University of California Riverside

ME170B Experimental Techniques: Lab 7

Refrigeration

Group A5

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Abstract

The purpose of this experiment is to analyze the efficiency of an R134A vapor compression refrigeration cycle, understand the working principles, and compare the actual cycle to the ideal vapor compression cycle. It is hypothesized that the actual refrigeration cycle will produce a lower compressor exit temperature and involve pressure drops during phase changes when compared to an isentropic ideal cycle with the same pressure ratio and refrigeration capacity. This experiment utilizes the CRD400 Refrigeration Cycle Demonstration Unit which takes temperature and pressure measurements for inlets and outlets of the four major components in the refrigeration cycle (evaporator, compressor, condenser, expansion valve). After allowing the system to reach equilibrium over twenty minutes, temperature and pressure measurements were recorded and combined with thermodynamic tables to calculate enthalpies, entropies, and ultimately, the coefficient of performance (COP), found to be 2.5 ± 0.3. A T-s diagram was constructed for the cycle and compared to the ideal refrigeration cycle's T-s diagram. Comparing the two T-s diagrams reveals that the actual cycle reaches a higher temperature and entropy state after compression, as well as changes in pressure through the condenser and evaporator. The deviations from the ideal conditions of isentropic compression, constant pressure heat exchange, and adiabatic expansion collectively impacts the system's efficiency, decreasing the coefficient of performance.

Introduction

One of the most critical design choices in refrigeration systems is the choice of the refrigerant, as this directly impacts the refrigeration capacity. Ideal refrigerants typically have low boiling points and low specific heats. Two common choices of refrigerants include R12 and R134a, with R134a more commonly used today due to environmental concerns.

R12, also known as Dichlorodifluoromethane, has the chemical formula CCl2F2, falling in the category of chloro-fluoro-carbons (CFCs). R12 used to be a common refrigerant used in cooling systems until the discovery of the depleting ozone layer as a result of R12 pollution. Although non-toxic, R12 has an ozone-depleting potential (ODP) of 1 and a global warming potential (GWP) of 2400. Ozone depletion occurs as R12 rises in the atmosphere and breaks apart due to solar radiation, releasing chlorine that destroys ozone (O3) molecules. The ozone layer is responsible for blocking ultraviolet radiation from the sun, which is damaging to most life and carcinogenic to humans. R12 is no longer manufactured or distributed in the United States but is still used in old systems that can only use R12 as refrigerant. This typically includes classic cars, or old refrigeration systems manufactured prior to the Montreal Protocol (1989), the international agreement to phase out ozone-depleting refrigerants, namely R12.

R134a, also known as 1,1,1,2-Tetrafluoroethane has the chemical formula CF3CH2F and was originally developed to replace R12 in refrigeration applications due to its ODP. As of 2023 R134a is also in the process of being phased out due to its high GWP. R134a and R12 have similar thermodynamic properties, so replacing R12 with R134a in refrigeration systems does not result in appreciable reductions in performance for the vapor-compression refrigeration cycle. Table 1 summarizes some properties of both refrigerants.

Property	R134A	R12
Chemical formula	CH_2FCF_3	CCl_2F_2
Boiling point	-26.11 ºC	-29.8 ºC
Critical Temperature	101.2 ºC	112 ºC
Saturation Vapor Pressure (45ºC)	11.6 bar	10.08 bar
Toxicity	Non-Toxic	Non-Toxic
Ozone Depleting Potential (ODP)	0	1
Global Warming Potential (GWP)	1300	2400

Table 1: Property comparison between R134a and R12 refrigeration

The CRD 400 Refrigeration Cycle Demonstration unit in this experiment uses R134a as the refrigerant and resembles commercial refrigeration equipment. The primary object of the lab is to understand the operational principles of the vapor compression refrigeration cycle, calculate the system's coefficient of performance (COP), and plot the actual refrigeration cycle on a T-s diagram. The CRD 400 measures temperature at seven points and pressure at four points of the cycle as well as the power consumption of the compressor. Using these measurements, the four stages of the cycle: evaporation, compression, condensation, and expansion are examined allowing for data analysis and the creation of a T-s diagram, resulting in a comprehensive understanding of the cycle's functionality.

