Heat Pump (Left)
Feasibility Study

Figure 1 Left Side Full Design

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1. Overview of Heat Pumps

1.1. History and Importance

William Cullen established artificial refrigeration in 1748 which is accepted as the very beginnings of the heat pump. In 1852 Lord Kelvin developed Cullen's idea and is generally credited with the scientific concept of the heat pump. It was Peter won Rittinger, however, who built the first heat pump system between 1855 and 1857. [1]

The objective of a heat pump is to absorb heat from a low temperature source and supply it to a warmer space. This is achieved by using a refrigerant which passes through many stages, outlined below. As thermal energy will only naturally flow from high to low temperatures, a source of power is required to drive the heat in the opposite direction, in this case a wind turbine.

A heat pump system is very important as with the addition of a reversing valve, the system can act as an air conditioning unit, therefore saving costs and space in the construction of a building. [2]

1.2. Components

There are four key components in a heat pump system: compressor, condenser, expansion valve and evaporator. Refrigerant as a gas at ambient temperature enters the compressor and is compressed using pistons to increase its temperature and pressure. This gas is then transported by insulated pipe into the warmer medium, in this case a water tank, and the thermal energy transferred by way of a heat exchanger. During this process the gas refrigerant will condense and cool ready for the expansion valve in which the pressure is rapidly decreased by limiting the amount of high pressure liquid that can pass through the system thus allowing the liquid to expand into a gas. The cold gas is then evaporated outside back to an ambient temperature gas ready for the cycle to begin again. [3]

1.3 Design Specifications

The brief required a small wind turbine system to be designed to provide domestic hot water. It was recommended that a low cost automotive compressor be adopted for the heat pump, and that the overall cost should be kept to a minimum as the final design is for private use.

Initial research was undertaken to understand key requirements for domestic systems, these are outlined below. [4],[5]

- Energy efficient: Coefficient Of Performance (COP) should be a reasonable value.
- Environmental protection: Minimal pollution to atmosphere and environment. Reduced emissions as the system will not require external ventilation.
- Ease of installation: Can be installed outdoors with no specialist equipment required.
- Low Operating Cost: Provide domestic water heating essentially for free or at substantial savings.
- Low Maintenance Costs: Minimal components means low failure rates, high reliability, easy servicing and increased life span.
- Durability: Last much longer than conventional HVAC (Heating, Ventilation and Air-conditioning) systems because they are protected from harsh outdoor weather.
- Low Noise
- Water temperature: Water should reach at least 65° before it is used as a legal requirement to combat legionnaire's disease. The tank does not have to be held at this temperature though as there will be a secondary water heating element later in the system.

1.4 Concept Designs

Design 1 – Overall Layout

Originally the design was as shown in figure 2 with the expansion valve at the tank outlet and the evaporator winding up the outside of the tower. This would mean a low pressure gas would have to be pumped up the tower, and the wound pipes would result in the vane of the rotor being fixed and unable to turn into the prevailing wind. Both of these factors make the design too inefficient.

