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3B1 Thermodynamics

MEU33B03 – Prof. Anthony Robinson

Performance Test on a Vapour-Compression Refrigeration System

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Abstract

The following lab report details a performance test of a vapor-compression refrigeration system. After running the cycle at four different compressor speeds, we were able to deduce the effects that different speeds have on the system's performance. We investigated the first law of thermodynamics and how it can be applied to refrigeration systems. We discussed the Carnot, ideal, and actual vapour-compression refrigeration cycle and their differences. We went through many stages of calculation to reach two key data points; the system's coefficient of performance (COP) and the compressor efficiency. We analysed this data and came to our conclusion, while also factoring in possible sources of error that were generated while conducting the experiment.

Introduction

The scientific basis of this lab follows the theory of vapour-compression cycles. By using a refrigeration apparatus, we were able to successfully execute these cycles and extract necessary data to calculate various quantitative measurements. Vapour-compression cycles are the most common cycle used in refrigeration, air conditioning, and heat pump systems. The process works based on the phase change of a working fluid, as it absorbs and releases heat while passing through the system's four main components. For this lab, we studied the refrigeration cycle. The working fluid of the system is R134a, which is the world's most used refrigerant.

In this lab, we will run the experiment four times each at a different compressor speed. We will investigate the different effects that this has on the system's overall performance, efficiency, and analyse its energy conservation properties through the first law of thermodynamics. The main objective is to conclude the coefficient of performance (COP) and compressor efficiencies for each different speed and compare them. These quantities rely a lot on the energy supplied to the refrigerant, energy supplied to the compressor, and thermal energy within the system's water. Our goal will be to find each of these values and compare them for each compressor speed.

Theory and Data Reduction

Thermodynamic Cycle

It is a known fact that in thermodynamics that heat will flow from high-temperature regions to cold-temperature regions. This heat transfer process is a natural phenomenon and requires no devices to make it possible. However, if we were to complete this cycle in the opposite direction, we would require a special device called a refrigerator.

To introduce the refrigeration cycle, we must discuss the scientific basis that led to its development. As we have covered previously in the 3B1 Thermodynamics module, the Carnot Cycle is a totally reversible cycle consisting of two reversible isothermal and two isentropic processes. It operates at the maximum efficiency possible for an engine cycle and serves as the standard for which actual engine cycles can be compared. As all 4 processes of the cycle are reversible, the entire cycle can also be reversed. This reversed process means that all the heat and work interactions within the cycle are also reversed. A refrigerator that works using the reversed Carnot cycle is known as a 'Carnot refrigerator'.

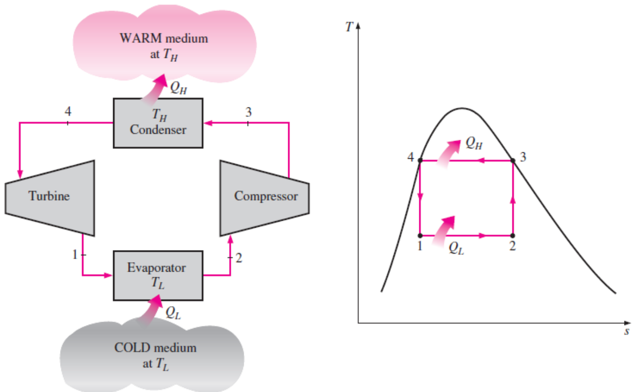


Figure 1: Schematic of the Carnot Refrigerator and a T-s diagram of the reversed Carnot Cycle

The coefficient of performance for a Carnot refrigeration cycle can be expressed as:

$$COP_{R,Carnot} = \frac{1}{\left(\frac{T_H}{T_L}\right) - 1}$$

Ultimately, to increase the COP of the system, we want to minimise the difference between the temperature of the cold sink (T_C) and the temperature of the hot sink (T_H). While this may seem fairly logical, in an actual scenario, where let's say that T_H is at room temperature and $T_C = 3°$, like in a fridge, keeping the difference in temperatures minimised is not always practical. A lot of the time, the Carnot efficiency therefore depends on the environment temperature. This is not practical, so the COP of a refrigerator does not always reflect a fair view of its actual performance.

Despite being the ideal way a refrigeration system should operate, there are many physical limitations that prevent this from being realistic. Processes 4-1 and 2-3 in the above figure can only be applied theoretically. This is because process 2-3 would involve compressing a liquid-vapour mixture, meaning the compressor would have to handle 2 phases, and process 4-1 requires the turbine to expand a refrigerant with a high moisture content.

This is why the Carnot Refrigerator was modified and modelled as the 'Ideal Vapour-Compression Refrigeration Cycle'. The 4-stage process used some different devices to achieve a cycle that is more practical than the Carnot cycle. As the fluid passes through the cycle, it circulates through a compressor, condenser, expansion valve, and evaporator. This addition of new devices such as the expansion valve, means that the cycle is no longer reversible.

Stage 1-2 is where the **isentropic compression** occurs. The compressor takes the refrigerant gas and increases its temperature and pressure. This is typically done via a work input in the form of an electric motor or engine, denoted as W_{in}. The low-pressure vapour is compressed into a high pressure, high temperature gas. The gas is then passed through to the condenser. This is **stage 2-3**, where the **isobaric heat rejection** occurs. The high-pressure gas releases the energy to its surroundings. This causes the refrigerant to condense into a high-pressure liquid. This is where the heat rejection phase occurs and is denoted by Q_{out}. **Stage 3-4** is the **isenthalpic expansion**. The high-pressure fluid then passes to the expansion valve where it expands rapidly, thus reducing its pressure. This causes the refrigerant to take the form of a low-pressure, low-temperature mixture of liquid and vapour. This expansion does not exchange any heat, therefore making it isenthalpic.

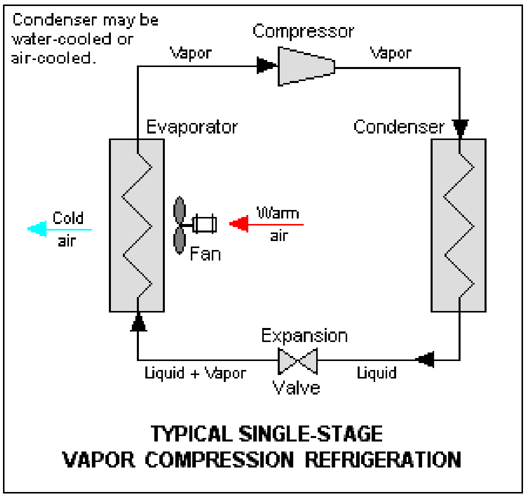


Figure 2: Schematic of a single-stage 4-step vapour-compression refrigeration cycle

The low-pressure refrigerant then passes to the evaporator where it absorbs the heat from its surroundings. This is **stage 4-1**, where **isobaric heat absorption** occurs. This heat absorption causes the refrigerant to change state completely into a low-pressure vapour. This is where the cooling occurs. The system continuously executes this 4-step process which allows for sustained cooling of a particular environment.

The coefficient of performance for an actual refrigeration cycle can be expressed as:

$$COP_R = \frac{q_L}{w_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1}$$

The below figures show a typical 4-step single-stage vapour-compression refrigeration cycle. Figure 1 shows an ideal P-h diagram of the refrigeration cycle. Note the isenthalpic expansion from points 2-3, and the isobaric condensation and evaporation from 1-2 and 3-4 respectively. Figure 3 shows an ideal T-s diagram of the refrigeration cycle. Note the isentropic compression from 4-1 and isothermal evaporation from 3-4.

