Galway – Mayo Institute of Technology GMIT

Preliminary Design of a Demonstration Heat Pump/AC Unit

MECHANICAL ENGINEERING: MAJOR PROJECT

FINAL REPORT

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INDEX

TABLE OF FIGURES

2. BACKGROUND

3. PROCEDURE

TABLE OF APPENDIXES

1. INTRODUCTION

The device that is going to be designed for this Major Project is a demonstration heat pump and air conditioning (AC) system that will be used to demonstrate other students acquiring the knowledge of these devices. It consists of a cheap system developed for the Motor Mechanic Workshop of GMIT used to explain how a real refrigerating system (i.e. in a car) works. The manufacture of the device will be accomplished in following years by other students.

First of all it has to be clear what this system does. The aim is to heat (if it acts as a heat pump) or to cool (if it acts as an air conditioner), and this device will be able to accomplish both tasks. Since heat pumps and air conditioning systems have the same physical principles, the objective is to join both in one dual device.

How do these systems work? Both of them work on refrigeration cycles. So before explaining the differences, it is critical to understand what these cycles consist on.

The basic idea is to make a fluid (called the refrigerant) run over and over in a circuit changing from liquid to vapour. When a fluid changes phases it absorbs or rejects heat from the surroundings, keeping its temperature and pressure constant. In order for the fluid to condense it needs to reject heat, and to evaporate it needs to absorb heat.

These processes can be easily observed in nature. Rain is formed because the sun heats water, this water evaporates (changes from liquid to vapour). When this process occurs, this hot vapour rises into the sky, getting colder and colder as it ascends. When it gets cold enough (has rejected enough heat) it condenses, changing from vapour into a liquid again, and forming raindrops that fall due to gravitational effect. So even in this simple process evaporation and condensation can be observed, this is, a fluid that changes from liquid to vapour and back. A similar cycle is what we want to create in these devices. But in our system, we want to take advantage of the heat that is rejected or absorbed in these state changes to heat or cool (i.e. a room).

Using these principles is how refrigeration cycles are developed. So we need to have both evaporation and condensation occurring continuously in our cycle in order to heat or cool our room. There will be these two clear processes where the fluid changes phases, so we will need an evaporator and a condenser. But we need to join them in order to create a cycle that will work continuously. This is why two other basic components will be needed: The compressor and the throttling valve. They will complete and close the cycle, and the refrigerant will run between four processes. Compression, condensation, throttling and evaporation.

These four processes will be explained briefly, and afterwards will be looked at more deeply.

1.) The refrigerant will be compressed, to rise its temperature and pressure

2.) Next it condensates, so it rejects heat to the surroundings without changing its high pressure and temperature. This rejected heat by the refrigerant will be the one that we will use to heat the room (it's the heat we need to work as a HEAT PUMP)

3.) A throttling process is then established, where the refrigerant lowers its pressure without losing any internal energy. This will join condensation and evaporation.

4.) And finally the fluid will evaporate, so it absorbs heat from the surroundings without changing its new low pressure and temperature. This absorbed heat by the refrigerant will be the one that we will use to cool the room (it's the heat we need to work as an AIR CONDITIONER). After this process, the refrigerant goes again into the compressor to continue with the cycle once again.

Even though our objective is to take advantage from the heat in condenser and evaporator, the compressor is the main element of the cycle, the "heart". It imposes high and low pressures, which are the most important parameters to define these cycles. The aim of this element is to elevate the pressure of the refrigerant, which is in gas state. Here is where the energy is added in form of mechanical work for this cycle to run, following Thermodynamics Laws.

Other auxiliary elements will be used, such as filters and dryers, and there will be a more extended explanation of the theory and elements used in the Theory Part in the report.

In conclusion, an air conditioner's aim is to extract heat from the surroundings (cooling them), and heat pump's is to reject heat to these surroundings (heating them). Having both heat pump and AC in only one device will make understanding easier and this project was needed for the Lecturers area.

This project will also analyze the heat transfer area of the device. It's easy to recognize where heat is exchanged, this is, what components actuate in this way: evaporator and condenser. So in these components it's important to know how efficiently this heat is exchanged, what are the heat transfer coefficients that drive this heat exchange, how big should evaporator and condenser be, etc.

2. BACKGROUND

2.1 Basic Theory of Heat Pumps

The relevant Theory applied in this project will be mainly based on Thermodynamics. As briefly explained in the Introduction, a heat pump is a device that works acquiring heat from a source and releasing heat in a different place, using an external for this process. Basically, pumping heat from a cold source to a hot one is what heat pumps do. This work input is absolutely necessary because as we can experiment, heat naturally flows from a hot object to a cold one. But to make heat go the other way around (“counter nature”) an external work source is needed. This is because heat pumps work according to a refrigeration cycle, as explained in the Introduction, and every cycle has to be in accordance with Modernised Clausius Statement on the Second Law of Thermodynamics:

> “It is impossible to construct a Heat Pump/Refrigerator which operates on a cycle, which requires no work input from the surroundings to transfer heat from a low temperature reservoir to the high temperature reservoir”.

The following schema in Figure 2.1 illustrates this statement. In order to transfer heat from the cold reservoir to the hot one, as a heat pump works, it can be represented as a heat engine running in reverse:

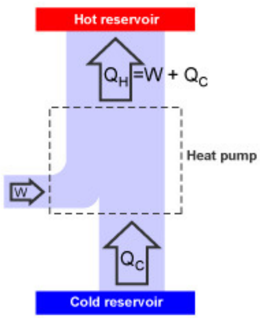


Figure 2.1. Energy Fluxes in a Heat Pump

From this graph the functioning of a heat pump can be extracted. The device absorbs heat from the cold reservoir (cooling the outside in this case) and by the addition of work (into the compressor, which makes the device work) it pumps heat into the hot reservoir (heating the inside in this case).

2.2 The Carnot Heat Pump (Reversible Heat Pump)

Carnot Heat Pump is an ideal one that works in a reversible cycle. The basic cycle consists of the four basic processes described before that would take place at the four main elements of the device. Figure 2.2 illustrates the thermodynamic refrigeration cycle:

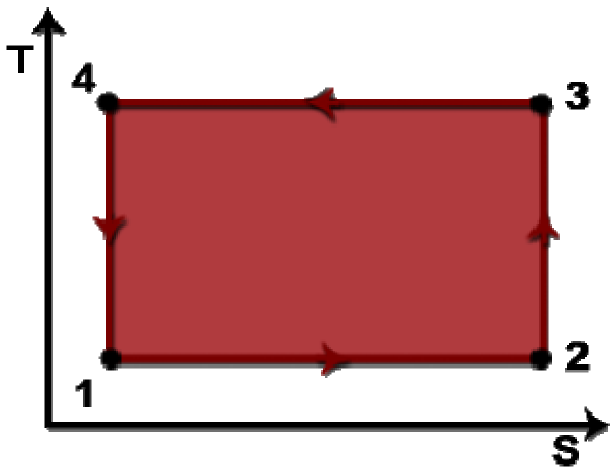


Figure 2.2. Thermodynamic Carnot Refrigeration Cycle

So the four processes described in the T-S diagram in the figure above are:

1-2 Reversible isothermal heat extraction at the low temperature source

2-3 Isentropic Compression

3-4 Reversible isothermal heat rejection at the high temperature source

4-1 Isentropic expansion

To define how good a cycle is, a new variable is defined: the Coefficient Of Performance (COP). It has different values depending of the objective of the device.

-A heat pump aim is to transfer as much heat as possible to the high temperature source with the minimum amount of work put in the system.

- A refrigerator aim is to remove as much heat as possible from the low temperature source with the minimum amount of work put in the system.

That ratio is what defines COPs:

Heat Pump	Refrigerator
$COP_{HP} = \frac{Q_A}{W_{NET}}$ $COP_{HP} = \frac{Q_A}{Q_A - Q_R}$	$COP_R = \frac{Q_R}{W_{NET}}$ $COP_R = \frac{Q_R}{Q_A - Q_R}$

Carnot COPs:

Carnot Heat Pump	Carnot Refrigerator
$COP_{HP} = \frac{T_H}{T_H - T_L}$	$COP_R = \frac{T}{T_H - T_L}$

It has to be clear that all reversible cycles operating between the same temperature reservoirs have the same thermal efficiency (COP), and this efficiency is the highest possible between these fixed temperature reservoirs.

So the efficiency of a real heat pump/refrigerator can't be higher that the equivalent Carnot cycle working between the same temperature limits. This means that Carnot COP is always higher than any other Heat Pump/Refrigerator's COP. This is because of the reversibility in all processes in Carnot cycle, impossible in reality.

This will have to be checked in the system for the project, and check results if don't match this last statement, as it would be impossible.

2.3 The Idealised Vapour Compression Cycle (Reversed Rankine Cycle)

This cycle, which is the one heat pumps and refrigerators work on, consists of 4 processes:

- 2 isobaric processes → (2-3) & (4-1),
- 1 isentropic compression → (1-2) ,
- 1 constant specific enthalpy process (throttling) →(3-4)

The following diagram (Figure 2.3) shows these four processes in a P-h chart, so these ideal processes are clearly stated:

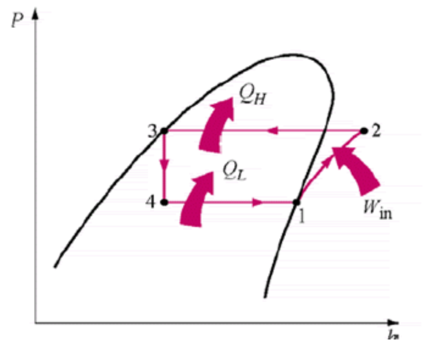


Figure 2.3. Processes in the Idealised Vapour Compression Cycle

The four processes described in the previous graph:

- 1-2 represents the isentropic compression (work input),
- 2-3 represents the isobaric heat rejection to the high temperature source,
- 3-4 represents the constant specific enthalpy throttling process
- 4-1 represents the isobaric heat absorption process from the low temperature source

Work, heat extracted and rejected are calculated by measuring the length of these lines from Figure 2.3:

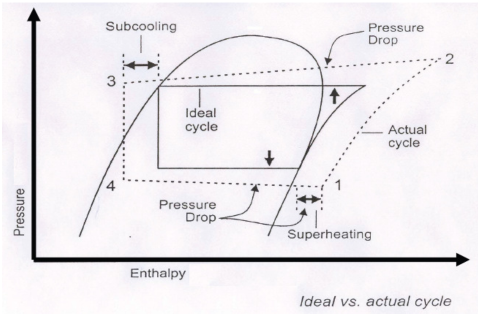
- Specific work required for compression: $h_2 - h_1$,
- Specific heat extracted from the low temperature source: $h_1 - h_4$,
- Specific heat rejected to the high temperature source: $h_2 - h_3$