Theory:

The refrigeration cycle begins at the evaporator, where heat is transferred to the cold refrigerant from the cold water stream. The refrigerant enters the evaporator as a saturated liquid-vapor mixture and by absorbing heat exits as a saturated vapor. The heat absorbed by the refrigerant is equal to the heat rejected by the cold water stream, achieving the desired refrigeration effect. The amount of heat transferred can be manipulated by the flow rate of refrigerant, with a higher flow rate resulting in more heat absorbed.

The heat transfer of refrigerant is given by Equation 1:

$$\dot{Q} = \dot{m} c_p (T_1 - T_4) \tag{1}$$

The refrigerant then enters the compressor as a saturated vapor and is compressed into a superheated state at high pressure. The ideal exit temperature of the compressor is given by the adiabatic relationship between temperature and pressure ratio, shown in Equation 2, where gamma is the adiabatic index for the refrigerant. The actual temperature can be measured or calculated based on the isentropic efficiency of the compressor. The compressor power is given by Equation 3.

$$\frac{T_2}{T_1} = (\frac{P_2}{P_1})^{\gamma} \tag{2}$$

$$\dot{W} = \dot{m} c_p (T_2 - T_1) \tag{3}$$

After compression, the refrigerant enters the condenser as superheated vapor, where it is cooled at a constant pressure by rejecting heat. The vapor condenses into a saturated liquid at high pressure. The heat rejected by the refrigerant is equal to the heat absorbed by the water stream and is given by Equation 4.

$$\dot{Q} = \dot{m}c_p(T_2 - T_3) \tag{4}$$

Finally, the saturated liquid refrigerant enters the expansion valve, where it is expanded isentropically and adiabatically. The expansion process occurs quickly, allowing no heat transfer, thus making it adiabatic but also irreversible, as the specific entropy increases greatly. The refrigerant cools as it expands, returning it to a low-temperature, low-pressure liquid-vapor state before being supplied to the evaporator. This step is also known as throttling, and the isenthalpic relationship is given by Equation 5.

$$h_4 = h_1 \tag{5}$$

In vapor-compression refrigeration systems, the coefficient of performance is defined as the ratio of refrigeration capacity to compressor power requirement. A typical refrigeration cycle will have a COP greater than 1, with higher values indicating greater efficiency and lower operating costs. The refrigerator's coefficient of performance can be given by Equation 6:

$$COP = \frac{\dot{m}c_p(T_1 - T_4)}{\dot{m}c_p(T_2 - T_1)} \tag{6}$$

In an actual vapor-compression system, the heat transfer between the warm and cold regions is not accomplished reversibly. In the actual refrigeration system, the refrigerant temperature at the evaporator and at the condenser are lower and higher than the cold and warm region temperatures respectively, as shown in Figures 1 and 2.

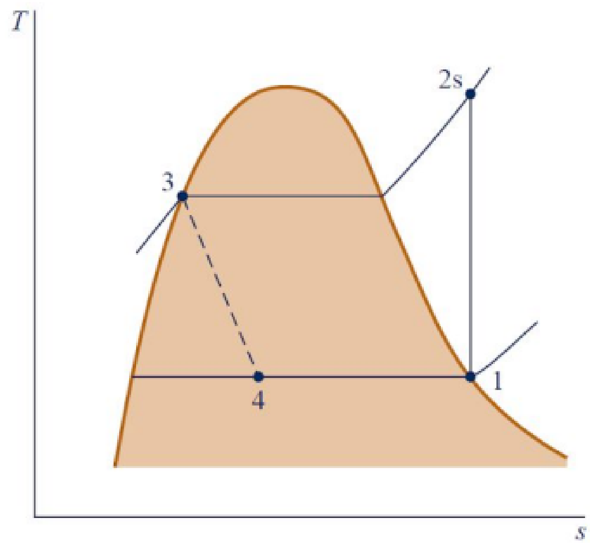


Figure 1: T-s diagram for an ideal vapor-compression cycle (refrigerator)