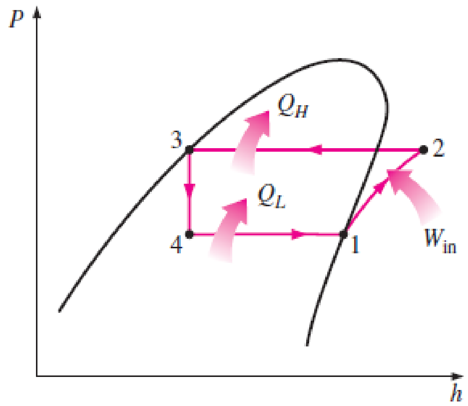


Figure 3: P-h diagram of an ideal refrigeration cycle

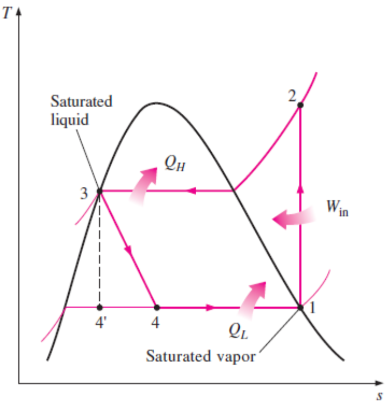


Figure 4: T-s diagram of an ideal refrigeration cycle

Below the differences between the Carnot refrigeration cycle and the ideal/standard one in a tabulated form:

Stage:	**Carnot Cycle**	**Ideal Cycle:**
1-2 & 3-4: Heat Rejection/Absorption	Isothermal	Isobaric (evaporator/condenser)
2-3 & 4-1: Compression/Expansion	Isentropic (turbine/compressor)	Isenthalpic (valve/compressor)
Reversibility:	Fully Reversible	Irreversible
Devices Needed:	Complex expansion device	Simple throttle/valve
Efficiency:	Maximised (unattainable)	Lower (physical constraints)
Practicality:	Not feasible	Standard in real systems

While the ideal cycle can be theoretically implemented in a real-world system, it too has its own limitations. In a real-world environment, there are many factors that can influence the irreversibility of the cycle such as real fluid behaviour fluid behaviour (friction which can cause pressure drops) and heat losses. There also may be non-ideal compression and expansion processes which will lead to reduced efficiency. This means that the ideal cycle has some added stages to account for these losses/imperfections [1]. This can be seen in the T-s diagram for an actual vapour-compression refrigeration cycle to the right:

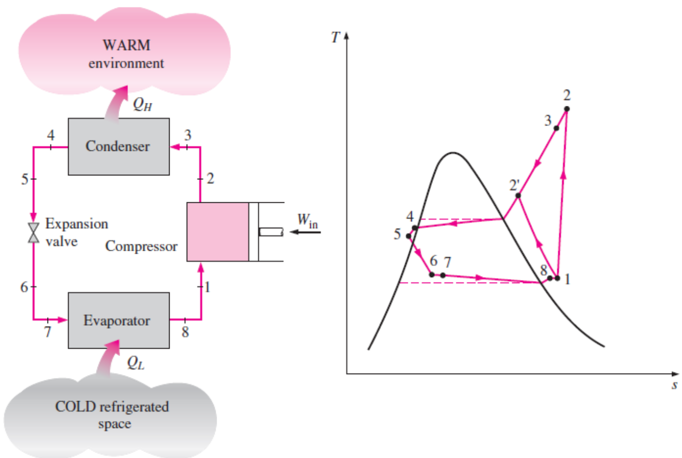


Figure 5: Schematic and T-s diagram of an actual vapour-compression refrigeration cycle

Data Reduction

The following section looks at the mathematical and quantitative aspect of the thermodynamics for an actual vapour-compression refrigeration cycle. Data for each compressor speed was calculated using Excel. However, below shows some manual calculation for a compressor speed of 50%, along with some notes and descriptions of what is happening at each stage.

To start, we can define our constants:

$$\rho_{R-134a} = 1206 \left[\frac{kg}{m^3}\right]; \; C_{p_{water}} = 4.18 \left[\frac{kJ}{kg}\right]; \; \rho_{water} = 1000 \left[\frac{kg}{m^3}\right]$$

Flow Rates:

The **mass flow rate** is taken by multiplying the volumetric flow rate **of the refrigerant fluid** with its density:

$$\dot{m}_{R-134a} = Q_{refrig}\left[\frac{m^3}{s}\right] \cdot \rho_{R-134a}\left[\frac{kg}{m^3}\right]$$

The **volumetric flow rate** at a compressor speed of 50% was:

$$Q_{R-134a} = 17\ [L/hr] = \frac{17}{(3600)(1000)} = \ 4.72 \times 10^{-6}\ \ [m^3/s]$$

$$\dot{m}_{R-134a} = (4.72 \times 10^{-6})(1206) = 5.695 \times 10^{-3}\ [kg/s]$$

The **mass flow rate of water** through the **evaporator** is:

$$\dot{m}_{water,evap} = Q_{water}\left[\frac{m^3}{s}\right] \cdot \rho_{water}\left[\frac{kg}{m^3}\right]$$

The **volumetric flow** was:

$$Q_{water} = 5.4\ [L/min] = \frac{5.4}{(60)(1000)} = 9 \times 10^{-5}\ \ [m^3/s]$$

$$\dot{m}_{water,evap} = (9 \times 10^{-5})(1000) = 0.0899\ [kg/s]$$

The **mass flow rate of water** through the **condenser** is:

$$\dot{m}_{water,cond} = Q_{water}\left[\frac{m^3}{s}\right] \cdot \rho_{water}\left[\frac{kg}{m^3}\right]$$

$$Q_{R-134a} = 2.7\ [L/min] = \frac{2.6}{(60)(1000)} = 4.33 \times 10^{-5}\ \ [m^3/s]$$

$$\dot{m}_{water,cond} = (4.33 \times 10^{-5})(1000) = 0.0433\ [kg/s]$$

Evaporator Energy Balance:

Next, we calculate the **heat gained by the refrigerant** as it passes through the **evaporator**, $\boldsymbol{Q_{in,R-134a}}$ $[\boldsymbol{W}]$. This is the stage is the isobaric evaporation of the liquid-vapour refrigerant takes place, turning it into a low-pressure vapour:

$$Q_{in,R-134a} = \dot{m}_{R-143a} \left[\frac{kg}{s}\right] \cdot \Delta h_{evap} \left[\frac{kJ}{kg}\right]$$

Where: $\Delta h_{evap} = h_3 - h_7$

N.B: refer to 'Calculations for Various Compressor Speeds'.

$$Q_{in,R-134a} = (5.695 \times 10^{-3})(170) = 0.968\ kJ/s\ = 968\ W$$

When the **water passes through the evaporator, it loses heat**. This loss of heat in the water is due to the relative difference in temperature between the two fluids, $\boldsymbol{Q_{in,water}}$ $[\boldsymbol{W}]$:

$$Q_{in,water} = \dot{m}_{water,evap} \left[\frac{kg}{s}\right] \cdot C_{p_{water}} \left[\frac{kJ}{kg \cdot K}\right] \cdot \Delta T_{water,evap}\ [K]$$

Where: $\Delta T_{water,evap} = T_8 - T_9$

N.B: refer to 'Calculations for Various Compressor Speeds'.

$$Q_{in,water} = (0.0899)(4.18)(2.4) = 0.886\ kJ/s\ =\ 901.88\ W$$

Condenser Energy Balance:

Next, we calculate the **heat lost by the refrigerant** as it passes through the **condenser**, $\boldsymbol{Q_{out,R-134a}}$ $[\boldsymbol{W}]$. This is the stage is the isobaric condensation of the high-temperature, high-pressure refrigerant takes place, turning it into a liquid:

$$Q_{out,R-134a} = \dot{m}_{R-143a} \left[\frac{kg}{s}\right] \cdot \Delta h_{cond} \left[\frac{kJ}{kg}\right]$$

Where: $\Delta h_{cond} = h_4 - h_5$

$$Q_{out,R-134a} = (5.695 \times 10^{-3})(186) = 1.059\ kJ/s\ = 1059\ W$$

When the **water passes through the condenser, it gains heat**. This gain of heat in the water is due to the relative difference in temperature between the two fluids, $\boldsymbol{Q_{out,water}}$ $[\boldsymbol{W}]$:

$$Q_{out,water} = \dot{m}_{water,cond} \left[\frac{kg}{s}\right] \cdot C_{p_{water}} \left[\frac{kJ}{kg \cdot K}\right] \cdot \Delta T_{water,cond}\ [K]$$

Where: $\Delta T_{water,evap} = T_2 - T_1$

$$Q_{out,water} = (0.0433)(4.18)(5.9) = 1.091\ kJ/s\ =\ 1067.86\ W$$

Compressor Energy Balance:

When the **refrigerant passes through the compressor**, it undergoes a phase change. The pressure applied causes the refrigerant temperature to increase. As the gas passes through the compressor, it **exits as a higher-temperature, higher-pressure gas**. The refrigerant gains energy from the compressor, $\boldsymbol{W_{in,R-134a}}$:

$$W_{in,R-134a} = \dot{m}_{R-143a}\left[\frac{kg}{s}\right] \cdot \Delta h_{comp}\left[\frac{kJ}{kg}\right]$$