Now that the fluxes are calculated, establishing COPs is the next step. For the cycle working as heat pump, the COP is the ratio between what we need (heat into the high temperature source) and work required for it (compression work). For the cycle working as refrigerator,, the COP is the ratio between what we need (heat out of the low temperature source) and work required for it (compression work). Hence,

-HEAT PUMP: $COP_{HP} = \frac{h_2 - h_3}{h_2 - h_1}$ -REFRIGERATOR: $COP_R = \frac{h_1 - h_4}{h_2 - h_1}$

2.4 The Actual Vapour Compression Cycle (Reversed Rankine Cycle)

Because of the impossibility to control the system with this accuracy, and the devices are not ideal, which means there are losses inside them, the real vapour compression cycle would look roughly like this:



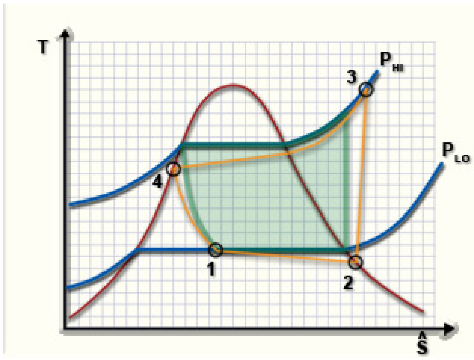


Figure 2.4. Ideal vs. Actual Vapour Compression Cycle (P-h and T-s)

As we can see, the pressure drops from 2-3 and from 4-1, in both condenser and evaporator. This is due to the impossibility to keep the pressure constant in real elements like these. Friction created inside the components induces the pressure loss in this heat exchange (latent heat should be at constant pressure and temperature, as it is inside the dome).

The subcooling in the condenser (process 2-3) is done to assure that the refrigerant is totally liquid, to avoid problems in the throttling valve.

There is also a superheating in the evaporator (process 4-1) to assure that the refrigerant is totally vapour when it enters the compressor. If there was some liquid left at this stage, it could seriously damage the compressor.

Finally, when compressing the refrigerant in process 1-2, the process will not be ideal, this means that the compression will not be an isentropic process.

2.5 Cooling and Heating with Heat Pump Systems

As briefly explained in the Introduction, this system will be able to cool and heat with the same elements because of the principles that work for both are the same. To go from AC to Heat Pump the only change will be to switch the position of a key element: The 4-way valve (or reversing valve). This will make the refrigerant run from one way to the opposite, making the indoor coil act as a condenser in the Heat Pump mode, and after switching the 4-way valve it will act as an evaporator to change to AC mode. Figure 2.5 illustrates this:

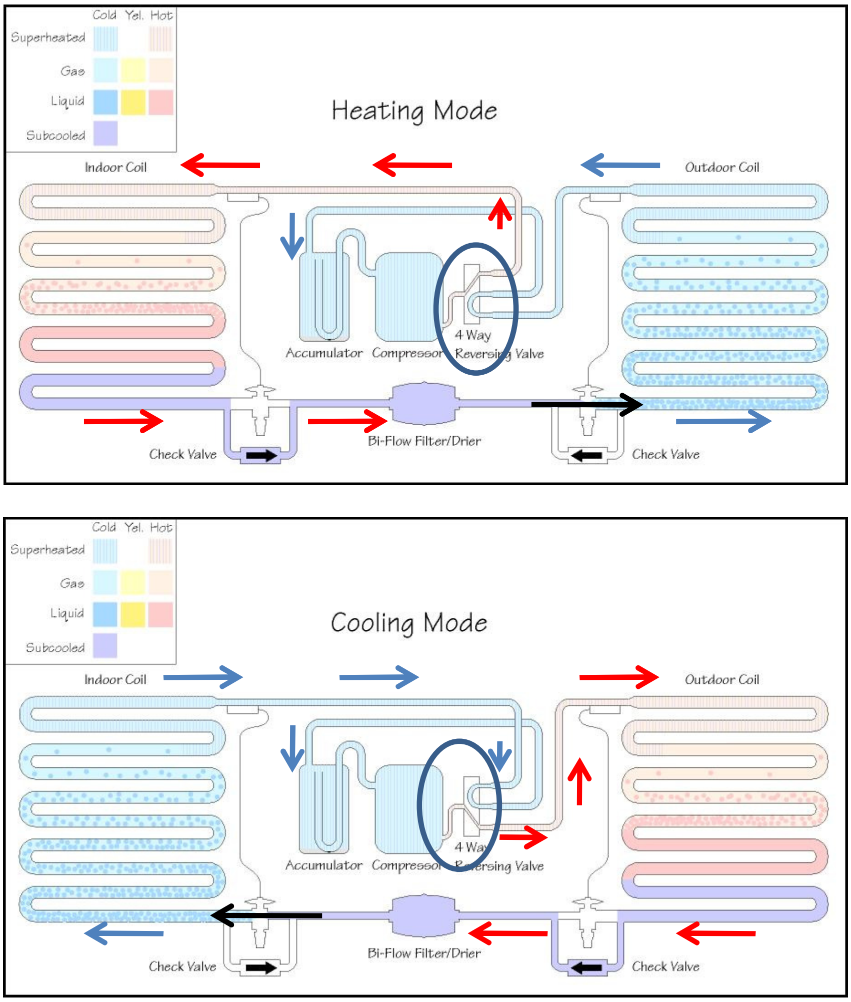


Figure 2.5. Heat Pump mode-AC mode

2.6 Heat Pump Components

In the next figure (Figure 2.6) we can see a typical schema of a Heat Pump that can also work as an AC system:

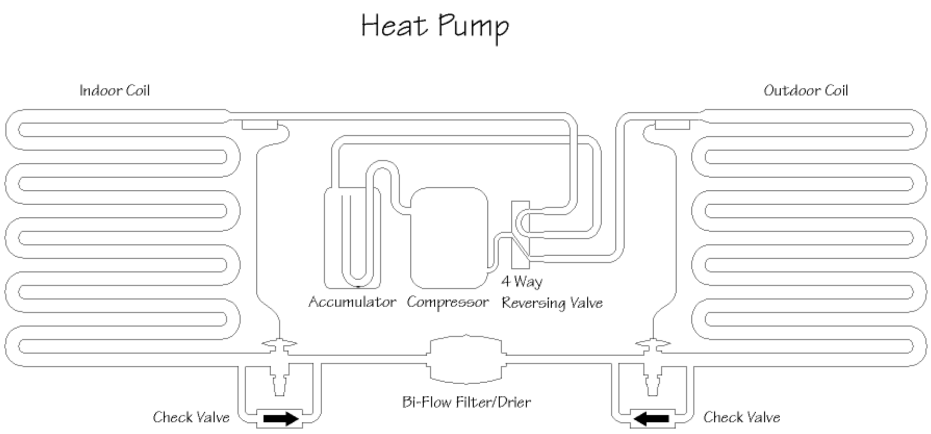


Figure 2.6. Heat Pump Components (Schematic)

In the Introduction the aims of the four basic elements were explained: Compressor, Condenser, Throttling Valve and Evaporator. But as we can see, in this Heat Pump/AC System there are other components, and each one has a specific function. A more exhaustive explanation of compressors will also be added, due to the key importance of this component:

COMPRESSOR: Since the compressor is the core component it is the most stressed during a heat pump's operation. Its function is to suck the refrigerant from the evaporator (low pressure) and compress it to its design pressure, which is typically above atmospheric pressure, and thereafter to deliver it to the condenser. Typical pressures in the suck and feed pipes depend on the refrigerant utilized and thus operating pressures/temperatures in the evaporator and condenser. The can range from: -10 to -20 ^{o}C and 0.1 to 0.5 MPa on the low side of the evaporator suck pipe, and from 60 to 100 ^{o}C and 1.1 to 2.5 MPa on the high side of the condenser feed pipe.

There are two main kinds of compressors:

-**Piston compressors**: Piston compressors or reciprocating compressors are driven by crankshafts as depicted and have been around for over 100 years and thus it is well known and tested technology. In case of heat pumps/refrigerators have a lifetime in excess of 20 years. Input powers can range from 50W to 10s of kW. Although they aren't very efficient, they are quite cheap, so this is going to be the kind of compressor used for this project.

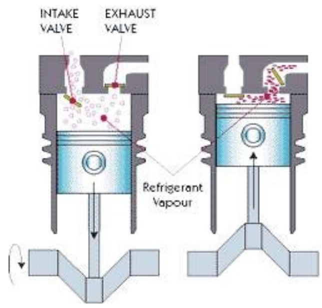


Figure 2.7. Piston Compressor Positions

-**Scroll compressors**: The scroll type compressor was devised and released in 1979 by the Copeland company and today is nearly installed in most heat pump/refrigerators as they are more reliable (lifetime of over 40 years) with a lower power consumption.

Scroll compressors consist of two metal spirals fitted inside one another, one of which is fixed and other which orbits as depicted in the figure. From the diagram depicted, the discharge pipe is fitted to the fixed spiral and the orbiting spiral rotates via an eccentric motor shaft, pockets of crescent-shape refrigerant gas is compressed and forwarded towards the discharge port at the centre of the spirals.