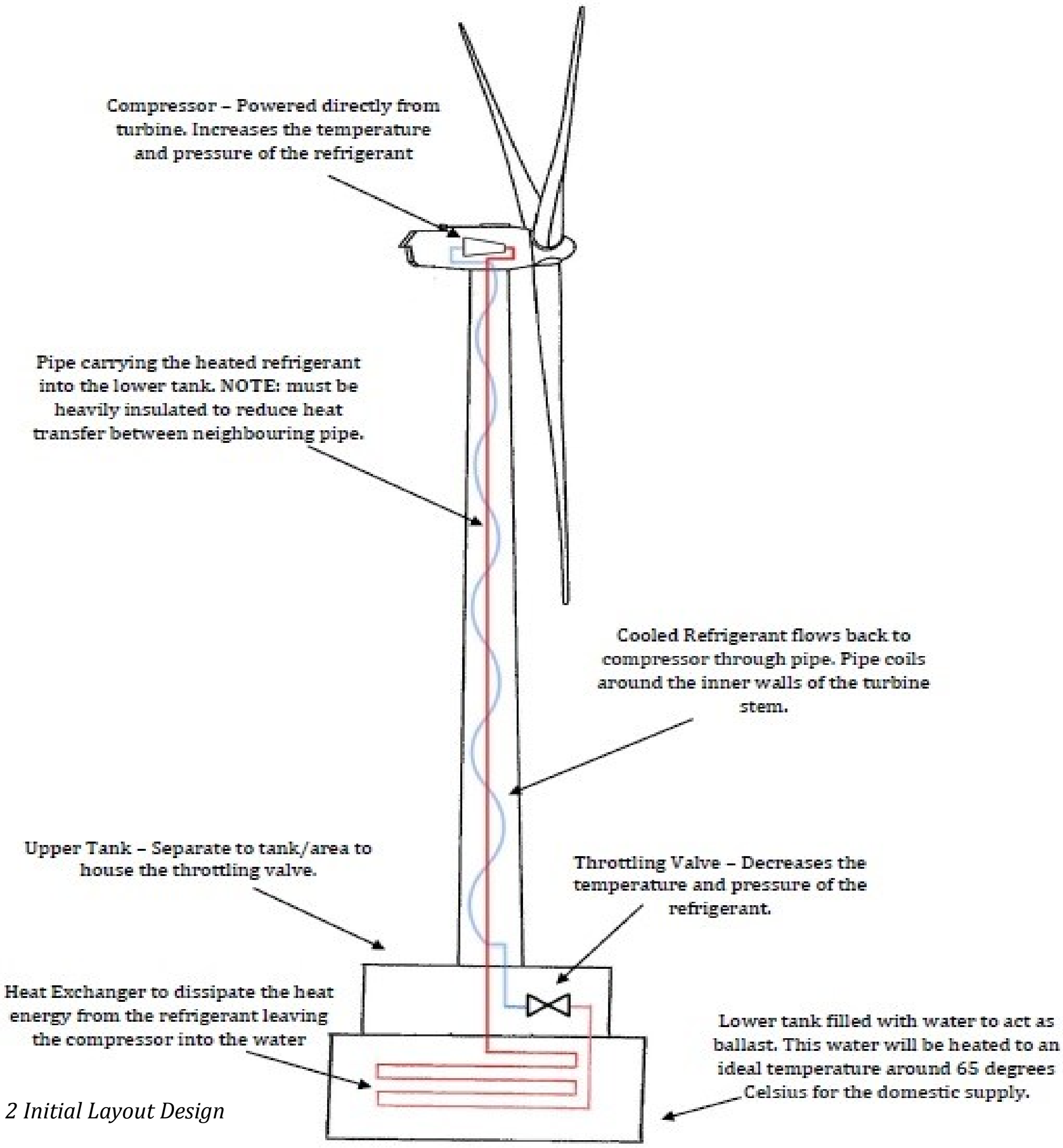

Figure 2 Initial Layout Design

Design 2 – Overcoming Twisting Pipes in the Rotation of the Vane

For efficiency, the vane should be able to rotate through 360 degrees into the prevailing wind direction. However, two separate pipes have to travel vertically between the water tank and the vane so only one pipe can lie on the vertical axis of rotation. To tackle this issue, various solutions were considered.

1. A flexible hose is used at one end of the low pressure pipe, so the pipe does not have to translate through an arc. This posed the problem of the hose becoming tangled around the solid high pressure pipe when the vane rotated too far. To counter this, the rotation of the vane could be limited to only 180 degrees. With this limitation however, a motor would be required to drive the vane against the wind if the blades were in an unfavourable orientation. This solution was not considered practical for this design.

2. The low pressure pipe feeds directly into the compressor at the top, and hangs vertically down the outside of the tower. At the bottom, on top of the water tank, it would feed into a toroid shaped tank wrapped around the tower. In order to allow rotation, the toroidal tank would be made of two halves, with the bottom half staying stationary, and the top half being free to rotate in direct accordance with the rotation of the vane. This idea was eventually discarded due to the manufacturing difficulties of the toroid tank, and the problem posed by keeping the tank sealed whilst keeping the rotation of the top half free so as to not apply significant torsion to the vertical pipe.