Where: $\Delta h_{comp} = h_4 - h_3$

$$W_{in,R-134a} = (5.695 \times 10^{-3})(16) = 0.091\ kJ/s\ = 91\ W$$

The **compressor is powered** by an external source of energy, $\boldsymbol{W_{in,electrical}}$:

$$W_{in,electrical} = V\ [V] \cdot I\ [A] = (24)(7.5) = 180\ W$$

Performance Analysis:

Total power in the **system**:

$$W_{total,system} = Q_{in,water} + W_{in,electrical} = 901.88 + 180 = 1081.88\ W$$

Total power in the **refrigerant**:

$$W_{total,R-134a} = Q_{in,R-134a} + W_{in,R-134a} = 968 + 91 = 1059\ W$$

Total power leaving the **system** through **water**:

$$Q_{out,water} = 1067.86\ W$$

Total power leaving the system through **refrigerant**:

$$Q_{out,R-134a} = 1059\ W$$

Compressor efficiency:

$$\eta_{compressor}[\%] = \frac{W_{in,R-134a}}{W_{in,electrical}} \times 100 = \frac{91}{180} \times 100 = 50.56\%$$

Coefficient of Performance (COP):

$$COP = \frac{Q_{in,water}}{W_{in,electrical}} = \frac{901.88}{180} = 5.01$$

The above calculations may vary slightly to those from the Excel spreadsheet. This is mainly due to rounding errors, as Excel takes many decimal places, which I tend to only round to one or two.

Experimental Apparatus and Procedure

Apparatus

In the lab, we used the Armfield RA1-MKII computer-controlled vapour-compression refrigeration unit. The idea was to assess what effect varying the compressor speed had on the system's overall performance. The mechanical system was linked to a computer which ran software capable of extracting data points such as temperatures, pressures, and flow rates. This made the final analysis very straightforward.

The system houses many different components, as seen in the labelled imaged below:

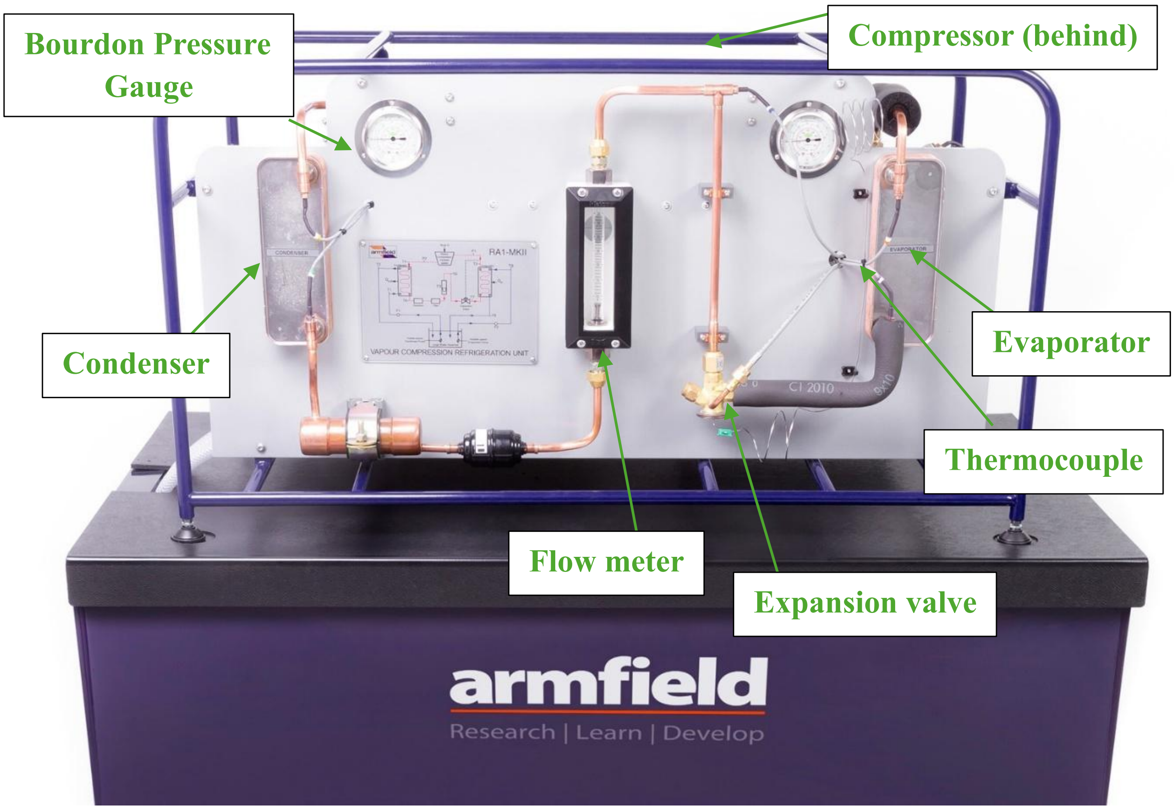
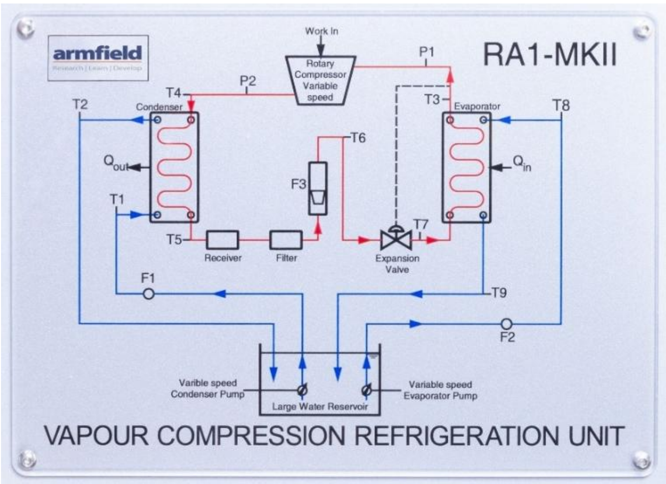


Figure 6: Labelled components of the vapour-compression refrigeration unit

Compared to a household or even an industrial refrigerator, it looks very different. But the operation process is very similar. The system uses an array of measurement instrumentation to extract the data and feed it to the computer software. The two bourdon pressure gauges records the pressure of the R-134a refrigerant as it passes through the different points of the system. The thermocouples, electronic devices capable of sensing temperature difference, record the temperatures at 9 different points in the



system. The flow meter is positioned in the middle of the apparatus, and it records the flow rate of the refrigerant within the system.

The bourdon pressure gauges work by using a flattened, curved metal tube. When internal pressure is applied, the curvature of the tube tends to decrease (it straightens). This pressure difference is then converted to a mechanical movement, which changes the position of a pointer on a calibrated scale. There are many sources of error in this type of gauge if not calibrated correctly. If reading directly from the gauge, human error can have an effect. If the mechanical apparatus that moves the gauge is faulty, the outputted value can include some degree of error. If the fluid flow is obstructed, or inconsistent that can also affect the accuracy of the data [2].

Figure 7: Schematic of the Armfield RA1-MKII vapour-compression refrigeration unit.

The thermocouples sense a temperature change by using two wires, which are attached to a metallic surface. Thomas Seebeck discovered that a circuit composed of two dissimilar metals will generate an electromagnetic field if the junctions at the end of those metals are kept at different temperatures. Due to each type of metal experiencing a different heat at an atomic level, electrons are pushed around the metal at different rates, causing a potential difference. This potential difference is then reflected as a temperature difference between the two points and is given as a reading by the thermocouple. The typical error in thermocouple readings varies between device models and specifications [3]. However, according to the 'NIST ITS-90 Thermocouple Database', depending on the thermocouple type (metallic compositions combination), the accuracy can range from anywhere between ±2.2°C and ±0.25°C [4]. Reasons for this margin of error can be due to many different sources. If the metallic compositions are wrong (too pure/not pure enough), the readings may contain error. If the wires are damaged, or have too much or too little electrical resistance, this may also lead to error.

The variable-area flowmeter is an instrument for measuring the flow of liquids and gases in pipelines. It includes a vertical tube through which the fluid flows whose diameter increases from the bottom to the top and a float which can move vertically in the tube. As the flow increases, this float moves to a higher position until its resistance to the fluid flow is balanced by the float's buoyed weight in the fluid, a value which is constant and independent of the flowrate. The position of the float is a measure of the flowrate. The flowrate values can be read on the scale [5]. This device can also output error. Similarly to the bourdon pressure gauge, the fluid mechanics can affect the data. The scale also may not be calibrated correctly.