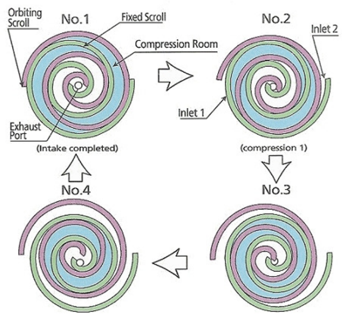


Figure 2.8. Scroll Compressor Positions

<u>Reversing valve</u>: this device changes the direction of the refrigerant flow. Here is another view of how the 4 way valve diverts flow; the pipe on the single stub side of the valve is always the discharge from the compressor. The discharge is diverted to the condenser, the middle stub is always suction going to the compressor, and the left over stub is always from the evaporator. The switching function is accomplished by the sliding back and forth of an internal barrel which has diverting passages. The systems own high side pressure is used to ram the barrel to the desired end by a pilot duty solenoid valve. Figure 2.7 illustrates this:

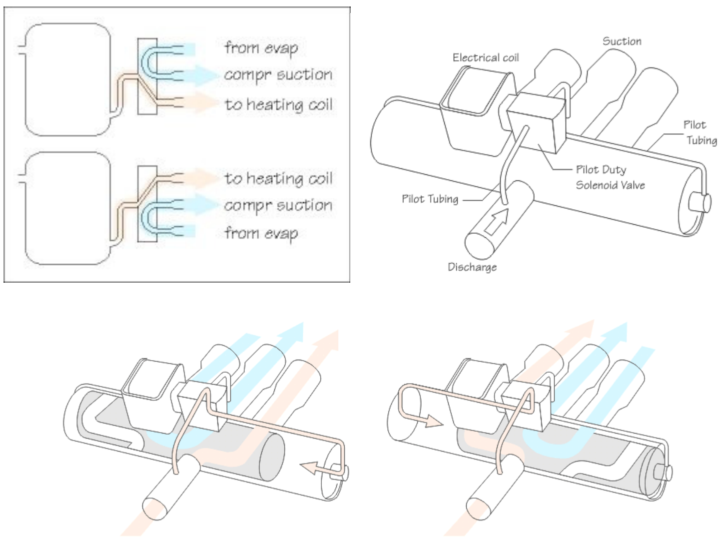


Figure 2.9. Reversing Valve Positions and Operating Method

Sight Glass/Moisture Indicator: this device controls both the existence of bubbles in the refrigerant (this means there is shortage of it) and indicate if there is any moisture in the refrigerant, negative for the circuit. An indicator on the glass changes colour in the presence of moisture, as we can see in the figure:

Figure 2.10. Sight Glass/Moisture Indicator

Bi-Flow Filter/Dryer: This kind of strainer collects particles in the refrigerant system. It allows refrigerant to flow in both directions because of its design, with two check valves that don't allow the filtered particles go again into the system when the refrigerant is flowing in the opposite direction. So it filters in both directions, ideal for our system:

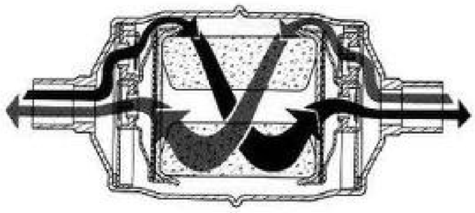


Figure 2.11. Bi-Flow Filter/Dryer flows schema

Accumulator/Receiver: The accumulator protects the compressor from liquid refrigerant in the suction line, which would damage the compressor, as it is designed to compress vapour and not liquid. This is accomplished by use of an inverted trap, shown in the figure. The design produces the stratification of the refrigerant. Saturated vapour, with lower density, moves to the top. Liquid refrigerant is maintained under it, and oil is kept in the bottom. The design has a little hole in the bottom of the inverted trap to suck this oil by a "Venturi Effect", necessary to lubricate the compressor.

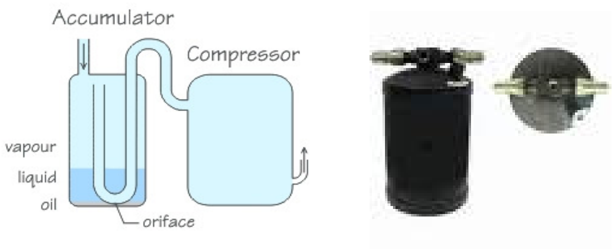


Figure 2.12. Accumulator/Receiver Schema and views

High and Low Pressostats: These are safety devices that turn on and off the heat pump depending on if the pressure before the compressor is too low or too high after it. When values return to normal, they switch the Heat Pump on again and continue to work normally.

Figure 2.13. High and Low Real Pressostat

Typical values for these pressures can be:

-Between 0.1 and 0.5 MPa in the low pressure side of the compressor (this means that minimum is ambient pressure, and usually over it). This pressure is measured just after exiting the evaporator and entering the compressor.

-Between 1.1 and 2.5 Mpa in the high pressure side. This pressure is now measured just after exiting the compressor and entering the condenser.

2.7 Refrigerants

The fluid used for the system is a refrigerant. This is a substance used in a heat cycle usually including, for enhanced efficiency, a reversible phase change from a gas to a liquid.

Ideally, a refrigerant will be non-toxic, non-flammable, operate at modest positive pressures (minimizing piping and component weights, compressor size and leakage into the system). It shall also be easily transportable, recyclable, environmental friendly, inexpensive to produce and compatible to the components of the system.

Traditionally, fluorocarbons, especially chlorofluorocarbons were used as refrigerants, but they were banned because of their ozone depletion effects. Other common refrigerants used in various applications are water, ammonia, sulfur dioxide (both toxic and only used in industrial applications), and non-halogenated hydrocarbons such as methane.

This Heat Pump-AC will use R-134a, which is a blended refrigerant, chlorine-free (so it has low ozone depletion potential). It's tetrafluoroethane (chemical formula CH_2FCF_3). It is used in automobile systems because of its high efficient properties, non-toxic and also because it is not contaminant.

It could be a possibility to use water as it is cheap and not contaminant. Because of the system working at pressures that are over atmospheric, water has a too low Saturation Temperature at Atmospheric Pressure, so it wouldn't work, opposite to R-134a that has a higher Saturation pressure than atmospheric and will flow. And the moat important reason in our system R-134a is used is because the manufacturer designed the compressor we are going to use to work specifically with this refrigerant.

3. PROCEDURE

3.1 Observation and Elements in the AC/Heat Pump System

Since the project is going to be a Heat Pump and AC system, it will be useful to look at an existing AC device bought by GMIT in the Motor Mechanic Workshop. This unit in the workshop only cools, so ours will be more complex and some extra components must be added to the device in order to add the heating capacity. The basic idea is to manufacture a cheaper system that also includes a heat pump in the same device. The elements that will have to be added in order to accomplish this modification will be clearer after the view of this system. Here are some pictures of the system and the elements are tagged:

Figure 3.1. AC Unit Front View

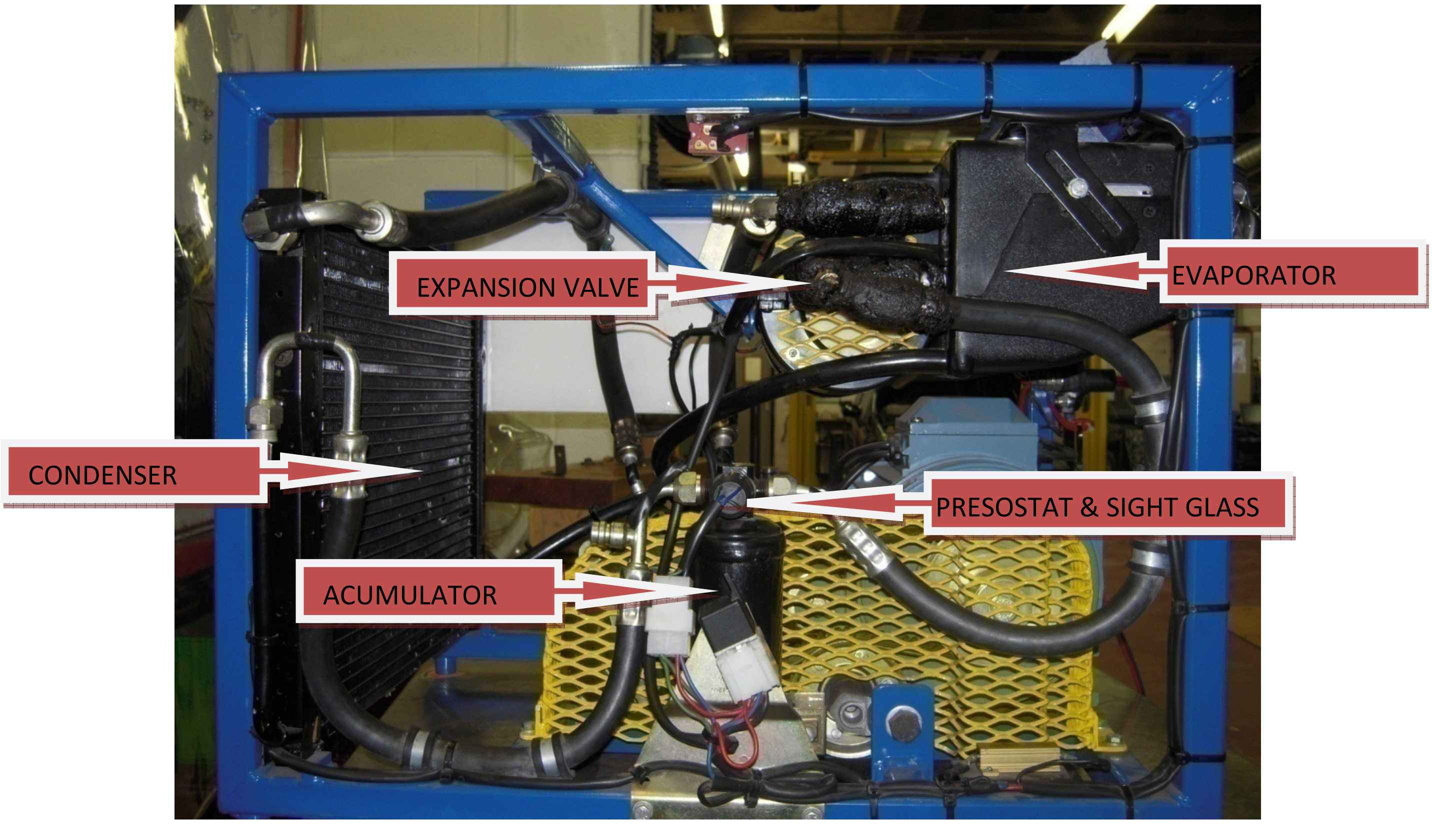


Figure 3.2. AC Unit Side View with elements tagged

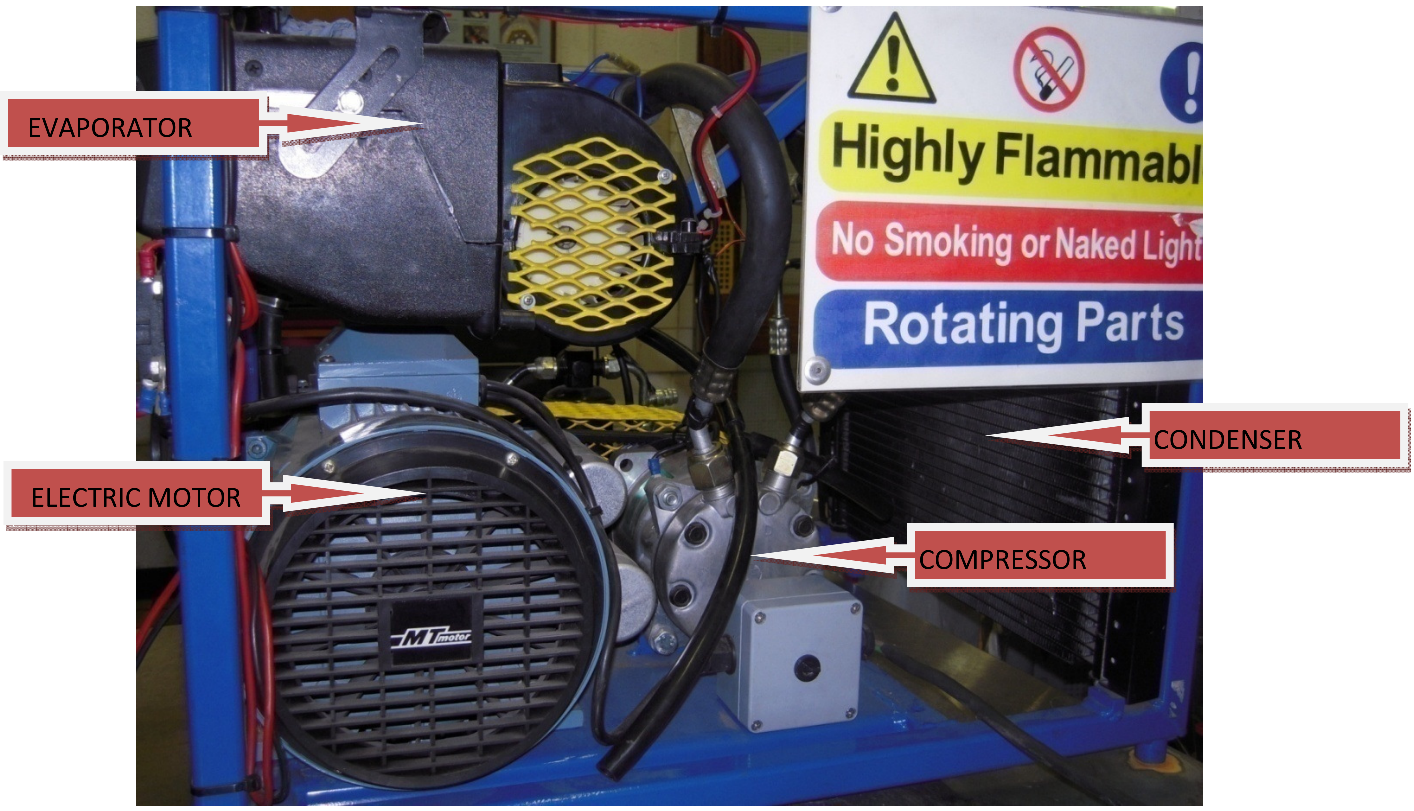


Figure 3.3. AC Unit Opposite Side View with elements tagged

Now that the elements in this previous system are identified, modifications and addition of new necessary elements can start. From the Theory part we can list the basic components:

1) Compressor (x1)
2) Indoor Coil (x1)
3) Outdoor Coil (x1)
4) Throttling Valves (x2) → 1 in previous system, <u>1 EXTRA NEEDED</u>
5) Check Valves (x2) → 1 in previous system, <u>1 EXTRA NEEDED</u>
6) Reversing Valve (x1) → <u>NEW ELEMENT NEEDED</u>
7) Accumulator (x1)
8) Bi-flow Filter/Dryer (x1)
9) Sight Glass/Moisture Indicator (x1)

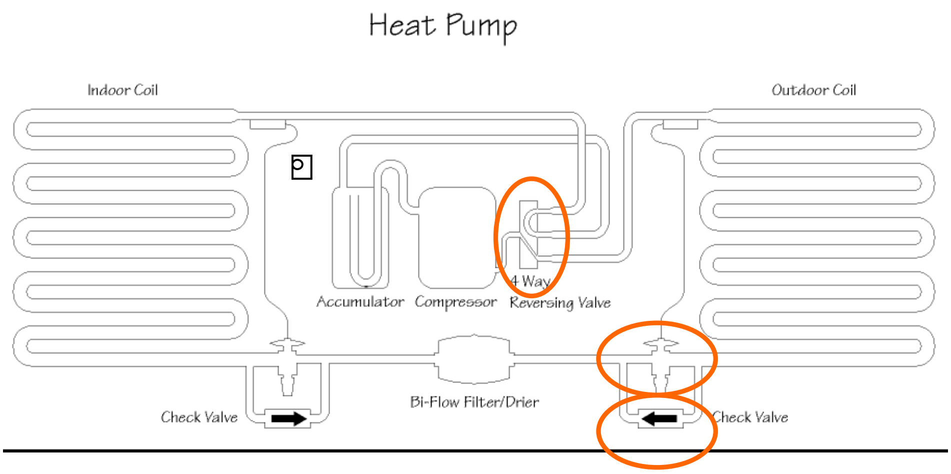


Figure 3.4. Heat Pump/AC system with extra elements tagged

The extra elements that an air conditioning system doesn't include and necessary for the heat pump capacity are circled in the figure above.

3.2 Acquiring of the Compressor and Low & High Pressures

Since the compressor is the key element of the system, the first thing should be acquiring one. In this project the compressor used will be a DELPHI 5106 SD7H15. It was taken from an automobile, an Opel Omega '97.

Since the High and Low pressure levels are the core of the theoretical calculations, the manufacturer book must be consulted. The manufacturer (Delphi) advices to have a high and low level pressures of <u>16.7 bar on the high side and 2.96 bar on the low side</u>. It also specifies a <u>10K superheating</u> is also necessary, and refrigerant used is <u>R-134a</u>. This info can be observed in the compressor performance graph, included in following pages.

3.3 Calculation of Ideal-Vapour Compression Cycle

In order to describe how the new system will behave, the first approach has to be the calculation of the ideal cycle that will rule the Heat Pump/AC system.

For this Ideal Cycle the only variables that describe it are the High and Low Pressures. These are taken from the advised by the manufacturer. The calculations are done with 16.7 bar(1.67Mpa) on the high side and 2.96 bar(0.296Mpa). Now that pressures are defined, the cycle can be represented in a Pressure-Enthalpy Diagram for the refrigerant in use, R-134a in this case:

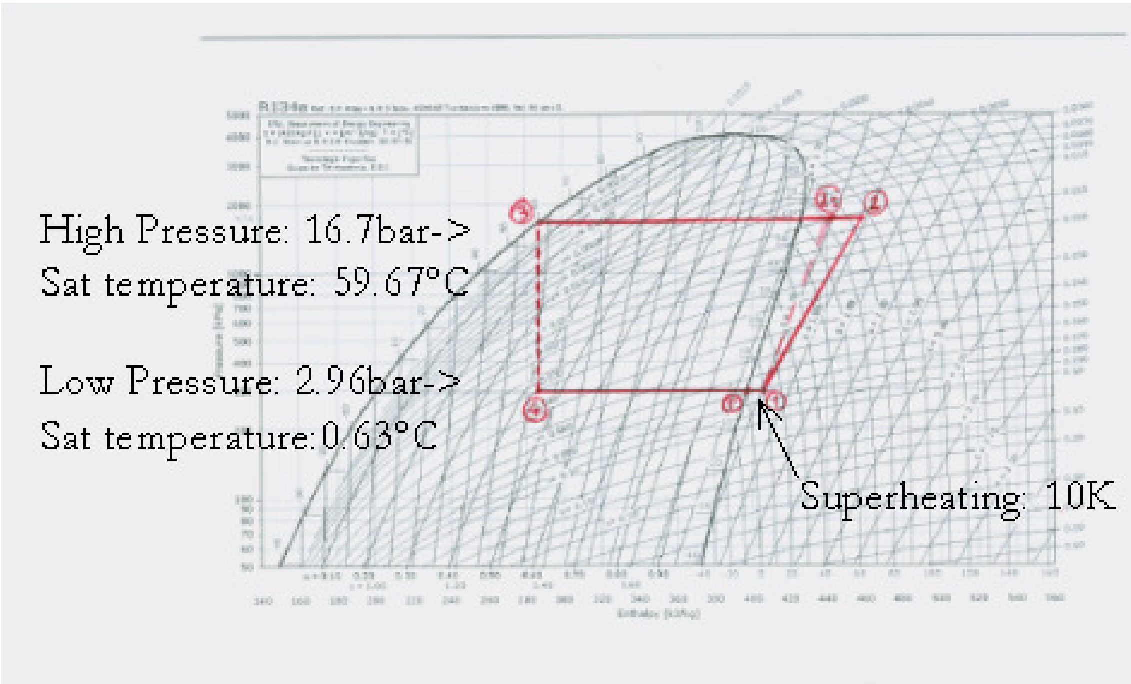


Figure 3.5. Pressure-Enthalpy Actual Refrigeration Diagram for R-134a

The previous diagram describes the 4 processes the refrigerant will go through in the refrigerating cycle:

1-2 → Compressor (1-2s ideal compressor, in discontinous line)

2-3 → Condensor

3-4 → Throttling Valve

4-1 →Evaporator (1'-1 is the superheating)

Next, Pressure and Enthalpy of each point must be calculated. To accomplish this we will use charts and properties we know about each point in advance:

Point 1' → $X_{1'} = 1$ (Saturated gas) ; $P_{1'} = P_4 = 2.96bar$

Point 1 → $P_1 = P_{1'} = P_4 = 2.96bar$; $T_1 = T_{1'} + 10K$

Point 2s → $P_{2s} = P_3 = 16.7bar$; $s_1 = s_{2s}$ (Isentropic process)

Point 2 → $P_2 = P_{2s} = P_3 = 16.7bar$ → to calculate this point first isentropic efficiency of the compressor is required

Point 3 → $X_3 = 0$ (saturated liquid) ; $P_{2s} = P_2 = P_3 = 16.7bar$

Point 4 → $h_3 = h_4$ (isenthalpic process) ; $P_1 = P_4 = 2.96bar$; $T_{1'} = T_4$

CALCULATIONS:

Knowing $P_3 = 16.7bar$ and $X_3 = 0$,we go to tables for saturated R-134a, and linear interpolate to extract values:

P(bar)	T (^oC)	h(kJ/kg)
16	57.92	134.02
16.7	T_3	h_3
18	62.91	142.22

h_3 **= 136.89 kJ/kg ,** T_3 **= 59.67 ^oC = 332.82 K**

Knowing $P_1 = P_{1'} = P_4 = 2.96bar$ and $X_{1'} = 1$, we go to tables for saturated R-134a, and linear interpolate to extract values:

$h_{1'}$ **= 247.59 kJ/kg ,** $T_{1'}$ **= 0.63 ^oC = 273.78 K**

Knowing $P_1 = P_{1'} = P_4 = 2.96bar$ and $T_1 = T_{1'} + 10K = 10.63$ ^{o}C, we go to tables for saturated R-134a, and linear interpolate to extract values:

T_1 **= 10.63 c = 283.78 K ,** h_1 **= 256.21 kJ/kg ,** s_1 **= 0.9497 kJ/kg K ,**
v_1 **=0.0722m^3 /kg**

Knowing $s_1 = s_{2s} = 0.9497 kJ / kgK$ and $P_{2s} = P_2 = P_3 = 16.7 bar$, we go to tables for saturated R-134a, and linear interpolate to extract values:

h_{2s} = **292.33 kJ/kg**

Knowing $P_{1'} = P_4 = 2.96 bar$, $T_{1'} = T_4$ and $h_3 = h_4$,we calculate Point 4:

h_4 **= 136.89 kJ/kg ,** T_4 **= 0.63 ^{o}C = 273.78 K**

Now all 5 points are defined, the Diagram can be completed and COPs can now be calculated (expected efficiencies).