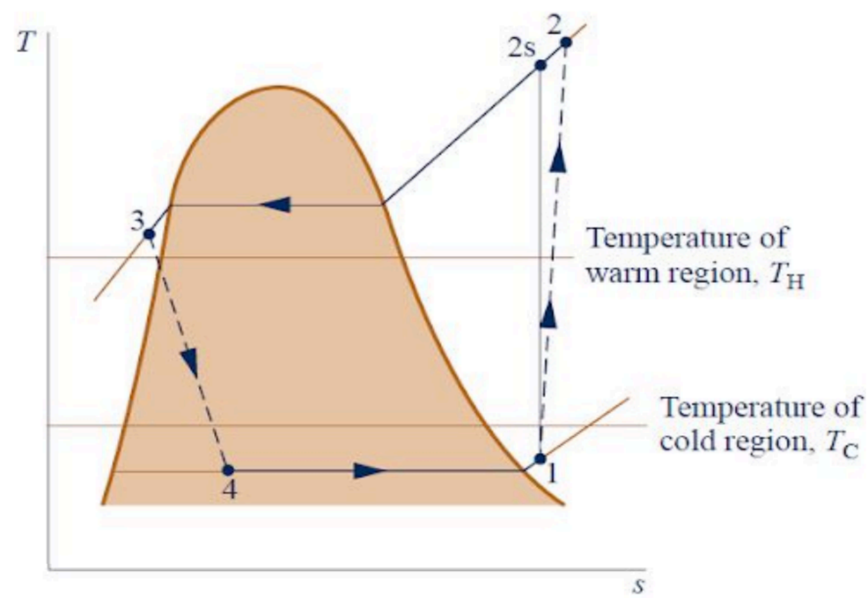


Figure 2: T-s diagram for an actual vapor-compression cycle (refrigerator)

<u>Methods</u>

Precaution: Exercise caution around the tubes connected to the compressor, as they become extremely hot during operation. Be sure to run water throughout the test and afterward to help with cooling. Avoid adjusting the display panels to prevent any malfunction of the equipment. Calibration was not required for this experiment, as the apparatus was pre-calibrated. Ensure that the water hose is securely in the drain, such that wastewater does not spill onto the floor.

Procedure: Begin by turning on the Refrigeration Cycle Demonstration unit CRD 400 (Figure 3) and opening the water valve. Adjust the flow rate using the left and right flow meters to the desired levels. Allow the system to warm up for approximately 20 minutes until it stabilizes. Once stable, take temperature measurements at critical points in the cycle: condenser input (T3), condenser output (T4), evaporator input (T6), and evaporator output (T7). To record these temperatures, connect the red and black temperature probes to their respective sockets, ensuring accurate measurements at each stage of the refrigeration cycle. Next, gather pressure readings at several points: compressor inlet and outlet, tank outlet, and regulator outlet. Once all necessary measurements have been recorded, turn off the system, but leave the water valves open. Allow the system to cool for at least 20 minutes. Following the cooling period, restart the system, allow it to warm up again for 20 minutes, and repeat the same procedure as many times as required. At the conclusion of the experiment, turn off both the refrigeration cycle unit and the water supply valve to complete the process.

Figure 3: Refrigeration Cycle Demonstration unit CRD 400

<u>Results</u>

To plot the T-s diagram for the actual refrigeration cycle, data from Tables 2 and 3 were used to calculate entropy values, with specific entropy values derived from the thermodynamic tables for saturated and superheated refrigerant R134a as provided in the textbook. When exact values were not available, linear interpolation was applied to obtain precise entropy readings. The resulting T-s diagram is presented in Figure 4, which, as observed, closely resembles the T-s diagram of an ideal vapor-compression cycle (Figure 2).

For the actual refrigeration cycle data (as shown in Table 2), there is an 18% +/- 0.2% reduction in enthalpy [calculated as Cp(T5-T6)] across the expansion valve (regulator), deviating from the isenthalpic behavior expected in an ideal refrigeration cycle. The measured entropy increase across the compressor amounted to 2.7% +/- 0.2%, with respect to the ideal adiabatic process (Δs = 0). Despite allowing time for the experiment to reach a steady state, the measured data continued to oscillate significantly, introducing uncertainty into our data collection..