Procedure

The experimental procedure was carried out as follows:

The system was turned on and connected to the computer. The condenser water pump (pump 1) was set to a flow rate speed (F1) of 40%. The evaporator water pump (pump 2) was set to a flow rate speed (F2) of 60%. The compressor speed was then set to 10% (2000 RPM) initially to allow the device to initialise and pump fluid around. We confirmed that the fluid was flowing by checking the variable area flow meter (F3). The 'Compressor On' button was then pressed, which sets it to a speed of 3000 RPM. It was let run for 30 seconds, and then it was changed to our first desired speed of 25%.

The system was run until the temperatures reached steady state. The refrigerant flow rate was read off the flow meter (F3) and entered into the software. This process was carried out for a compressor speed of 25%, 50%, 75%, and 100%. We ensured that each time the process was run, the temperatures were allowed to reach steady state. Once each compressor speed was run, the speed was set to 50% and the system was allowed to settle. Once the system was settled, the 'Stop' button was clicked to stop recording the data, and 'Compressor On' was clicked again to stop running the apparatus.

Results

The enthalpies for each compressor setting of 25%, 50%, 75% and 100% were calculated based on their respective p-h charts. The calculations and charts can be viewed at: 'Enthalpy Calculations for Various Compressor Speeds'.

Compressor Speed	h_3 (kJ/kg)	h_4 (kJ/kg)	h_5 (kJ/kg)	h_7 (kJ/kg)
25%	412	425	240	240
50%	413	429	243	243
75%	412	432	235	235
100%	411.5	433	237	237

This above table shows the enthalpy values for each compressor speed. Note that between h_5 and h_7, an isenthalpic expansion occurs, hence why there is no enthalpy change. These results are fairly accurate, however there is of course a margin of error that should be accounted for. As the data was interpolated on the p-h charts, human error should be factored. It is hard to get a precise reading from these charts, hence why most of the figures are whole numbers.

By finding these enthalpy values, we were able to calculate many different measures of performance within the system. These calculations were computed manually for a compressor speed of 50% and can be seen in the 'Data Reduction' section.

$T_{water\ evap}$ (°C)	$T_{water\ cond}$ (°C)
2.0	4.8
2.4	5.9
2.8	6.8
3.1	7.8

Using the formulae previously used when calculating values at a compressor speed of 50%, and also Excel, the remaining compressor speed values can be calculated in a tabular fashion:

Table Colour Legend for Compressor Speeds:	25%	50%	75%	100%

Flow Rates:

Mass Flow Rate of Refrigerant [kg/s]	Mass Flow Rate of Water Through Evaporator [kg/s]	Mass Flow Rate of Water Through Condenser [kg/s]
0.00469	0.089746094	0.043701172
0.005695	0.089892578	0.043701172
0.0067	0.089306641	0.044580078
0.00804	0.089453125	0.044726563

Evaporator Energy Balance:

Heat Gained by R-134a in Evaporator [W]	Heat Lost by Water in Evaporator [W]
806.68	759.8294859
968.15	913.6641584
1185.9	1053.124657
1402.98	1177.515538

Condenser Energy Balance:

Heat Lost by R-134a in Condenser [W]	Heat Gained by Water in Condenser [W]
867.65	874.5960591
1059.27	1070.800967
1319.9	1270.97404
1575.84	1449.507389

Compressor Energy Balance:

Work Done on the R-134a in the Compressor [W]	Electrical Power into the Compressor [W]
60.97	131.015625
91.12	180.984375
134	230.2734375
172.86	288.7265625

Performance Analysis:

Total Power in the System [W]	Total Power in the R-134a [W]	Total Power Leaving the System Through Water [W]	Total Power Leaving the System Through R-134a [W]
890.8451109	867.65	874.5960591	867.65
1094.648533	1059.27	1070.800967	1059.27
1283.398094	1319.9	1270.97404	1319.9
1466.2421	1575.84	1449.507389	1575.84

Compressor Efficiency [%]	Coefficient of Performance (COP)
46.53643411	5.799533345
50.34688768	5.048304078
58.19168787	4.573365769
59.86979463	4.07830692

The above data was calculated using the enthalpies, pressures, temperatures, and other values supplied by the computer linked to the refrigeration system.

We can see from the above chart that the COP drops off significantly when it reaches an operating speed of 100%. This could be due to many reasons. While the compressor is operating at 100%, many would think that this means there is a linear relation between the power consumption and cooling performance. However, the relation is not linear due to many factors such as fluid flow, friction, heat losses, and performative cooling limits. Often the compressor power rises faster than the cooling, which leads to a decreased COP. This increased compressor speed, may also mean that there is less time for the heat exchange to occur. As we discussed previously, in order to increase the coefficient of performance the temperature difference between the hot and cold sinks should be minimised. However, we discussed that this is not necessarily a fair evaluation of the system due to feasibility issues.

Aside from the interpretation of the p-h diagrams, there are also many other possible sources of experimental uncertainty that have to be accounted for. Sensor and instrument inaccuracy of the thermocouples, flow meter, and bourdon pressure gauges can occur if the devices are faulty or inaccurately calibrated. There may be heat loss to the environment if the pipes or devices are not insulated correctly or are damaged. Mechanical instruments such as valves may be loose or drift over time with use. The sampling rate of the data may be too low for minor fluctuations to be picked up.

Discussion

First Law of Thermodynamics

Thermodynamic Energy Balances:

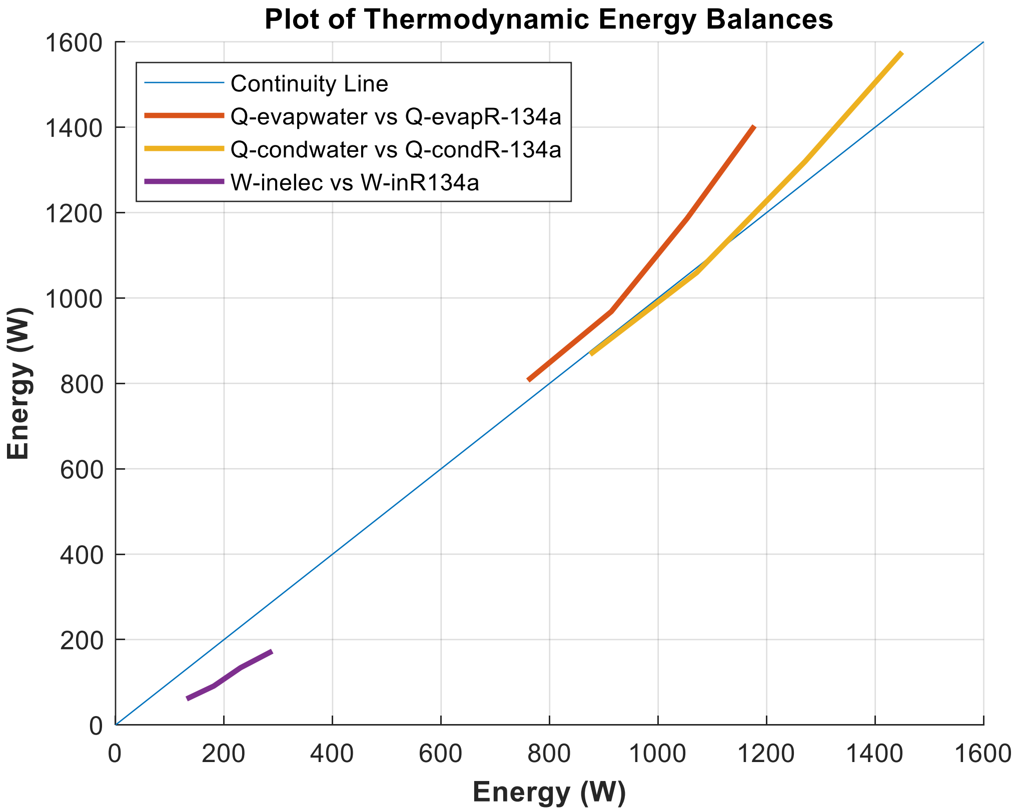


Figure 8: Plot of the energy balances within the refrigeration cycle.

The above plot shows the energy balances across the system during the refrigeration cycle. The energy transfer within the evaporator, condenser, and the compressor are all plotted. Based on the first law of thermodynamics, every watt supplied to the system should be conserved between mediums. However, due to the system not being an ideal, nor a Carnot, refrigeration cycle, a perfect conservation of energy does not exist. As we discussed previously, this is due to a multitude of factors that are present in an actual cycle such as heat loss, friction, and error. The continuity line, in blue, has a slope of 1 and represents a system that is operating with perfect energy conservation. It serves as a refence point to the other processes that are graphed, all of whom incur energy losses.

