Rate	Temperature Reading (°F)	48	127	88.1	51	46
	Pressure Reading (psig)	47	175	175	55	53
	Rotameter reading	13.8	Watts Reading	985	Refrigerant Level	8.5
Fourth Flow Rate	Temperature Reading (°F)	58	129	86.3	48	60
	Pressure Reading (psig)	46	174	171	52.5	50
	Rotameter reading	12.8	Watts Reading	967	Refrigerant Level	8.5
Fifth Flow Rate	Temperature Reading (°F)	60	138	84.2	46	61
	Pressure Reading (psig)	45	182.5	184	51	48
	Rotameter reading	12	Watts Reading	970	Refrigerant Level	8.5

Table 2: Temperature and Pressure readings for the TVX mode

	Readings	Point 1	Point 2	Point 3	Point 4	Point 5
	Initial Temperature (°F)	76	76	69	74	75
	Initial Pressure (Psig)	75	72.5	74	74	82
S5:High S6:High	Temperature Reading (°F)	47	132	90	51	45
	Pressure Reading (psig)	40	173	173	54	52.5
	Rotameter reading	13.8	Watts Reading	985	Refrigerant Level	11.2
S5:High S6:Medium	Temperature Reading (°F)	44	133	88	48	40
	Pressure Reading (psig)	44	167	166	50	50
	Rotameter reading	12.8	Watts Reading	937	Refrigerant Level	11.3
S5:Medium S6:High	Temperature Reading (°F)	47	136	106	45	45.8
	Pressure Reading (psig)	50	200	205	57.5	53
	Rotameter reading	12.5	Watts Reading	1022	Refrigerant Level	11.9
S5:High S6:Low	Temperature Reading (°F)	42.5	142	87	46	40
	Pressure Reading (psig)	45	168	156	50	49
	Rotameter reading	12.4	Watts Reading	920	Refrigerant Level	12

Table 3: Temperature and Pressure readings for the second normal capillary mode run

	Readings	Point 1	Point 2	Point 3	Point 4	Point 5
	Initial Temperature (°F)	76	76	69	74	75
	Initial Pressure (Psig)	75	72.5	74	74	82
First Flow Rate	Temperature Reading (°F)	49	140	103	43	45
	Pressure Reading (psig)	50	155	155	59	54
	Rotameter reading	14.3	Watts Reading	945	Refrigerant Level	13.9
Second Flow Rate	Temperature Reading (°F)	47	141	101	42	44
	Pressure Reading (psig)	50	155	154	56	53
	Rotameter reading	14.1	Watts Reading	945	Refrigerant Level	13.9
Third Flow Rate	Temperature Reading (°F)	55	146	87	49.5	56
	Pressure Reading (psig)	48	157.5	157	54	52
	Rotameter reading	13.5	Watts Reading	940	Refrigerant Level	13.8
Fourth Flow Rate	Temperature Reading (°F)	62.5	148	92	43	60
	Pressure Reading (psig)	44	156	157	47.5	45
	Rotameter reading	11.8	Watts Reading	905	Refrigerant Level	13.8
Fifth Flow Rate	Temperature Reading (°F)	65.5	150	91	42	64
	Pressure Reading (psig)	40	159	160	45	45
	Rotameter reading	10.4	Watts Reading	885	Refrigerant Level	13.8

Table 4: Temperature and Pressure readings for the first normal capillary run

Figures

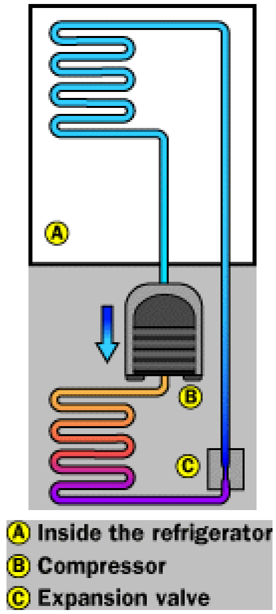

Figure 1: The Refrigeration cycle

:

Figure 2:

Figure 2:

Appendix B

Equations and Sample Calculations

Equations

1. Here

Sample Calculations

- Here

To: Dr. Zhang

From: Salim Fettaka, Matthew Galarneau, Group 4

Date: March 9, 2009

Subject: CHG3122, Refrigeration

A refrigeration system works by taking heat from a cold reservoir and rejecting that heat to the hot reservoir. The evaporator absorbs the heat from the environment and the condenser rejects the heat to the environment. The coefficient of performance in a Carnot refrigerator does not depend on the working fluid. This is not true for actual refrigeration systems as each fluid has different irreversibilities that are inherent in the vapour-compression cycle. Since the fluid has some effect on the coefficient of performance, it is necessary to prevent air from getting into the system. This is done by choosing a fluid whose vapour pressure at the evaporator temperature is greater than atmospheric. The fluid must also be picked so that the temperature in the condenser is not too high, as this will increase operating costs. One of the most important choices for the fluid are the safety considerations. Fluid toxicity, flammability, cost and corrosion properties are very important properties when considering a refrigerant.

This experiment consisted of the Scott Air Conditioning and Refrigeration Education system with Freon R-12 as the working fluid. This fluid meets all the above recommendations for fluid selection but is no longer used as refrigerant as it is known to damage the ozone layer. The experiment was performed with the system operating in two different modes. The capillary mode, (constant enthalpy expansion) was run with varying refrigerant flow rates. It was repeated for a second run with varying refrigerant levels. The thermostatic (TXV) mode (constant entropy expansion) was run with varying condenser and evaporator fan speeds. Since the compressor could trip if the condenser and evaporator fan speeds were set to low, the low setting was never used. A separate fan was used to cool down the overheating compressor to prevent it from tripping. Although the compressor never tripped, temperature and pressure readings were taken rapidly just in case.

Some key findings from this experiment include: The coefficient of performance is proportional to the fan speed on the evaporator and condenser when the experiment was operated in TXV mode. The coefficient of performance of the system, when operated in capillary mode, was proportional to the flow rate of the refrigerant in the system. It was also observed that the level of refrigerant had an inverse effect on the coefficient of performance. As the refrigerant level increased, the coefficient of performance decreased almost linearly in all 3 experimental trials.

Equipment and Procedure:

In this experiment we use the Scott Air Conditioning Education system with dichloro-difluoro-methane (CCl_2F_2) (Freon-12) as the working fluid. A schematic diagram of the apparatus is shown in figure 1 of Appendix A.

As show on figure 1 of Appendix A, the system is composed of : a compressor, a condenser, an evaporator, a capillary, a thermostatic expansion valve, a temperature sensor, valves, a moisture and liquid indicator, a calibrated rotameter, a drier, a liquid refrigerant receiver tank, an oil and refrigerant accumulator tank and an oil storage tank. Various pressure and temperature gauges are used and are located at five places in the system: the evaporator outlet, the condenser inlet, the condenser outlet, the evaporator inlet, and the compressor inlet.

Before we begin the experiment, we record the temperature and pressure of the gauges to make sure that the system is in thermal equilibrium with the surroundings (temperature is the same in all gauges), and determine the wattage requirements of the fans in all combinations.

We run the system under two operating modes: normal capillary and thermostatic expansion. First we begin with the normal capillary mode, setting both fans to high speed; we take readings of the rotameter, refrigerant level, wattage requirements, pressure and temperature at 5 different flow rates. We have to allow wait some time during the start up to allow the system to reach steady state and to avoid air bubbles inside the rotameter. Then we switch to the thermostatic expansion TXV mode, we take the same readings for four fan settings: high-high, high-medium, medium-high, medium-medium. Finally, we switch the apparatus back to the capillary mode with both fans set to high, but this time we decrease the amount of refrigerant in the system.

We should note that the Freon-12 refrigerant used in the apparatus does contain some oil to lubricate the compressor, and thus its properties are different from those of pure CCl_2F_2, and the necessary corrections have been made into the Excel spreadsheet.

Summary of Results:

1. Tables 1, 2, and 3 of Appendix B list the hot T_H and cold T_C reservoirs temperatures of the Scott Air Conditioning and Refrigeration system for the first capillary. We observe that the average $T_H = 568.74°R$ and $T_C = 504.07°R$ and table 3 for the second run $T_H = 568.70°R$ and $T_C = 502.07°R$. Table 3 lists the values for the thermostatic expansion TXV mode, the average $T_H = 575.33°R$ and $T_C = 512.47°R$.
2. The refrigeration capacity RC values for the first capillary run, the thermostatic expansion TXV mode, and the second capillary run are listed on tables 1,2, and 3 respectively. The average refrigeration capacity for the first run under capillary mode is $RC = 73.30\,{}^{Btu}\!/\!_{\min}$ and for the second run is $RC = 68.93\,{}^{Btu}\!/\!_{\min}$. We note that as the flow rate of the refrigerant increases the value of the refrigeration capacity also increases. As for the thermostatic expansion TXV mode the average $RC = 84.83\,{}^{Btu}\!/\!_{\min}$, and the maximum is reached when the condenser fan S5 is at a medium velocity and the evaporator fan S6 is at medium velocity. The minimum refrigeration capacity is when both the condenser S5 and evaporator S6 fans are set to medium velocity.
3. Figures 1,3, and 4 of Appendix C plot the variation of the refrigeration capacity as a function of the refrigerant flow rate for the first capillary run, the TXV mode, and the second capillary run respectively. We observe that for both capillary runs, RC increases linearly with the mass flow rate of the refrigerant. This is expected since the refrigeration capacity increases with the quantity of refrigerant available to remove the heat from the system. However, in the case of the thermostatic expansion TXV mode not the same linear patter is observed.

4. Tables 1, 2, and 3 list the power to the compressor W for the first capillary run, the thermostatic expansion mode, and the second capillary run. We observe that the power input of the compressor increases as the flow rate increases in the capillary mode. In the case of the TXV mode, the maximum happens when both the condenser S5 and evaporator S6 fans are set to medium velocity. The minimum value is when the condenser S5 and evaporator S6 are set to high.
5. Given that the actual coefficient of performance is ratio of the refrigeration capacity RC on the power to the compressor W, we expect that COP_{actual} follows the same trend as RC and W for the capillary mode. As shown on figures 4, and 5 of Appendix C the COP_{actual} decreases as the flow rate of the refrigerant increases. The actual coefficient of performance values ranged from 1.795 to 1.556. In the case of the thermostatic expansion mode, the maximum is reached when the condenser fan S5 is high, and the evaporator fan S6 is medium, and the minimum happens when the condenser fan S5 is set to medium, and the evaporator fan S6 is high.
6. The fluid coefficient of performance COP_{fluid}, for the first capillary run, the TXV mode, and the second capillary run are listed in tables 1,2, and 3 respectively. We note that COP_{fluid} depends on the enthalpy of the working fluid at 4 points in the system. For the first capillary run, the average is 8.61, and 6.51 for the second capillary run. As for the thermostatic expansion mode, the average is 5.74.
7. The maximum coefficient of performance $COP_{\max}$ is the ratio of the energy absorbed by the cold reservoir over the difference between the energy absorbed by the hot reservoir and cold reservoir. We note that for the first capillary run, the average $COP_{\max}$ is 7.80, and 7.54

for the second capillary run. In the case of the thermostatic expansion mode, the average is 8.16.

8. Figures xx plot the T-S, T-H and P-H diagrams for the first capillary run. These diagrams are in accordance with the ideal isenthalpic vapour-compression refrigeration cycle diagram.
9. T-S, T-H and P-H diagrams for the first run of the TXV mode is shown on figure xx. These diagrams exhibit the properties of an isentropic vapour-compression refrigeration cycle.
10. Tables 1, and 3 of Appendix B, list the compression efficiency and overall compression efficiency for the first and second capillary runs respectively. The average η_{comp} for the first capillary run is 0.10 as opposed to 0.07 for the second run. The compression efficiency η_{comp} is lower in the second capillary run than in the first, which is expected since the system contains less refrigerant. However the cycle efficiency η_{cycle} for both runs is the same. Table 2 shows the value of the values of η_{comp}, and η_{cycle} for the TXV mode. The average η_{comp} is 0.22 and η_{cycle} is 0.23.
11. The compression ratio CR for both capillary runs is shown on figures xx of Appendix C. There is linear relationship between CR and the refrigerant flow rate, CR decrease as the refrigerant flow rate increases. For the TXV mode, the compression ratio is at its minimum when the condenser S5 and evaporator fan S6 are set to high, and the maximum happens when the condenser fan S5 is set to medium, and the evaporator fan S6 is high.

Discussions

The capillary runs demonstrated that when the fluid flow rate is increased, the refrigeration capacity of the system will also increase. This is analogous to a heat exchanger where the flow rate of the fluid allows for more heat transfer to the fluid. With an increase in the Reynolds number (due to the increase in fluid flow), an increase in the Nusselt number will occur. Better heat transfer is achieved as the Nusselt number determines the convective heat transfer coefficient to the fluid.

The actual coefficient performance is plotted as a function of the level of refrigerant for both capillary runs on figures 4, 5 of Appendix C. We observe for the first capillary run that COP_{actual} is higher when the level of the refrigerant is low and then it starts decreasing as the level of the refrigerant increases. The same trend is observed in the second capillary run. Figure 6 shows the COP_{actual} for the TXV mode, in general the coefficient of performance is higher than for the capillary mode.

The T-H, T-S diagrams plot the refrigeration cycle's profile and are in accordance with theoretical expectations. In the capillary mode the expansion is isenthalpic, whereas the thermostatic expansion mode is isentropic. According to our experimental data show on figures xx, the system is not perfectly ideal and there are deviations from both isentropic and isenthalpic profiles. Although deviations do exist, they are minor and the isenthalpic and isentropic assumptions still hold. This is due to the fact that ideal systems are hard to achieve and sometimes impossible experimentally.

The drop in refrigeration capacity due to the decrease of the condenser fan setting was much larger than when the evaporator fan setting was decreased. The increase in refrigeration capacity with the increase of condenser fan speed follows the convective heat transfers principles. As the fan speed is increased, the condenser is better able to reject heat to the environment. This allows for better condensation of the two phase fluid that leaves the compressor. When the condenser is able to reject more heat, the compressor does not need to compress the fluid as much as more of the two phase fluid will condense in the condenser. The TXV mode also demonstrated a higher compressor power requirement when the fan speeds were decreased (Appendix C – Figure 11). The observed trend shows that the work needed in the compressor is reduced when more heat is rejected from the condenser to the environment. Since the compressor is run using an electric motor, the energy requirements of the system can be reduced with a better means of heat transfer at the condenser.

The ability for the condenser to reject heat to the environment played a major role in the effectiveness of the system. The work requirements for the compressor are at a minimum when the condenser fan is at maximum .The current refrigeration system capacity is limited since air has a poor convective heat transfer coefficient compared to other fluids. The use of water as a medium for the condenser to reject heat to would have increased the performance of the system drastically. This is one of the many reasons why geothermal heat pumps are becoming more popular as they can provide the same refrigeration capacity but require less work input to the compressor. The water used in a typical geothermal system is around 15^{O}C while on a hot summer day, the ambient air temperatures can exceed 30^{O}C. It can be difficult for an air conditioning unit to reject heat to the environment when the environment is already hot.

Although our experimental results are overall in accordance with theoretical data, a certain amount of variation exists due to in part to the fluctuation in the level of refrigerant during the experiment. Other sources of error include taking temperature and pressure readings before the system reached steady state, and the precision of the measuring devices. To decrease the experimental error, we could increase the number of data points for each run to confirm our results.

Answers to Question Set 1:

1. We use the temperature at point 4 (after the expansion valve) as a basis for the corrections of temperatures and pressures at other points to take into account the oil mixed with the refrigerant. The excel program computes the saturation pressure at this point and use it to correct the pressure, enthalpy, and entropy at the rest of the points. The properties at point 4 are obtained using the equation $M = (1 - x^V)M^L + x^V M^V$ where M is either H or S, V is the vapour mole fraction, and L is the liquid mole fraction. For the capillary expansion mode, we assume the expansion to be isenthalpic, and thus the value of enthalpy is the same at points 3 and 4. In the case of the thermostatic expansion (TXV) mode the expansion is isentropic and the value of entropy is the same at points 3 and 4. Although we should note that in practice theses modes are not perfectly isenthalpic or isentropic.

2. The circulation rate of the refrigerant ($\dot{m}_f$) tend to increase the performance of the refrigeration system. As shown on figures xx of Appendix C, the refrigeration capacity increases in a linear fashion with mass flow rate of the refrigerant for both runs of the capillary mode. However for the TXV mode the performance of the refrigeration didn't vary with circulation rate of the refrigerant. The actual coefficient of performance of the system increases with circulation rate for both capillary runs. This is expected given that the refrigeration capacity increases (Eq. (2) of the lab manual.

3. The effect of level of refrigerant on the system is observed in figures 4, 5 and 6 or Appendix C. A mostly linear downwards trend is observed in the coefficient of performance as the amount of refrigerant is increased. Both capillary trials and the TXV trial show this relationship.

4. P-H and T-S diagrams for the first run of the TXV mode are presented in figures 7 – 10 in Appendix C. To simulate an increase in ambient room temperature, all temperature values were increased by 5 degrees Rankin and the excel applet was run to re-calculate entropies and enthalpies of the system. The following observations can be seen on the new P-H and T-S diagrams (figures 8 and 10 respectively):

   - The system is closer to the two phase region at point 3 in the P-H diagram when the temperature is increased.
   - The temperature difference between 3' and 3 on the T-S diagrams is shorter if the ambient temperature is increased.

The effects of an increased ambient temperature seem to have the greatest effect at point 3. This is located right after the condenser in the refrigeration system. These results demonstrate that an increase in lab temperature might reduce the ability for the condenser to condense the two phase fluid since the P-H diagram shows point 3 closer to the two phase region. This is due to its reduced ability to transfer heat to the environment as the temperature difference from the condenser to the environment is now 5 degrees less.

Conclusions and Recommendations

The Scott Air Conditioning and Refrigeration Education system showed very predictable results. When the system was operated in the capillary mode, the refrigeration capacity increased in an almost linear trend with an increase in refrigerant flow rate. This trend was observed in both capillary runs.

The level of refrigerant appeared to have a negative effect on the two capillary runs. As the level of refrigerant was increased a downward linear trend is observed in the refrigeration capacity. This might occur because the refrigerant flow rate is at a minimum when the fluid level is at its highest value.

The TXV mode also provided very predictable results. As the fan speed was decreased, the refrigeration capacity also decreased as with the coefficient of performance. When the fan settings on the condenser were changed the system noticed the largest change in refrigeration capacity. The work required by the compressor was inversely proportional to the fan speed. As the fan speed was decreased, the compressor required more power to compress the two phase fluid.

To increase the accuracy of the results a few recommendations are noted:

- The use of a data logger with pressure and temperature sensors will eliminate rounding error associated by visual inspection of gauges.
- The evaporator was observed to have some damage to its fins. This ultimately reduces its ability to absorb heat from the surroundings and reduces its efficiency. This unit might require replacing.
- The compressor was very close to overheating and a fan was required to cool the unit. Instead of this set up we recommend using the cool air from the evaporator to cool the unit. This might allow the TXV mode of the experiment to be run with a low fan setting on the condenser.
- Both the evaporator and condenser have a copper pipe replaced with a transparent plastic (or glass) pipe to allow for visual inspection of the quality of the fluid. Although this allows

for very good visual observations of the system, it reduces the efficiency of the units. If the evaporator is ever to replaced due to damaged fins, it is recommend the unit be replaced with one that does not have the transparent heat pipe

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Appendix A – Drawings

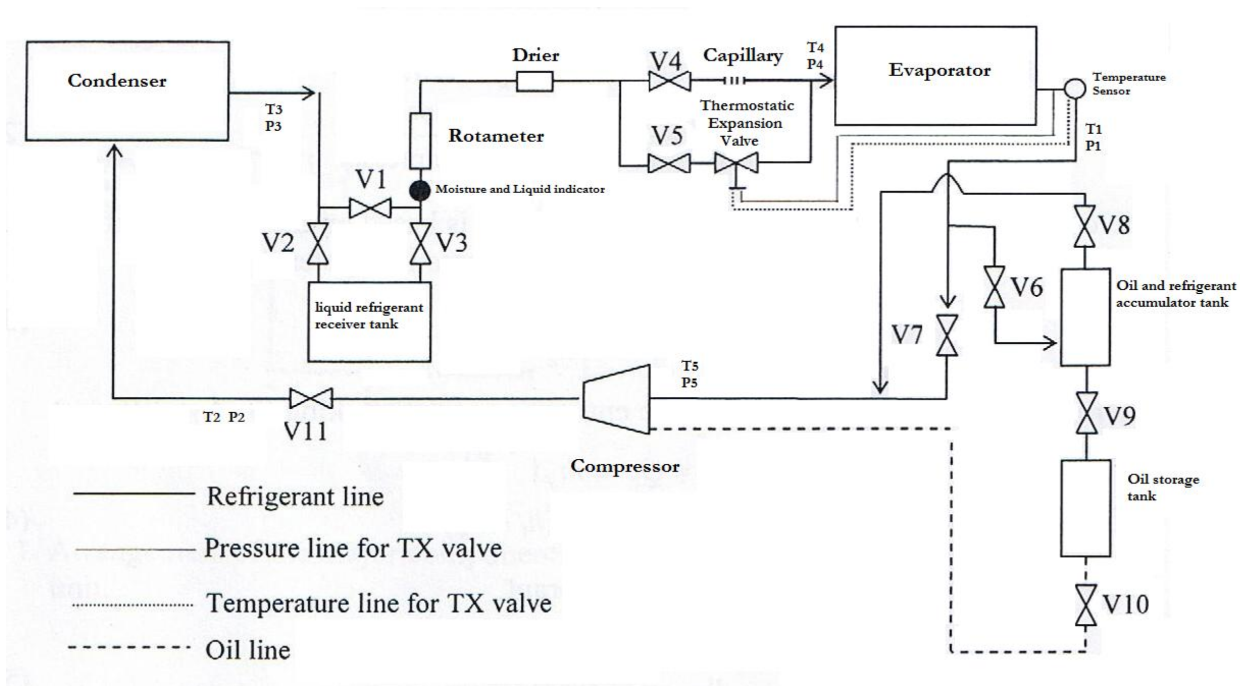


Figure 1 – Schematic diagram of the Scott Air Conditioning and Refrigeration Education system. With temperature and pressure measuring points labelled.

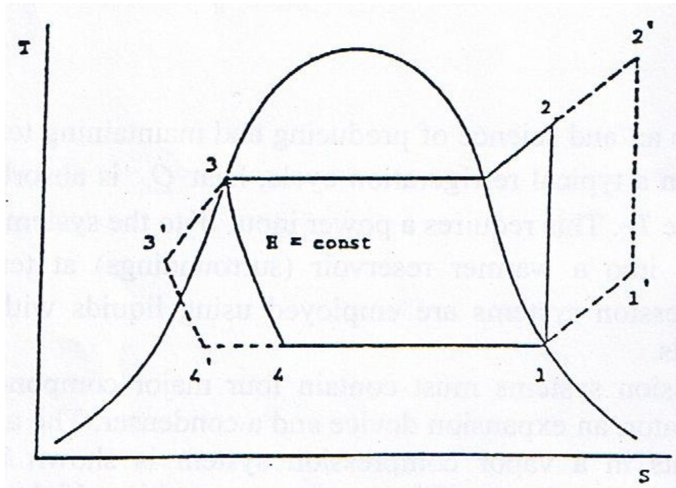


Figure 2 – TS diagram representing the various points in a refrigeration system.

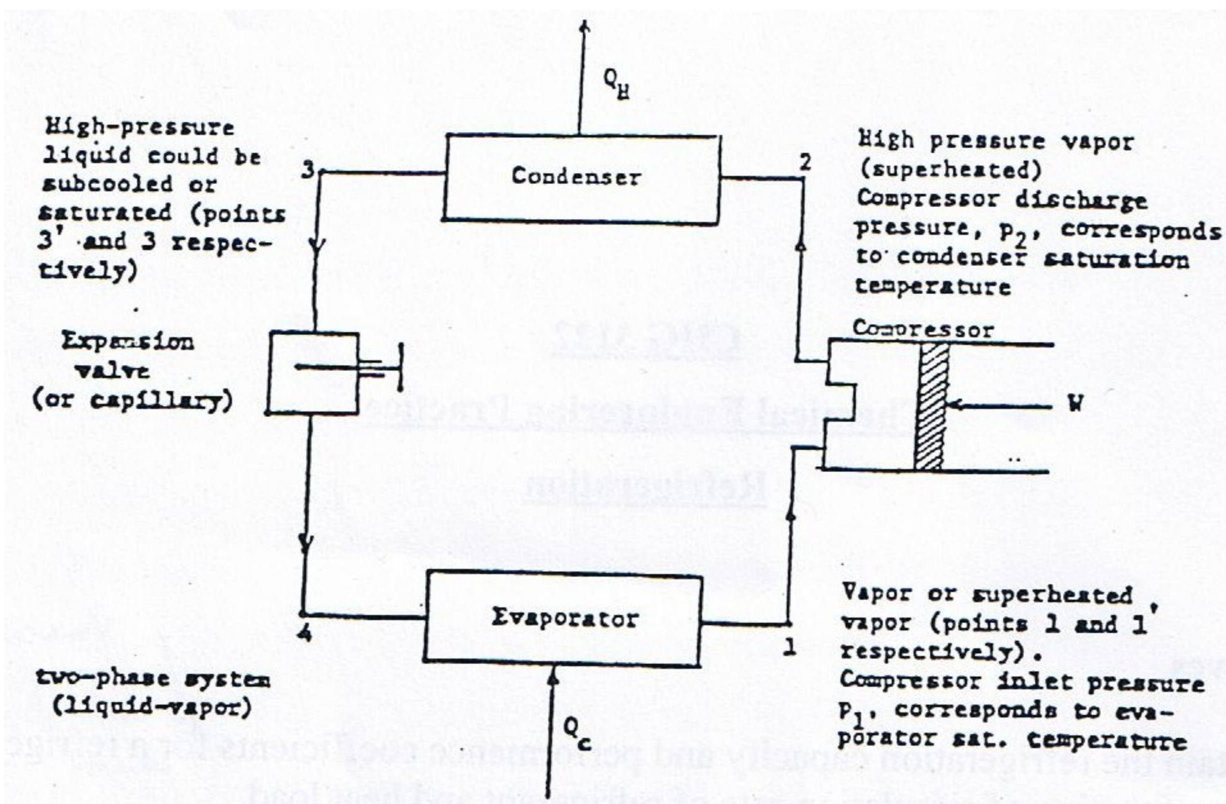


Figure 3 – Major components in a refrigeration system. Heat from the environment is transferred to the evaporator and is rejected to the environment from the condenser.

Appendix B – Tables

Table 1 – Refrigeration system parameters as flow rates vary in the first run of the capillary mode.

	Flow Rate (lbm/min)				
	1.5	1.45	1.39	1.29	1.21
TH (*R*)	568.45	570.82	568.58	568.38	567.47
TC (R)	506.47	506.47	505.47	502.47	499.47
COPmax	8.17	7.87	8.01	7.62	7.35
Qc (Btu/min)	79.6	76.65	75.16	69.45	65.66
W (Btu/min)	44.36	44.36	44.07	42.65	41.51
COPactual	*1.795*	1.728	1.705	1.628	1.582
COPfluid	12.244	7.803	9.51	7.186	6.321
RC	79.6	76.65	75.16	69.45	65.66
ηComp	0.117	0.122	0.097	0.074	0.077
ηCycle	0.22	0.22	0.213	0.214	0.215
CR	2.993	2.964	3.002	3.236	3.253

Table 2 – Refrigeration system parameters as fan speeds vary in the TXV mode.

	Fan Settings			
	S5: High S6: High	S5: High S6:Medium	S5: Medium S6: High	S5: Medium S6:Medium
Flow Rate (Btu/min)	1.72	1.72	1.72	1.72
TH (R)	571.36	572.88	580.88	576.21
TC (R)	510.47	512.47	514.47	512.47
COPmax	8.38	8.48	7.75	8.04
Qc (Btu/min)	87.41	85.94	83.04	82.92
W (Btu/min)	44.36	45.33	46.63	47.49
COPactual	1.97	1.9	1.78	1.75
COPfluid	5.44	6.28	5.33	5.91
RC (Btu/min)	87.41	85.94	83.04	82.92
ηComp	0.21	0.22	0.23	0.22
ηCycle	0.24	0.22	0.23	0.22
CR	2.77	2.72	3.01	2.93

Table 3 – Refrigeration system parameters are varying flow rates in the second run of the capillary mode.

	Flow Rate (lbm/min)				
	1.39	1.32	1.29	1.21	1.18
TH (*R)*	568.84	569.84	569.38	567.98	567.47
TC (R)	504.47	504.47	502.47	499.47	499.47
COPmax	7.84	7.72	7.51	7.29	7.35
Qc (Btu/min)	74.59	70.45	69.08	65.92	64.61
W (Btu/min)	43.11	42.94	42.94	42.54	41.51
COPactual	1.73	1.641	1.609	1.55	1.556

COPfluid	7.396	6.246	6.238	6.333	6.345
RC	74.59	70.45	69.08	65.92	64.61
ηComp	0.097	0.081	0.07	0.056	0.05
ηCycle	0.221	0.213	0.214	0.213	0.212
CR	3.126	3.167	3.143	3.275	3.253

Appendix C – Figures

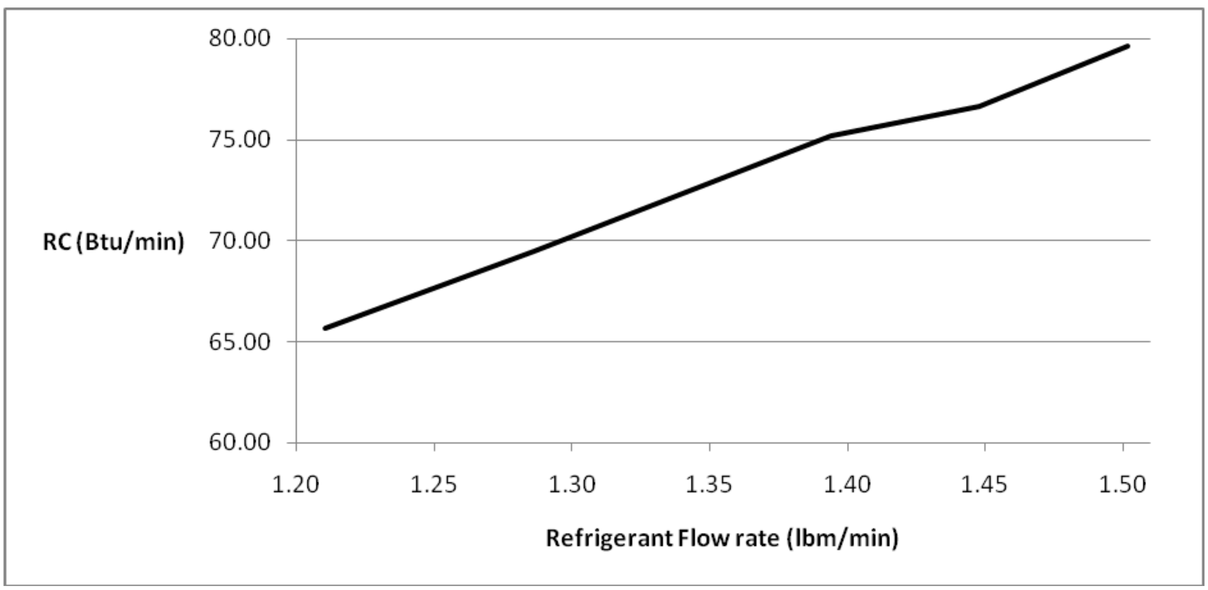


Figure 1 – Refrigeration capacity of the first capillary run of the system as a function of refrigerant flow rate.

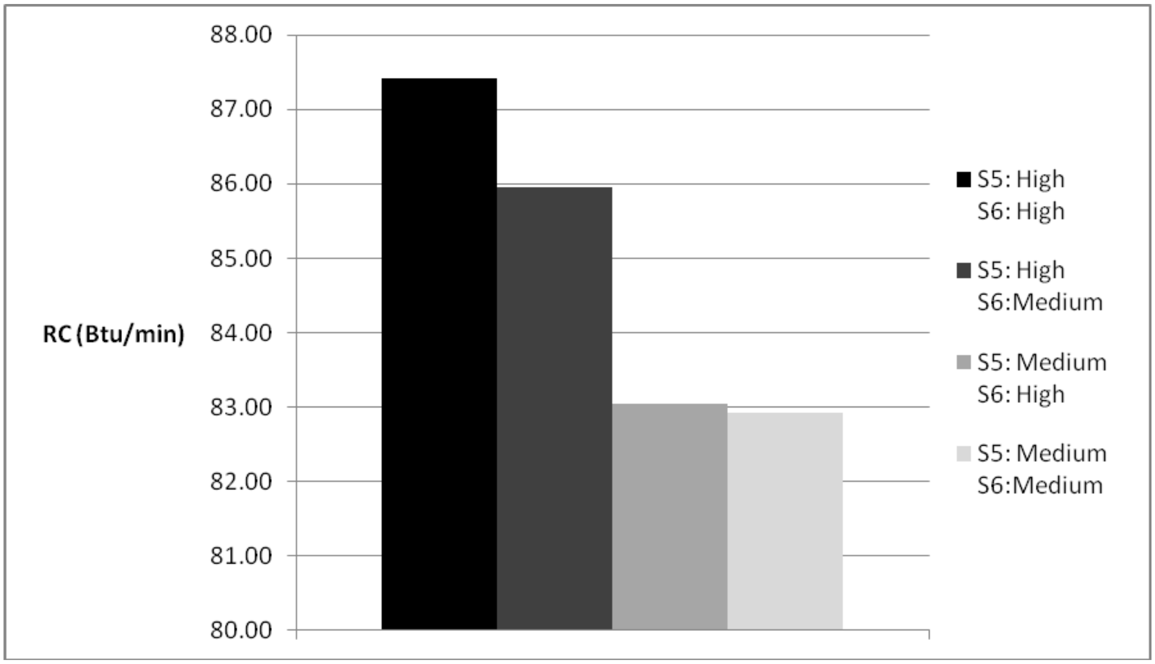


Figure 2 – The effects of condenser and evaporator fan speed on refrigeration capacity during the TXV mode experiment.

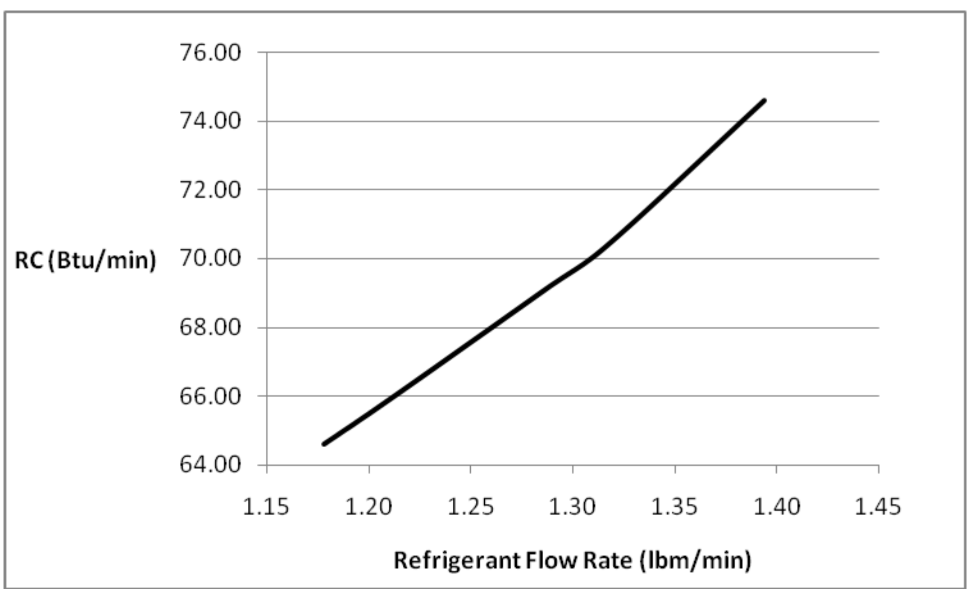


Figure 3 – Refrigeration capacity of the second capillary run of the system as a function of refrigerant flow rate.

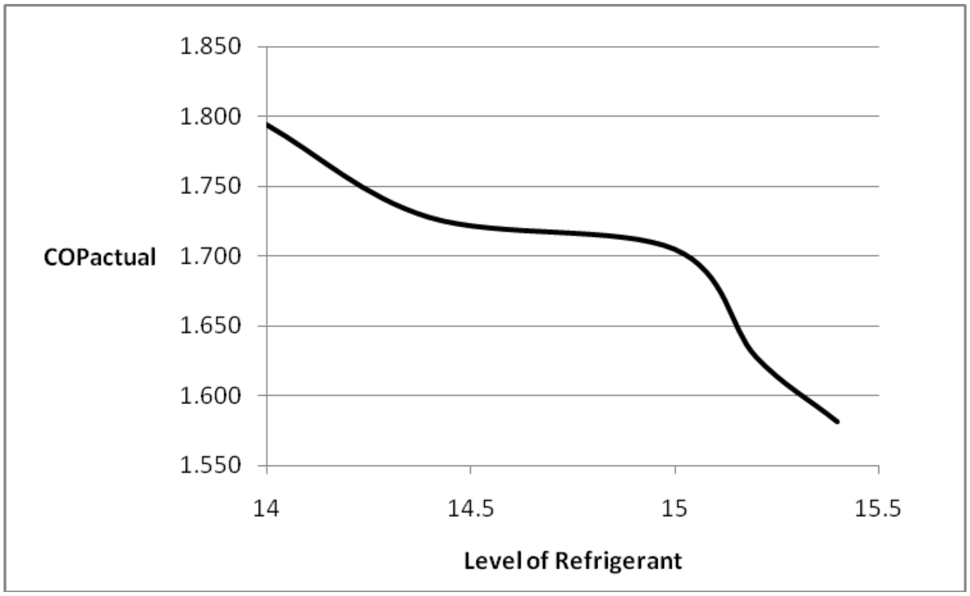


Figure 4 – Coefficient of performance of the first capillary run of the system as a function of refrigerant level.

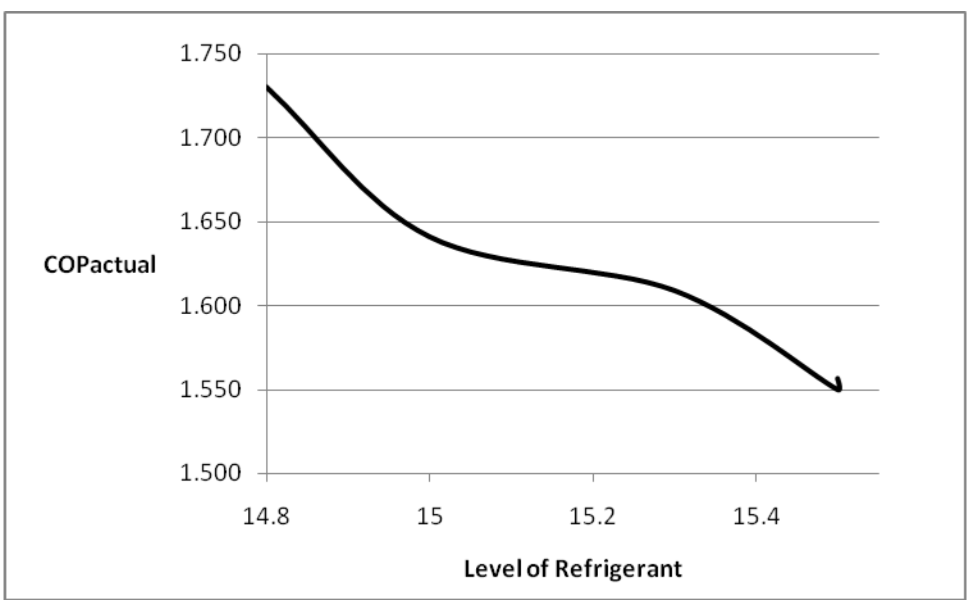


Figure 5 – Coefficient of performance of the second capillary run of the system as a function of refrigerant level.

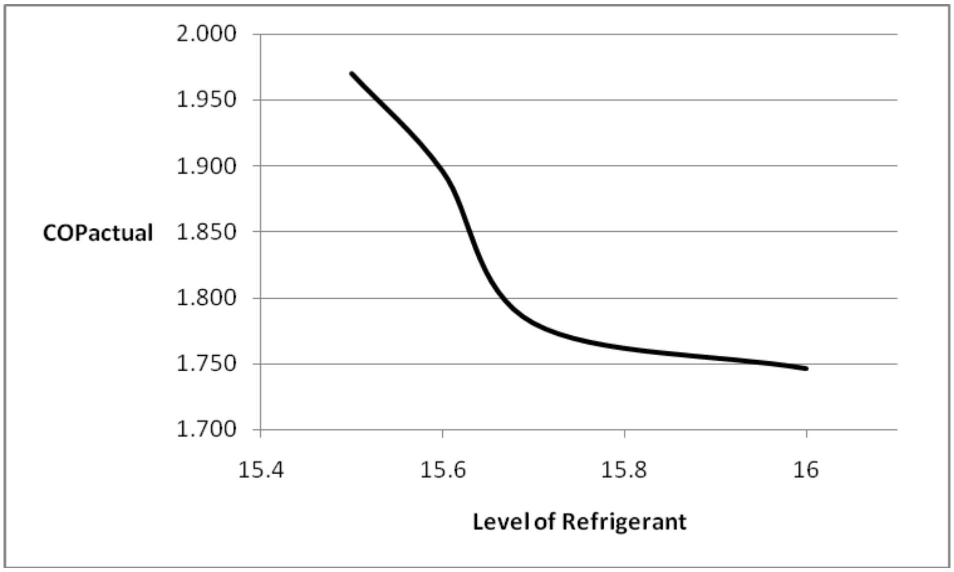


Figure 6 – Coefficient of performance of the TXV mode of the system as a function of refrigerant level.

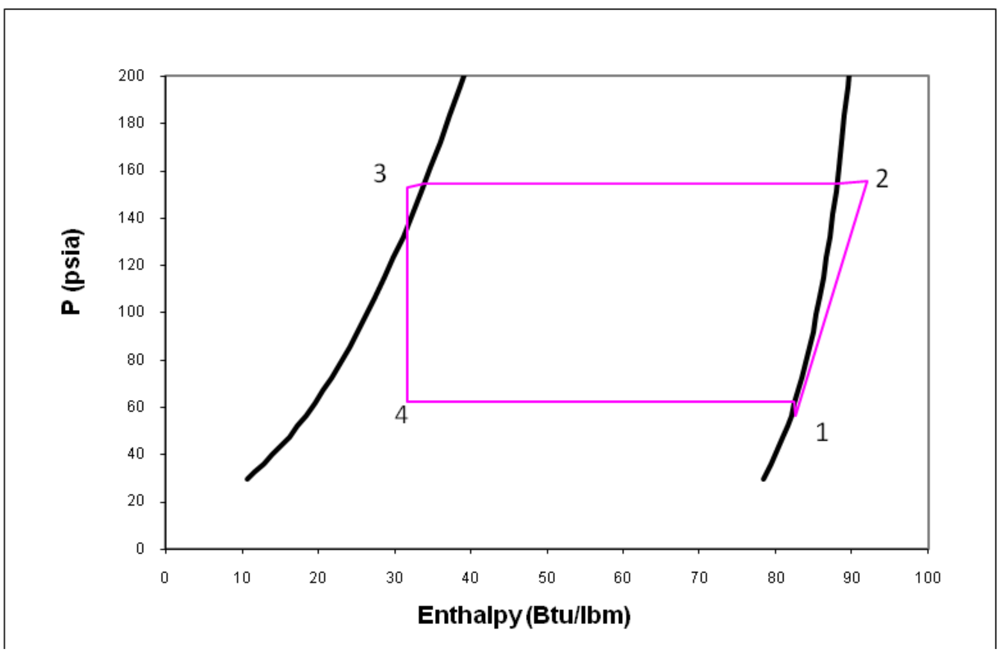


Figure 7– P-H diagram for the first run of the TXV mode.

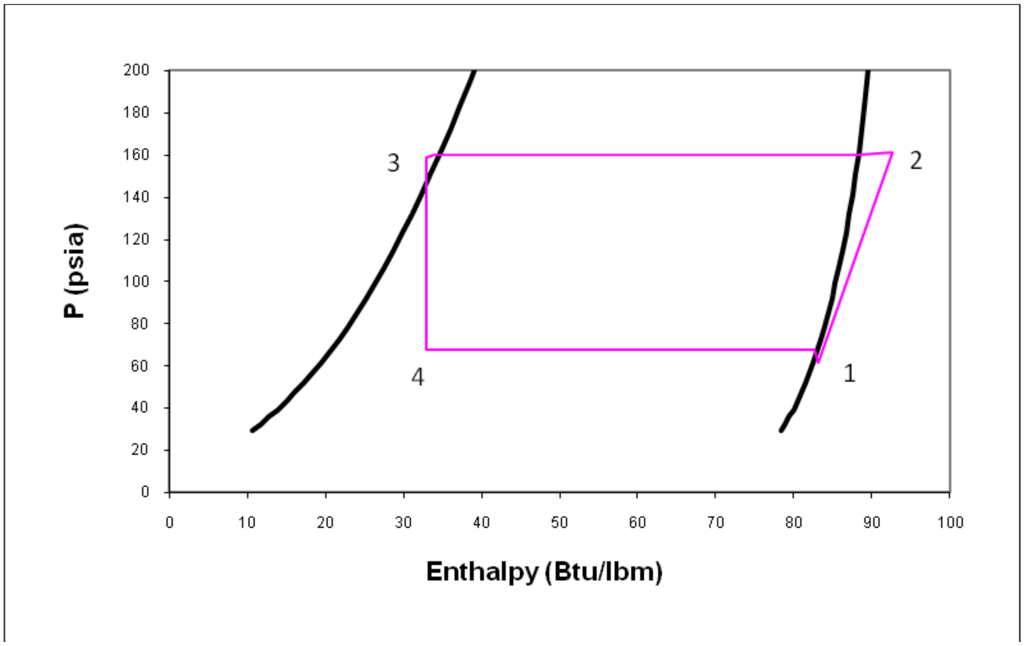


Figure 8– P-H diagram for the first run of the TXV mode assuming the system is operating 5 degrees above lab conditions.

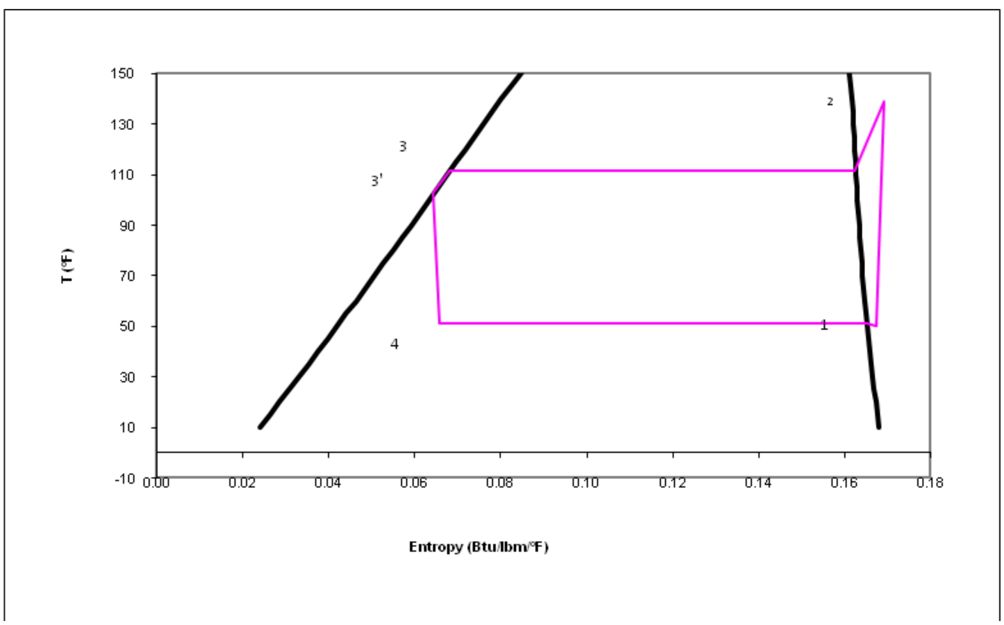


Figure 9– T-S diagram for the first run of the TXV mode.

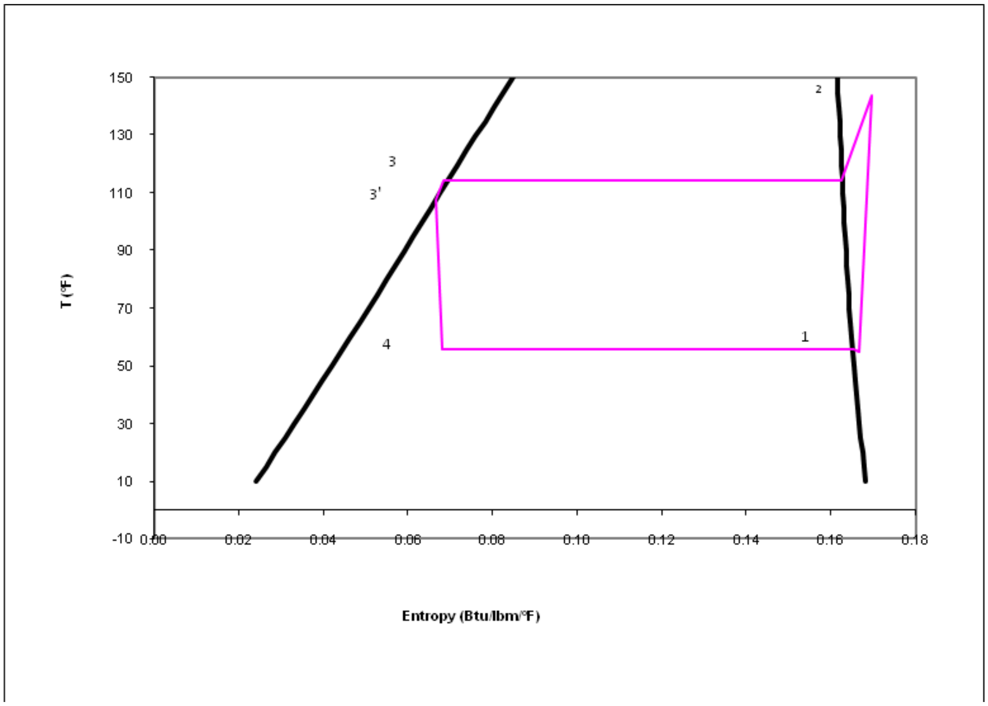


Figure 10– T-S diagram for the first run of the TXV mode assuming the system is operating 5 degrees above lab conditions.

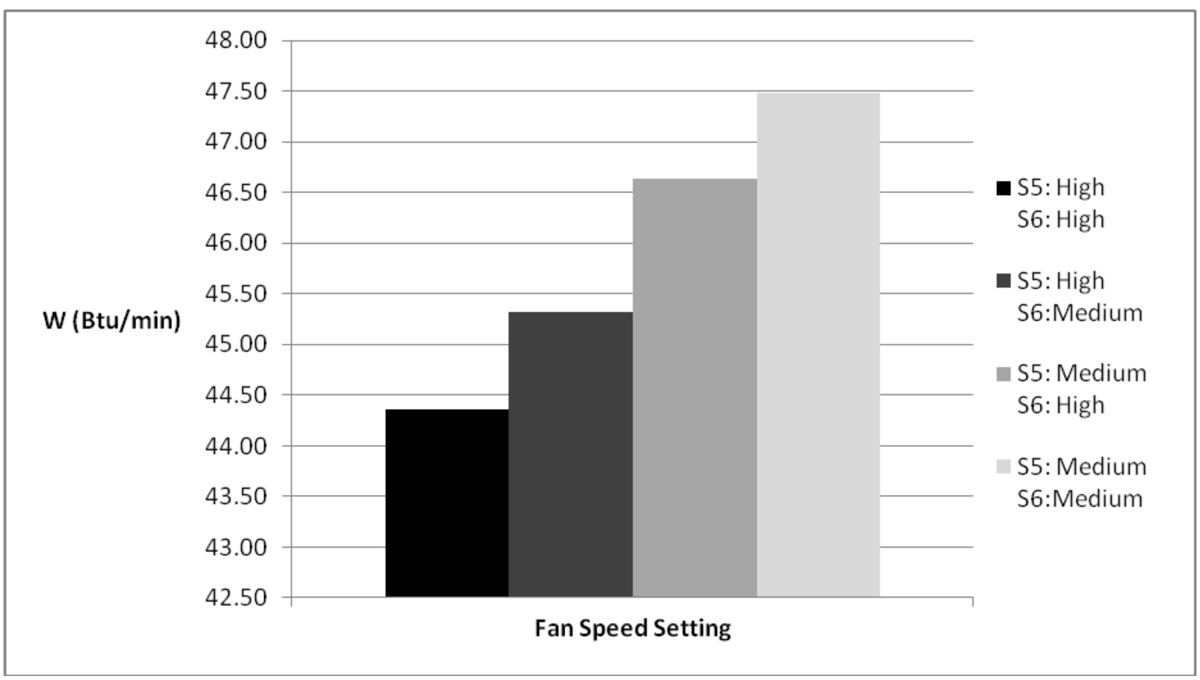


Figure 11– The effects of condenser and evaporator fan speed on compressor work input during the TXV mode experiment.

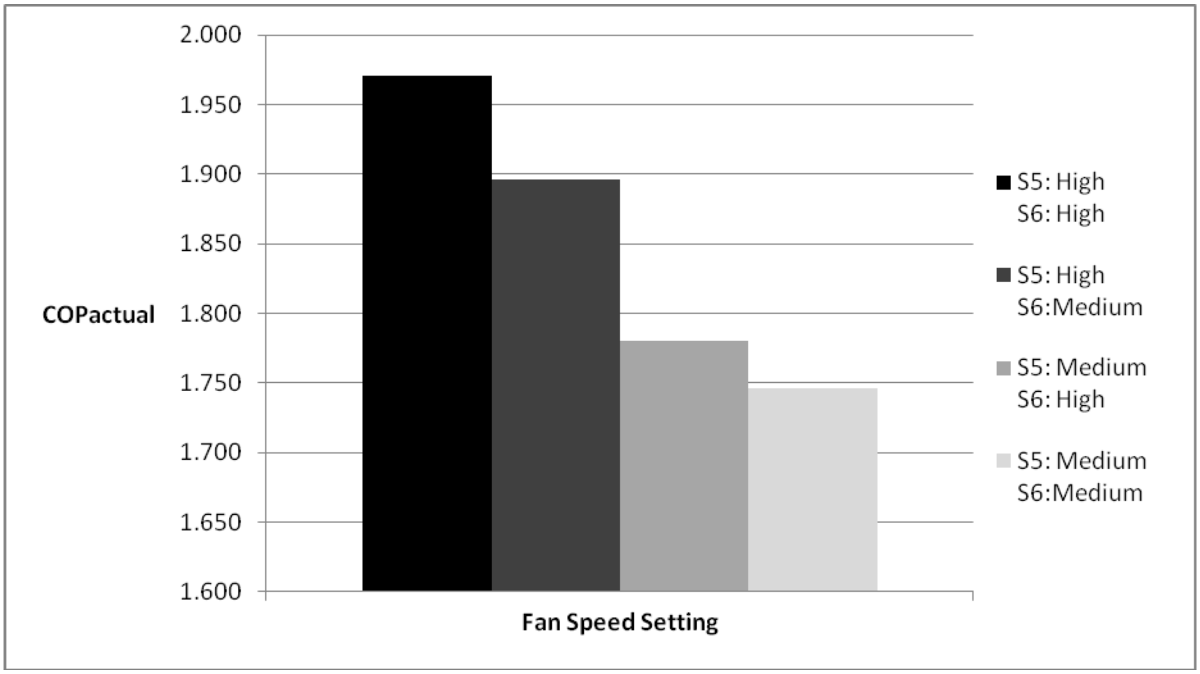


Figure 11– The effects of condenser and evaporator fan speed on the coefficient of performance during the TXV mode experiment.