	Temp (˚C)	Pressures (Bar)	Entropy
T0: Water: Exchanger Input	22.8˚C	N/A	N/A
T1: Water: Condenser Output	33.2˚C	N/A	N/A
T2: Water: Evaporator Output	13.2˚C	N/A	N/A
T3: R134a: Condenser Input	49˚C	10 bar	0.939 kJ/kg*K
T4: R134a: Condenser Output	37.5˚C	10 bar	0.3772 kJ/kg*K
T5: R134a: Regulator Input	28.5˚C	2.4bar	0.33 kJ/ kg*k
T6: R134a: Evaporator Input	3.3˚C	-	-
T7: R134a: Evaporator Output	8.5˚C	2.25 bar	0.914775 kJ/kg*K

Table 2: Gathered Experimental data and calculated entropies

The observed coefficient of performance (COP) for the Refrigeration Cycle Demonstration unit CRD 400 was calculated to be 3.4 ± 0.3, aligning with expectations for a vapor-compression cycle, where the COP is typically greater than one. In practical applications, an actual refrigeration cycle presents certain advantages and disadvantages. The main advantage is that the average temperature of the refrigerant at the evaporator is lower than in an ideal cycle, leading to an enhanced cooling effect. However, a key drawback is that the refrigerant's average temperature at the condenser is higher than the ambient temperature, requiring more work for compression to reach the necessary high temperature and pressure. This increased compression work reduces the system's coefficient of performance, leading to higher energy consumption and operating costs compared to an idealized refrigeration cycle.

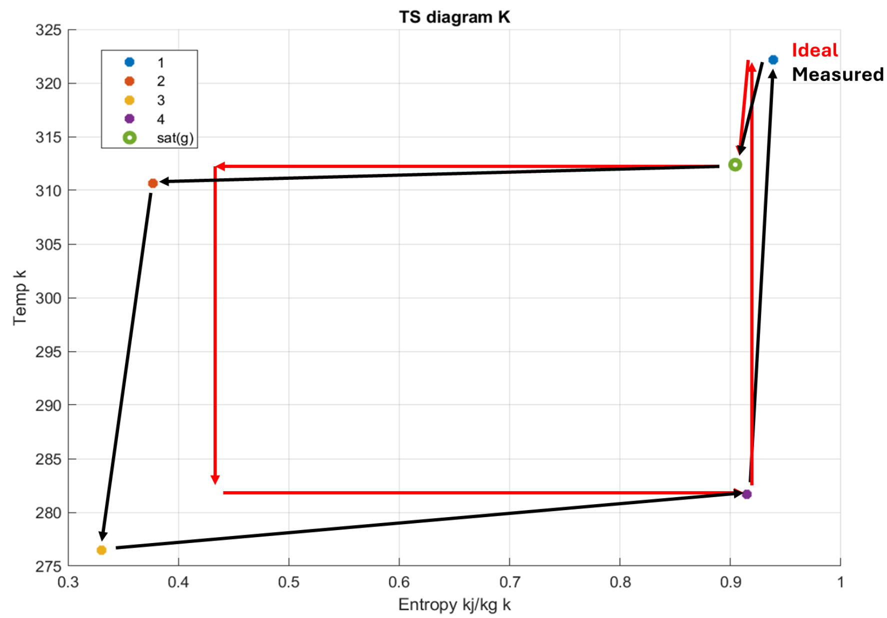


Figure 4: Actual T-s diagram for Trial 1 data

Figure 4 illustrates the deviations between the ideal and actual cycle. One major difference can be seen in the compression line. In the ideal cycle, the compression line is vertical, as ideal compression is isentropic. However, in the real cycle, we can see an increase in entropy during compression as the line drifts to the right. Another key observation from the graph is that the lowest temperature of the real cycle is lower than that of the ideal cycle. This is due to the various irreversibilities that cause a loss of internal energy in the refrigerant as it travels through the pipes of the system, and is reflected in the drop in enthalpy.

Discussion

The results indicate the expected losses in the system, primarily the 18% +/- 0.2% drop in enthalpy and the 3.6%+/- 0.2% increase in entropy. This confirms the previously stated hypothesis that the actual refrigeration cycle will produce a higher compressor exit temperature and involve pressure drops during phase changes when compared to an isentropic ideal cycle with the same pressure ratio and refrigeration capacity.

