Mechanical Heat Pump

201, Thursday B-1

Meredith Addison, James Hughes, Taylor Kellis,
Michael Luketich, Kyler Madara, Nicholas Yuhas

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Nomenclature

Variables	Description	Units
T_1	Temperature of refrigerant exiting evaporator	K
T_2	Temperature of refrigerant exiting compressor	K
T_3	Temperature of refrigerant exiting condenser	K
T_4	Temperature of refrigerant exiting expansion valve	K
T_5	Temperature of water entering condenser	K
T_6	Temperature of water exiting condenser	K
CP	Coefficient of Performance	unit-less
m	Mass flow rate of refrigerant	g/s
W	Power input	W
h_n	Enthalpy at T_n (where n = cycle stage)	kJ/kg
Q_L	Heat loss	J/s
C_P	Heat capacity of refrigerant	J/(g K)

1.0 Introduction and Background

A mechanical heat pump is a popular process used for heating and cooling. It is often used in space heating, process heating, dehumidification, and heat recovery. It can be a low-cost and high-efficiency method of both heating in the winter and air conditioning in the summer for residential and commercial buildings [1]. Industrial uses include drying and dehumidification in timber and paper industries. Additionally, they can be added to distillation processes to make them more efficient. As a whole, heat pumps trim energy consumption for a variety of processes, making them cheaper and better for the environment [2].

The process consists of four stages. First, air flows into the evaporator and transfers heat to the refrigerant. The heated refrigerant, which is now vapor, flows into the compressor where it is super-heated and then flows into the heat exchanger. At the same time, water flows into the heat exchanger. This causes a phase change, where the refrigerant is condensed and flows through an expansion valve back into the evaporator [3]. During the experiment, the following parameters were recorded: the temperatures at each stage for the water and refrigerant, condenser and evaporator pressures, the power input to the system, the water flow rate and the refrigerant flow rate. The measurements were then used to complete the technical objectives.

The first objective was to evaluate the effect of the condenser water flow rate on three different parameters, which include temperature of water exiting the condenser, rate of energy delivered by the mechanical heat pump, and coefficient of performance of the system, all at steady-state. The temperature of the water delivered by the mechanical heat pump is T6 on the display (T_h). The rate of energy delivered by the mechanical heat pump is the power output (W). The coefficient of performance can be calculated using Equation 1:

$$COP = \frac{Q_W}{W} = \frac{m_W C_{p_w} \Delta T_w}{W} \quad (1)$$

C_{pw} is the specific heat capacity of water, ΔT_w is the difference in the exiting and entering water temperatures from the exchanger, and m_w is the mass flow rate of the water.

The second objective was to generate temperature-enthalpy (T-H) and pressure-enthalpy (P-H) diagrams for the R134a vapor-compression cycle operating at steady-state with the condenser water flow rate at 40 g/s and 15 g/s. This can be achieved by plotting the measured temperature and pressure on the given Pressure-Enthalpy diagram. First, the evaporator temperature and temperature after expansion vs. evaporator pressure were plotted. Then temperature after compression and condenser temperature vs. condenser pressure were plotted, and the points were connected.

The third objective was to determine the steady-state rate of heat loss to the environment from the compressor at each condenser flow rate investigated. Heat loss can be calculated from Equation 2:

$$\dot{W} - \dot{m}(h_2 - h_1) = \dot{Q}_L \quad (2)$$

Q is the rate of heat loss, W is the power input, m is the flow rate of the refrigerant, and the h values represent the enthalpy change in the refrigerant across the compressor. The heat loss contributes to determining the efficiency of the process.

Mechanical heat pumps are becoming a popular and efficient process for many uses across industries. Heat pumps are four-stage cycles consisting of an evaporator, compressor, condenser and expansion valve. The main objective of the experiment was to understand this process and determine at what water flow rate the heat pump is most efficient. This was done through completing the three objectives, the first being to evaluate the effect of the water condenser flow rate through calculating the coefficient of performance. The second objective was to generate temperature-enthalpy and pressure-enthalpy diagrams; finally, the third was to determine the rate of heat loss to the environment from the compressor.

2.0 Experimental Methodology

2.1 *Equipment and Apparatus* – A mechanical heat pump was used to complete the technical objectives in the experiment by enabling measurements to be taken to calculate heat loss from the compressor and to evaluate the effect of the condenser water flow rate on the coefficient of performance of the heat pump. The actual mechanical heat pump equipment used is shown in Figure 1 and Figure 2.

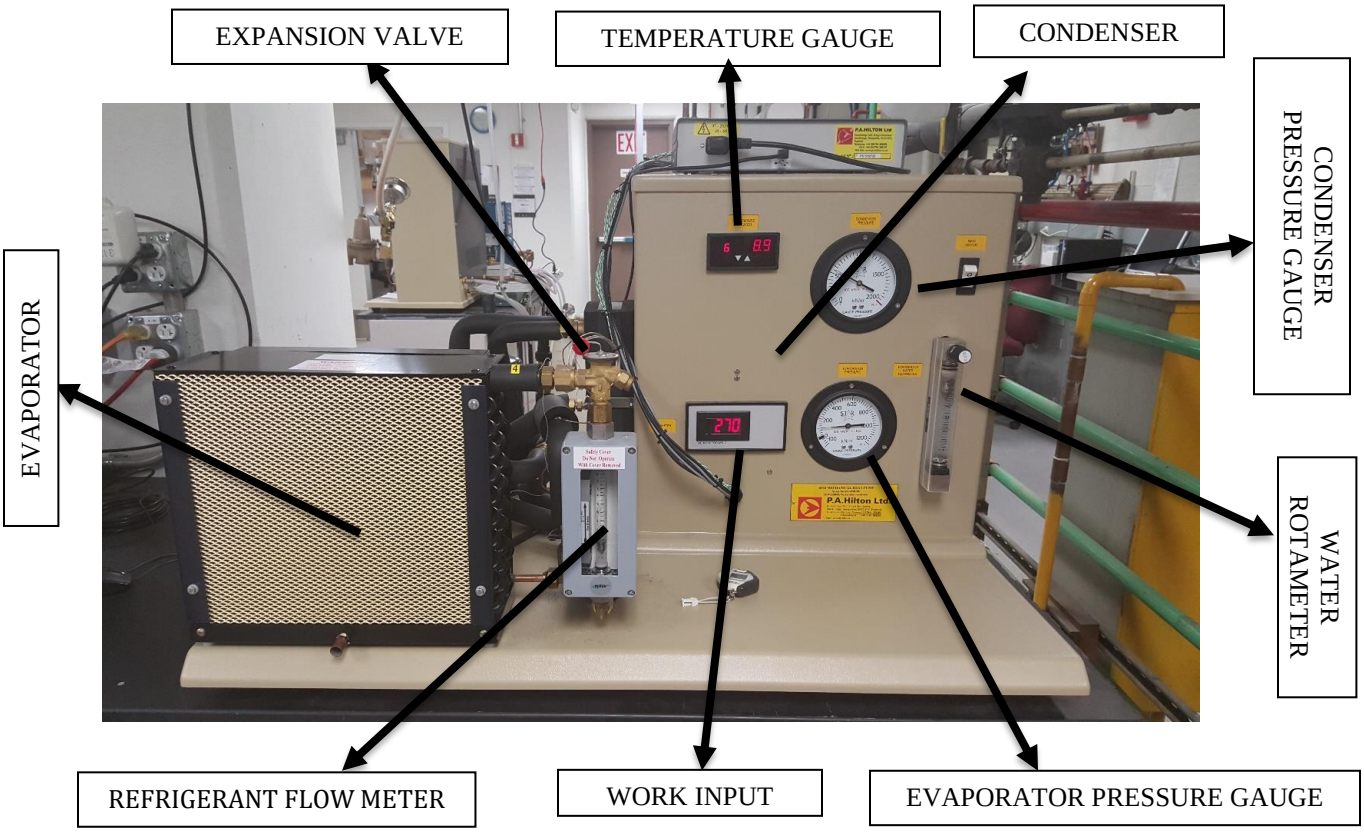


Fig. 1. Equipment used in the experiment

A bird's eye view of the remaining components of the heat pump behind the evaporator is shown in Figure 2.