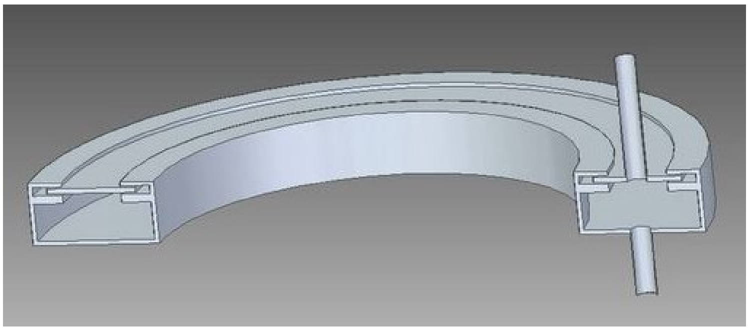

Figure 3 Toroidal Tank

3. By mounting the compressor at 90° to the shaft driven from the rotor blades, the rotation would be transferred through two bevel gears. This would allow the compressor and pipes to remain stationary while the vane and rotor could rotate around the crown bevel gear that is attached to the compressor. However, this idea was discarded as if the tail fin did not provide significant resistance to rotation, the rotor would simply continually rotate around the bevel gear rather than drive the compressor. A solution to this problem would require significant further engineering, seeing costs spiral and the project put over budget.

