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Refrigeration Cycle

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Abstract

Refrigeration technologies have enabled great advances to the modern society. Contrary to what one might think the first civilizations that used some sort of technique to refrigerate a room go back to prehistoric times. On these ancient times ice was stored and packed with hay and other types of insulating materials in order to have a cool environment to preserve food during hot days. The uses of the now known refrigeration cycle have broadened to more than just food preservation. Up to this day cryogenics exists, where super low temperatures can be achieved and the quest to absolute zero has just begun.

This experiment will replicate the vapor compression cycle that was first proposed in 1805 by Oliver Evans. The upcoming report will show the acquired understanding of the refrigeration cycle and its applications. In order to analyze this system several assumptions where made such like a isentropic process at the compressor a isenthalpic expansion in the throttling valve. Diagrams will be provided to depict these thermodynamic processes in addition to computing the heat transferred to the system and the work input to the compressor.

It was found that lowering fan speeds of condenser and evaporator in the experiment reduced the compressor and coefficient of performance of the cycle. It was suggested that this may have been because at higher fan speed, convection coefficient increased, increasing heat transfer in the evaporator and condenser with the surroundings, thus reducing the work of the compressor to the refrigerant. Vapor compression cycle was more efficient with fans of evaporator and condenser at highest speeds.

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Symbol List and Abbreviations

Table 1 Symbol Table

Symbols:	Meaning
G0-4	Readings from gages
$\mathbf{h_{0\text{-}4}}$	Enthalpy R134a
$\boldsymbol{h_{0s}}$	Isentropic Enthalpy
$\mathbf{P_{0\text{-}4}}$	Pressure (psi)
$\mathbf{Q_{in}}$	Heat Energy In (Btu/lbm)
$\mathbf{Q_{out}}$	Heat Energy Out (Btu/lbm)
$\mathbf{s_{0\text{-}4}}$	Entropy R134a
$\boldsymbol{T_c}$	Coolest Temperature in Cycle
$\boldsymbol{T_h}$	Hottest Temperature in Cycle
$\mathbf{v_{0\text{-}4}}$	Specific Weight
$\mathbf{W_{compressor}}$	Compressor Work (Btu/lbm)
$\mathbf{W_{cycle}}$	Cycle Work (Btu/lbm)
$\boldsymbol{\beta}$	Refrigeration Performance
$\boldsymbol{\beta_{max}}$	Refrigeration Performance based on Carnot Cycle
ŋ	Compressor Efficiency

Figure List

Table List

Introduction

A vapor refrigeration cycle is composed of several components which can be seen in the Figure 1 below. These components are the evaporator, compressor, condenser and throttling valve (Brokowski). The figure 1 shows the simplified stages at which the cycle occurs. The refrigerant in this case R134a receives heat from the surroundings at the evaporator and is passed to the compressor ideally as a saturated vapor. At the stage one the compressor increases the pressure in the refrigerant which is accompanied also by an increase in temperature. Ideally this compression is isentropic; this heated fluid is then passed to the condenser in order to reject heat to the surroundings. Here at stage two the temperature is lowered and then is passed to the stage three. This stage is where a sudden change in pressure takes the saturated liquid and is further cooled this cooling ideally occurs as an isenthalpic process. The loop is closed at stage four where the cooled refrigerant is then returned to evaporator.