Total Power In vs Total Power Out:

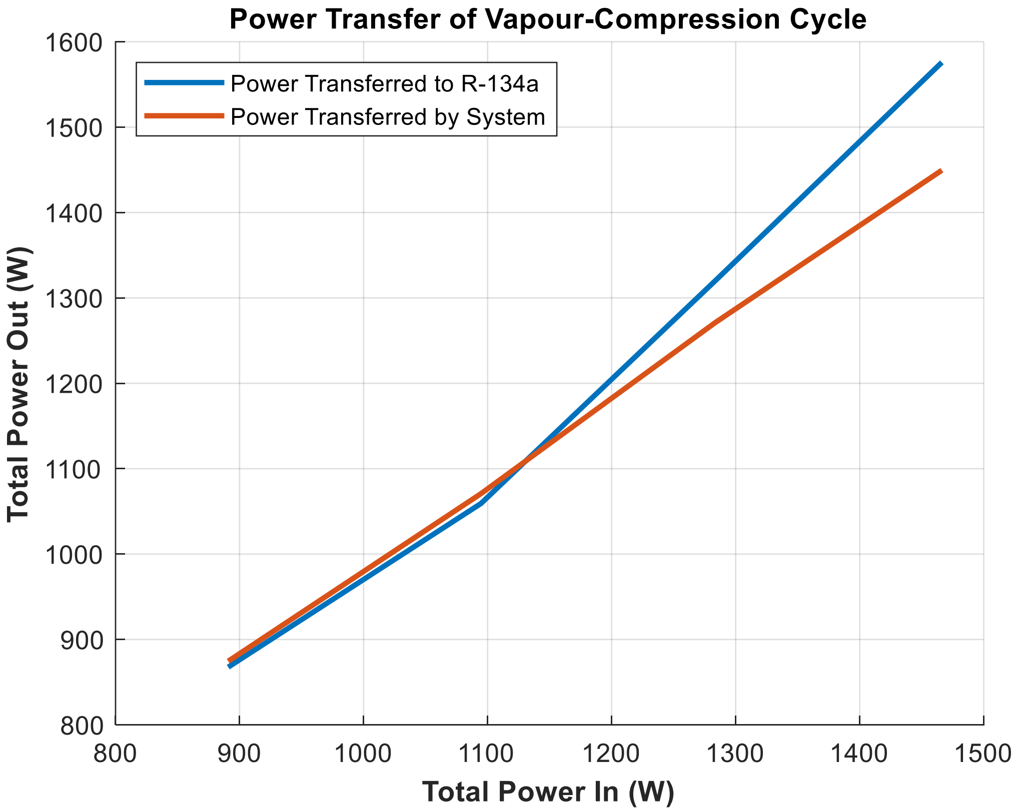
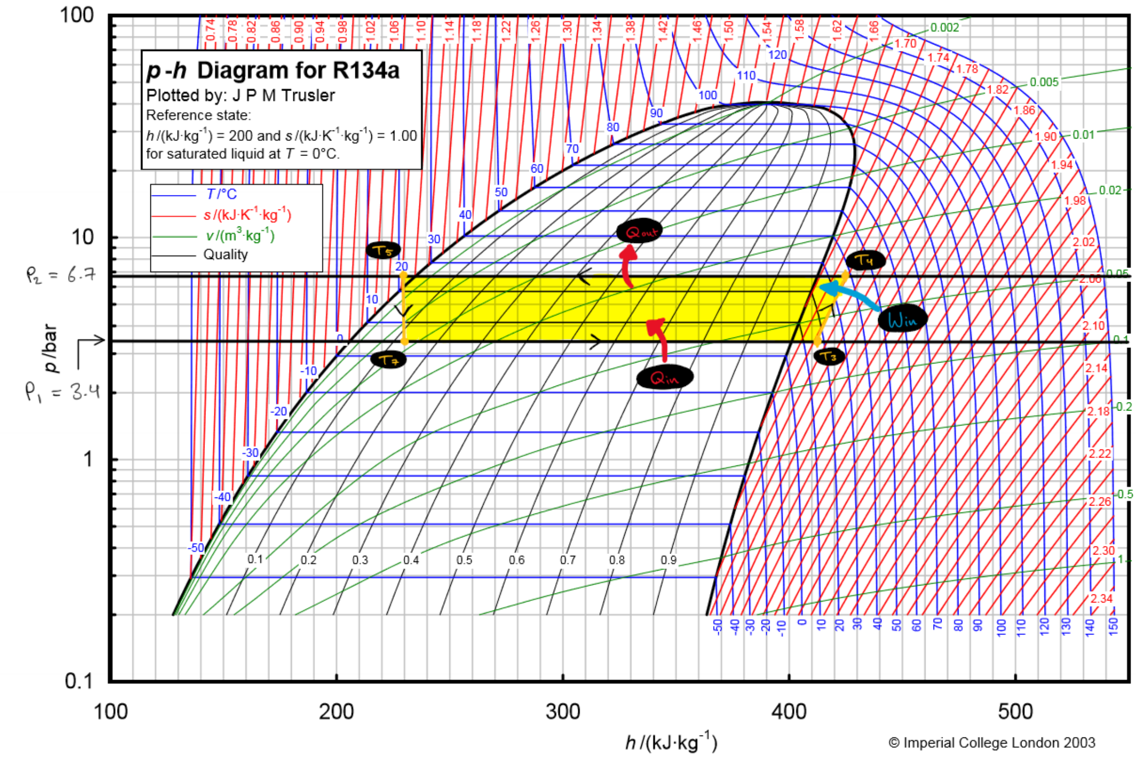


Figure 9: Total power leaving system vs total power entering system

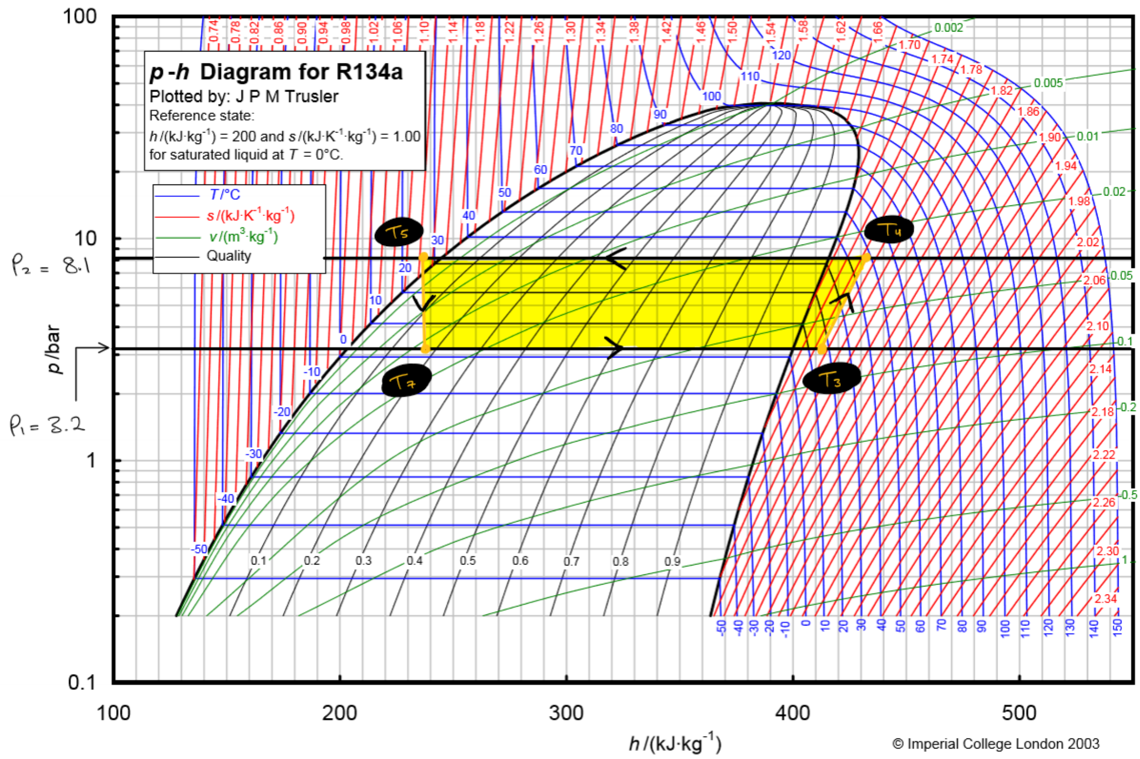
Like with the plot for energy balances, we would expect a perfectly linear relation between the power input and power output of the system based on the first law of thermodynamics. We would expect the power into the system to be equal to the power leaving the system, and the power put into the refrigerant (blue line), to be equal to the power transferred by the system (orange line). However, due to it being an actual vapour-compression cycle, as opposed to a Carnot or ideal one, there is of course an imperfect balance in power transfer. From the above graph, we can see that at the beginning of the plot, the two lines are relatively close to another, and both start with the power input being almost equal the power leaving the system. The lines begin to diverge until around 1100 W on the input side. This corresponds to a compressor speed of around 55%. From this point on, the power transferred to the refrigerant greatly increases. The power transfer to the system holds a linear relationship for each stage of the cycle, but the power input is always greater than the power leaving the system. This is expected, and this discrepancy can be attributed to friction, heat loss, and other forms of energy loss in the system.