2. Final Design

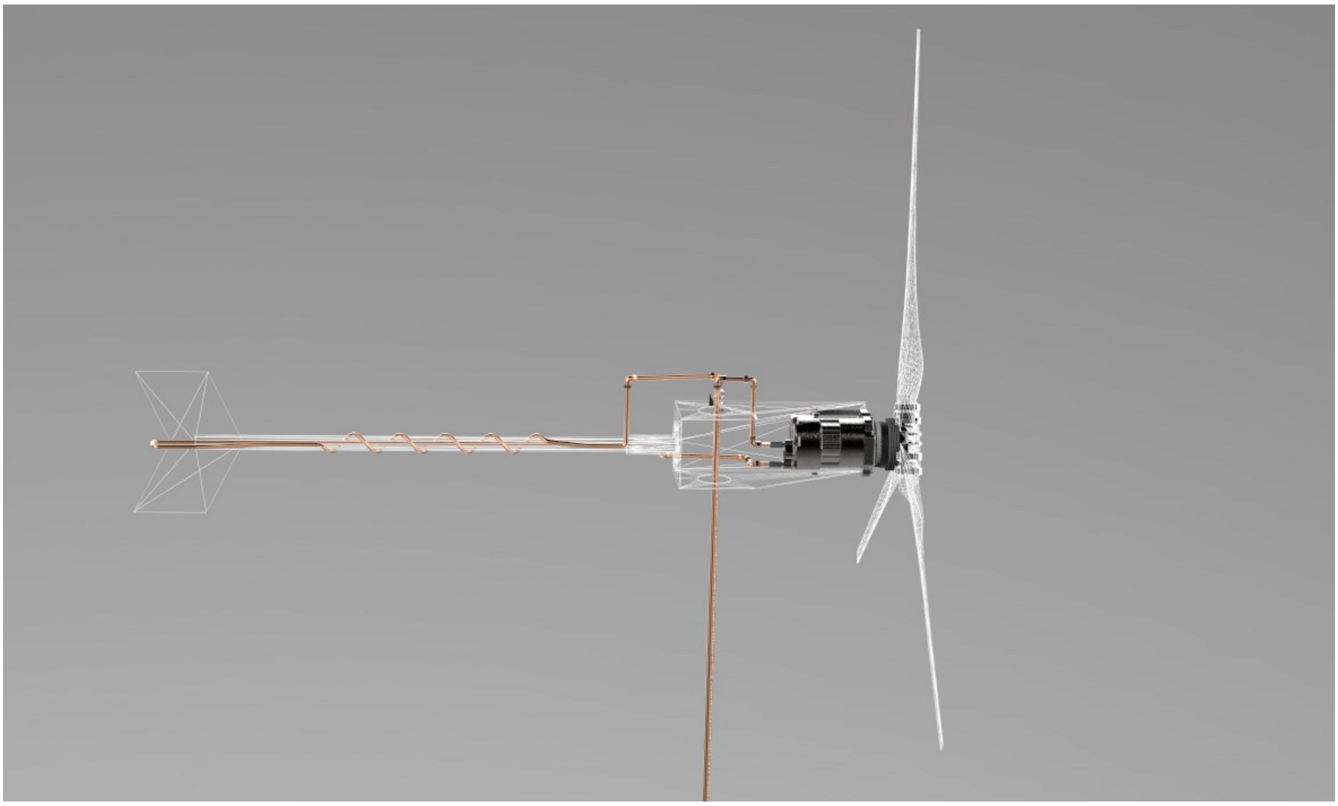

Figure 4 - Final Design of Heat Pump within Nacelle

2.1 Compressor

The compressor is the key component in the heat pump system. First, it takes in the low temperature and pressure refrigerant vapour from the evaporator. Then by piston action the refrigerant vapour is compressed in order to increase the pressure and temperature. Finally, the refrigerant exits into the heat exchanger and condenser. A flow chart of the process is shown in figure 5.

Figure 5 Compressor Process Flow Chart

There are two kinds of compressor, reciprocating and rotary, categorised according to their volume. A reciprocating compressor can be one of three styles; camshaft connecting rod compressor, wobble plate compressor and radiation compressor. The camshaft connecting rod is more difficult to install and the rotating speed is limited. The radiation type is also difficult to install and its performance at high speeds is poor. [6] For this project the wobble plate compressor has been chosen. A Sanden SD7V16 has been purchased for the prototype which is lightweight and has variable capacity, see figure 6. [7]

Figure 6 Sanden SD7V16 Compressor

This model can rotate between 800 and 8000rpm with a displacement range of 8.1cc to 161.3cc. [8] The compressor has seven pistons which can be understood from its name "SD7V16" the 7 for the number of pistons and the 16 for the maximum value of displacement (160). [9]