3.4 Calculation of Carnot COP and Actual COPs

Now all 4 points are defined, COPs will be calculated. COPs indicate the efficiency of refrigeration devices, by indicating what factor multiplies the work put into the system to obtain a certain quantity of heat (extracted or rejected, depending on if the device works as a Heat Pump or a refrigerator).

For example, if $COP_{HP} = 3$ and $COP_R = 2$, for every 10kW inverted in the system as Work in the compressor, the device will reject 3x10 = 30kW in Heat Pump Mode and 2x10 = 20kW in Refrigerator Mode.

Moreover, as explained in the Theory, Actual COPs can never be greater than Carnot's COPs, as this last values are the highest possible between two fixed pressure values.

Actual COPs

Heat Pump	Refrigerator
$COP_{HP} = \frac{Q_A}{W_{NET}}$ $COP_{HP} = \frac{Q_A}{Q_A - Q_R} =$ $= \frac{h_2 - h_3}{h_2 - h_1} = \frac{292.33 - 136.89}{292.33 - 256.21} =$ **4.30**	$COP_R = \frac{Q_R}{W_{NET}}$ $COP_R = \frac{Q_R}{Q_A - Q_R} =$ $= \frac{h_1 - h_4}{h_2 - h_1} = \frac{256.21 - 136.89}{292.33 - 256.21} =$ **3.30**

Carnot COPs:

Carnot Heat Pump	Carnot Refrigerator
$COP_{HP,Carnot} = \frac{T_H}{T_H - T_L} =$ $COP_{HP,Carnot} = \frac{T_3}{T_3 - T_4} =$ $COP_{HP,Carnot} =$ **5.64**	$COP_{R,Carnot} = \frac{T}{T_H - T_L} =$ $COP_{R,Carnot} = \frac{T_4}{T_3 - T_4} =$ $COP_{R,Carnot} =$ **4.64**

From these values a conclusion can be extracted: calculations are correct because of the relation between COPs is maintained in both cases ($COP_{HP} = COP_R + 1$) and Carnot COPs are higher than the Actual ones. If this wasn't accomplished, calculations would be wrong for sure.

3.5 Defining the Motor to drive the Compressor

In order to choose the right motor to drive the compressor, the first step is looking at the charts the manufacturer (Delphi) provides, to check the performance of this compressor, this is, efficiencies and power consumptions.

Figure 3.1 is a chart where these values can be extracted from:

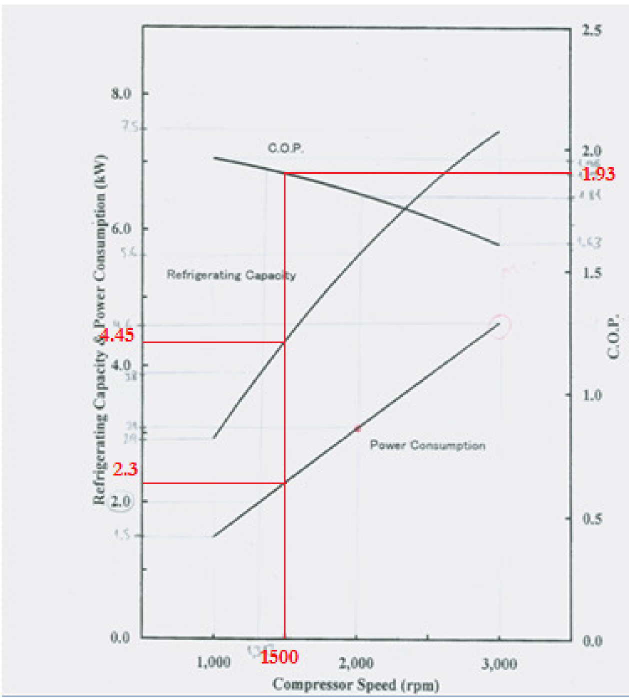


Figure 3.6. Compressor DELPHI 5106 SD7H15 Performance

Observing the chart, the COP (which is the variable we want to keep high) goes down with high revolutions of the motor, as power consumption rises. This is non-efficient, so the idea is to keep the revs as low as possible, to also minimize power consumption and keep COP as high as possible.

To have an idea of what the chart sais, the following values at minimum, mid and top revolutions are taken from the graph:

Compressor Speed: 1000rpm (min)

-Power consumption**: 1.5 kW** -Refrigerating Capacity**: 2.9 kW**

-COP**: 1.96 →HIGHEST**

Compressor Speed: 2000rpm (mid)

-Power consumption**: 3.1 kW** -Refrigerating Capacity**: 5.6 kW**

-COP**: 1.81 → COP DESCENDS**

Compressor Speed: 3000rpm (max)

-Power consumption**: 4.6 kW** -Refrigerating Capacity**: 7.5 kW**

-COP**: 1.63 → LOWEST**

This decrease in COP and increase in power consumption is due to the compressing cycles inside the compressor at high revolutions try to pump much more refrigerant at once, increasing resistance from it.

So keeping the motor as small as possible is the objective. In this case a 4-pole AC motor will be installed because of its low cost, as we need to keep the investment at minimum. And also because of it's low revs, as we want to keep COPs as high as possible. Since this kind of motor has a fixed rotation speed of 1500 rpm, we calculate values at this speed:

Compressor Speed: 1500rpm (CHOSEN MOTOR)

-Power consumption**: 2.3 kW**

-Refrigerating Capacity**: 4.45 kW**

-COP**: 1.93**

With these values we can now calculate new needed parameters, as the refrigerant mass flow, isentropic efficiency of the compressor, compression ratio and continue on with the heat transfer part of the project.

3.6 Refrigerant Mass Flow

One very important parameter needed is the refrigerant mass flow. This is how much R134-a is circulating through the device per unit time. This parameter is needed in order to calculate powers, as all the energy units from the cycle previously calculated are per unit mass (kJ/kg). Mass flow is expressed in kg/s , so it expresses how much mass is circulating per second. To obtain this value, we are going to focus in the manufacturer data and our calculations.

Since we know how much heat is extracted from the cold reservoir in the evaporator from the compressor performance (this is the Refrigeration Capacity of 4.45kW) and we also know the enthalpies at the entrance and exit of the evaporator:

$$\dot{Q}_{evap} = \dot{m}(h_4 - h_1)$$

$$\dot{m} = \frac{\dot{Q}_{evap}}{h_4 - h_1} = \frac{4.45kW}{256.21 - 136.89kJ/kg}$$

$\dot{m}$ **= 0.0373 kg/s = 37.3 g/s**

3.7 Isentropic Efficiency of the Compressor

Now that refrigerant mass flow is calculated, we are going to proceed to calculate the isentropic efficiency of the compressor. This dimensionless parameter establishes how good is our real compressor related to an ideal reversible one. It can be calculated extracting work values from a P-v diagram like this:

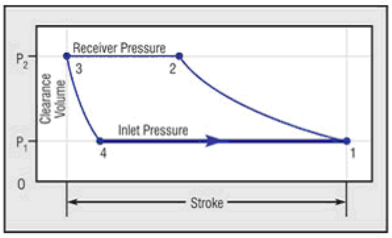


Figure 3.7. P-v Compressor Diagram

The problem is that is this case obtaining values for these calculations wasn't possible, so an alternative method is going to be used. Based on manufacturer's data and our calculations of an ideal compressor (1-2s in the diagram) we calculate the work inputs ratio.

Our manufacturer Sanden Compressors establishes that for our 1500rpm motor the compressor consumes 2.3 kW. This is the total power consumption, so to know how much work is really going into the refrigerant, we subtract an average 10 percent of mechanical losses. So the compressing power consumption is:

$$\dot{W}_{comp,real} = \dot{W}_{comp,manufacturer} - 10\% \text{ mechanical losses} = 0.9 \cdot \dot{W}_{comp,manufacturer}$$

$$\dot{W}_{comp\ ,real} = 0.9 \cdot 2.3kW = 2.07\ kW$$

So only 2.07 kW are really invested into the refrigerant from the 2.3 kW consumed. Next we calculate the work consumed by the ideal reversible compressor working between the same pressure differences from the enthalpy difference and the refrigerant mass flow calculated previously:

$$\dot{W}_{comp,rev} = \dot{m}(h_{2s} - h_1)$$

$$\dot{W}_{comp,rev} = 0.0373 \cdot (292.33 - 256.21) = 1.347kW$$

This means that an ideal reversible compressor would only need 1.347 kW to compress the refrigerant between the same pressure difference than our real one. With these values we proceed to calculate the isentropic efficiency (η_s) of the compressor by calculating the ratio between the ideal and real work inputs:

$$\eta_s = \frac{\dot{W}_{comp,rev}}{\dot{W}_{comp,real}} = \frac{1.347kW}{2.07kW}$$

η_s **= 0.6507 = 65.07%**

From this calculations we extract the conclusion that with 65.07% of the real work consumed by the compressor an ideal one would accomplish the same task. So isentropic efficiency is quite low, but as expected from these kind of reciprocating compressor (around 60%).

3.8 Final point of the diagram (Point 2) and Compression Ratio

Obtaining the isentropic efficiency of the compressor was not only important to know how good it was accomplishing its task, but also to complete the cycle diagram. The real point where the refrigerant exits the compressor (not the ideal one, Point 2s) will conclude defining the cycle. So now we proceed to calculate Point 2 in the diagram.