Influence of Compressor Speed on Refrigeration Unit Performance

p-h diagram for the lowest compressor speed of 25%:



p-h diagram for the highest compressor speed of 100%:



For a compressor speed of 25%, the corresponding p-h diagram shows a relatively small cycle compared to that for a higher compressor speed of 100%. I shaded the area of the cycle in yellow. As we can see, the distance between the pressure bars at a compressor speed of 25% is much smaller than that of 100%. As the vertical lines on the graph represent the compression and expansion of the refrigerant, we can see how the increase in compressor speed has a direct correlation to an increase in pressure. For this lab, we assumed that the expansion process across the valve (h_5 to h_7) was an isenthalpic process. However, this was idealised. In an actual cycle, we would see enthalpy fluctuations due to inconsistent pressure changes, energy loss, and even due to valve defects. As the plot widens, there is an increase in heat rejection and also heat input to the system. We can see that the pressure across the evaporator (P_1) stays more or less equal at both 25% and 100% compressor speed. Meanwhile, the pressure across the condenser (P_2) increases as the compressor speed increases.

Compressor Efficiency (Ƞ) vs Compressor Speed (%):

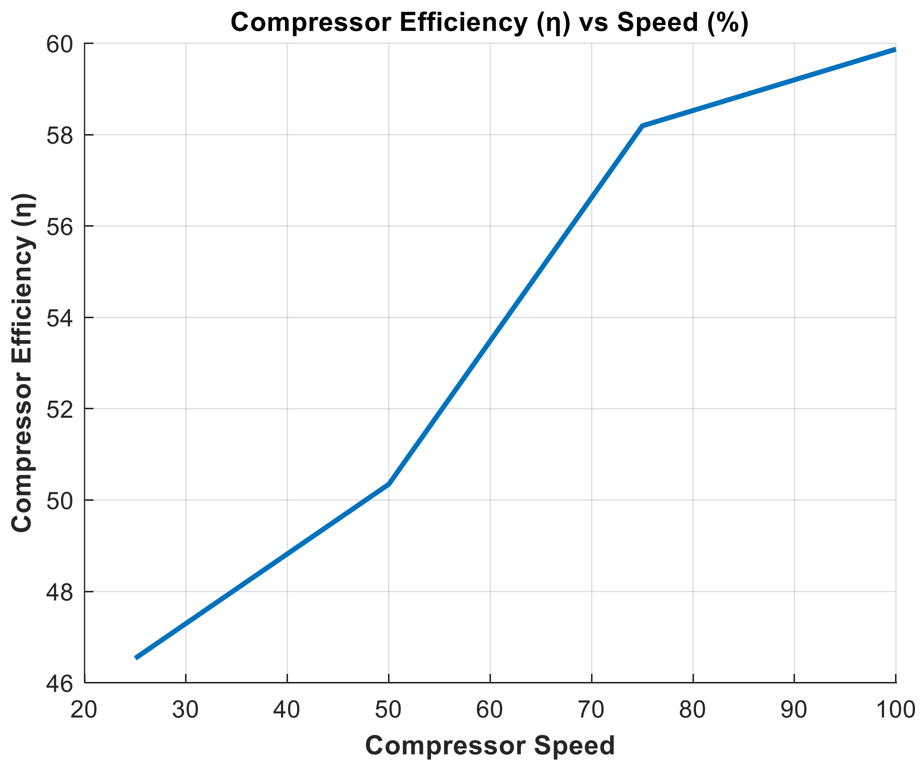


Figure 10: Plot of the compressor efficiency vs the compressor speed of 25-100%

Coefficient of Performance (COP) vs Compressor Speed (%):

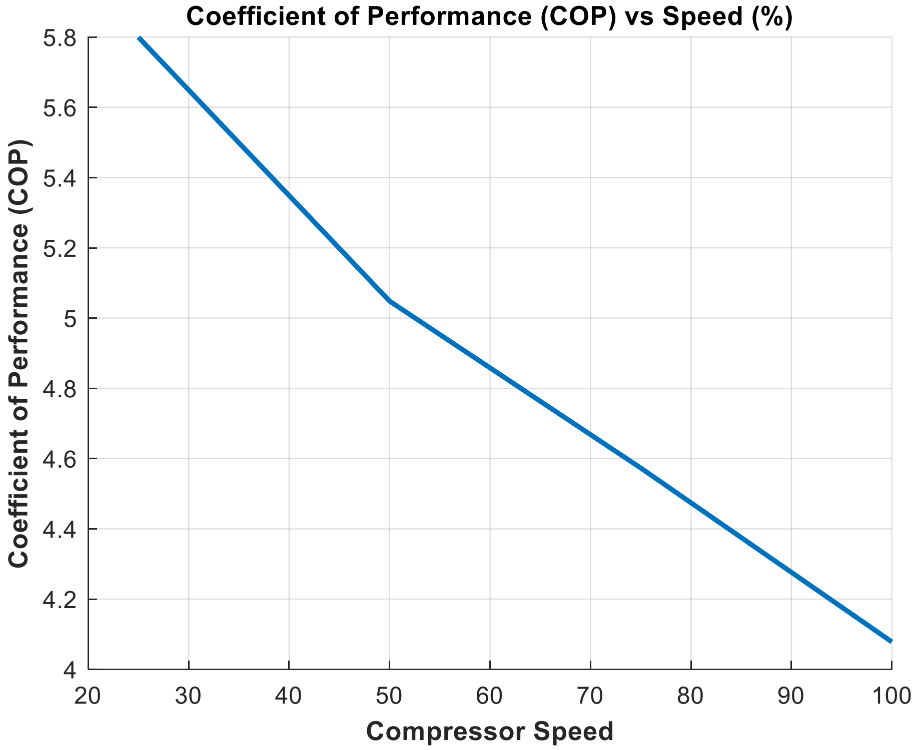


Figure 11: COP vs compressor speed of 25-100%

Conclusion

The objective from this experiment was to investigate the effects that different compressor speeds have on a vapour-compression refrigeration cycle. Through the different sections of the report, we showed what these effects were and how they effect performance and efficiency. By carrying out the experiment, we were able to extract data from the refrigeration system which allowed us to find four key enthalpy values. These values were used to plot four different p-h diagrams at different compressor speeds. We concluded, through the first law of thermodynamics, that by increasing the compressor speed (increasing the power input to the system), the system became more pressurised, which in turn caused an increased cooling phenomenon. Of course, we also had to factor in the effects of error in the processes. These were described and accounted for thoroughly.

We also investigated the energy balances of the system by plotting our data. While our data suggested that as the compressor speed increased compressor efficiency also increased, the coefficient of performance (COP) drastically decreases in the buildup to working at maximum speed. It is worth noting that while the efficiency did increase from 75% to 100%, the rate that it changed was much less compared to the increase in efficiency from 50% to 75%. This perhaps indicates that the system efficiency was beginning to plateau. This is likely due to mechanical limitations in the apparatus and energy loss through heat. We also concluded that a real refrigeration cycle varies a lot compared to an ideal or Carnot cycle due to energy losses in real world application.