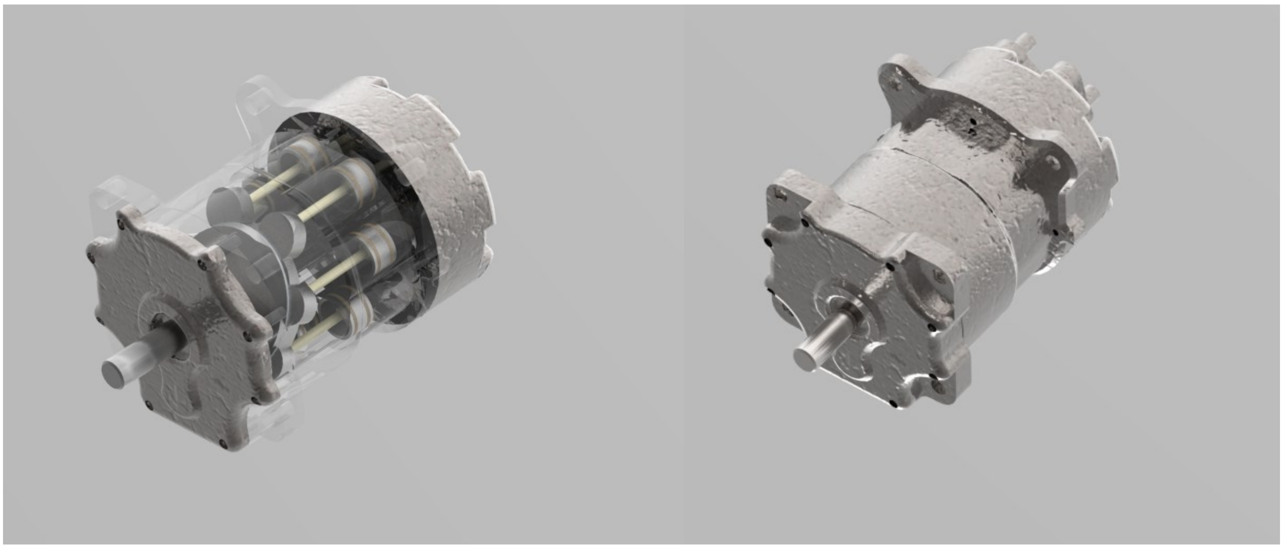

Figure 5 - Rendered Model of the Compressor

2. 2 Expansion Valve

An adjustable expansion valve (AXV) was chosen for the system as it will allow for the temperature and pressure of the refrigerant to be easily varied. This is very important as the water must not be allowed to overheat as this would cause a danger to the user. The AXV ensures that the evaporator is under a constant inlet pressure under all load conditions so that even as the temperature of the water changes it will not adversely affect the rest of the system. An AXV is less expensive to purchase and less complex to install than a traditional thermal expansion valve both of which are important to minimise the overall cost of the project. The use of an AXV will ensure more efficient running of the compressor, which in turn will reduce the power consumption; both are key features of this sustainable system. [10]

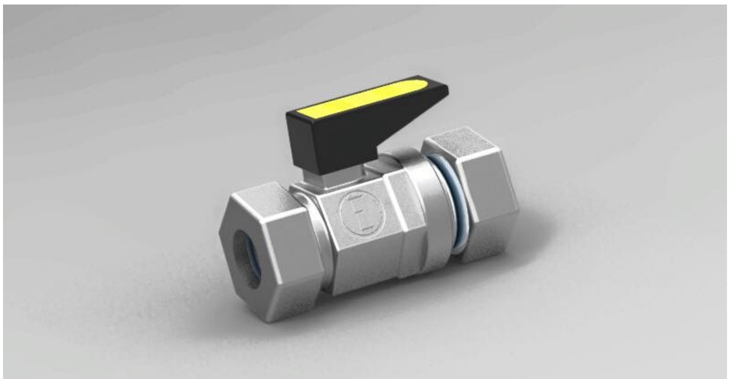

Figure 6 - Rendered Model of Expansion Valve

2.3 Piping

As stated in section 1.4, the vane must be able to rotate freely. It was decided that to keep within the budget of this project, both the pipes and the heat exchanger will rotate with the turbine vane, and the nacelle team will design some supports to ensure that the torsion does not damage the pipes.

2.3.1 Material

The material for the pipes must be carefully chosen according to the headings in table 1 below. The most important of these is thermal conductivity as the pipe will constitute the heat exchanger whilst inside the water tank. Clearly copper is the better choice for this project even though it can cost twice as much as Aluminium tubing. [11], [13]

Table 1 - Water Tank Piping Materials for Consideration

Material	Strength	Ductility	Corrosion Resistance	Thermal Conductivity (W/mK) [3]	Ease of Fabrication	Cost	Density
Copper	High	High	Good	381	Excellent	Medium	Medium
Aluminium	Medium-High	High	Excellent	211	Excellent	Low-Medium	Very Low

2.3.2 Manufacturing

The manufacturing of the pipes is mainly dependent on the number of bends required to achieve the area of contact between the pipes and the water for heat transfer. The pipe can be bent using various manufacturing methods such as press bending, rotary draw (CNC) bending, roll bending and heat-induction bending, all of which are widely used by companies throughout the United Kingdom, and can be done at a reasonable price. Another method is to use straight lengths of pipe with bent joints. This is the cheaper method, however, as the number of bends increases, so too must the number of joints. This is unfavourable for two reasons: increased cost due to extra parts and a much higher risk of pipe leakage. The latter of these problems is of most concern as maintenance of piping carrying refrigerant fluids and gases can be very expensive due to health and safety implications.

Despite cold bending being more expensive, the trade-off between a long lasting pipe network at a slightly higher cost far outweighs the problems involved in system failure and thus maintenance. Since this project is aiming for a long-life wind turbine, it is clear that making use of cold bending is the superior choice.

For the purpose of the prototype heat pump system, the piping in and around the compressor housing was connected using elbow pipe fittings since appropriate bending facilities were not readily available. These elbow joints allow for connections between the piping around the nacelle at right angles. As mentioned before, leaks would likely be an issue here and the joints proved to be quite expensive even for the scaled-down prototype due to the high number of joints necessary. As a result of the right angled nature of these joints, the refrigerant would likely separate as it flows through them, thus leading to small flow losses and a reduced system efficiency. Another factor to be considered is that these separation points will spawn small vortices, thus increasing the likelihood of stall-type phenomena in the joints.