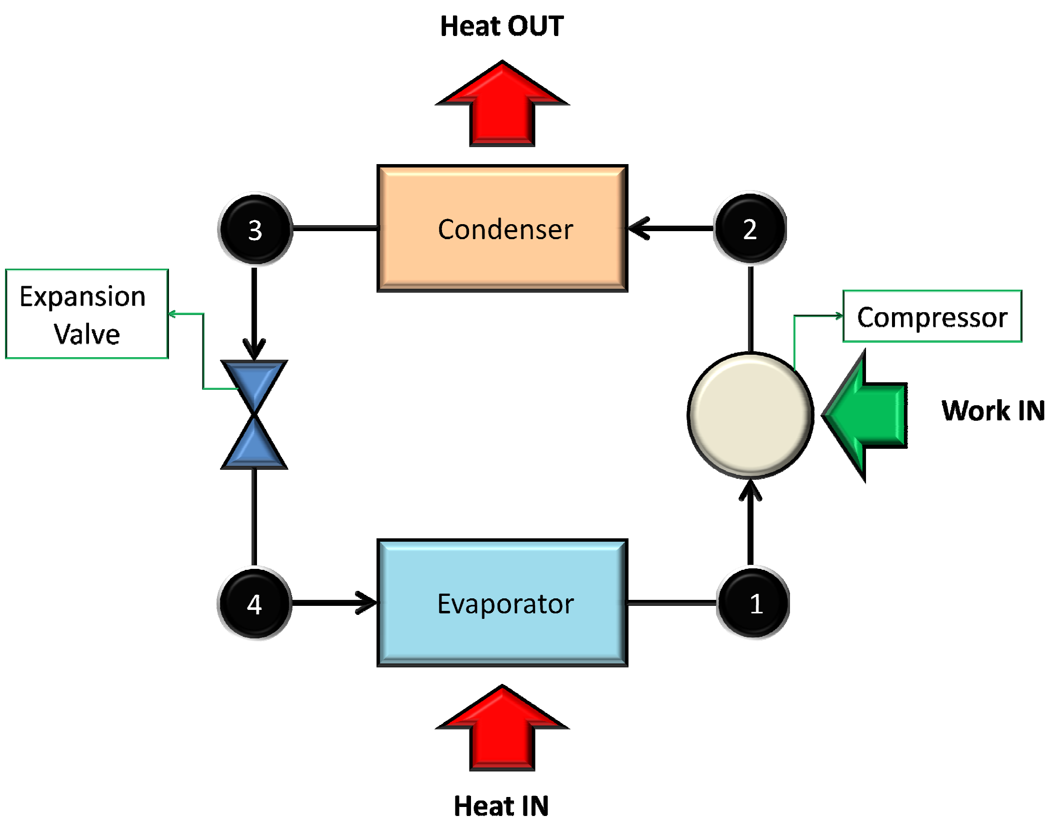

Figure 1: Schematic for vapor compression cycle

Experimental Setup

This experiment consists on taking pressure and temperature measurements at selected points. These points are marked as G0, G1, G2, G3 and G4. These correspond respectively to the pressure gages installed after the evaporator, compressor, expansion valve and evaporator. Temperatures are also going to be collected from these discrete sections in addition to data on the physical state of the refrigerant; which can be seen trough small windows installed in the equipment. The Figure 2 below shows the equipment setup.