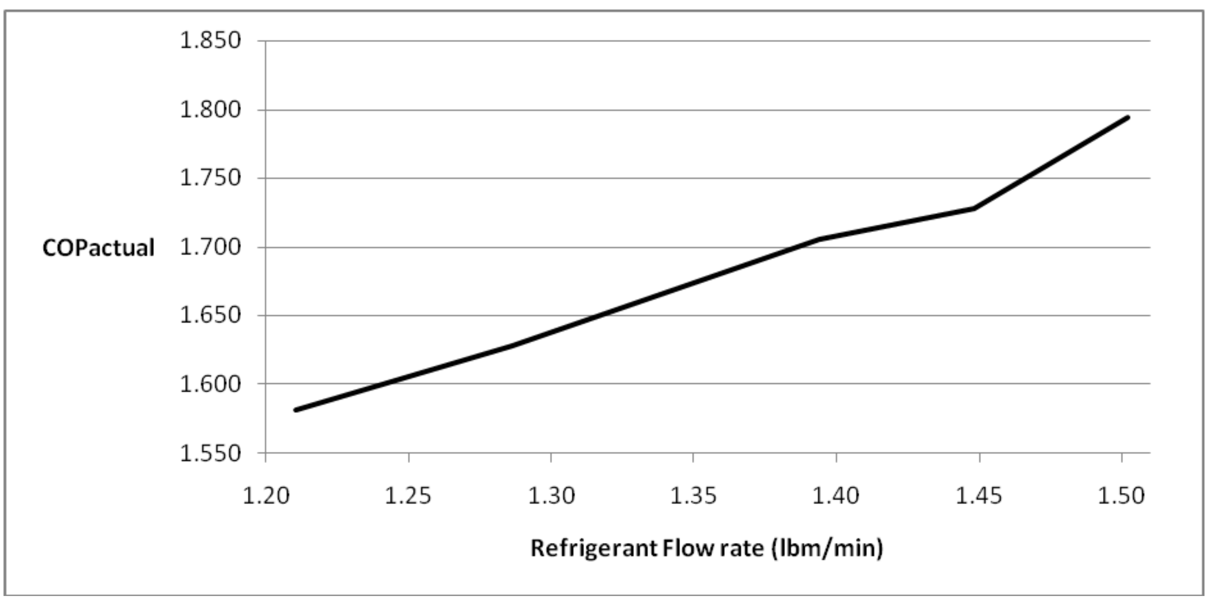


Figure 11– The effects of refrigerant flow rate on the coefficient of performance during the first capillary experiment.

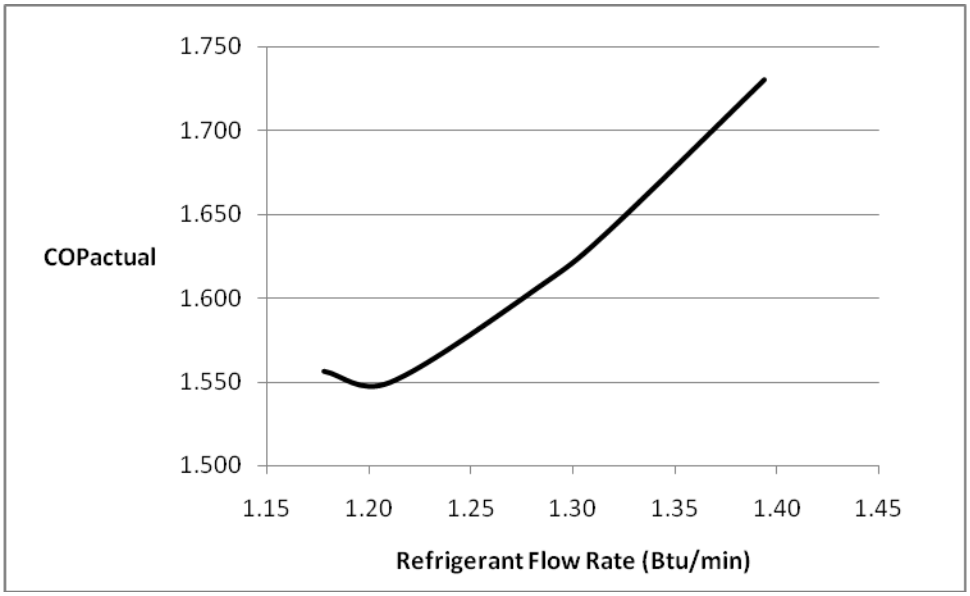


Figure 11– The effects of refrigerant flow rate on the coefficient of performance during the second capillary experiment.

Appendix D- Sample Calculations

All the below calculations are done for the first capillary run at the first selected flow rate.

Conversion of rotameter reading to refrigerant mass flow rate.
From the experiment, the equation relating rotameter reading to flow rate was given as

$$\text{Refrigerant flow rate } (lb_m/min) = 0.108x(\text{rotameter reading}) + 0.098$$

$$0.108x(13)+0.098 = 1.502 lbm/min$$

Conversion factors used in calculations:
Temperature:

1 ^oF = 460.67^oR = 255.93^oK

Pressure

1 psi = 6894.76Pa = 51.71mmHg

Power:

1hp = 745.70watts = 2544.43btu/min

To calculate the Carnot efficiency:

$$COP_{max} = \left(\frac{\dot{Q}_C}{W}\right)_{ideal} = \frac{\dot{Q}_C}{\dot{Q}_H - \dot{Q}_C} = \frac{T_C}{T_H - T_C} \qquad \text{(Equation 1)}$$

Where:

$\dot{Q}_C$ is the rate of energy absorbed in the cold reservoir per unit mass of refrigerant
$\dot{Q}_H$ is the rate of energy rejected in a hot reservoir per unit mass of refrigerant

W is the power input to the compressor per unit mass of refrigerant

T_c and T_H are the temperatures of the cold and hot reservoirs respectively.

$$COP_{max} = \frac{506.47}{568.45 - 506.47} = 8.1716$$

Refrigeration capacity is given by

$$RC = \dot{Q}_c = m_f(\boldsymbol{h_1} - \boldsymbol{h_4}) \qquad \text{(Equation 4)}$$

Where $\dot{m}_f$ is the circulation rate of the refrigerant.

$$RC = \dot{Q}_c = 1.502\left({}^{lbm}/_{\mathrm{min}}\right)\left(84.28\,{}^{Btu}/_{min} - 31.28\,{}^{Btu}/_{min}\right) = 79.60\frac{Btu}{\mathrm{min}}$$

To calculate the actual coefficient of performance:

$$COP_{actual} = \frac{\dot{Q}_C}{W} = \frac{refrigeration\ capacity}{power\ to\ the\ compressor} \qquad \text{(Equation 2)}$$

$$COP_{actual} = \frac{79.60\frac{Btu}{min}}{44.36\frac{Btu}{min}} = 1.7945$$

To calculate the coefficient of performance of the fluid:

$$COP_{fluid} = \frac{\boldsymbol{h_1} - \boldsymbol{h_4}}{\boldsymbol{h_2} - \boldsymbol{h_1}}$$ (Equation 3)

Where h1,h2,h3 and h4 are specific enthalpies of the working fluid at different points in the system (shown in figure 3)

$$COP_{fluid} = \frac{\frac{84.28Btu}{lb_m} - \frac{31.28Btu}{lb_m}}{\frac{88.61Btu}{lb_m} - \frac{84.28Btu}{lb_m}} = \mathbf{12.244}$$

The compression efficiency is given by:

$$\eta_{comp} = \frac{\dot{m}_f(\boldsymbol{h_2^s} - \boldsymbol{h_1})}{W}$$ (Equation 5)

$$\eta_{comp} = \frac{1.502\frac{lbm}{\min\left(\frac{87.74Btu}{lb_m} - \frac{84.28Btu}{lb_m}\right)}}{\frac{44.36Btu}{min}} = \mathbf{0.1173}$$

The cycle efficiency is given by:

$$\eta_{cycle} = \frac{COP_{actual}}{COP_{\max}}$$ (Equation 6)

$$\eta_{cycle} = \frac{1.7945}{8.1716} = 0.2196$$

The compression ratio is given by:

$$CR = \frac{P_2}{P_1}$$ (Equation 7)

Where P_1 and P_2 are the pressures at point 1 and point 2 respectively on figure 3.

$$CR = \frac{150.19}{50.19} = 2.993$$

Appendix E: Raw and non-essential data

Table 1 – Power usage (watts) of system fans at various fan speeds.

Condenser	Evaporator		
	High	*Medium*	*Low*
High	120	105	102
Medium	100	95	90
Low	100	85	80

Table 2 – Rotameter, system power and level of refrigerant for each run during the experiment.

		Rotameter Reading	System Power *(watts)*	Level of Refrigerant
Capillary (First Run)	1st flow rate	13	900	14
	2nd flow rate	12.5	900	14.4
	3rd flow rate	12	895	15

	4th flow rate	11	870	15.2
	5th flow rate	10.3	850	15.4
TXV mode	High:High	15	900	15.5
	High:Medium	15	902	15.6
	Medium:High	15	920	15.7
	Medium:Medium	15	930	16
Capillary (Second Run)	1st flow rate	12	878	14.8
	2nd flow rate	11.3	875	15
	3rd flow rate	11	875	15.3
	4th flow rate	10.3	868	15.5
	5th flow rate	10	850	15.5

Table 3 – First capillary run lab results.

	Point 1	Point 2	Point 3	Point 4	Point 5
First Flow Rate					
Temperature Reading *(°F)*	60	115	98	46	62
Pressure Reading *(psig)*	45	142.5	142	52	48
Second Flow Rate					
Temperature Reading (°F)	60	132	98	46	62
Pressure Reading *(psig)*	45	145	145	50	47.5
Third Flow Rate					
Temperature Reading *(°F)*	63	127	96	45	66
Pressure Reading *(psig)*	44	142	141	50	48
Fourth Flow Rate					
Temperature Reading *(°F)*	66	143	98	42	58
Pressure Reading *(psig)*	40	141	141	47	45

Fifth Flow Rate					
Temperature Reading *(°F)*	64	148	96	39	69
Pressure Reading *(psig)*	40	140	140	45	42

Table 4 – TXV run lab results.

	Point 1	Point 2	Point 3	Point 4	Point 5
S5: High S6: High					
Temperature Reading (°F)	50	139	100	50	44
Pressure Reading (psig)	50	147	147	55	52
S5: High S6:Medium					
Temperature Reading (°F)	48	128	102	52	44
Pressure Reading (psig)	50	148	148	55	53
S5: Medium S6: High					
Temperature Reading (°F)	50	141	110	54	44
Pressure Reading (psig)	50	165	165	57	53
S5: Medium S6:Medium					
Temperature Reading (°F)	48	132	109	52	44
Pressure Reading (psig)	**50**	**160**	**150**	**55**	**54**

Table 5 – Second capillary run lab results.

	Point 1	Point 2	Point 3	Point 4	Point 5
First Flow Rate					
Temperature Reading (°F)	63	138	98	44	62
Pressure Reading (psig)	43	143	143	50	45
Second Flow Rate					
Temperature Reading (°F)	66	150	100	44	66
Pressure Reading (psig)	42	144	144	49	45
Third Flow Rate					
Temperature Reading (°F)	68	152	100	42	68
Pressure Reading (psig)	41	142	142	46	44
Fourth Flow Rate					
Temperature Reading (°F)	69	153	98	39	69

Pressure Reading (psig)	40	141	141	45	42
Fifth Flow Rate					
Temperature Reading (°F)	70	154	97	39	70
Pressure Reading (psig)	39	139	139	44	42

To: Dr. Kruczek
From: Mukhtar Al-Ismaily, Sherif Yared, Group 4a
Date: March 12, 2007
Subject: CHG3122, Refrigeration

In a refrigeration cycle, hot and cold reservoirs are used and heat is interchanged between them. Heat from the cold reservoir, $\dot{Q}_C$ is absorbed at a temperature T_C. This heat then is transferred to the hot reservoir with heat $\dot{Q}_H$ at a temperature T_H by using a power input W to the system. In this experiment a vapour-compression system is employed, using Freon-12 (also called R-12) as the working fluid. Some objectives that needed to be obtained in this experiment were: Determining the refrigeration capacity and coefficient of performance for the refrigeration cycle as the circulation rate of R-12 was changed. This was done for two modes, the capillary valve and the thermostatic expansion valve (TXV). A comparison of the performance of the refrigeration unit for each of these two modes was also done and some conclusions and trends were seen.

A Scott Air Conditioning and Refrigeration Education system was used for this experiment, with R-12 acting as the working fluid. The experiment ran under two modes: normal capillary and thermostatic expansion. Five strategically set pressure and temperature taps were used in this system, so the required parameters could be obtained. One major problem that did arise was the fact that the compressor that was being used was not capable of handling high temperatures and as a result shut down every 30-40 minutes. This caused some delay in the experiment since the compressor had to be left to cool down until the operation could be started up again. Knowing that the compressor would shut down quite often, readings for temperature and pressure had to be taken quiet fast and any error in those readings would lead to inaccurate results. One major error would be having incorrect saturation temperatures and being mislead to whether the fluid was in the vapour or fluid regime.

Overall some key conclusions can be stated about this experiment as such: An increase in the circulation rate of the refrigerant did increase the performance of the unit in both the capillary and TXV mode. COP_{actual} values were found to be around 1.59 averaged for both capillary runs and approximately 1.61 for the TXV mode. Also the refrigeration capacity (heat load) did show an increase as the circulation rate was increased giving values of RC: 76.41 Btu/min for averaged capillary runs, and about 84 Btu/min for the TXV mode. One final observation that was made was that as the amount of refrigeration was increased in the system the performance of the fluid decreased. Values for COP_{fluid} for both capillary runs were averaged to be around 6.46, and for the TXV mode approximately 7.23.

Equipments and Procedure:

This experiment was performed using a Scott Air Conditioning Education system (see Figure 1) with Freon-12 (R-12) as the refrigerant fluid. Two of the available operating modes in the system are run: Normal capillary and thermostatic expansion (TXV). The system's main refrigeration components are as follows: A compressor (A), a condenser (B), an evaporator (C), a capillary (D), a thermostatic expansion valve (E), a temperature sensor (F), valves (V) a moisture and liquid indicator (G), a calibrated rotameter (H), a drier (I), a liquid refrigerant receiver tank (J), an oil and refrigerant accumulator tank (K) and an oil storage tank (L). The pressure/temperature gauges are all placed in strategic locations throughout the system as labelled (T and P) in the figure:

1 – Evaporator outlet, 2 – Condenser inlet, 3 – Condenser outlet, 4 – Evaporator inlet, and 5 – Compressor inlet.

The system was initially ensured to being at thermal equilibrium (Temperature in all 5 gauges are the same). After that, it was necessary to determine the work done by the compressor by recording the wattage of all fan setting combinations (See Appendix D, Table 10). Both fans are then set to high, and the experiment is started.

For the first set of runs, a lower amount of refrigerant was used and the capillary expansion valve was also used for the expansion step. For this set of runs, valve 5 is closed, and valve 4 is used to control the circulation rate (Figure 1). The recording of 5 temperatures and 5 pressures were taken at all 5 different flow rates (giving 25 temperatures, 25 pressures, 5 flow rates and 5 power readings). These readings were taken once the system has reached equilibrium. The same procedure as above was repeated, only for a higher amount of refrigerant (note the refrigerant level readings at Appendix A, Table 4).

For the third set of runs, valve 4 is closed, and valve 5 is then used to control the circulation rate since the experiment was performed under the TXV mode (for the expansion of course). The same recordings were taken, only at different 4 fan settings which where: high-high, high-medium, medium-high and medium-medium (recorded readings shown in Appendix D, Tables 11, 12, and 13). The system works as follows: The refrigerant (vapour) leaves the evaporator then enters a compressor then exits as a high pressure vapour. Then this high pressure vapour enters the condenser which results in having a high pressure saturated liquid and also results in a rejection of heat into the hot reservoir. This high pressure, saturated liquid then enters the expansion valve (capillary or thermostatic), thus resulting into a low pressure liquid. This decrease in pressure also results in a decrease in temperature (which is colder than the surrounding air). This low pressure, low temperature liquid then enters an evaporator which then becomes vapour and also some energy is absorbed into the cold reservoir. The Freon returns to its original state.

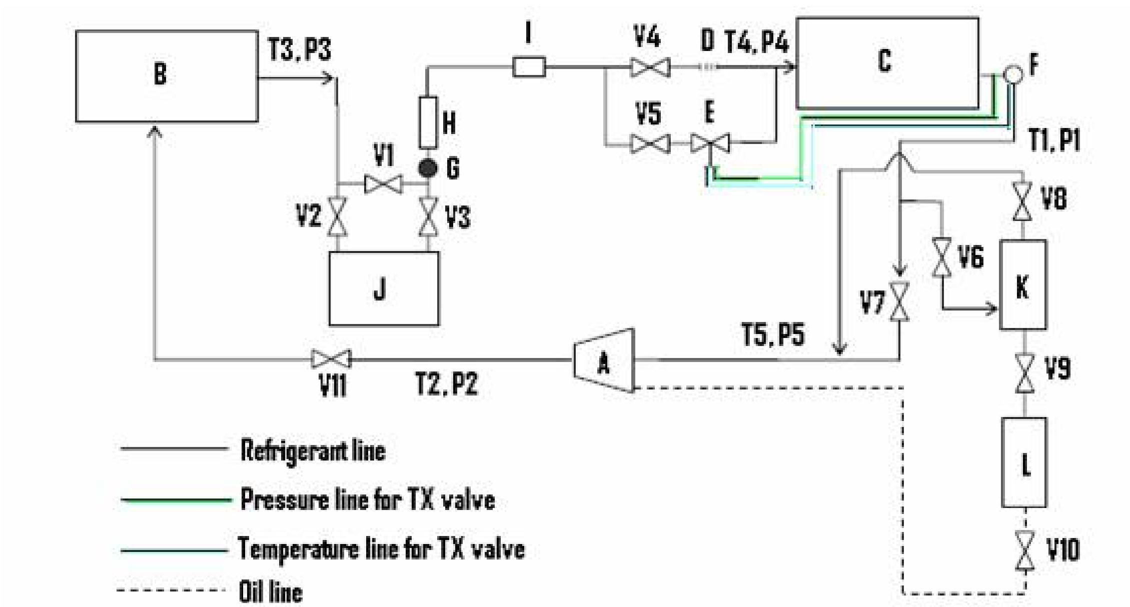


Figure 1 – A general schematic of the Scott Air Refrigeration system being used to carry out experiment

Summary of Results:

1. The temperatures for the hot and cold reservoirs for the refrigeration system that was tested was calculated at both capillary runs (1 & 2), and at the thermostatic expansion valve (TXV) mode. The averages are shown below for both capillary runs and the TXV mode:

- Capillary Run 1: $T_C = 509.17°R$; $T_H = 578.89°R$
- Capillary Run 2: $T_C = 509.77°R$; $T_H = 575.16°R$
- TXV Mode: $T_C = 515.42°R$; $T_H = 582.51°R$

2. The refrigeration capacity was found for both modes (capillary and TXV) using (Eq. 4). The averaged values for the changing flow rates for the corresponding modes are shown as follows:

- Capillary Run 1: $RC = \dot{Q}_C = 78.90\, {}^{Btu}\!/_{\min}$
- Capillary Run 2: $RC = \dot{Q}_C = 73.91\, {}^{Btu}\!/_{\min}$
- TXV mode: $RC = \dot{Q}_C = 83.87\, {}^{Btu}\!/_{\min}$

3. The performance of the refrigeration system that was tested was found at both capillary runs (1 & 2), and at the thermostatic expansion valve (TXV) mode. The average values for the COP_{actual}, COP_{max}, and COP_{fluid} were found for each mode by averaging over the changing flow rates that were adjusted using the corresponding capillary or TXV valve. These values are shown as follows:

- Capillary Run 1: $COP_{\max} = 7.36$; $COP_{actual} = 1.62$; $COP_{fluid} = 7.55$
- Capillary Run 2: $COP_{\max} = 7.82$; $COP_{actual} = 1.55$; $COP_{fluid} = 5.38$
- TXV Mode: $COP_{\max} = 7.72$; $COP_{actual} = 1.61$; $COP_{fluid} = 7.23$

4. The efficiency of the compression stage for both the capillary and TXV mode was found using (Eq.5). The averaged efficiencies for each run for varying flow rates is shown below:

- Capillary Run 1: $\eta_{comp} = 0.186 \Rightarrow 18.6\%$
- Capillary Run 2: $\eta_{comp} = 0.143 \Rightarrow 14.3\%$
- TXV mode: $\eta_{comp} = 0.200 \Rightarrow 20.0\%$

5. The overall efficiency of the cycle was also determined for both modes using (Eq. 6). The averaged efficiencies are listed below as follows:

- Capillary Run 1: $\eta_{cycle} = 0.221 \Rightarrow 22.1\%$
- Capillary Run 2: $\eta_{cycle} = 0.199 \Rightarrow 19.9\%$
- TXV mode: $\eta_{cycle} = 0.210 \Rightarrow 21.0\%$

6. The compression ratio was obtained for the different operating modes and the averaged values for each run are listed as follows (by using Eq. 7):

- Capillary Run 1: $CR = 3.24$
- Capillary Run 2: $CR = 2.90$
- TXV mode: $CR = 2.90$

7. The circulation rate of refrigerant $\dot{m}_f$ was plotted as a function of performance for both modes (i.e. the capillary and TXV mode). It can be seen that for the first capillary run that as the circulation rate was increased so did the performance, and this behaviour was in an almost linear fashion. For the second capillary run, the behaviour was quite similar. Again an increase in circulation rate did show an increase in performance; however the relationship observed was not so linear. For the TXV mode it was shown that the circulation rate of the refrigerant did not affect the performance, and no trend was observed (Appendix C, Figure 3).

8. The amount of refrigerant in the system was plotted as a function of fluid performance. Looking at the two capillary runs (Appendix C, Figure 4) it can be seen that at lower levels of refrigerant the performance was higher. As the level of refrigerant was increased for the second capillary run the behaviour seen is that the performance of the fluid decreased quite a bit. Comparing the capillary mode at the first run with the TXV mode, which had similar levels of refrigerant, a general trend can be seen where the performance of the fluid for the capillary was higher then for the TXV.
9. If the temperature in the pilot plant was to increase by a few degrees, the temperature and pressures at the strategic location would differ. Some observations can be made by taking the first experimental run for the TXV mode on the T-S and p-H diagrams and redrawing them with this new increase in temperature. In general the plot for the T-S diagram shifted up and resulted in points 4 – 1 and 3 – 2 to land on higher isotherms. For the p-H diagram the cycle also shifted upwards with points 4 – 1 and 3 – 2 being on higher isobars. The evaporation and condensation processes in general took place at higher temperatures. Referring to Figures 5-8 Appendix C shows these observations.

Discussion:

The temperature reading at point 4 (see Appendix C, Figure 2), was used as the basis for correcting all other temperatures readings. The reason to this is that at that point, the refrigerant is at both the liquid and vapour phase (2 phases), and since the fluid only consists of one component, then according to Gibb's phase rule only one intensive property is required to solve for the system and to obtain all remaining parameters, which for this case is the temperature reading. Another reason to selecting this as the basis point is because it happens to be the starting point of the whole experiment; right after it leaves its storage tank.

As shown in Appendix C, Figure 3, the actual coefficient of performance of the system increased with an increase in circulation rate on both capillary runs, whilst exhibiting a more linear behaviour in the first capillary run. Assuming all other parameters remained unchanged; an increase in the circulation rate (increase fluid velocity) would lead to a decrease in pressure according to Bernoulli's equation. This decrease in pressure would lead to a decrease temperature, would have an immediate effect on T_H since the control valve lies at that point (from 581ºR to 574ºR, see Tables 1, 4, and 7 in Appendix A). Since all parameters have to remain constant, T_C would be forced to increase in order to maintain constant entropy/enthalpy values at all measured points. According to Table 1 in Appendix A, these changes in T_C (503ºR, 506ºR...etc) would lead to in increase in the heat absorbed in the cold reservoir, hence increasing Q_C (68 Btu/min, 77 Btu/min... Etc). Increasing Q_C would lead to an increase in the actual coefficient of performance as shown by Eq. 2 Appendix B. Q_C and the refrigeration capacity are the same, so increasing the flow rate should eventually lead to an increase Q_C as shown by Eq. 4 Appendix B

The change in coefficient of performance with circulation rate in the thermostatic mode shows no trend according to Figure 3, Appendix C. The reason to this is a decrease in flow rate in the valve leads to no change in the overall pressure of the system; therefore Bernoulli's Equation doesn’t apply in this case. This would lead to no significant change in temperature readings with respect to flow rate, hence not exhibiting any significant change in the coefficient of performance of the refrigeration system.

As shown in Figure 4 in Appendix C, an increase in amount (level) of the refrigerant used shows a decrease in the coefficient of performance of the fluid itself. Also noticed is that at the second capillary run, the averaged coefficient of performance (5.38) was significantly lower in comparison to the first run (7.55) and TXV mode (7.23), see Tables 3, 6, and 9 in Appendix A.

R-12 was the first halo-carbon refrigerant to go into general use. Its immiscibility with oil is one of its best characteristics. Nonetheless increasing its amount causes it to be in greater excess. The greater amount will tend to reduce the heat transfer rate since it's resistance due to convection increases because of its amount. As a result, this will lead to a lesser performance of the Freon as a refrigerant and the system itself. In the second run of Capillary mode when extra refrigerant was used, the maximum coefficient of performance was lower than the first one which is expected.

In the first TXV run, the effect of increasing the ambient temperature is shown in Figures 5-6 and 7-8 in Appendix C. This increase (by approximately 12.5 F) would lead to an increase in temperatures of the hot and cold reservoirs to 589ºR and 528ºR, respectively. Assuming the change in the ambient air temperature will have no effect on the compressor ratio, the change will have a considerable effect on two major components in the refrigeration system: the *Evaporator* and the *Condenser*. In the evaporation process, increasing the temperature of the cold reservoir will lead to an increase in the driving force involved of transferring heat into the fluid. This results in the production of a greater amount superheated vapour exiting the evaporator, before even entering the compressor. The enthalpy and entropy values should decrease at points 1, 1' in Figure 2 Appendix C. On the other hand, the opposite occurs in the condenser, since the increase in temperature of the hot reservoir (heat sink) will reduce the driving force that rejects the heat. This leads to an increase in entropy and enthalpy values at points 3 and 3'.

The main source of error in this experiment results was the compressor overheating. It turned off a lot of times which forced the group to rush in taking the readings that may have been inaccurate which may have led to great deviations in the calculated thermodynamic properties. Another source of error is that the piping within the system wasn't well insulated, hence leading to heat loss throughout nearly all steps shown in Figure _ Appendix C, especially at steps when the process is suppose to be adiabatic (e.g. 3'-4').

Conclusion:

In the vapour-compression system that was test, the refrigerant R-12 did behave as expected in some manners. Mainly when the circulation rate of R-12 was varied and increased an increase in the performance of the unit was observed. For the capillary run an overall linear fashion was seen for the first run, while for the second capillary run a more curved fit was observed. The TXV mode however did not really show any set trend and it could not be seen weather or not the circulation rate had any effect on its performance. It was also seen that overall the refrigeration capacity was higher for the TXV mode then for both capillary runs shown that a higher heat load is required to be removed from the cold reservoir. When the efficiency of the compression cycle was determined it was seen that the TXV mode exhibited higher efficiency then the capillary mode. However when the overall efficiency of the system was found both modes, the TXV and capillary hovered around almost the same efficiency. Some errors in the set up of the experiment and the actual data obtained did skew off the results obtained. Some recommendations on improving the quality of the results and the efficiency of the experiment can be made as follows:

- Since the compressor kept on overheating and turning off, it would've been best if the condenser was partially submerged in a tank of cold water, assuming the fact the compressor is well insulated.
- Thermally insulating the pipelines or having them replaced with pipes of better insulation will appear as a big improvement in preventing heat loss in all steps of the refrigeration process.

- Using digital pressure and temperature transducers that would result in more accurate readings in the strategic points.
- Since the rotameter readings taken kept on oscillating in each run, replacing the rotameter with a newer and more improved one would've improved the results obtained.

Appendix A

<u>Tabulated Results</u>

Table 1 - Changes in the heat absorbed in the evaporator (at T_C) and released in the condenser (T_H) at various circulation rates. [Capillary Run 1]

Flow *(lbm/min)*	**Amount** *(inch)*	**Compressor Power** *(Btu/min)*	**T_C** *(°R)*	**T_H** *(°R)*	**Q_C** *(Btu/min)*	**Q_H** *Btu/min*
1.718	9.2	48.624	513.170	574.085	85.040	95.966
1.686	9.7	48.737	512.170	578.037	83.976	95.360
1.61	9.8	48.737	511.170	580.287	79.921	91.561
1.502	9.9	48.510	506.170	580.900	77.258	87.060
1.297	9.9	48.624	503.170	581.157	68.292	76.973
	Average		509.170	578.893	78.897	89.384

Table 2 - Changes in enthalpy and pressure values at strategic points in the system, under various flow rates (see Appendix C, Figure _). [Capillary Run 1]

Flow *(lbm/min)*	**h'_1** *(Btu/lbm)*	**h'_2** *(Btu/lbm)*	**h^s_2** *(Btu/lbm)*	**h'_3** *(Btu/lbm)*	**h'_4** *(Btu/lbm)*	**P_1** *(psig)*	**P_2** *(psig)*
1.718	82.610	88.971	88.185	33.111	33.111	55.174	159.174
1.686	82.395	89.149	88.485	32.575	32.575	57.094	167.094
1.61	82.200	89.429	88.653	32.559	32.559	55.027	172.027
1.502	82.989	89.515	88.698	31.553	31.553	45.889	173.889
1.297	83.699	90.394	88.717	31.038	31.038	48.960	169.960

Table 3 - Changes in certain coefficients of performances and efficiencies in the system, under various circulation rates. [Capillary Run 1]

Flow *(lbm/min)*	**COP_{max}**	**COP_{actual}**	**COP_{fluid}**	**RC**	**η_{comp}**	**η_{cycle}**	**CR**
1.718	8.424	1.749	7.783	85.040	0.197	0.208	2.885
1.686	7.776	1.723	7.377	83.976	0.211	0.222	2.927
1.61	7.396	1.640	6.867	79.921	0.213	0.222	3.126
1.502	6.773	1.593	7.882	77.258	0.177	0.235	3.789
1.297	6.452	1.404	7.866	68.292	0.134	0.218	3.471
Average	7.364	1.622	7.555	78.897	0.186	0.221	3.240

Table 4 - Changes in the heat absorbed in the evaporator (at T_C) and released in the condenser (T_H) at various circulation rates. [Capillary Run 2]

Flow *(lbm/min)*	**Amount** *(inch)*	**Compressor Power** *(Btu/min)*	**T_C** *(°R)*	**T_H** *(°R)*	**Q_C** *(Btu/min)*	**Q_H** *Btu/min*
1.686	12.2	48.908	515.170	576.075	80.386	96.052
1.610	12.4	48.737	513.170	575.745	76.846	91.664
1.480	12.8	48.283	510.170	576.822	74.018	89.244
1.286	12.9	46.065	504.170	575.863	67.251	79.308
1.372	13.2	46.065	506.170	571.284	71.039	82.594
	Average		509.770	575.158	73.908	87.772

Table 5 - Changes in enthalpy and pressure values at strategic points in the system, under various flow rates (see Appendix C, Figure _). [Capillary Run 2]

Flow *(lbm/min)*	**h'_1** *(Btu/lbm)*	**h'_2** *(Btu/lbm)*	**h^s_2** *(Btu/lbm)*	**h'_3** *(Btu/lbm)*	**h'_4** *(Btu/lbm)*	**P_1** *(psig)*	**P_2** *(psig)*
1.686	82.823	92.117	88.337	35.133	35.133	60.375	163.375
1.610	82.352	91.556	88.312	34.622	34.622	58.174	162.174
1.480	83.596	93.881	88.394	33.597	33.597	54.973	164.973
1.286	84.884	94.260	88.321	32.590	32.590	50.924	161.924
1.372	85.128	93.547	87.968	33.365	33.365	51.889	147.889

Table 6 - Changes in certain coefficients of performances and efficiencies in the system, under various circulation rates. [Capillary Run 2]

Flow *(lbm/min)*	**COP_{max}**	**COP_{actual}**	**COP_{fluid}**	**RC**	**η_{comp}**	**η_{cycle}**	**CR**
1.686	8.459	1.644	5.131	80.386	0.190	0.194	2.706
1.610	8.201	1.577	5.186	76.846	0.197	0.192	2.788
1.480	7.654	1.533	4.861	74.018	0.147	0.200	3.001
1.286	7.032	1.460	5.578	67.251	0.096	0.208	3.180
1.372	7.774	1.542	6.148	71.039	0.085	0.198	2.850
Average	7.824	1.551	5.381	73.908	0.143	0.199	2.905

Table 7 - Changes in the heat absorbed in the evaporator (at T_C) and released in the condenser (T_H) at various circulation rates. [TXV]

Flow (lbm/min)	**Amount** (inch)	**Compressor Power** (Btu/min)	**T_C** (°R)	**T_H** (°R)	**Q_C** (Btu/min)	**Q_H** Btu/min
1.718	8.800	51.752	515.170	582.233	85.204	96.864
1.718	9.100	51.752	509.170	579.109	85.121	96.939
1.729	9.200	52.036	517.170	588.808	82.451	94.691
1.750	9.300	52.889	520.170	579.877	82.695	93.468
	Average		515.420	582.507	83.868	95.491

Table 8 - Changes in enthalpy and pressure values at strategic points in the system, under various flow rates (see Appendix C, Figure _). [TXV]

Flow (lbm/min)	**h'_1** (Btu/lbm)	**h'_2** (Btu/lbm)	**h^s_2** (Btu/lbm)	**h'_3** (Btu/lbm)	**h'_4** (Btu/lbm)	**P_1** (psig)	**P_2** (psig)
1.718	82.645	89.432	88.796	33.050	33.050	61.375	176.375
1.718	82.620	89.499	88.566	33.074	33.074	54.932	169.932
1.729	82.735	89.815	89.262	35.043	35.043	62.632	191.632
1.750	83.122	89.277	88.623	35.879	35.879	67.121	172.121

Table 9 - Changes in certain coefficients of performances and efficiencies in the system, under various circulation rates. [TXV]

Flow (lbm/min)	**COP_{max}**	**COP_{actual}**	**COP_{fluid}**	**RC**	**η_{comp}**	**η_{cycle}**	**CR**
1.718	7.682	1.646	7.307	85.204	0.204	0.214	2.874
1.718	7.280	1.645	7.202	85.121	0.197	0.226	3.093
1.729	7.219	1.585	6.736	82.451	0.217	0.219	3.060
1.750	8.712	1.564	7.676	82.695	0.182	0.179	2.564
Average	7.723	1.610	7.230	83.868	0.200	0.210	2.898

Appendix B

Sample Calculations

The first thing to establish is that all measurements taken in the lab were taken with different units for different pieces of instrumentation. All the values used to do the calculations for this report have been changed accordingly into Metric Units for clarity and ease of use.

The following conversion factors used are listed as follows:

Pressure:

1 atm $= 101.325kPa = 760torr = 760mmHg = 29.9212inHg = 14.696\,psi$

1 psi $= 6.89kPa = 6890Pa$

psia = psig + 14.696 psi

Temperature:

$$K = °C + 273.15 = °R/1.8 \approx °C + 273 \qquad °C = (°F - 32)/1.8$$
$$°R = °F + 459.67 \approx °F + 460 = 1.8K \qquad °F = 1.8°C + 32$$

Universal Gas Constant:

$$R = 8.314\frac{J \cdot Pa}{mol \cdot K} = 0.08206\frac{L \cdot atm}{mol \cdot K} = \frac{1.987\,Btu}{lbmol \cdot °R}$$

Power:

1 hp $= 0.746kW = 550\, {ft \cdot lbf}/{s} = 33{,}000\, {ft \cdot lbf}/{\min} = 2545\, {Btu}/{h}$

1 W $= 1.34 * 10^{-3}hp = {J}/{s} = {N \cdot m}/{s} = 9.49 \cdot 10^{-4}\, {Btu}/{s}$

Note: All sample calculations that follow are based on the first capillary run at the first flow rate. Calculations for the TXV mode are identical, and all values that are calculated can be found in Tables [] (Appendix A – Tabulated Results)

Calculation of Performance for an Ideal and Actual Refrigeration System

1. To calculate the coefficient of performance (COP) for ideal (Carnot) efficiency system, the points at where the temperatures for cold and hot reservoirs are evaluated, need to be defined. For this experiment the constant isotherm at the point were the cycle crosses the T-S curve to the ideal point of the condenser outlet is used for the hot reservoir temperature, T_H. While the constant isotherm at the inlet of the evaporator to the outlet point that crosses the T-S curve is used for the cold reservoir temperature, T_C. The performance for the ideal system is shown below:

$$COP_{\max} = \left(\frac{\dot{Q}_C}{W}\right)_{ideal} = \frac{\dot{Q}_C}{\dot{Q}_H - \dot{Q}_C} = \frac{T_C}{T_H - T_C}$$

$$Where\ T_C = 513.17°R;\ T_H = 574.09°R \qquad \text{Eq. 1}$$

$$\therefore \quad COP_{\max} = \frac{513.17°R}{574.09°R - 513.17°R} = 8.42$$

2. The actual coefficient of performance (COP_{actual}) for the vapour pressure system that was actually tested is the ratio of the actual refrigerating effect and work expanded by the compressor. Using Eq. 2, this can be determined as follows:

$$COP_{actual} = \frac{\dot{Q}_C}{W} = \frac{refrigeration\ capacity}{power\ to\ the\ compressor} \qquad \text{Eq. 2}$$

To calculate the performance for the actual system, the refrigeration capacity needs to be calculated first by Eq. 4. as follows:

$$\dot{Q}_C = \dot{m}_f \left(h_1^{'} - h_4^{'}\right) \qquad Where \;\; \dot{m}_f = 1.718 \, {}^{lbm}\!/_{\min} \qquad \text{Eq. 4}$$

$$\therefore \; \dot{Q}_C = 1.718 \, {}^{lbm}\!/_{\min} \left(82.61 \, {}^{Btu}\!/_{lbm} - 33.11 \, {}^{Btu}\!/_{lbm}\right) = 85.04 \, {}^{Btu}\!/_{\min}$$

Substituting the refrigeration capacity back into Eq. 1 and converting the power term to appropriate units, yields the actual coefficient of performance as follows:

$$COP_{actual} = \frac{\dot{Q}_C}{W} = \frac{85.04 \, {}^{Btu}\!/_{\min}}{(940 - 855) watts * 9.49 * 10^{-4} \, {}^{Btu}\!/_{s} * 60 {}^{s}\!/_{\min}} = 1.75$$

Note: The power consumption for the compressor is found by subtracting the power at each fan setting, from the total power output that was read. For the capillary runs the fan power was on high-high setting for all five flow rates and gave corresponding power of 85 watts. For the TXV mode the setting for the fan changed for all four flow rates and corresponded to high-high, high-medium, medium-high, and medium-medium settings respectively.

3. The Coefficient of performance of the fluid is determined by taking the ratio of the specific enthalpies of the working fluid at different stages of the refrigeration system. The COP_{fluid} can be obtained by using Eq. 3 as follows:

$$COP_{fluid} = \frac{h_1^{'} - h_4^{'}}{h_2^{'} - h_1^{'}} \qquad \text{Eq. 3}$$

$$COP_{fluid} = \frac{(82.61 - 33.11)\,{}^{Btu}\!/_{lbm}}{(88.97 - 82.61)\,{}^{Btu}\!/_{lbm}} = 7.78$$

4. Next the efficiency of the compression stage must be calculated. This is done by using Eq. 5 from the lab handout and is shown as follows:

$$\eta_{comp} = \frac{\dot{m}_f \left(h_2^s - h_1^{'}\right)}{\dot{W}} \qquad \text{Eq. 5}$$

$$\eta_{comp} = \frac{1.718\,{}^{lbm}\!/_{\min}\,(88.19 - 82.61)\,{}^{Btu}\!/_{lbm}}{(940 - 85)watts * 9.49 \cdot 10^{-4}\ {}^{Btu}\!/_{s} * {}^{60s}\!/_{\min}} = 0.197$$

5. The efficiency of the overall refrigeration cycle can be determined by taking the ratio of the actual coefficient of performance over the ideal (Carnot) coefficient of performance and is calculated below by using Eq. 6:

$$\eta_{cycle} = \frac{COP_{actual}}{COP_{\max}} = \frac{1.75}{8.42} = 0.208 \qquad \text{Eq. 6}$$

6. Finally the compression ratio can be found by dividing the pressure at point two which is the compressor outlet, over the pressure at point one – the compressor inlet. This calculation is shown by use of Eq. 7 below as follows:

$$CR = \frac{p_2^{'}}{p_1^{'}} = \frac{159.17\,psia}{55.17\,psia} = 2.88 \qquad \text{Eq. 7}$$

Appendix C

Related Figures

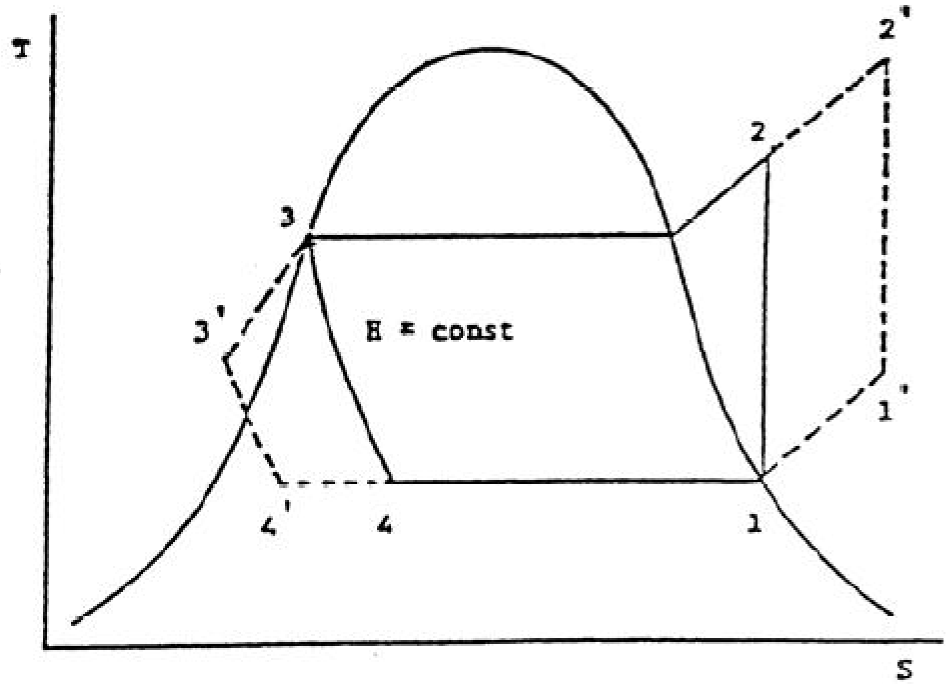


Figure 2 – A T-S diagram of the various steps carried out in the refrigeration process and the 2-phase envelope.

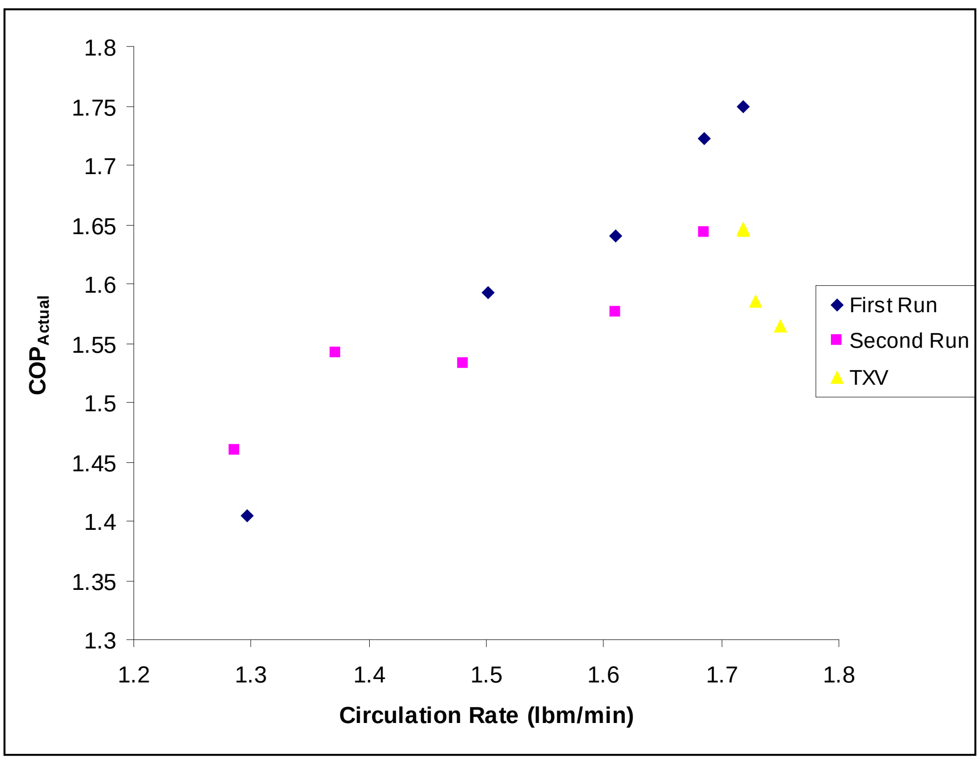


Figure 3 – Changes in the coefficient of performance of the refrigeration system with respect to the circulation rate of the refrigerant, on all 3 selected runs.

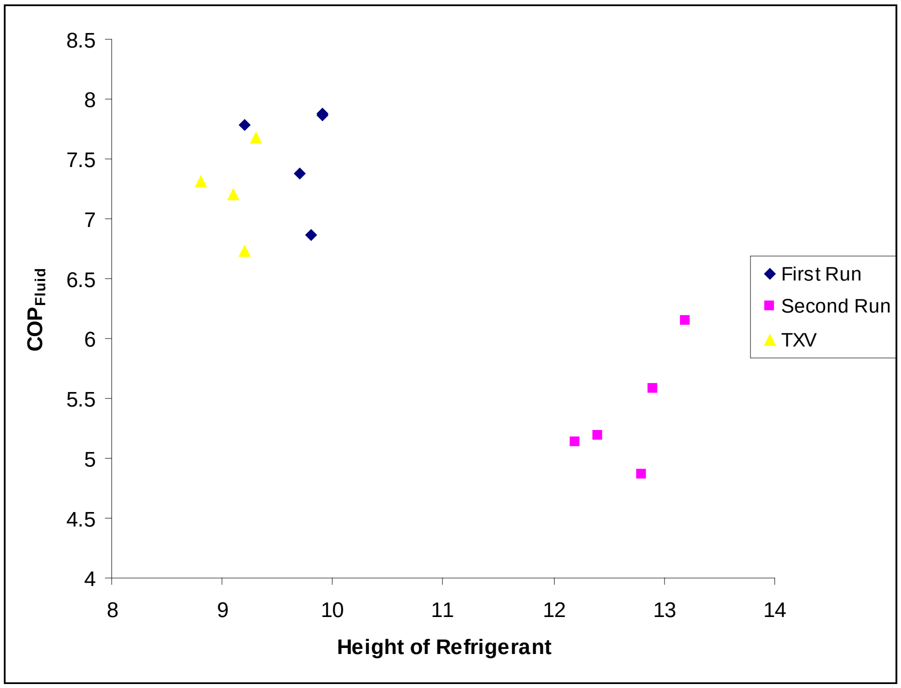
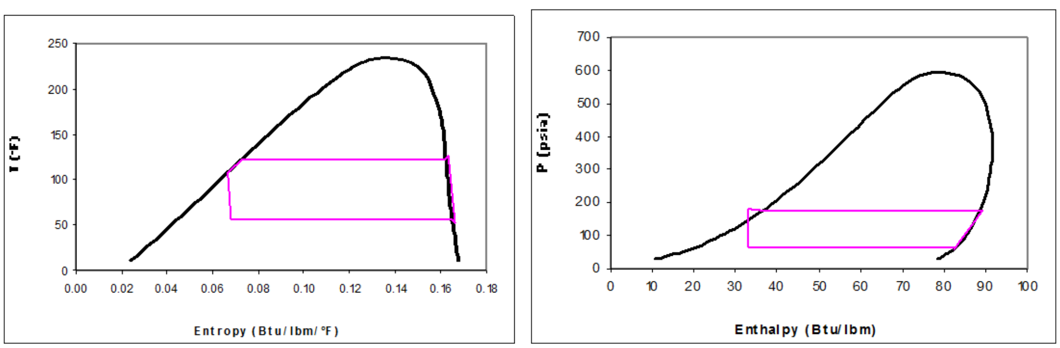
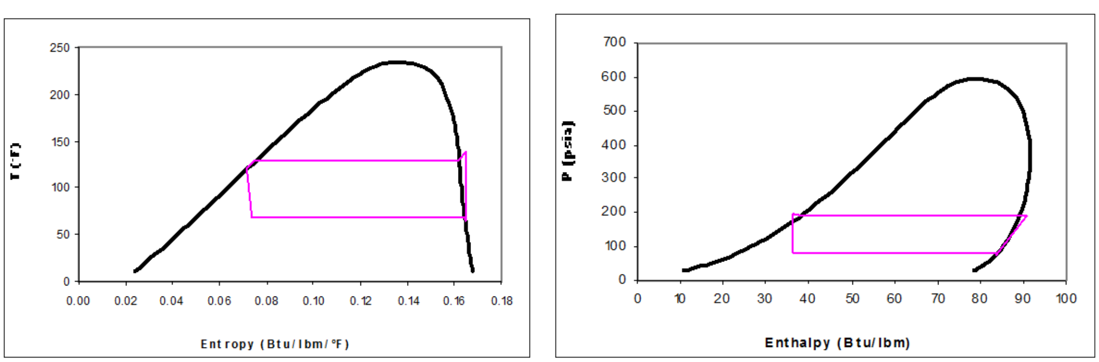


Figure 4 - Changes in the coefficient of performance of the refrigerant itself with respect to the amount of refrigerant used, on all 3 selected runs.



Figures 5 and 6 - T-S and p-H plots of the refrigerant in the first run of the TXV mode, at a room temperature of 77.5ºF.



Figures 7 and 8 - T-S and p-H plots of the refrigerant in the first run of the TXV mode, at a room temperature of 90ºF.

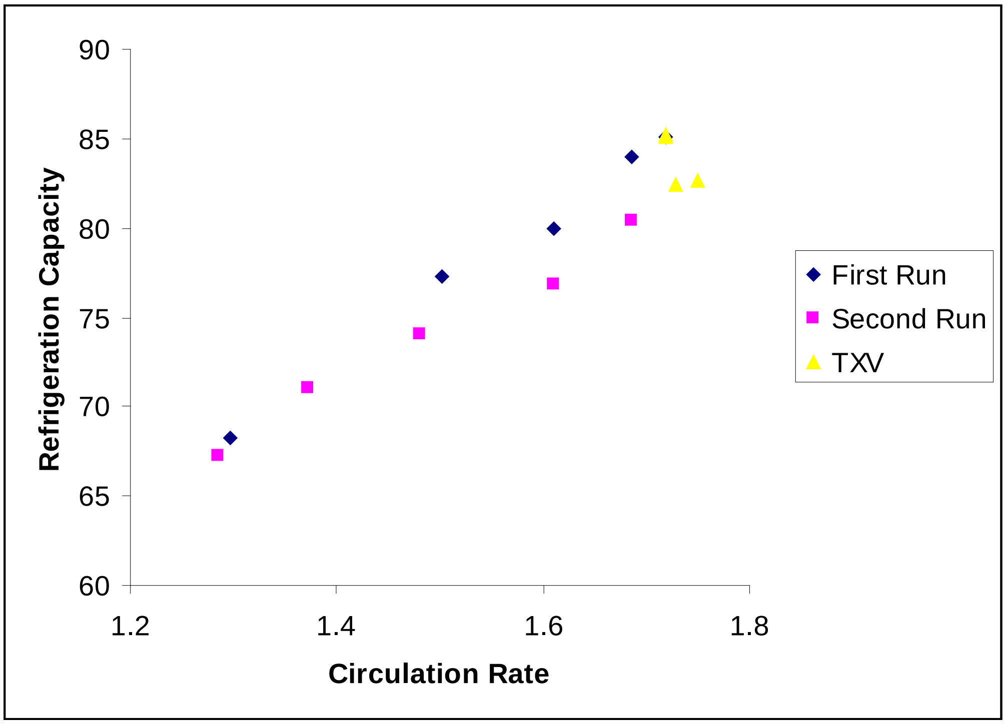


Figure 9 - Changes in the refrigeration capacity of the system with respect to the circulation rate of the refrigerant, on all 3 selected runs.

Appendix D

Raw and Nonessential Data

Table 10 - Condenser power at at different condenser and evaporator fan settings.

S-5 Condenser	S-6 Evaporator (watts)		
	High	**Medium**	**Low**
High	85	75	70
Medium	80	55	45
Low	65	50	40

Table 11 - Measured values at the various strategic points in the refrigeration system under different. [First Capillary Run]

	Rotameter Reading	Refridgerant Level	Temperature *(°F)*	Pressure *(psig)*	Power *(watts)*
Point 1	15	9.2	48	50	940
Point 2	15	9.2	118	156	940
Point 3	15	9.2	94	137	940
Point 4	15	9.2	52	59	940
Point 5	15	9.2	45	55	940
Point 1	14.7	9.7	47	50	942
Point 2	14.7	9.7	121	162	942
Point 3	14.7	9.7	92	144	942
Point 4	14.7	9.7	51	56	942
Point 5	14.7	9.7	42	53	942
Point 1	14	9.8	45	48	942
Point 2	14	9.8	124	167	942
Point 3	14	9.8	92	149	942
Point 4	14	9.8	50	55	942
Point 5	14	9.8	41	50	942
Point 1	13	9.9	48	39	938
Point 2	13	9.9	125	169	938
Point 3	13	9.9	88	150	938
Point 4	13	9.9	45	50	938
Point 5	13	9.9	53	48	938
Point 1	11.1	9.9	54	41	940
Point 2	11.1	9.9	130	164	940
Point 3	11.1	9.9	86	154	940
Point 4	11.1	9.9	42	46	940
Point 5	11.1	9.9	60	49	940

Table 12 - Measured values at the various strategic points in the refrigeration system under 5 different sub-runs. [Second Capillary Run].