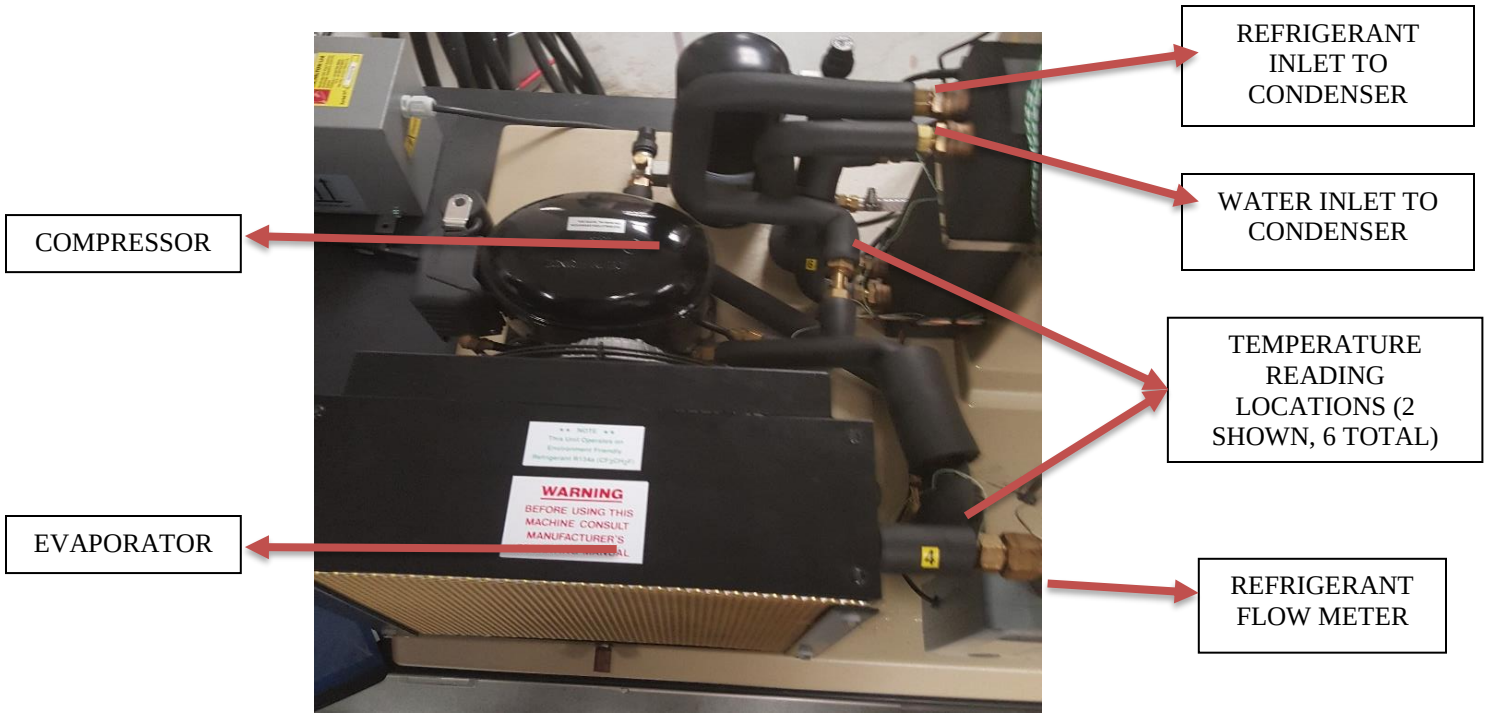


Fig. 2. Bird's eye view from above the evaporator

The heat pump functions with two main flows, the refrigerant flow and the water flow. During the experiment, refrigerant is cycled through the key parts of the heat pump: evaporator, compressor, condenser, and expansion valve, in the order listed. As the refrigerant flows through the evaporator, it experiences a phase change going from a liquid to gas by absorbing the heat from the air taken in from the room by the evaporator. After the vaporized refrigerant passes through the evaporator, its temperature is measured before it enters the compressor. Pressure on the refrigerant is increased, putting it in a superheated vapor state. From the compressor, the refrigerant temperature is measured again, and then it enters the condenser. The refrigerant is not the only flow rate to pass through the condenser. An external water source has its temperature measured and flows into the condenser as well. At this time, heat transfer occurs because the water source has a lower temperature than the refrigerant, so the refrigerant condenses as its heat is transferred to the water. While this occurs, the water increases in temperature and the once superheated vapor refrigerant condenses and becomes a liquid. The heated water exits the condenser and then has its final temperature is measured. At the same time, the refrigerant also exits the condenser and has its temperature

measured. The refrigerant then enters an expansion valve which decreases pressure. The liquid refrigerant becomes a vapor-liquid mixture and has its temperature is recorded once more before it reenters the evaporator, to start the cycle over again. Figure 3 is a flow chart that summarizes the refrigerants path and where its temperatures were measured through the mechanical heat pump.

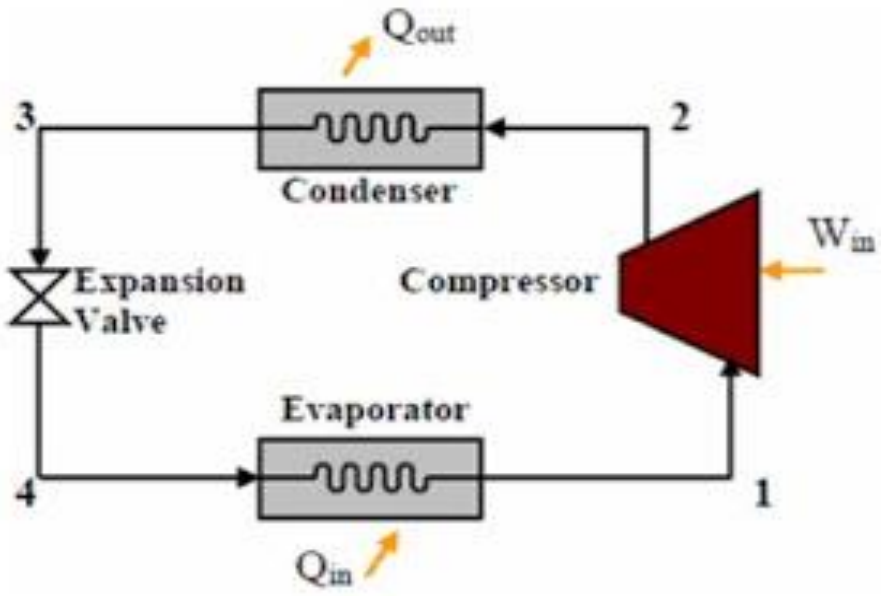


Fig. 3. Refrigerant Vapor Compression Cycle Flow Chart [4]

2.2 *Experimental Procedures* – The first technical objective was to evaluate the effect of the condenser inlet water flow rate at steady-state on the exiting water temperature, rate of energy delivered, and the coefficient of performance of the mechanical heat pump. The data needed to evaluate the water's flow rate on these parameters were found by varying the inlet condenser water flow rate while keeping the remaining independent experimental variables constant.

To start the experiment, the water rotameter was opened all the way to allow maximum flow, and the inlet water pressure was adjusted on a needle valve to keep its pressure between 20-60 psig. At this time the condenser inlet water flow rate was set at 50 g/s and the needle valve pressure was adjusted to keep the pressure between 20-60 psig. The system was run until steady-state was reached. Steady-

state for the system was defined when the six temperatures (T_1: refrigerant exiting evaporator; T_2: refrigerant exiting compressor; T_3: refrigerant exiting condenser; T_4: refrigerant exiting expansion valve; T_5: water entering condenser; T_6: water exiting condenser), condenser and evaporator pressures, water and refrigerant flow rates, and power input to the compressor all stabilized. These variables were measured using a digital display, pressure gauges, and flow meters which were monitored manually until steady-state was reached. Once steady-state was achieved, the variables were recorded.

The next run was at a condenser inlet water flow rate of 45 g/s where the same steps were followed and data was recorded at steady-state. After the 45 g/s run, the condenser inlet water flow rate was reduced again by 5 g/s, continuing until the final run at 15 g/s reached steady-state and was recorded. Eight runs were recorded and analysis was performed on the data to accomplish technical objective one.

Technical objectives two and three were accomplished by using the data recorded for technical objective one (shown in Tables 3, 4, 5, and 6), so no additional procedures were carried out.

3.0 Results

During the experiment, the data was recorded and imported to an Excel spreadsheet. Each variable was measured for a water flow rate varying from 50 (g/s) to 15 (g/s), with an interval of 5 (g/s) in between each. Based on the values that were collected, a chart was constructed by plotting water flow rate versus exiting water temperature (T_6). Figure 4 and Figure 5 show these charts for Trial 1 and Trial 2.