From the definition of isentropic efficiency, we can write:

$$\eta_s = \frac{W_{comp,rev}}{W_{comp,real}}$$

If we now substitute work inputs by enthalpy differences:

$$\eta_s = \frac{h_{2s} - h_1}{h_2 - h_1}$$

And extract h_2 from this formula:

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_s}$$

$$h_2 = 256.21 + \frac{292.33 - 256.21}{0.6507}$$ = **311.72 kJ/kg**

Now that we know enthalpy in Point 2(h_2 = 311.72 kJ/kg) and pressure ($P_{2s} = P_2 = P_3 = 16.7bar$), we go to tables for saturated R-134a, and linear interpolate to extract values:

T_2 **= 86.7ºC ,** v_2 **= 0.0141 m ^{3} /kg**

With these last calculations, the whole cycle is defined, and Points 1, 1', 2s, 2, 3 and 4 are completely defined, so the diagram is complete.

Next step will be calculate the compression ratio, this is, how many times smaller is the same mass of the high pressure compressed refrigerant exiting the compressor in comparison to the low pressure refrigerant entering it. These compressors have a maximum compression ratio of 6. This means that for security reasons a constant value mass of refrigerant exiting the compressor must be 6 times smaller that the one entering, as maximum. So now we proceed to calculate this ratio (specific volume entering compressor over specific volume exiting):

$$\text{Compression Ratio} = \frac{v_1}{v_2} = \frac{0.0722 m^3/kg}{0.0141 m^3/kg}$$

v_2 **= Compression Ratio = 5.12**

Analyzing this result, we can conclude that our compressor is operating securely, as it doesn't exceed the compression ratio established.

3.9 Condenser and Evaporator Air Mass Flow Rate

Now that the thermodynamic diagram is completely defined, next step is to calculate how much air do condenser and evaporator need in order to exchange the calculated heat. To achieve this, an Energy Balance from 1st Law of Thermodynamics will be done on both condenser and evaporator.

$$\dot{Q} - \dot{W} = \sum_{IN} \dot{m}\left(h + \frac{V^2}{2} + gz\right)_{IN} - \sum_{OUT} \dot{m}\left(h + \frac{V^2}{2} + gz\right)_{OUT} = \Delta U_{sist}$$

And from here we can assume:

-No kinetic energy variations ($\frac{V^2}{2}$ = 0)

-No potential energy variations (*gz* = 0)

-Stationary system, there is no work/energy accumulated in the system (ΔU_{sist} = 0)

After using the assumptions, the balance is:

$$\sum_{IN} \dot{m}\, h_{IN} = \sum_{OUT} \dot{m}\, h_{OUT}$$

And substituting from what we know:

$$\dot{m}_{R134a}(\Delta h_{R134a}) = \dot{m}_{air} \cdot c_{air} \cdot \Delta T_{air}$$

With this last equation we calculate Air Mass Flow Rates ($\dot{m}_{air}$) for both condenser and evaporator

$$\dot{m}_{air} = \frac{\dot{m}_{R134a}(\Delta h_{R134a})}{c_{air} \cdot \Delta T_{air}}$$

-specific heat of air (c_{air})= 1.006 kJ/kgK

<u>**CONDENSER:**</u> The air temperatures entering and exiting the condenser are now needed. This component is the one that rejects heat, heating the space required. We take the worst case scenario of winter, where Met Eireann says that average temperature in Ireland's winter is about 8 ^oC, and we take an even lower temperature to ensure or calculations. This ambient air temperature is going to be taken as 5 ^oC. The exiting temperature we want will be of 45 ^oC to heat the space fast. So with all data we substitute in the equation:

$$\dot{m}_{air,cond} = \frac{\dot{m}_{R134a}(h_2 - h_3)}{c_{air} \cdot \Delta T_{air}} = \frac{0.0373(311.72 - 136.89)}{1.006(45-5)} =$$

$\dot{m}_{air,cond}$ **= 0.162 kg/s**

EVAPORATOR: The air temperatures entering and exiting the evaporator are also needed. This component is the one that absorbs heat, cooling the space desired. We take the worst case scenario of summer, where Met Eireann says that average temperature in Ireland's summer is about 18-20 °C, and we take an even higher temperature to ensure or calculations. This ambient air temperature is going to be taken as 25 °C. The exiting temperature we want will be of 15 °C to cool the space fast. So with all data we substitute in the equation:

$$\dot{m}_{air,evap} = \frac{\dot{m}_{R134a}(h_1 - h_4)}{c_{air} \cdot \Delta T_{air}} = \frac{0.0373(256.21 - 136.89)}{1.006(25 - 15)} =$$

$\dot{m}_{air,evap}$ **= 0.442 kg/s**

3.10 Condenser and Evaporator Heat Transfer Calculations

In this part of the project we are going to focus in both heat exchangers: evaporator and condenser. The aim is to find how good are they in developing their task of absorb/reject heat respectively, what rates of heat transfer will they have and how big are they going to be.

To describe how good are they, we will calculate the effectiveness of the exchangers. This is defined as the ratio between the maximum temperature change possible in this exchanger and the real one. Rates of heat transfer will be described with U-vales, which describe how well is the heat transferred. And to see how big are we going to design the exchangers we will look at commercial catalogs, in order to pick one directly from the shelf.

Both condenser and evaporator are going to be designed as cross-flow unmixed fluids heat exchangers, where one fluid is R134-a refrigerant and the other one is air. Next figure is a schematic explanation of both fluids movement. In the condenser refrigerant comes in hot and cools down, affecting the air around it, that comes in cold and heats up. In the evaporator the opposite process happens, where refrigerant comes in cold and heats up, affecting the air around it, that comes in hot and cools down.

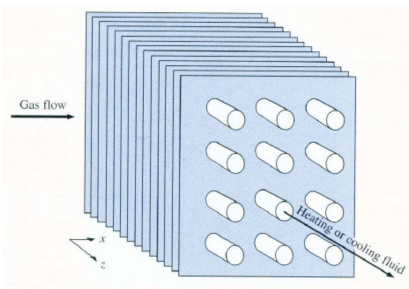


Figure 3.8. Cross-Flow heat exchanger (Refrigerant Fluid and Gas Flows)

U-value and area of the condenser are related by the following equation:

$$A_{cond/evap} = \frac{\dot{Q}_{out/in}}{U \cdot F \cdot \Delta T_m}$$

Where $A_{cond/evap}$ is the area of the condenser/evaporator expressed in m^2, $\dot{Q}_{out/in}$ is the heat rejected/absorbed by the refrigerant in kW, U is the U-value explained above expressed in kW/m^2 K, F is a correction factor obtained from the graph shown underneath, and ΔT_m is the Log Mean Temperature Difference.

The reason why instead of using simply the temperature difference between entrance and exit is that heat exchangers like this don't exchange heat isothermally, as both refrigerant and air are constantly changing temperatures. So by using the Log Mean Temperature Difference we take account of this factor.

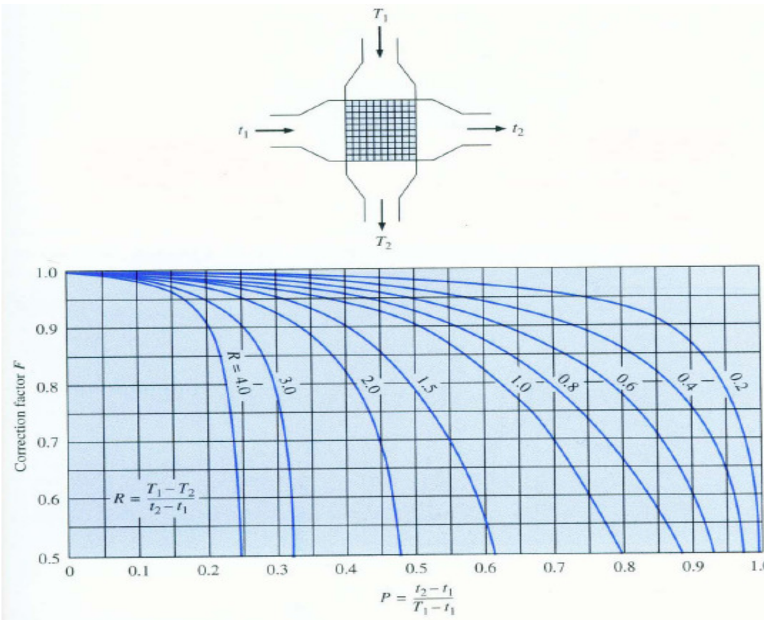


Figure 3.9. Correction Factor (F) Plots for single pass counter-flow heat exchanger, both fluids unmixed

CONDENSER

First we calculate how much heat is going to be rejected by the condenser in our design. This is calculated by multiplying refrigerant mass flow by enthalpy difference between entering and exiting the exchanger:

$$\dot{Q}_{out} = \dot{m}_{R134a}(h_2 - h_3) = 0.0373(311.72 - 136.89)$$

$\dot{Q}_{out}$ **= 6.52 kW**

This means that refrigerant rejects 6.52 kW of heating power in our system in order to heat the intended space.