	Rotameter Reading	**Refridgerant Level**	**Temperature** *(°F)*	**Pressure** *(psig)*	**Power** *(watts)*
Point 1	15	9.2	51	54	940
Point 2	15	9.2	140	159	940
Point 3	15	9.2	102	140	940
Point 4	15	9.2	54	60	940
Point 5	15	9.2	45	55	940
Point 1	14.7	9.7	47	50	942
Point 2	14.7	9.7	136	156	942
Point 3	14.7	9.7	100	138	942
Point 4	14.7	9.7	52	56	942
Point 5	14.7	9.7	43	53	942
Point 1	14	9.8	55	48	942
Point 2	14	9.8	152	160	942
Point 3	14	9.8	96	141	942
Point 4	14	9.8	49	54	942
Point 5	14	9.8	54	50	942
Point 1	13	9.9	63	43	938
Point 2	13	9.9	154	156	938
Point 3	13	9.9	92	139	938
Point 4	13	9.9	43	47	938
Point 5	13	9.9	65	45	938
Point 1	11.1	9.9	65	45	940
Point 2	11.1	9.9	146	143	940
Point 3	11.1	9.9	95	135	940
Point 4	11.1	9.9	45	50	940
Point 5	11.1	9.9	66	46	940

Table 13 - Measured values at the various strategic points in the refrigeration system under 4 different sub-runs: High-High, Medium-High, High-Medium, Medium-Medium. [TXV Run]

	Rotameter Reading	**Refridgerant Level**	**Temperature** *(°F)*	**Pressure** *(psig)*	**Power** *(watts)*
Point 1	15	8.8	50	53	995
Point 2	15	8.8	125	170	995
Point 3	15	8.8	94	152	995
Point 4	15	8.8	54	58	995
Point 5	15	8.8	45	55	995
Point 1	15	9.1	48	51	985
Point 2	15	9.1	124	168	985
Point 3	15	9.1	94	149	985
Point 4	15	9.1	48	56	985
Point 5	15	9.1	45	56	985
Point 1	15.1	9.2	51	54	995
Point 2	15.1	9.2	131	185	995
Point 3	15.1	9.2	102	167	995
Point 4	15.1	9.2	56	60	995
Point 5	15.1	9.2	47	57	995
Point 1	15.3	9.3	55	58	985
Point 2	15.3	9.3	123	165	985
Point 3	15.3	9.3	105	145	985
Point 4	15.3	9.3	59	63	985
Point 5	15.3	9.3	50	60	985

Appendix E

References

1. M.J. Moran and H.N. Shapiro, "*Fundamentals of Engineering Thermodynamics, 3rd Ed.*" John Wiley & Sons, Inc., New York, NY, 1995

2. N. de Nevers, *"Fluid Mechanics for Chemical Engineers, 3rd Ed."* New York, NY; McGraw-Hill, 2005.

3. Smith J.M., and Van Ness H. J, *"Introduction to Chemical Engineering Thermodynamics, 7th Ed."* New York, NY; McGraw-Hill, 2005

4. CHG3122 Winter 2007 Lab Handout

Refrigeration

Chemical Engineering Practice

CHG3122

Julie Faubert-Smith

Gabriel Potvin

To: Dr. Zhang
From: Julie Faubert-Smith, Gabriel Potvin, Group 4
Date: March 12th 2008
Subject: CHG3122, Refrigeration Experiment

The purpose of the present experiment was the calculation of refrigeration capacity and coefficients of performance for the Scott Air Conditioning and Refrigeration Education system using dichloro-difluoro-methane (CCl_2F_2) as the working fluid for different rates of refrigerant circulation and heat load. The system performance in normal capillary and thermostatic expansion will so be compared.

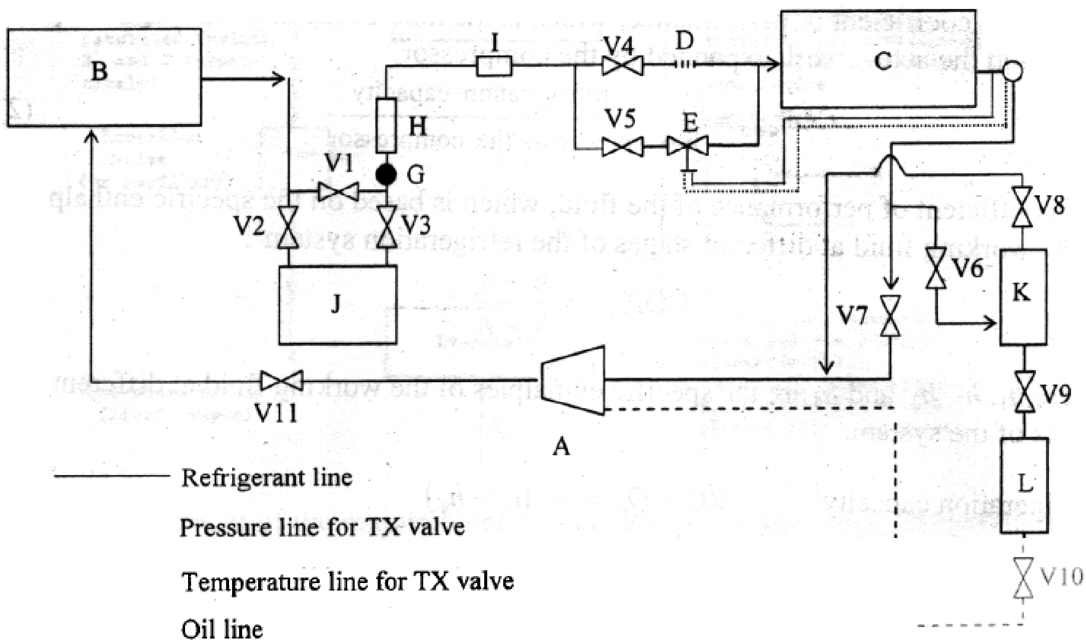
Pressure and temperature at 5 strategic locations and wattage requirements for both fans at all settings are recorded prior to beginning the experiment. In normal capillary mode, the rotameter reading, refrigerant level, wattage reading and pressure and temperature at the five locations are recorded for 5 different refrigerant flow rates. The experiment is repeated with a decrease in refrigerant. After every change in flow rate, a three minute time period is allowed prior to taking readings in order to allow the system to reach steady state. The same readings are taken in thermostatic expansion mode for four different fan settings. Note that the working fluid properties are not those of pure CCl_2F_2 since it is mixed with oil, and that the appropriate corrections are performed automatically by the provided Excel spreadsheet. Since data was collected at a limited amount of points, to ensure the validity of results, more readings should be taken at each run condition.

Results generally reflect theoretical expectations. Although the assumption of isenthalpic and isentropic expansion in capillary and TXV modes respectively is adequate for the present experiment, actual expansion is not ideal. In capillary mode, the refrigeration capacity (Q_c), power consumed by the compressor (work) and heat loss increase with the refrigerant's flow rate. When the total amount of refrigerant is reduced, the fluid performance and the compression efficiency increase. The condenser (S-5) and evaporator (S-6) fan settings affect the performance of the TXV system. The heat loss from the compressor is at its maximum when the condenser is running on medium and evaporator on high. The TXV system's efficiency (compression and cycle) and compression ratio are at their maximum when both the condenser and evaporator are running on high. The compression work is relatively constant in both systems with a value of about 40-50 Btu/min in capillary mode and 45-55 Btu/min for TXV mode. TXV therefore has a better refrigeration performance as expected, although higher values for the coefficients of performance were expected.

Gabriel Potvin   Julie Faubert-Smith

Equipment and Procedure

The experiment is performed using the Scott Air Conditioning and Refrigeration Education system, which operates in normal capillary or thermostatic explansion modes, using dichloro-difluoro-methane (CCl_2F_2) as the working fluid. The system is set up as depicted in the following diagram (taken from the protocol provided by the laboratory). Note that pressure and temperature gauges are placed at five strategic locations and are not displayed on the diagram.



With system components:

A – Compressor	G – Moisure and Liquid Indicator
B – Condenser	H – Calibrated Rotameter
C – Evaporator	I – Drier
D – Capillary	J – Liquid Refrigerant Receiver Tank
E – Thermostatic Expansion Valve	K – Liquid Accumulator Tank
F – Temperature sensor	L – Oil Storage Tank

Pressure and temperature at the 5 locations and wattage requirements for both fans at all settings are recorded prior to beginning the experiment. In normal capillary mode, the rotameter reading, refrigerant level, wattage reading and pressure and temperature at the five strategic locations are recorded for 5 different flow rates. After every change in flow rate, a three minute time period is allowed prior to taking readings in order to allow the system to reach steady state. The same readings are taken in thermostatic expansion mode for four different fan settings. The experiment is then repeated a final time in normal capillary mode with a decreased quantity of refrigerant. Note that the working fluid properties are not those of pure CCl_2F_2 since it is mixed with oil, and that the appropriate corrections are performed automatically by the provided Excel spreadsheet.

Results

All parameters were calculated using the provided Excel spreadsheet. Details regarding these calculations are found in the Excel Program Detail found at the end of the experimental protocol.

1. The temperature at point 4 (after the expansion valve) is used as a basis for the correction of temperatures and pressures at other points. This correction is necessary in order to account for the effect of oil on the refrigerant properties. The saturation pressure is calculated at point 4 and it is used to correct the pressures, enthalpies and entropies at all other points (necessary for example for the calculation of T_H and T_C which use the pressures).
2. The quality at point 4 is calculated using the generic equation $M = (1-x^v)M^l + x^vM^v$, where M can represent in the present case H or S, the subscripts v and l indicated the vapor or liquid states respectively, and x is the fraction or quality of the two-phase system. In the case of the capillary tube, the process is assumed to be isenthalpic, and constant H is used to determine the thermodynamic data using the compiled tables. In the case of the TXV, the process is assumed isentropic, and hence a constant value of S is used while reading the thermodynamic tables. Although these assumptions provide a reasonably good approximation of actual behaviour, in reality these processes are not completely isenthalpic or isentropic.
3. The thermodynamic graphs for a single run are shown for both the capillary and TXV experimental setups in the present report. The graphs drawn for all other runs are found in the appendix, and all unusual data will be discussed.
4. T-S, T-H and P-H diagrams are found in Figure 1 for the first run condition of the capillary experiment. These diagrams reflect the expected profiles for vapor-compression refrigeration cycles. The experimental refrigeration cycle is drawn in pink and the temperature or pressure profiles are drawn in black.
5. T-S, T-H and P-H diagrams are found in Figure 2 for the first run condition of the TXV experiment. These diagrams reflect the expected profiles for vapor-compression refrigeration cycles in TXV mode. The experimental refrigeration cycle is drawn in pink and the temperature or pressure profiles are drawn in black.
6. Work and refrigeration capacity are plotted in figure 3, maximum, fluid and actual coefficients of performance are plotted in figure 4, compression and cycle efficiency are plotted in figure 5 and compression ratio is plotted in figure 6 all vs. mass flow rate for both runs of the capillary experiment at the 5 run conditions. Note that refrigerant was removed before the second run of the capillary experiment. An outlying point is observed at 1.4588 lbm/min for the first run in the COP_{max}, compression efficiency and compression ratio plots.
7. In figures 3-6 while taking the runs individually, there seems to be a slight increase in refrigeration performance as the circulation rate of the refrigerant increases. This is expected given that the refrigeration capacity increases with the quantity of refrigerant available within a time interval, and more heat can be removed from the system for a same absolute quantity of refrigerant. Due to the small number of data points collected as well as the presence of outliers, results have limited reliability.
8. While comparing both runs in figures 3-6, the only difference observed is the higher compression efficiency in the second run, which is expected given the fact

that the system contains less refrigerant. The cycle efficiency remains the same as expected and the overall performance of the refrigeration system remains the same. Again, given the small number of points, results have limited reliability.

9. Work and refrigeration capacity are plotted in figure 7, maximum, fluid and actual coefficients of performance are plotted in figure 8, compression and cycle efficiency are plotted in figure 9 and compression ratio is plotted in figure 10 all vs. the four fan settings during the TXV experiment.
10. Based on figures 7-10, the performance is generally unaffected by the fan settings, although a decrease in compression ratio is observed when the power of either or both fans is reduced. Fluctuations in efficiency are observed, which are due to the fluctuating level of refrigerant in the system. Although in theory a slight reduction in performance is expected when the fan power is reduced the difference is probably not discernible with the present experimental setup. Again, given the small number of data points, reliability of results is limited.
11. Since it was not insulated, heat loss from the compressor occurred. Rate of heat loss due to the compressor is plotted vs. flow rate in figure 11 for both experiments in normal capillary mode. Heat loss is plotted vs. the fan settings for the experiment in TXV mode in figure 12. The same limits on reliability of results apply, and the outliers observed in the capillary experiment results cannot be judged insignificant without further experiments.
12. The experimental T-S and P-H plots for the first run condition of the TXV experiments are compared to the theoretical plots obtained following an increase of 10ºF in ambient temperature in figure 13. As the temperature increases, the thermodynamic behaviour of the refrigeration system becomes more different from the ideal behaviour. This is expected given the higher quantity of energy to be extracted from the system. A completely reversible system (Carnot engine) is ideal, and the profile tends towards this ideal situation as the difference between T_H and T_C decreases. By increasing ambient temperature, this interval is increased.

Discussion

It should be noted that although experimental results are generally in accordance with theoretical expectations in the present experiment, given the small number of data points collected, trends discussed in this discussion may not be significant. More data points should be collected to ensure significance of results.

The T-H, T-S and P-S diagrams provide information on the refrigeration cycle's behaviour and profiles are in accordance with theoretical expectations. In the present refrigeration experiment, two different mechanisms of expansion were used: normal capillary, in which expansion is isenthalpic, and using a thermostatic expansion valve (TXV), which is considered isentropic. In figures 1 and 2 it is apparent that the system does not behave ideally, as a deviation from isentropic or isenthalpic behavior is observed. This is expected as the assumptions of isenthalpic and isentropic expansion apply to ideal systems, which are extremely difficult, if not impossible, to reproduce experimentally (Perry, 2997). However, given that the divergences from ideal behaviour are negligible for the current experiment, the assumptions of isenthalpic and isentropic behaviour are used for all calculations.

The performance of the refrigeration system is characterized by measuring parameters such as Carnot, actual and fluid coefficients of performance, refrigeration capacity, compression and cycle efficiency as well as the compression ratio. These parameters allow given refrigeration cycles, in this case the normal capillary and TXV cycles, to be compared. The effect of changing parameters such as refrigerant flow rate, refrigerant quantity (capillary) and fan settings (TXV) were also evaluated. The effect of each of these parameters on the refrigeration cycle performance will be further discussed.

In normal capillary mode, it was found that the performance of the refrigeration cycle increases as the flow rate of the refrigerant increases. This is expected considering that more heat can be removed from the system over a given time interval due to the increased flow of refrigerant. In the second run in capillary mode, some refrigerant was removed, which increased the compression efficiency and reduced the coefficient of performance of the fluid. This was also expected given that the rate of heat loss in the compressor is lower when less refrigerant is circulated through the system and hence more energy is directed into work, hence the higher compression efficiency. The rate of heat loss increases with the flow rate of the refrigerant. The overall cycle efficiency does not change with the refrigerant flow rate or amount of refrigerant, which again is expected. The heat capacity and compression work seem to increase slightly with the refrigerant flow rate, but are not affected by the total amount of refrigerant in the system. Compression ratios in both runs are very similar, however an outlying point is observed at around 1.45 lbm/min during the first run. To determine the validity of this point the experiment should be repeated.

In TXV mode, the effects of four different fan settings on the refrigeration cycle were evaluated. When the fans S-5 (condenser) and S-6 (evaporator) are both operating on high, the cycle and compression efficiencies are at their maximum, which is expected since with higher air flow rates, more heat transfer is possible. The compression efficiency is at it's lowest when the condenser is on high and the evaporator is at medium, and the cycle efficiency is at its minimum when both the condenser and evaporator are operating at medium setting. This is also expected. When the condenser fan is set to high, the condensation (vapour to liquid) is more efficient, and when the evaporator is set to a lower setting, evaporation (liquid to vapour) is less efficient, which results in a higher proportion of liquid in the working fluid. Since for the present experiment liquids can be assumed incompressible, it is expected that compression efficiency is low. When both fans are set to the lower setting, less air is circulated, which leads to less heat transfer, hence the lowest cycle efficiency (de Nevers, 2005).

The compression ratio in TXV mode is at its maximum when both the condenser and evaporator operate on high, and is reduced significantly when the power of either or both of the fans is reduced. The heat lost from the compressor is constant for each fan setting at about 28 Btu/min, except when the condenser (S-5) is set to medium, and the evaporator (S-6) is on high, there is a considerable increase of heat loss to about 36 Btu/min. The experiment should be repeated to confirm the validity of this data point. The heat capacity and compression work do not seem to be affected by the fan settings.

Using the calculated parameters described, the TXV and capillary modes can be compared. In capillary mode, the compression and cycle efficiencies, as well as the compression rations are slightly higher than in TXV mode. The TXV has a higher refrigeration capacity of 90-95 Btu/min compared to 70-82 Btu/min in capillary mode, using slightly more power (compression work), which varies from approximately 40-50 Btu/min in capillary mode and 45-55 Btu/min for TXV mode. Actual coefficients of performance are very similar, although higher coefficients were expected for the system in TXV mode. As expected, the system operating in TXV mode has slightly better performance (based on the refrigeration capacity).

In both capillary and TXV modes, the required fluid compression work is a factor that must be considered when choosing a refrigerant in order to minimize the system's external power use. The compression work is a function of the fluid's specific enthalpies and a reduced compression work shows that the refrigerant is adequate.

Because of the fluctuating level of refrigerant in the cycle, there is a significant amount of variation in the results obtained throughout the entire experiment. Other sources of variation include collecting the data before the system reaches steady-state at new conditions, lack of precision in the measurements and the usual experimenter error. This data fluctuation could be misleading and lead to false conclusions about the system's behavior. More data points should be collected at each run condition to verify the validity of results.

Conclusions/Recommendations

- Expansion in normal capillary and TXV modes is not completely isentropic and isenthalpic respectively as would be expected in an ideal system. However, the divergences are small enough that the approximation can be made.

- In capillary mode, the refrigeration capacity (Q_c), power consumed by the compressor (work) and heat loss increase with the refrigerant's flow rate. When the total amount of refrigerant is reduced, the fluid performance and the compression efficiency increase.

- The condenser (S-5) and evaporator (S-6) fan settings affect the performance of the TXV system. The heat loss from the compressor is at its maximum when the condenser is running on medium and evaporator on high. The TXV system's efficiency (compression and cycle) and compression ratio are at their maximum when both the condenser and evaporator are running on high.

- The compression work is relatively constant in both systems with a value of about 40-50 Btu/min in capillary mode and 45-55 Btu/min for TXV mode.

- TXV mode has a greater refrigeration capacity of 90-95 Btu/min compared to 70-82 But/min for the capillary mode. Although TXV has a better refrigeration performance, due to it's higher cost in capital and maintenance (compared to the capillary mode), costs and benefits, as well as requirements for a given process, should be evaluated prior to system design.

- In order to ensure the validity of results, more data points should be collected at each run condition and longer time intervals should be used to ensure system achieves steady-state.

References

de Nevers, N. “Fluid Mechanics for Chemical Engineers” Third ed., McGraw-Hill, New York NY, 2005

Perry, R.H. and Green, D.W. “Perry’s Chemical Engineers’ Handbook” Seventh ed., McGraw-Hill, New York, NY, 1997.

Appendix

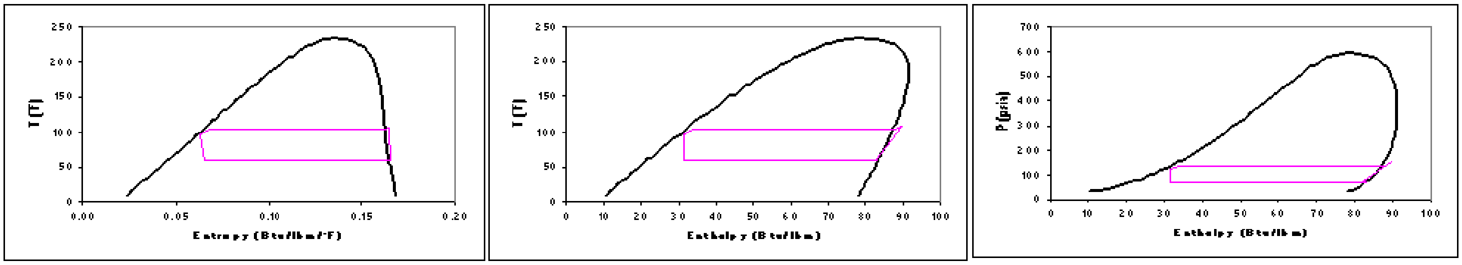


Figure 1. Experimental vapor-compression refrigeration cycle for the first run condition of the capillary experiment. The refrigeration cycle profile, in pink, is drawn on T-S, T-H and P-H diagrams.

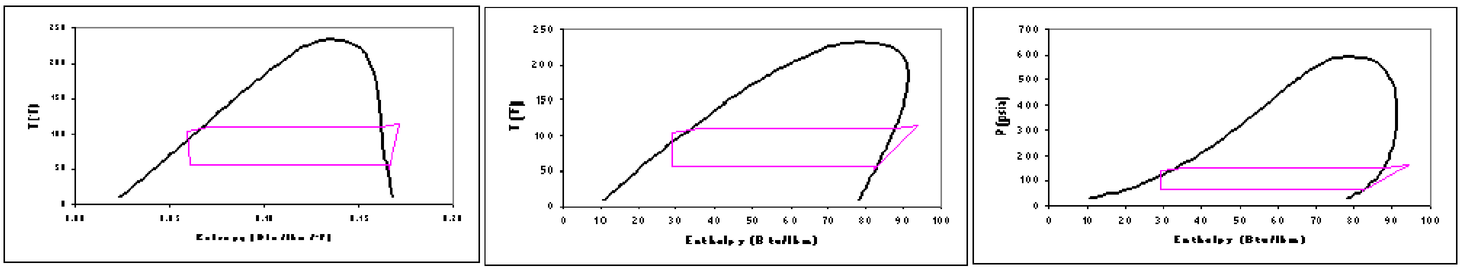


Figure 2. Experimental vapor-compression refrigeration cycle for the first run condition of the TXV experiment. The refrigeration cycle profile, in pink, is drawn on T-S, T-H and P-H diagrams.

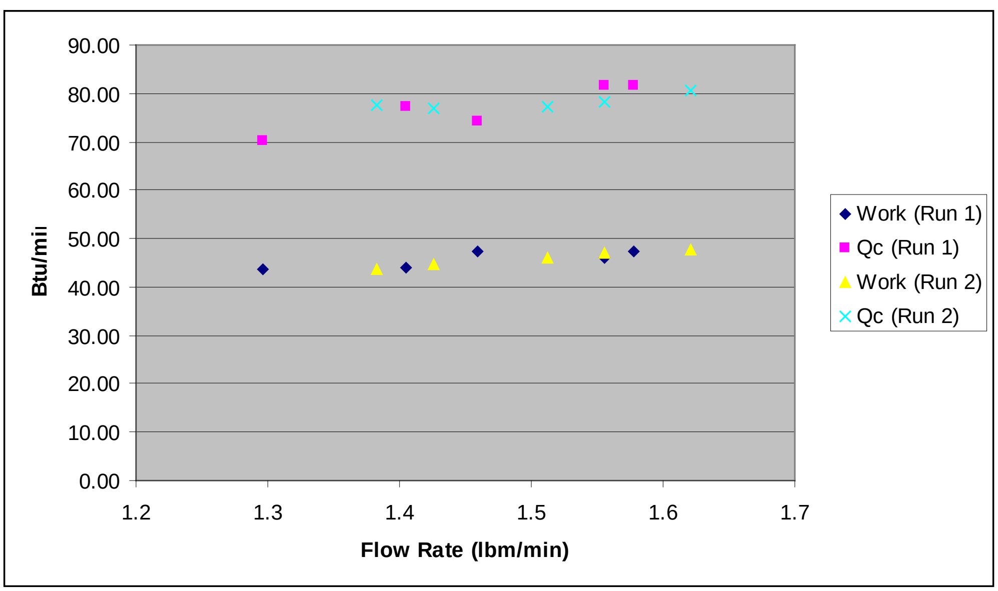


Figure 3. Compression work and refrigeration capacity plotted vs. mass flow rate for both runs of the capillary experiment at the five run conditions.

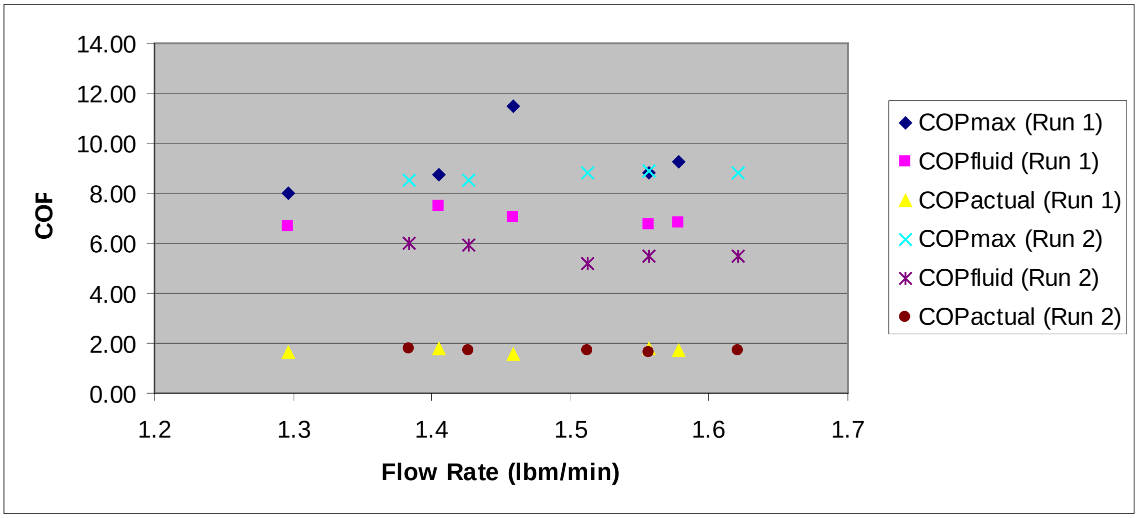


Figure 4. Maximum, fluid and actual coefficient of performance (COP) plotted vs. mass flow rate for both runs of the capillary experiment at the five run conditions.

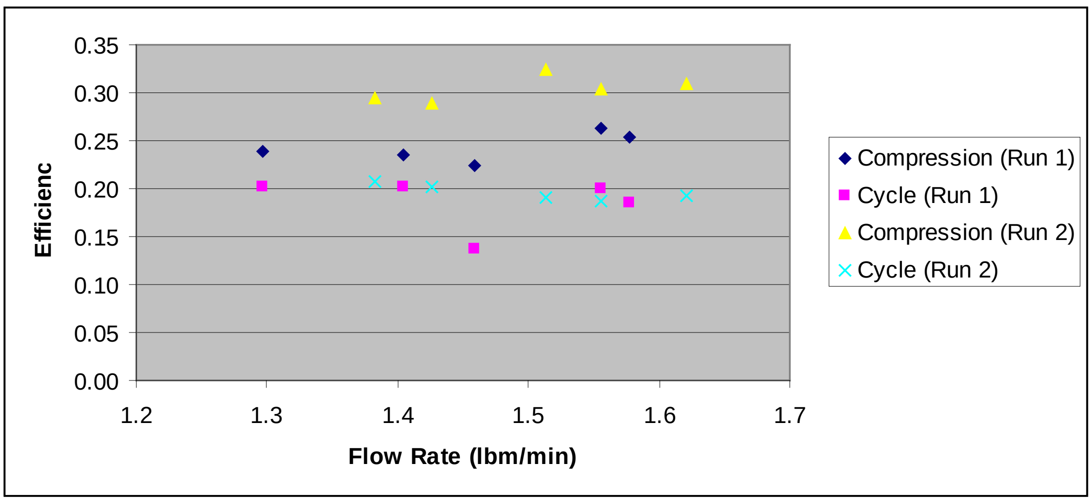


Figure 5. Compression and cycle efficiency plotted vs. mass flow rate for both runs of the capillary experiment at the five run conditions.

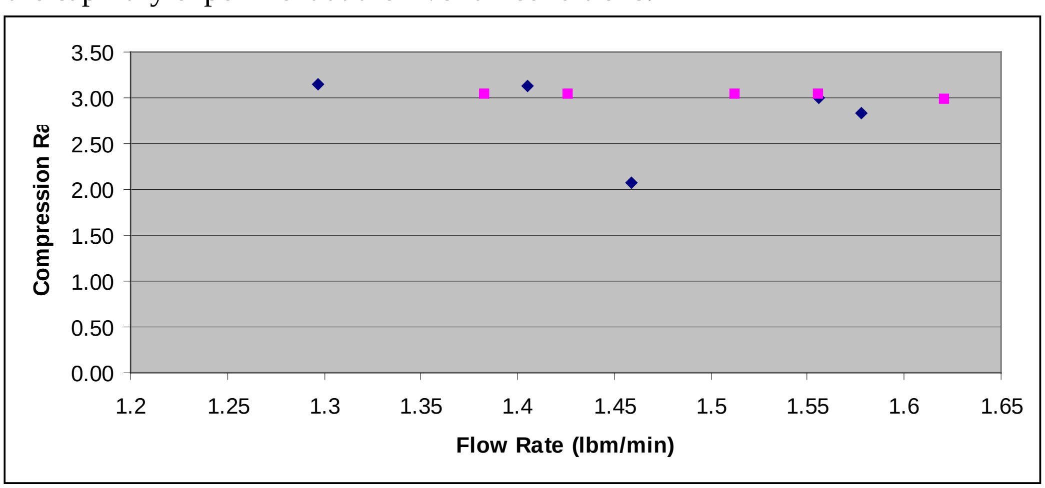


Figure 6. Compression ratio plotted vs. mass flow rate for both runs of the capillary experiment at the five run conditions.

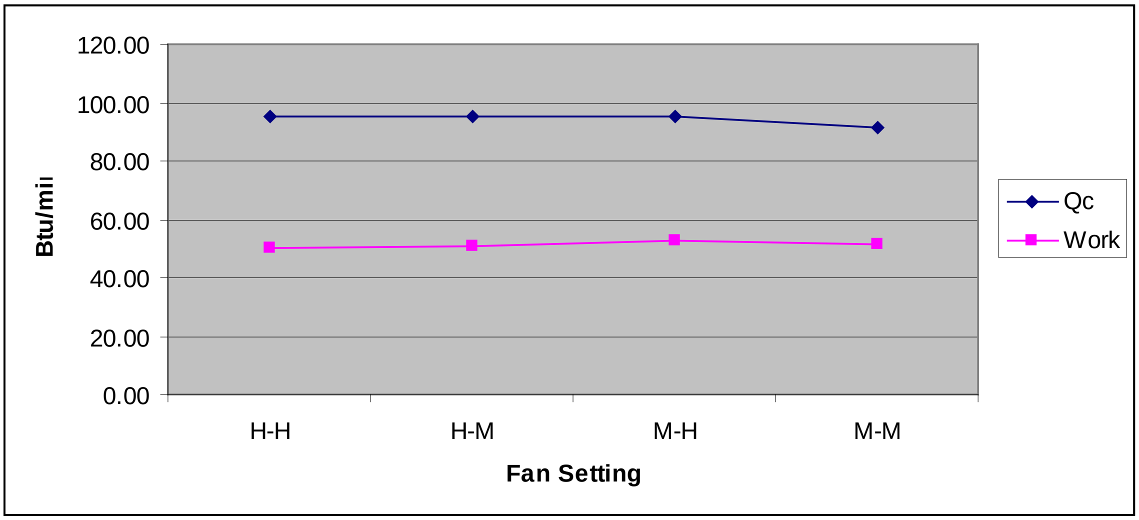


Figure 7. Work and refrigeration capacity plotted vs. each fan setting during the TXV experiment. The first letter of the labels on the x axis represents the setting for fan S-5 (condenser) and the second letter represents the setting for fan S-6 (evaporator).

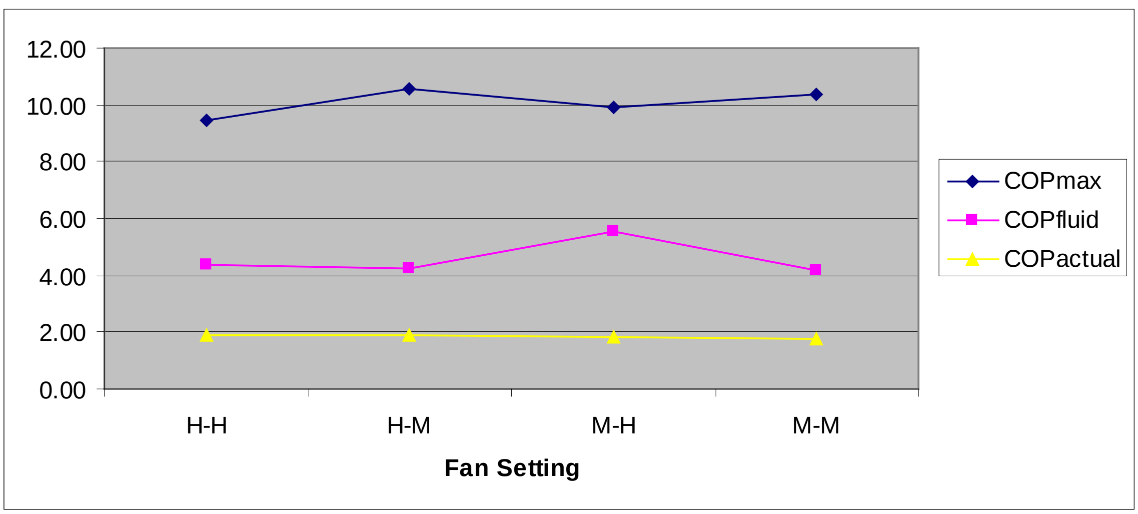


Figure 8. Maximum, fluid and actual coefficients of performance (COP) plotted vs. each fan setting during the TXV experiment. The first letter of the labels on the x axis represents the setting for fan S-5 (condenser) and the second letter represents the setting for fan S-6 (evaporator).

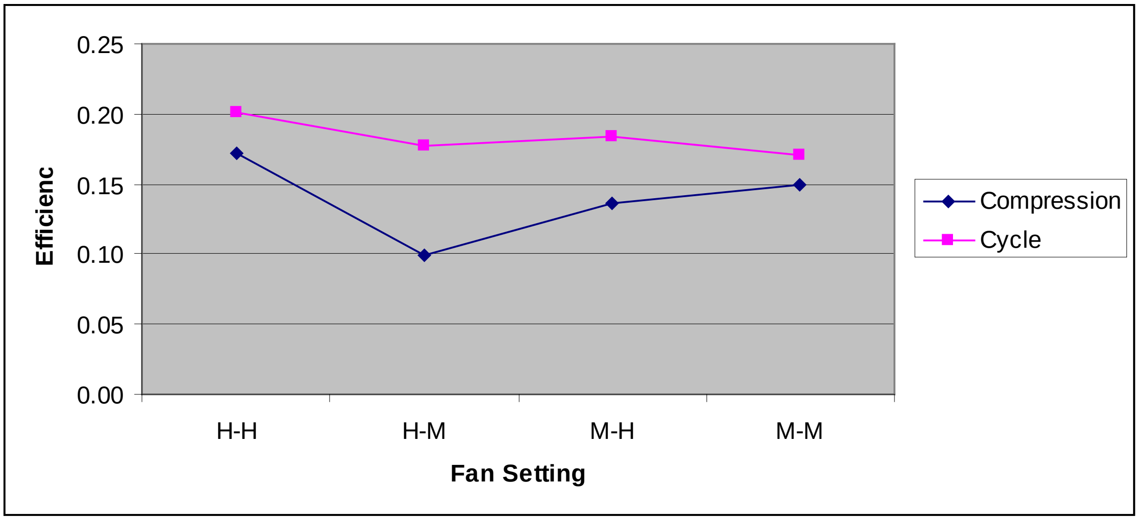


Figure 9. Compression and cycle efficiencies plotted vs. each fan setting during the TXV experiment. The first letter of the labels on the x axis represents the setting for fan S-5 (condenser) and the second letter represents the setting for fan S-6 (evaporator).

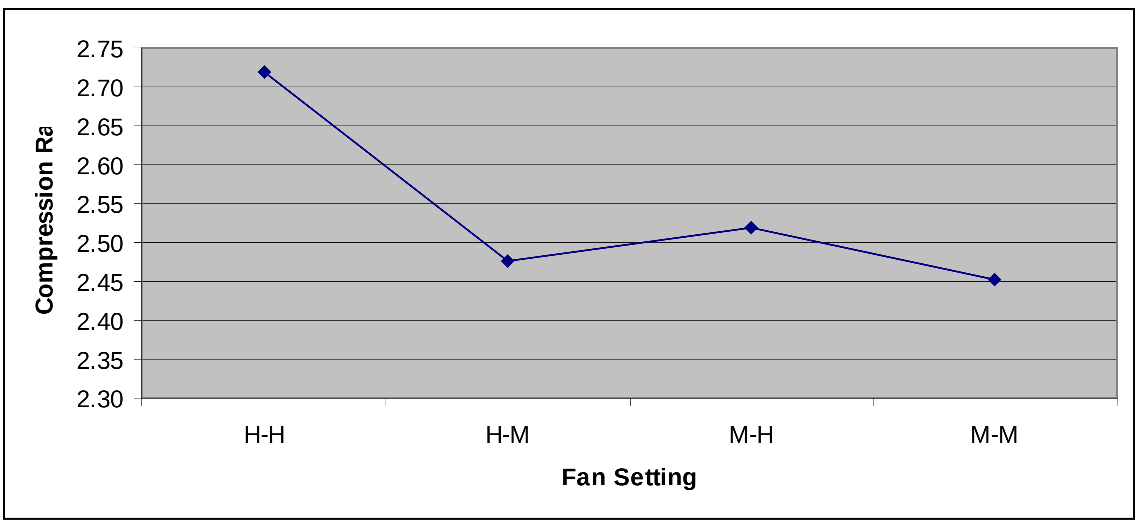


Figure 10. Compression ratio plotted vs. each fan setting during the TXV experiment. The first letter of the labels on the x axis represents the setting for fan S-5 (condenser) and the second letter represents the setting for fan S-6 (evaporator).

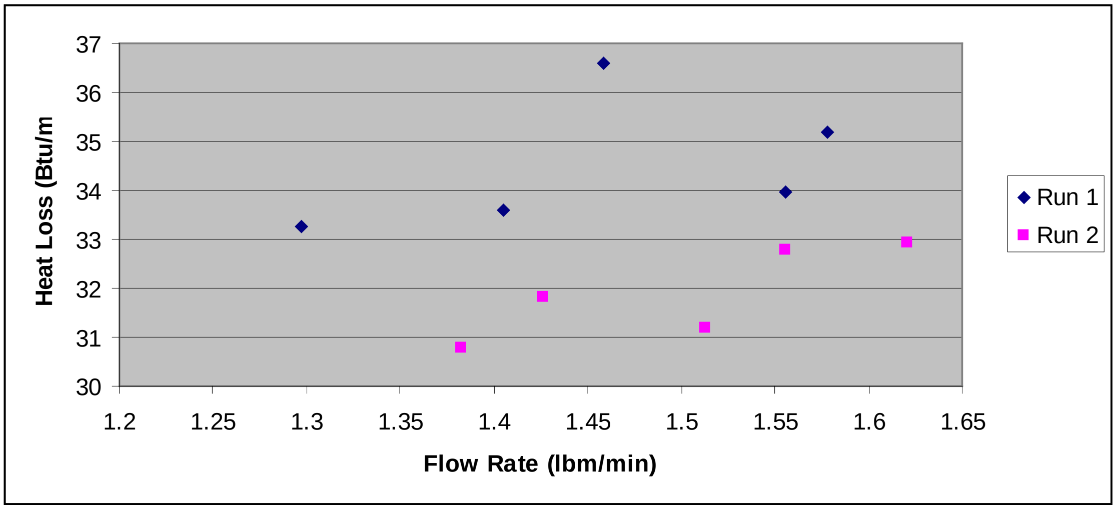


Figure 11. Rate of heat loss from the compressor plotted vs. refrigerant flow rate for both runs of the experiment in normal capillary mode.

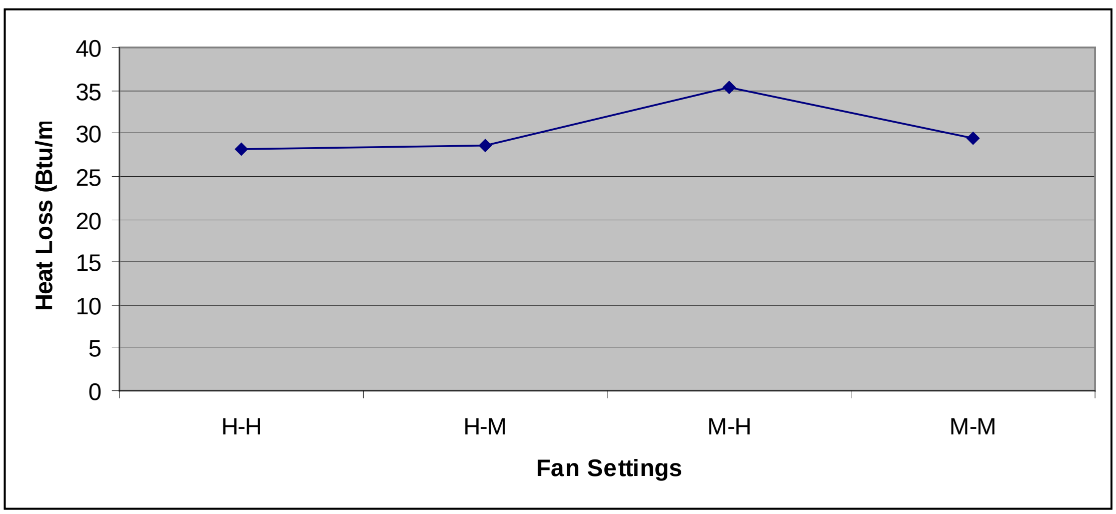


Figure 12. Rate of heat loss from the compressor plotted vs. each fan setting during the TXV experiment. The first letter of the labels on the x axis represents the setting for fan S-5 (condenser) and the second letter represents the setting for fan S-6 (evaporator).

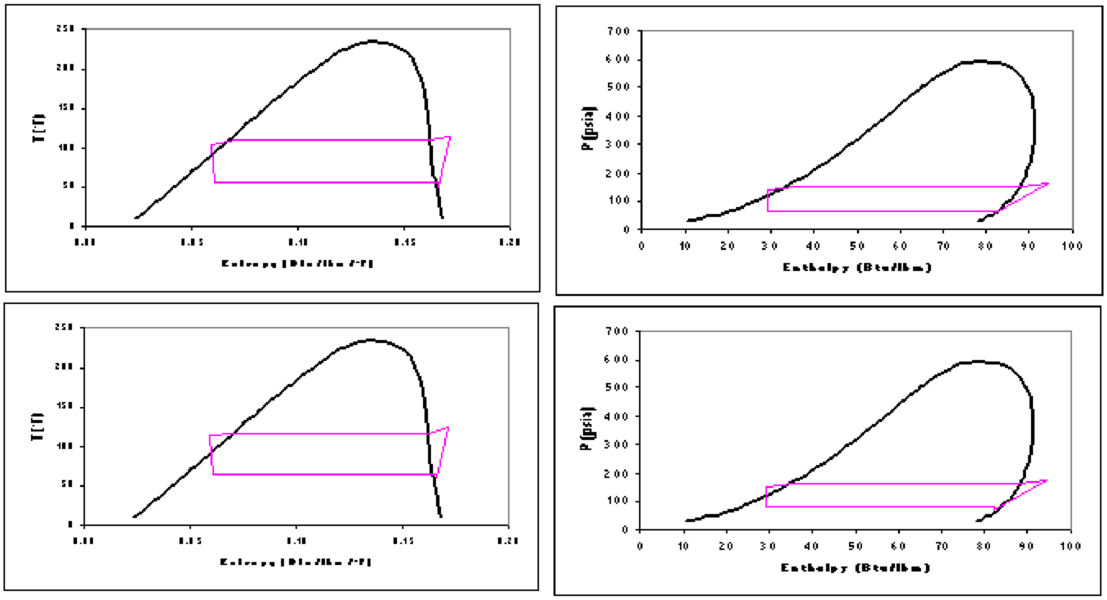


Figure 13. Experimental T-S and P-H plots for the first run condition during the TXV experiment (top) and theoretical plots following an increase in ambient temperature of 10ºF (bottom).

Refrigeration

Chemical Engineering Practice

CHG 3122

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From: Abubakar Waraich, Taha Khan, Group 3

Date: July 14, 2010

Subject: CHG 3122, Refrigeration

Refrigeration is best known for its use in the air conditioning of buildings and in the treatment, transportation, preservation of food and for large scale industrial applications. The term refrigeration implies the maintenance of a temperature below that of the surrounding. In a typical refrigeration cycle, heat Q_C is absorbed from a colder reservoir at temperature T_C. Vapour- compression systems must contain four major components. These are compressor, an evaporator, an expansion device and a condenser. The arrangement of the four components in vapour-compression system is shown in Figure#1 (Appendix A). The vapour-compression system is a system, in which the compressor operates isentropically, the heat transfer takes place only in the evaporator and the condenser, pressure drops only in the expansion device, and the expansion step is isenthalpic if the capillary tube is used or isentropic if the turbine is used. The experiment was performed by using the Scott Air Conditioning and Refrigeration system with the refrigerant working fluid Freon 12 also referred to as R-12.

The objectives of the refrigeration experiment are:

1. Study the refrigeration capacity and performance coefficients for a refrigeration unit as a function of circulation rate of refrigerant and heat load.
2. Compare the performance of the refrigeration unit in normal capillary mode and thermostatic expansion valve (TXV) mode.

The experiment was performed using the Scott Air conditioning and Refrigeration system with Freon-12 as the working fluid. The experiment was run under two conditions: the capillary mode and the thermal expansion mode. Five different pressure and temperature taps were setup in the experiment to measure the pressure and temperature for different set of circulation rates. There were some discrepancies encountered in the experiment such as the compressor was not able to handle very high temperatures due to repetitive changing circulation rates and the speed of the fan. Therefore, a fan was used to cool down the compressor, which delayed our experiment by 15-20 mins. For these circumstances, the pressure and temperature readings were taken consequently throughout the experiment so that the compressor does not shut down during the readings.

An increase in the circulation rate of the refrigerant lead to a increase in the performance of the unit in both the capillary and TXV mode. The average actual coefficient of performance values was found to be 1.69 for capillary run#1, 1.62 for capillary run#2 and 1.85 for the TXV mode. The refrigeration capacity resulted in an increase as the circulation rate was increased giving average values of RC: 73.30 Btu/min for capillary run#1, 68.93 Btu/min for capillary run#2 and 84.83 Btu/min for the TXV mode. Another observation that was analyzed in the refrigeration system was that when the actual coefficient of performance increased, the coefficient of performance of the fluid, COP_{fluid} also increased for capillary run#1&2, while it fluctuated for TXVmode since the TXV flow rates were the same. The average COP_{fluid} values ranged at 8.61 for capillary run#1, 6.51 for capillary run#2 and 5.74 for TXV mode.

Equipment

The refrigeration experiment was performed using the Scott Air Conditioning and Refrigeration Education system with Freon-12 often referred to as R-12 (dichloro-difluoro-methane, CCl_2F_2) as a working fluid. The refrigeration system was setup to be run in two different operating modes: normal capillary mode and thermostatic expansion valve (TXV) mode. The components of the refrigeration system as in (Figure#1) are: Compressor (A), Condenser (B), evaporator (C), Capillary (D), thermostatic expansion valve (E), temperature sensor (F), valve (V), moisture and liquid indicator (G), calibrated rotameter (H), drier (I), liquid refrigerant receiver tank (J), oil and refrigerant accumulator tank (K) and oil storage tank (L).

The pressure and temperature gauges are installed at different strategic locations throughout the refrigeration system. There is a fan installed behind the condenser and another one behind the evaporator. The wattmeter, current meter, potentiometer and several electrical switches are also installed along the system.

Procedure

1. Record the room temperature and pressure for the laboratory pilot plant.
2. Get familiarize with the equipment and the excel program to make sure the readings are recorded for the pressure and temperature at different strategic locations, watt, refrigerant level, and rotameter reading. To prevent any damage to the unit, use precautions outlines in the lab manual.
3. Make sure the refrigeration system is set at thermal equilibrium. Ensure the temperature gauges at the five different locations are all the same.
4. Record the wattage requirements for all fan setting combination High-High, High-medium, High-low and vice-versa while the compressor is shut-off.(Appendix B, table#11)

Capillary Mode Operation: (Two Trials)

5. Begin the experiment by setting the fan setting to high-high for the condenser and evaporator.
6. Set the system to capillary mode by turning the valve 5 closed and opening the valve 4 for the expansion step. The valve 4 was used to control the circulation rate of the refrigerant. Use a lower amount of refrigerant to conduct the first run of capillary mode.
7. Set the circulation rate of refrigerant at a maximum rotameter reading, which was decided by the demonstrator. Make sure the system stabilizes by checking if the pressure is between 120 psig and 140 psig and there are no vapour bubbles in the rotameter. Once the system has stabilized, record the five temperature and

pressure readings at different locations (Appendix B, table#13) for power consumption, rotameter reading, and level of refrigerant and flow rate throughout the refrigeration system. (Figure#1).

8. Conduct the second run of capillary mode by using a higher amount of refrigerant and repeat the steps 5-7. (Appendix B, table#14)

   Thermal expansion valve (TXV) mode:

9. Set the system to thermal expansion mode by turning the valve 4 closed and opening the valve 5 for the expansion step. The valve 5 is then used to control the circulation rate of the refrigerant. The same readings were then recorded for four different fan settings: High-high, High-Medium, Medium-High and Medium-Medium. (Appendix B, table#15)

The refrigerant system works using the vapour-compression cycle (Appendix A, figure#8). The refrigerant leaves the evaporator as a vapour and then enters the compressor where it leaves as high pressure vapour. Then the high pressure vapour then enters a condenser where it forms a high pressure saturated liquid and releases off heat to the hot reservoir. This high pressure liquid then enters the expansion or capillary valve which results in a low pressure liquid. This low pressure also results in releasing off cooler temperature to the surroundings, where it encounters the two- phase system (liquid-vapour). This low pressure liquid then enters the evaporator to return back to its original state as a vapour by absorbing heat into the cold reservoir.

Summary of Results

1. The temperatures for the hot and cold reservoir are calculated using the temperature at point 2-3 and point 4-1 respectively for capillary and expansion valve modes. An average was taken for the temperatures at hot and cold reservoir for different circulation rates. The average hot and cold reservoir temperatures are as follows:

Capillary (run#1):	$T_C = 568.74\ °R$	$T_H = 504.07\ °R$
Capillary (run#2):	$T_C = 568.70\ °R$	$T_H = 502.07\ °R$
TXV	$T_C = 575.33\ °R$	$T_H = 512.47\ °R$

2. The ideal (Carnot) efficiency was determined by using the equation (1.), which is based on the ratio of hot reservoir temperature to the difference between the hot and cold reservoir. The average Carnot efficiency was calculated at circulation rates for capillary and TXV mode which are as follows:

Capillary (run#1)	$COP_{MAX} = 7.80$
Capillary (run#2)	$COP_{MAX} = 7.54$
TXV	$COP_{MAX} = 8.16$

3. The refrigeration capacity (RC) is calculated using the enthalpies at point 1 and 4 and multiplying it by the circulation rate of the refrigerant (equation#4), which is also the cold reservoir heat. The average refrigeration capacity was calculated for both the capillary and the TXV modes and are as follows:

Capillary (run#1)	$RC = Q_C = 73.30\ {}^{Btu}/_{min}$
Capillary (run#2)	$RC = Q_C = 68.93\ {}^{Btu}/_{min}$
TXV	$RC = Q_C = 84.83\ {}^{Btu}/_{min}$

4. The actual coefficient of performance was determined by using the equation (2.), which is the ratio of the refrigeration capacity to the power to compressor. The

average actual coefficient of performance were calculated at different circulation rates for capillary and TXV mode and are as follows:

Capillary (run#1)	$COP_{actual} = 1.69$
Capillary (run#2)	$COP_{actual} = 1.62$
TXV	$COP_{actual} = 1.85$

5. The coefficient of performance of the fluid was determined by using equation (3.), which is based on the specific enthalpies of the working fluid at different stages of the refrigeration system. The average coefficient of performance of the fluid was calculated at different circulation rates for capillary and TXV mode and are as follows:

Capillary (run#1)	$COP_{fluid} = 8.61$
Capillary (run#2)	$COP_{fluid} = 6.51$
TXV	$COP_{fluid} = 5.74$

6. The compression efficiency was determined by the equation (5.), which is based on the ratio of specific enthalpies of the working fluid at 1 and 2 stages multiplied by circulation rate to the work done by the compressor. The average compression efficiency was calculated at different circulation rates for capillary and TXV mode and are as follows:

Capillary (run#1)	$\eta_{comp} = 0.10$
Capillary (run#2)	$\eta_{comp} = 0.07$
TXV	$\eta_{comp} = 0.22$

7. The cycle efficiency was determined by the equation (6.), which is based on the ratio of actual coefficient of performance to the Ideal (Carnot) efficiency also denoted as COP_{max}. The average cycle efficiency was calculated at different circulation rates for capillary and TXV mode and are as follows:

Capillary (run#1) $\eta_{cycle} = 0.22$
Capillary (run#2) $\eta_{cycle} = 0.21$
TXV $\eta_{cycle} = 0.23$

8. The compression ratio (CR) is determined by using the equation (7.), which is based on the ratio of the pressure at stage 2 to the pressure at stage 1 in the refrigeration system. The average compression ratio was calculated for different circulation rates at the capillary and TXV mode.

Capillary (run#1) $CR = 3.09$
Capillary (run#2) $CR = 3.19$
TXV $CR = 2.85$

Discussions

The performance of the refrigeration system increased as the circulation rate increases. The results obtained (Appendix A, tables 3, 6&9) for the capillary mode (run#1 and run#2) and TXV mode was very close to each other.

The actual coefficient of performance was plotted against the circulation rate (Appendix A, figure#1), the actual coefficient of performance showed linear characteristic trends as the circulation rate was increased for capillary runs#1 &2. However, the TXV mode showed no characteristic trends as opposed to the capillary modes. The TXV mode showed constant performance, since there were no changes in the circulation rate of the refrigerant system.

The coefficient of performance of the fluid was plotted against the circulation rate (Appendix, Figure#11). The results obtained were different for the capillary modes and TXV mode. The capillary mode (run#1&2) both showed a linear characteristic trend for coefficient of performance of the fluid, while the TXV mode showed a constant characteristic pattern. The capillary mode had a better performance in the run#2 while the capillary mode in run#1 was less efficient. The TXV mode had the same performance as the capillary mode (run#2), which shows the experiment for coefficient of performance of the fluid was carried out correctly.

The coefficient of performance of the fluid was plotted against the refrigerant level (Appendix A, figure#3). The results obtained showed exponential decline characteristic trend as the level of refrigerant was increased, with increasing circulation rate of refrigerant. The capillary mode (run#1) had more outliers with the coefficient of performance of the fluid, this might be due to delays caused due to stabilizing the refrigerant system during the experiment. The capillary mode (run#2) had better results compared to the first run. The TXV mode also showed an exponential decline, as the level of refrigerant was increased in the refrigeration system.

The refrigeration capacity was plotted against the circulation rate (Appendix A, figure#4). The results obtained for capillary mode (run#1&2) showed a straight linear characteristic trend while the TXV showed constant trend behaviour. The capillary mode (run#1) was able to remove more heat easily at higher circulation rate compared to capillary mode (run#2), which had a lower circulation rate. The TXV mode did not show any valid characteristic pattern, as the circulation rate never changed. The TXV mode followed a constant behavior at a constant circulation rate.

The compression efficiency was plotted against the circulation rate (Appendix A, figure#5). The result obtained for capillary mode (run#1&2) showed a straight linear characteristic trend while the TXV showed constant trend behavior. The capillary mode (run#1) had less compression efficiency compared to the capillary mode (run#2). This shows that as we increase the circulation rate in capillary mode (run#2), the compression efficiency also gets more efficient. The TXV mode had more compression efficiency than capillary mode since it was running at a higher circulation rate and did not show any circulation rate changes.