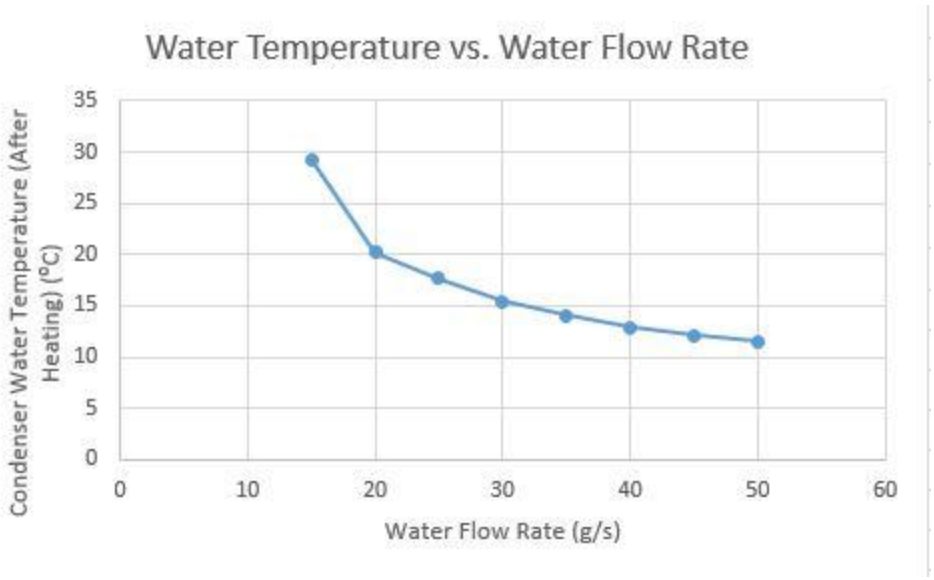


Fig. 4. Trial 1, Plot of Condenser Water Temperature versus Water Flow Rate

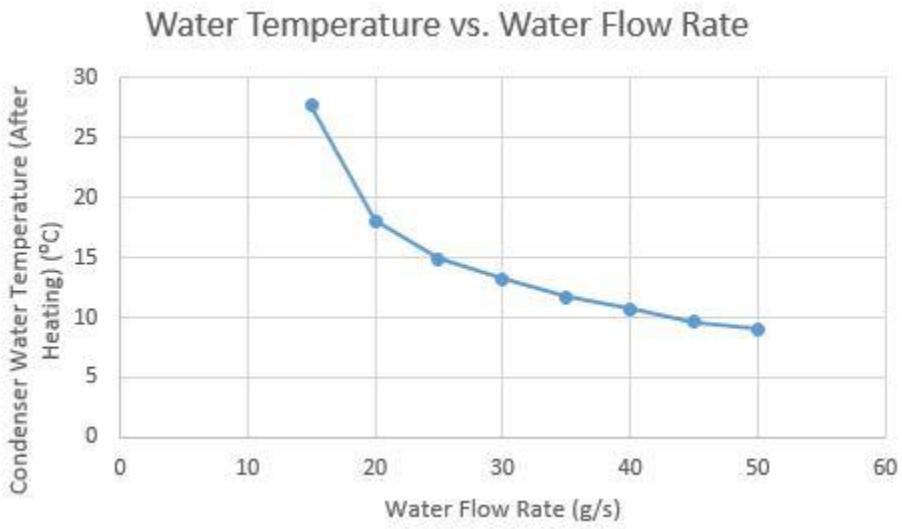


Fig. 5. Trial 2, Plot of Condenser Water Temperature versus Water Flow Rate

Next, the energy delivered to the water from the refrigerant was plotted versus the water flow rate. Figure 6 and Figure 7 show these relationships for Trial 1 and Trial 2, respectively.

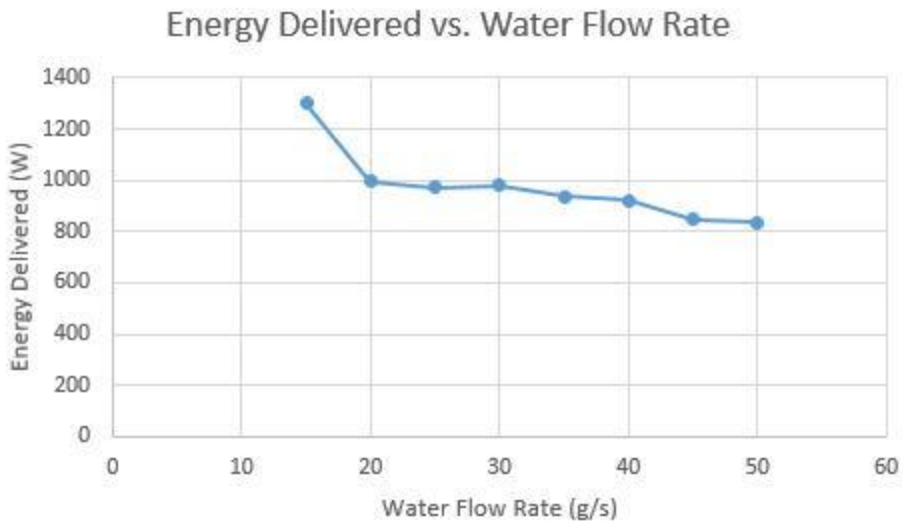


Fig. 6. Trial 1, Plot of Energy Delivered versus the Water Flow Rate

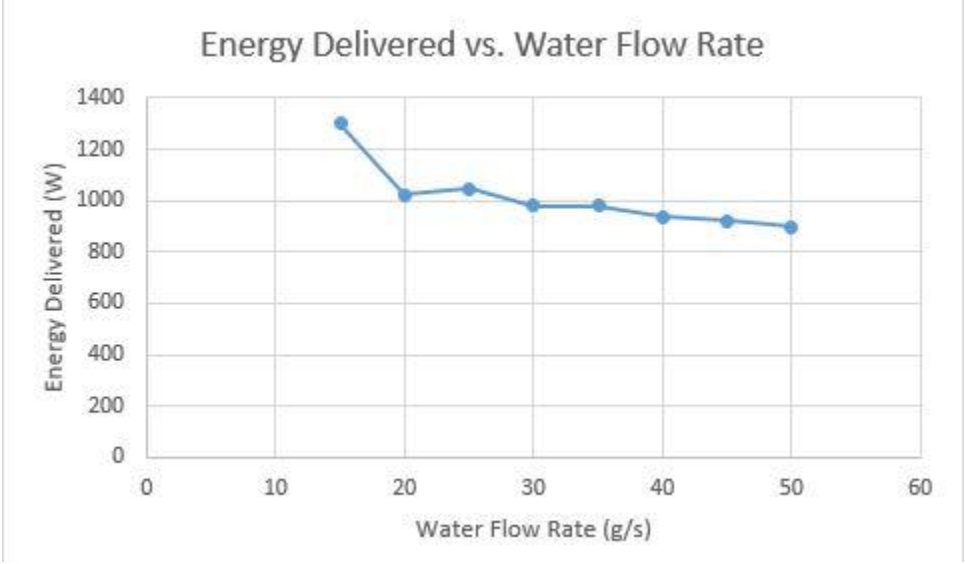


Fig. 7. Trial 2, Plot of Energy Delivered versus the Water Flow Rate

The final part of the first objective was to analyze the effects of coefficient of performance versus water flow rate. Equation 1 was used to calculate the coefficient of performance for each water flow rate. These calculated values can be found in Appendix A-1. Figure 8 and Figure 9 show coefficient of performance versus water flow rate.

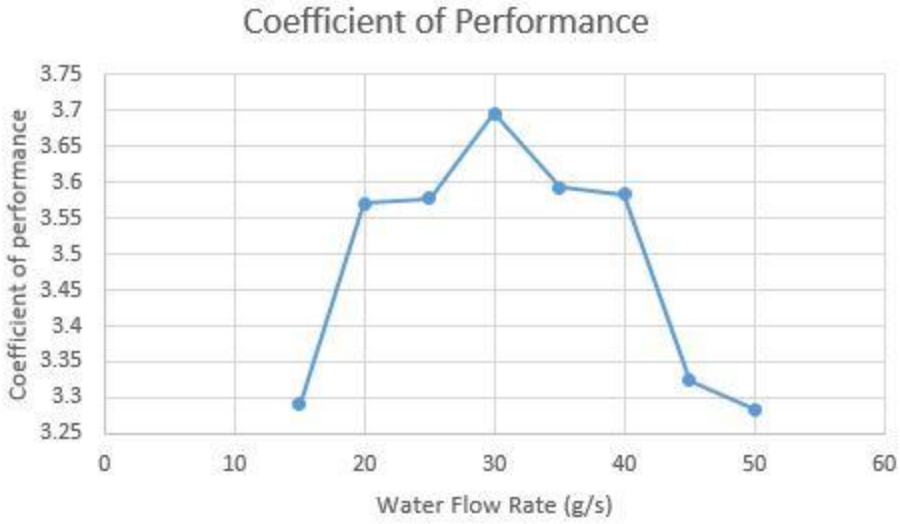


Fig. 8. Trial 1, Plot of the Water Flow Rate versus the corresponding Coefficient of Performance