Once this value is calculated we continue towards effectiveness, U-value and condenser area. First thing is drawing a sketch of the heat exchanger, with inlet and outlet temperatures. Air temperatures are the same taken when calculating air mass flows. Why these temperatures are taken was explained before. And refrigerant R134-a temperatures are taken from the P-h diagram:

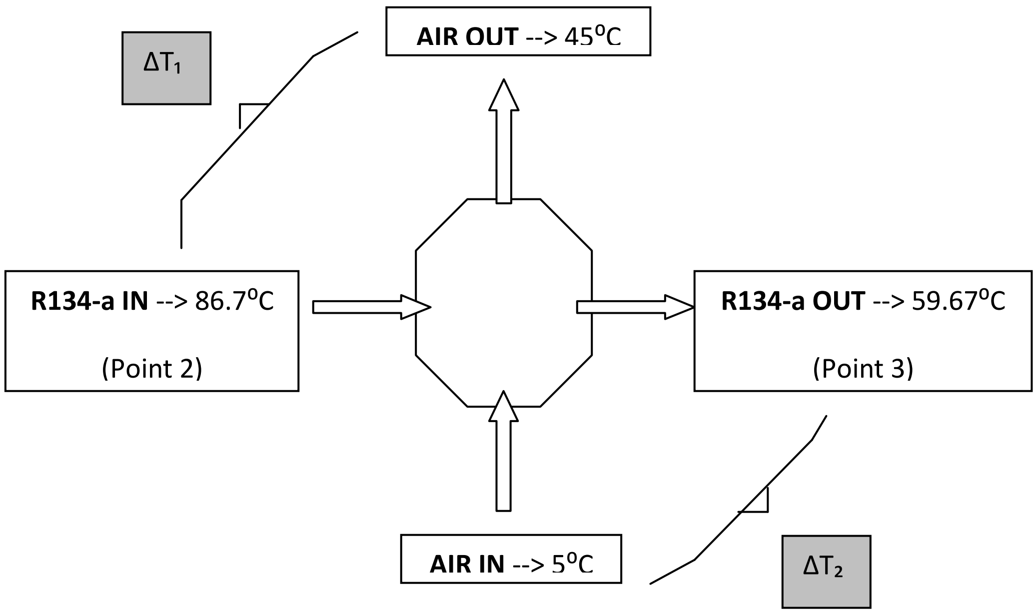


Figure 3.10. Condenser Inlet and Outlet Temperatures Sketch

This is the objective: Air will come in the condenser at 5°C (278.15K) and has to exit it at 45°C (318.15K). This will be achieved by the cooling of the refrigerant, that will enter the condenser at Point 2 at 86.7°C (359.85K) and exit it at Point 3 at 59.37°C (332.82K).

Now log mean temperature difference is calculated. By definition,

$$\Delta T_m = \frac{\Delta T_2 - \Delta T_1}{\ln(\frac{\Delta T_2}{\Delta T_1})}$$

where ΔT_1 is the change in high temperature for the refrigerant and the air, and ΔT_2 is the change in low temperatures for the refrigerant and the air, as shown in the sketch above. Hence,

$$\Delta T_m = \frac{(59.67-5)-(86.7-45)}{\ln(\frac{59.67-5}{86.7-45})} = \mathbf{47.89\ K}$$

Next step is calculating correction factor (F). Looking at the plots, we need to calculate 2 parameters to obtain our F value, P and R.

$$P = \frac{T_{air.high} - T_{air.low}}{T_{R134a.high} - T_{air.low}} = \frac{45-5}{86.7-5} = 0.49$$
$$R = \frac{T_{R134a.high} - T_{R134a.low}}{T_{air.high} - T_{air.low}} = \frac{86.7-59.67}{45-5} = 0.68$$

With P and R, correction factor F is approximately 0.94.

Once the variables are calculated, we go back to the initial equation and substitute:

$$A_{cond} = \frac{\dot{Q}_{out}}{U \cdot F \cdot \Delta T_m} = \frac{6.52kW}{U \cdot 0.94 \cdot 47.89K}$$

$$A_{cond} = \frac{0.1448kW / K}{U}$$

So now U-value and condenser area are related. In order to design the condenser we have two options: state a normalized condenser area or a U-value. In this case the area is going to be picked. The condenser is going to be a normalized 20x30 cm. So the condenser area is:

$$A_{cond} = 0.2m \cdot 0.3m =$$

A_{cond} **= 0.06m** ^{2}

And with the area already stated, we proceed to calculate U-value of the condenser:

$$U_{cond} = \frac{0.1448kW / K}{A_{cond}} =$$

U_{cond} **= 2.413 kW/m** ^{2} **K**

With the temperature diagram we can also define the effectiveness that will be obtained with this exchanger temperatures. Effectiveness (ε) describes how much temperature change is really achieved respect to the maximum possible:

$$\varepsilon_{cond} = \frac{\Delta T \ of \ fluid \ with \ l\arg er \ temperature \ change}{Difference \ between \ inlet \ temperatures} = \frac{45-5}{86.7-5} =$$

ε_{cond} **= 0.4896 = 48.96%**

EVAPORATOR

The same steps as with the condenser are going to be followed. The heat that is going to be absorbed by the evaporator in our design was told by the manufacturer.

$$\dot{Q}_{in} = \mathbf{4.45\ kW}$$

This means that refrigerant absorbs 4.45 kW of cooling power to our system in order to cool the intended space.

Once this value is calculated we continue towards effectiveness, U-value and evaporator area. Again first thing is drawing a sketch of the heat exchanger, with inlet and outlet temperatures. Air temperatures are the same taken when calculating air mass flows. Why these temperatures are taken was explained before. And refrigerant R134-a temperatures are taken from the P-h diagram:

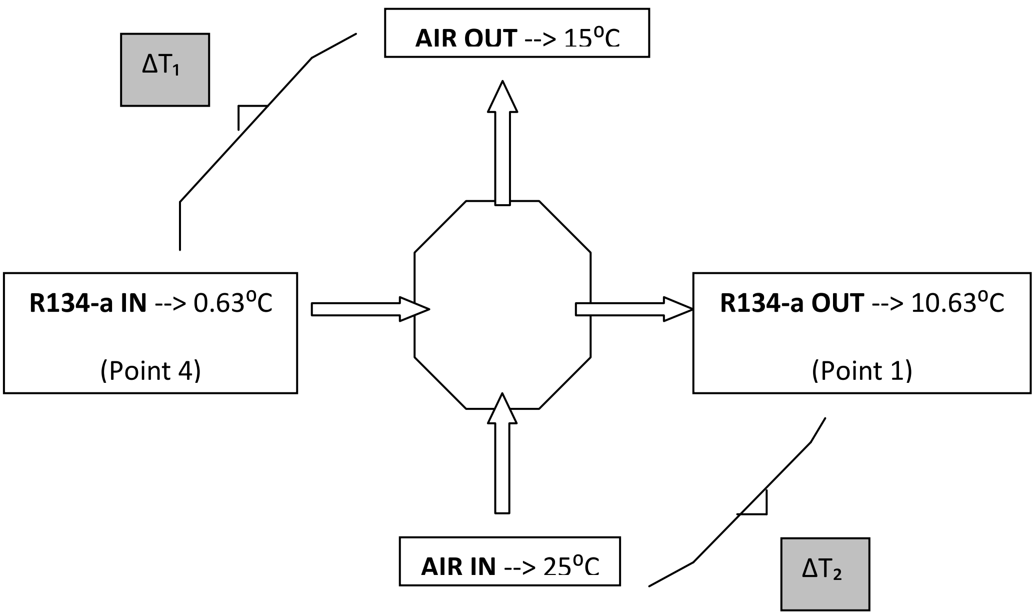


Figure 3.11. Evaporator Inlet and Outlet Temperatures Sketch

This is the objective: Air will come in the evaporator at 25°C (298.15K) and has to exit it at 15°C (288.15K). This will be achieved by the heating of the refrigerant, that will enter the Evaporator at Point 4 at 0.63°C (359.85K) and exit it at Point 1 at 10.63°C (332.82K).

Now log mean temperature difference is calculated. By definition,

$$\Delta T_m = \frac{\Delta T_2 - \Delta T_1}{\ln(\frac{\Delta T_2}{\Delta T_1})}$$

where ΔT_1 is the change in high temperature for the refrigerant and the air, and ΔT_2 is the change in low temperatures for the refrigerant and the air, as shown in the sketch above. But in this case air and refrigerant have the same temperature change (10ºC). When this occurs, the log mean temperature difference can be taken as an average of ΔT_1 and ΔT_2 .Hence,

$$\Delta T_m = \frac{\Delta T_2 + \Delta T_1}{2} = \frac{(15-0.63)+(25-10.63)}{2}$$ = **14.37 K**

And since the process in the evaporator is almost completely inside the dome, it occurs at constant temperature and pressure. Because of this correction factor F can be taken as 1.

Once the variables are calculated, we go back to the initial equation and substitute:

$$A_{evap} = \frac{\dot{Q}_{in}}{U \cdot F \cdot \Delta T_m} = \frac{4.45kW}{U \cdot 1 \cdot 14.37K}$$

$$A_{cond} = \frac{0.310kW/K}{U}$$

So now U-value and condenser area are related. In order to design the evaporator we have two options: state a normalized condenser area or a U-value. In this case the area is going to be picked. To make the design cheaper, the evaporator is going to be the same as the condenser, a normalized <u>20x30 cm</u>. So the evaporator area is again:

A_{evap} = **0.06m** ^{2}

And with the area already stated, we proceed to calculate U-value of the evaporator:

$$U_{evap} = \frac{0.310kW/K}{A_{evap}} =$$

U_{evap} = **5.167 kW/m** ^{2} **K**

With the temperature diagram we can also define the effectiveness that will be obtained with this exchanger temperatures. Effectiveness (ε) describes how much temperature change is really achieved respect to the maximum possible:

$$\varepsilon_{evap} = \frac{\Delta T\ of\ \ fluid\ \ with\ l\arg er\ temperature\ change}{Difference\ between\ inlet\ temperatures} = \frac{25-15}{25-0.63} =$$

ε_{evap} = **0.4103 = 41.03%**

The following table resumes results obtained in this part:

	CONDENSER	***EVAPORATOR***
Dimensions Exchanger	**20x30 cm**	**20x30 cm**
U-value	**2.413 kW/m^2 K**	**5.167 kW/m^2 K**
Effectiveness	**48.96%**	**41.03%**

3.11 Tubing Calculations

After having calculated these heat transfer parameters for the exchangers, the tubing is going to be designed. We log on to a manufacturer's webpage and choose one. In this case the tubing was taken from Lawton Tubes. In order to choose the correct diameter tubes the manufacturer provides an equation for the thickness of the tubing, critical in order to have a strong enough design:

$$t = \frac{p \cdot d}{p + 20F}$$

where t is the wall thickness in mm, p is the design pressure in bar, d is the outside diameter of the tube in mm and F is the design stress in N/mm^2.

The tubing is going to be made of copper, with an outside diameter for both high and low pressure sides of 6mm, the smallest this manufacturer provides. This diameter is taken to make the design as cheap as possible.

From manufacturer's charts, for our high pressure of 16.7 bar it states a design stress (F) of 40 N/mm^2, and for our low pressure side of 2.96 bar it states a design stress (F) of 41 N/mm^2.