The coefficient performance was plotted as a function of heat load, (Appendix A, figure#7). The results obtained for the capillary mode (run#1 and 2) and TXV mode, both showed linear characteristic behavior as the circulation rate and the refrigerant rate of refrigerant was increased. The actual coefficient of performance followed a linear trend with the amount of heat load, this shows that the refrigeration system was working efficiently while absorbing energy in a cold reservoir.

The cycle efficiency was plotted as a function of circulation rate of refrigerant (Appendix A, figure#9). The result obtained for capillary mode (run#1&2) showed a straight linear characteristic trend while the TXV showed constant trend behavior. The TXV mode was working more efficiently, when the level of refrigerant was increased with respect to increase in circulation rate of refrigerant. The capillary mode (run#1&2) also showed similar linear trends with compared to each other; however they were less efficient for the refrigeration cycle.

The compression ratio was plotted as a function of circulation rate of refrigerant (Appendix A, figure#10). The results obtained for capillary mode (run#1&2) showed an exponential decline for capillary mode (run#1&2) while the TXV mode showed a constant decline trend. The capillary mode followed the same trend even when the circulation rate and refrigerant rate was increased. The TXV mode showed constant trend with increasing flow rate and had a lower compression ratio compared to the capillary mode (run#1&2).

Difference between the ideal and actual vapour-compression refrigeration cycles:

1. In actual, the refrigerant enters the compressor at point 1, slightly superheated vapor,
   instead of saturated vapor in the ideal cycle.
2. The suction line (the line connecting the evaporator to the compressor) is very long.
   Thus pressure drop and heat transfer to the surroundings can be significant.
3. The compressor is not internally reversible in practice, which increase entropy. However, using a multi-stage compressor with intercooler, or cooling the refrigerant during the compression process, will result in lower entropy, state 2'.
4. In actual, the refrigerant leaves condenser as sub-cooled liquid. Sub-cooling increases the cooling capacity and will prevent entering any vapor (bubbles) to the expansion valve.
5. Heat rejection and addition in the condenser and evaporator do not occur in constant pressure (and temperature) as a result of pressure drop in the refrigerant.

Set#2 Problems

Question#1:

The quality of vapour at point 4 can be determined by using the principles of Raoult's Law. At point 4, the refrigerant is at two phases, liquid and vapour phase, therefore, by using the Raoult's law and the temperature and pressure values given, we can calculate the dew point and bubble point. For dew point, the vapour has a quality of 1 and for bubble point, the liquid has a quality of 0. The properties at point 4 are calculated by assuming an isenthalpic process for expansion, thus the enthalpies at point 3 and 4 are equal. Since, the compression refrigeration system is not an internally reversible cycle, it involves throttling which is an irreversible process. If the expansion valve was replaced by an isentropic turbine, the refrigerant would enter the evaporator at point 4. As a result the refrigeration capacity would increase (area under point 4) and the net work input would decrease (turbine would produce some work). However, replacing the expansion valve by a turbine is not practical as it involves cost considerations and complexity. Therefore, it is better to assume an isenthalpic process for expansion, as it is much cheaper and easier to use capillary tube for expansion.

Question#2:

The fan settings used for the TXV mode were: High-High, High-medium, Medium-High and Medium-Medium.

For High-high fan setting, the actual coefficient of performance is at 1.97, which shows the highest performance peak level. For High-medium, the actual coefficient of performance is at 1.90, which results in the high performance peak level. For Medium-

High, the actual coefficient of performance is at 1.78, which results in the low performance peak level. Finally, for Medium-Medium, the actual coefficient of performance is at 1.75, which is the lowest performance peak level.

Therefore, we draw the comparison that the TXV mode reaches the highest peak performance at the High-High fan setting, where it achieves the highest rate of energy absorbed in a cold reservoir per unit mass of refrigerant, Q_C and the highest power (work). (Appendix A, table#9).

Question#3:

The actual coefficient of performance increases as the circulation rate of refrigerant increases for both Capillary mode (run#1&2) and TXV mode (Appendix A, Figure#2). The capillary mode (run1&2) both exhibit a linear characteristic behaviour while the TXV mode shows a constant increase in behavior. As we increase the circulation rate of the refrigerant (the velocity also increases), this increases the pressure according to Bernoulli's equation. Therefore, as we decrease the pressure, the temperature also decreases, which will have an effect on the temperature of the hot reservoir (Appendix A, tables 1, 4, & 7). However, all the parameters are constant, the temperature of the cold reservoir would need to be increased to maintain constant enthalpy/entropy at all the points. Therefore, changes in the temperature of cold reservoir T_C (Appendix A, table#1) would lead to an increase in the heat absorbed in the cold reservoir, Q_C (Appendix A, table#1,4,7). Increasing Q_C would lead to an increase in the actual coefficient of performance in the refrigeration system. Therefore an increase in flow rate will also result in the increase of refrigeration capacity or Q_C. The coefficient of performance with circulation rate showed a constant behaviour (Appendix A, figure#2). As the flow rate decreased in the valve, there was no pressure change in the refrigeration system. Therefore, there was no change applied to the actual coefficient of performance of the refrigeration system.

Question#4:

The compressor used in the experiment was not properly insulated which resulted in overheating of the compressor. Therefore, there was more heat loss resulting from the non-insulated compressor. The rate of heat loss was evaluated using the difference between the enthalpies at point 2 & 3 and multiplying it by the circulation rate. The compression efficiency was plotted as a function of heat loss, (Appendix A, figure#6). The capillary mode (run#1&2) followed a semi-parabolic behaviour whereas the TXV mode followed a constant trend. The heat loss Q_H was greater than the heat gain Q_C, which shows that since the compressor was not properly insulated, there was more heat loss in the TXV mode compared to the capillary (run#1&2). Such discrepancies might lead to changes in the calculated thermodynamic properties.

There was non-insulated piping used for the compressor and evaporator, this might have resulted in more heat loss in the refrigeration system.

Conclusion and Recommendation

The refrigerant R-12, was used as the working fluid to set up vapour-compression cycle in a refrigeration system. The R-12 was circulated throughout the refrigeration system, which resulted in increase circulation rate with an increase in actual coefficient of performance. The capillary mode (run#1&2) were performed and were shown to have linear characteristic behaviour while the TXV showed a constant trend based on the performance of the refrigeration system. The refrigeration capacity (RC) obtained was higher than the refrigeration capacity obtained for the capillary mode (run#1&2). The TXV mode had a higher cycle and compression efficiency compared to the capillary mode (run#1&2). The compression ratio obtained for the capillary mode (run#2) was higher than the TXV mode and the capillary mode (run#1). The sources of error could be minimized so that we can get better results which will allow the vapour-compression cycle to be more efficient.

Recommendations

1. Using a multi-stage compressor with intercooler, or cooling the refrigerant during the compression process, will result in lower entropy, point 2.
2. A two-stage cascade refrigeration cycle improves the coefficient of performance of a refrigeration cycle. Moreover, the refrigerants can be selected to have reasonable evaporator and condenser pressures in the two or more temperature ranges.

3. Ensuring that the compressor used for the experiment is perfectly insulated so that cooling water can be used to submerge the compressor in a water tank to bring down the overheating of the compressor.
4. Using digital pressure and temperature transducers to accurately record the recordings quickly in a LABVIEW program.
5. Using ozone friendly working fluids such as R22, R134a, R744, R290 etc. which will be better for the environment.
6. The rotameter readings kept on fluctuating during the experiment, it will be better to have new rotameter or a computer generated program to stabilize the refrigeration system ahead of time without causing any delays.

APPENDIX A
TABLES AND FIGURES

Table#1: The amount of heat lost and gain in the refrigeration system for different circulation rates at Capillary mode (run#1)

Circulation rate (lb_m/min)	Refrigerant Height (inch)	T_H (°R)	T_C (°R)	W (Btu/min)	Q_c (Btu/min)	Q_H (Btu/min)
1.502	14	568.45	506.47	44.36	79.60	86.10
1.448	14.4	570.82	506.47	44.36	76.65	86.48
1.394	15	568.58	505.47	44.07	75.16	83.06
1.286	15.2	568.38	502.47	42.65	69.45	79.11
1.210	15.4	567.47	499.47	41.51	65.66	76.05
Average		568.74	504.07	43.39	73.30	82.16

Table#2: The changes in the amount of enthalpies at different strategic stages of the refrigeration system for Capillary(run#1):

Circulation rate (lb_m/min)	h_1 (Btu/lb)	h_2 (Btu/lb)	h_2^s (Btu/lb)	h^3 (Btu/lb)	h^4 (Btu/lb)	P_1 (psig)	P_2 (psig)
1.502	84.28	88.61	87.74	31.28	31.28	50.19	150.19
1.448	84.20	90.98	87.93	31.26	31.26	52.19	154.69
1.394	84.70	90.37	87.75	30.78	30.78	50.20	150.70
1.286	85.29	92.80	87.74	31.28	31.28	46.29	149.79
1.210	85.04	93.62	87.67	30.79	30.79	45.50	148.00

Table#3: The performance of the refrigeration system using various parameters for different circulation rates at normal capillary mode (run#1).

Circulation rate (lb_m/min)	COP_{max}	COP_{actual}	COP_{fluid}	RC (Btu/min)	η_{Comp}	η_{Cycle}	CR
1.502	8.17	1.79	12.24	79.60	0.12	0.22	2.99
1.448	7.87	1.73	7.80	76.65	0.12	0.22	2.96
1.394	8.01	1.71	9.51	75.16	0.10	0.21	3.00
1.286	7.62	1.63	7.19	69.45	0.07	0.21	3.24
1.210	7.35	1.58	6.32	65.66	0.08	0.22	3.25
Average	7.80	1.69	8.61	73.30	0.10	0.22	3.09

Table#4: The amount of heat lost and gained in the refrigeration system for different circulation rates at Capillary mode (run#2):

Circulation rate (lb_m/min)	Refrigerant Height (inch)	T_H (°R)	T_C (°R)	W (Btu/min)	Q_c (Btu/min)	Q_H (Btu/min)
1.394	14.8	568.84	504.47	43.11	74.59	84.67
1.3184	15	569.84	504.47	42.94	70.45	81.73
1.286	15.3	569.38	502.47	42.94	69.08	80.15
1.2104	15.5	567.98	499.47	42.54	65.92	76.33
1.178	15.5	567.47	499.47	41.51	64.61	74.79
Average		568.70	502.07	42.61	68.93	79.53

Table#5: The changes in the amount of enthalpies at different strategic stages of the refrigeration system for Capillary (run#2):

Circulation rate (lb_m/min)	h_1 (Btu/lb)	h_2 (Btu/lb)	h_2^s (Btu/lb)	h^3 (Btu/lb)	h^4 (Btu/lb)	P_1 (psig)	P_2 (psig)
1.394	84.78	92.02	87.77	31.28	31.28	48.22	150.72
1.3184	85.21	93.76	87.85	31.77	31.77	48.22	152.72
1.286	85.49	94.10	87.82	31.77	31.77	48.29	151.79
1.2104	85.74	94.34	87.71	31.29	31.29	45.50	149.00
1.178	85.89	94.53	87.67	31.04	31.04	45.50	148.00

Table#6: The performance of the refrigeration system using various parameters for different circulation rates at normal capillary mode (run#2).

Circulation rate (lb_m/min)	COP_{max}	COP_{actual}	COP_{fluid}	RC	η_{Comp}	η_{Cycle}	CR
1.394	7.84	1.73	7.40	74.59	0.10	0.22	3.13
1.318	7.72	1.64	6.25	70.45	0.08	0.21	3.17
1.286	7.51	1.61	6.24	69.08	0.07	0.21	3.14
1.210	7.29	1.55	6.33	65.92	0.06	0.21	3.27
1.178	7.35	1.56	6.35	64.61	0.05	0.21	3.25
Average	7.54	1.62	6.51	68.93	0.07	0.21	3.19

Table#7: The amount of heat lost and gain in the refrigeration system for different circulation rates at TXV mode.

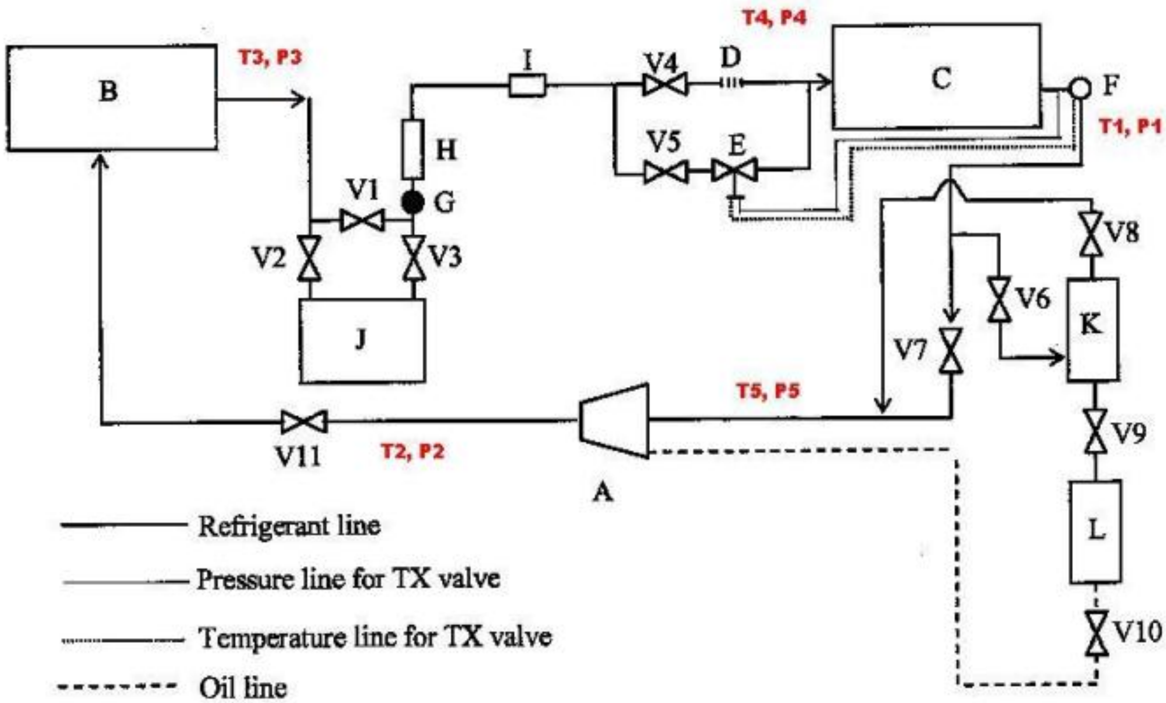
Fan setting	Circulation rate (lb_m/min)	Refrigerant Height (inch)	T_H (°R)	T_C (°R)	W (Btu/min)	Q_c (Btu/min)	Q_H (Btu/min)
H-H	1.718	15.5	571.36	510.47	44.36	87.41	103.49
H-M	1.718	15.6	572.88	512.47	45.33	85.94	99.64
M-H	1.718	15.7	580.88	514.47	46.63	83.04	98.62
M-M	1.718	16.0	576.21	512.47	47.49	82.92	96.94
	Average		575.33	512.47	45.95	84.83	99.67

Table#8: The changes in the amount of enthalpies at different strategic stages of the refrigeration system for TXV mode.

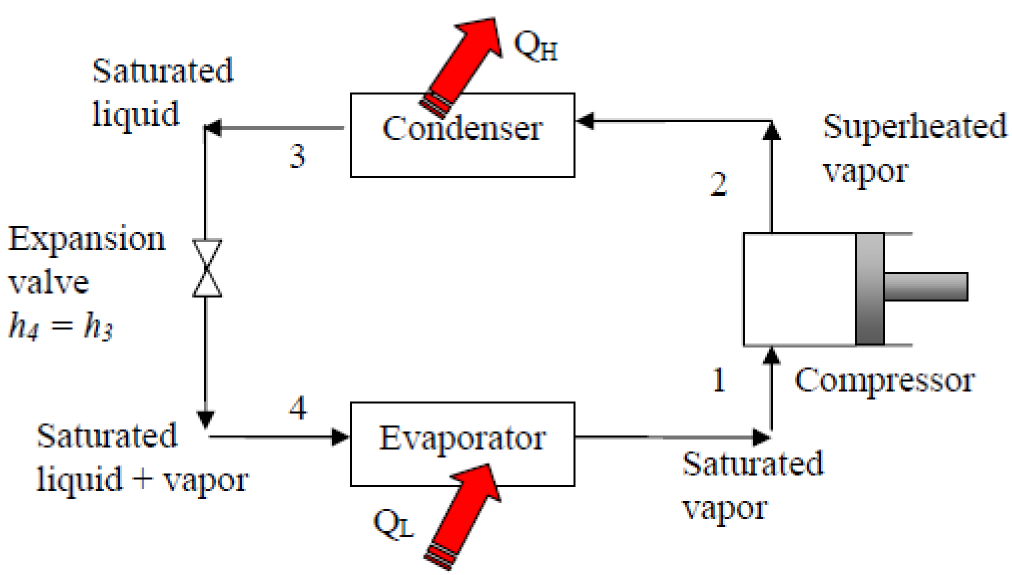
Fan setting	Circulation rate (lb_m/min)	h_1 (Btu/lb)	h_2 (Btu/lb)	h_2^s (Btu/lb)	h^3 (Btu/lb)	h^4 (Btu/lb)	P_1 (psig)	P_2 (psig)
H-H	1.718	82.64	92.00	87.97	31.76	31.76	56.29	155.79
H-M	1.718	82.27	90.24	88.09	32.25	32.25	58.42	158.92
M-H	1.718	82.54	91.61	88.70	34.21	34.21	58.60	176.10
M-M	1.718	82.27	90.44	88.35	34.01	34.01	58.42	170.92

Table#9: The performance of the refrigeration system using various parameters for different circulation rates at TXV mode.

Fan setting	Circulation rate (lb_m/min)	COP_{max}	COP_{actual}	COP_{fluid}	RC	η_{Comp}	η_{Cycle}	CR
H-H	1.718	8.38	1.97	5.44	87.41	0.21	0.24	2.77
H-M	1.718	8.48	1.90	6.28	85.94	0.22	0.22	2.72
M-H	1.718	7.75	1.78	5.33	83.04	0.23	0.23	3.01
M-M	1.718	8.04	1.75	5.91	82.92	0.22	0.22	2.93
	Average	8.16	1.85	5.74	84.83	0.22	0.23	2.85

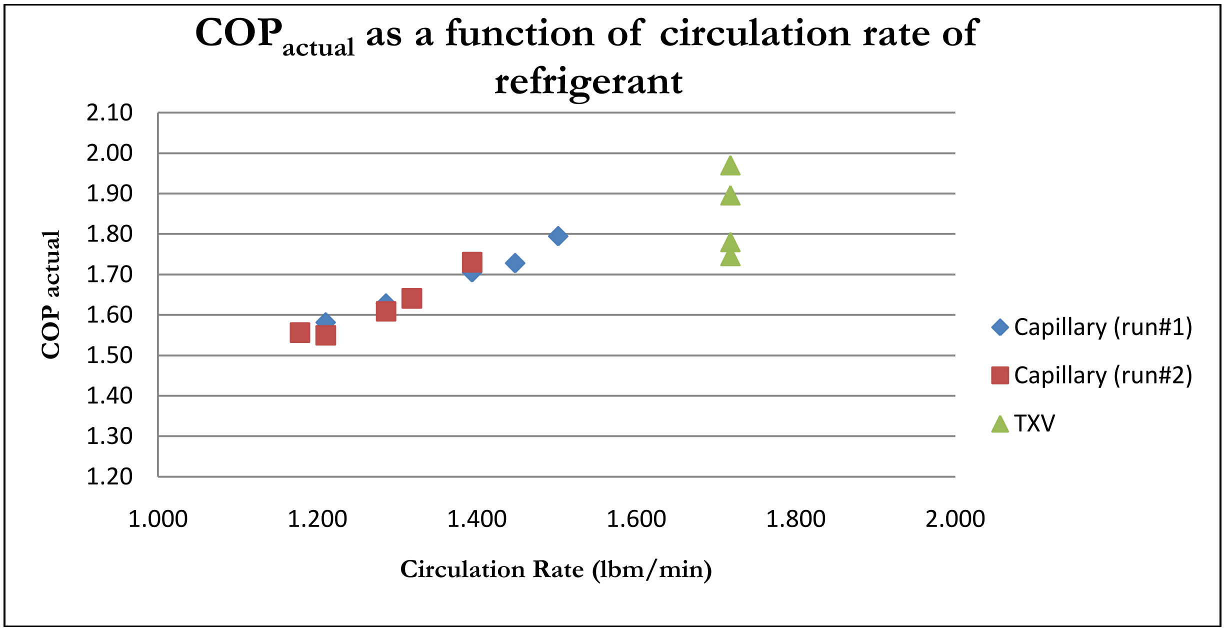


Figure#1: Schematic diagram for the Scott Air Conditioning and Refrigeration System, where
1 – Evaporator outlet, 2 – Condenser inlet, 3 – Condenser outlet, 4 – Evaporator inlet, and 5 – Compressor inlet.

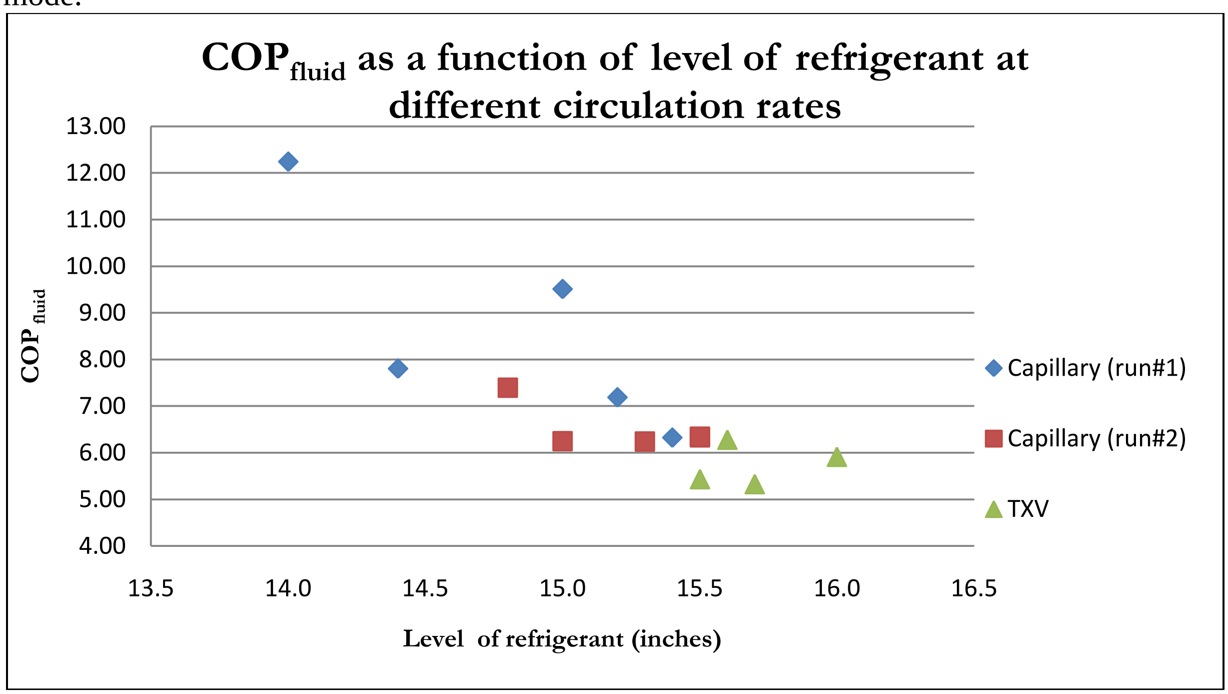


Figure#1(a): Schematic diagram of vapour-compression refrigeration cycle

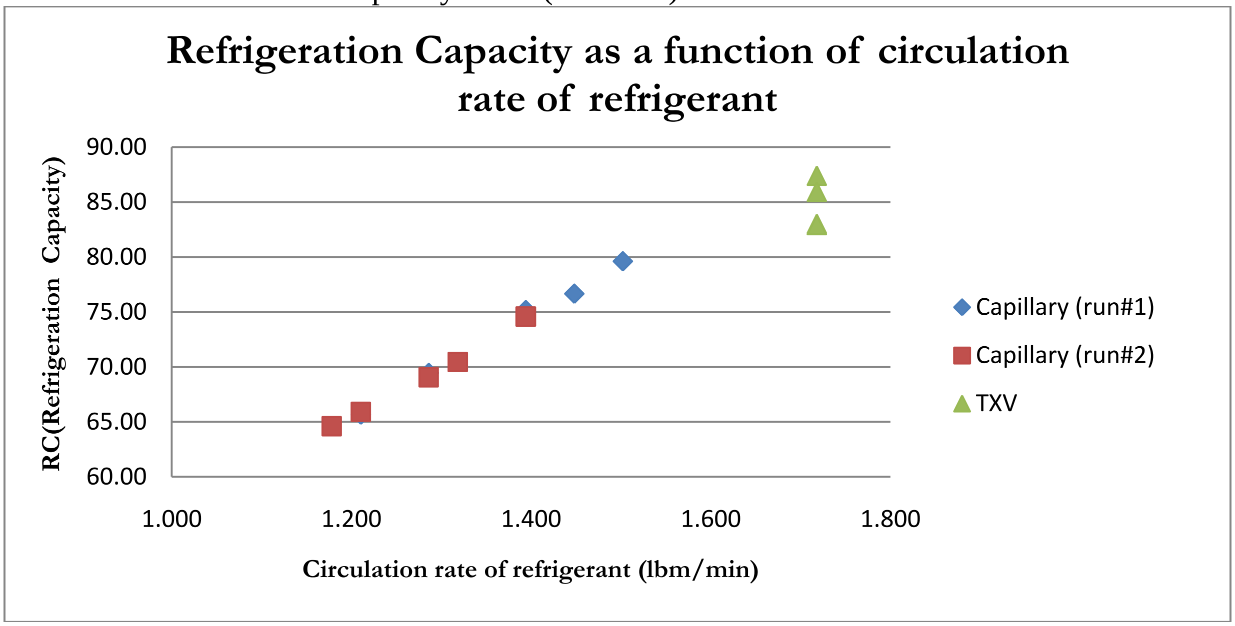
Figure#2: Plot for the actual coefficient of performance as a function of circulation rate of refrigeration system for different circulation rates at normal capillary rate (run#1&2) and TXV mode.



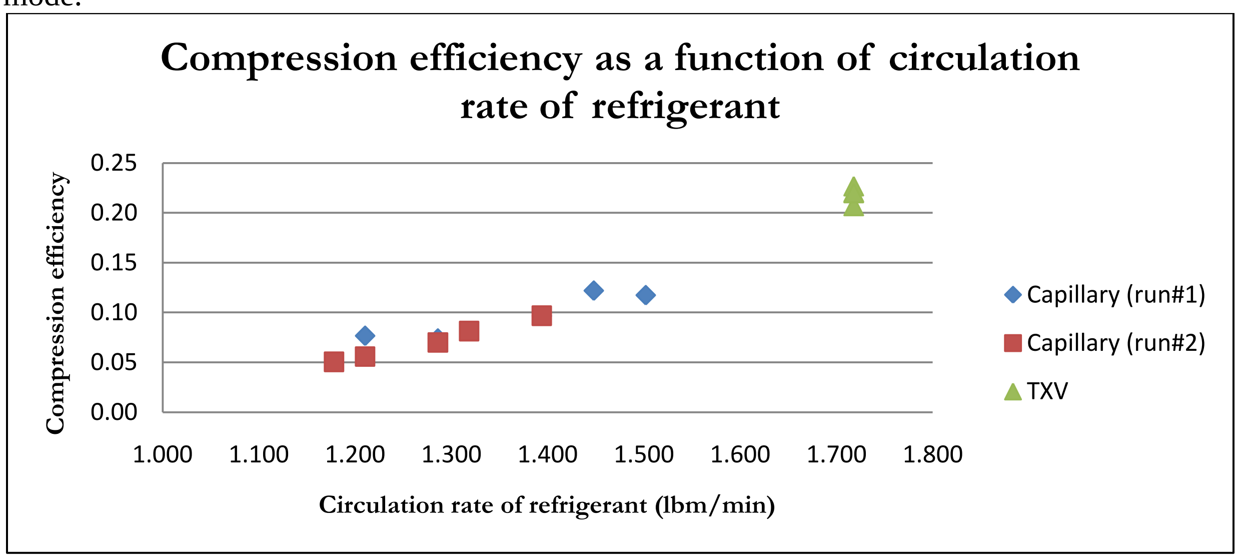
Figure#3: Plot for the coefficient of performance of the fluid as a function of level of refrigerant for different circulation rates at normal capillary mode (run#1&2) and TXV mode.



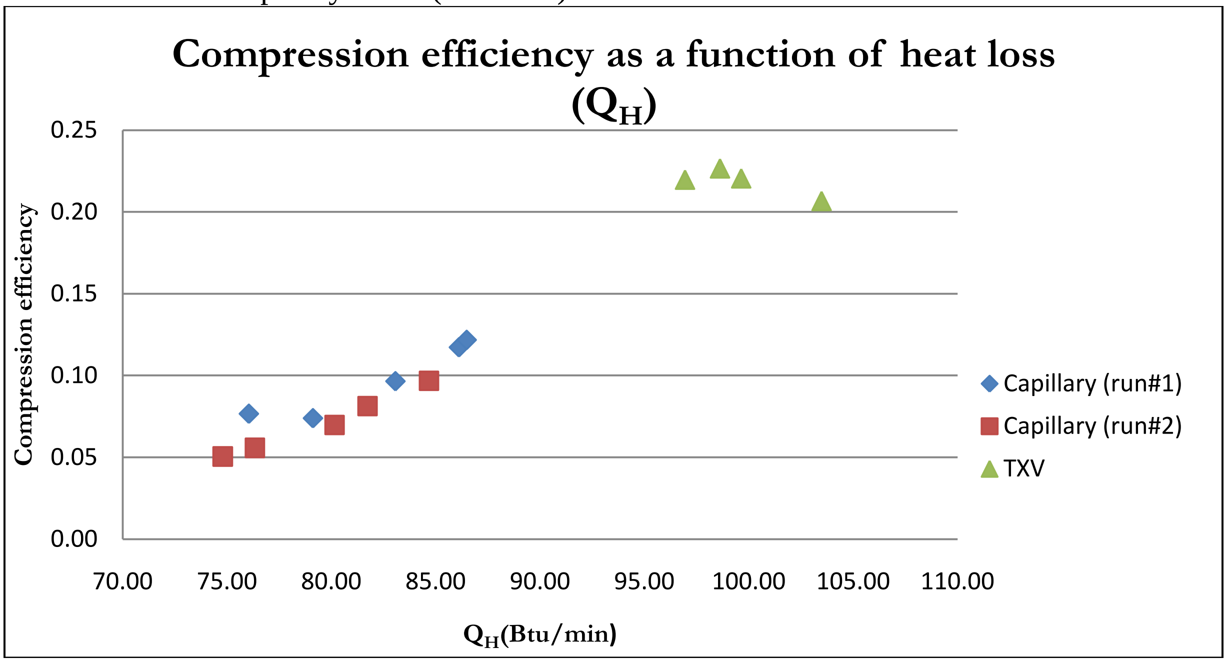
Figure#4: Plot for the refrigeration capacity as a function of circulation rate for different circulation rates at normal capillary mode (run#1&2) and TXV mode.



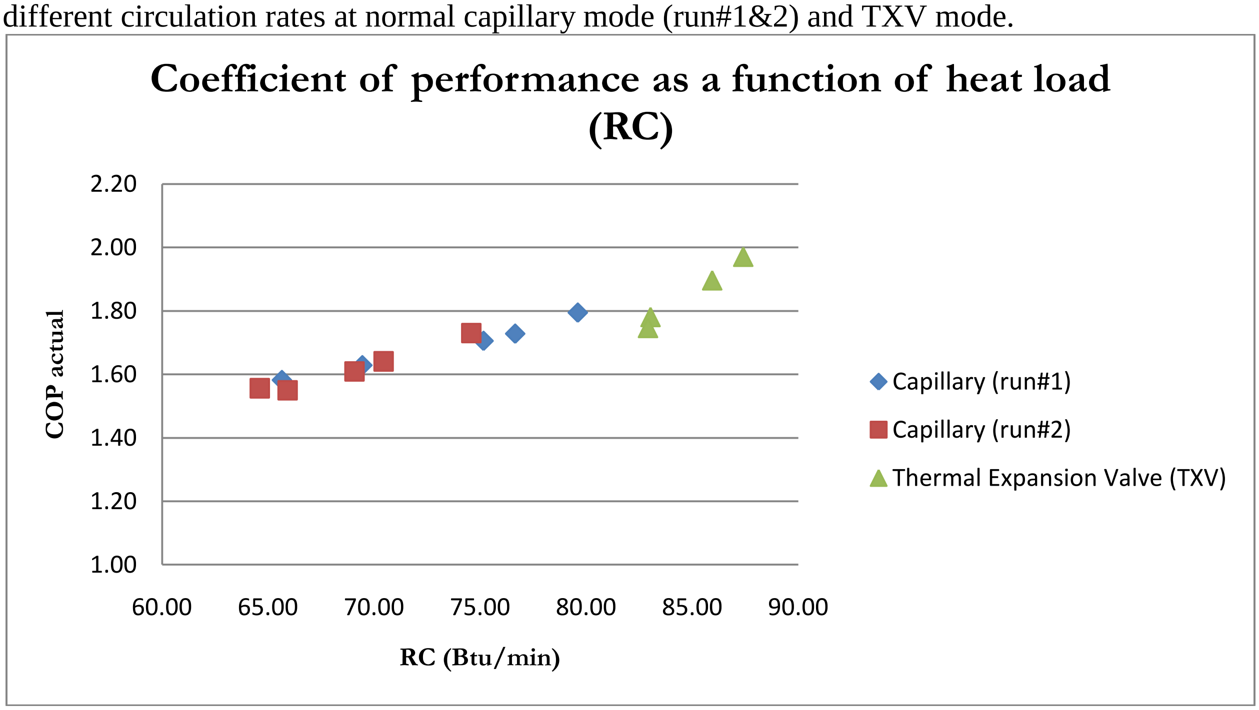
Figure#5: Plot for the compression efficiency as a function of circulation rate of refrigerant for different circulation rates at normal capillary mode (run#1&2) and TXV mode.



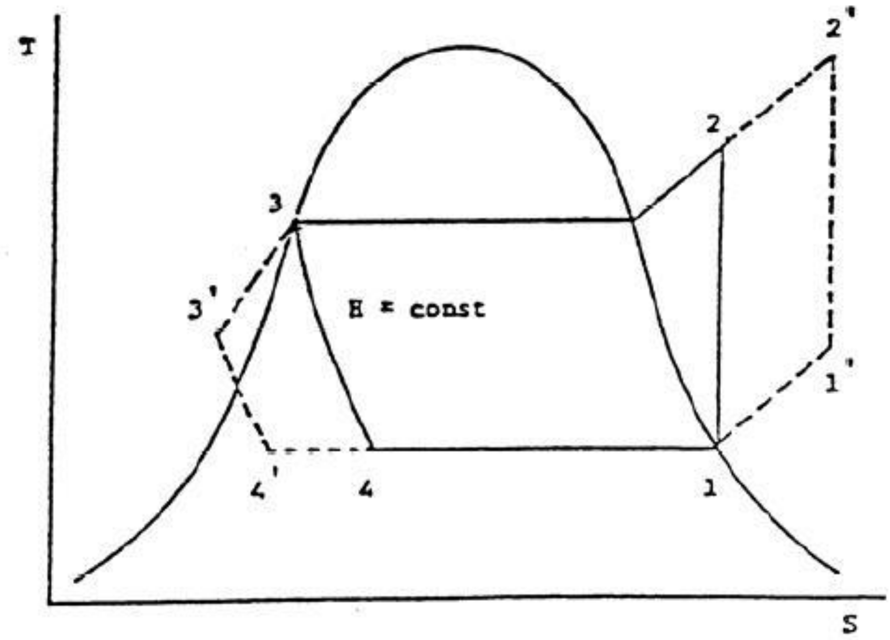
Figure#6: Plot for compression efficiency as a function of heat loss for different circulation rate at capillary mode (run#1&2) and TXV mode.



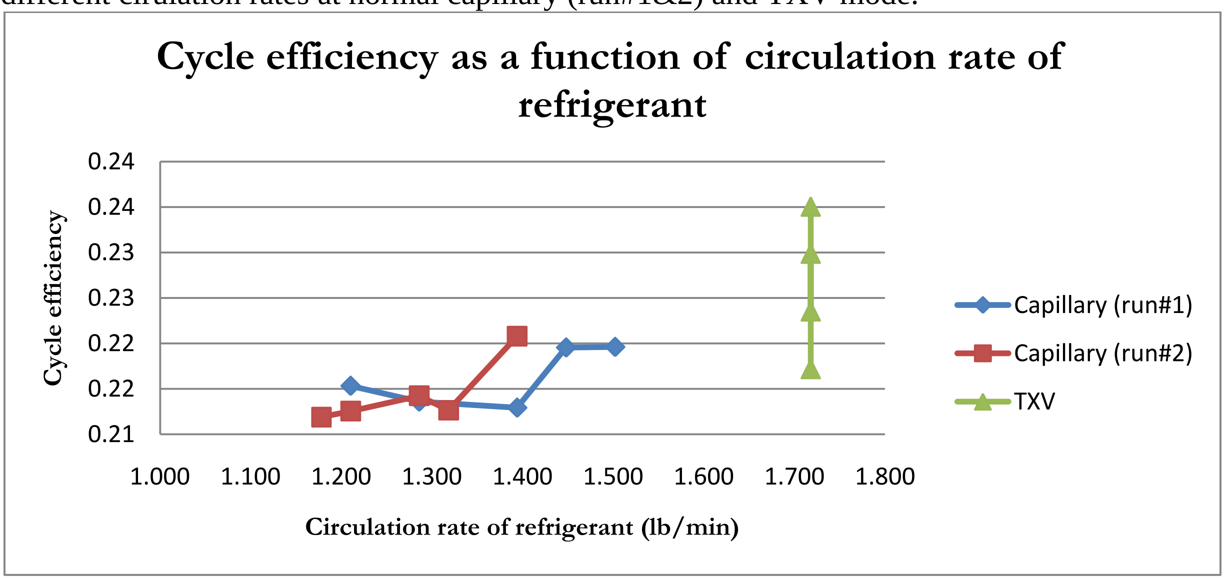
Figure#7: Plot for actual coefficient of performance as a function of heat load for different circulation rates at normal capillary mode (run#1&2) and TXV mode.



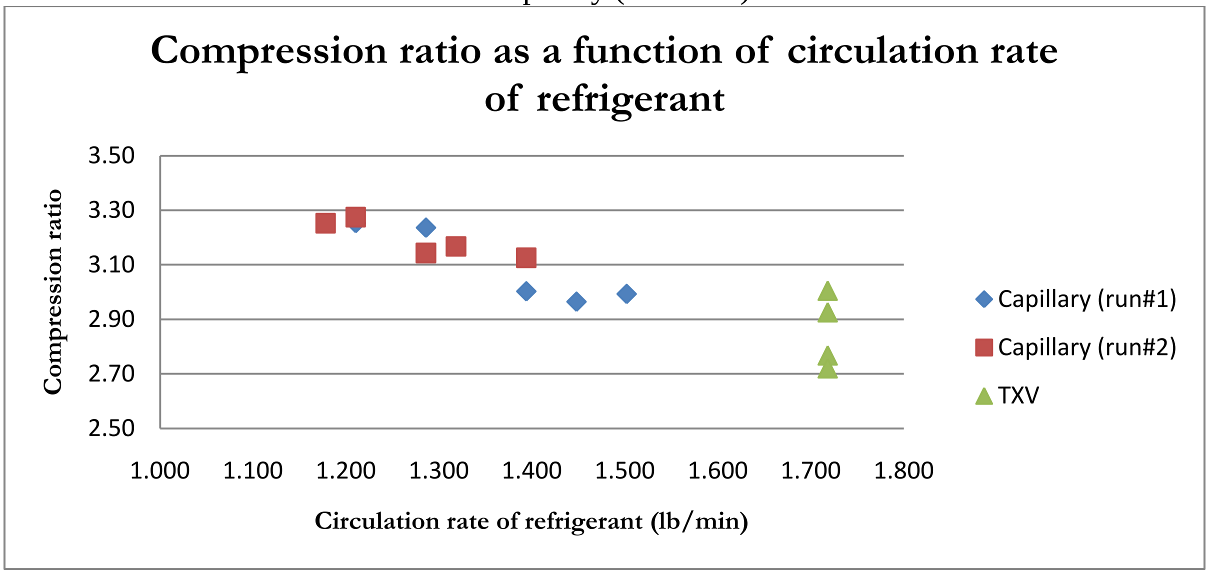
Figure#8: Representation of an ideal vapour-compression cycle on the temperature-entropy diagram.



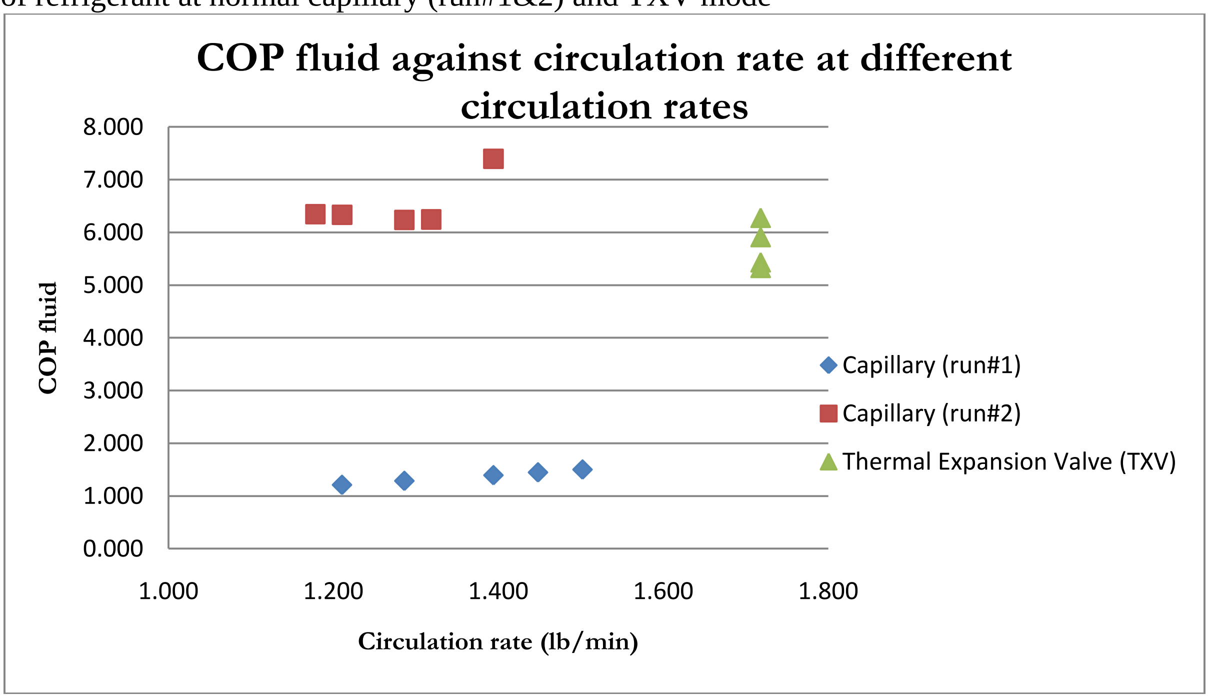
Figure#9: Plot for the cycle efficiency as a function of circulation rate of refrigerant for different cirulation rates at normal capillary (run#1&2) and TXV mode.



Figure#10: Plot for the compression ratio as a function of circulation rate of refrigerant for different circulation rate at normal capillary (run#1&2) and TXV mode.



Figure#11: Plot for the coefficient of performance of fluid as a function of circulation rate of refrigerant at normal capillary (run#1&2) and TXV mode



APPENDIX B
RAW & UNESSENTIAL DATA

Table#10: The temperature and pressure at room temperature

Ambient Temperature (°F)	75
Ambient Pressure (mmHg)	756.4

Table#11: Condenser and evaporator power at different fan settings.

		S-6 Evaporator		
		High	Medium	Low
S-5 Condenser	High	110	100	95
	Medium	100	95	90
	Low	95	85	80

Table#12: Initial temperature and pressures for different strategic positions for normal capillary mode (run#1&2) and TXV mode.

	Point 1	**Point 2**	**Point 3**	**Point 4**	**Point 5**
Initial Temperature (°F)	72	71	70	72	72
Initial Pressure (Psig)	73	72	74	73	80

Table#13: Measured values for different strategic points for different flow rates at normal capillary mode (run#1)

	Point	Circulation rate (lbm/min)	Rotameter Reading (Inch)	Refrigerant level (inch)	Temp. (°F)	Pressure (psig)	Power (W)
First Flow Rate	1	1.502	13	14	60	45	900
	2	1.502	13	14	115	142.5	900
	3	1.502	13	14	98	142	900
	4	1.502	13	14	46	52	900
	5	1.502	13	14	62	48	900
Second Flow Rate	1	1.448	12.5	14.4	60	45	900
	2	1.448	12.5	14.4	132	145	900
	3	1.448	12.5	14.4	98	145	900
	4	1.448	12.5	14.4	46	50	900
	5	1.448	12.5	14.4	62	47.5	900
Third Flow Rate	1	1.394	12	15	63	44	895
	2	1.394	12	15	127	142	895
	3	1.394	12	15	96	141	895
	4	1.394	12	15	45	50	895
	5	1.394	12	15	66	48	895
Fourth Flow Rate	1	1.286	11	15.2	66	40	870
	2	1.286	11	15.2	143	141	870
	3	1.286	11	15.2	98	141	870
	4	1.286	11	15.2	42	47	870
	5	1.286	11	15.2	58	45	870
Fifth Flow Rate	1	1.210	10.3	15.4	64	40	850
	2	1.210	10.3	15.4	148	140	850
	3	1.210	10.3	15.4	96	140	850
	4	1.210	10.3	15.4	39	45	850
	5	1.210	10.3	15.4	69	42	850

Table#14: Measured values for different strategic points for different flow rates at normal capillary mode (run#2)

	Point	Circulation rate (lbm/min)	Rotameter Reading (Inch)	Refrigerant level (inch)	Temp. (°F)	Pressure (psig)	Power (W)
First Flow Rate	1	1.394	12	14.8	63	43	878
	2	1.394	12	14.8	138	143	878
	3	1.394	12	14.8	98	143	878
	4	1.394	12	14.8	44	50	878
	5	1.394	12	14.8	62	45	878
Second Flow Rate	1	1.318	11.3	15	66	42	875
	2	1.318	11.3	15	150	144	875
	3	1.318	11.3	15	100	144	875
	4	1.318	11.3	15	44	49	875

	5	1.318	11.3	15	66	45	875
Third Flow Rate	1	1.286	11	15.3	68	41	875
	2	1.286	11	15.3	152	142	875
	3	1.286	11	15.3	100	142	875
	4	1.286	11	15.3	42	46	875
	5	1.286	11	15.3	68	44	875
Fourth Flow Rate	1	1.210	10.3	15.5	69	40	868
	2	1.210	10.3	15.5	153	141	868
	3	1.210	10.3	15.5	98	141	868
	4	1.210	10.3	15.5	39	45	868
	5	1.210	10.3	15.5	69	42	868
Fifth Flow Rate	1	1.178	10	15.5	70	39	850
	2	1.178	10	15.5	154	139	850
	3	1.178	10	15.5	97	139	850
	4	1.178	10	15.5	39	44	850
	5	1.178	10	15.5	70	42	850

Table#15: Measured values for different strategic points for different flow rates at TXV mode.

	Point	**Circulation rate (lbm/min)**	**Rotameter Reading (Inch)**	**Refrigerant level (inch)**	**Temp. (°F)**	**Pressure (psig)**	**Power (W)**
High-High	1	1.718	15	15.5	50	50	900
	2	1.718	15	15.5	139	147	900
	3	1.718	15	15.5	100	147	900
	4	1.718	15	15.5	50	55	900
	5	1.718	15	15.5	44	52	900
High-Medium	1	1.718	15	15.6	48	50	902
	2	1.718	15	15.6	128	148	902
	3	1.718	15	15.6	102	148	902
	4	1.718	15	15.6	52	55	902
	5	1.718	15	15.6	44	53	902
Medium-High	1	1.718	15	15.7	50	50	920
	2	1.718	15	15.7	141	165	920
	3	1.718	15	15.7	110	165	920
	4	1.718	15	15.7	54	57	920
	5	1.718	15	15.7	44	53	920
Medium-Medium	1	1.718	15	16	48	50	930
	2	1.718	15	16	132	160	930
	3	1.718	15	16	109	150	930
	4	1.718	15	16	52	55	930
	5	1.718	15	16	44	54	930

APPENDIX C
SAMPLE CALCULATIONS

Pressure:
1 atm = 101.325 kPa = 760 torr = 760 mmHg = 14.696 psi
1 psi = 6.89 kPa = 6890 Pa
Psia = psig + 14.696 psi

Temperature:
K = °C + 273.15 = °R/1.8 = °C + 273.15 °C = (°F-32)/1.8
°R = °F + 459.67 = °F + 460 = 1.8 K °F = 1.8°C+32

Universal Gas Constant:
$R=8.314\ \frac{J*Pa}{mol*K}=0.08206\ \frac{L*atm}{mol*K}=\frac{1.987\ Btu}{lb\ mol*°R}$

Power:
1 Watt = $9.49 * 10^{-4}\ Btu/s$

All sample calculations have been calculated by using the normal capillary mode (run#2).

The calculations for the normal capillary mode (run#1) and Thermal expansion valve mode (TXV), both follow the same calculations as for normal capillary mode (run#1) and can be found in Appendix A (table and figures).

Calculations for the performance of the Refrigeration system, is characterized using the various parameters below:

1. Ideal (Carnot) Efficiency:

   To determine the ideal (Carnot) efficiency, we need to define the ratio of Q_C, which is the rate of energy absorbed in a cold reservoir per unit mass of refrigerant to the difference between Q_C and Q_H, where Q_H is the rate of energy rejected in a hot reservoir per unit mass of refrigerant. The ideal efficiency is as follows:

$$COP_{MAX} = \left(\frac{Q_C}{W}\right)_{ideal} = \frac{Q_C}{Q_H - Q_C} = \frac{T_C}{T_H - T_C}$$

$where\ T_C = 506.47\ °R, T_H = 568.45°R$

$COP_{MAX} = \frac{506.47\ °R}{568.45°R - 506.47\ °R} = 8.17$
equation (1.)

2. The actual coefficient of performance for the vapour pressure system, which is the ratio of the actual refrigerating effect and the actual work expanded by the compressor:

$$COP_{actual} = \frac{Q_C}{W} = \frac{refrigeration\ capacity}{power\ to\ the\ compressor}$$

equation (2.)

To calculate the actual coefficient of performance, we need to first calculate the refrigeration capacity or the heat loss.

$RC = Q_C = m_f(h_1 - h_4), where\ m_f = 1.502\ lb_m/min$
$h_1 = 84.28 \frac{Btu}{min}, h_4 = 31.28\ \frac{Btu}{min}$
$RC = \left(1.502\ \frac{lb_m}{min}\right)\left(84.28 \frac{Btu}{min} - 31.28\ \frac{Btu}{min}\right) = 79.60\ \frac{Btu}{min}$ equation (4.)

Substituting refrigeration capacity in equation (4.) to equation (2.) and then calculate the power to the compressor, by converting the power to appropriate units. The power consumption is obtained by subtracting the power output by the fan-setting high-high power.

$$COP_{actual} = \frac{Q_C}{W} = \frac{79.60\ \frac{Btu}{min}}{(900 - 120)watts * 9.49 * 10^{-4}\ Btu/s * {}^{60s}/_{min}} = 1.79$$

3. The coefficient of performance of the fluid equation (3.), which is based on the specific enthalpies of the working fluid at different stages of the refrigeration system:

$$COP_{fluid} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$where\ h_1 = 84.28 \frac{Btu}{min}, h_4 = 31.28 \frac{Btu}{min}, h_2 = 88.61 \frac{Btu}{min}$$

$$COP_{fluid} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{84.28 \frac{Btu}{min} - 31.28 \frac{Btu}{min}}{88.61 \frac{Btu}{min} - 84.28 \frac{Btu}{min}} = 12.24$$

equation (3.)

4. Compression efficiency:

To calculate the compression efficiency, which is defined as the ratio between the specific enthalpies of the working fluid at different stages of the refrigeration system to the work done by the compressor.

$$n_{comp} = \frac{m_f(h_2^s - h_1)}{W},$$

$$where\ m_f = 1.502 \frac{lb_m}{min}, h_2^s = 87.74 \frac{Btu}{min}, h_1 = 84.28 \frac{Btu}{min}$$

$$n_{comp} = \frac{m_f(h_2^s - h_1)}{W} = \frac{\left(1.502 \frac{lb_m}{min}\right) * \left(87.74 \frac{Btu}{min} - 84.28 \frac{Btu}{min}\right)}{(900 - 120)watts * 9.49 * 10^{-4}\ Btu/s * {}^{60s}/_{min}}$$

$$= 0.12$$

equation (5.)

5. Cycle efficiency:

To calculate the cycle efficiency, which is defined as the ratio between the actual coefficient of performance to the ideal (Carnot) efficiency.

$$n_{cycle} = \frac{COP_{actual}}{COP_{MAX}}, where\ COP_{actual} = 1.79, COP_{MAX} = 8.17$$

$$n_{cycle} = \frac{COP_{actual}}{COP_{MAX}} = \frac{1.79}{8.17} = 0.22$$

equation (6.)

6. Compression ratio:

To calculate the compression ratio, which is defined as the ratio between the pressure at point 2 to the pressure at point 1.

$$CR = \frac{P_2}{P_1}\ , where\ P_2 = 150.19\ psig\ , P_1 = 50.19\ psig$$

$$CR = \frac{P_2}{P_1} = \frac{150.19\ psig}{50.19\ psig} = 2.99$$

equation (7.)

APPENDIX D
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Refrigeration

Chemical Engineering Practice

CHG 3122

Hani Alsaed

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Shafiq Ahmad

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To: Somaieh Salehpour

From: Hani Alsaed and Shafiq Ahmad, Group 3

Date: July 14, 2010

Subject: CHG3122, Refrigeration

A refrigeration system works by taking heat from a cold reservoir and rejecting that heat to the hot reservoir. The evaporator absorbs the heat from the environment and the condenser rejects the heat to the environment. The coefficient of performance in a Carnot refrigerator does not depend on the working fluid. This is not true for actual refrigeration systems as each fluid has different irreversibilities that are inherent in the vapour-compression cycle. Since the fluid has some effect on the coefficient of performance, it is necessary to prevent air from getting into the system. This is done by choosing a fluid whose vapour pressure at the evaporator temperature is greater than atmospheric. The fluid must also be picked so that the temperature in the condenser is not too high, as this will increase operating costs. One of the most important choices for the fluid is the safety considerations. Fluid toxicity, flammability, cost and corrosion properties are very important properties when considering a refrigerant.

This experiment consisted of the Scott Air Conditioning and Refrigeration Education system with Freon R-12 as the working fluid. This fluid meets all the above recommendations for fluid selection but is no longer used as refrigerant as it is known to damage the ozone layer. The experiment was performed with the system operating in two different modes. The capillary mode, (constant enthalpy expansion) was run with varying refrigerant flow rates. It was repeated for a second run with varying refrigerant levels. The thermostatic (TXV) mode (constant entropy expansion) was run with varying condenser and evaporator fan speeds. Since the compressor could trip if the condenser and evaporator fan speeds were set to low, the low setting was never used. A separate fan was used to cool down the overheating compressor to prevent it from tripping. Although the compressor never tripped, temperature and pressure readings were taken rapidly just in case.