2.4 Piping Insulation

Within our heat pump system there is a significant length of pipe that serves two functions: to act as the heat exchanger element within the water tank and to allow the refrigerant to travel between components. Obviously the primary aim with the heat exchanger is to maximise energy transfer between the refrigerant and the water supply therefore a piping material of high thermal conductivity is required. Copper was chosen for its desirable properties, as well as being the industry standard for its ease of fabrication. To cut down design costs copper pipe has been used throughout the entire design. Besides the heat exchanger, heat loss from the refrigerant is undesirable as it a source of inefficiency in the system and so losses should be reduced to a minimum. To negate the high thermal conductivity of copper pipes, extensive thermal insulation has been used to reduce heat loss. This is in the form of pipe lagging which consists of hollow tubes of an insulating material that simply wrap around the copper pipes.

There are several different types of insulating material that can be used and are widely available from hardware stores. Insulating materials primarily exist as either wools/fibres or as solid foam. The latter is normally manufactured from Polyethylene which has a melting point of approximately 115°C. This makes it unsuitable for the high temperature applications of our heat pump due to risk of melting. Some form of wool insulation is therefore the optimal choice for our design.

Table 2 - Insulation Materials for Consideration

Insulation Material	Environmental Impact	Insulation Properties	Skin Irritability	Combustibility	Cost
Sheeps Wool	Natural Product: None	Excellent 0.0354W/m K	None	Not readily combustible	High
Glass Wool	Recycled Glass: Environmentally Friendly	Excellent While Dry	Needs to be handled with gloves	Fire retardant	Low
Mineral Wool	Quarried Rock/Release of Chemicals: Large Impact	Excellent	Can cause temporary irritation	Very fire resistant	Low/Mid

Examining the table it is clear that glass wool insulation would optimise the two key design considerations; environmental impact, and cost. Steps should be made by the nacelle team to ensure that the insulation is kept dry at all times.

2.5 Fixings and Connectors

Various fixings were considered for attaching the copper pipes to the compressor inlet and outlet. The inlet was simple as the refrigerant will be at atmospheric pressure coming from the evaporator and so a jubilee clip will suffice to connect the copper pipe to the flexible hose. The jubilee clip has the advantage of being a very cheap and easy to replace component, commonly made of galvanised steel, making it corrosion resistant. It is possible that the connection will not be sound enough for compressed flow though and will result in unwanted leakage. This becomes less of an issue as the piping is inserted further into the hose, providing the jubilee clip keeps the piping fixed securely.

At the output of the compressor the refrigerant pressure will be at its highest (approximately 16.7MPa), thus, the flexible hosing has been removed and a reducing coupler fixing has been used between the 15mm output pipe and the 10mm copper pipe. This is a very cheap and secure solution which is easy to maintain or replace. [14]

2.6 Adding the Refrigerant

The system refrigerant is supplied in a canister equipped with a pressure gauge. If not for this gauge, an AC manifold set would be necessary in order to carry out a safe refrigerant import. Firstly, it is important to ensure the system is empty by using a vacuum pump. After this the canister can be attached to the system via a low pressure tight hose, and the refrigerant can be input until the canister pressure gauge reaches its “full” position - rather than “low” or “overfull”. The canister can then simply be disconnected, leaving the system with sufficient refrigerant levels.

In the event of a leak it is possible to add dye to the system in order to identify it. It is important to turn the system off prior to dye input as leaks can normally be identified and located without having to make use of it simply by listening. After dye input, if necessary, test pressures are engaged and analysis of the system components is carried out in order to detect the leak.

3 Calculations

3.1 Data Collection

The data for the calculations shown in table 3 was obtained from a variety of sources. The power input to the compressor is the quoted power received from the rotor team, based on a wind speed of 14m/s.

The isentropic efficiency was estimated to be 0.88, and the mechanical efficiency of the compressor to be 82%. These figures are the lowest allowable in the range of ‘normal values’ given [15], and were chosen as they allow for a reasonable factor of safety for the design.