The results confirm that our experimental refrigeration cycle deviates from the properties of an ideal refrigeration cycle, as predicted by the hypothesis. The deviation from ideal conditions includes an 18%+/- 0.2% loss in enthalpy across the expansion chamber. The 18%+/- 0.2% loss in enthalpy across the expansion chamber likely implicates losses due to the partial vaporization of the refrigerant, causing the gas to act as a mixture, requiring the need to interpolate the values of enthalpy and entropy. The same was observed after the compressor but led to a smaller deviation of 3.6%+/- 0.2% compared to an isentropic cycle with no change in entropy. Additionally, the results imply many sources of energy loss throughout the fluid flow in the pipes. It is possible the elevation changes in the refrigeration system also played a role in the energy losses as the system contained several vertical pipe fixtures.

While this experiment provided a solid understanding of the vapor compression cycle, it is important to recognize the limitations within the analysis. One primary concern is the stray heat transfer between the refrigerant and the environment as it passes through copper tubes, where making the system more compact could potentially reduce this heat loss and associated pressure head losses, thereby preserving more of the refrigerant's internal energy. Additionally, factors like the smoothness of the tubes affected refrigerant flow, and these effects were beyond the scope of this class to analyze fully. The interpolation of values from thermodynamic tables, which cannot provide continuous data, also introduced some uncertainty, as we measured pressures and temperatures only at specific points in the cycle.

These limitations in data collection, combined with oscillations in temperature and pressure readings even at steady state, contributed to an overall error margin of ±0.2%. This variability impacted the precision of calculated parameters, such as enthalpy and entropy, and required that the coefficient of performance (COP) account for this margin. The resulting ±0.2% error margin underscores the sensitivity of the system to minor data inconsistencies, highlighting the challenges of achieving exact measurements and the inherent uncertainties present in experimental analysis.

Other limitations to this version of the experiment are associated with the data collection. Altering the pressure ratio of the compressor may have provided great insights into the effect of compression on the performance of the refrigeration cycle. Additionally altering the fluid flow rates of both the water and refrigerant would provide interesting insights into the refrigeration capacity. Another interesting variation would be to alter the refrigerant substance, to observe how differences in thermodynamic properties may influence refrigeration capacity and coefficient of performance. Unfortunately conducting multiple trials at steady state with these variations would exceed the time resources for the purpose of this laboratory course. Despite these limitations, conducting this experiment with one set of variables still provides profound insights into the concept of the vapor compression refrigeration process.

Conclusion

In conclusion, this experiment found that the primary deviations for an actual vapor compression cycle compared to an ideal cycle were the increase in entropy and energy losses through the pipes, as predicted. Additionally, deviations included changes in pressure during heat exchange in the evaporator and condenser, along with incomplete phase changes in the evaporator and the condenser. Despite this, the actual cycle very closely resembles the ideal cycle when plotted on the T-s diagram, implicating the validity of the theory.

While the results of the T-s diagram were quite similar to an actual vapor-compression refrigeration system, without the saturation curve for the Refrigerant 134a, it is difficult to follow a constant pressure line as would be seen in T-s diagrams. A slight increase in temperature at the evaporator inlet and outlet shows that there was some heat being transferred within the copper tubes and other components, therefore, better insulation would result in no/less temperature change between components. Assumptions were made for State 4 at the evaporator, we assumed to have 50-50 liquid-vapor Refrigerant 134a. Therefore, a better method of determining the liquid-vapor ratio would give more accurate results. The average and standard deviation of the coefficient of performance for the Refrigeration Cycle Demonstration unit CRD 400 was calculated to be 3.42 ± 0.30, which agrees with the theory that COP values for refrigeration systems are greater than 1; however, without a theoretical COP value of the CRD 400, we are not able to compare how accurate our results are. Moreover, we also learned how different refrigerants would affect the system by comparing R134A to R12, and why R134A was chosen for this experiment.