Some key findings from this experiment include: The coefficient of performance is proportional to the fan speed on the evaporator and condenser when the experiment was operated in TXV mode. The coefficient of performance of the system, when operated in capillary mode, was proportional to the flow rate of the refrigerant in the system. It was also observed that the level of refrigerant had an inverse effect on the coefficient of performance. As the refrigerant level increased, the coefficient of performance decreased almost linearly in all 3 experimental trials.

Equipment and Procedure:

In this experiment we use the Scott Air Conditioning Education system with dichloro-difluoro-methane (CCl_2F_2) (Freon-12) as the working fluid. A schematic diagram of the apparatus is shown in figure 1 of Appendix A.

As show on figure 1 of Appendix A, the system is composed of : a compressor, a condenser, an evaporator, a capillary, a thermostatic expansion valve, a temperature sensor, valves, a moisture and liquid indicator, a calibrated rotameter, a drier, a liquid refrigerant receiver tank, an oil and refrigerant accumulator tank and an oil storage tank. Various pressure and temperature gauges are used and are located at five places in the system: the evaporator outlet, the condenser inlet, the condenser outlet, the evaporator inlet, and the compressor inlet.

Before we begin the experiment, we record the temperature and pressure of the gauges to make sure that the system is in thermal equilibrium with the surroundings (temperature is the same in all gauges), and determine the wattage requirements of the fans in all combinations.

We run the system under two operating modes: normal capillary and thermostatic expansion. First we begin with the normal capillary mode, setting both fans to high speed; we take readings of the rotameter, refrigerant level, wattage requirements, pressure and temperature at 5 different flow rates. We have to allow wait some time during the start up to allow the system to reach steady state and to avoid air bubbles inside the rotameter. Then we switch to the thermostatic expansion TXV mode, we take the same readings for four fan settings: high-high, high-medium, medium-high, medium-medium. Finally, we switch the apparatus back to the capillary mode with both fans set to high, but this time we decrease the amount of refrigerant in the system.

We should note that the Freon-12 refrigerant used in the apparatus does contain some oil to lubricate the compressor, and thus its properties are different from those of pure CCl_2F_2 , and the necessary corrections have been made into the Excel spreadsheet.

Summary of Results:

1. Tables 1of Appendix B lists the hot T_H and cold T_C reservoirs temperatures of the Scott Air Conditioning and Refrigeration system for the first capillary. We observe that the average $T_H = 569.91°R$ and $T_C = 508.27°R$ and table 3 for the second run $T_H = 572.49°R$ and $T_C = 512.87°R$. Table 2 lists the values for the thermostatic expansion TXV mode, the averages are $T_H = 573.41°R$ and $T_C = 509.76°R$.
2. The refrigeration capacity RC values for the first capillary run, the thermostatic expansion TXV mode, and the second capillary run are listed on tables 1, 2, and 3 respectively. The average refrigeration capacity for the first run under capillary mode is $RC = 76.14\,{}^{Btu}\!/_{\min}$ and for the second run is $RC = 76.60\,{}^{Btu}\!/_{\min}$. We note that as the flow rate of the refrigerant increases the value of the refrigeration capacity also increases. As for the thermostatic expansion TXV mode the average $RC = 75.24\,{}^{Btu}\!/_{\min}$, and the maximum is reached when the condenser fan S5 is at a high velocity and the evaporator fan S6 is at high velocity. The minimum refrigeration capacity is when both the condenser S5 and evaporator S6 fans are set to medium velocity.
3. Figures 1and 3 of Appendix C plot the variation of the refrigeration capacity as a function of the refrigerant flow rate for the first capillary run and the second capillary run respectively. We observe that for both capillary runs, RC increases linearly with the mass

flow rate of the refrigerant. This is expected since the refrigeration capacity increases with the quantity of refrigerant available to remove the heat from the system. However, in the case of the thermostatic expansion TXV mode not the same linear patter is observed.

4. Tables 1, 2, and 3 lists the power to the compressor W for the first capillary run, the thermostatic expansion mode, and the second capillary run. We observe that the power input of the compressor increases as the flow rate increases in the capillary mode. In the case of the TXV mode, the maximum happens when both the condenser S5 and evaporator S6 fans are set to medium velocity. The minimum value is when the condenser S5 is high and evaporator S6 is medium.

5. Given that the actual coefficient of performance is ratio of the refrigeration capacity RC on the power to the compressor W, we expect that COP_{actual} follows the same trend as RC and W for the capillary mode. As shown on figures 16, and 17 of Appendix C the COP_{actual} increases as the flow rate of the refrigerant increases. The actual coefficient of performance values ranged from 1.817 to 1.617. In the case of the thermostatic expansion mode, the maximum is reached when the condenser fan S5 and the evaporator fan S6 are both set to high velocity, and the minimum happen when the condenser fan S5 and the evaporator fan S6 are both set to medium velocity. This can be seen in figure 15 of Appendix C.

6. The fluid coefficient of performance COP_{fluid}, for the first capillary run, the TXV mode, and the second capillary run are listed in tables 1, 2, and 3 respectively. We note that COP_{fluid} depends on the enthalpy of the working fluid at 4 points in the system. For the first capillary run, the average is 12.62, and 7.87 for the second capillary run. As for the thermostatic expansion mode, the average is 5.99.

7. The maximum coefficient of performance $COP_{\max}$ is the ratio of the energy absorbed by the cold reservoir over the difference between the energy absorbed by the hot reservoir and cold reservoir. We note that for the first capillary run, the average $COP_{\max}$ is 8.27, and 8.61 for the second capillary run. In the case of the thermostatic expansion mode, the average is 8.01.

8. Figures 7, 8, and 9 of Appendix C plot the P-H, T-S, and T-H diagrams for the first capillary run. These diagrams are in accordance with the ideal isenthalpic vapour-compression refrigeration cycle diagram.

9. T-S, T-H and P-H diagrams for the first run of the TXV mode is shown on figures 10-13. These diagrams exhibit the properties of an isentropic vapour-compression refrigeration cycle.

10. Tables 1, and 3 of Appendix B, list the compression efficiency and overall compression efficiency for the first and second capillary runs respectively. The average η_{comp} for the first capillary run is 0.10 as opposed to 0.16 for the second run. However the cycle efficiency η_{cycle} for the first capillary run is 0.21 as opposed to 0.19 for the second run. Table 2 shows the value of the values of η_{comp}, and η_{cycle} for the TXV mode. The average η_{comp} is 0.19 and η_{cycle} is 0.22.

11. The compression ratio CR for both capillary runs is shown on figure 18 of Appendix C. There is linear relationship between CR and the refrigerant flow rate, CR decrease as the refrigerant flow rate increases. For the TXV mode, the compression ratio is at its minimum when the condenser S5 and evaporator fan S6 are set to high, and the

maximum happen when the condenser fan S5 and the evaporator fan S6 are set to medium.

Discussions

The capillary runs demonstrated that when the fluid flow rate is increased, the refrigeration capacity of the system will also increase. This is analogous to a heat exchanger where the flow rate of the fluid allows for more heat transfer to the fluid. With an increase in the Reynolds number (due to the increase in fluid flow), an increase in the Nusselt number will occur. Better heat transfer is achieved as the Nusselt number determines the convective heat transfer coefficient to the fluid.

The actual coefficient performance is plotted as a function of the level of refrigerant for both capillary runs on figures 4, 5 of Appendix C. We observe for the first capillary run that COP_{actual} is higher when the level of the refrigerant is low and then it starts decreasing as the level of the refrigerant increases. The same trend is observed in the second capillary run. Figure 6 shows the COP_{actual} for the TXV mode, in general the coefficient of performance is higher than for the capillary mode.

The T-H, T-S diagrams plot the refrigeration cycle's profile and are in accordance with theoretical expectations. In the capillary mode the expansion is isenthalpic, whereas the thermostatic expansion mode is isentropic. According to our experimental data show on figures 7-13, the system is not perfectly ideal and there are deviations from both isentropic and isenthalpic profiles. Although deviations do exist, they are minor and the isenthalpic and isentropic assumptions still hold. This is due to the fact that ideal systems are hard to achieve and sometimes impossible experimentally.

The drop in refrigeration capacity due to the decrease of the condenser fan setting was much larger than when the evaporator fan setting was decreased. The increase in refrigeration capacity with the increase of condenser fan speed follows the convective heat transfers principles. As the fan speed is increased, the condenser is better able to reject heat to the environment. This allows for better condensation of the two phase fluid that leaves the compressor. When the condenser is able to reject more heat, the compressor does not need to compress the fluid as much as more of the two phase fluid will condense in the condenser. The TXV mode also demonstrated a higher compressor power requirement when the fan speeds were decreased (Appendix C – Figure 14). The observed trend in Figure 15 of Appendix C shows that the work needed in the compressor is reduced when more heat is rejected from the condenser to the environment. Since the compressor is run using an electric motor, the energy requirements of the system can be reduced with a better means of heat transfer at the condenser.

The ability for the condenser to reject heat to the environment played a major role in the effectiveness of the system. The work requirements for the compressor are at a minimum when the condenser fan is at maximum .The current refrigeration system capacity is limited since air has a poor convective heat transfer coefficient compared to other fluids. The use of water as a medium for the condenser to reject heat to would have increased the performance of the system drastically. This is one of the many reasons why geothermal heat pumps are becoming more popular as they can provide the same refrigeration capacity but require less work input to the compressor. The water used in a typical geothermal system is around $15^{O}C$ while on a hot summer day, the ambient air temperatures can exceed $30^{O}C$. It can be difficult for an air conditioning unit to reject heat to the environment when the environment is already hot.

Although our experimental results are overall in accordance with theoretical data, a certain amount of variation exists due to in part to the fluctuation in the level of refrigerant during the experiment. Other sources of error include taking temperature and pressure readings before the system reached steady state, and the precision of the measuring devices. To decrease the experimental error, we could increase the number of data points for each run to confirm our results.

Answers to Question Set 1:

1. We use the temperature at point 4 (after the expansion valve) as a basis for the corrections of temperatures and pressures at other points to take into account the oil mixed with the refrigerant. The excel program computes the saturation pressure at this point and use it to correct the pressure, enthalpy, and entropy at the rest of the points. The properties at point 4 are obtained using the equation $M = (1 - x^{V})M^{L} + x^{V}M^{V}$ where M is either H or S, V is the vapour mole fraction, and L is the liquid mole fraction. For the capillary expansion mode, we assume the expansion to be isenthalpic, and thus the value of enthalpy is the same at points 3 and 4. In the case of the thermostatic expansion (TXV) mode the expansion is isentropic and the value of entropy is the same at points 3 and 4. Although we should note that in practice theses modes are not perfectly isenthalpic or isentropic.

2. The circulation rate of the refrigerant ($\dot{m}_f$) tend to increase the performance of the refrigeration system. As shown on figures 1 and 3 of Appendix C, the refrigeration capacity increases in a linear fashion with mass flow rate of the refrigerant for both runs of the capillary mode. As shown on figures 16, and 17 the actual coefficient of performance of the system increases with circulation rate for both capillary runs. This is expected given that the refrigeration capacity increases and COP_{actual} is a function of Qc according to Eq. (2) of the lab manual. However for the TXV mode the performance of the refrigeration didn't vary with circulation rate of the refrigerant.

3. The effect of level of refrigerant on the system is observed in figures 4 and 5 Appendix C. A mostly linear downwards trend is observed in the coefficient of performance as the amount of refrigerant is increased. Both capillary trials and the TXV trial show this relationship.

4. P-H and T-S diagrams for the first run of the TXV mode are presented in figures 10-13 in Appendix C. To simulate an increase in ambient room temperature, all temperature values were increased by 5 degrees Rankin and the excel applet was run to re-calculate entropies and enthalpies of the system. The following observations can be seen on the new P-H and T-S diagrams (figures 11 and 13 respectively):

    - The system is closer to the two phase region at point 3 in the P-H diagram when the temperature is increased.
    - The temperature difference between 3' and 3 on the T-S diagrams is shorter if the ambient temperature is increased.

The effects of an increased ambient temperature seem to have the greatest effect at point 3. This is located right after the condenser in the refrigeration system. These results demonstrate that an increase in lab temperature might reduce the ability for the condenser to condense the two phase fluid since the P-H diagram shows point 3 closer to the two phase region. This is due to its reduced ability to transfer heat to the environment as the temperature difference from the condenser to the environment is now 5 degrees less.

Conclusions and Recommendations

The Scott Air Conditioning and Refrigeration Education system showed very predictable results. When the system was operated in the capillary mode, the refrigeration capacity increased in an almost linear trend with an increase in refrigerant flow rate. This trend was observed in both capillary runs.

The level of refrigerant appeared to have a negative effect on the two capillary runs. As the level of refrigerant was increased a downward linear trend is observed in the refrigeration capacity. This might occur because the refrigerant flow rate is at a minimum when the fluid level is at its highest value.

The TXV mode also provided very predictable results. As the fan speed was decreased, the refrigeration capacity also decreased as with the coefficient of performance. When the fan settings on the condenser were changed the system noticed the largest change in refrigeration capacity. The work required by the compressor was inversely proportional to the fan speed. As the fan speed was decreased, the compressor required more power to compress the two phase fluid.

To increase the accuracy of the results a few recommendations are noted:

- The use of a data logger with pressure and temperature sensors will eliminate rounding error associated by visual inspection of gauges.
- The compressor was very close to overheating and a fan was required to cool the unit. Instead of this set up we recommend using the cool air from the evaporator to cool the unit. This might allow the TXV mode of the experiment to be run with a low fan setting on the condenser.
- Both the evaporator and condenser have a copper pipe replaced with a transparent plastic (or glass) pipe to allow for visual inspection of the quality of the fluid. Although this allows for very good visual observations of the system, it reduces the efficiency of the units. If the evaporator is ever to replaced due to damaged fins, it is recommend the unit be replaced with one that does not have the transparent heat pipe

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Refrigeration

Chemical Engineering Practice

CHG 3122

Danielle Major
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Table of contents

To: Dr. Macchi
From: Danielle Major and Nasima Mohammed, Group 2
Date: June 20th, 2006
Subject: CHG3122, Refrigeration Laboratory

Refrigeration is a process that consists of removing heat from one media and transferring it to another.

This experiment consists of running a refrigeration cycle in order to attain specific objectives. The first objective is to obtain the refrigeration capacity and performance coefficients for the refrigeration unit as a function of circulation rate of refrigerant and heat load. The other objective is to compare the performance of the refrigeration unit in normal capillary mode and thermostatic expansion valve (TXV) mode.

The experiment was set up with a Scott Air Conditioning and Refrigeration Education system with Freon-12 as the working fluid. The temperature and pressure readings were taken at different points throughout the system that will enable us to perform the needed calculations in determining the unit performance.

The main problem that was encountered was the reading of the data due to the compressor shutting down often because of overheating. The students had to rush in taking the data which might have affected the accuracy, especially if the data was close to the saturation line, which could lead to a big difference in the thermodynamic properties between liquid and vapour.

Equipment and Procedure

This experiment was performed using a Scott Air Conditioning Education system with Freon-12 (R-12) as the working fluid (see figure __, Appendix B). The system described above is run at two different modes: normal capillary and thermostatic expansion. The system consists of: a compressor (A), a condenser (B), an evaporator (C), a capillary (D), a thermostatic expansion valve (E), a temperature sensor (F), valves (V) a moisture and liquid indicator (G), a calibrated rotameter (H), a drier (I), a liquid refrigerant receiver tank (J), an oil and refrigerant accumulator tank (K) and an oil storage tank (L). The temperature and pressure gauges are located at: point 1 – evaporator outlet, point 2 – condenser inlet, point 3 – condenser outlet, point 4 – evaporator inlet and point 5 – compressor inlet.

The circulation rate of the refrigeration operating at normal capillary mode was controlled by valve 4 (valve 5 should be closed). At the TXV operating mode, valve 4 was closed and valve 5 was opened.

At the beginning of this experiment the compressor was off and the system was in thermal equilibrium with the surroundings. The temperature and pressure readings at the five points were taken. The power consumption for the fans in 9 different settings was taken. The system was then set to the normal capillary mode (isenthalpic) by opening valve 4 and the fans set to high. The compressor was turned on and the circulation rate of the fluid was set to a level corresponding to the maximum reading of the rotameter. Five different temperature and pressure measurements were recorded at five different refrigerant circulation rates. After this first capillary run, we repeated this procedure a second time with a different amount of refrigerant.

The system was then switched to the TXV mode and the temperature and pressure readings were collected at the five different points and at 4 different fan speeds: high-high, high-medium, medium-high and medium-medium.

Summary of Results

- For the first capillary run the average values for T_H and T_C were T_H= , T_C= The average performance of the refrigeration system at the first capillary run are COP_{max}=2.658, COP_{actual}=1.512, COP_{fluid}=0.034, RC=85.891 Btu/min, η_{comp}=0.161, η_{cycle}=0.875, CR=3.616.
- For the second capillary run the average values for T_H and T_C were T_H= , T_C= The average values for performance of the refrigeration system at the second capillary run are COP_{max}=2.684, COP_{actual}=1.485, COP_{fluid}=-0.025, RC=79.729 Btu/min, η_{comp}=0.157, η_{cycle}=0.813, CR=3.452.
- For the TXV mode the average values for T_H and T_C were T_H= , T_C= The average values for performance of the refrigeration system at the TXV mode are COP_{max}=1.083, COP_{actual}=0.000, COP_{fluid}=0.0282, RC=74.52 Btu/min, η_{comp}=0.201, η_{cycle}=0.000, CR=3.336.

Discussion

Conclusions and Recommendations

References

Laboratory handout

Appendix A

Tabulated and graphical results

Table1: Power, Pressure and Temperature for Capillary first run

Circulation	Power used(watts)	Corrected Temperature (°R)					Pressure(psia)				
		1	2	3	4	5	1	2	3	4	5
C1	990	513	585	553	515	507	59.82002	178.82	175.82	66.82002	53.82002
C2	990	509	589	549	512	507	56.5584	186.5584	183.5584	63.5584	52.0584
C3	1020	508	593	547	512.5	507	58.09359	200.0936	198.0936	64.09359	53.09359
C4	1005	526	597	544	508	519	52.39592	208.3959	206.3959	59.39592	47.39592
C5	990	529	603	542	504	531	50.44054	220.4405	218.4405	55.44054	46.44054

Table2: Power, Pressure and Temperature for Capillary second run

Circulation	Power used(watts)	Corrected Temperature (°R)					Pressure(psia)				
		1	2	3	4	5	1	2	3	4	5
C1	980	510	591	556	504	505	48.44054	166.4405	164.4405	55.44054	44.44054
C2	975	509	598	554	513	505	57.63213	180.6321	178.6321	64.63213	53.13213
C3	935	528	600	549	508	529	53.39592	178.3959	176.3959	59.39592	48.39592
C4	920	532	611	547	505	532	51.41032	181.4103	179.4103	56.41032	46.41032
C5	905	523	617	546	503	532	47.48331	181.4833	179.4833	54.48331	44.48331

Table3: Power, Pressure and Temperature for TXV mode

Circulation	Power used(watts)	Corrected Temperature (°R)					Pressure(psia)				
		1	2	3	4	5	1	2	3	4	5
High-High (C1)	790	520	607	564	512	513	58.5584	161.5584	159.5584	63.5584	51.5584
High-Medium (C2)	880	509	611	563	509	511	55.41688	158.4169	156.4169	60.41688	49.91688
Medium-High (C3)	955	522	607	575	503	515	48.48331	176.4833	174.4833	54.48331	43.48331
Medium-Medium (C4)	900	513	605	573	509	513	54.41688	179.4169	177.4169	60.41688	50.41688

Table4: Enthalpy and entropy for the corresponding points and circulations for Capillary first run

Circulation	Enthalpy(Btu/lbm)					Entropy(Btu/lbm.R)				
	1	2	3	4	5	1	2	3	4	5
C1	82.9146809	89.1261132	29.19806	29.19806	82.31719	0.166843	0.162645	0.059573	0.060652	0.167325
C2	82.4858061	89.4624761	28.19433	28.19433	82.38819	0.166882	0.162734	0.057715	0.058804	0.167983
C3	82.28608253	89.607178	27.6669	27.6669	82.3464	0.16607	0.162198	0.056696	0.057755	0.167594
C4	85.03643474	89.9321216	26.91933	26.91933	84.25452	0.173013	0.162304	0.055297	0.056441	0.17308
C5	85.54053531	90.4374445	26.41141	26.41141	85.98911	0.174566	0.162544	0.054328	0.055582	0.176701

Table5: Enthalpy and entropy for the corresponding points and circulations for Capillary second run

Circulation	Enthalpy(Btu/lbm)					Entropy(Btu/lbm.R)				
	1	2	3	4	5	1	2	3	4	5
C1	82.9529013	90.4375099	29.97594	29.97594	82.42502	0.170211	0.165724	0.061034	0.062678	0.170491
C2	82.44330354	91.009964	29.43512	29.43512	82.06751	0.166504	0.165716	0.059989	0.061191	0.167032
C3	85.27914893	91.3860361	28.2165	28.2165	85.62433	0.173177	0.16649	0.057787	0.059002	0.175372
C4	85.92700952	92.9482459	27.71966	27.71966	86.13248	0.174997	0.168874	0.056866	0.058143	0.176981
C5	84.81374116	93.8291883	27.47641	27.47641	86.21311	0.174125	0.170353	0.05642	0.057746	0.177788

Table6: Enthalpy and entropy for the corresponding points and circulations for TXV mode

Circulation	Enthalpy(Btu/lbm)					Entropy(Btu/lbm.R)				
	1	2	3	4	5	1	2	3	4	5
High-High	83.94492285	93.009543	31.98455	31.98455	83.24375	0.169174	0.170388	0.064651	0.066221	0.169812
High-Medium	82.53120139	93.7220078	31.74588	31.74588	83.03161	0.16729	0.171807	0.064245	0.065912	0.169901
Medium-High	84.63154798	92.5072703	34.71799	34.71799	83.85757	0.173453	0.168477	0.069377	0.072191	0.173633
Medium-Medium	83.12856813	92.1061352	34.19787	34.19787	83.29018	0.168745	0.167619	0.068456	0.070742	0.170251

Table7: The performance of the refrigeration system in the Capillary first run

Circulation	COPmax	COPactual	COPfluid	RC(Btu/min)	ηcomp	ηcycle	CR
C1	1.3333	1.6700487	0.096192	94.02557947	0.187638	1.252536	2.9893
C2	1.23077	1.5941902	0.013992	89.75466766	0.198033	1.29528	3.298509
C3	1.2368	1.4956267	-0.00824	86.75711366	0.198359	1.20923	3.444331
C4	3.8235	1.4723964	0.159715	84.15356091	0.119671	0.385088	3.977331
C5	5.667	1.3279113	-0.0916	74.76287206	0.103069	0.234337	4.370305

Table8: The performance of the refrigeration system in the Capillary second run

Circulation	COPmax	COPactual	COPfluid	RC(Btu/min)	ηcomp	ηcycle	CR
C1	1.088889	1.5611989	0.070529	87.00936543	0.189108	1.433754	3.435976
C2	1.090909	1.5391555	0.043866	85.3431779	0.214191	1.410893	3.134226
C3	3.35	1.5191449	-0.05652	80.7778911	0.124349	0.453476	3.341003
C4	5.071429	1.4547326	-0.02926	76.1119245	0.115539	0.286849	3.528675
C5	2.818182	1.3484539	-0.15522	69.40110732	0.141725	0.478484	3.822044

Table9: The performance of the refrigeration system in the TXV mode

Fan setting	COPmax	COPactual	COPfluid	RC(Btu/min)	ηcomp	ηcycle	CR
High-High	1.372093	0.0013715	0.077353	80.29	0.220809	0.001	3
High-Medium	0.90566	-0.000817	-0.04472	72.99	0.229996	-0.0009	3.173913
Medium-High	1.173077	0.0012435	0.098274	76.59	0.161701	0.00106	3.5
Medium-Medium	0.881356	-0.00025	-0.018	68.21	0.192868	-0.00028	3.652174

Appendix B

Sample Calculations

Appendix C

Raw Experimental Data

Please see attached diskette for the raw experimental data
under filename: rawdata.xls

Appendix D

Figures

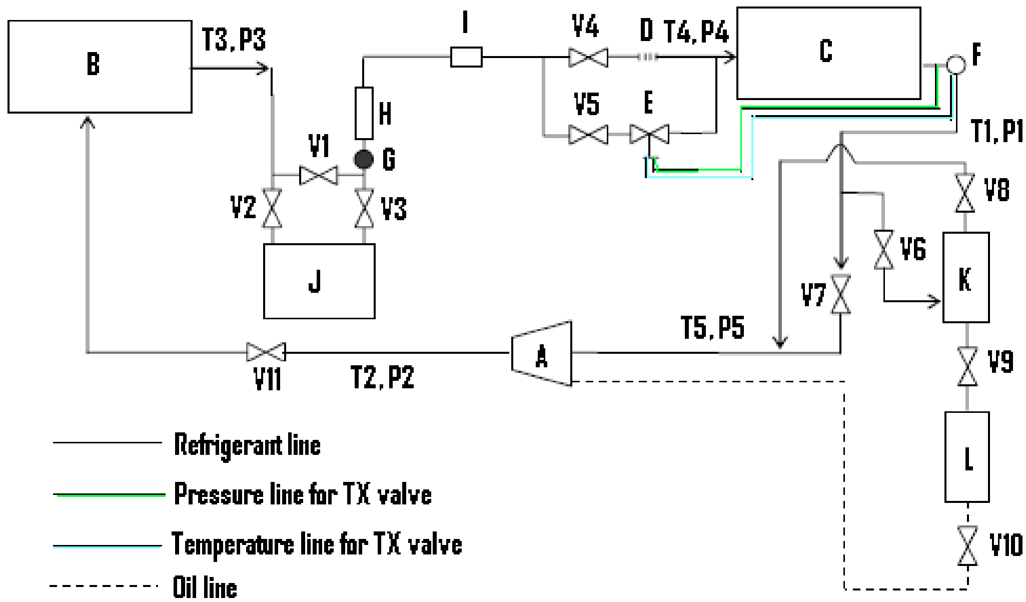


Figure 1– A schematic diagram of the Scott Air Conditioning and Refrigeration Education system used in the experiment.

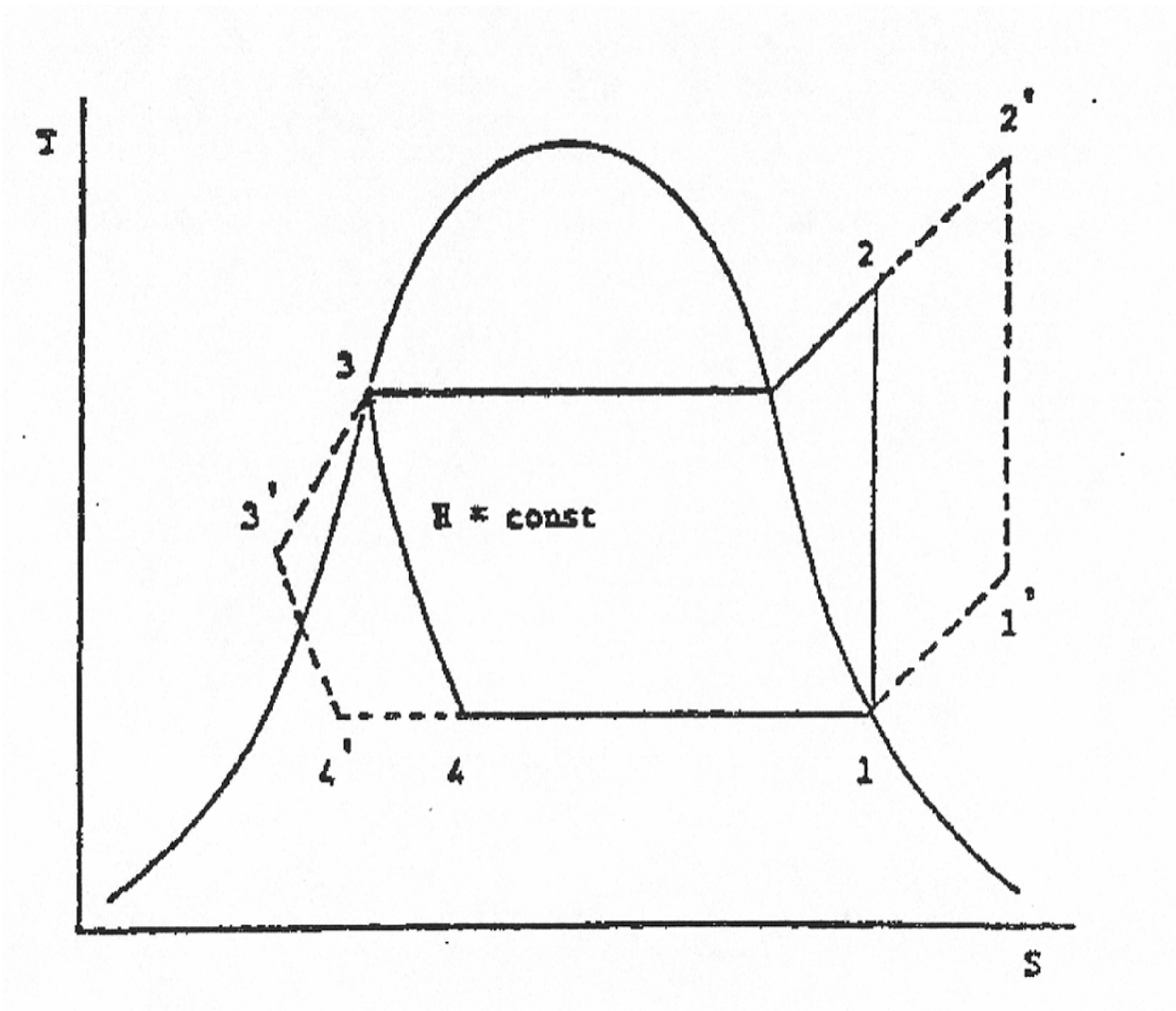


Figure2: An ideal vapour compression cycle on temperature-entropy diagram

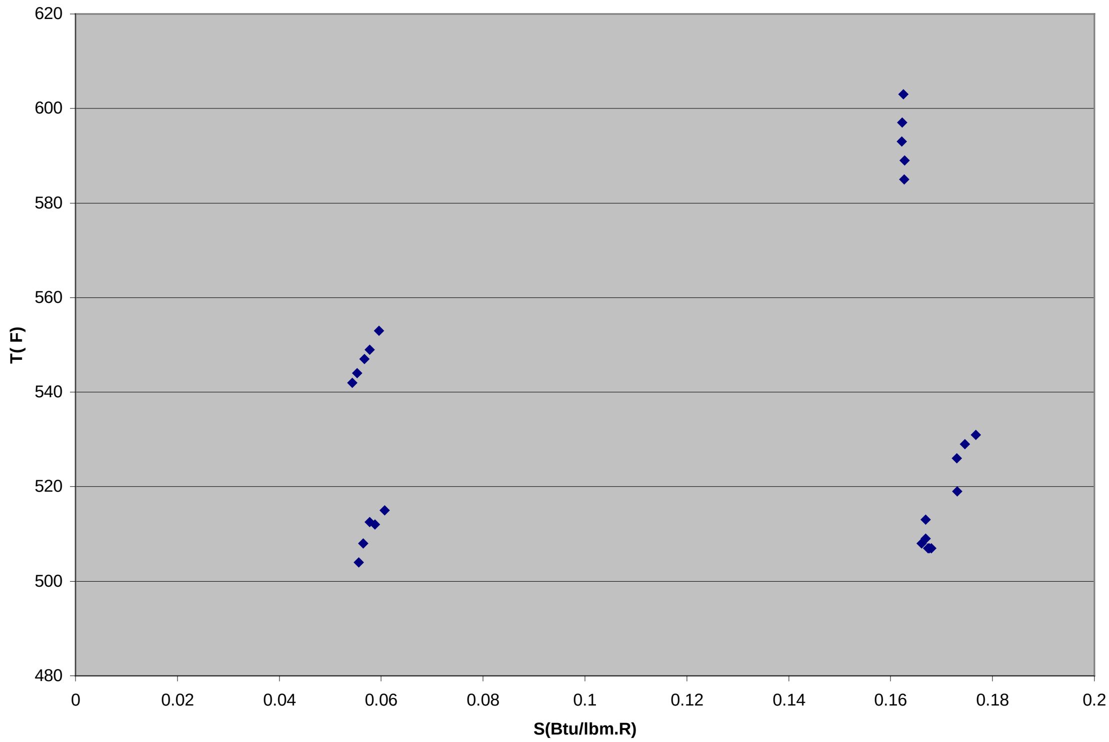


Figure3: Entropy –Temperature graph for the first Capillary run mode at the five points in the cycle.

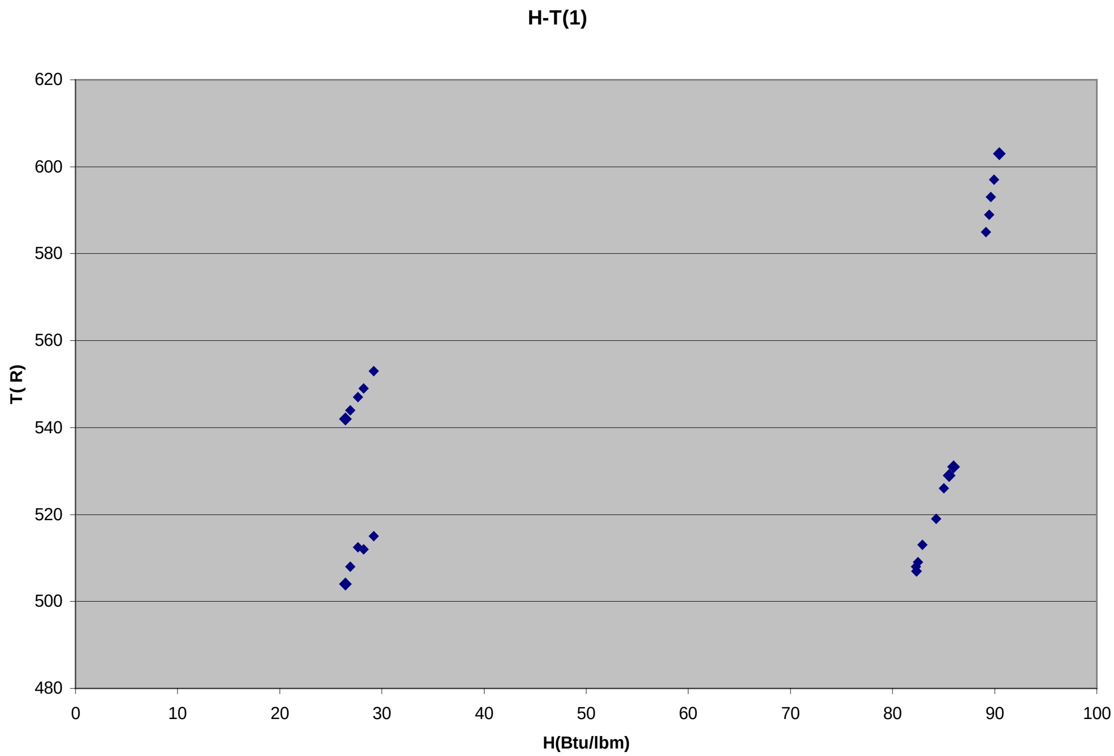
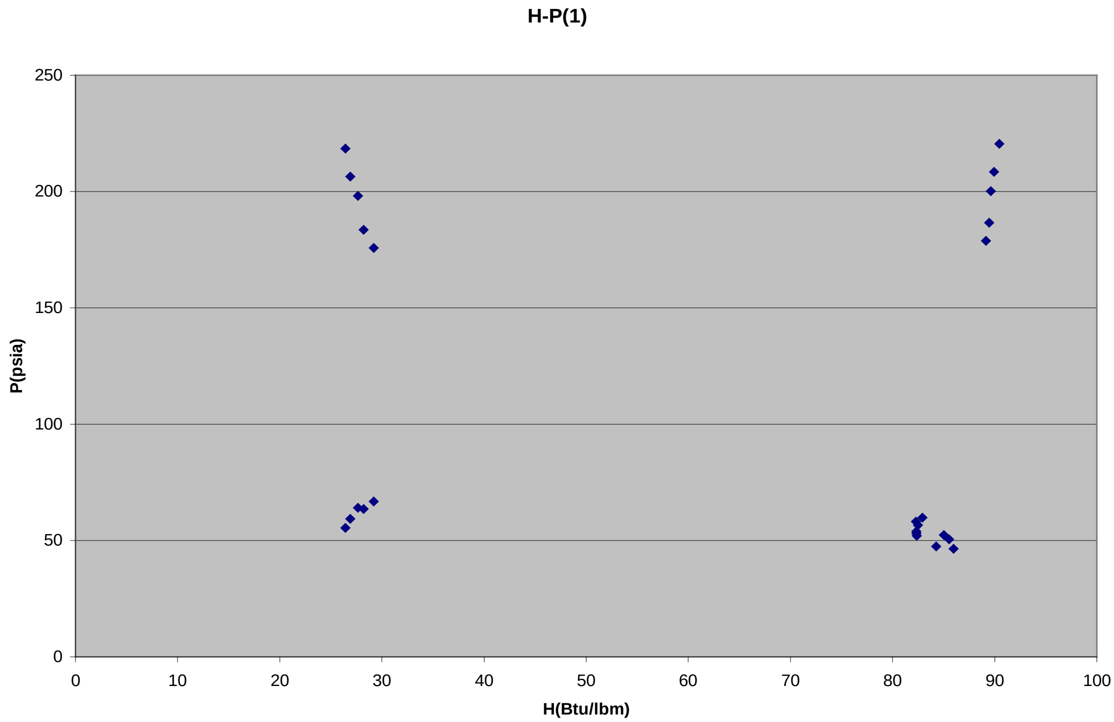
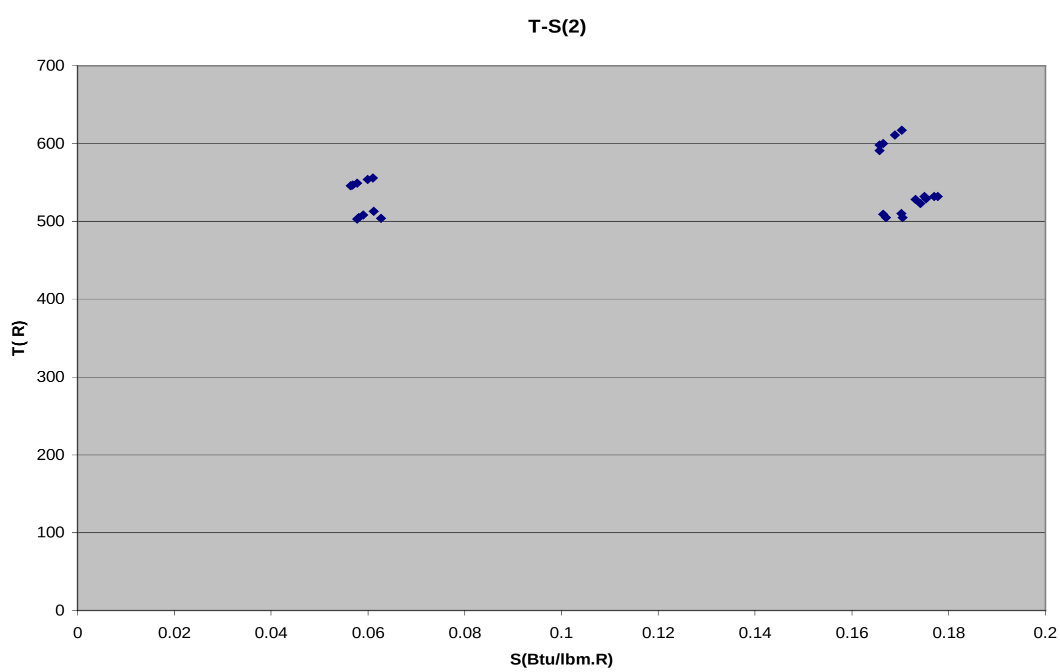


Figure4: Enthalpy –Temperature graph for the first Capillary run mode at the five points in the cycle.



<u>Figure5</u>: Enthalpy –Pressure graph for the first Capillary run mode at the five points in the cycle.



<u>Figure6</u>: Entropy –Temperature graph for the second Capillary run mode at the five points in the cycle.

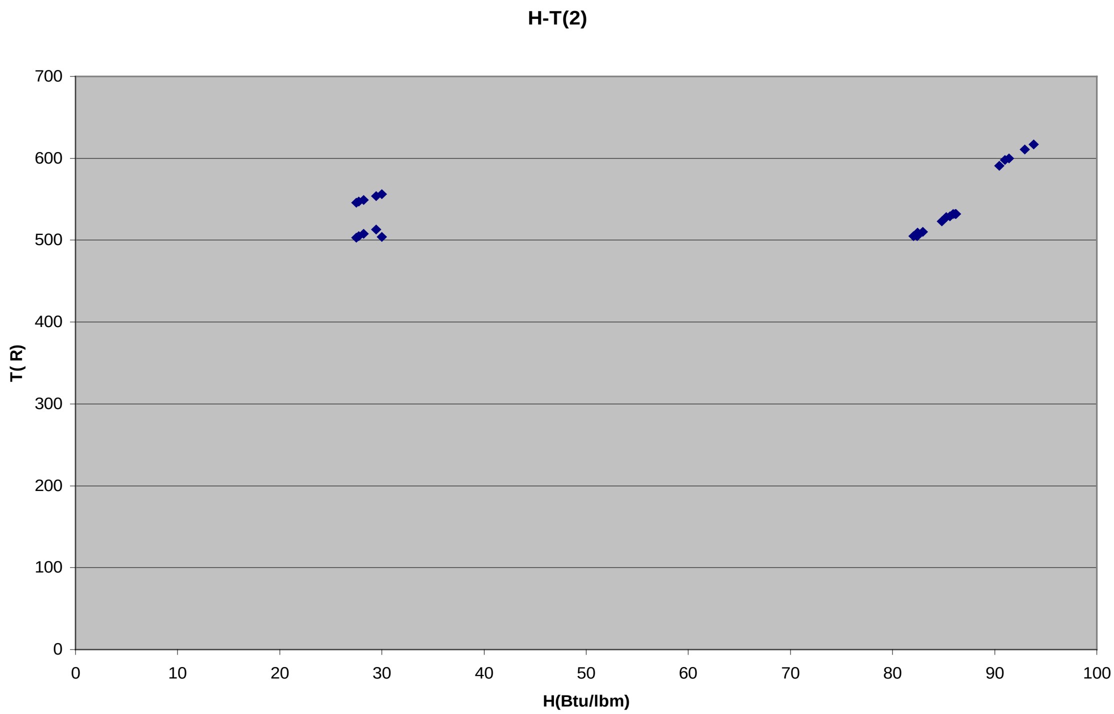


Figure7: Enthalpy –Temperature graph for the second Capillary run mode at the five points in the cycle.

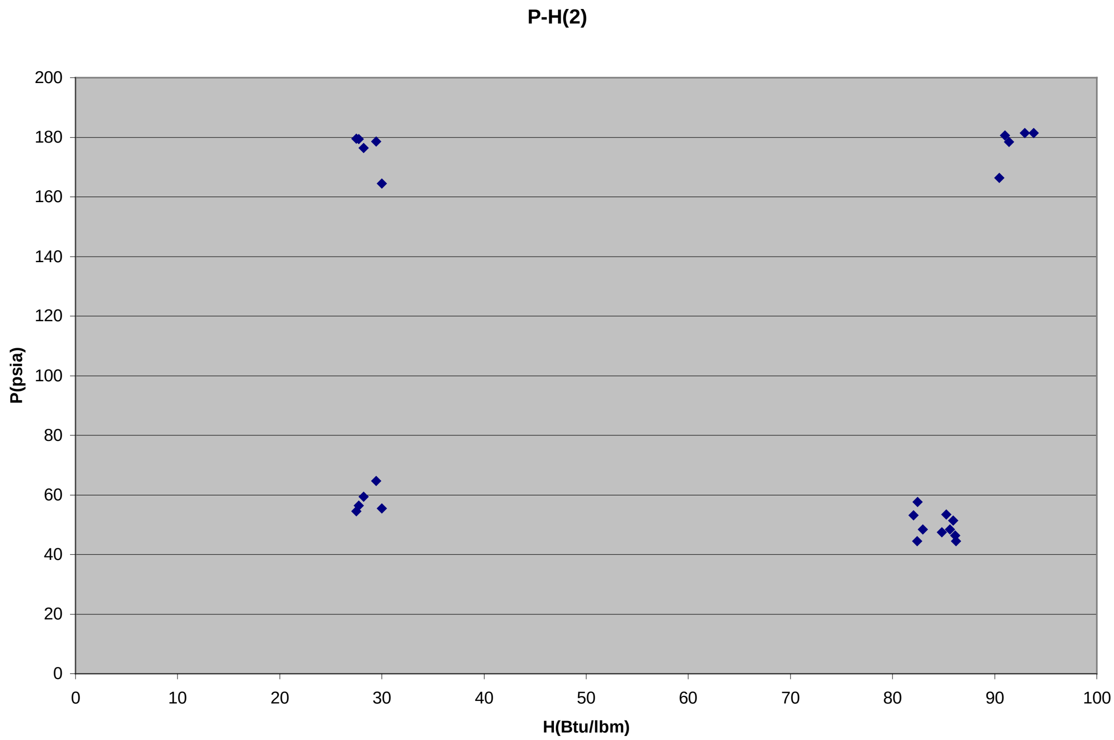


Figure8: Enthalpy –Pressure graph for the second Capillary run mode at the five points in the cycle.

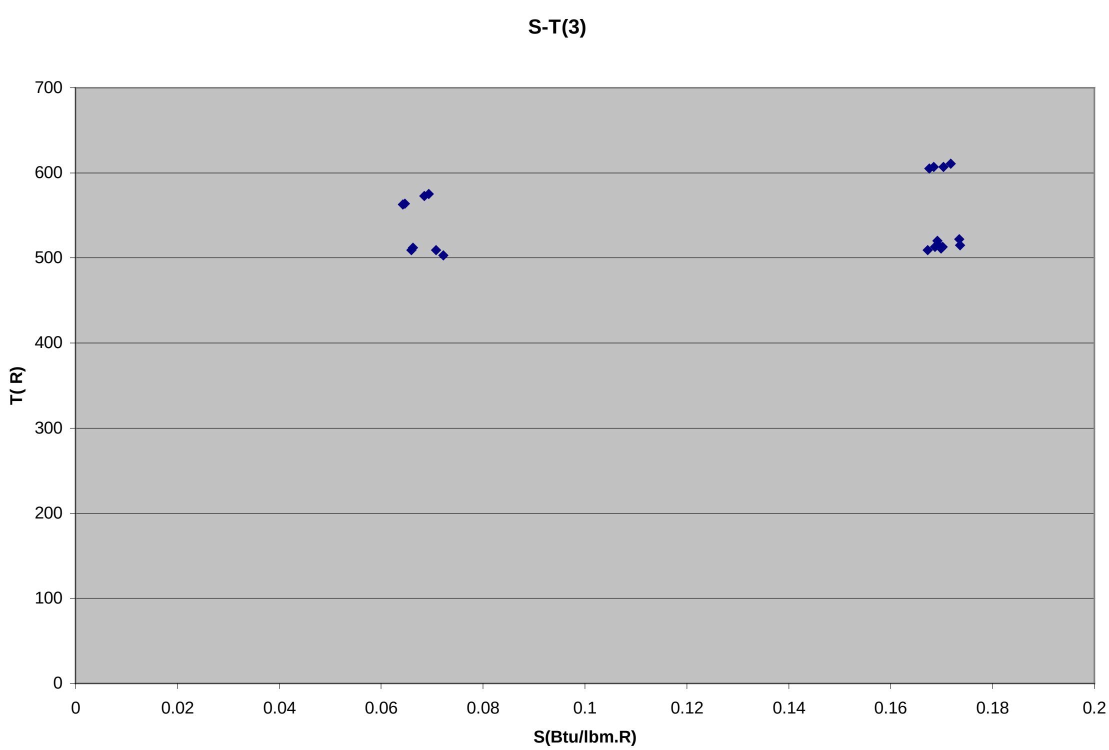


Figure9: Entropy-Temperature graph for the TXV mode at the five points in the cycle.

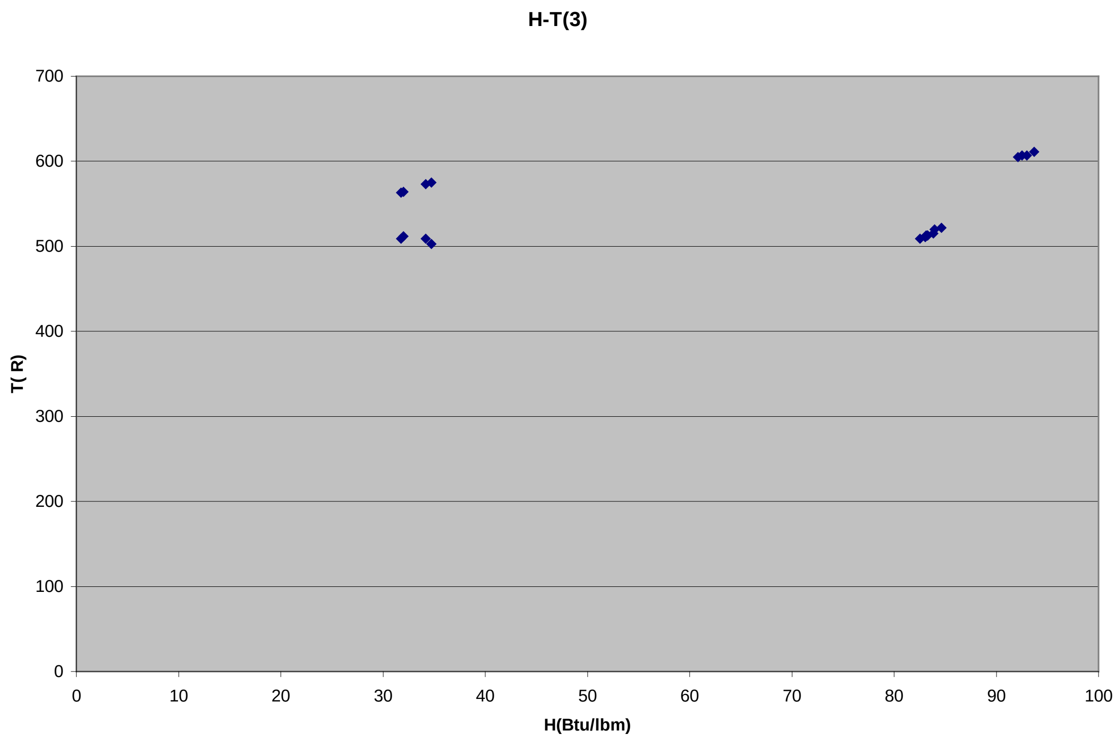


Figure10: Enthalpy-Temperature graph for the TXV mode at the five points in the cycle.

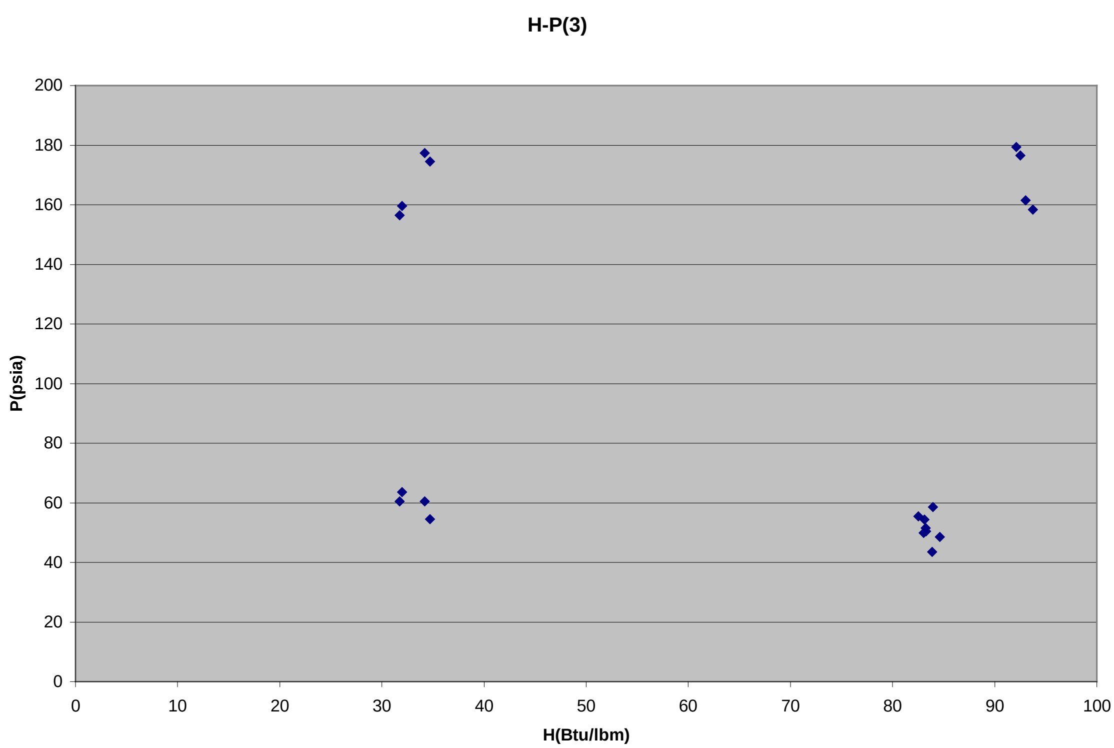


Figure11: Enthalpy-Pressure graph for the TXV mode at the five points in the cycle.

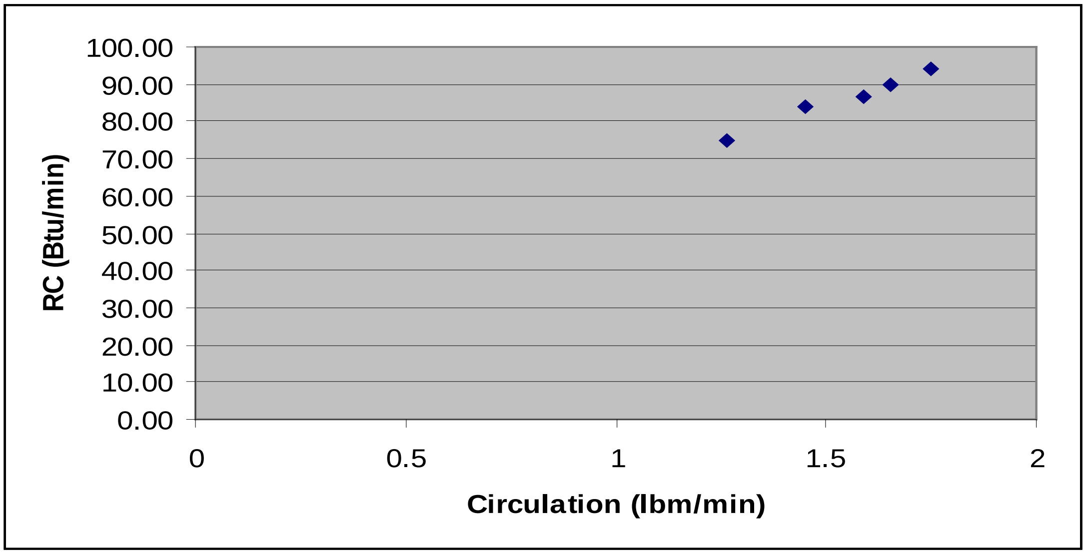


Figure 12: Experimental Refrigeration Capacity versus refrigerant circulation rate for the first run in the Capillary mode.