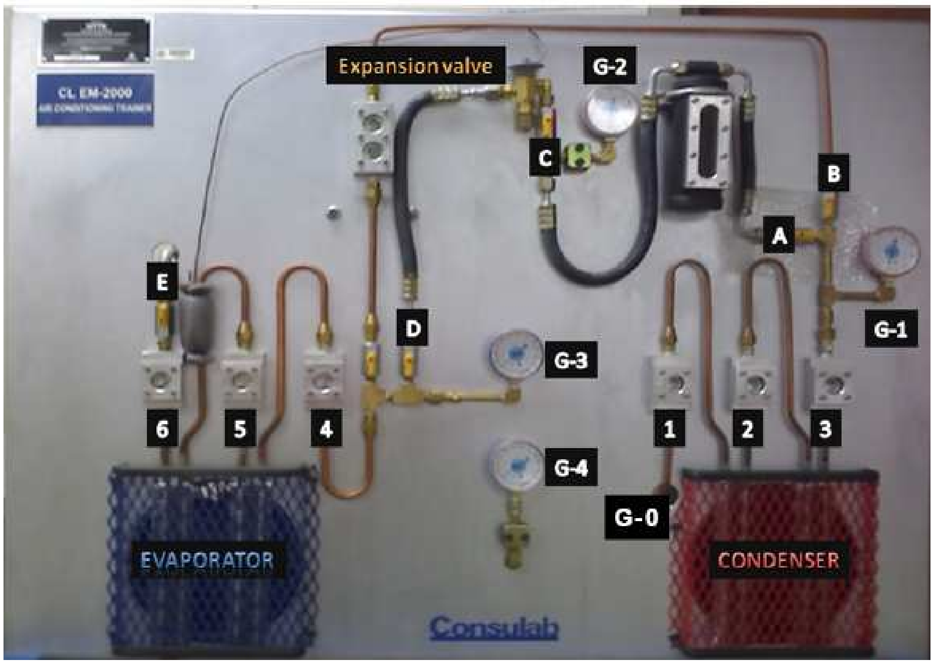

Figure 2: Vapor compression demonstrative equipment

In addition to the components mentioned above there is a certain number of accessories that are being connected to the refrigerant lines. These accessories include check valves tee valves and other types of fittings within others. The losses produced by these types of equipments will be neglected, since the main purpose of the experiment is the thermal behavior of the system. Readings obtained for max fan speed are presented in tables 2 and 3 while readings for low fan speed of evaporator and condenser are presented in table 4 and 5 in experimental data.

Experimental Data

Table 2: Fans @ High Speed

	Location	Pressure (psi)	Temperature (^oF)
Station	G0	225	185.18
	G1	225	138
	G2	225	140
	G3	20	22
	G4	21	23
	Location	Physical State	
Sight glass	1	vapor	
	2	two phase liquid vapor	
	3	two phase liquid vapor, more liquid	
	4	two phase liquid vapor, more vapor	
	5	vapor	
	6	vapor	

Table 3: Properties for Fans @ High Speed

	Fans @ High Speed				
Stages	0	1	2	3	4
P (psi)	240	240	240	35	36
v (ft^3/lb)	0.2851	0.0152	0.0152	0.6331	1.3203
T (^oF)	185.18	138	140	22	23
s (Btu/lb R)	0.2432	0.1142	0.1142	0.1242	0.2202
h (Btu/lb)	134.70	58.61	58.61	58.61	104.89

Table 4: Fans @ Low Speed

	Location	Pressure (psi)	Temperature (^oF)
Station	G0	260	202.1
	G1	260	149
	G2	260	150
	G3	22	25
	G4	23	25
	Location	**Physical State**	
Sight glass	1	vapor + droplets	
	2	two phase liquid vapor, more vapor	
	3	two phase liquid vapor, more liquid	
	4	two phase liquid vapor, more vapor	
	5	vapor + some liquid	
	6	vapor	

Table 5: Properties for Fans @ Low Speed

	Fans @ Low Speed				
Stages	0	1	2	3	4
P (psi)	260	260	260	22	23
v (ft^3/lb)	0.29860	0.01560	0.01560	0.64599	1.26660
T (^oF)	202.1	149	150	25	25
s (Btu/lb R)	0.2503	0.1210	0.1210	0.1322	0.2200
h (Btu/lb)	139.32	62.76	62.76	62.76	105.32

Analysis and Results

The following assumptions were made in the thermodynamics cycle of refrigeration and are define according to Figure 3. Heat transfer from state G0 to state G1 occurs at constant pressure in the condenser. State one and two are the same state and assumed to be a saturated liquid. From state two to state three it assumed to be an isenthalpic expansion. Heat transfer from state G3 to state G4 occurs at a constant pressure in the evaporator. State four is assumed to be a saturated vapor. Compression from state four to zero ($G0_s$) is isentropic.