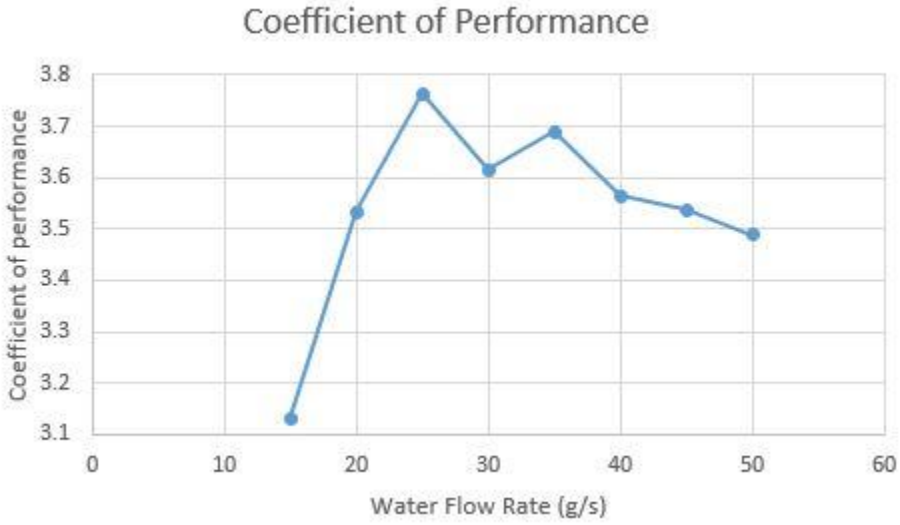


Fig. 9. Trial 2, Plot of the Water Flow Rate versus the corresponding Coefficient of Performance

The second technical objective was to create the Pressure-Enthalpy diagrams and Temperature-Enthalpy diagrams when the water flow rate was 40 g/s and 15 g/s. The Pressure-Enthalpy diagrams were created to calculate the enthalpy at each of the four r134a temperatures. The enthalpies were found by plotting the temperature of the refrigerant exiting the evaporator (T_1) and the temperature exiting the expansion valve

(T_4) versus the evaporator pressure, and the temperature exiting the compressor (T_2) and the temperature exiting the condenser (T_3) versus the condenser pressure. These four points were plotted on a general r134a Pressure-Enthalpy diagram, and the enthalpy at each point was recorded. The Pressure-Enthalpy diagrams generated are shown in Figures 10, 11, 12, and 13. The enthalpies could then be plotted versus their respective temperatures. Figures 14, 15, 16, and 17 display the Temperature-Enthalpy diagrams for Trial 1 15 g/s, Trial 1 40 (g/s), Trial 2 15 (g/s) and Trial 2 40 (g/s), respectively.