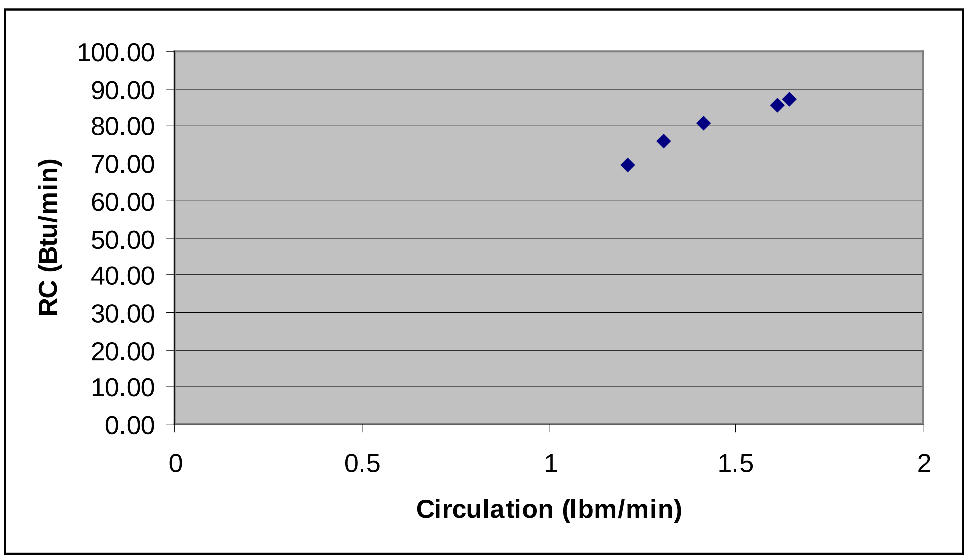


Figure 13: Experimental Refrigeration Capacity versus refrigerant circulation rate for the second run in the Capillary mode.

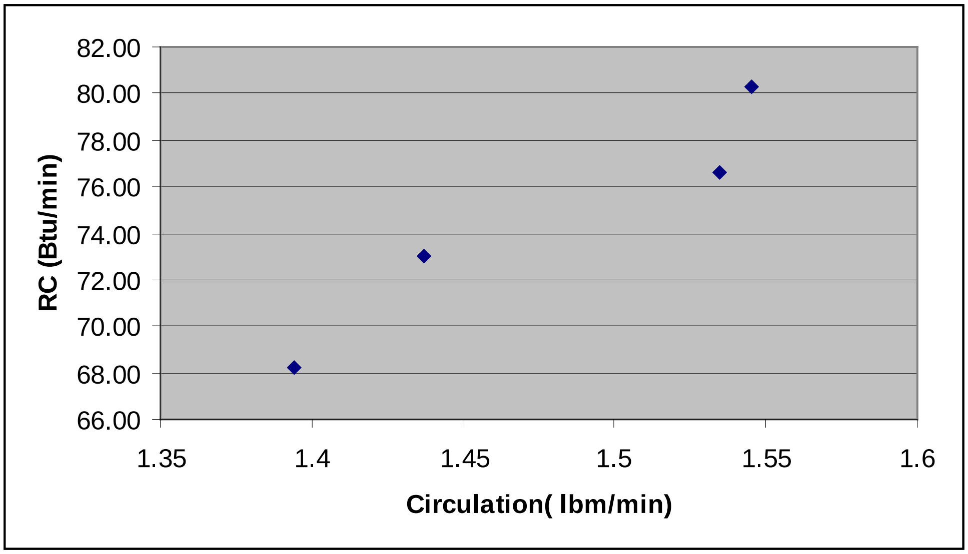


Figure 14: Experimental Refrigeration Capacity versus refrigerant circulation rate for the TXV mode.

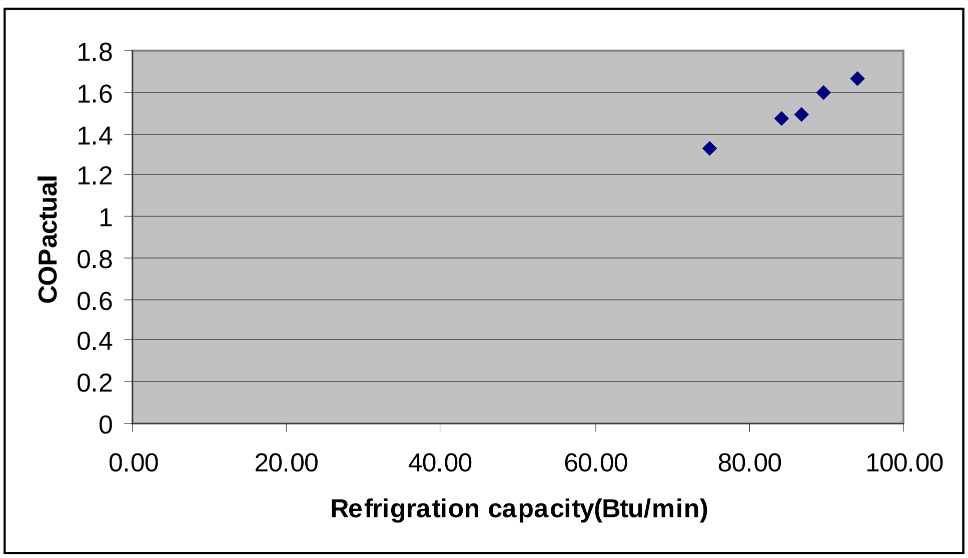


Figure15: The experimental actual coefficient of performance versus refrigerant capacity rate for the first run in the Capillary mode.

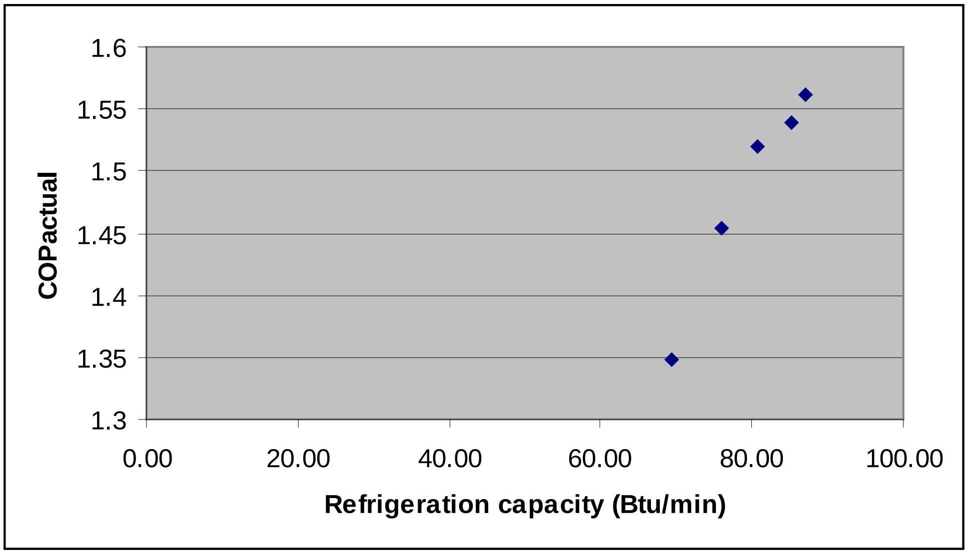


Figure16: The experimental actual coefficient of performance versus refrigerant capacity rate for the second run in the Capillary mode

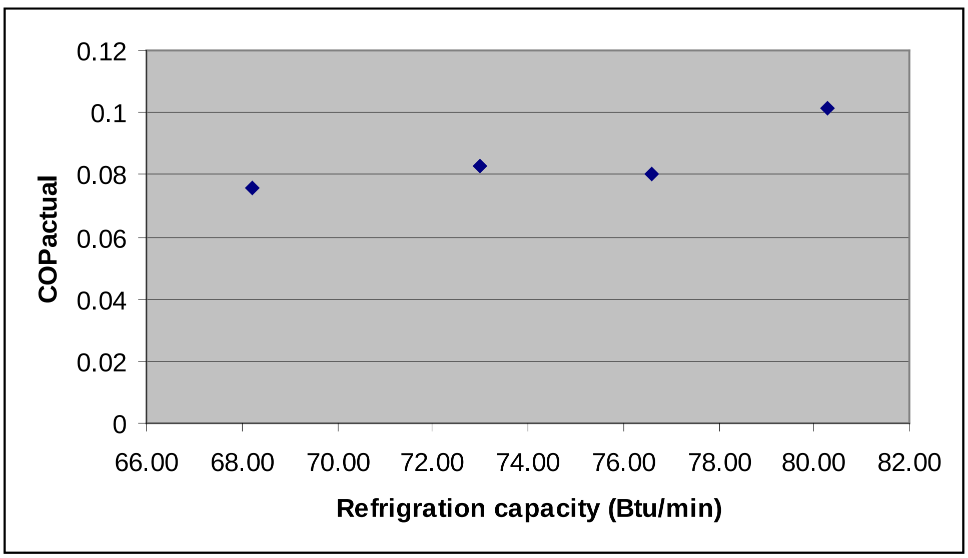


Figure17: The experimental actual coefficient of performance versus refrigerant capacity for the TXV mode

Refrigeration

Chemical Engineering Practice

CHG 3122

Ragunath Singaravelu
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Ziquan Chen
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To: Dr. Macchi & TA: Mohamed Hashi

From: Ragunath Singaravelu, Alison Reiche, and Ziquan Chen, Group 2

Date: July 14, 2008

Subject: CHG 3122, Refrigeration Experiment, Set 2

There were a few main objectives of this experiment. Firstly, the refrigeration capacity and performance coefficients for a refrigeration unit were obtained as a function of circulation rate of refrigeration and heat load. Additionally, a comparison was made between the performance of the unit in normal capillary mode and thermostatic expansion valve (TXV) mode. Furthermore, the rates of heat loss from the compressor were calculated. Lastly, the effect of fan settings used in TXV mode on the performance of the refrigeration system was also investigated.

The refrigeration unit consisted of a compressor, an evaporator, a condenser, and both a capillary and thermostatic expansion valve with a working fluid of Freon-12. By strategically placing pressure, temperature, and power gauges along the refrigeration unit, enthalpies and entropies at the different stages of the refrigeration cycle were calculated. The wattage requirements of the fans were also measured at all fan settings, which enabled calculation of the useful work. Furthermore, a rotameter allowed for the measurement of refrigerant circulation rates. For the TXV mode, runs were done at different fan settings in order to investigate the fans` effect on the performance of the unit. In normal capillary mode, for two levels of refrigerant, five experimental runs were done at different circulation rates of refrigerant. The calculated enthalpies and entropies allowed for evaluation of the actual coefficient of performance, the coefficient of performance of the refrigerant, the Carnot efficiency, the refrigeration capacity, the compression efficiency, the cycle efficiency, and the compression ratio – all of which were used to assess the performance of the unit with the two different modes of expansion.

It was found that cycle efficiency, refrigeration capacity, COP_{max}, and COP_{actual} decreased as both the evaporator and condenser fan speeds decreased when operating in TXV mode. $COP_{fluid,}$ CR, and compression efficiency did not follow the same trend. Compression efficiency, $COP_{max,}$ COP_{fluid}, RC, and CR were found to be lower for TXV mode in comparison with normal capillary mode contrary to expected results. COP_{actual} values for TXV mode and capillary mode were similar while cycle efficiency values were larger for TXV mode. Of all the parameters that were examined, only cycle efficiency followed the expected trends. Heat loss rates were found to be higher for the run with the higher refrigerant level (13.9) using normal capillary mode.

Sincerely,

Alison Reiche

Ziquan Chen

Ragunath Singaravelu

Equipment and Procedure

The previous protocol [1] was modified as follows:

The experiment was done using a Scott Air Conditioning and Refrigeration Education system with Freon-12 (R-12) as a working fluid. Three operating modes can be run on this system: normal capillary, thermostatic expansion, and reverse. Only the normal capillary and thermostatic expansions modes were used in this investigation.

The following are the system components appearing in Figure 1: compressor (A), condenser (B), evaporator (C), capillary (D), thermostatic expansion valve (E), temperature sensor (F), valves (V), moisture and liquid (G), calibrated rotameter (H), drier (I), liquid refrigerant receiver tank (J), oil and refrigerant accumulator tank (K), and oil storage tank (L). Other components include: temperature and pressure gauges located in strategic places in the system, two variable speed fans (one behind the condenser and one behind the evaporator), wattmeter, current meter, potentiometer, and several other electrical switches.

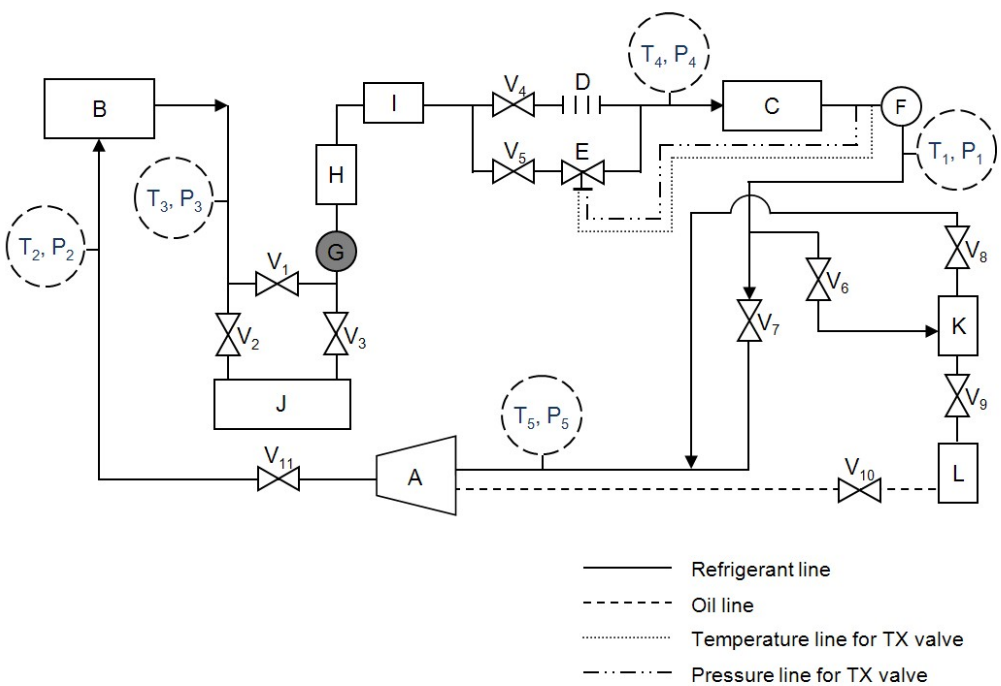

Figure 1 - Schematic representation of Scott Air Conditioning and Refrigeration system

In normal capillary mode, the expansion step takes place in a capillary feed tube. As the liquid at high pressure enters the capillary tube, pressure drops due to friction resulting in the partial evaporation of the liquid. Most of the time, the length of the capillary is matched to the capacity of the evaporator. In this system, however, the capillary tube is made to be too long to ensure that no liquid refrigerant enters the compressor. Valve V4 controls the circulation rate of the refrigerant in the system operating in the normal capillary mode.

In TXV mode, the expansion step takes place in a thermostatic expansion (TX) feed valve. The system can be switched to the TXV mode by closing the valve V4 and opening the valve V5. The TX valve is connected to a temperature-sensing bulb, which can be found at the evaporator outlet. The bulb contains superheated Freon-12. The pressure of R-12 varies with the temperature of the refrigerant leaving the evaporator. In addition, the TX valve is directly connected to the outlet of the evaporator. The pressure of R-12 in the bulb acts on a diaphragm inside the TX valve. The movement of the diaphragm is controlled by the flow of the refrigerant through the TX valve.

Precautions should be taken to prevent damage to the unit. The wattmeter (S3) should be kept in high power position. For readings lower then 750 W, switch to low power, take the readings, and then switch to high power. Start up must be done with the accumulator in line to prevent liquid carry-over to the compressor. The compressor must be stopped if the output pressure exceeds 200 psig. Finally, the fans should not be turned off if the compressor is running unless someone remains at the controls.

The flow chart (Figure 1) was studied and flow was followed through the refrigeration unit. The settings for the valves and the electrical switches given in Table 1 for setting up the system in normal capillary mode were used. The compressor (S-8) remained off. Before, the experiment, the system was left to reach thermal equilibrium with the environment. Pressure and temperature measurements were made in 5 strategic locations. Ambient pressure and temperature values were also measured. Wattage requirements for both fans in all positions were recorded with the compressor off. The compressor was then turned on with both fans (S-5, S-6) set on high and the system set to normal capillary mode. The circulation rate of the refrigerant was set at a level corresponding to the maximum rotameter reading. The compressor outlet was between 120 and 140 psig and it was made sure that no vapour bubbles were in the rotameter. The pressures, temperatures, and power consumption rates were recorded once steady state was reached.

The circulation rate of the refrigerant was then decreased. When the system stabilized, temperatures, pressures, and power consumption were recorded. The procedure was repeated with four different circulation rates of refrigerant.

The system was then switched to TXV mode with both fans at high speed. Once steady state was reached, the rotameter reading, temperatures, pressures and the power consumption were recorded. The experiment was repeated with three other fan settings.

The system was returned to normal capillary mode keeping both fans at high speed. A new amount of refrigerant was used, and the experiments were repeated with the previously used circulation rates.

Table 1 – Valve and switch positions in different operating modes for the Model 9086 Scott Air conditioning and refrigeration education system

SWITCH OR VALVE		SWITCH OR VALVE POSITION FOR OPERATING MODE				
No.	Name	Prestart	Normal Capillary	Normal TXV	Reverse Cycle	Stop[4]
V-1	Receiver Bypass		open	open		
V-2	Receiver Inlet					open[4]
V-3	Receiver Outlet					
V-4	Capillary Control		open			
V-5	TXV Control			open		open
V-6	Accumulator Inlet		open	open		open
V-7	Accumulator Bypass				open	
V-8	Accumulator Outlet		open	open		
V-9	Oil Drain		open	open		
V-10	Oil Outlet		open	open		
V-11	Reverse Capillary				open	
S-1	Key Lock	CCW(off)	CW(on)			CCW(off)
S-2	Main Power	down(off)	up(on)			down(off)
S-3	Wattmeter Select	HIGH POWER				
S-4	Voltmeter Select	LINE VOLTAGE				
S-5	Condenser Fan	O (off)	as desired		High (H) or Medium (M)	O (off)
S-6	Evaporator Fan	O (off)	as desired		High (H) or Medium (M)	O (off)
S-7	Cycle Control	NORMAL			REVERSE	NORMAL
S-8	Compressor	down(off)	up(on)			down(off)
S-10 S-11 S-12 S-13	Faulting Switches	IN				

NOTES:
1. All valves closed except as indicated.
2. Open valves as indicated before energizing switches.
3. Energize switches left to right; deenergize right to left.
4. Allow refrigerant level to build up for five minutes; then close V-2 and deenergize electrical panel.

Summary of Results

1. As the condenser fan (S5) speed decreased from high to low while maintaining the evaporator fan (S6) at medium speed, refrigeration capacity decreased from 78.85 Btu/min to 71.99 Btu/min as expected (Figure 5). COP_{max} decreased from 8.63 to 7.35 (Figure 2), COP_{actual} decreased from 1.74 to 1.38 (Figure 3), and COP_{fluid} increased from 7.83 to 8.23 (Figure 4). The values for COP_{fluid} did not follow the theoretical trend. The compression ratio was found to increase from 2.78 to 3.49, also contrary to predicted results (Figure 8). The compression efficiency was found to unexpectedly decrease from 0.223 to 0.167 (Figure 5), and the cycle efficiency lowered from 0.202 to 0.187 (Figure 7) as predicted.

2. As the evaporator fan (S6) speed decreased from high to medium while maintaining the condenser fan (S5) at high speed, refrigeration capacity decreased from 78.85 Btu/min to 75.21 Btu/min in agreement with expected trends (Figure 5). As predicted, COP_{max} decreased from 8.63 to 8.52 (Figure 2), COP_{actual} decreased from 1.74 to 1.49 (Figure 3), and COP_{fluid} decreased from 7.83 to 7.76 (Figure 4). The compression ratio was found to increase from 2.78 to 2.83 contrary to predicted results (Figure 8). The compression efficiency was found to decrease from 0.223 to 0.193 (Figure 6), and the cycle efficiency was found to decrease from 0.202 to 0.175 as expected (Figure 7).

3. When the evaporator fan (S6) speed decreased from medium to low while the condenser fan speed (S5) was maintained at a low setting, refrigeration capacity decreased from 71.99 Btu/min to 64.56 Btu/min in agreement with expected trends (Figure 5). As predicted, COP_{max} decreased from 7.35 to 6.28 (Figure 2), COP_{actual} decreased from 1.38 to 1.23 (Figure 3), and COP_{fluid} decreased from 8.23 to 6.93 (Figure 4). The compression ratio was found to increase from 3.49 to 3.52 contrary to predicted results (Figure 8). The compression efficiency was found to increase from 0.167 to 0.178 contrary to the expected trend (Figure 6), and the cycle efficiency was found to decrease from 0.187 to 0.175 as predicted (Figure 7).

4. Contrary to the expected results, $COP_{max,}$ COP_{fluid}, RC, compression efficiency, and CR were found to be lower for TXV mode (Table 2, Table 3). COP_{actual} values for TXV mode and capillary mode were similar while cycle efficiency values were larger for TXV mode (Table 2, Table 3). Of all the parameters that were examined, only cycle efficiency followed the expected trends.

5. In general, the run with the higher refrigerant level (13.9) displayed higher heat loss rates than the run with the 11.0 refrigerant level. At 13.9 (Table 3), the normal capillary mode data displayed heat loss values ranging from 8.25 Btu/min to 9.33 Btu/min for the flow rate range of 1.243 lb_m/min to 1.599 lb_m/min. While for the 11.0 refrigerant level (Table 4), the normal capillary mode data displayed heat loss values ranging from 4.65 Btu/min to 8.42 Btu/min for the flow rate range of 1.340 lb_m/min to 1.664 lb_m/min. The TXV mode data showed heat loss values in the range of 2.20 Btu/min to 5.44 Btu/min for the four different fan speed settings. No trend was seen between the rate of heat loss and the fan speeds.

Discussion

The quality of vapour (x^v) in general is calculated by the following equation:

$$H = (1 - x^v)H^l + x^v H^v \qquad (1)$$

To determine the quality for point 4, H refers to the specific enthalpy at point 4 while H^l and H^v are saturated enthalpies of liquid and vapour at point 4 respectively [2]. The saturated enthalpies are calculated using the saturation temperature at point 4. The measured temperature at point 4 is assumed to be the saturation temperature. Additionally, the expansion is assumed to be isenthalpic as the H_4 value used in this calculation is equivalent to H_3. However, due to the fact that the real processes are irreversible, this assumption may not be valid for this refrigeration unit. Furthermore, there is assumed to be no oil in the two phase zone, so the enthalpy values can be derived from the thermodynamic properties of saturated Freon-12.

The condenser and evaporator fan settings were varied in the TXV operating mode to determine their effect on the performance of the refrigeration system. The condenser fan functions to increase air flow across the coil and consequently increases the rate of heat transfer. The evaporator fan performs a similar function. Therefore at the highest fan setting, refrigeration capacity, coefficients of performance (COP), efficiency, and the compression ratio should be greatest. Upon decreasing the condenser fan (S5) speed from high to low while maintaining the evaporator fan (S6) at medium speed, refrigeration capacity decreased from 78.85 Btu/min to 71.99 Btu/min as expected (Figure 5). COP_{max} decreased from 8.63 to 7.35 (Figure 2), COP_{actual} decreased from 1.74 to 1.38 (Figure 3), and COP_{fluid} increased from 7.83 to 8.23 (Figure 4). The values for COP_{fluid} did not follow the expected trends. This can be explained by the inherent sources of error of this investigation, which will be discussed later. The compression ratio was found to increase from 2.78 to 3.49 also contrary to predicted results (Figure 8). The compression efficiency was found to decrease from 0.223 to 0.167 (Figure 5), and the cycle efficiency was found to decrease from 0.202 to 0.187 (Figure 7) as expected.

Upon decreasing the evaporator fan (S6) speed from high to medium while maintaining the condenser fan (S5) at high speed, refrigeration capacity decreased from

78.85 Btu/min to 75.21 Btu/min in agreement with expected trends (Figure 5). As predicted, COP_{max} decreased from 8.63 to 8.52 (Figure 2), COP_{actual} decreased from 1.74 to 1.49 (Figure 3), and COP_{fluid} decreased from 7.83 to 7.76 (Figure 4). The compression ratio was found to increase from 2.78 to 2.83 contrary to predicted results (Figure 8). The compression efficiency was found to decrease from 0.223 to 0.193 (Figure 6), and the cycle efficiency, as expected, was found to decrease from 0.202 to 0.175 (Figure 7).

Upon decreasing the evaporator fan (S6) speed from medium to low while maintaining the condenser fan (S5) at low speed, refrigeration capacity decreased from 71.99 Btu/min to 64.56 Btu/min in agreement with expected trends (Figure 5). As predicted, COP_{max} decreased from 7.35 to 6.28 (Figure 2), COP_{actual} decreased from 1.38 to 1.23 (Figure 3), and COP_{fluid} decreased from 8.23 to 6.93 (Figure 4). The compression ratio was found to increase from 3.49 to 3.52 contrary to predicted results (Figure 8). The compression efficiency was found to increase from 0.167 to 0.178 contrary to the expected trend (Figure 6), and the cycle efficiency was found to decrease from 0.187 to 0.175 as predicted (Figure 7).

Overall, it was predicted that the highest fan setting, refrigeration capacity, coefficients of performance (COP), efficiency, and the compression ratio should be greatest at the highest fan speed. This agreed with most of the obtained results except for CR, compression efficiency, COP_{fluid} and cycle efficiency. The deviations may be accounted for by inaccurate pressure gauges, fluctuations in flow rates, overheating of the compressor, and possible oil in the refrigerant (skewing enthalpy and entropy value calculations – though adjustments were made in calculations in order to try and account for its presence).

The most common expansion device is the thermostatic expansion valve (TXV) [3]. The TXV is considered to be an improvement on passive devices such as the capillary valve as it actively controls the refrigerant flow rate. Previous studies have shown that the energy efficiency ratio (EER) was 2-3% higher for TXV when compared to capillary valves [3]. Therefore, it is expected that coefficient of performance values, refrigerant capacity, efficiencies, and compression ratios should be higher for the TXV valve relative to the normal capillary mode. It is difficult to compare the two modes of expansions based on our experimental data due to the differences in refrigerant level and refrigerant flow rate. For this reason, the normal capillary mode with an average refrigerant level of 13.9 was used for comparison with the TXV mode which was run using a refrigerant level of approximately 14.4. For the normal capillary mode runs, the condenser and evaporator fans were both set on high, so only the TXV run using those settings could be used for the comparison. The normal capillary mode trials with refrigerant flow rates of 1.599 lb_m/min and 1.480 lb_m/min were used for comparison since they most closely matched the refrigerant flow rate for the TXV run (1.491 lb_m/min).

Contrary to the expected results, $COP_{max,}$ COP_{fluid}, RC, compression efficiency, and CR were found to be lower for TXV mode (Table 2, Table 3). COP_{actual} values for TXV mode and capillary mode were similar while cycle efficiency values were larger for TXV mode (Table 2, Table 3). Of all the parameters that were examined, only cycle efficiency followed the expected trends. Aside from the aforementioned sources of error, the assumption that the expansion was isenthalpic in the calculations may have led to skewed results.

The compressor used in this experiment was not insulated resulting in heat loss from the system. The rate of heat loss was evaluated for each experimental run. For normal capillary mode, the experimental heat loss exhibits a parabolic curve over the range of tested flow rates (Figure 9). Two separate runs were performed with different levels of refrigerant. In general, the run with the higher refrigerant level (13.9) displayed higher heat loss rates than the run with the 11.0 refrigerant level. At 13.9 (Table 3), the normal capillary mode data displayed heat loss values ranging from 8.25 Btu/min to 9.33 Btu/min for the flow rate range of 1.243 lb_m/min to 1.599 lb_m/min. While for the 11.0 refrigerant level (Table 4), the normal capillary mode data displayed heat loss values ranging from 4.65 Btu/min to 8.42 Btu/min for the flow rate range of 1.340 lb_m/min to 1.664 lb_m/min. The TXV mode data showed heat loss values in the range of 2.20 Btu/min to 5.44 Btu/min for the four different fan speed settings. No trend is seen between the rate of heat loss and the fan speeds.

Conclusions and Recommendations

1. Experimental data showed a decrease in cycle efficiency, refrigeration capacity, COP_{max}, and COP_{actual} as both the evaporator and condenser fan speeds were decreased in TXV mode. This was in agreement with the expected results. The other parameters (COP_{fluid}, CR, and compression efficiency) didn't followed predicted trends.

2. Compression efficiency, $COP_{max,}$ COP_{fluid}, RC, and CR were found to be lower for TXV mode in comparison with normal capillary mode contrary to expected results. COP_{actual} values for TXV mode and capillary mode were similar while cycle efficiency values were larger for TXV mode. Of all the parameters that were examined, only cycle efficiency followed the expected trends.

3. The run with the higher refrigerant level (13.9) displayed higher heat loss rates than the run with the 11.0 refrigerant level for normal capillary mode.

4. Runs should be done using the same refrigerant level and circulation rate for both TXV and normal capillary modes with high fan speeds, so that a more accurate comparison of individual parameters can be made.

5. Performing repeated trials for each mode with more flow rates would improve the overall reliability of the data obtained in this experiment.

References

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2. Smith, J.M., Van Ness, H.C., and M.M. Abbott, "Introduction to Chemical Engineering Thermodynamics," 7 ed. McGraw Hill, New York, NY (2005) pp. 225.

3. Kim, B. H., and D.L. O'Neal, "Effect of Refrigerant Flow Control on the Heating Performance of a Variable-Speed Heat Pump Operating at Low Outdoor Temperature," Journal of Solar Energy Engineering. 127 (2): 277-286 (2005).

Appendix

Tabulated and Graphical Results

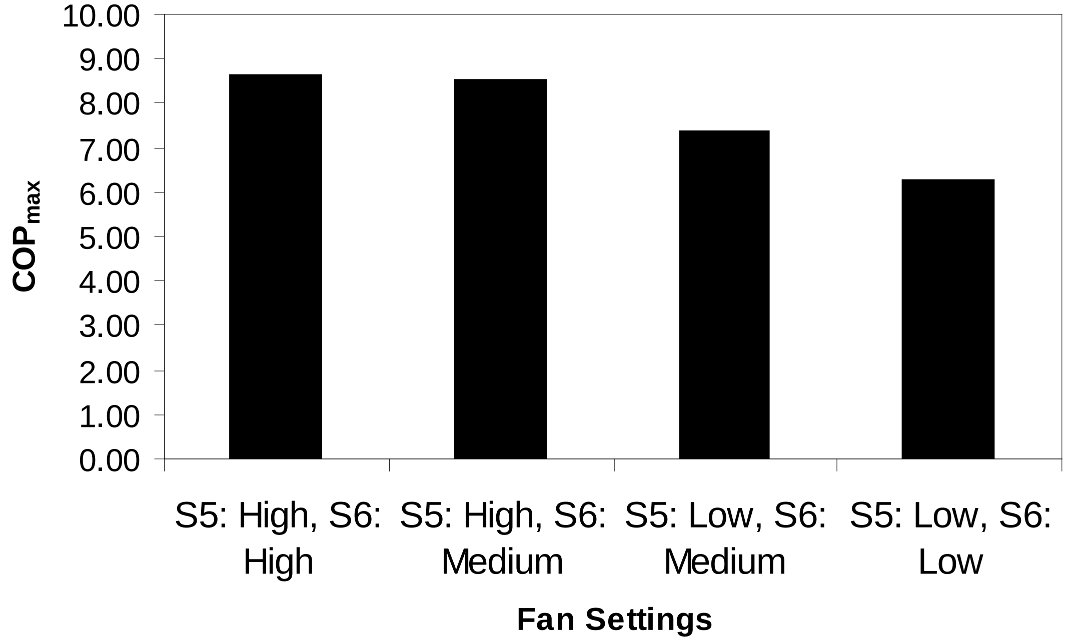


Figure 2: Ideal (Carnot) efficiency at varying fan speeds (S5: Condenser, S6: Evaporator)

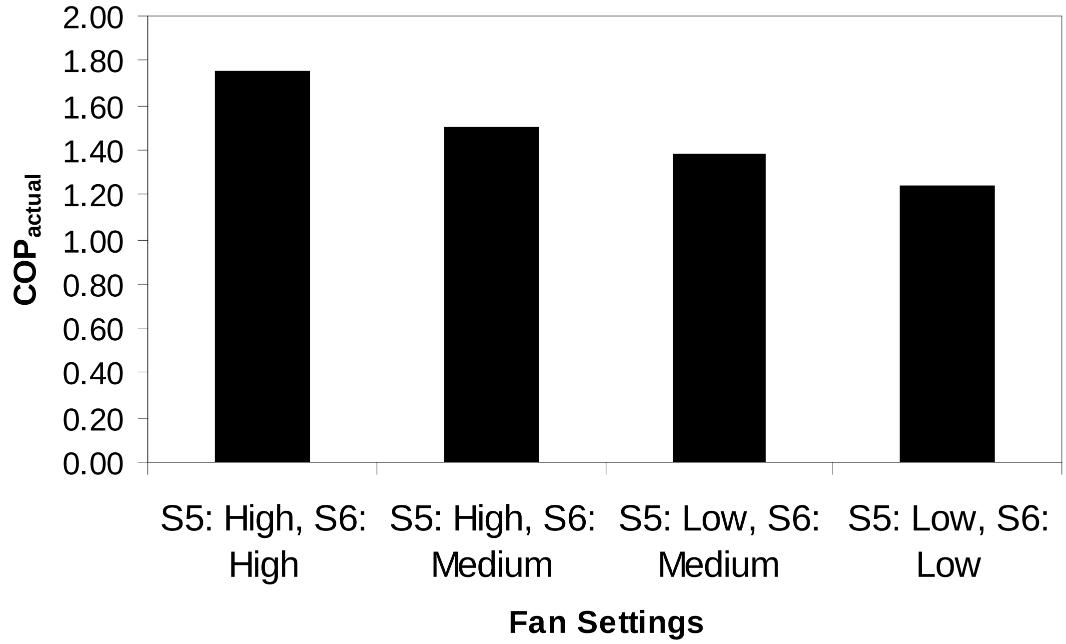


Figure 3: Actual coefficient of performance at varying fan speeds (S5: Condenser, S6: Evaporator)

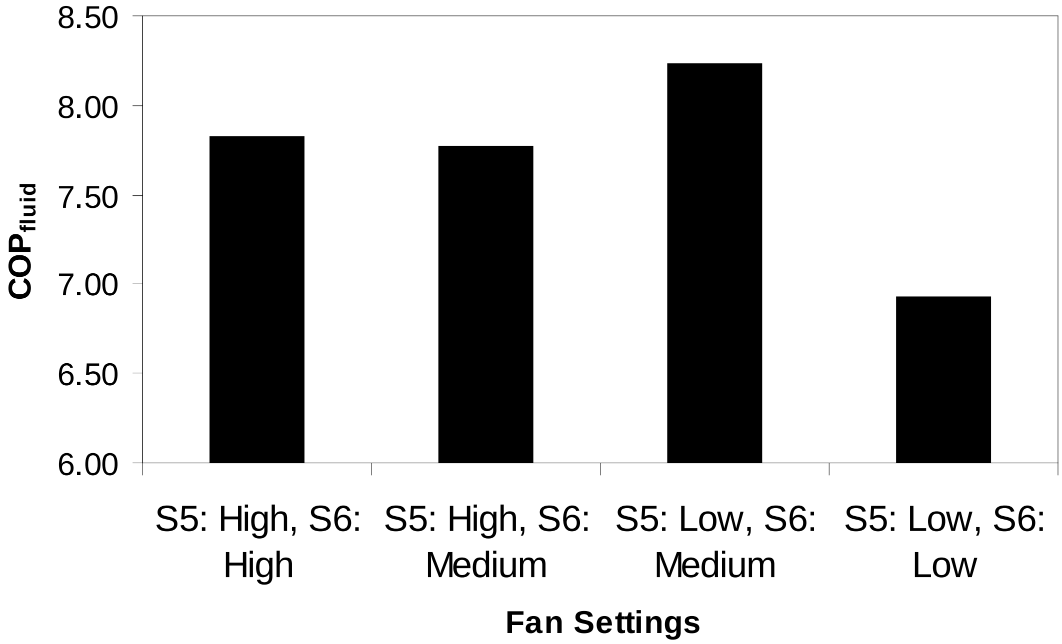


Figure 4: Coefficient of performance of Freon-12 at varying fan speeds (S5: Condenser, S6: Evaporator)

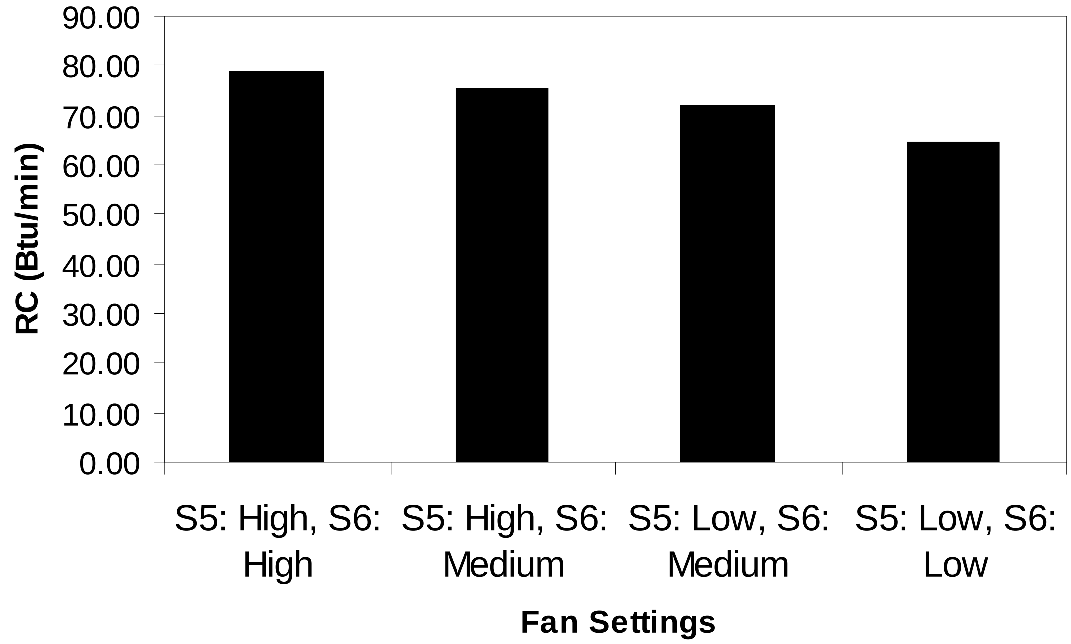


Figure 5: Refrigeration capacity at varying fan speeds (S5: Condenser, S6: Evaporator)

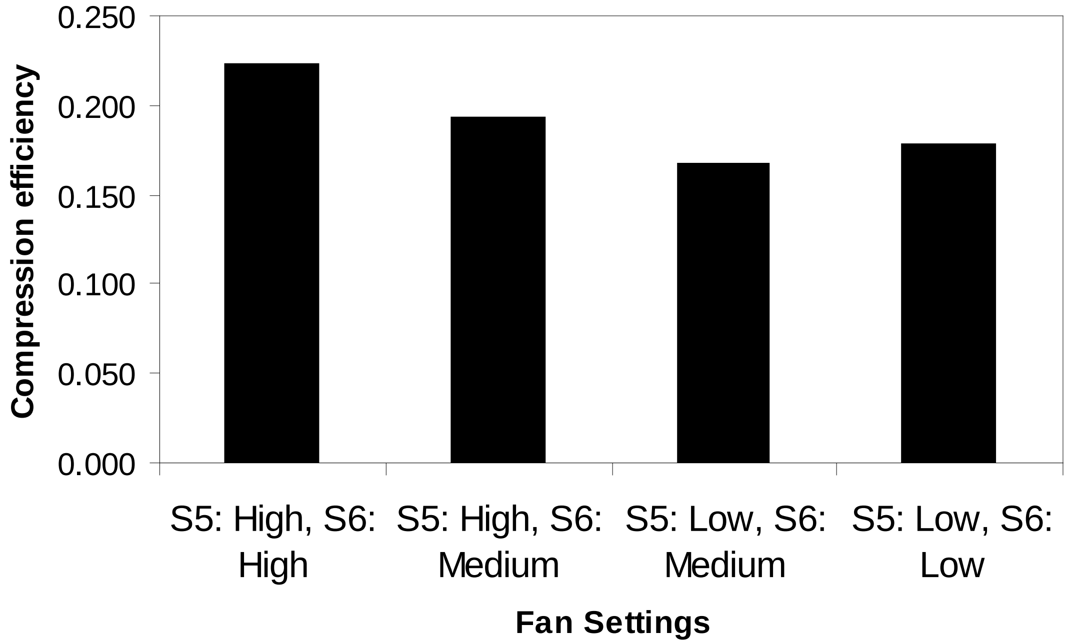


Figure 6: Compression efficiency at varying fan speeds (S5: Condenser, S6: Evaporator)

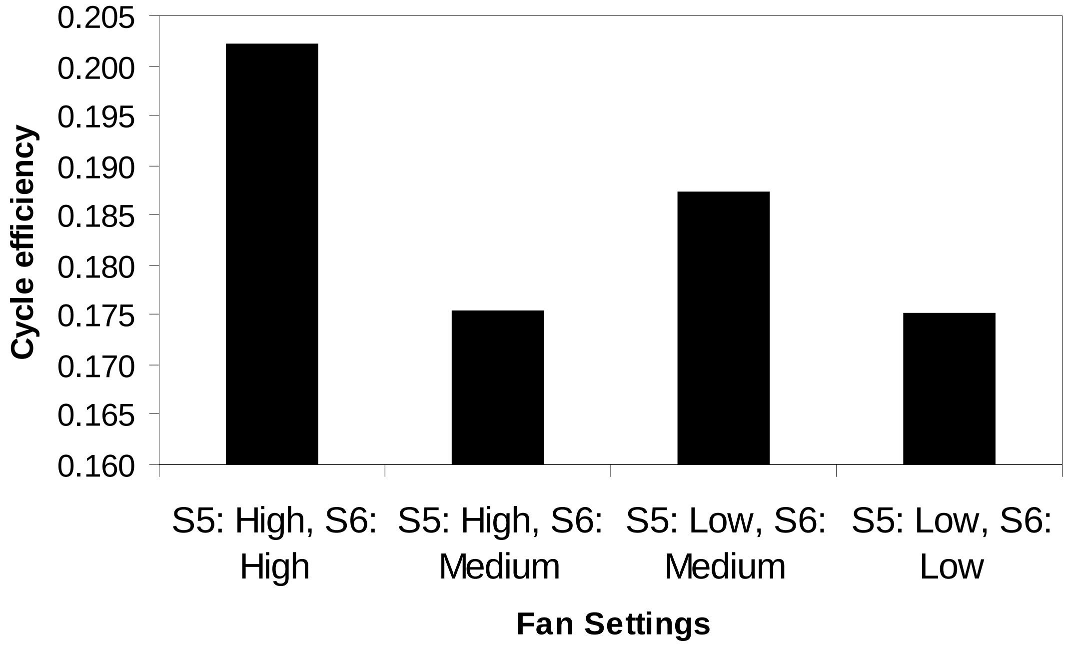


Figure 7: Cycle efficiency at varying fan speeds (S5: Condenser, S6: Evaporator)

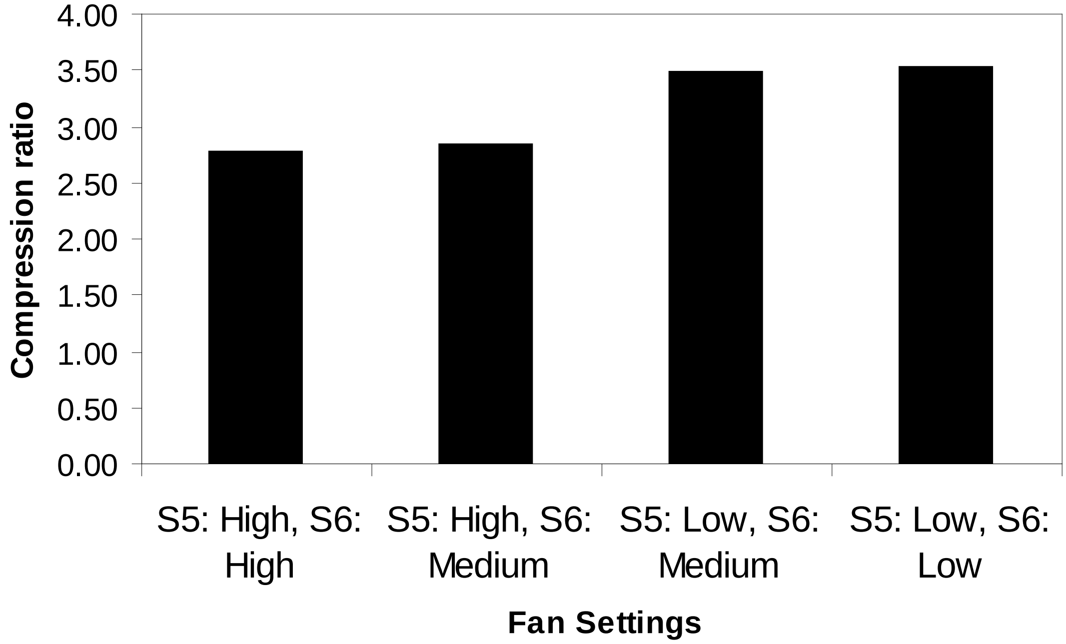


Figure 8: Compression ratio at varying fan speeds (S5: Condenser, S6: Evaporator)

Table 2 – TXV mode: parameter values at average refrigerant level of 14.4

Fan Speeds	Refrigerant flow rate (lb_m/min)	Parameters							
		COP_{max}	COP_{actual}	COP_{fluid}	RC (Btu/min)	η_{comp}	η_{cycle}	CR	Rate of heat loss from compressor (Btu/min)
S5: High S6: High	1.491	8.63	1.74	7.83	78.85	0.223	0.202	2.78	2.25
S5: High S6: Medium	1.416	8.52	1.49	7.76	75.21	0.193	0.175	2.83	2.20
S5: Low S6: Medium	1.416	7.35	1.38	8.23	71.99	0.167	0.187	3.49	5.44
S5: Low S6: Low	1.286	6.28	1.23	6.93	64.56	0.178	0.175	3.52	3.60

Table 3 – Normal capillary mode: parameter values with average refrigerant level ~ 13.9

Refrigerant flow rate (lb_m/min)	Parameters							
	COP_{max}	COP_{actual}	COP_{fluid}	RC (Btu/min)	η_{comp}	η_{cycle}	CR	Rate of heat loss from compressor (Btu/min)
1.599	10.03	1.97	20.58	90.85	0.313	0.197	2.86	9.99
1.480	9.13	1.59	15.92	81.55	0.262	0.174	2.90	8.26
1.426	8.87	1.55	16.53	79.43	0.255	0.175	2.92	8.25
1.308	8.54	1.50	18.27	75.22	0.250	0.176	3.00	8.39
1.243	8.28	1.49	22.18	73.08	0.258	0.180	3.16	9.33

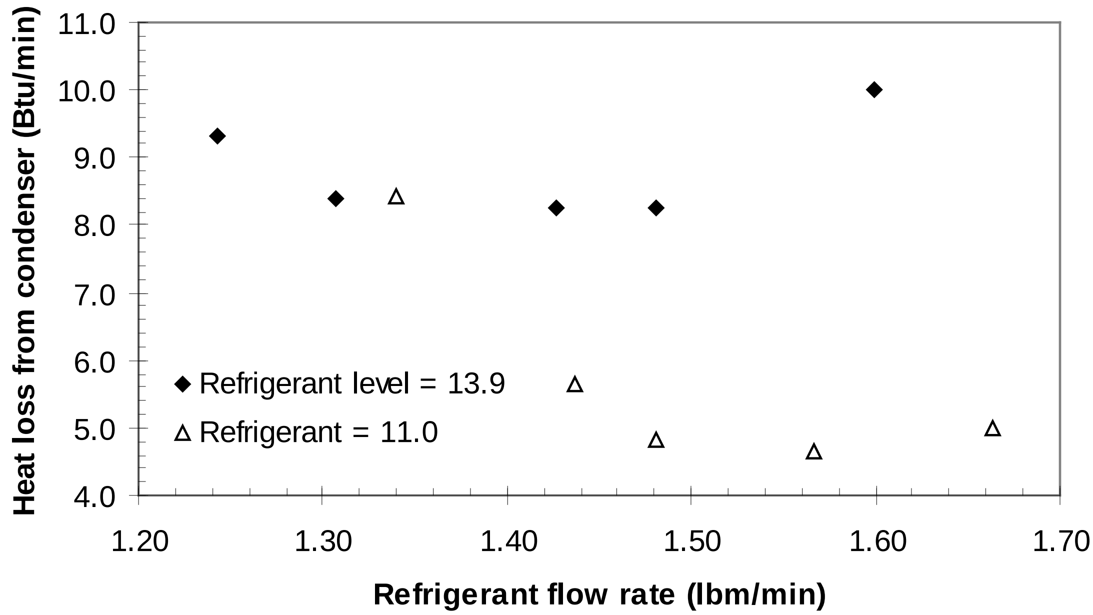


Figure 9: Heat loss from compressor at varying flow rates (S5: Condenser, S6: Evaporator) for normal capillary mode.

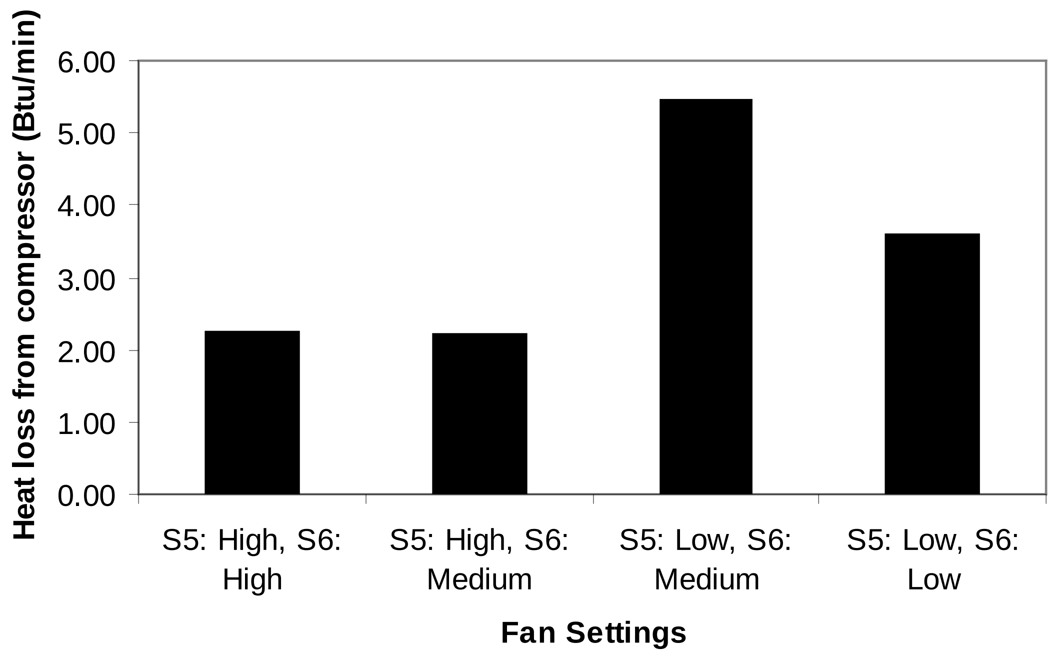


Figure 10: Heat loss from compressor at varying fan speeds (S5: Condenser, S6: Evaporator) for TXV mode.

Table 4 – Normal capillary mode: parameter values with average refrigerant level ~ 11.0

Refrigerant flow rate (lb_m/min)	Parameters							
	COP_{max}	COP_{actual}	COP_{fluid}	RC (Btu/min)	η_{comp}	η_{cycle}	CR	Rate of heat loss from compressor (Btu/min)
1.664	9.12	1.84	8.80	89.18	0.313	0.202	3.09	4.99
1.567	9.52	1.58	9.14	86.24	0.258	0.166	3.04	4.65
1.480	9.42	1.50	9.09	82.11	0.254	0.160	3.19	4.82
1.437	9.18	1.49	9.69	80.74	0.258	0.162	3.30	5.65
1.340	8.70	1.46	13.95	78.30	0.263	0.168	3.46	8.42

Raw and Nonessential Data

Table 5 – Initial temperature and pressure readings*

	Point 1	Point 2	Point 3	Point 4	Point 5
Initial Temperature (°F)	66.2	72.3	70.0	71.3	66.0
Initial Pressure (Psig)	59.0	57.5	59.9	58.0	66.0

*-raw data was taken at an ambient temperature of 73.8°F and an ambient pressure of 750.2 mmHg

Table 6 – Wattage requirements for both fans with compressor off*

S5 Setting	S6:High	S6:Medium	S6:Low
High	110 W	100 W	93 W
Medium	103 W	92 W	88 W
Low	100 W	80 W	77 W

*- S5 and S6 refer to condenser and evaporator fans respectively

Table 7 - Normal capillary mode: raw data with average refrigerant level ~ 13.9

Trial	Capillary	Point 1	Point 2	Point 3	Point 4	Point 5	Rotameter Reading	Power Reading (W)	Level of Refrigerant
1	Temperature Reading (°F)	58.7	108	85	46	66	13.9	920	13.7
	Pressure Reading (psig)	44	139	136	51	47			
2	Temperature Reading (°F)	57	112	91	46	66	12.8	900	13.9
	Pressure Reading (psig)	44	143	142	50	47			
3	Temperature Reading (°F)	59.5	114	90	45	67	12.3	900	13.9
	Pressure Reading (psig)	45	145	144	50	47			
4	Temperature Reading (°F)	65	118	86	43	70	11.2	880	14
	Pressure Reading (psig)	42	142.5	142.5	47	45			
5	Temperature Reading (°F)	68	118	83	40	72	10.60	860	14
	Pressure Reading (psig)	39	141	141	44	43			

Table 8 - Normal capillary mode: raw data with average refrigerant level ~ 11.0

Trial	Capillary	Point 1	Point 2	Point 3	Point 4	Point 5	Rotameter Reading	Power Reading (W)	Level of Refrigerant
1	Temperature Reading (°F)	45	122	90	50	49	14.5	960	11
	Pressure Reading (psig)	47	162.5	162.5	54	52			
2	Temperature Reading (°F)	45	121	84.1	49	49	13.6	960	10.9
	Pressure Reading (psig)	47	160	162	53	51			
3	Temperature Reading (°F)	44	122	82	48	52	12.8	960	11.1
	Pressure Reading (psig)	46	165	166	52	50			
4	Temperature Reading (°F)	47	123.6	81	47	60	12.4	952	11
	Pressure Reading (psig)	45	167.5	169	51	49			
5	Temperature Reading (°F)	59	125	79	44	65	11.5	940	11
	Pressure Reading (psig)	43	169	170	48	46			

Table 9 - TXV mode: raw data at average refrigerant level of 14.4

Fan Speeds	Capillary	Point 1	Point 2	Point 3	Point 4	Point 5	Rotameter Reading	Power Reading (W)	Level of Refrigerant
S5: High S6: High	Temperature Reading (°F)	44	122	92	48	45	12.9	905	14.1
	Pressure Reading (psig)	46	146	146	50	50			
S5: High S6: Medium	Temperature Reading (°F)	42	121	90	46	51	12.2	885	14.3
	Pressure Reading (psig)	44	145	145	47	48			
S5: Low S6: Medium	Temperature Reading (°F)	46	128	102	46	57	12.2	920	14.4
	Pressure Reading (psig)	45	180	180	49	49			
S5: Low S6: Low	Temperature Reading (°F)	39	128	101	43	54	11	920	14.7
	Pressure Reading (psig)	43	175	178	46	46			

Refrigeration

Chemical Engineering Practice

CHG 3122

IBRAHIM AL-MUFTAH
#3631449

To: Dr. Kruczek
From: Ibrahim Al-Muftah, Group 2b
Date: April 3, 2007
Subject: CHG 3122, Refrigeration Experiment

The main objective of this experiment is to obtain the refrigeration capacity and performance coefficients (COPs) for a refrigeration unit as a function of circulation rate of refrigeration and heat load. Another objective is comparing the performance of the refrigeration unit in two modes; the normal capillary mode and the thermostatic expansion valve (TXV) mode.