With this data we proceed to calculate the tube thickness:

-HIGH PRESSURE

$$t = \frac{p \cdot d}{p + 20F} = \frac{16.7 \cdot 6}{16.7 + 20 \cdot 40} = 0.1226mm$$

-LOW PRESSURE

$$t = \frac{p \cdot d}{p + 20F} = \frac{2.96 \cdot 6}{2.96 + 20 \cdot 41} = 0.0215mm$$

From these thicknesses we take the closest the manufacturer provides and suits our needs. This wall thickness will be 0.8mm. The next sketch is a view of the cross-sectional of our 6mm copper tube:

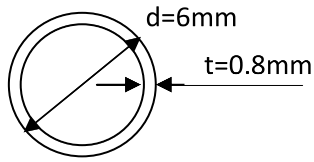


Figure 3.12. Copper Tubing Cross-Section

3.12 Heat transfer coefficients for air in Condenser and Evaporator

Now that the tubing is designed, we can continue to calculate the heat transfer coefficients (h) that rule the heat exchange between condenser and evaporator with air. This will give us an idea of the values expected for how good is this heat transfer between the tubing and ambient air, something like U-values explained before.

Both condenser and evaporator are going to be crossed-flow unmixed fluids heat exchangers, and to calculate h for them we will take the exchangers as a bank of staggered pipes. Heat transfer coefficients will be calculated then for flow of air over staggered copper pipes.

CONDENSER

For the condenser air will come in at 5°C (T_∞) at a mass flow rate of 0.162 kg/s. The temperature of the refrigerant in the condenser will be assumed the same as the outside of the copper tubing in contact with the air. This temperature will be taken as isothermal as if all the process occurred inside the dome, although it's not completely correct, but it will be a good approximation. Then wall temperature of the copper will be T_w = 59.67°C. With all this data first we draw a sketch of the situation:

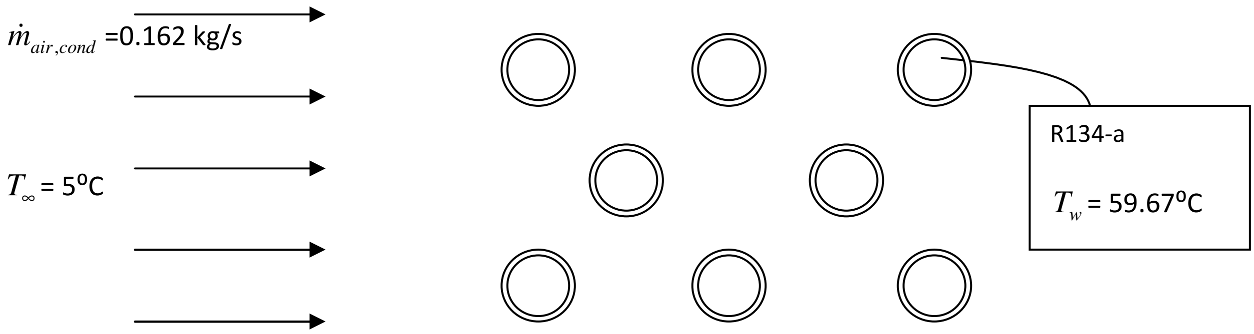


Figure 3.13. Condenser Modelled as Air Flow over Staggered Pipes

With this clear, calculations can be done. From empirical correlations it can be estimated that for air flow over staggered banks of pipes:

$$Nu = 0.33 \cdot C_h \cdot \text{Re}^{0.6}$$

where Nu is Nusselt Number, a non-dimensional number proportional to h, C_h is a correction factor equal to 1.1 for staggered pipes, and Re is Reynolds Number, another non-dimensional number that takes account of the kinetic factor affecting the fluid.

The process to calculate h starts with calculating Film Temperature (T_f). This is the average temperature between the fluid and the wall in contact. The objective of taking this temperature instead of taking the fluid nor the wall temperature is to have a unique temperature to calculate properties of the fluid. Hence:

$$T_f = \frac{T_w + T_\infty}{2} = \frac{59.67 + 5}{2} = 32.335°\text{C} = 305.485\text{K}$$

With this film temperature we can now calculate properties of the fluid, necessary for Reynolds Number. Properties are taking linear interpolating in air properties at ambient pressure:

$$\nu = 16.25 \cdot 10^{-6} m^2 / s \;\; ; \;\; k = 0.02666 W / mK \;\; ; \;\; \text{Pr} = 0.707$$

Since we also need air velocity instead of mass flow for Reynolds number, we calculate V:

$$V = \frac{\dot{m}_{air,cond}}{\rho_{air \;\; at \; T=5°C} \cdot A} = \frac{0.162 kg / s}{1.2803 kg / m^3 \cdot 0.06 m^2} = 2.11 m / s$$

Reynolds number can now be calculated. By definition, Re number is:

$$\text{Re}_D = \frac{V \cdot D}{\nu} = \frac{2.11 m / s \cdot 0.006 m}{16.25 \cdot 10^{-6} m^2 / s} = 779.08$$

With these last calculations we can enter the values to the initial correlation:

$$Nu = 0.33 \cdot C_h \cdot \mathrm{Re}^{0.6} = 0.33 \cdot 1.1 \cdot 779.08^{0.6} = 19.718$$

And from this we can finally extract h from Nusselt number, because:

$$h = \frac{k \cdot Nu}{D} = \frac{0.02666 \cdot 19.718}{0.006} =$$

$$\boxed{h_{air,cond} = \mathbf{86.61\ W/m^2\ K}}$$

This value of h is logical, as for forced convection h can range between 50 and 250 W/m ^{2} K

EVAPORATOR

The same procedure as before will be used for the evaporator. Air will come in at 25°C (T_∞) at a mass flow rate of 0.442 kg/s. The temperature of the refrigerant in the evaporator will be assumed the same as the outside of the copper tubing in contact with the air. This temperature will be taken as isothermal as if all the process occurred inside the dome, although it's not completely correct, but it will be a good approximation. Then wall temperature of the copper will be T_w = 0.63°C. With all this data first we draw a sketch of the situation:

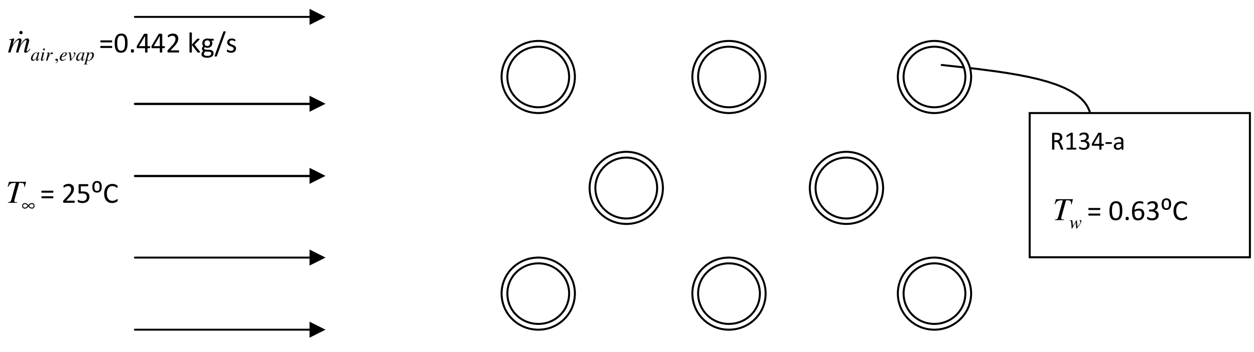


Figure 3.14. Evaporator Modelled as Air Flow over Staggered Pipes

With this clear, calculations can be done. From empirical correlations it can be estimated that for air flow over staggered banks of pipes:

$$Nu = 0.33 \cdot C_h \cdot \mathrm{Re}^{0.6}$$

where Nu is Nusselt Number, a non-dimensional number proportional to h, C_h is a correction factor equal to 1.1 for staggered pipes, and Re is Reynolds Number, another non-dimensional number that takes account of the kinetic factor affecting the fluid.

The process to calculate h starts with calculating Film Temperature (T_f) . This is the average temperature between the fluid and the wall in contact. The objective of taking this temperature instead of taking the fluid nor the wall temperature is to have a unique temperature to calculate properties of the fluid. Hence:

$$T_f = \frac{T_w + T_\infty}{2} = \frac{0.63 + 25}{2} = 12.815°\mathrm{C} = 285.965\mathrm{K}$$

With this film temperature we can now calculate properties of the fluid, necessary for Reynolds Number. Properties are taking linear interpolating in air properties at ambient pressure:

$$\nu = 14.46 \cdot 10^{-6} m^2 / s \;\; ; \;\; k = 0.02513 W / mK \;\; ; \;\; \mathrm{Pr} = 0.712$$

Since we also need air velocity instead of mass flow for Reynolds number, we calculate V:

$$V = \frac{\dot{m}_{air,evap}}{\rho_{air \;\; at\, T=25°\mathrm{C}} \cdot A} = \frac{0.442 kg / s}{1.1861 kg / m^3 \cdot 0.06 m^2} = 6.21 m / s$$

Reynolds number can now be calculated. By definition, Re number is:

$$\mathrm{Re}_D = \frac{V \cdot D}{\nu} = \frac{6.21 m / s \cdot 0.006 m}{14.46 \cdot 10^{-6} m^2 / s} = 2576.76$$

With this last calculations we can enter the values to the initial correlation:

$$Nu = 0.33 \cdot C_h \cdot \mathrm{Re}^{0.6} = 0.33 \cdot 1.1 \cdot 2576.76^{0.6} = 40.416$$

And from this we can finally extract h from Nusselt number, because:

$$h = \frac{k \cdot Nu}{D} = \frac{0.02513 \cdot 40.416}{0.006} =$$

$$\boxed{h_{air,evap} = \mathbf{169.27\ W/m^2\ K}}$$

This value of h is also logical, as for forced convection h can range between 50 and 250 W/m^2 K

4. CONCLUSION

The preliminary design of the unit is now complete, and underneath is a summary of results obtained:

- Motor: 4-pole AC motor rotating at 1500 rpm
- Compressor Power: 2.3 kW
- Compressor Isentropic Efficiency: 65.07%
- COP Heat Pump: 2.93
- COP Air Con: 1.93
- Refrigerant R134a Mass flow: 0.0373 kg/s
- Cross-Flow Coils Dimensions: 20x30cm
- U-value Condenser: 2.413 kW/m^{2}K
- Air mass flow Condenser: 0.162 kg/s
- Effectiveness Condenser: 48.96%
- U-value Evaporator: 5.617 kW/m^{2}K
- Air mass flow Evaporator: 0.442 kg/s
- Effectiveness Evaporator: 41.03%
- Copper Tubing: Ø6mm, thickness 0.8mm
- Condenser-air h: 86.71 W/ m^{2}K
- Evaporator-air h: 169.27 W/ m^{2}K

This Project will be continued by other students in following years in order to manufacture the unit, based on all this preliminary design.

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