Appendix

Enthalpy Calculations for Various Compressor Speeds:

$\Delta h_{evap} = h_3 - h_7$

$\Delta h_{comp} = h_4 - h_3$

$\Delta h_{cond} = h_4 - h_5$

$\Delta h_{expan} = 0\ (isenthalpic)$

Compressor speed of 25%:

T_3 (°C)	T_4 (°C)	T_5 (°C)	T_7 (°C)	P_1 (bar)	P_2 (bar)
16.7	37.1	21.6	7.7	3.4	6.7

T_8 (°C)	T_9 (°C)	T_1 (°C)	T_2 (°C)	Motor Current (A)
17.2	15.1	17.4	22.2	5.5

$\Delta T_{water,evap} = T_8 - T_9 = 17.2 - 15.1 = 2.1\ °C$

$\Delta T_{water,cond} = T_2 - T_1 = 22.2 - 17.4 = 4.8\ °C$

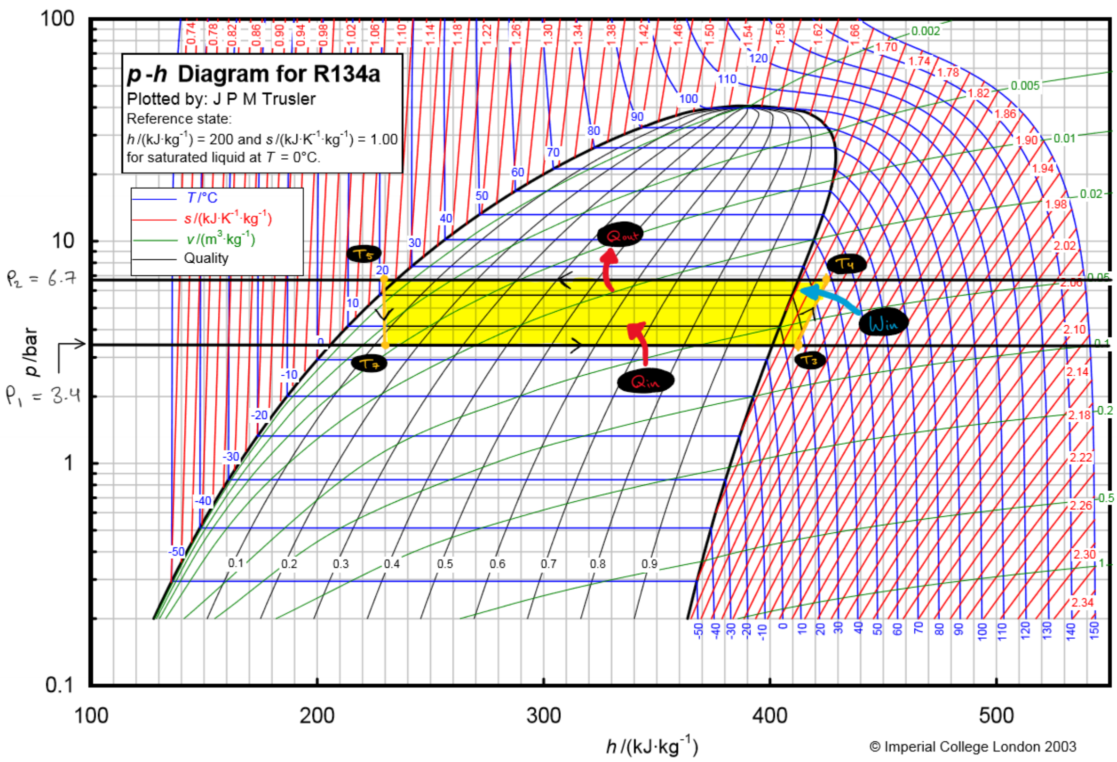


Figure 12: p-h diagram for R-134a at a compressor speed of 25%

From the above figure, we can approximate the enthalpies at each point to be:

h_3 (kJ/kg)	h_4 (kJ/kg)	h_5 (kJ/kg)	h_7 (kJ/kg)
412	425	240	240

$$\Delta h_{evap} = h_3 - h_7 = 412 - 240 = 172\ kJ/kg$$
$$\Delta h_{comp} = h_4 - h_3 = 425 - 412 = 13\ kJ/kg$$
$$\Delta h_{cond} = h_4 - h_5 = 425 - 250 = 172\ kJ/kg$$

Compressor Speed of 50%:

T_3 (°C)	T_4 (°C)	T_5 (°C)	T_7 (°C)	P_1 (bar)	P_2 (bar)
16.7	41.5	23.5	7.3	3.4	7.1

T_8 (°C)	T_9 (°C)	T_1 (°C)	T_2 (°C)	Motor Current (A)
17.2	14.8	17.5	23.3	7.4

$$\Delta T_{water,evap} = T_8 - T_9 = 17.2 - 14.8 = 2.4\ °C$$
$$\Delta T_{water,cond} = T_2 - T_1 = 5.8\ °C$$

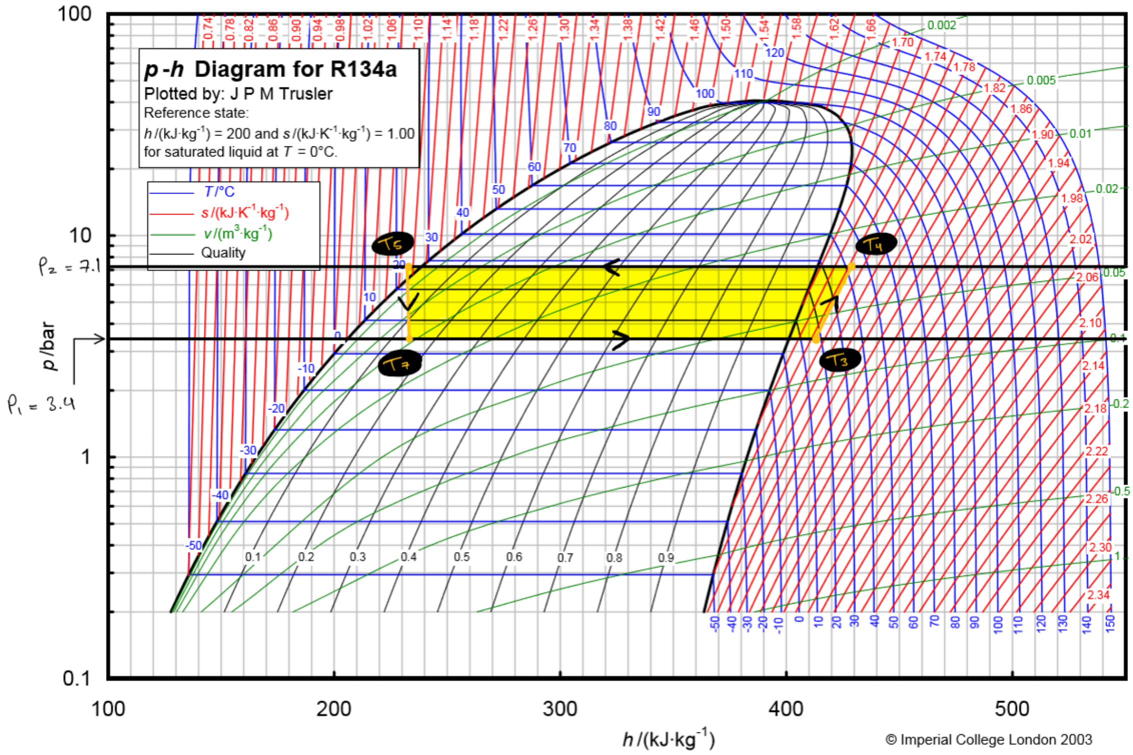


Figure 13: p-h diagram for R-134a at a compressor speed of 50%

From the above figure, we can approximate the enthalpies at each point to be:

h_3 (kJ/kg)	h_4 (kJ/kg)	h_5 (kJ/kg)	h_7 (kJ/kg)
413	429	243	243

$$\Delta h_{evap} = h_3 - h_7 = 413 - 243 = 170\ kJ/kg$$
$$\Delta h_{comp} = h_4 - h_3 = 429 - 413 = \ 16\ kJ/kg$$

$\Delta h_{cond} = h_4 - h_5 = 429 - 243 = 186\ kJ/kg$

Compressor Speed of 75%:

T_3 (°C)	T_4 (°C)	T_5 (°C)	T_7 (°C)	P_1 (bar)	P_2 (bar)
16.3	45.1	25.0	6.9	3.3	7.6