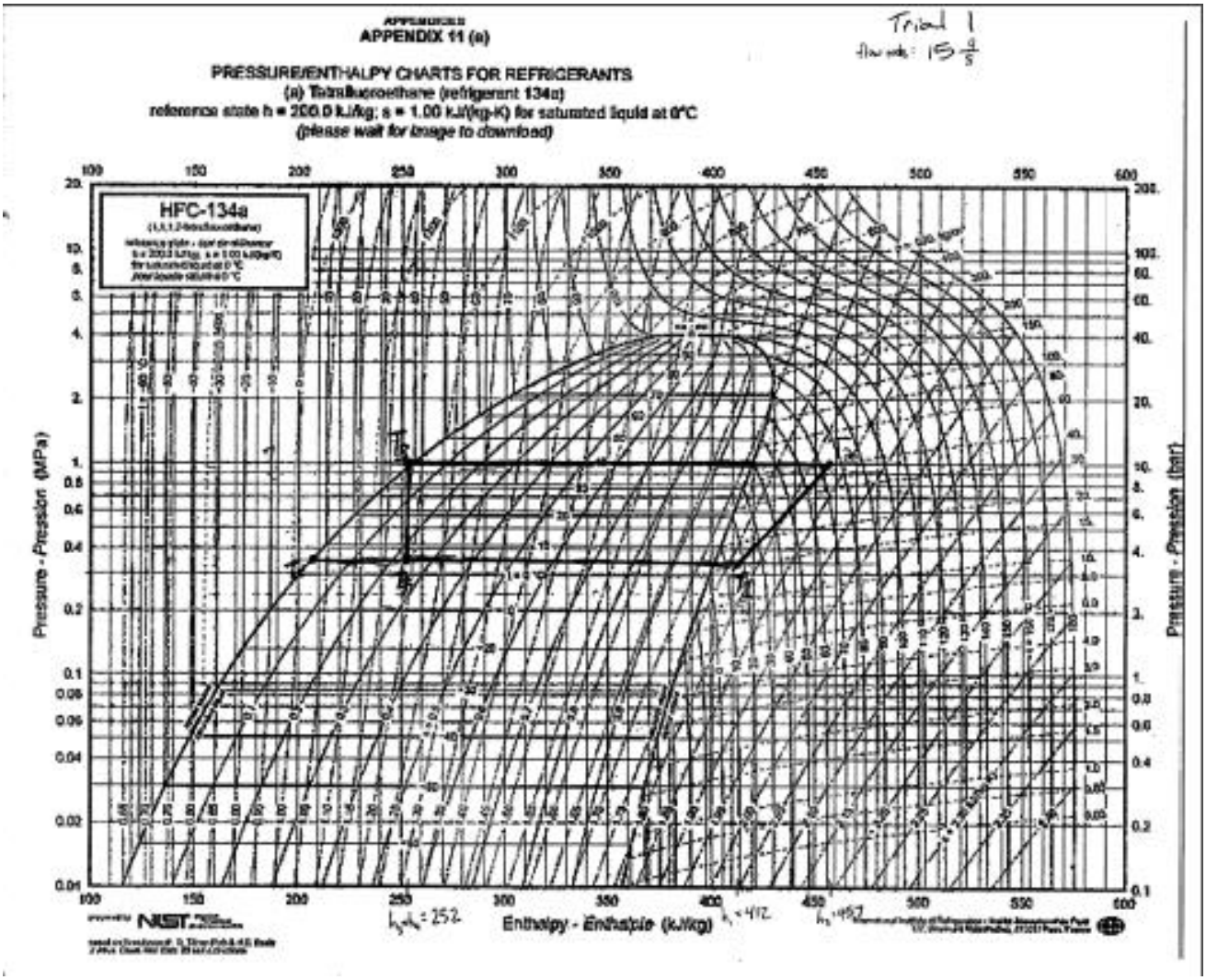


Fig. 10. Trial 1, Pressure-Enthalpy diagram plotted for 15 g/s

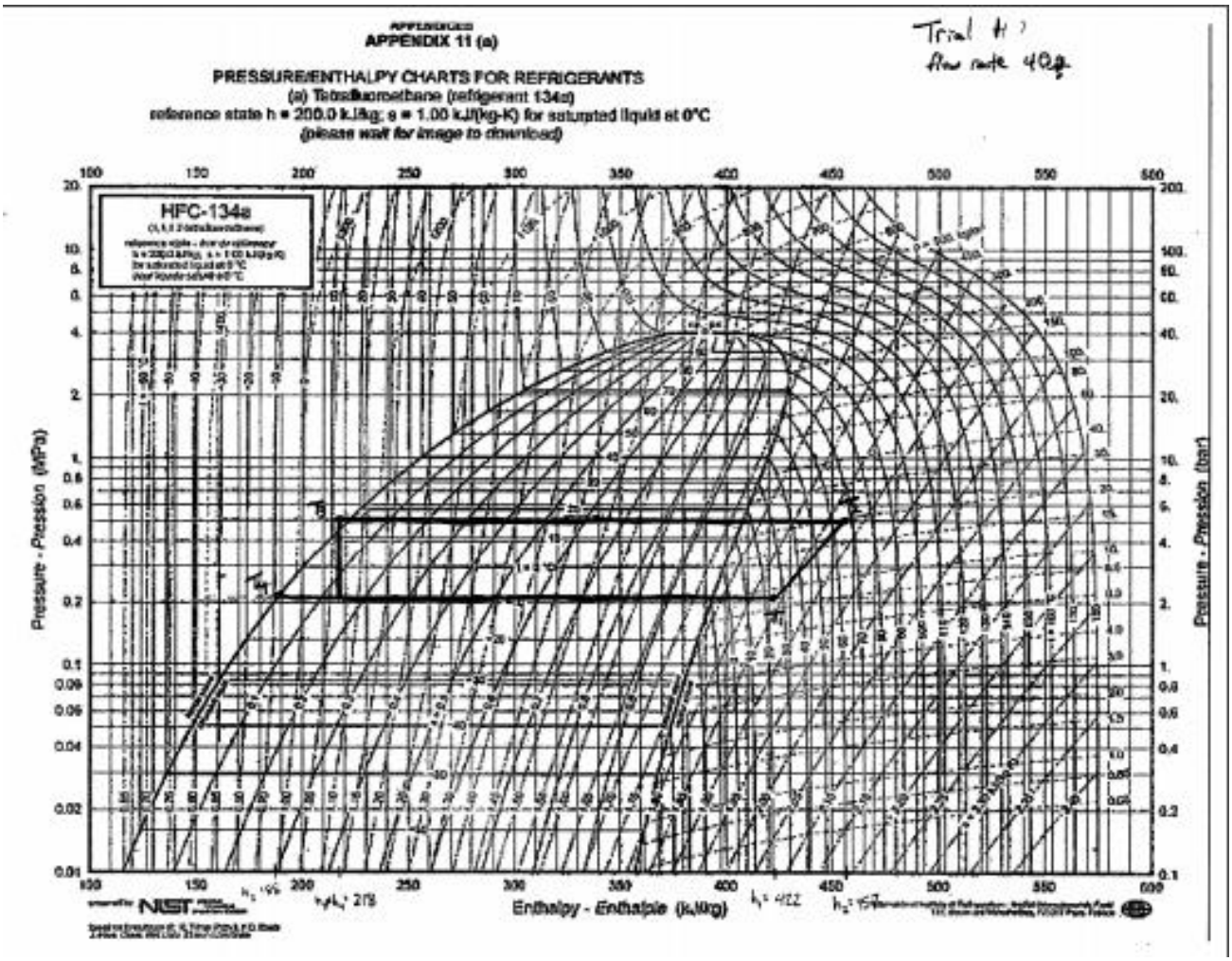


Fig. 11. Trial 1, Pressure-Enthalpy diagram plotted for 40 g/s

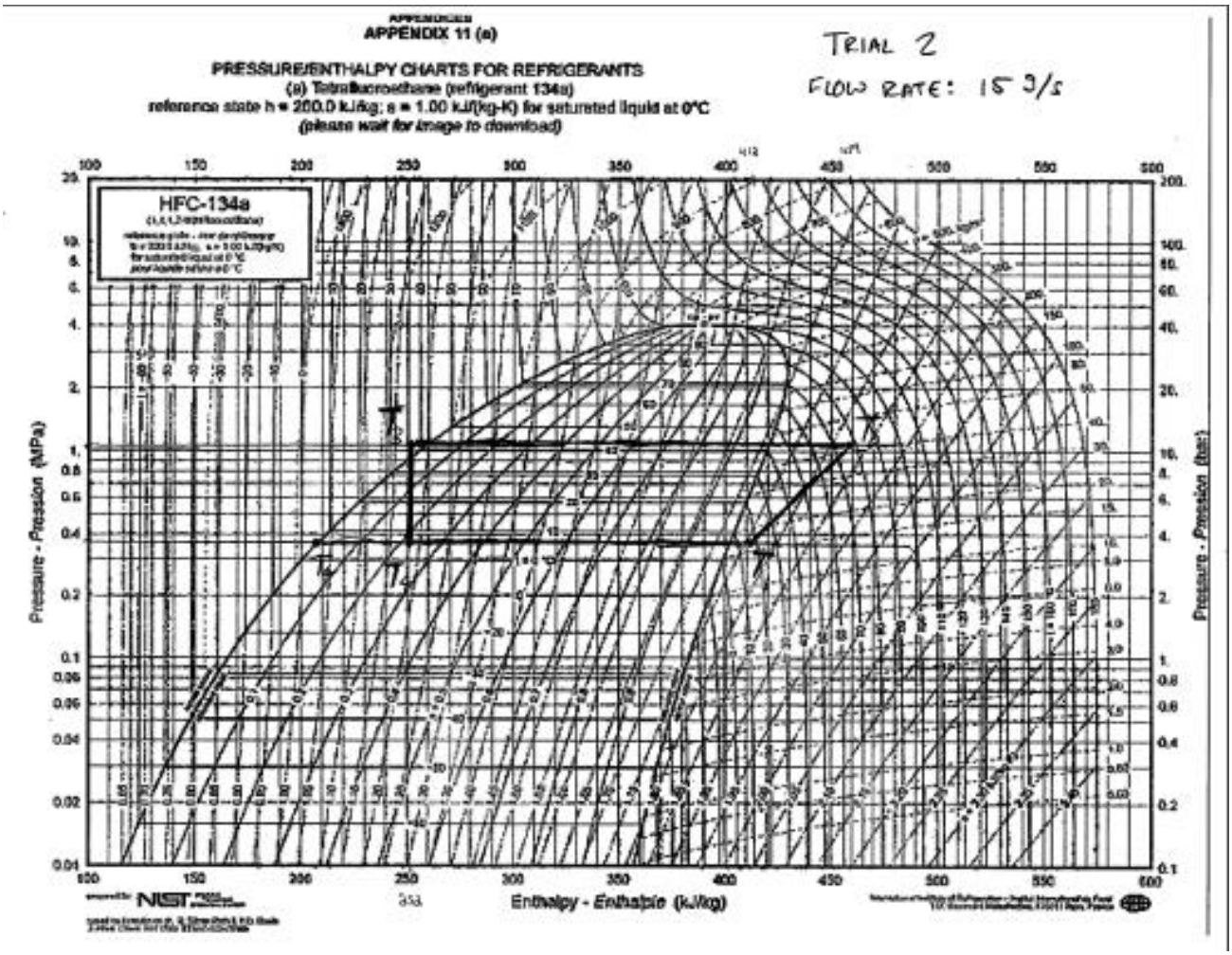


Fig. 12. Trial 2, Pressure-Enthalpy diagram plotted for 15 g/s

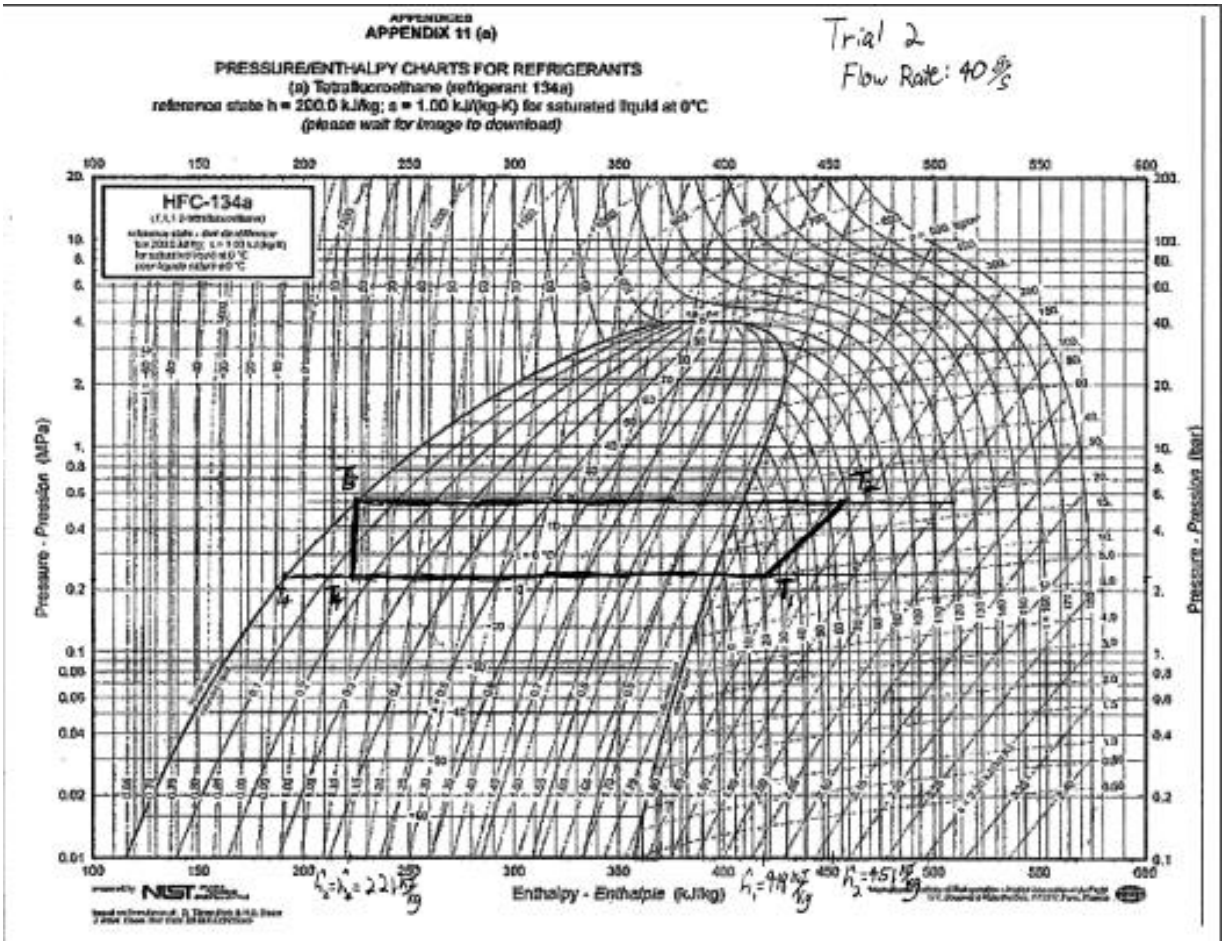


Fig. 13. Trial 2, Pressure-Enthalpy diagram plotted for 40 g/s

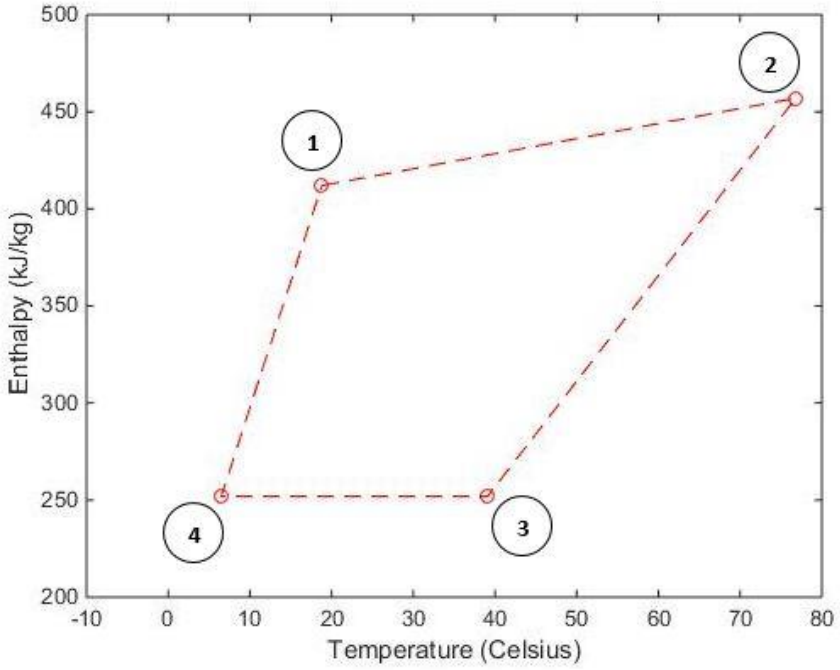


Fig. 14. Trial 1, Temperature-Enthalpy diagram plotted for 15 g/s