The quoted discharge pressure for the compressor is found from the technical graph for the compressor from the manufacturer’s site seen in the appendix 11.

The revolutions per second of the compressor were found based on the quoted 1050rpm of the rotor at an optimised wind speed of $14ms^{-1}$.

The volume of the tank was chosen as $2m^3$, as this amount would compensate for when the wind is not blowing and so the water is not being heated.

The heat losses from the tank at ambient temperatures of -15°C and 10°C are quoted values from the nacelle team, based on a water temperature of 65°C. For the purpose of these calculations these heat losses were also used for the tank during the initial stage of the heating process, in order to act as a safety factor for the heat pump design.

The temperature of the water is selected as 65°C, however due to the inconsistent nature of the power supply through the wind turbine the temperature may dip below this value for periods of time. This temperature is a legal requirement as it prevents Legionnaire’s Disease from propagating.

Table 3 Input Data for Calculations

Input Data			
Data	**Symbol**	**Value**	**Units**
Power Input	P_T	1.55	kW
Isentropic Efficiency	η_I	0.88	
Discharge Pressure	P	16	Bar
Mechanical Efficiency	η_M	0.82	
RPS of Compressor		17.5	RPS
Compressor cylinder Outer Diameter	d_c	2.9	cm
Compressor cylinder area	A_c	6.605192975	cm^2
Volume of Water in the Tank	V_w	2	m^3
Specific Heat of Water	c_w	4.19	kJ/kgK
Density of Water	ρ_w	999.97	kg/m^3
Mass of Water	m_w	1999.94	kg
Heat Loss from Tank at outside temp 10°C	Q_{Lost10}	0.126	kW
Heat Loss from Tank at outside temp -15°C	$Q_{Lost-15}$	0.18327	kW
Temperature of Water	T_w	65	°C

3.2 Temperature of Refrigerant at Compressor Outlet

The basic heat pump configuration is as shown in figure 8 [16], where the refrigerant enters the compressor as a superheated gas, is compressed further into a superheated vapour, and then condenses in the heat exchanger releasing energy in the form of heat. Passing through the expansion valve causes a drop in the refrigerant's enthalpy, pressure and temperature, and flowing through the evaporator coils the refrigerant gains heat raising the temperature until it is a superheated gas once more.

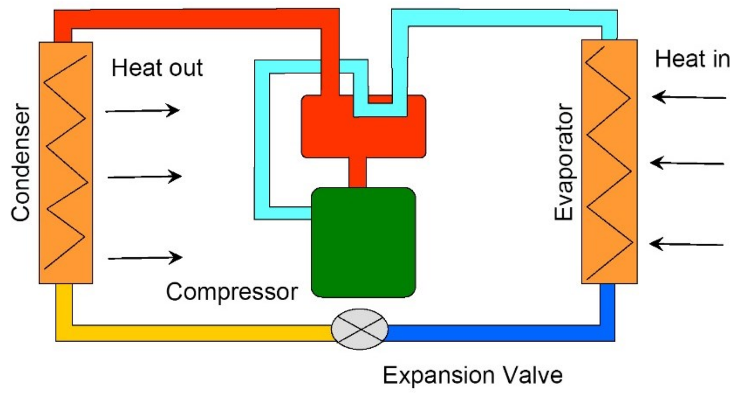

Figure 7 Basic Heat Pump Configuration

It is assumed that during the initial heating of the water the cycle in figure 9 is followed, and that the refrigerant will give out all of its heat energy until it reaches a saturated liquid state at 57.88°C. However once the water also reaches this temperature, and surpasses it, the refrigerant will no longer reach a saturated liquid state. For the purposes of simplifying the calculations in this model it has been assumed that past the water temperature of 57.88°C, the refrigerant will leave the tank still in its superheated vapour form at 65°C.