In refrigeration cycle both the hot and cold reservoirs are used and heat flows from one reservoir to the other interchangeably. Heat from the cold reservoir denoted by $\dot{Q}_C$ is absorbed at a temperature T_C. Then it is transferred to the hot reservoir with heat denoted as $\dot{Q}_H$ at a temperature T_H by using a power input 'W' to the system.

The experiment was straight forward; Freon-12 was used as a working fluid. Temperatures and pressures readings were taken at five strategic points "where pressures were corrected" that along the refrigerant's path through the system. These recordings were done twice in the capillary mode and once for the TXV mode. For the capillary mode, each run was done with different amount of refrigerant in the system, where the flow rate of the refrigerant is varied. On the other the TXV mode was run at constant flow rate with varying fan speeds across the condenser and evaporator. One of the problems that faced us was the condenser got hot very fast; therefore, it shut down. This problem led us to hurry to take the date from the 5-stratigic points in the TXV mode. Finally, the focus of this report will be on answering the questions of set 2.

Overall some key conclusions can be stated about this experiment; An increase in the circulation rate of the refrigerant did increase the performance of the unit in both the capillary and TXV mode. The TXV mode was the most economical one, because it had the highest compressor efficiency, highest cycle efficiency, and the lowest heat loss in comparison to the capillary mode. On the other hand, the capillary mode had higher refrigeration capacity and actual coefficient of performance. A final observation states that as the amount of refrigeration was increased in the system and the performance of the fluid decreased, which is proven by the values we got from the experiment.

Equipments & Procedure:

A Scott Air Conditioner was used as the foundation of this experiment with Freon-12 as a working fluid. The experiment will be run at two different modes: normal capillary and thermostatic expansion. The experimental setup consists of :

five pressure gauges, five temperature gauges, a set of fans, Compressor (A), condenser (B), evaporator (C), capillary tube (D), TXV (E), temperature sensor for automatic TXV control(F) , moisture and liquid indicator (G), calibrated rotameter (H), drier (I), liquid refrigerant receiver tank (J), oil and refrigerant accumulator tank (K), oil storage tank (L). The pressure & temperature gauges are all placed in strategic locations throughout the system and are labeled by (T and P) in (Figure 1). The numbers below refer to these strategic locations of the tempreture & pressure gauges in the refrigeration cycle:

1. Evaporator outlet.
2. Condenser inlet.
3. Condenser outlet.
4. Evaporator inlet.
5. Compressor inlet.

Before starting the experiment, the system was in thermal equilibrium with surrounding and temperatures along with pressures were taken at the five strategic points.

In the first mode “ normal capillary “ the expansion took place in a capillary feed tube, which is a simple metering device .the capillary tube acts as a fluid expansion vessel for the means of flow rate measurement for the Freon -12. The circulation rate of refrigerant in the system operating in normal capillary mode is controlled by valve 4. To switch to the TXV mode, valve V4 is closed and valve V5 should be opened. Finally, the main function of the normal capillary is to create a pressure drop due to the friction which will cause the fluid to evaporate; and five measurements of temperatures and pressures recorded at five different circulation rates of the refrigerant.

The second mode was the thermostatic expansion feed valve (TXV). Temperatures and pressures at the five strategic points were recorded at four different fans sits with allowing the system to reach steady state: High-high, medium-medium, medium-high, and low-high When steady state was reached, the temperature, pressure, fluid level, voltage and rotameter readings were measured. The difference between TXV and normal capillary is that in TXV is connected to a temperature sensing bubble. This bulb contained superheated Freon and caused for the pressure to change the temperature of the leaving Freon.

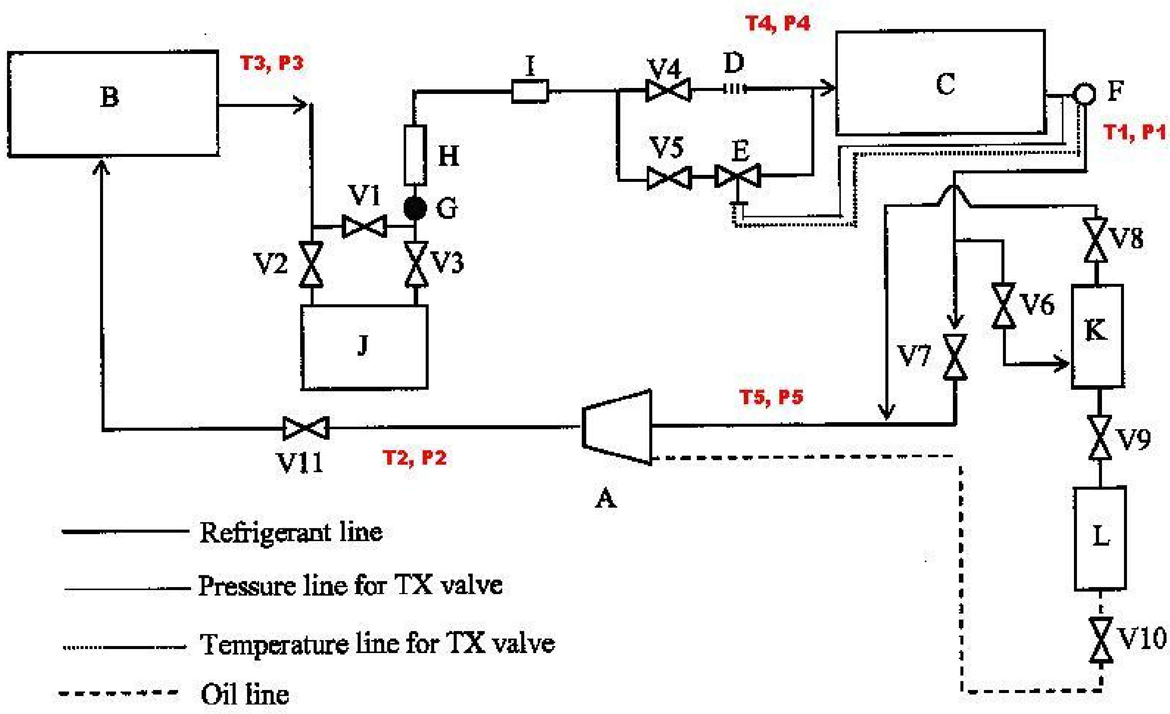


FIGURE 1: Schematic Diagram of the Refrigeration System in the Experiment.

Summery of results:

- COP_{max} increased with decreasing circulation rate of refrigerant.
- $COP_{actual,}$ increased and COP_{fluid} decreased with increasing circulation rate of refrigerant for the capillary mode.
- $COP_{actual,}$ decreased and COP_{fluid} increased with increasing circulation rate of refrigerant for the TXV mode.
- The Maximum COP_{max} was 8.20 and the maximum COP_{fluid} was 9.85 at refrigerant flow rate of 1.588 lbm/min in the capillary first run.
- The Maximum $COP_{actual,}$ was 1.76 at refrigerant flow rate of 1.588 lbm/min in the capillary first run.
- The Maximum COP_{max} was 7.53 at refrigerant flow rate of 1.588 lbm/min and the maximum COP_{fluid} was 11.69 at 1.296 lbm/min in the capillary 2nd run.
- The Maximum $COP_{actual,}$ was 1.66 at refrigerant flow rate of 1.588 Ibm/min in the capillary 2nd run.
- COP_{actual} was smaller than COP_{fluid} for any given refrigerant rate.
- CR increased with decreasing circulation rate of refrigerant for the capillary mode.
- CR increased with increasing circulation rate of refrigerant for the TXV mode.
- η_{cycle} increased as the refrigerant flow rate increases for the capillary and TXV modes.

- Overall efficiency of cycle η_{cycle}:

Capillary 1st Run: $\eta_{cycle} = 0.221 \Rightarrow 22.1\%$

Capillary 2nd Run: $\eta_{cycle} = 0.228 \Rightarrow 22.8\%$

TXV mode: $\eta_{cycle} = 0.210 \Rightarrow 22.9\%$

- The efficiency of the compression stage η_{comp}:

Capillary 1st Run: $\eta_{comp} = 0.118 \Rightarrow 11.8\%$

Capillary 2nd Run: $\eta_{comp} = 0.138 \Rightarrow 13.8\%$

TXV mode: $\eta_{comp} = 0.199 \Rightarrow 19.9\%$

- The compression ratio CR:

Capillary 1st Run: $CR = 3.21$

Capillary 2nd Run: $CR = 2.90$

TXV mode: $CR = 2.74$

- η_{comp} increases as the refrigerant flow rate increases for the capillary only.
- η_{comp} decreased as the refrigerant flow rate increased for the TXV mode only.
- RC increasd for both capillary mode and the TXV one as the refrigerant circulation rate increased.
- TXV mode is the most economical one because it gives the maximum cycle efficiency.

Discussion:

The different coefficients of performance calculated experienced an expected scenario, were the fluid COP was always higher than the actual one since it neglects heat losses in the system while the actual one doesn't.

Although it was difficult to observe the effect of the refrigerant circulation rate; it should have a major effect on the refrigeration process and its performance. As it is known from Bernoulli's equation, increasing the velocity of a fluid in a section leads to decreasing its pressure, if all other parameters been unchanged. That would lead to a higher heat absorbed from the cold reservoir and less heat rejected from the hot reservoir because the lower pressure will cause the refrigerant to evaporate quicker and the vapour will be superheated. Superheated vapour doesn't corresponds to pressure/temperature relationship; And since there will be no liquid remain to boil off to vapour, no more vapour pressure can be generated when heat is added. The vapour will take on sensible heat when heated and the temperature will rise. This eventually will increase the coefficient of performance of the refrigeration unit and all other performance parameters.

R-12 was the first halo-carbon refrigerant to go into general use. Its immiscibility with oil is one of its best characteristics. Nonetheless-having it in excess will tend to reduce the heat transfer rate and as a result will lead to a poor refrigeration unit performance. In the second run of Capillary mode when extra refrigerant was used, the maximum coefficient of performance was lowered than the first one which was expected.

Assuming the change in the ambient air temperature will have no effect on the compressor performance. This illustration will focus on the effects that change will have on the other two major components in the refrigeration system, the Evaporator and the Condenser.

The evaporator function is very simple. It absorbs heat into the system from the surroundings. This could lead to the simple conclusion that increasing the surrounding temperature will drive this process forward and will result a super saturated vapour with higher sensible energy “h1”. Higher enthalpy means higher Entropy as well.

The condenser rejects heat into the surroundings. So, as concluded in the previous paragraph. Increasing the surroundings temperature will decrease the rate of this rejection and eventually increase “h3” and S3.

The main source of error in this experiment results was the compressor overheating. It turned off a lot of times which forced the group to rush in taking the readings that may have been inaccurate which may have led to great deviations in the calculated thermodynamic properties. Another source of error is that the piping within the system wasn't well insulated, hence leading to heat loss throughout nearly all steps shown in Figure _ Appendix C, especially at steps when the process is suppose to be adiabatic (e.g. 3'-4').

Conclusion & Recommendations:

- The refrigeration capacity increased along with increasing refrigerant flow rate for the two different modes.
- To increases the refrigeration unit performance quality, we have to increase the refrigerant circulation.
- The system will perform better if the oil was removed completely from the refrigerant.
- TXV mode was more economical that the capillary one.
- The equipment used for the refrigeration system was old, and that affect the collection data as well as the results.
- We recommend using digital equipment to obtain better result of the data.
- We recommend using new temperatures and pressures gauges for this system, because some of these gauges give wrong readings.
- We recommend using a thermal insulator on the pipes, because by doing that we can eliminate the heat loss in the pipe. Therefore, obtaining good data for the T-S diagrams.
- Using insulated compressor will reduce the heat loss and increase the cycle efficient.

Refrences:

- Geankoplis,C,N. (2003),Transport Process and Separation process principles" Fourth ed., Prenice Hall, New Jesey,.
- Dr. Kruczek, Lab Handout

Appendix A

Tabulated Results

Table 1: Results obtained for first run in Capillary mode

	First run in Capillary mode				
Flow (lbm/min)	1.5884	1.502	1.394	1.286	1.178
Level of refrigerant	10	10	10	10.1	10.3
W (watts)	820	820	810	790	780
Tc (R)	507.67	506.67	503.67	500.67	499.67
Th (R)	569.56	571.28	571.55	571.64	572.65
Qc (Btu/min)	82.29	79.16	76.43	72.28	67.51
Qh(Btu/min)	90.65	88.42	84.55	80.09	75.29
COP_{max}	8.20	7.84	7.42	7.06	6.85
COP_{actual}	1.76	1.70	1.66	1.61	1.52
h_{1} (Btu/min)	82.83	82.97	84.35	84.99	85.60
h_{2} (Btu/min)	88.09	89.14	90.18	91.07	92.20
h_2^s (Btu/min)	87.83	87.97	87.99	88.00	88.07
h_{3} (Btu/min)	31.02	30.27	29.52	28.79	28.29
h_{4} (Btu/min)	31.02	30.27	29.52	28.79	28.29
P_{1} Psia	51.40	51.39	48.44	46.61	45.69
P_{2} Psia	152.40	156.39	156.44	156.61	158.69
COP_{fluid}	9.85	8.55	9.41	9.25	8.68
RC (Btu/min)	82.29	79.16	76.43	72.28	67.51
η_{comp}	0.17	0.16	0.11	0.09	0.07
η_{cycle}	0.22	0.22	0.22	0.23	0.22
CR	2.97	3.04	3.23	3.36	3.47

Table 2 : Results obtained for Second run in Capillary mode:

	Second run in Capillary mode				
Flow (lbm/min)	1.5884	1.502	1.394	1.286	1.178
Level of refrigerant	8.8	8.8	8.9	8.9	8.9
W (watts)	869	868	850	828	810
Tc (F)	509.67	507.67	499.67	499.67	499.67
Th (F)	577.28	576.55	574.81	576.45	577.39
Qc (Btu/min)	82.06	79.71	76.90	72.64	67.50
Qh(Btu/min)	92.53	89.10	84.47	79.18	73.28
COP_{max}	7.54	7.37	6.65	6.51	6.43
COP_{actual}	1.66	1.61	1.59	1.54	1.47
h_1 (Btu/min)	81.88	82.55	83.93	84.75	85.31
h_2 (Btu/min)	88.47	88.80	89.35	89.83	90.21
h_2^s (Btu/min)	88.43	88.37	88.24	88.37	88.44
h_3 (Btu/min)	30.22	29.48	28.76	28.26	28.01
h_4 (Btu/min)	30.22	29.48	28.76	28.26	28.01
P_1 Psia	54.45	51.40	45.19	45.69	45.69
P_2 Psia	167.45	166.40	163.19	166.69	168.69
COP_{fluid}	7.84	8.49	10.16	11.11	11.69
RC (Btu/min)	82.06	79.71	76.90	72.64	67.50
η_{comp}	0.21	0.18	0.12	0.10	0.08
η_{cycle}	0.22	0.22	0.24	0.24	0.23
CR	3.08	3.24	3.61	3.65	3.69

Table 3 : Results obtained for TXV mode:

	TXV mode			
Flow (lbm/min)	1.6748	1.7396	1.7504	1.7612
Level of refrigerant	8.5	8.5	8.7	8.8
W (watts)	870	880	880	893
Tc (F)	510.67	515.67	516.67	515.67
Th (F)	577.53	577.96	578.02	579.34
Qc (Btu/min)	83.55	87.97	87.02	85.64
Qh(Btu/min)	94.24	97.48	97.08	96.45
COP_{max}	7.64	8.28	8.42	8.10
COP_{actual}	1.69	1.76	1.74	1.69
h_1 (Btu/min)	82.35	83.02	82.93	82.59
h_2 (Btu/min)	88.73	88.49	88.68	88.73
h_2^s (Btu/min)	88.45	88.48	88.48	88.58
h_3 (Btu/min)	32.46	32.45	33.22	33.97
h_4 (Btu/min)	32.46	32.45	33.22	33.97
P_1 Psia	56.50	67.93	63.06	60.93
P_2 Psia	168.50	166.93	170.06	172.93
COP_{fluid}	7.82	9.25	8.65	7.92
RC (Btu/min)	83.55	87.97	87.02	85.64
η_{comp}	0.21	0.19	0.19	0.21
η_{cycle}	0.22	0.21	0.21	0.21
CR	2.98	2.46	2.70	2.84

Appendix B

Figures

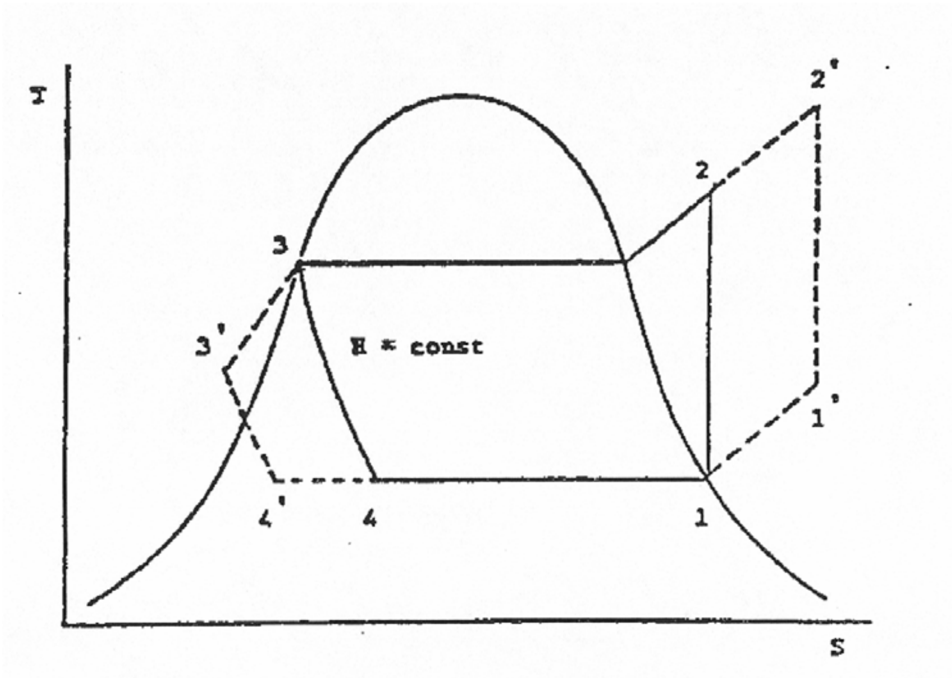


Figure 2 : Representation of an ideal vapour compression cycle on the temperature-entropy diagram.

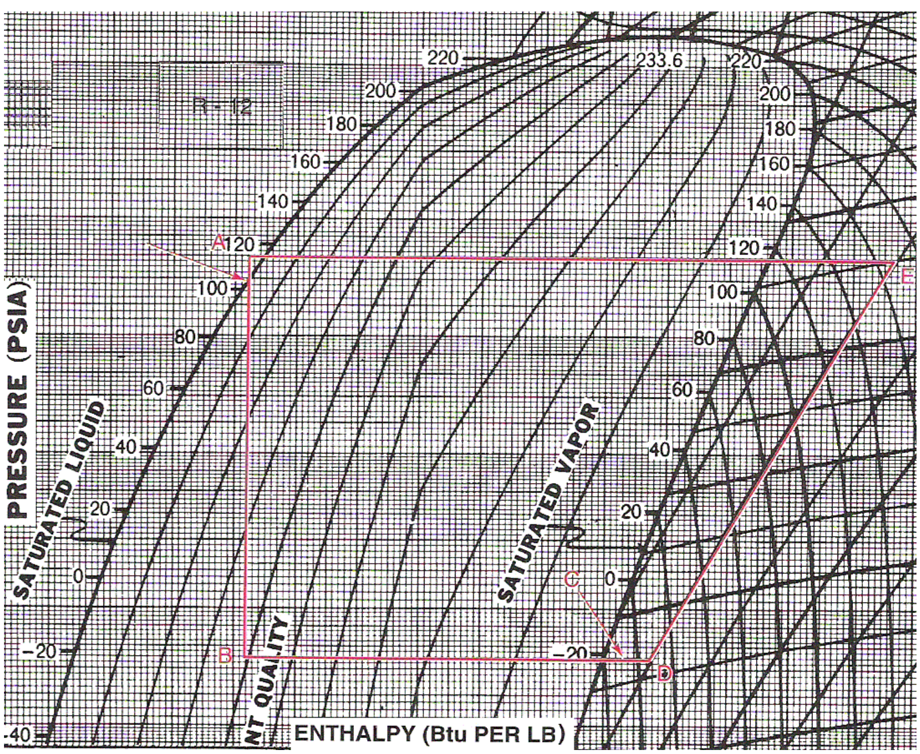


Figure 3: Representation of an ideal vapour compression cycle on the pressure – enthalpy diagram.

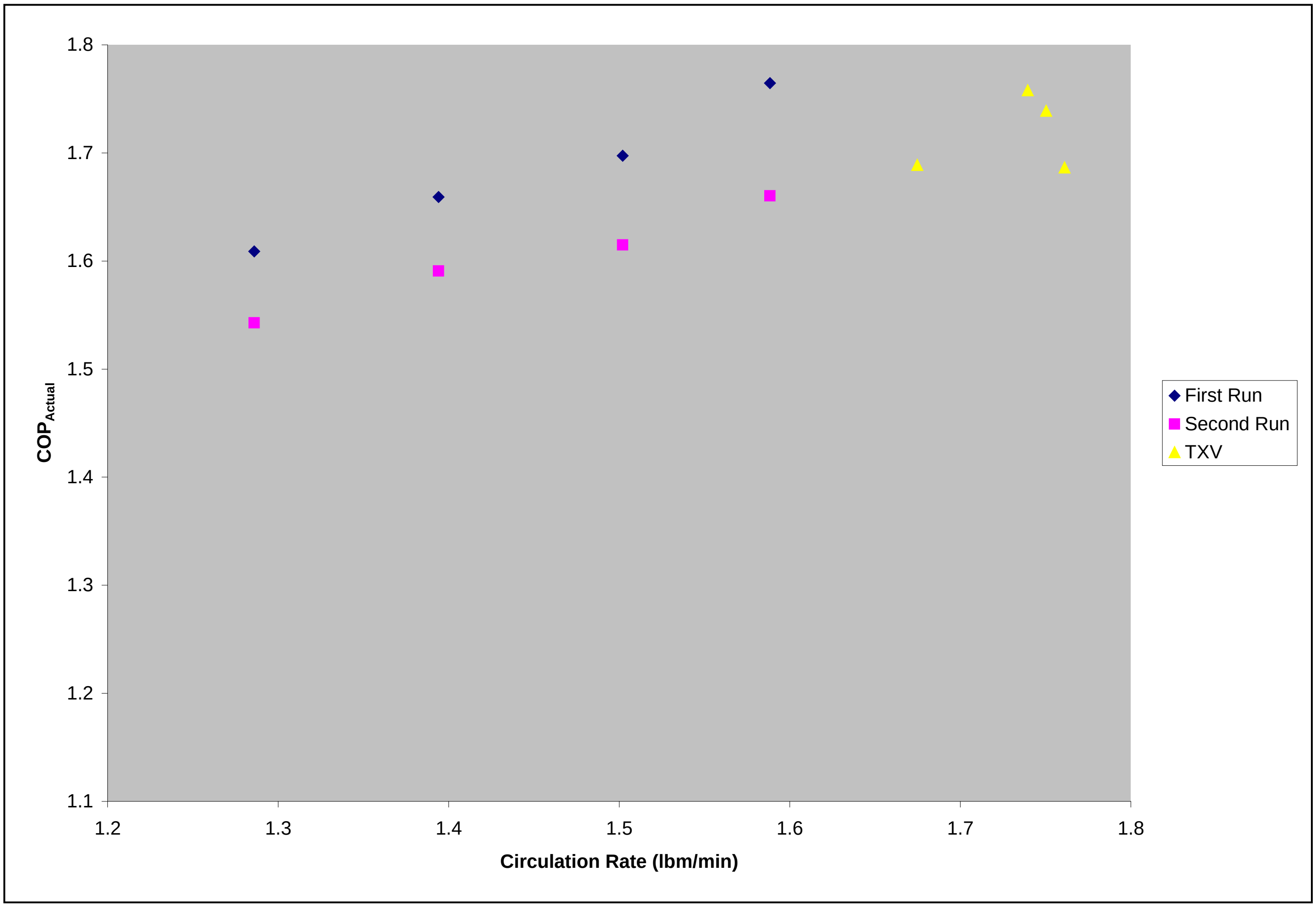


Figure 4: The coefficient of performance of the refrigeration system with respect to the circulation rate of the refrigerant, on all 3 selected runs.

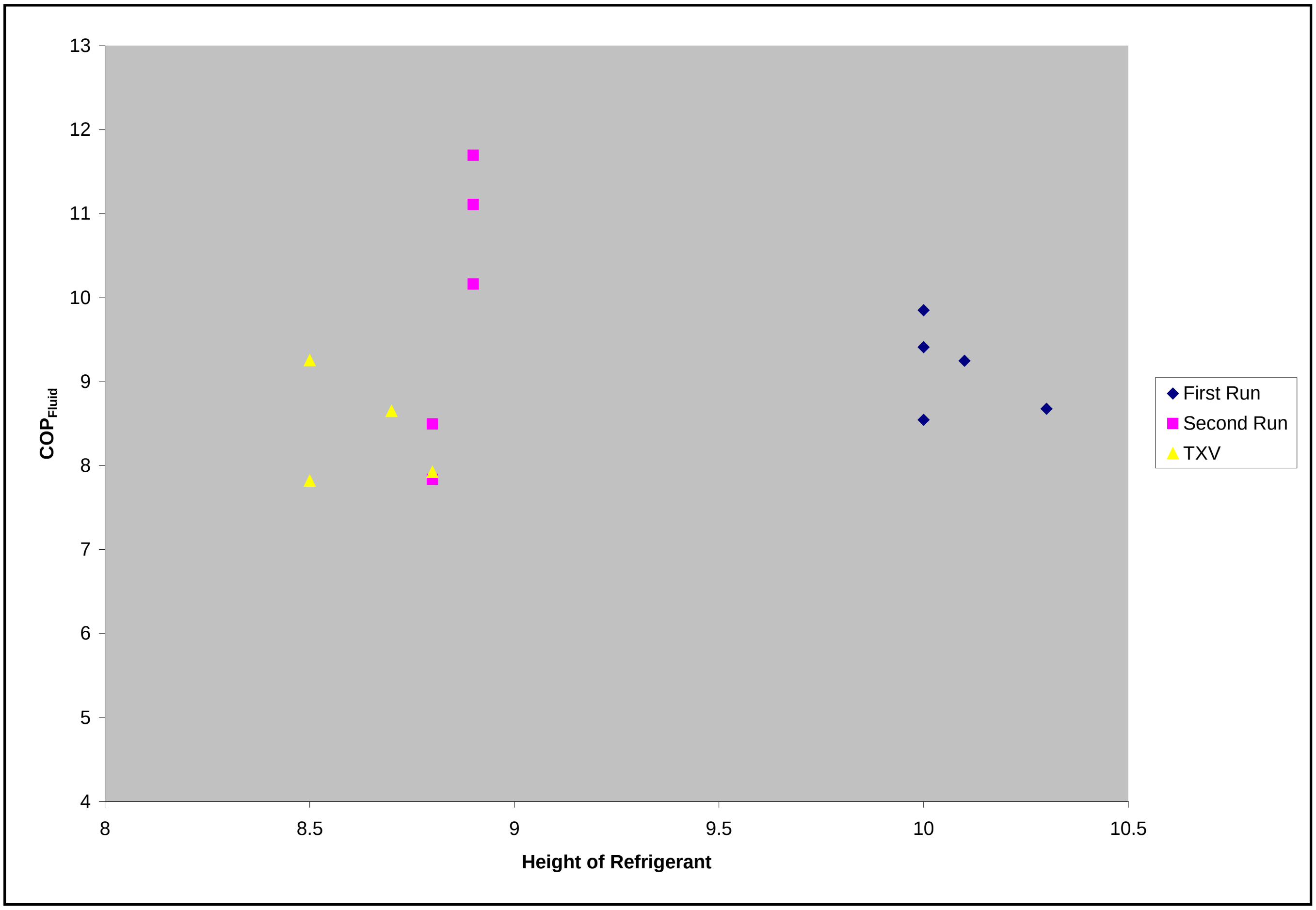
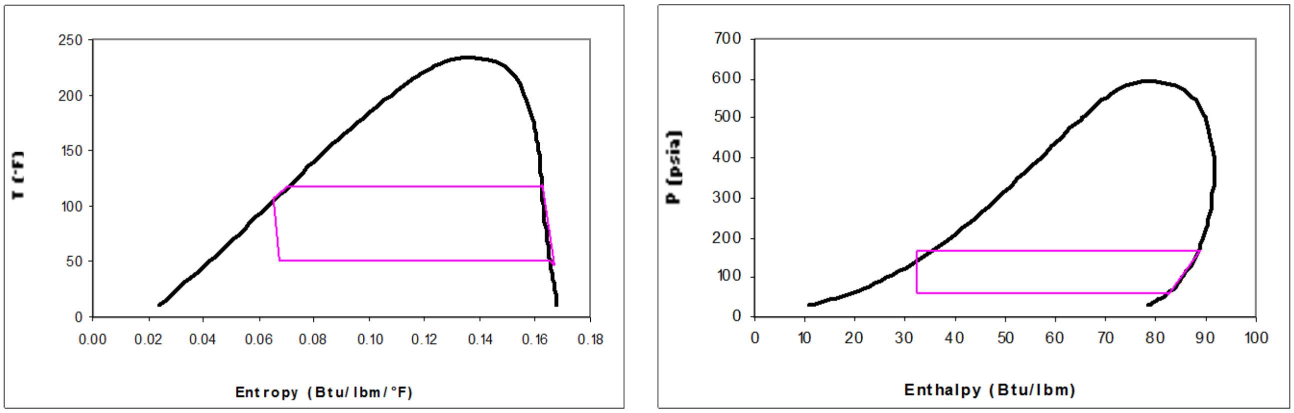
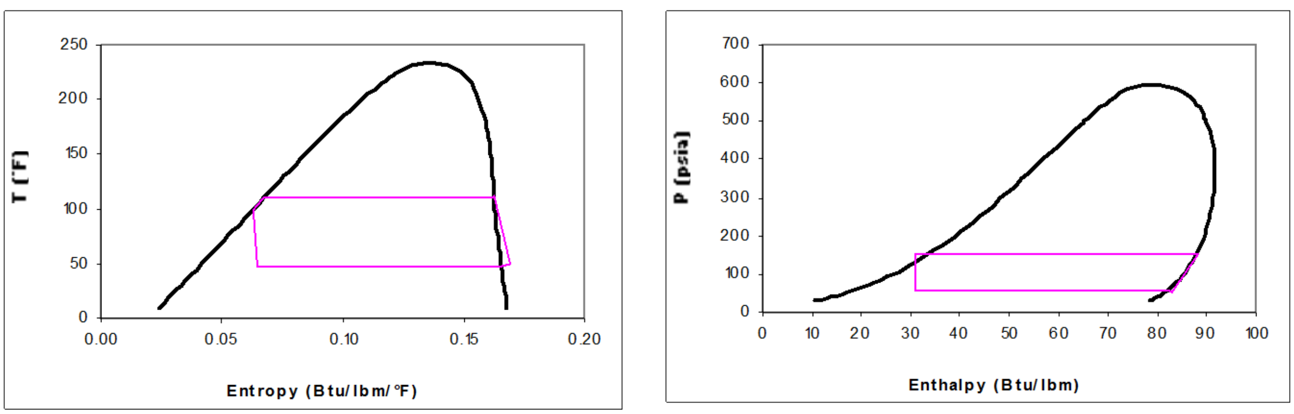


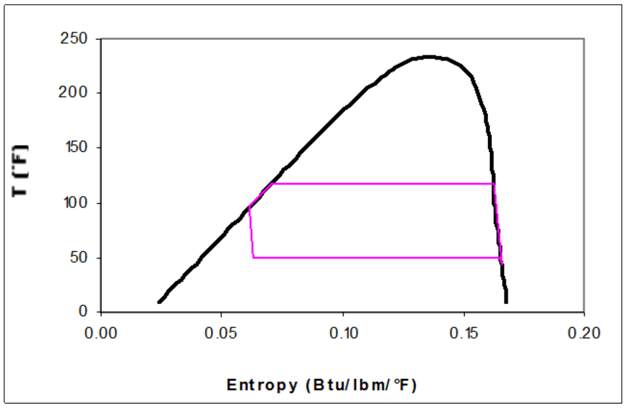
Figure 5: The Height of the refrigerant with respect to the circulation rate of the refrigerant, on all 3 selected runs.

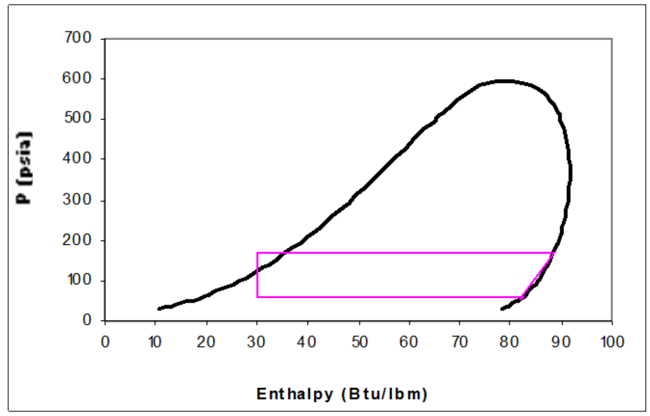


Figures 6 and 7: T-S and p-H plots of the refrigerant in the first run of the TXV mode. (Room Temp = 74 F^{o})



Figures 8 and 9: T-S and p-H plots of the refrigerant in the first run of the 1st Capillary mode. (Room Temp = 74 F^{o})





Figures 10 and 11: T-S and p-H plots of the refrigerant in the first run of the 2nd Capillary mode. (Room Temp = 74 F^{o})

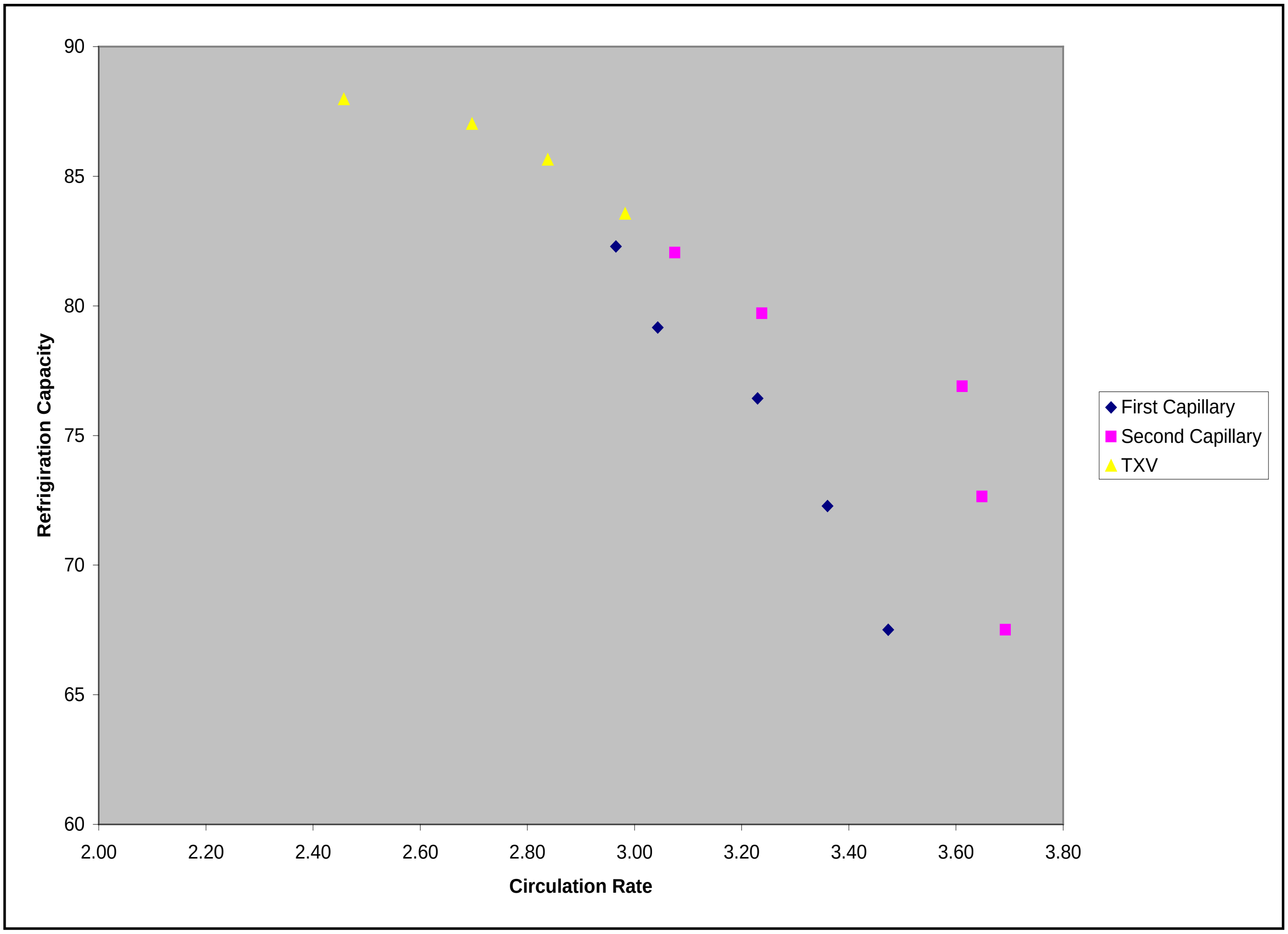


Figure 12: Changes in the refrigeration capacity of the system with respect to the circulation rate of the refrigerant, on all 3 selected runs.

Appendix C

Sample Calculations

Sample Calculation:

- This calculation at capillary first run at refrigerant circulation rate of 1.588 (Ibm/min)

1- Ideal (Carnot) efficiency:

$$T_h = T_3 = 100F^o = 569.56R^o$$

$$T_c = T_1 = 50F^o = 507.67R^o$$

$$COP_{\max} = \left(\frac{\dot{Q}_c}{W}\right)_{ideal} = \frac{T_c}{T_h - T_c} = \frac{507.67}{569.56 - 507.67}\frac{R^o}{R^o} = 8.20$$

Where,
T_h= Temperature of the hot reservoir
T_C= Temperature of the cold reservoir.
Q_C = Rate of energy absorbed in a cold reservoir per unit mass of refrigerant.
Q_H = Rate of energy rejected in a hot reservoir per unit mass of refrigerant.
W = Power input to the compressor per unit mass of refrigerant.

2-Actual coefficient of performance:

$$COP_{actual} = \frac{\dot{Q}_c}{W} = \frac{82.29}{46.63}\frac{Btu/Ibm}{Btu/Ibm} = 1.76$$

Where,
Q_C = Refrigeration capacity.
W = Power to the compressor.

3-Coefficient of performance of the fluid:

$$COP_{Fluid} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{82.83 - 31.02}{88.09 - 82.83}\frac{Btu/Ibm}{Btu/Ibm} = 9.85$$

Where,
h1= Specific enthalpy of the working fluid at point 1
h2= Specific enthalpy of the working fluid at point 2
h3= Specific enthalpy of the working fluid at point 3
h4= Specific enthalpy of the working fluid at point 4

4-Refrigeration capacity :

$$RC = \dot{Q}_c = \dot{m}_f (h_1 - h_4) = 1.588 \frac{Ibm}{\min} * (82.83 - 31.02) \frac{Btu}{\min} = 82.27 \frac{Btu}{\min}$$

Where,

m_f = The circulation rate of the refrigerant.
RC= The refrigeration capacity.

5-Compression efficiency:

$$\eta_{comp} = \frac{\dot{m}_f (h_2^s - h_1)}{W} = \frac{1.588 \, {Ibm}/{\min} \, (87.83 - 82.83) \, {Btu}/{Ibm}}{46.63 \, {Btu}/{\min}} * 100 = 17.02\%$$

h_2^s = is the enthalpy exiting the compressor at constant entropy and it equal to h_2,because the compressor work as isentropic.

6- Cycle efficiency:

$$\eta_{cycle} = \frac{COP_{actual}}{COP_{\max}} = \frac{1.76}{8.20} * 100 = 21.5\%$$

7- Compression ratio:

$$CR = \frac{P_2}{P_1} = \frac{152.39}{51.40} = 2.96$$

Where,

CR = the compression ratio
P_2 = pressure at point 2
P_1 = Pressure at point 1

Appendix D

Raw Data

Ambient Temperature (°F)=	74
Ambient Pressure (mmHg)=	755

Table 1 – Different fan sits Energy consumption in watts.

		S-6 Evaporator		
		High	Medium	Low
S-5 Condenser	High	90	78	70
	Medium	70	65	45
	Low	68	60	40

Table 2 – Initial conditions before running the first run in the Capillary mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Initial Temperature (°F)	74	74	72	74	74
Temperature corrections (°F)	0	0	-2	0	0
Initial Pressure (Psig)	73	70	54	72	80
Absoulute pressure (psia)	87.60	84.60	68.60	86.60	94.60

Table 3 – Initial conditions before running the Second run in the Capillary mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Initial Temperature (°F)	74	74	72	74	74
Temperature corrections (°F)	0	0	-2	0	0
Initial Pressure (Psig)	73	70	54	72	80
Absoulute pressure (psia)	87.60	84.60	68.60	86.60	94.60

Table 4 – Initial conditions before running the TXV mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Initial Temperature (°F)	74	74	72	74	74
Temperature corrections (°F)	0	0	-2	0	0
Initial Pressure (Psig)	73	70	54	72	80
Absoulute pressure (psia)	87.60	84.60	68.60	86.60	94.60

Table 5 – first cycle from first run with capillary mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	50	112	98	48	55
Pressure Reading (psig)	45	143	124	52	48
Absolute pressure (psia)	59.60	157.60	138.60	66.60	62.60
Psat (psia)	61.45	155.18	131.98	59.40	66.82
Tsat (°F)	39.68	110.63	109.14	48.00	35.15
Corrected Pressure (psia)	51.40	152.40	149.40	59.40	47.40
Corrected Temperature (°F)	50.0	112.0	100.0	48.0	55.0
Enthalpy (Btu/lbm)	82.8321	88.0917	31.0246	31.0246	83.6938
Entropy (Btu/lbm.R)	0.1691	0.1628	0.0630	0.0645	0.1720
Volume (ft^3/lbm)	0.8195	0.2679	0.0127	0.1377	0.9198

Table 6 – Second cycle from first run with capillary mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	51	120	95	47	50
Pressure Reading (psig)	44	146	126	50	46
Absolute pressure (psia)	58.60	160.60	140.60	64.60	60.60
Psat (psia)	62.50	172.16	126.60	58.39	61.45
Tsat (°F)	39.67	112.59	110.63	47.00	33.97
Corrected Pressure (psia)	51.39	156.39	152.39	58.39	46.39
Corrected Temperature (°F)	51.0	120.0	97.0	47.0	50.0
Enthalpy (Btu/lbm)	82.9717	89.1387	30.2682	30.2682	83.0379
Entropy (Btu/lbm.R)	0.1693	0.1643	0.0616	0.0631	0.1710
Volume (ft^3/lbm)	0.8233	0.2710	0.0126	0.1339	0.9250

Table 7 – Third cycle from first run with capillary mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	60	127	92	44	60
Pressure Reading (psig)	42	147	128	48	45
Absolute pressure (psia)	56.60	161.60	142.60	62.60	59.60
Psat (psia)	72.53	188.07	121.37	55.44	72.53
Tsat (°F)	36.36	112.62	111.15	44.00	31.62
Corrected Pressure (psia)	48.44	156.44	153.44	55.44	44.44
Corrected Temperature (°F)	60.0	127.0	94.0	44.0	60.0
Enthalpy (Btu/lbm)	84.3519	90.1775	29.5235	29.5235	84.5184
Entropy (Btu/lbm.R)	0.1729	0.1660	0.0603	0.0618	0.1746
Volume (ft^3/lbm)	0.9134	0.2816	0.0125	0.1386	1.0063

Table 8 – fourth cycle from first run with capillary mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	64	133	89	41	64
Pressure Reading (psig)	40	147	128	45	38
Absolute pressure (psia)	54.60	161.60	142.60	59.60	52.60
Psat (psia)	77.35	202.51	116.31	52.61	77.35
Tsat (°F)	34.22	112.70	111.23	41.00	22.70
Corrected Pressure (psia)	46.61	156.61	153.61	52.61	37.61
Corrected Temperature (°F)	64.0	133.0	91.0	41.0	64.0
Enthalpy (Btu/lbm)	84.9912	91.0684	28.7862	28.7862	85.3753
Entropy (Btu/lbm.R)	0.1748	0.1675	0.0589	0.0605	0.1787
Volume (ft^3/lbm)	0.9666	0.2895	0.0125	0.1437	1.2209

Table 9 – fifth cycle from first run with capillary mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	68	141	87	40	69
Pressure Reading (psig)	38	148	129	43	41
Absolute pressure (psia)	52.60	162.60	143.60	57.60	55.60
Psat (psia)	82.41	222.94	113.02	51.69	83.71
Tsat (°F)	33.13	113.70	112.25	40.00	28.16
Corrected Pressure (psia)	45.69	158.69	155.69	51.69	41.69
Corrected Temperature (°F)	68.0	141.0	89.0	40.0	69.0
Enthalpy (Btu/lbm)	85.5951	92.2001	28.2892	28.2892	85.9058
Entropy (Btu/lbm.R)	0.1762	0.1693	0.0580	0.0595	0.1782
Volume (ft^3/lbm)	1.0000	0.2946	0.0124	0.1424	1.1077

Table 10 – first cycle from second run with capillary mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	44	118	95	50	42
Pressure Reading (psig)	46	156	139	52	47
Absolute pressure (psia)	60.60	170.60	153.60	66.60	61.60
Psat (psia)	55.44	167.80	126.60	61.45	53.54
Tsat (°F)	42.97	117.84	117.37	50.00	36.37
Corrected Pressure (psia)	54.45	167.45	166.45	61.45	48.45
Corrected Temperature (°F)	44.0	118.0	97.0	50.0	42.0
Enthalpy (Btu/lbm)	81.8761	88.4696	30.2162	30.2162	81.8431
Entropy (Btu/lbm.R)	0.1663	0.1623	0.0615	0.0629	0.1680
Volume (ft^3/lbm)	0.7403	0.2406	0.0126	0.1213	0.8477

Table 11 – second cycle from second run with capillary mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	48	120	92	48	49
Pressure Reading (psig)	45	157	139	52	47
Absolute pressure (psia)	59.60	171.60	153.60	66.60	61.60
Psat (psia)	59.40	172.16	121.37	59.40	60.42
Tsat (°F)	39.68	117.35	116.41	48.00	33.98
Corrected Pressure (psia)	51.40	166.40	164.40	59.40	46.40
Corrected Temperature (°F)	48.0	120.0	94.0	48.0	49.0
Enthalpy (Btu/lbm)	82.5539	88.8013	29.4828	29.4828	82.8984
Entropy (Btu/lbm.R)	0.1685	0.1629	0.0601	0.0615	0.1708
Volume (ft^3/lbm)	0.8120	0.2465	0.0125	0.1215	0.9211

Table12 – third cycle from second run with capillary mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	56	123	89	40	56
Pressure Reading (psig)	42	157	138	47.5	45
Absolute pressure (psia)	56.60	171.60	152.60	62.10	59.60
Psat (psia)	67.93	178.86	116.31	51.69	67.93
Tsat (°F)	32.53	115.85	114.42	40.00	27.51
Corrected Pressure (psia)	45.19	163.19	160.19	51.69	41.19
Corrected Temperature (°F)	56.0	123.0	91.0	40.0	56.0
Enthalpy (Btu/lbm)	83.9259	89.3546	28.7613	28.7613	84.0954
Entropy (Btu/lbm.R)	0.1732	0.1641	0.0589	0.0605	0.1749
Volume (ft^3/lbm)	0.9747	0.2589	0.0125	0.1480	1.0810

Table 13 – fourth cycle from second run with capillary mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	62	127	87	40	61
Pressure Reading (psig)	40	158	139	45	42
Absolute pressure (psia)	54.60	172.60	153.60	59.60	56.60
Psat (psia)	74.91	188.07	113.02	51.69	73.71
Tsat (°F)	33.13	117.48	116.08	40.00	26.86
Corrected Pressure (psia)	45.69	166.69	163.69	51.69	40.69
Corrected Temperature (°F)	62.0	127.0	89.0	40.0	61.0
Enthalpy (Btu/lbm)	84.7476	89.8320	28.2596	28.2596	84.8187
Entropy (Btu/lbm.R)	0.1746	0.1647	0.0579	0.0595	0.1765
Volume (ft^3/lbm)	0.9820	0.2568	0.0124	0.1421	1.1122

Table 14 – fifth cycle from second run with capillary mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	66	130	86	40	65
Pressure Reading (psig)	38	158	139	43	41
Absolute pressure (psia)	52.60	172.60	153.60	57.60	55.60
Psat (psia)	79.85	195.20	111.39	51.69	78.59
Tsat (°F)	33.13	118.41	117.02	40.00	28.16
Corrected Pressure (psia)	45.69	168.69	165.69	51.69	41.69
Corrected Temperature (°F)	66.0	130.0	88.0	40.0	65.0
Enthalpy (Btu/lbm)	85.3120	90.2125	28.0087	28.0087	85.3398
Entropy (Btu/lbm.R)	0.1757	0.1652	0.0575	0.0590	0.1771
Volume (ft^3/lbm)	0.9942	0.2567	0.0124	0.1391	1.0957

Table 15 – first cycle from TXV mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	48	120	104	51	46
Pressure Reading (psig)	50	159	141	55	54
Absolute pressure (psia)	64.60	173.60	155.60	69.60	68.60
Psat (psia)	59.40	172.16	143.25	62.50	57.39
Tsat (°F)	45.09	118.32	117.40	51.00	41.96
Corrected Pressure (psia)	56.50	168.50	166.50	62.50	53.50
Corrected Temperature (°F)	48.0	120.0	106.0	51.0	46.0
Enthalpy (Btu/lbm)	82.3492	88.7305	32.4618	32.4618	82.1914
Entropy (Btu/lbm.R)	0.1666	0.1627	0.0655	0.0672	0.1672
Volume (ft^3/lbm)	0.7221	0.2416	0.0128	0.1401	0.7652

Table 16 – second cycle from TXV mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	56	118	104	56	48
Pressure Reading (psig)	55	151	138	54	56
Absolute pressure (psia)	69.60	165.60	152.60	68.60	70.60
Psat (psia)	67.93	167.80	143.25	67.93	59.40
Tsat (°F)	56.00	117.60	118.98	56.00	50.46
Corrected Pressure (psia)	67.93	166.93	169.93	67.93	61.93
Corrected Temperature (°F)	56.0	118.0	106.0	56.0	48.0
Enthalpy (Btu/lbm)	83.0211	88.4870	32.4500	32.4500	19.0485
Entropy (Btu/lbm.R)	0.1651	0.1623	0.0654	0.0669	0.0409
Volume (ft^3/lbm)	0.5950	0.2418	0.0128	0.1206	0.0117

Table 17 – third cycle from TXV mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	54	120	107	57	48
Pressure Reading (psig)	56	160	141	61	58
Absolute pressure (psia)	70.60	174.60	155.60	75.60	72.60
Psat (psia)	65.72	172.16	149.13	69.06	59.40
Tsat (°F)	51.53	119.04	117.66	57.00	46.67
Corrected Pressure (psia)	63.06	170.06	167.06	69.06	58.06
Corrected Temperature (°F)	54.0	120.0	109.0	57.0	48.0
Enthalpy (Btu/lbm)	82.9282	88.6779	33.2167	33.2167	82.2873
Entropy (Btu/lbm.R)	0.1660	0.1625	0.0668	0.0684	0.1661
Volume (ft^3/lbm)	0.6483	0.2380	0.0129	0.1241	0.6973

Table 18 – fourth cycle from TXV mode.

	Point 1	Point 2	Point 3	Point 4	Point 5
Temperature Reading (°F)	51	121	110	56	47
Pressure Reading (psig)	53	162	143	59	56
Absolute pressure (psia)	67.60	176.60	157.60	73.60	70.60
Psat (psia)	62.50	174.38	155.18	67.93	58.39
Tsat (°F)	49.50	120.35	118.98	56.00	45.54
Corrected Pressure (psia)	60.93	172.93	169.93	67.93	56.93
Corrected Temperature (°F)	51.0	121.0	112.0	56.0	47.0
Enthalpy (Btu/lbm)	82.5922	88.7296	33.9678	33.9678	82.1930
Entropy (Btu/lbm.R)	0.1659	0.1624	0.0681	0.0699	0.1662
Volume (ft^3/lbm)	0.6664	0.2333	0.0130	0.1348	0.7111

To: Dr. Macchi.

From: Pankhuri Anand, Suhaib Alamash, Group 6.

Date: July 21, 2011.

Subject: CHG 3122, Refrigeration.

The experiment was performed by using the Scott Air Conditioning and Refrigeration system with the refrigerant working fluid Freon 12 also referred to as R-12. There were two main objectives of this lab. Firstly, to obtain the refrigeration capacity and performance coefficients for a refrigeration unit as a function of circulation rate of refrigerant and heat load. Secondly, to compare the performance of the refrigeration unit in normal capillary mode and thermostatic expansion valve (TXV) mode. This report will focus primarily on Problem Set- 1 provided in the lab handout.

The procedure was performed by running the experiment under two conditions: the capillary mode and thermal expansion valve mode (TXV). For capillary run #1, five strategic points were placed along the system to record temperature and pressure for five different circulation rates. The data was collected at the inlet and outlet of both the evaporator and the condenser as well as only the inlet of the compressor. The procedure was repeated for capillary run# 2 with higher amount of refrigerant. In the case of TXV mode, the flow rate was constant with varying fan speeds across the condenser and evaporator. There are several factors that affected the results in this experiment. The major challenge was to measure temperature and pressure reading at five different points along with the level of refrigerant and flow rate. Also, the compressor was not able to handle very high temperatures since the circulation rate was changing constantly. Therefore, a fan was used to cool down the compressor.

As the amount of refrigerant in the system was reduced, there was an increase in the overall efficiency and various performance coefficients in both the capillary and TXV mode. The average actual coefficient of performance values was found to be 1.700 for capillary run#1, 1.675 for capillary run#2 and 1.505 for the TXV mode. The calculated average refrigeration capacities were 82.086 Btu/min for capillary run#1, 73.080 Btu/min for capillary run#2 and 69.459 Btu/min for the TXV mode. There was a linear relationship established between the circulation rate and different performance efficient. The average COP fluid values were 6.937 for capillary run#1, 5.621 for capillary run#2 and 4.576 for TXV mode.