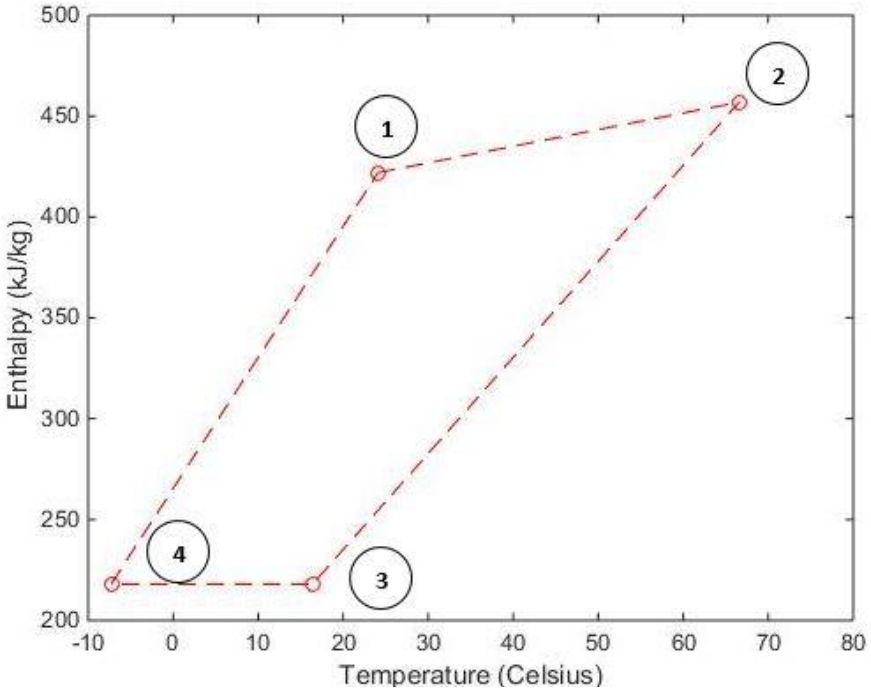


Fig. 15. Trial 1, Pressure-Enthalpy diagram plotted for 40 g/s

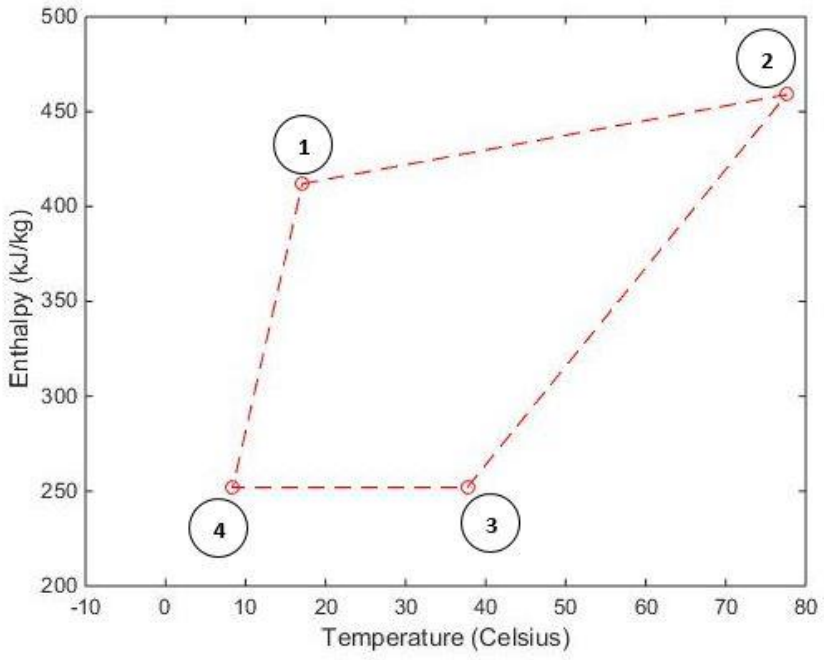


Fig. 16. Trial 2, Pressure-Enthalpy diagram plotted for 15 g/s

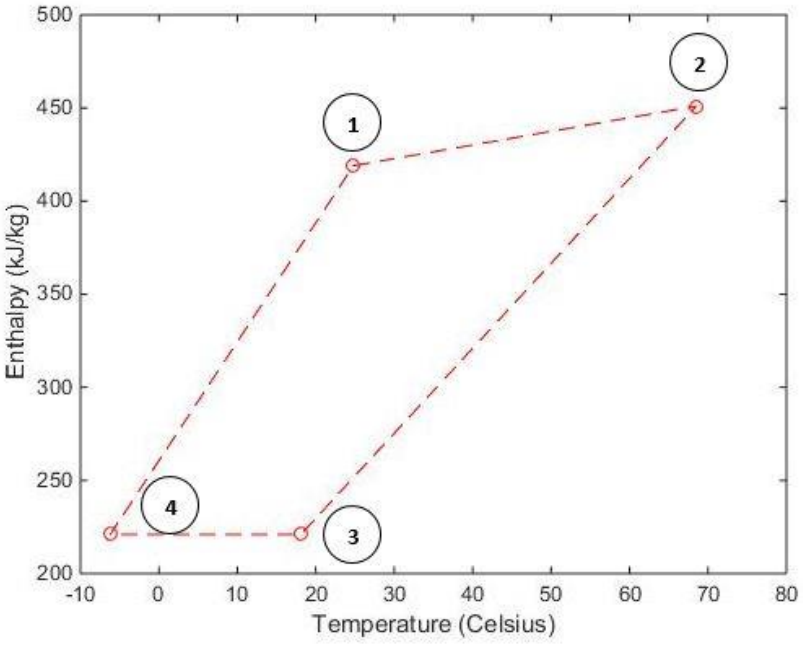


Fig. 17. Trial 2, Pressure-Enthalpy diagram plotted for 40 g/s

The final objective was to calculate the heat loss from the compressor. Equation 2 was used calculate the heat loss. Figure 18 and Figure 19 show the heat losses for Trial 1 and Trial 2 water flow rates.