To improve the effectiveness of the experiment, it is recommended to conduct multiple trials with different parameters. This could include pressure ratio, mass flow rate of refrigerant, mass flow rate of water as well as the type of refrigerant. Altering the pressure ratio would allow for valuable insights into how the power of the compression affects the coefficient of performance as well as the heat rejected. Increasing the mass flow rates of refrigerant and water would allow for a higher rate of heat exchange between the two substances, effectively increasing the refrigeration capacity. It would be interesting to see the effects of this on the deviation of the system from ideal conditions. Additionally, swapping the refrigerant R134a for another chemical such as propane or R12 would allow for insights into the refrigerant properties and how they affect the performance of the refrigeration cycle.

References:

[1] M. J. Moran, Fundamentals of Engineering Thermodynamics, 7th Edition, John Wiley & Sons, New York, 2011.

[2] Khemani, Haresh. “Refrigerant 134a Properties & Replacement of R12”. 2010 https://www.brighthubengineering.com/hvac/63578-refrigerant-r134a-properties-and-replacement-for-r12/

[3] Engineering Toolbox https://www.engineeringtoolbox.com/refrigerants-d_902.html

Statement of Contribution:

Elijah Perez: Experimental design, data collection, data analysis, lab report

Soham Saha: Experimental design, data collection, lab report

Alex Pham: Experimental design, data collection, lab report

Appendix:

T = Temp of fluid
P = Pressure
$\dot{Q}$ = Heat rate
$\dot{W}$ = Power
C_p = Heat Capacity
$\dot{m}$ = mass flow rate
h = Enthalpy
s = Entropy

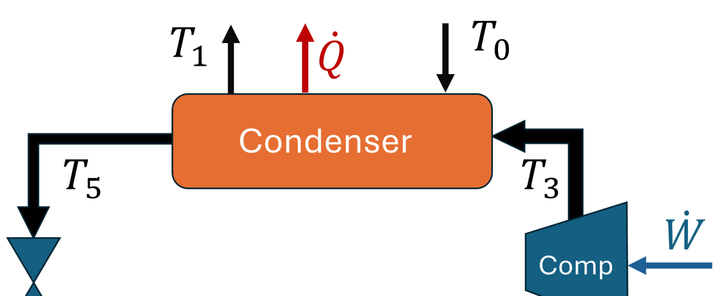
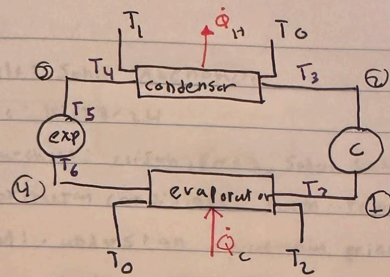


Figure 5: Skimated that corresponds to CRD 400 Refrigeration Cycle Demonstration unit



- $E_F = \dfrac{\text{evap (Power)}}{\text{comp (power)}} = \dfrac{\cancel{\dot m\, cp\,(T_0 - T_1)}}{\cancel{\dot m\, cp\,(T_2 - T_0)}} = \dfrac{cp \cdot \dot m\,(T_0 - T_2)}{cp \cdot \dot m\,(T_3 - T_2)}$ $(T_4 - T_3)$
$= cp\,\dot m\,(h_0 - h_4)$

- $cop = \dfrac{\text{cond (Power)}}{\text{comp (power)}} = \dfrac{\dot m\, cp\,(T_1 - T_0)}{\dot m\, cp\,(T_3 - T_2)}$

- $\text{comp (power)} = V \cdot I$

- $\text{global } \eta = \dfrac{\dot m\, cp\,(T_3 - T_2)}{V \cdot I}$

Ideal cycle

$$cop = \frac{Q_c}{W} = \frac{Q_c}{Q_H - Q_c} = \cancel{\frac{E_2 - E_1}{t}}$$

$$= \frac{\dot m\, c_p\,(T_2 - T_0)}{[\dot m\, cp\,(T_0 - T_1)] - [\dot m\, cp\,(T_2 - T$$

$$= \frac{T_2 - T_0}{(T_0 - T_1) + (-T_2 + T_0)}$$

$$\boxed{cop = \frac{T_2 - T_0}{2T_0 - T_1 - T_2}}$$