T_8 (°C)	T_9 (°C)	T_1 (°C)	T_2 (°C)	Motor Current (A)
17.4	14.6	17.7	24.6	9.6

$\Delta T_{water,evap} = T_8 - T_9 = 17.4 - 14.6 = 2.8\ °C$

$\Delta T_{water,cond} = T_2 - T_1 = 24.6 - 17.7 = 6.9\ °C$

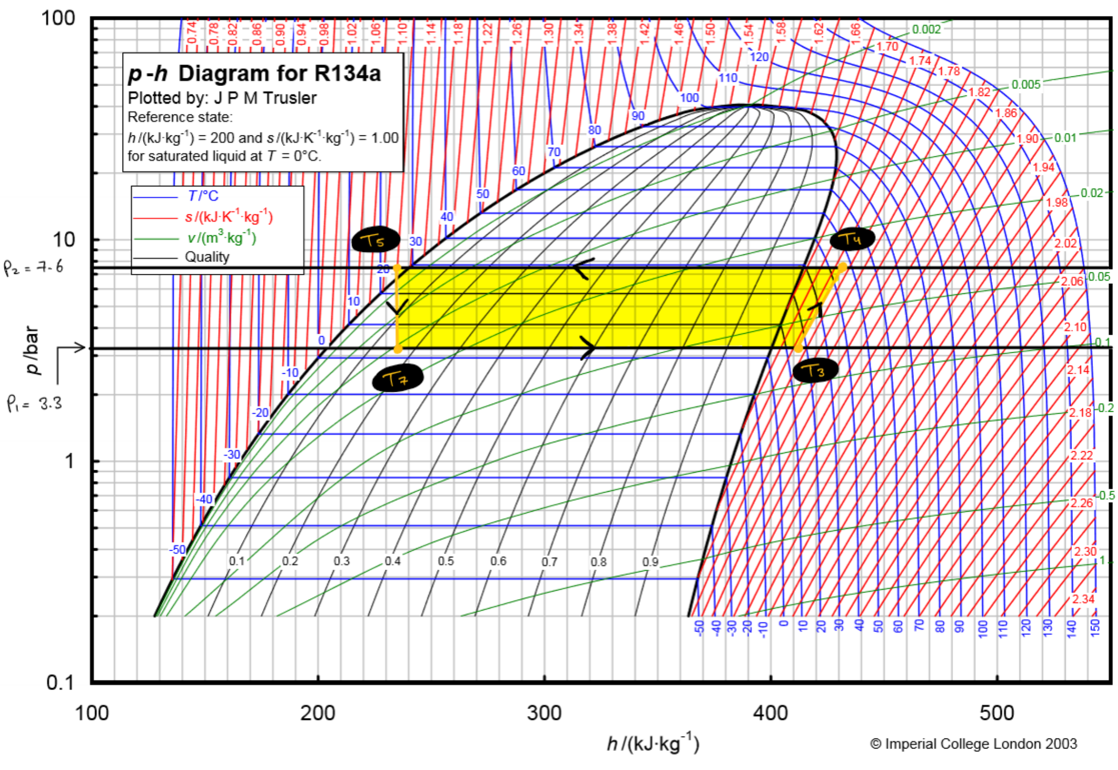


Figure 14: p-h diagram for R-134a at a compressor speed of 75%

From the above figure, we can approximate the enthalpies at each point to be:

h_3 (kJ/kg)	h_4 (kJ/kg)	h_5 (kJ/kg)	h_7 (kJ/kg)
412	432	235	235

$\Delta h_{evap} = h_3 - h_7 = \ kJ/kg$

$\Delta h_{comp} = h_4 - h_3 = \ kJ/kg$

$\Delta h_{cond} = h_4 - h_5 = \ kJ/kg$

Compressor Speed of 100%:

T_3 (°C)	T_4 (°C)	T_5 (°C)	T_7 (°C)	P_1 (bar)	P_2 (bar)
15.7	47.3	26.5	6.6	3.2	8.1

T_8 (°C)	T_9 (°C)	T_1 (°C)	T_2 (°C)	Motor Current (A)
17.5	14.4	17.9	25.6	11.9

$$\Delta T_{water,evap} = T_8 - T_9 = 17.5 - 14.4 = 3.1\ °C$$
$$\Delta T_{water,cond} = T_2 - T_1 = 25.6 - 17.9 = 7.7\ °C$$

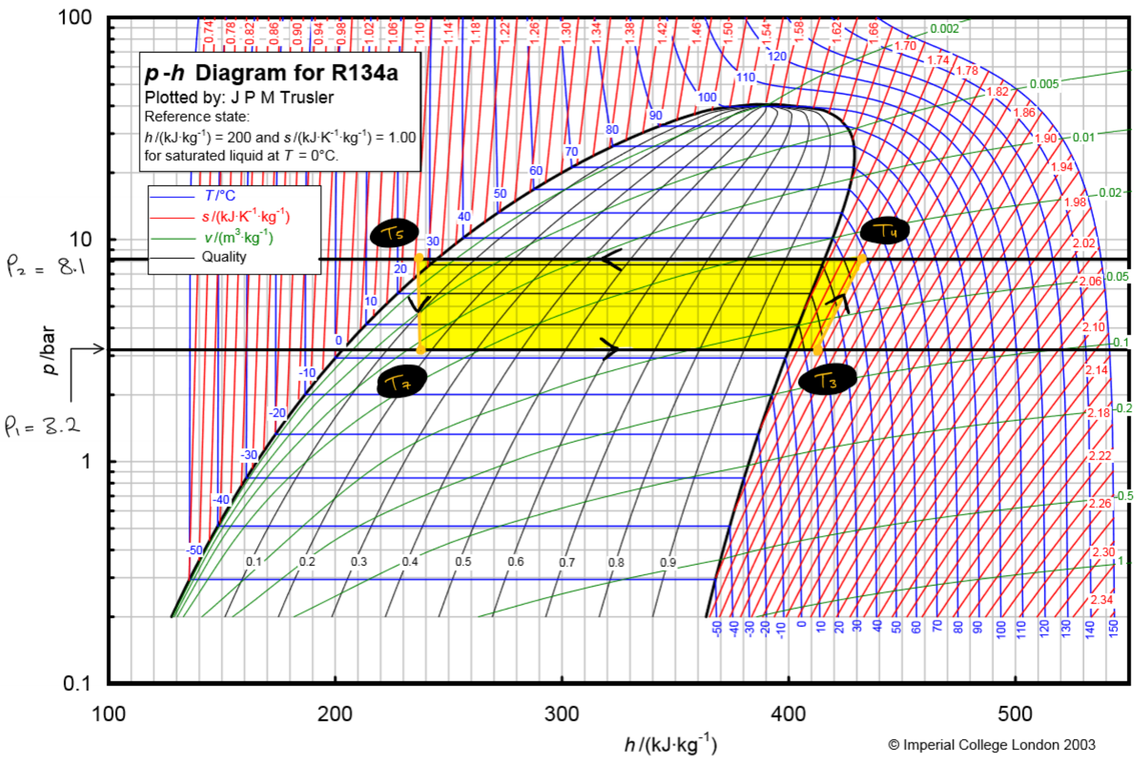


Figure 15: p-h diagram for R-134a at a compressor speed of 100%

From the above figure, we can approximate the enthalpies at each point to be:

h_3 (kJ/kg)	h_4 (kJ/kg)	h_5 (kJ/kg)	h_7 (kJ/kg)
411.5	433	237	237

$$\Delta h_{evap} = h_3 - h_7 = kJ/kg$$
$$\Delta h_{comp} = h_4 - h_3 = \ kJ/kg$$
$$\Delta h_{cond} = h_4 - h_5 = \ kJ/kg$$

References

Text References:

[1]: Çengel, Y.A. and Boles, M.A., 2015. ‘Thermodynamics: An Engineering Approach’. 8th ed. New York: McGraw-Hill Education.

[2]: Hawkins, John T., 1866. ‘The Bourdon Pressure Gauge’. Journal of the Franklin Institute.

[3]: Bajzek, Thomas J., 2005. ‘Thermocouples: A Sensor for Measuring Temperature’. IEEE Xplore.

[4]: NIST ITS-90, 2024. ‘NIST Temperature Scale Database, Standard Reference Database, (SRD) 60, Version 3.0. https://srdata.nist.gov/its90/main/

[5]: ABB, 2001. ‘Variable Area Flowmeter Basic Fundamentals and Descriptions’. Technical Specifications D184B003U46 Rev. 01.
https://library.e.abb.com/public/c125698f006840e9c1256a7f003f7ae1/D184B003U46.pdf

Figure References:

Figure 1, 3, 4, 5: Çengel, Y.A. and Boles, M.A., 2015. ‘Thermodynamics: An Engineering Approach’. 8th ed. New York: McGraw-Hill Education.

Figure 2: https://en.wikipedia.org/wiki/Vapor-compression_refrigeration