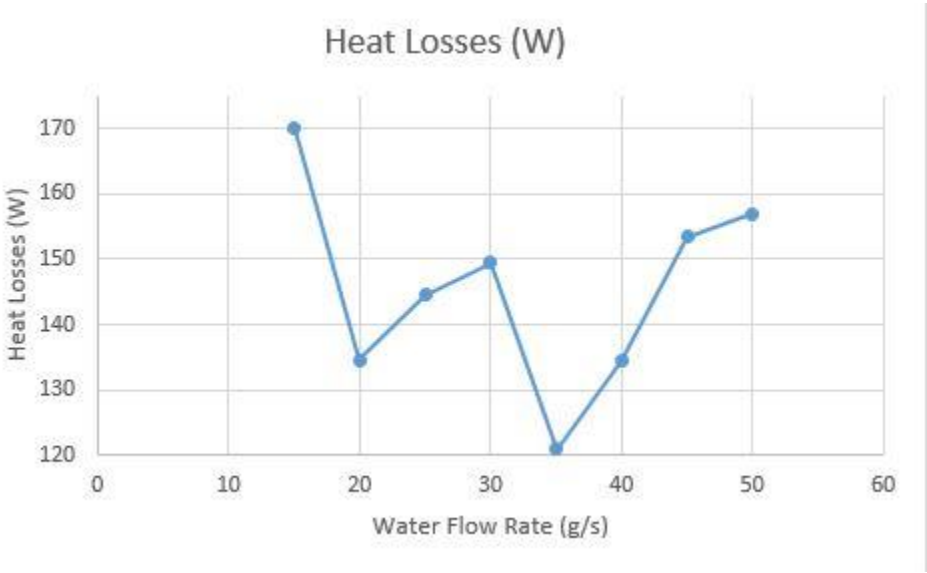


Figure 18. Trial 1, Heat lost to the environment

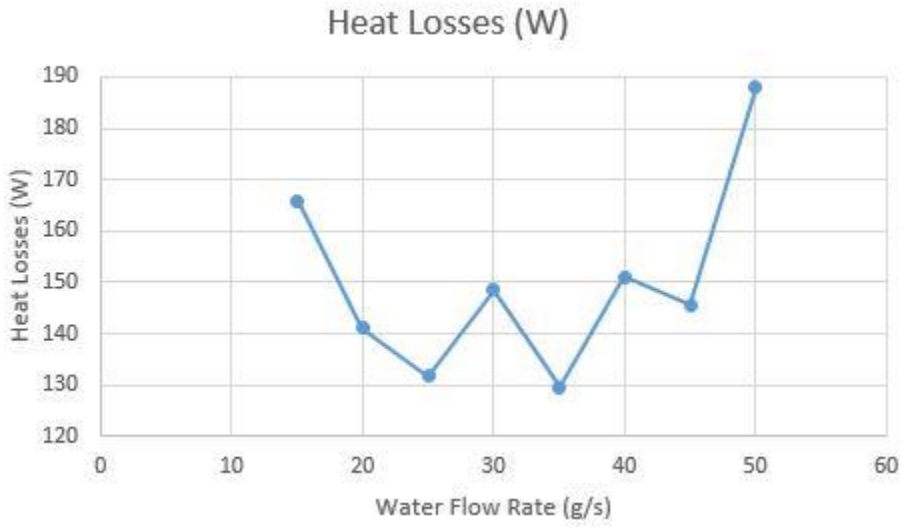


Figure 19. Trial 2, Heat lost to the environment

4.0 Analysis and Discussion

Figure 4 and Figure 5 compare the effects of condenser water flow rate on the temperature of the water exiting the condenser. In both trials, the exiting temperature decreased as the condenser water flow rate increased. This is because as the flow rate increased, more water exchanged heat with r134a. As a result, a larger mass of water gained heat per second, so the temperature could not rise as high as it did with the lower flow rates. The plots in Figure 4 and Figure 5 suggest a non-linear, negative correlation between condenser water flow rate and the exiting water temperature.

Figure 6 and Figure 7 show the effects of condenser water flow rate on energy delivered. In both trials, the generated graphs show a spike in delivered energy at 15 g/s followed by a slight decrease in each other flow rates analyzed. Because temperature is related to energy delivered, it makes sense that there was a spike at 15 g/s, since there was also a temperature spike at that flow rate. As the rest of the flow rates increased, the energy delivered tended to decrease. Since the flow rates are higher, there is more water flowing through the cycle. If there is more water, the energy would need to be delivered to more material, which would decrease the amount of energy delivered from the decreasing temperature difference between the entering and exiting water temperatures.

The final part of the first technical objective was to analyze the effects of the condenser water flow rate on the coefficient of performance (COP). Figure 8 and Figure 9 show a parabolic trend for the COP as water flow rate increases. Each value is between 3 and 4, which makes sense because the heat pump is taking energy from the air to transfer heat, so more total heat will be transferred to the water than the power that is inputted to the compressor. Since the parabola is concave down, we can assume that there is a maximum in the graph which shows when the heat pump is transferring heat most efficiently. For the first trial, this point was at 30 g/s. For the second trial, this point was at 25 g/s. From this data, we can conclude that there exists a maximum-

efficiency flow rate in between 25 g/s and 30 g/s where the mechanical heat pump transfers heat in the most efficient manner.

The second technical objective was to generate temperature and pressure enthalpy diagrams at steady state when the water flow rate was set at 40 g/s and 15 g/s for the r134a vapor-compression cycle. T_1 through T_4 along with the condenser and evaporator pressures were needed to generate these plots using a Pressure-Enthalpy chart for refrigerant r134a.

T_4 and T_1 correspond to temperatures of the refrigerant before and after the evaporator, respectively, while T_2 and T_3 correspond to temperatures of the refrigerant before and after the condenser. The pressures for the condenser and evaporator were measured in kPa and happened to be gauge pressures, which do not include atmospheric pressure. For each pressure, 101.325 kPa was added to account for atmospheric pressure in order to make the plots.

The enthalpy graphs were made for both flow rates by plotting the intersection points of the absolute pressure for the evaporator with T_1 and T_4 and the absolute pressure for the condenser with T_2 and T_3. Points 1 through 4 were connected in ascending order, representing the different stages that the refrigerant r134a traveled through, as represented in Figures 10, 11, 12, and 13. Enthalpies can be read from the figures by drawing lines straight down to the x-axis, where enthalpy values are found. Enthalpies that correspond to T_1 and T_2 were the highest and enthalpies that correspond to T_3 and T_4 were the lowest (and were equal).

After the enthalpies were recorded for each temperature point, Matlab was used to generate Temperature-Enthalpy diagrams. These graphs are represented by Figures 14, 15, 16, and 17. The 40 g/s water flow rate enthalpy graphs for both trials increase with temperature. The enthalpies are highest for T_1 and T_2 because the refrigerant is a vapor at T_1 and a superheated vapor at T_2. When the refrigerant travels through the evaporator, it evaporates and enters a gaseous phase at T_1. When this vapor travels through the

compressor, it compresses and the temperature rises at T_2, creating a superheated vapor with an even higher enthalpy than refrigerant in the vapor phase. At T_3, the refrigerant condenses when it transfers its heat to the water flowing through the heat exchanger. Here, the refrigerant r134a became a subcooled liquid, and its enthalpy decreased. The refrigerant then travels back to the evaporator through an expansion valve where the pressure is lowered. At T_4, the refrigerant is in vapor-liquid equilibrium. Enthalpies generally are understood to be higher in the vapor phases and lower in the solid or liquid phases, which agree with the findings in this experiment.

For the 15 g/s water flow rate, the results were the same except that T_3 was greater than T_1. This indicated that there was not as much heat transfer from the refrigerant to the water when the water was moving at this slower flow. During this flow rate, there was less water for the refrigerant to transfer its heat, so the refrigerant temperature did not decrease as much as other trials when the water flow rate was higher than 15 g/s. This also explains why the coefficient of performance wasn't very high at this flow rate. The water was flowing slow enough to allow efficient heat transfer from the refrigerant to the water so the temperature of the refrigerant did not decrease nearly as much as the other flow rates. At higher flow rates, there was more water for the refrigerant to transfer heat to so the temperature of the refrigerant decreased more than it did in the trials when the water flow rate was slower. This trend shows that as water flow rate increases, temperature T_3 decreases.

The third technical objective was to calculate the heat loss (Q_L) from the compressor in the process. In an ideal system, all of the power input should be converted to compressing the refrigerant. This means that the difference between the enthalpy before and after compression multiplied by the flow rate of the refrigerant should be equal to the power input. Any difference between these values is the result of heat loss, as Equation 2 shows.

Both the enthalpy before and after compression were measured from the Pressure-Enthalpy charts created in Technical Objective 2. These enthalpy values were measured

in units of kJ/kg and multiplied by g/s; therefore, unit conversions were needed so that these values could be subtracted from units of watts for power.