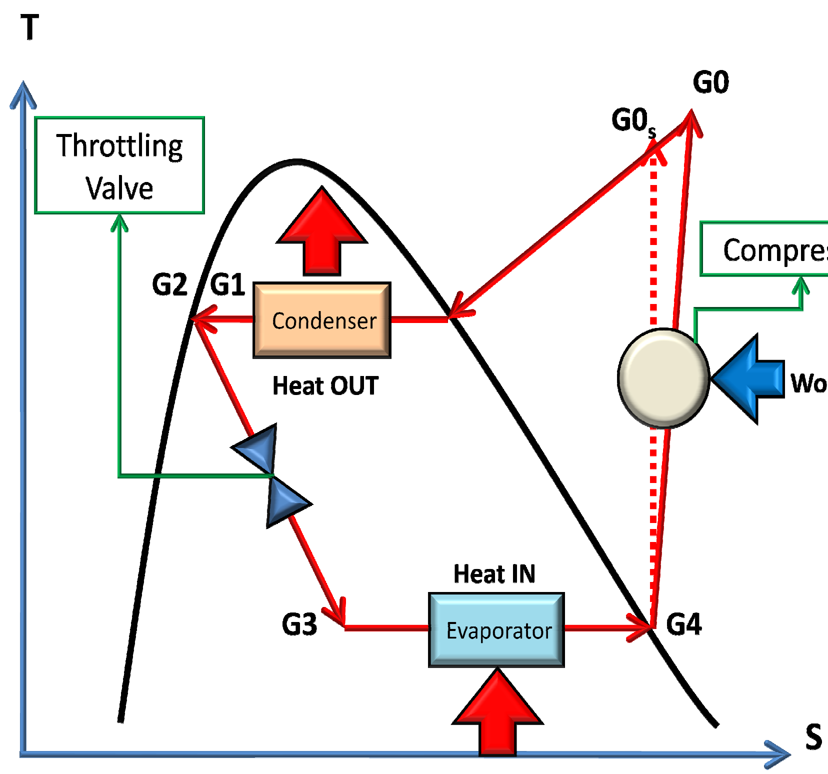

Figure 3: Thermodynamic diagram with overlapped components.

The properties of each stage were read from thermodynamics tables (Moran & Shapiro, 2004) using interpolated values of the temperature and pressure measurements. All measurements were corroborated win the program (EES- Engineering Equation Solver). The programming code is provided in the appendix. The following example calculations were made using low speed fan thermodynamic properties obtained by EES. The heat transfer of the condenser is the heat loss, in other words Q_{out} and is given by;

$$Q_{out} = h_0 - h_2 = 136.4 - 63.86 = 72.56\, BTU/lbm$$

And the heat gain by the evaporator is equal to Q_{in} and is given by;

$$Q_{in} = h_4 - h_3 = 106.7 - 63.86 = 42.83\, BTU/lbm$$

The work done by the compressor is the same as the work in the cycle and is given by;

$$W_{compressor} = h_0 - h_4 = 136.4 - 106.7 = 29.73\, BTU/lbm$$

The coefficient of performance (COP) of the cycle is given by;

$$\beta = \frac{Q_{in}}{Wcycle} = \frac{Q_{in}}{Wcompressor} = \frac{42.83}{29.73} = 1.441$$

The max coefficient of performance is based on the Carnot cycle and was calculated with:

$$\beta_{max} = \frac{T_c}{T_c + T_h} = \frac{25 + 459.67}{(202.2 + 459.67) - (25\ + 459.67)} = 2.767$$

Compressor efficiency was calculated in both cases by the following formula:

$$\eta_{\text{compressor}} = \frac{h_{0s} - h_4}{\text{Wcompressor}} = \frac{124.7 - 106.7}{29.73} x100 = 60.45\%$$

Table 6 presents results obtained in both conditions using Engineering Equations Solver (EES).

Table 6: Summary of Results

	Fans at Maximum Speed	**Fans at lower Speed**
Qin (Evaporator) [Btu/lbm]	46.5	42.83
Qout (Condenser) [Btu/lbm]	73.4	72.56
$W_{compressor}$ [Btu/lbm]	26.95	29.73
$\boldsymbol{\beta}$	1.725	1.441
$\boldsymbol{\beta_{max}}$	2.949	2.737
η	63.67%	60.45%

Figure 4, Figure 5, Figure 6 and Figure 7 shows the various relationship of the refrigeration system for fan at max and lower speeds as required by experiment manual.