Equipments and Procedure:

This experiment was done using Scott Ail Conditioning Education system (Figure 1, Appendix A) with Freon-12 as refrigerant fluid. Two operating modes were performed: Capillary and thermostatic expansion (TXV). The system's major refrigeration apparatuses are as the following A compressor (A), a condenser (B), an evaporator (C), a capillary (D), a thermostatic expansion valve (E), a temperature sensor (F), valves (V) a moisture and liquid indicator (G), a calibrated rotameter (H), a drier (I), a liquid refrigerant receiver tank (J), an oil and refrigerant accumulator tank (K) and an oil storage tank (L). The pressure/temperature gauges are all placed in strategic locations throughout the system as labelled (T and P) in the figure: 1 – Evaporator outlet, 2 – Condenser inlet, 3 – Condenser outlet, 4 – Evaporator inlet, and 5 – Compressor inlet.

Before beginning the experiment , the system was ensured to being at thermal equilibrium where temperature at all gauges are the same. Then, it was important to record the work done by the compressor by reading the wattage of all fan setting combination. Then , both of fans were set to high and so the experiment is started.

A lower amount of refrigerant was used for the first set of runs. For this set of runs, valve 4 is used to control the circulation rate while valve 5 is closed. Then , 5 temperatures and pressures were recorded once the system has reached equilibrium. The same procedure was then repeated for higher amount of refrigerant.

When performing under TXV mode , valve 4 was closed and valve 5 was used to control the circulation rate. The same recordings were taken except at 4 different fan setting which were: high-high, high-medium, medium-high and medium-medium.

(Please refer to Figure 1, Appendix A for set up of the apparatus)

Summary of Results (Problem Set# 1 Questions):

1. The corrected temperature is acquired by taking the difference between the initial and the ambient temperature. Therefore, the corrected pressure, P_1', is calculated using the following equation:

$$P_1' = P_1 - (P_4 - P_4^{sat}) - (P_1^{initial} - P_4^{initial})$$

   where, P_1 is the measured pressure, $(P_4 - P_4^{sat})$ term is the correction factor for any compressor oil being in the system and the $(P_1^{initial} - P_4^{initial})$ term accounts for the beginning deviation in the pressure measurement at point 1. The temperature at point 4 is used as the basis for correction because it lies within the two-phase region i.e. liquid-vapour. Since the temperature does not significantly change inside this region, it is fare to assume the T_4 readings recorded are quite accurate. These temperature values are then used to find the pressure at point 4 which are used for the correcting pressure at all other points.

2. The effect of the circulation rate of refrigerant (m_f) on the performance of the system is graphically summarized in Figure 2 and 4, Appendix A. It is observed that COP max is increasing linearly with increasing circulation. Whereas, the COP actual is barely increasing linearly as the circulation rate increases. Also, the COP fluid values obtained closely match with COP max but fluctuating. The compression ratio (CR) is slightly decreasing also following the linear trend.

3. Comparing the data from Capillary run #1 and Capillary run #2, it can be seen that having a higher amount of refrigerant in the system increases refrigeration capacity and decreases the COP maximum, actual and fluid. The higher amount of the refrigerant in the system had no observed effect on the compression efficiency. Moreover, Cycle

efficiency and compression ratio is increased with the amount of the refrigerant in the system.

4. It is assumed that the system is at thermodynamic equilibrium. An increase in ambient temperature would cause the temperature and pressure values to increase relatively. Since enthalpy and entropy are directly proportional to temperature, both properties will increase with temperature (Figure 6, Appendix A). Also, the refrigeration capacity decreases with an increase in the ambient temperature. In general the condenser operates at a higher temperature than the ambient temperature. Therefore, the increase in ambient temperature causes a decrease in heat ejected by the condenser. This is because a lower temperature difference between the fluid and the ambient air is present at the condenser. The decrease in heat given off by the condenser generates a higher temperature refrigerant thereby leaving the condenser and evaporator at a higher temperature. This would decrease the driving force for the heat transfer at the evaporator and in turn, the lower heat absorbed (refrigeration capacity) is expected and observed. The power required by the compressor would not be affected. Therefore, the COP actual and COP fluid decreases while COP max increases. As a result, cycle efficiency i.e. the ratio of COP actual and COP max decreases.

Discussion:

This experiment was designed to obtain performance coefficients at different circulation rate. It was done in two different modes, capillary mode and TXV mode. Starting by the capillary run #1, it can be said that the experiment followed an expected trend with respect to the coefficient of performance max, actual and fluid. An increase in the circulation rate increased both the maximum and actual coefficients of performance; COP max and COP actual. A linear relationship was observed for COP max as per the ideal Carnot Cycle while the COP actual was slightly increasing linearly. This is because, the amount of energy absorbed in the cold reservoir (T_C) does increase with increasing flow rate while the temperature of the hot reservoir (T_H) is decreased and the amount of power fed to the compressor is increased. Hence, a slight upward trend in the actual performance coefficient can be noticed. Since COP of the fluid is dependant only on the enthalpies at different points that increase with increasing temperatures, the COP fluid has similar values to the COP max. The COP actual is the observed performance and thus expected to fall below the others since there is heat loss along the system whereas the COP fluid is the ideal assuming no heat loss throughout and the most pronounced at the compressor. Also, the COP max is the absolute upper bound of any refrigeration system (Figure 2, Appendix A).

When plotting the compression and cycle efficiency (Figure 3 and 5, Appendix A), both of them have nearly constant value. This is attributed to very small range of the flow rates used in the experiment. By definition, Compression efficiency (CR) is the ratio of the minimum shaft work required for compression (isentropic work) and the actual work of the compressor. As the amount of refrigerant increased, the power required for compressor also increased thereby increased the actual and isentropic work. The change in the amount of refrigerant does not have any impact on the efficiency. The cycle efficiency (η cycle) is the ratio of COP actual to COP max, two variables that increased with increasing circulation rate. The graph shows that η cycle decreases with increasing circulation rate. The values range from 0.209 to 0.261 with an average of 0.238. The main reason behind the decreased efficiency would be due to the the increased fluid friction in the system as circulation rate increased. On the other hand, values recorded for refrigeration capacity (RC) display high correlation to the flow rate since more fluid travelling through the system increases, the amount of cooling the system can supply to the cold sink. It can be seen that the efficiency of the system seems to increase with increasing flow rate. Yet this is counteracted by the additional power needed to run the compressor and by the amount of heat that can be transferred to and from the system at the condenser and evaporator respectively.

The refrigeration cycle can be described as: First, the refrigerant in the vapour phase flowed from the evaporator and entered the compressor. The high pressure vapour was formed in the compressor and passed to the condenser where saturated high pressure liquid was created. At some stage in this step, heat was lost to the hot reservoir. The high pressure and saturated liquid was then passed through the expansion valve where a pressure drop was noticed which lead to drop in temperature as well. A two- phase system i.e. liquid- vapour region was observed at this point. The new pressure and temperature values entered the evaporator in which heat was captivated from surroundings. Here, a pure vapour solution was observed and thus the Freon returned back to its original condition.

There were inevitably some considerable sources of error in the experiment. The major challenge was to maintain the flow rate by controlling the valve. The system exceeded the prescribed maximum pressure of 200 psig for the compressor. Ideally, heat is lost in the hot reservoir (condenser) and gained in the cold reservoir (evaporator). The system had to be restarted for each run since the system was too hot which delayed the experiment. It was important to maintain the flow rate below the threshold. Therefore, an external fan was used to cool down the compressor and avoid overheating. The equations used to determine the performance of the system do not account for the heat lost to the surroundings from the tubing. Thus, there was heat loss from the compressor and then entire piping system was not insulated which further lead to heat losses. It was difficult to record quick and accurate reading of five different temperature and pressure values along with the level of refrigerant and power. Overall, the system did not allow significant range for flow rates which greatly limited the resulting analysis.

Conclusion and Recommendations:

1. The refrigerant R-12, was used as the working fluid to set up vapor-compression cycle in a refrigeration system. The increase in circulation rate increased the actual and maximum coefficient of performance. Whereas, an increase in amounts of refrigerant decreased COP actual and max.
2. It was observed that the performance coefficients for capillary mode (run#1 and 2) showed a linear characteristic behavior.
3. The average refrigeration capacities (RC) of capillary mode (run#1 and 2) are higher than the refrigeration capacity obtained for the TXV mode. The capillary mode (run#1) had a higher cycle and compression efficiency compared to the capillary (run#2) and TXV mode.
4. The compression ratio calculated for the TXV mode was higher than capillary mode (run#1 and 2).

The sources of error could be minimized by applying the following recommendations so that we can get better results. This will allow the vapor-compression cycle to be more efficient:

1. It is important to ensure that the piping system and compressor used for the experiment are perfectly insulated to prevent heat is loss to the surrounding environment.
2. Using a multi-stage compressor with intercooler, or cooling the refrigerant during the compression process, will result in lower entropy, point 2.

3. A good suggestion would be to use the digital pressure and temperature transducers in order to accurately record the values quickly in a LABVIEW program.
4. The current refrigerant can be replaced with ozone friendly working fluids such as R22, R134a, R744 or R290 will be better for the environment.
5. Also, the rotameter readings were constantly fluctuating during the experiment. It will be better to change the rotameter or use a computer generated software/program to stabilize and maintain the refrigeration system ahead of time without causing any delays during the experiment.

References:

1. Perry, J. H. Perry's Chemical Engineers' Handbook. New York: McGraw-Hill. 1984
2. Smith, J.M. “Introduction to Chemical Engineering Thermodynamics” – 7th edition, 2005 New York, McGraw Hill
3. Macchi, A. CHG3122 Chemical Engineering Practice Summer 2011, Lab E. Chemical Engineering Department, University of Ottawa.

Appendix A

Tabulated and Graphical Results.

Table 1: The amount of heat lost and gain in the refrigeration system for different circulation rates at Capillary mode (run#1)

Circulation rate	Height (inch)	T_H	T_C	W	Q_c	Q_H
1.718	7.6	572.687	514.170	840	87.591	442.703
1.610	7.7	579.380	511.170	855	83.271	456.849
1.502	7.8	584.019	509.170	850	78.729	469.455
1.394	7.9	587.860	506.170	850	75.420	491.183
1.556	8.1	586.650	511.170	850	85.420	504.199
1.556	**7.8**	**582.119**	**510.370**	**849**	**82.086**	**472.878**

Table 2: The changes in the amount of enthalpies at different strategic stages of the refrigeration system for Capillary (run#1):

Circulation Rate	COPmax	COPactual	COPfluid	RC	ηCompression	ηCycle	CR
1.718	8.787	1.834	7.017	87.591	0.203	0.209	2.656
1.610	7.494	1.713	6.797	83.271	0.198	0.229	3.034
1.502	6.803	1.629	6.745	78.729	0.198	0.239	3.333
1.394	6.196	1.560	6.703	75.420	0.182	0.252	3.604
1.556	6.772	1.767	7.422	85.420	0.190	0.261	3.306
1.556	**7.210**	**1.700**	**6.937**	**82.086**	**0.194**	**0.238**	**3.187**

Table 3: The performance of the refrigeration system using various parameters for different circulation rates at capillary mode (run#1).

Circulation Rate	h1	h2	h2s	h3	h4	p1	p2
1.718	82.369	89.632	88.012	31.382	31.382	59.768	158.768
1.610	82.100	89.710	88.072	30.379	30.379	56.527	171.527
1.502	81.906	89.677	88.289	29.490	29.490	54.432	181.432
1.394	82.332	90.404	88.647	28.229	28.229	53.389	192.389
1.556	82.980	90.330	88.900	28.080	28.080	57.030	188.530

Table 4: The amount of heat lost and gain in the refrigeration system for different circulation rates at Capillary mode (run#2).

Circulation rate	Height (inch)	T_H	T_C	W	Q_c	Q_H
1.610	11.0	568.384	511.67	787	80.048	94.630
1.502	11.4	569.075	509.17	782	75.657	89.478
1.394	11.2	570.792	506.67	770	71.058	84.779
1.286	11.4	569.849	502.67	755	69.446	81.148
1.232	11.4	570.620	501.67	740	69.193	80.670
1.405	**11.3**	569.744	506.37	**767**	**73.080**	**86.141**

Table 5: The changes in the amount of enthalpies at different strategic stages of the refrigeration system for Capillary (run#2):

Circulation Rate	COPmax	COPactual	COPfluid	RC	ηCompression	ηCycle	CR
1.610	9.022	1.789	5.489	80.048	0.209	0.198	2.653
1.502	8.500	1.701	5.474	75.657	0.202	0.200	2.782
1.394	7.902	1.623	5.179	71.058	0.198	0.205	2.804
1.286	7.483	1.617	5.934	69.446	0.132	0.216	3.103
1.232	7.276	1.644	6.029	69.193	0.104	0.226	3.207
1.405	**8.036**	**1.675**	**5.621**	**73.080**	**0.169**	**0.209**	**2.910**

Table 6: The performance of the refrigeration system using various parameters for different circulation rates at capillary mode (run#2).

Circulation Rate	h1	h2	h2s	h3	h4	p1	p2
1.610	82.223	91.280	88.029	32.504	32.504	56.558	150.058
1.502	82.002	91.203	87.973	31.631	31.631	54.432	54.432
1.394	81.848	91.691	88.073	30.874	30.874	54.888	153.888
1.286	83.748	92.848	88.165	29.747	29.747	48.983	92.848
1.232	84.678	93.994	88.220	28.515	28.515	48.039	154.039
1.405	82.900	92.203	88.092	30.654	30.654	52.580	121.053

Table 7: The amount of heat lost and gain in the refrigeration system for different circulation rates at TXV mode.

Circulation rate	Height (inch)	T_H	T_C	W	Q_c	Q_H
1.340	10.5	575.682	501.670	800	70.884	87.209
1.297	10.6	577.049	503.170	800	67.832	83.738
1.394	10.7	588.709	507.670	815	72.777	85.889
1.329	10.5	585.308	505.170	833	66.342	82.347
1.340	**10.6**	**581.687**	**504.420**	**812**	**69.459**	**84.796**

Table 8: The changes in the amount of enthalpies at different strategic stages of the refrigeration system for TXV mode.

Circulation Rate	COPmax	COPactual	COPfluid	RC	ηCompression	ηCycle	CR
1.340	6.778	1.558	4.342	70.884	0.188	0.230	3.386
1.297	6.811	1.491	4.265	67.832	0.189	0.219	3.254
1.394	6.265	1.570	5.550	72.777	0.153	0.251	3.491
1.329	6.304	1.400	4.145	66.342	0.198	0.222	3.491
1.340	**6.539**	**1.505**	**4.576**	**69.459**	**0.182**	**0.230**	**3.405**

Table 9: The performance of the refrigeration system using various parameters for different circulation rates at TXV mode..

Circulation Rate	h1	h2	h2s	h3	h4	p1	p2
1.340	82.121	94.303	88.505	29.222	29.222	49.039	166.039
1.297	81.675	93.941	88.291	29.368	29.368	51.460	51.460
1.394	83.865	93.272	88.946	31.658	31.658	55.396	193.396
1.329	81.828	93.868	88.898	31.916	31.916	53.400	93.868
1.340	**82.372**	**93.846**	**88.660**	**30.541**	**30.541**	**52.324**	**126.191**

Figures and Graphs:

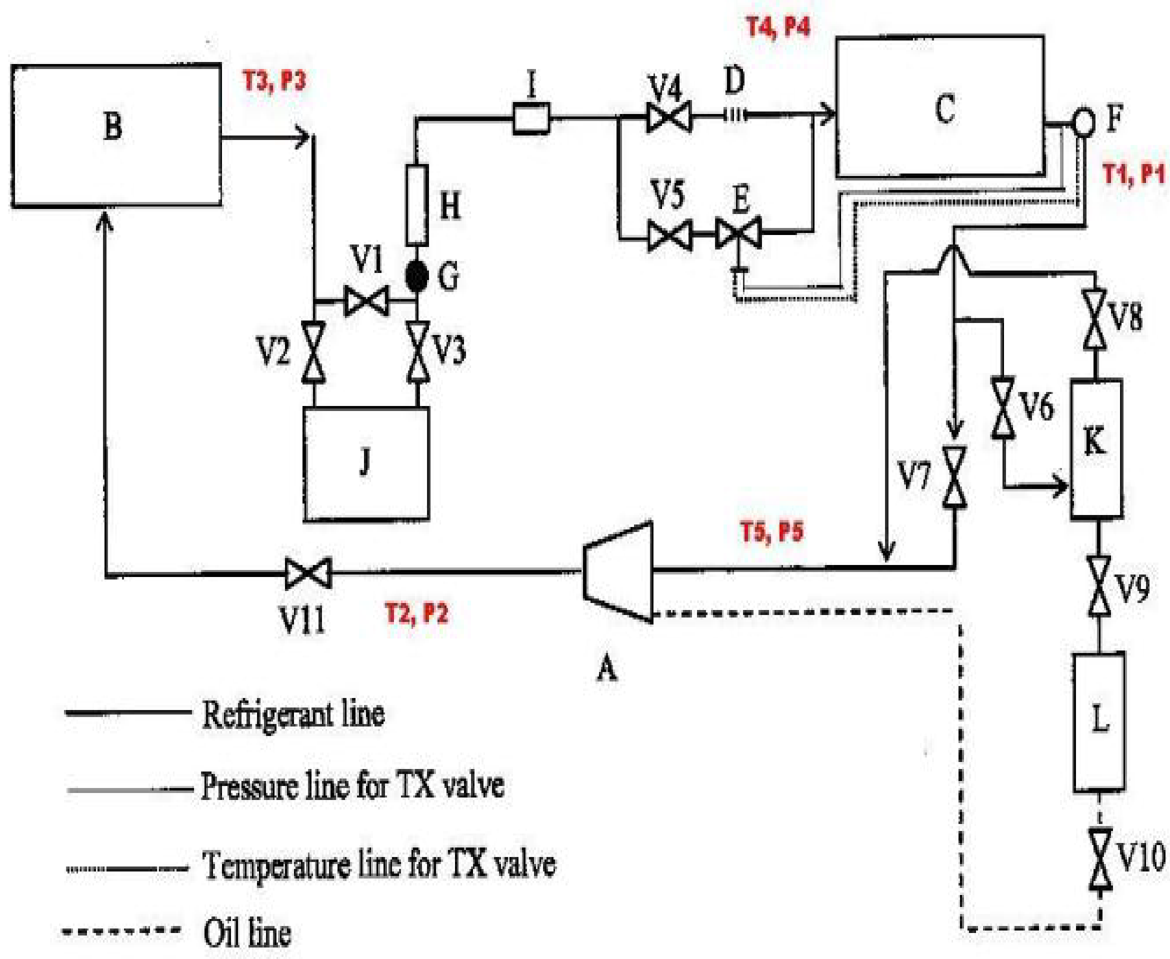


Figure1: Schematic diagram for the Scott Air Conditioning and Refrigeration System, where 1 – Evaporator outlet, 2 – Condenser inlet, 3 – Condenser outlet, 4 – Evaporator inlet, and 5 – Compressor inlet.

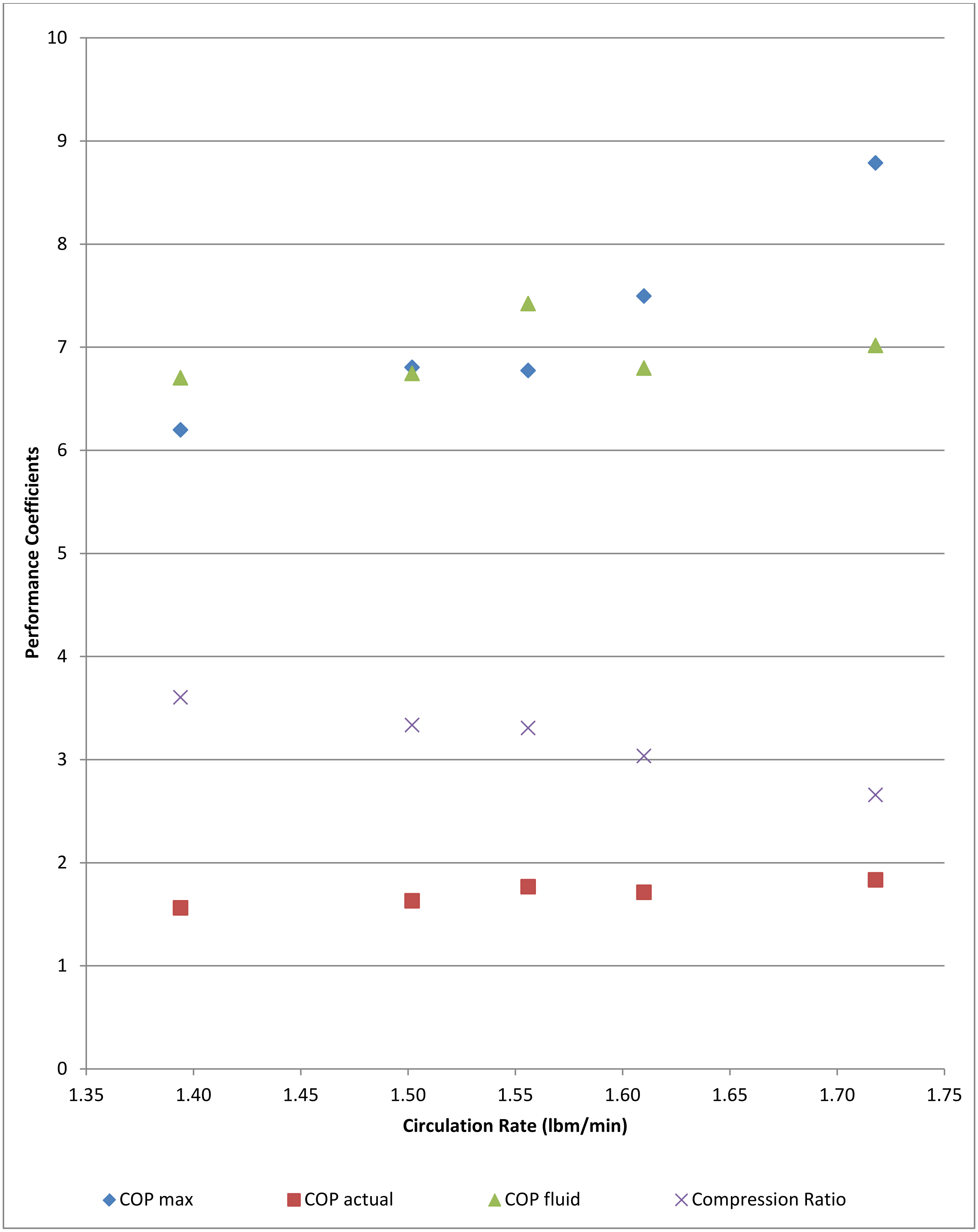


Figure 2: Performance Coefficients as a function of circulation rate (lbm/min) for Capillary run#1 mode.

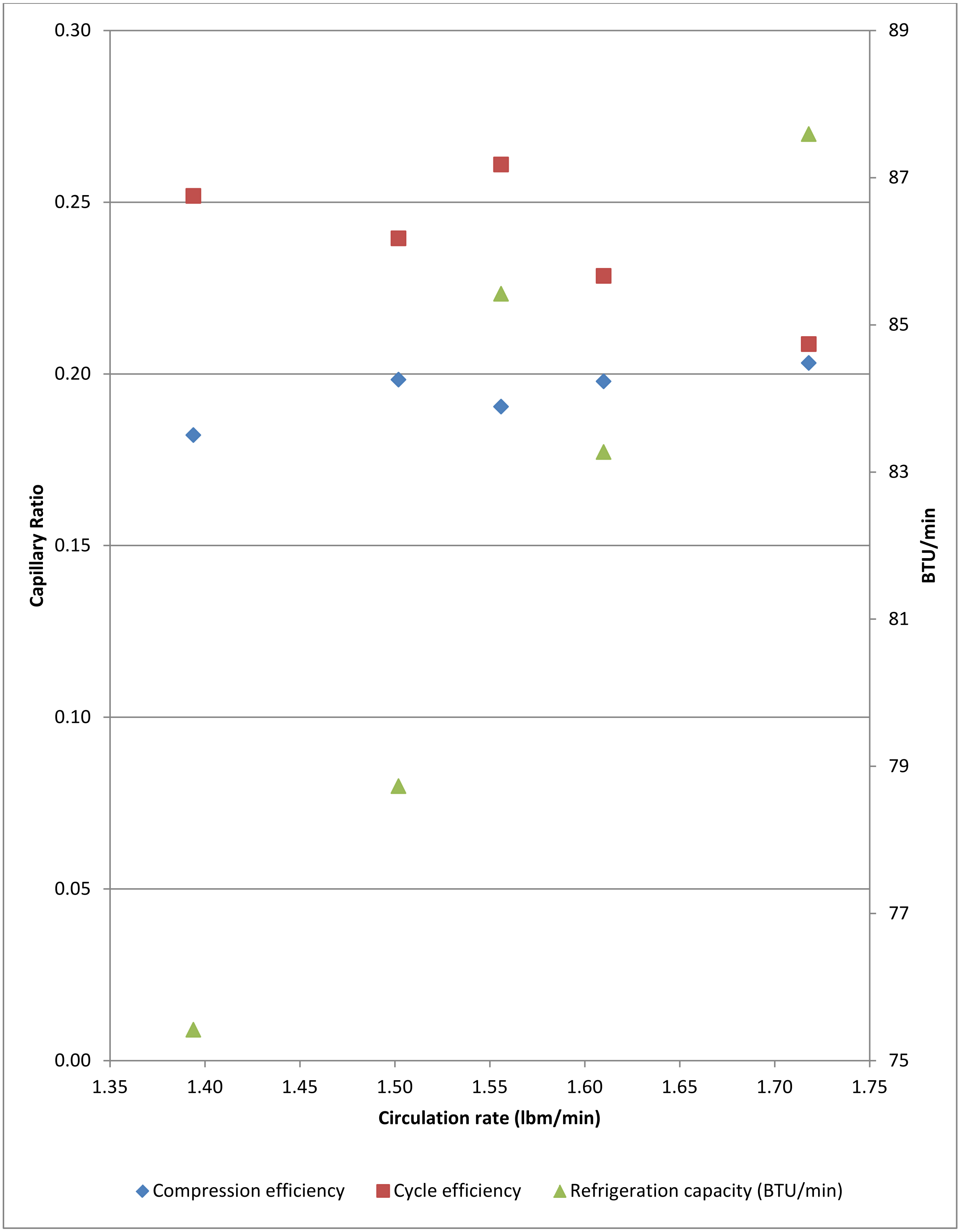


Figure 3: Compression and cycle efficiency along with refrigeration capacity (BTU/min) as a function of circulation rate (lbm/min) for Capillary run#1 mode.

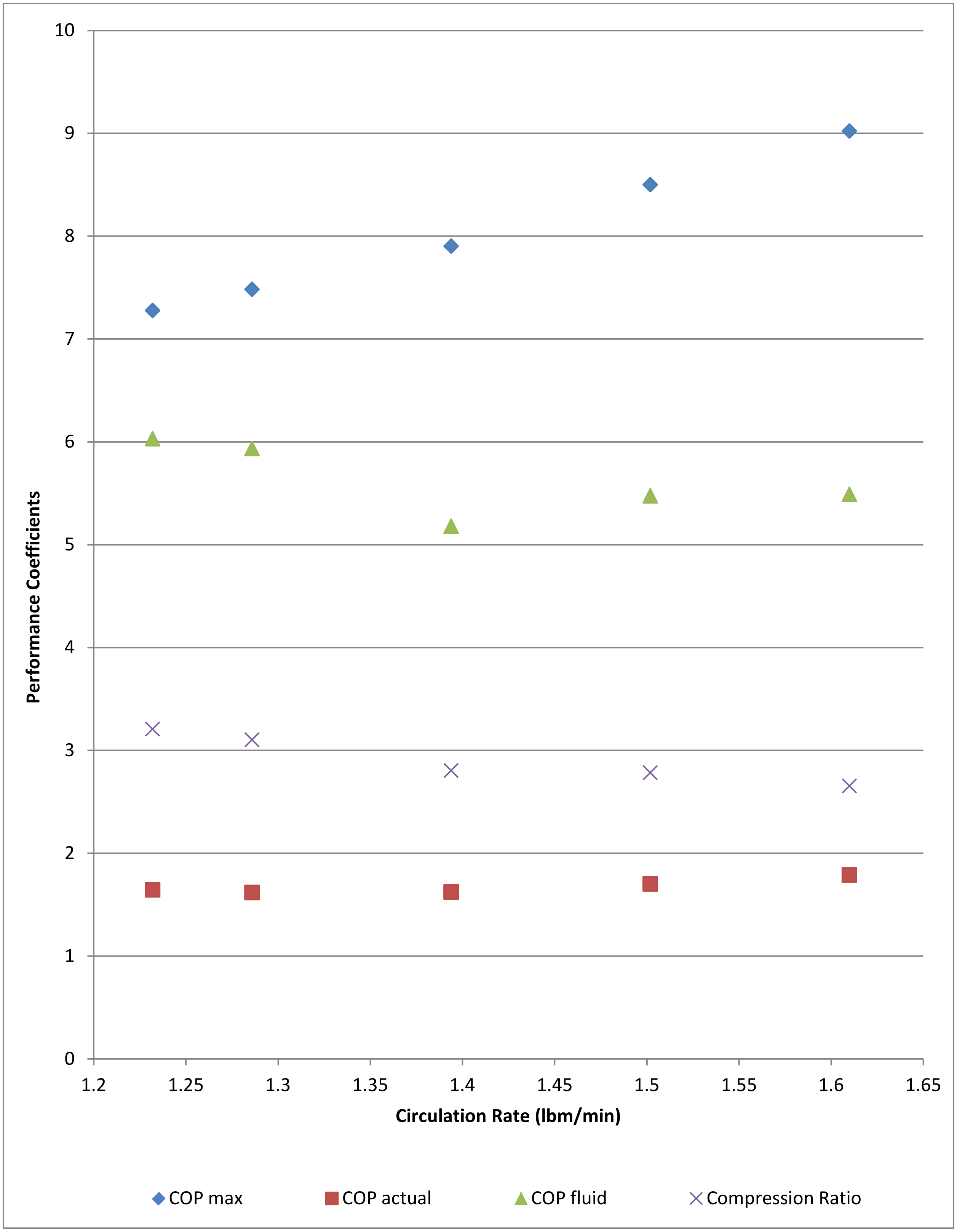


Figure 4: Performance Coefficients as a function of circulation rate (lbm/min) for Capillary run#2 mode.

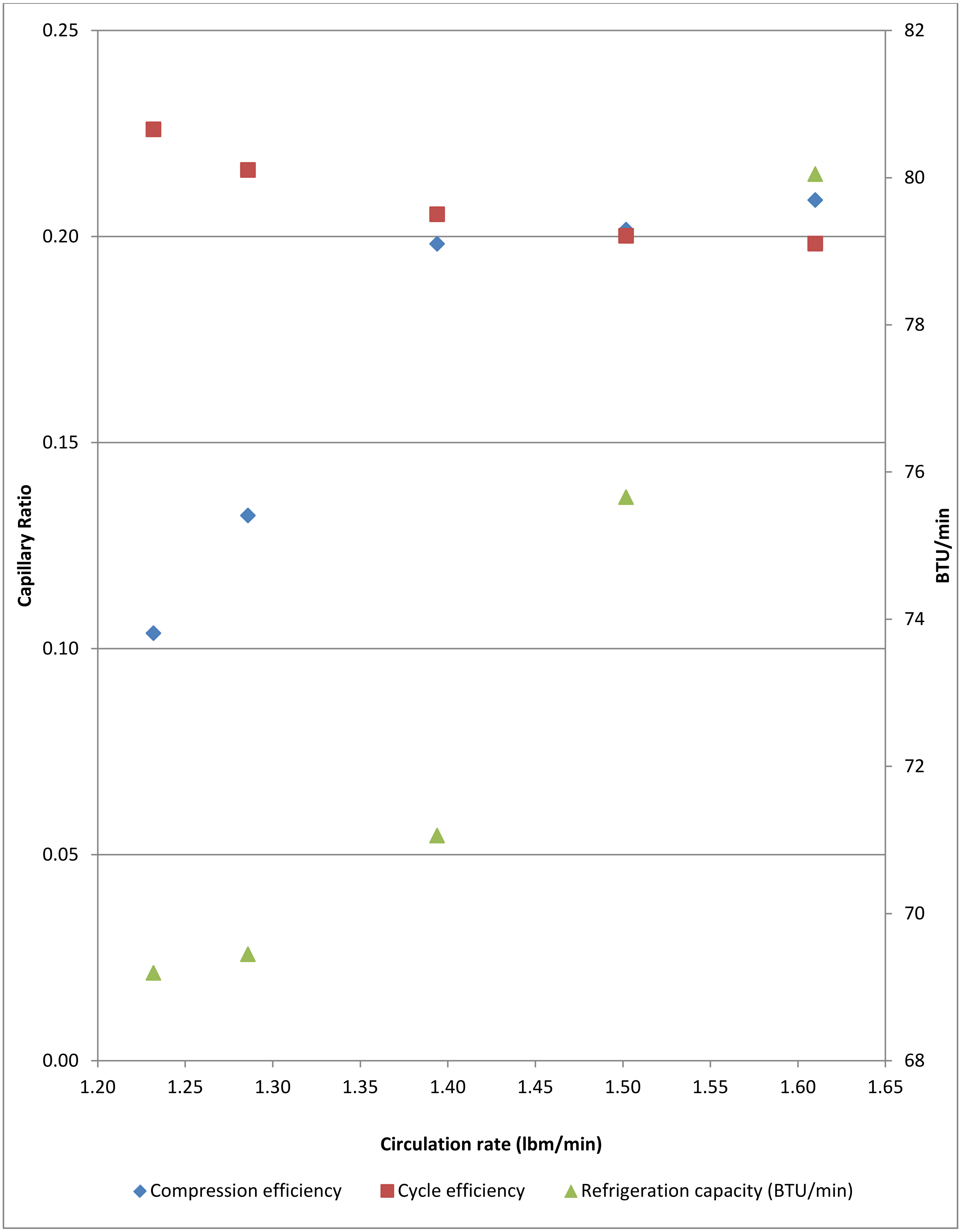
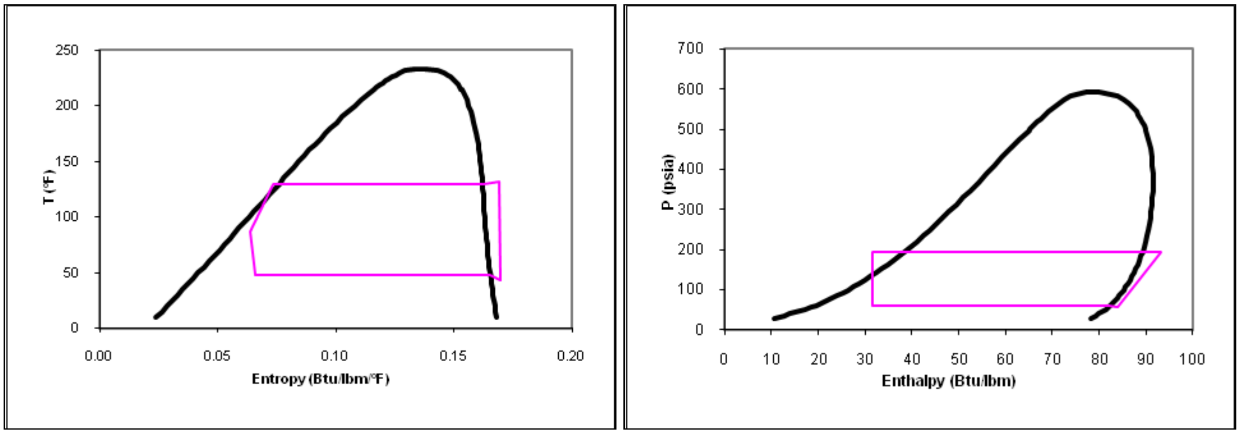


Figure 5: Compression and cycle efficiency along with refrigeration capacity (BTU/min) as a function of circulation rate (lbm/min) for Capillary run#2 mode.

Low Ambient Temperature:



High Ambient Temperature:

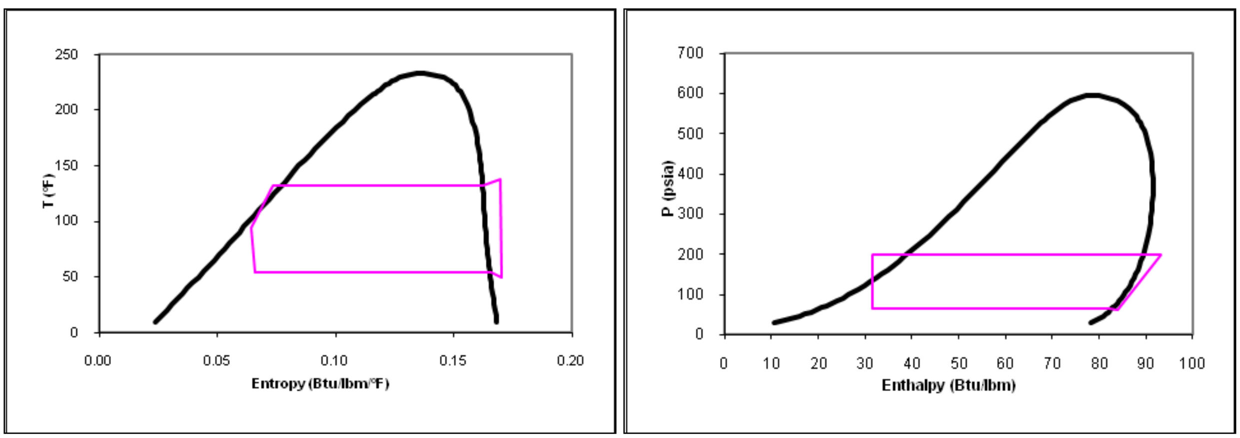


Figure 7: The effect of ambient temperature on T-S and P-H graphs.

Appendix B

Sample Calculations.

The sample calculation in all of the following cases are done for capillary run #1 , first flow rate.

Conversions Factors and Constants

Conversion of Temperature from Fahrenheit (F) to Rankin (R):

$$T(R) = T(F) + 459.69$$
$$T(R) = 113.02 + 459.69 = 572.69\ R$$

Conversion of Watts (W) to Btu/min:

$$W(Btu/\text{min}) = W(W)\frac{60\ \text{sec}}{\text{min}} \cdot \frac{0.00094783\ \text{Btu}}{J}$$
$$W(Btu/\text{min}) = 840\ W\ \frac{60\ \text{sec}}{\text{min}} \cdot \frac{0.00094783\ \text{Btu}}{J}$$
$$W(Btu/\text{min}) = 47.771\ Btu/\text{min}$$

Conversion of Pressure from mmHg to psi:

$$P(psi) = \frac{P(Torr)}{760\ Torr} \cdot 14.696\ psi$$
$$P(pis) = \frac{757.3\ Torr}{760\ Torr} \cdot 14.696\ psi = 14.637\ psi$$

Constants:

$$R = 1.98588\ \ Btu/lb\,mol\ R$$
$$P_{ambient} = 757.3\ mmHg \qquad P_{\text{standard}} = 760\ mmHg$$
$$T_{ambient} = 73.5\ {}^oF \qquad T_{\text{standard}} = 32\ {}^oF$$
$$1\ Btu = 1\ 055.055\ 9\ \text{J}$$

Maximum Coefficient of Performance

The maximum coefficient of performance, COP_{max}, is found by the following:

$$COP_{\max} = \frac{T_c}{T_h - T_c} = \frac{T_1}{T_3 - T_1}$$

where,

$T_c = T_1$ = the cold reservoir (evaporator) temperature in Rankin

$T_h = T_3$ = the hot reservoir (condenser) temperature in Rankin

$$COP_{\max} = \frac{T_c}{T_h - T_c} = \frac{T_1}{T_3 - T_1} = \frac{514.17}{572.69 - 514.17} = 8.787$$

Refrigeration Capacity

The maximum refrigeration capacity, RC, is determined as follows:

$$RC = Q_c = m_f (h_1 - h_4)$$

where,

Q_c = the rate of energy absorbed by evaporator per unit mass of Freon in Btu/min

m_f = the mass flow rate of Freon in lbm/min

h_i = the specific enthalpy at the point i in Btu/lbm

$$RC = Q_c = m_f (h_1 - h_4) = (1.718)(82.37 - 31.38) = 87.59\ Btu/\min$$

Actual Coefficient of Performance

The actual coefficient of performance, COP_{actual}, is determined as follows:

$$COP_{actual} = \frac{Q_c}{W}$$

where,

Q_c = the rate of energy absorbed by evaporator per unit mass of Freon in Btu/min

W = the energy supplied to the compressor in Btu/min

$$COP_{actual} = \frac{Q_c}{W} = \frac{87.59}{47.771} = 1.834$$

Coefficient of Performance of Freon

The coefficient of performance of the fluid (Freon), COP_{Freon}, can be determined by the following equation:

$$COP_{Freon} = \frac{h_1 - h_4}{h_2 - h_1}$$

where, h_i = the specific enthalpy at the point i in Btu/lbm

$$COP_{Freon} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{82.37 - 31.38}{89.63 - 82.73} = 7.016$$

Compression Efficiency

The compression efficiency, η_{comp}, of the Freon fluid is determined as follows:

$$\eta_{comp} = \frac{m_f (h_2^s - h_1)}{W}$$

where,

m_f = the mass flow rate of Freon in lbm/min

h_2^s = the isentropic specific enthalpy calculated at the point 2 in Btu/lbm

h_1 = the specific enthalpy at the point 1 in Btu/lbm

W = the energy supplied to the compressor in Btu/min

$$\eta_{comp} = \frac{m_f (h_2^s - h_1)}{W} = \frac{1.718(88.01 - 82.37)}{47.771} = 0.203$$

Cycle Efficiency

The cycle efficiency, η_{cycle}, is determined as follows:

$$\eta_{cycle} = \frac{COP_{actual}}{COP_{\max}}$$

where,

COP_{actual} = actual coefficient of performance

COP_{max} = maximum coefficient of performance

$$\eta_{cycle} = \frac{COP_{actual}}{COP_{\max}} = \frac{1.834}{8.787} = 0.209$$

Compression Ratio

The compression ratio, CR, is determined by the following:

$$CR = \frac{P_2}{P_1} = \frac{158.77}{59.77} = 2.656$$

where, P_1 = pressure at point 1 in psia

P_2 = pressure at point 2 in psia

Appendix C

Raw and Non-Essential Data.

Table 10: Raw Data for Capillary Run#1 Mode at Five Different Flow Rates

	Point 1	Point 2	Point 3	Point 4	Point 5
Initial Temperature (°F)	72	72	70	71	72
Temperature corrections (°F)	-1.5	-1.5	-3.5	-2.5	-1.5
Initial Pressure (Psig)	73	71	74	72.5	80
Absolute pressure (psia)	87.64	85.64	88.64	87.14	94.64
Temperature Reading (°F)	48	116	83	52	55
Pressure Reading (psig)	50	147	147	56	57
Absolute pressure (psia)	64.64	161.64	161.64	70.64	71.64
Psat (psia)	60.93	166.72	109.00	66.27	68.50
Tsat (°F)	48.37	113.74	112.29	54.50	48.37
Corrected Pressure (psia)	59.77	158.77	155.77	66.27	59.77
Corrected Temperature (°F)	49.5	117.5	86.5	54.5	56.5
Enthalpy (Btu/lbm)	82.3659	89.6322	31.3819	31.3819	18.3205
Entropy (Btu/lbm.R)	0.1658	0.1650	0.0636	0.0649	0.0395
Volume (ft^3/lbm)	0.6781	0.2736	0.0127	0.1116	0.0117
Quality (Vapour = 1; Liquid = 0)	1	1	0	0.170	0
Temperature Reading (°F)	45	120	79	49	40
Pressure Reading (psig)	47	160	163	53	52
Absolute pressure (psia)	61.64	174.64	177.64	67.64	66.64
Psat (psia)	57.89	175.49	102.78	63.03	53.07
Tsat (°F)	45.12	119.71	119.71	51.50	43.05
Corrected Pressure (psia)	56.53	171.53	171.53	63.03	54.53
Corrected Temperature (°F)	46.5	121.5	82.5	51.5	41.5
Enthalpy (Btu/lbm)	82.0997	89.7096	30.3785	30.3785	17.8781
Entropy (Btu/lbm.R)	0.1658	0.1651	0.0618	0.0630	0.0386
Volume (ft^3/lbm)	0.6945	0.2727	0.0126	0.1091	0.0116
Quality (Vapour = 1; Liquid = 0)	1	1	0	0.161	0
Temperature Reading (°F)	42	124	76	47	40
Pressure Reading (psig)	45	170	174	51	47
Absolute pressure (psia)	59.64	184.64	188.64	65.64	61.64
Psat (psia)	54.96	184.58	98.29	60.93	53.07
Tsat (°F)	42.95	124.13	124.57	49.50	37.49
Corrected Pressure (psia)	54.43	181.43	182.43	60.93	49.43
Corrected Temperature (°F)	43.5	125.5	79.5	49.5	41.5
Enthalpy (Btu/lbm)	81.9064	89.6770	29.4904	29.4904	17.4752
Entropy (Btu/lbm.R)	0.1660	0.1646	0.0602	0.0613	0.0378
Volume (ft^3/lbm)	0.7204	0.2611	0.0125	0.1058	0.0116
Quality (Vapour = 1; Liquid = 0)	1	1	0	0.151	0
Temperature Reading (°F)	47	130	76	44	52

Pressure Reading (psig)	44	181	181	48	45
Absolute pressure (psia)	58.64	195.64	195.64	62.64	59.64
Psat (psia)	59.90	198.83	98.29	57.89	65.17
Tsat (°F)	41.84	128.83	127.56	46.50	35.14
Corrected Pressure (psia)	53.39	192.39	189.39	57.89	47.39
Corrected Temperature (°F)	48.5	131.5	79.5	46.5	53.5
Enthalpy (Btu/lbm)	82.3319	90.4040	28.2288	28.2288	82.6237
Entropy (Btu/lbm.R)	0.1676	0.1651	0.0578	0.0590	0.1681
Volume (ft^3/lbm)	0.7758	0.2485	0.0124	0.1057	0.7838
Quality (Vapour = 1; Liquid = 0)	1	1	0	0.143	1
Temperature Reading (°F)	45	128	77	49	40
Pressure Reading (psig)	47.5	177	179	53	50
Absolute pressure (psia)	62.14	191.64	193.64	67.64	64.64
Psat (psia)	57.89	194.00	99.77	63.03	53.07
Tsat (°F)	45.63	127.19	126.77	51.50	40.91
Corrected Pressure (psia)	57.03	188.53	187.53	63.03	52.53
Corrected Temperature (°F)	46.5	129.5	80.5	51.5	41.5
Enthalpy (Btu/lbm)	82.9826	90.3296	28.0828	28.0828	82.6080
Entropy (Btu/lbm.R)	0.1694	0.1645	0.0575	0.0589	0.1684
Volume (ft^3/lbm)	0.8285	0.2340	0.0124	0.1196	0.8006

Table 11: Ambient Lab Temperature and Pressure

Ambient Temperature (°F)	73.5
Ambient Pressure (mmHg)	757.3

Table 12: Raw Data for Capillary Run#2 Mode at Five Different Flow Rates

	Point 1	Point 2	Point 3	Point 4	Point 5
Initial Temperature (°F)	72	72	70	71	72
Temperature corrections (°F)	-1.5	-1.5	-3.5	-2.5	-1.5
Initial Pressure (Psig)	73	71	74	72.5	80
Absolute pressure (psia)	87.64	85.64	88.64	87.14	94.64
Temperature Reading (°F)	45	110	92	49.5	40
Pressure Reading (psig)	47.5	139	139	54	50
Absolute pressure (psia)	62.14	153.64	153.64	68.64	64.64
Psat (psia)	57.89	154.16	123.97	63.56	53.07
Tsat (°F)	45.15	109.47	107.96	52.00	40.41
Corrected Pressure (psia)	56.56	150.06	147.06	63.56	52.06
Corrected Temperature (°F)	46.5	111.5	95.5	52.0	41.5
Enthalpy (Btu/lbm)	82.2231	91.2803	32.5040	32.5040	18.0981
Entropy (Btu/lbm.R)	0.1657	0.1679	0.0656	0.0671	0.0390
Volume (ft^3/lbm)	0.6825	0.2905	0.0128	0.1259	0.0116
Quality (Vapour = 1; Liquid = 0)	1	1	0	0.192	0
Temperature Reading (°F)	43	111	89	47	45.5
Pressure Reading (psig)	45	140	140	51	47
Absolute pressure (psia)	59.64	154.64	154.64	65.64	61.64
Psat (psia)	55.92	156.20	118.82	60.93	58.39
Tsat (°F)	42.95	110.15	108.65	49.50	37.49
Corrected Pressure (psia)	54.43	151.43	148.43	60.93	49.43
Corrected Temperature (°F)	44.5	112.5	92.5	49.5	47.0
Enthalpy (Btu/lbm)	82.0017	91.2033	31.6309	31.6309	17.6604
Entropy (Btu/lbm.R)	0.1664	0.1679	0.0641	0.0655	0.0382
Volume (ft^3/lbm)	0.7389	0.2936	0.0127	0.1270	0.0116
Quality (Vapour = 1; Liquid = 0)	1	1	0	0.186	0
Temperature Reading (°F)	55	111	85	44.5	60.5
Pressure Reading (psig)	44	141	143	47	45.5
Absolute pressure (psia)	58.64	155.64	157.64	61.64	60.14
Psat (psia)	68.50	156.20	112.20	58.39	74.91
Tsat (°F)	43.42	111.37	110.88	47.00	37.44
Corrected Pressure (psia)	54.89	153.89	152.89	58.39	49.39
Corrected Temperature (°F)	56.5	112.5	88.5	47.0	62.0
Enthalpy (Btu/lbm)	81.8478	91.6911	30.8739	30.8739	81.9525
Entropy (Btu/lbm.R)	0.1660	0.1685	0.0627	0.0641	0.1661
Volume (ft^3/lbm)	0.7282	0.2915	0.0127	0.1272	0.7235
Quality (Vapour = 1; Liquid = 0)	1	1	0	0.180	1

Temperature Reading (°F)	62	112	82	40.5	66
Pressure Reading (psig)	40	141	143	45	42
Absolute pressure (psia)	54.64	155.64	157.64	59.64	56.64
Psat (psia)	76.74	158.27	107.42	54.48	81.76
Tsat (°F)	36.98	110.43	109.93	43.00	31.05
Corrected Pressure (psia)	48.98	151.98	150.98	54.48	43.98
Corrected Temperature (°F)	63.5	113.5	85.5	43.0	67.5
Enthalpy (Btu/lbm)	83.7481	92.8479	29.7465	29.7465	83.6640
Entropy (Btu/lbm.R)	0.1702	0.1703	0.0606	0.0620	0.1700
Volume (ft^3/lbm)	0.8036	0.2976	0.0125	0.1196	0.8016
Quality (Vapour = 1; Liquid = 0)	1	1	0	0.165	1
Temperature Reading (°F)	65	112	80	39.5	69
Pressure Reading (psig)	39	143	144	44	40
Absolute pressure (psia)	53.64	157.64	158.64	58.64	54.64
Psat (psia)	80.48	158.27	104.31	53.54	85.69
Tsat (°F)	35.90	111.44	110.45	42.00	28.61
Corrected Pressure (psia)	48.04	154.04	152.04	53.54	42.04
Corrected Temperature (°F)	66.5	113.5	83.5	42.0	70.5
Enthalpy (Btu/lbm)	84.6782	93.9935	28.5150	28.5150	84.6156
Entropy (Btu/lbm.R)	0.1727	0.1720	0.0584	0.0598	0.1724
Volume (ft^3/lbm)	0.8683	0.3015	0.0124	0.1265	0.8578

Table 13: Raw Data for TVX Mode at Constant Flow Rate

	Point 1	Point 2	Point 3	Point 4	Point 5
Initial Temperature (°F)	72	72	70	71	72
Temperature corrections (°F)	-1.5	-1.5	-3.5	-2.5	-1.5
Initial Pressure (Psig)	73	71	74	72.5	80
Absolute pressure (psia)	87.64	85.64	88.64	87.14	94.64
Temperature Reading (°F)	62	118	77	39.5	62
Pressure Reading (psig)	39	154	152	43	41
Absolute pressure (psia)	53.64	168.64	166.64	57.64	55.64
Psat (psia)	76.74	171.07	99.77	53.54	76.74
Tsat (°F)	37.05	117.18	114.83	42.00	31.12
Corrected Pressure (psia)	49.04	166.04	161.04	53.54	44.04
Corrected Temperature (°F)	63.5	119.5	80.5	42.0	63.5
Enthalpy (Btu/lbm)	82.1207	94.3035	29.2219	29.2219	18.3333
Entropy (Btu/lbm.R)	0.1661	0.1719	0.0597	0.0609	0.0395
Volume (ft^3/lbm)	0.7132	0.2877	0.0125	0.1104	0.0117
Quality (Vapour = 1; Liquid = 0)	1	1	0	0.154	0

Temperature Reading (°F)	36.5	118	78	41	42
Pressure Reading (psig)	41	155	156	44	47
Absolute pressure (psia)	55.64	169.64	170.64	58.64	61.64
Psat (psia)	49.88	171.07	101.27	54.96	54.96
Tsat (°F)	39.75	117.84	116.91	43.50	38.65
Corrected Pressure (psia)	51.46	167.46	165.46	54.96	50.46
Corrected Temperature (°F)	38.0	119.5	81.5	43.5	43.5
Enthalpy (Btu/lbm)	81.6755	93.9411	29.3680	29.3680	16.8685
Entropy (Btu/lbm.R)	0.1661	0.1717	0.0599	0.0613	0.0366
Volume (ft^3/lbm)	0.7483	0.2961	0.0125	0.1215	0.0116
Quality (Vapour = 1; Liquid = 0)	1	1	0	0.164	0
Temperature Reading (°F)	42	130	84	45.5	50
Pressure Reading (psig)	45	181	183	48.5	47
Absolute pressure (psia)	59.64	195.64	197.64	63.14	61.64
Psat (psia)	54.96	198.83	110.59	59.40	63.03
Tsat (°F)	43.95	129.25	128.83	48.00	38.58
Corrected Pressure (psia)	55.40	193.40	192.40	59.40	50.40
Corrected Temperature (°F)	43.5	131.5	87.5	48.0	51.5
Enthalpy (Btu/lbm)	83.8654	93.2715	31.6582	31.6582	83.9134
Entropy (Btu/lbm.R)	0.1698	0.1693	0.0640	0.0658	0.1701
Volume (ft^3/lbm)	0.7743	0.2567	0.0127	0.1436	0.7833
Quality (Vapour = 1; Liquid = 0)	1	1	0	0.199	1
Temperature Reading (°F)	39	126	83	43	40
Pressure Reading (psig)	43	174	174	46	45
Absolute pressure (psia)	57.64	188.64	188.64	60.64	59.64
Psat (psia)	52.14	189.25	109.00	56.90	53.07
Tsat (°F)	41.85	126.28	124.99	45.50	36.32
Corrected Pressure (psia)	53.40	186.40	183.40	56.90	48.40
Corrected Temperature (°F)	40.5	127.5	86.5	45.5	41.5
Enthalpy (Btu/lbm)	81.8276	93.8683	31.9163	31.9163	17.7846
Entropy (Btu/lbm.R)	0.1664	0.1703	0.0644	0.0664	0.0384
Volume (ft^3/lbm)	0.7530	0.2618	0.0127	0.1551	0.0116
Quality (Vapour = 1; Liquid = 0)	1	1	0	0.209	0