Figures 18 and 19 show the heat losses versus the flow rates. The heat loss to the environment had a parabolic shape, meaning the heat losses were greatest at 15 g/s and 50 g/s. This corresponds to the COP graphs. The COP was the least at both 15 g/s and 50 g/s. The heat loss was greatest at 50 g/s because the water was moving too fast to be heated efficiently. This also explained the COP being so low at this flow rate. The heat loss was also greatest at 15 g/s because it was moving too slowly. Since there was not enough water to match the heat being transferred, the heat was lost to the surroundings. The heat loss should be least when the COP is the greatest. The heat loss for both trials were minimum at 35 g/s. These values differed slightly from the COP maximums which occurred at 30 g/s and 25 g/s. Human error could be the cause in this difference. Because these results rely on Pressure-Enthalpy diagrams, these plots could display slightly wrong results. This could result in a heat loss that differs from what is expected based on the COP. The heat loss came to a minimum at this point because the majority of the heat transferred was successfully used to heat the water. These trends occurred for both trials. This process shows that it is more efficient when the flow rate is between 30 g/s and 35 g/s and least efficient as the flow rates increase or decrease. Since the process is less efficient in heating the water at a lower flow rate, it is expected that this is caused by excess heat lost to the environment.

5.0 Summary and Conclusions

The experiment conducted with the mechanical heat pump showed empirical data that successfully assessed the technical objectives. The first technical objective was to evaluate the effect of condenser water flow rate on the following parameters: exiting water temperature, energy delivered, and coefficient of performance. After evaluating the effect of condenser water flow rate on the temperature of water exiting the mechanical heat pump at steady state, Figures 4 and 5 showed that as condenser water flow rate increased, the temperature of water exiting decreased. For the rate of energy delivered,

the results for trials 1 and 2 both showed a spike in power, as well as temperature, when the condenser water flow rate was 15 g/s. The rest of the data, in Figures 6 and 7 showed a decrease in the rest of the flow rates. The coefficient of performance was calculated using Equation 1. It was found that the COP followed a concave down, parabolic trend as water flow rate increased, suggesting that there exists a maximum in efficiency somewhere between the flow rates investigated.

The second technical objective was to generate Temperature-Enthalpy (T-H) and Pressure-Enthalpy (P-H) diagrams for the r134a vapor-compression cycle operating at steady-state with the condenser water flow rate set to 40 g/s and 15 g/s.

From the Temperature-Enthalpy and Pressure-Enthalpy diagrams, it was concluded that T_1 and T_2 had the highest enthalpies: T_1 being the temperature exiting the evaporator and T_2 being before the condenser. These two temperatures had the highest enthalpies because they were vapor (T_1) and superheated vapor (T_2). T_3 and T_4 were found to be equal and had lower enthalpies. This was because they were in vapor-liquid equilibrium (T_3) or subcooled liquid (T_4).

The third and final technical objective was to determine the steady-state rate of heat loss to the environment from the compressor at each condenser flow rate investigated. As observed in Figure 18 and Figure 19, the heat loss to the environment followed a parabolic, concave up trend as water flow rate increased. This suggests that there is a minimum in heat loss at one of the flow rates investigated. Although this was thought to be at the same point as the maximum COP, this was not the case, which could be due to human error in generating the Pressure-Enthalpy diagrams.

6.0 References

[1] “Mechanical heat pump” *Industrial Heat Pumps,* 2008.

[2] Forsen, Martin, “Heat Pumps: Technology and Environmental Impact,” *European Heat Pump Association,* July 2005.

[3] “Basic Heat Pump Theory,” *Heat-pump-pro.com.*

[4]Salah, Ahmad, “What Is An Absorption Chiller?” *Absorption Chiller*.

Appendix A-1
Examples and Guidelines for Equations

Table 1. Trial 1, raw temperature data

1						
Time (min)	Temp 1 (C) - Evaporator	Temp 2 (C)- After Compression	Temp 3 (C)-Condenser (After Heat Exchanger)	Temp 4 (C)- After expanison into evaporator	Temp 5 (C)-Condenser H2O going in	Temp 6 (C)-Condenser H2O going out
19	24	60.3	15.5	-9	5	9
7.783	24	65	15.8	-7.9	5.1	9.6
4.767	24.1	66.6	16.4	-7.3	5.2	10.7
8	24.2	67.9	17.1	-6.8	5.3	11.7
5.217	24.2	68.5	18	-5.9	5.4	13.2
4.5	24.4	69.2	19	-5	5.6	14.9
7.5	24.1	70.2	20.9	-4.1	6.2	18.1
7.5	18.8	76.8	39.1	6.5	7	27.7

Table 2. Trial 1, other raw data

1				
Cond. Pressure (kPa)	Evap. Pressure (kPa)	Water Flow Rate (g/s)	Refr. Flow Rate (g/s)	Power (W)
400	105	50	3.5	255
405	120	45	3.5	255
415	122	40	3.5	257
435	125	35	3.5	261
445	130	30	3.5	265
460	140	25	3.75	272
500	150	20	3.8	279
910	250	15	5	395

Table 3. Trial 2, raw temperature data

2						
Time (min)	Temp 1 (C) - Evaporator	Temp 2 (C)- After Compression	Temp 3 (C)-Condenser (After Heat Exchanger)	Temp 4 (C)- After expanison into evaporator	Temp 5 (C)-Condenser H2O going in	Temp 6 (C)-Condenser H2O going out
26.6	24.3	64.3	16.8	-7.3	7.2	11.5
8.0013	24.6	67.1	17.5	-6.7	7.2	12.1
9.75	24.8	68.5	18.1	-6.2	7.3	12.9
6	24.7	69.2	18.7	-5.7	7.4	14.1
5	24.5	69.7	19.4	-5.2	7.6	15.4
6	24.5	70.2	20.7	-4.2	7.7	17.7
7	24.6	71.2	22.9	-3	8	20.2
5	17.1	77.6	37.7	8.4	8.6	29.3

Table 4. Trial 2, other raw data

2				
Cond. Pressure (kPa)	Evap. Pressure (kPa)	Water Flow Rate (g/s)	Refr. Flow Rate (g/s)	Power (W)
420	120	50	3.5	258
435	125	45	3.5	261
445	125	40	3.5	263
450	130	35	3.5	266
460	140	30	3.5	271
496	148	25	3.75	278
540	155	20	4	289
975	275	15	5.3	415

```
clear
clc

%The temperatures below were listed in ascending order, in Celsius
temps = [T1,T2,T3,T4,T1];

%The enthalpies were listed corresponding to each temperature value in the
%pump
enthalpies = [h1,h2,h3,h4,h1];

%These lines of code plot the Enthalpy against Temperature
plot(temps,enthalpies,'r--o'),xlabel('Temperature (Celsius)'),ylabel('Enthalpy (kJ/kg)');
axis([-10 80 200 500]);

hold on;
```

Fig. 20. Matlab code used to generate Figures 14, 15, 16, and 17

Appendix A-2
Examples and Guidelines for Figures and Tables

The following calculations were made using data from the first trial with the condenser flow rate at 40 g/s.

The first calculation that had to be done was for the coefficient of performance (COP) for the heat pump was

$$COP = \frac{\left(40\frac{g}{s}\right) * \left(4.186\frac{J}{g * ℃}\right) * |10.7℃ - 5.2℃|}{257\,J/s} = 3.5833. \quad (3)$$

After completing the first technical objective, the next step was to generate a temperature-enthalpy graph using data obtained from the given pressure-enthalpy graph. The enthalpies found were

$$\widehat{h_1} = 422\frac{kJ}{kg}, \quad (4)$$

$$\widehat{h_2} = 457\frac{kJ}{kg}, \quad (5)$$

$$\widehat{h_3} = \widehat{h_4} = 252\frac{kJ}{kg}. \quad (6)$$

Matlab was used to graph these enthalpies against their corresponding temperatures (T_1, T_2, T_3, and T_4). The code that generated Figure 14 is given in Figure 19.

```
clear
clc

%The temperatures below were listed in ascending order, in Celsius
temps = [24.1,66.6,16.4,-7.3,24.1];

%The enthalpies were listed corresponding to each temperature value in the
%pump
enthalpies = [422,457,218,218,422];

%These lines of code plot the Enthalpy against Temperature
plot(temps,enthalpies,'r--o'),xlabel('Temperature (Celsius)'),ylabel('Enthalpy (kJ/kg)');
axis([-10 80 200 500]);

hold on;
```

Fig. 21. Matlab code used to generate a temperature-enthalpy graph for Trial 1 at 40 g/s

The third and final technical objective was to determine the steady-state rate of heat loss to the environment from the compressor at each investigated flow rate (40 g/s and 15 g/s). The calculation done for this was

$$\dot{Q}_L = \left(257\frac{J}{s}\right) - \left(3.5\frac{g}{s}\right) * \left(457\frac{J}{g} - 422\frac{J}{g}\right) = 134.5\ W. \qquad (7)$$