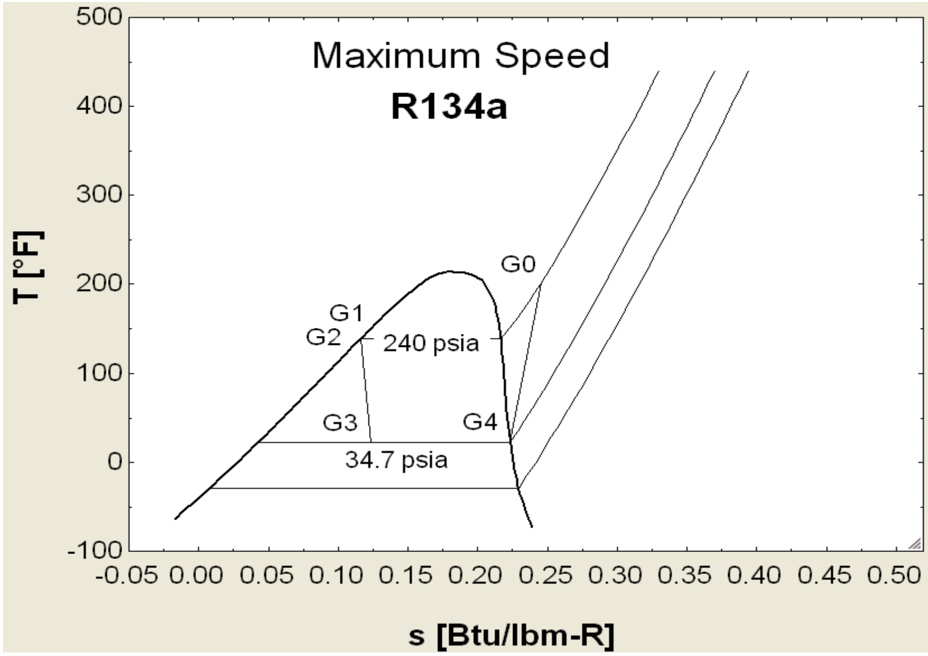

Figure 4 - T vs. s graph for maximum fan speed in condenser and evaporator

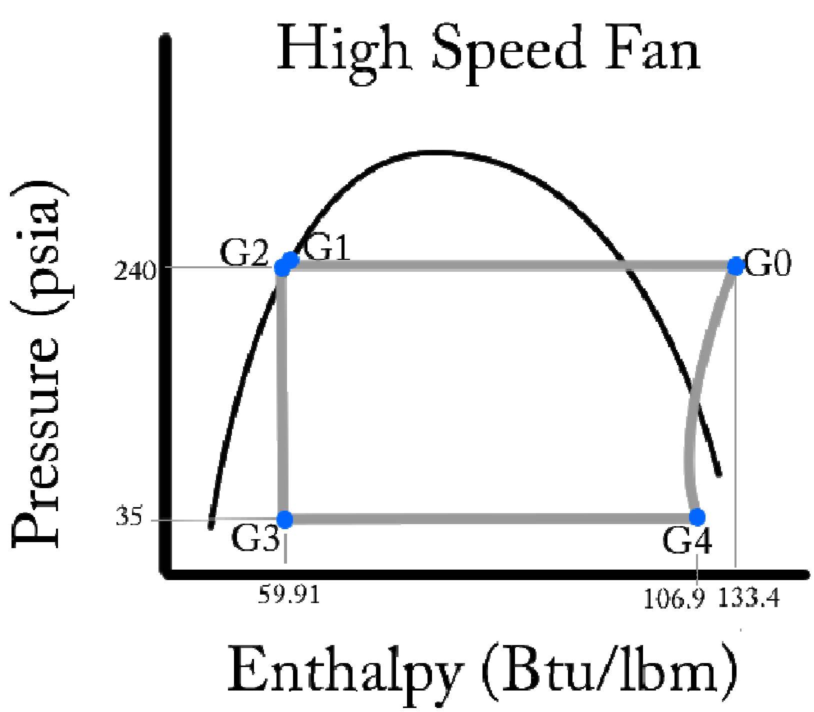

Figure 5 - Maximum fan speed pressure vs. enthalpy graph

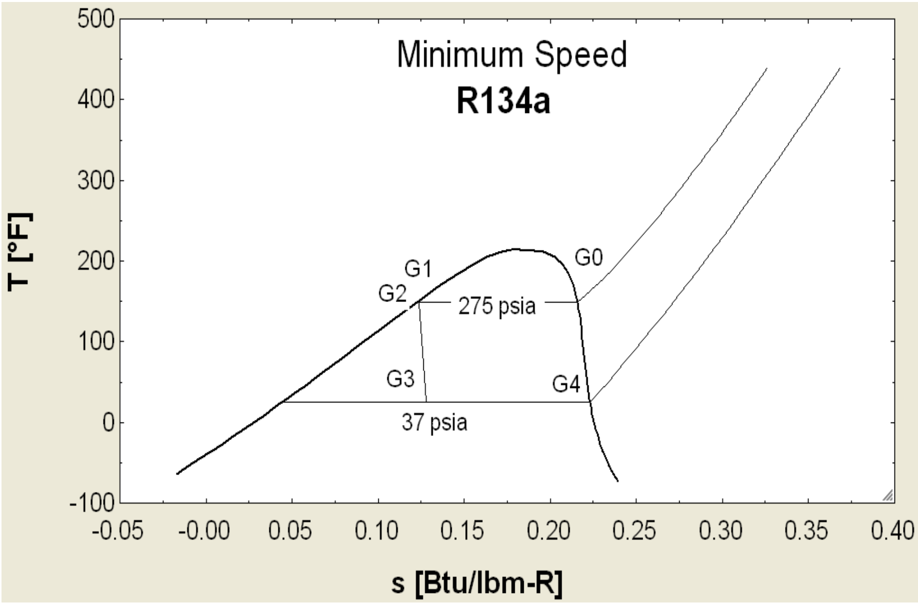

Figure 6 - Minimum fan speed temperature vs. entropy graph

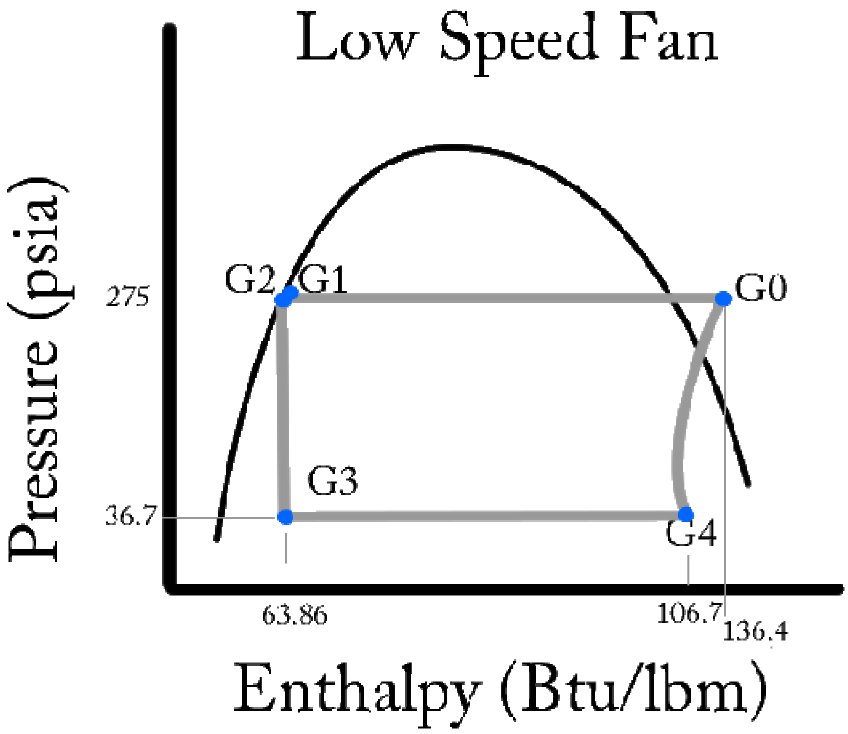

Figure 7 - Minimum fan speed pressure vs. enthalpy graph

Discussion

It was seen that the refrigerant enters the evaporator as a two phase liquid vapor mixture. Ideally this two-phase liquid is turned to steam with the heat addition received in the evaporator heat exchanger. Had it been the case where droplets of liquid refrigerant entered the compressor could significantly hinder the performance of this component. This wet compression of the refrigerant could potentially damage this component via cavitation. Ideally this compression is done isentropically, with no heat addition from its surroundings and internally reversible. Unfortunately this is not the case of a real vapor compression cycle and due to this issue the efficiency of the system is lowered.

Once the compressor has increased the pressure of the refrigerant it is passed through the condenser, it is here that heat is rejected to the surroundings. In this component the refrigerant entered in both conditions as a superheated vapor and should leave as a saturated liquid. The next step for the vapor compression refrigeration cycle is the isenthalpic throttling valve. This expansion cooled the refrigerant and lowers its pressure to the pressure inside the evaporator. This cooled two-phase liquid vapor mixture is then returned to the evaporator to close the loop.

It was seen when fans of condenser and evaporator were at the maximum speed, heat entering (Q_{in}) and exiting (Q_{out}) in the system were increased. This was expected and can be explained as convection coefficient is a function directly proportional to the velocity of the air passing through the heat exchangers in the evaporator and condenser thus increasing heat transfer to the surroundings. It was found that compressor did more work when fans were at lower speed than higher. This can be explained with the fact that the cycle was operating with a variable orifice expansion valve. In the second condition, fans at low speed, the valve opened to increase the amount of refrigerant passing through the lines, therefore more work was needed in the compressor. This valve effectively helped the system to better regulate the flow of the refrigerant even when conditions were not optimum.

Conclusion

This experiment was carried out in order to observe the properties and functions of a simple vapor refrigeration cycle with a variable orifice expansion valve. Temperature and pressure measurements were taken with specially designed gages and a thermocouple located at the critical points. With the properties of the refrigerant R134a at all the stages of the refrigeration cycle known, the heat transfer at the condenser and evaporator were calculated. The heat absorbed by the system (evaporator) with the fans at their maximum velocity was determined to be 46.5 Btu/lbm. The heat given off by the condenser was calculated to being 73.4 Btu/lbm. The work done by the compressor was also calculated to being 26.95 Btu/lbm. With this, the refrigeration performance of the cycle was calculated and was equal to 1.725. It was also seen that by running the cycle with the fans at a lower speed the coefficient of performance lowered to 1.441. This was due to an increase in work done by the compressor due to the opening of the variable orifice expansion valve. The respective values for the heat input, output and the work done by the compressor are: 42.83, 72.56, 29.73 (all in Btu/lbm) respectively.

The variable orifice expansion valve provided way to control the amount of refrigerant passing through the lines. With ideal operating conditions, fans at high speed, the amount of refrigerant needed was less and so the valve permitted a lower volume flow rate of refrigerant to enter. On the contrary with the evaporator and the condenser having their fans at lower speeds resulted in a increase of the refrigerant flow, making the compressor require more work as it needs now to supply more refrigerant.

Another observation made was that the temperatures achieved in the lines that connect the expansion valve to the evaporator showed high amounts of frozen moisture as can be seen in **Error! Reference source not found.**. This ice formation on the lines of the heat exchanger produces an insulation layer that hiders the heat transfer. By acquiring a dehumidifying unit for the laboratory this effect could be significantly diminished.

References

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Appendix