The calculations for the fluid is based on the values obtained from the Dupont Suva Refrigerant tables for R-134a. The refrigerant will be entering the compressor at atmospheric pressure (101.325kPa) and at ambient air temperature. Two cases have been examined for the ambient air temperature; a worst case scenario of -15°C, and an average UK air temperature of 10°C. The temperatures of the refrigerant at the outlet of the compressor are 86.93°C and 112.18°C respectively. The full calculations along with the refrigerant tables can be viewed in appendices 1,2,3,4 and 12.

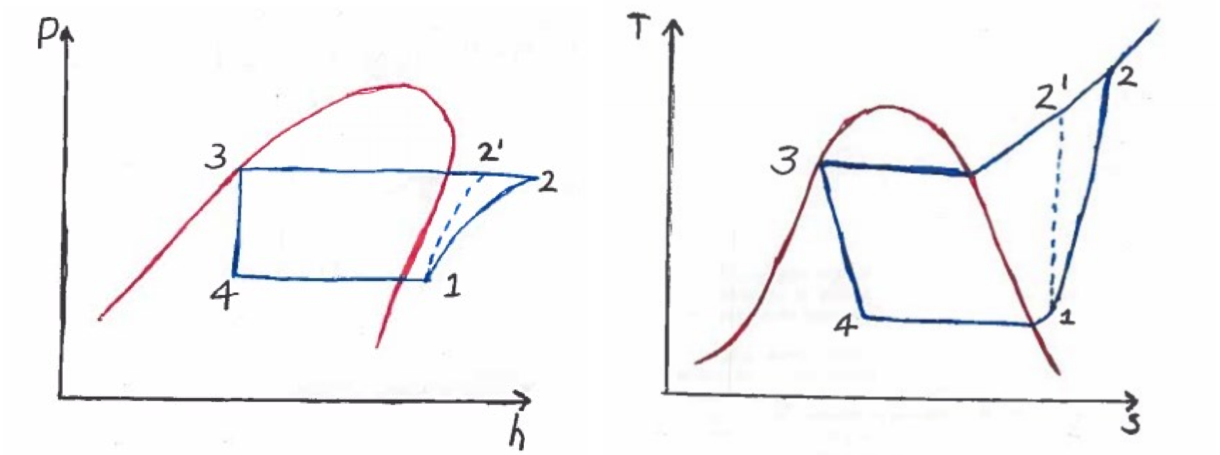

Figure 8 Refrigerant Pressure and Temperature Diagrams

3.3 Fixed Compressor Displacement and Heat Addition to Water

To find the value of the displacement it is necessary to find the mass flow rate of the refrigerant and the actual power input to the compressor from the rotor. The mass flow rate is converted into volume flow rate so the volume of refrigerant pumped by each piston in the compressor can be found, and therefore what displacement the piston must go through to reach the desired water temperature. Also from the mass flow rate, the heat output to the water and the heat input to the refrigerant from the air around the evaporator can be calculated. The results for operation at -15°C and 10°C are shown in tables 4 and 5 respectively, with full calculations for section 3.3 shown in appendices 1 and 2.

Table 4 Results for Heat Pump Operating at -15°C

Output	
Power input (kW)	1.271
Mass flow rate (kg/s)	0.0182
Heat Input to Evaporator with Tank T< 57.88°C (kW)	1.9565
Heat Input to Evaporator with Tank T> 57.88°C (kW)	-0.7986
Heat Output in Condenser with Tank T< 57.88°C (kW)	3.2275
Heat Output in Condenser with Tank T> 57.88°C (kW)	0.4724
COPHP < 57.88°C	2.5393
COPHP > 57.88°C	0.3716
Volume flow rate (m^3 / s)	0.000271
Volume flow rate (cm^3 / s)	270.5637
Displacement Fixed (cc)	15.4608

Table 5 Results for Heat Pump Operating at 10°C

Output	
Power input (kW)	1.271
Mass flow rate (kg/s)	0.0164
Heat Input to Evaporator with Tank T< 57.88°C (kW)	2.0917
Heat Input to Evaporator with Tank T> 57.88°C (kW)	-0.3833
Heat Output in Condenser with Tank T< 57.88°C (kW)	3.3627
Heat Output in Condenser with Tank T> 57.88°C (kW)	0.8877
COPHP < 57.88°C	2.6457
COPHP > 57.88°C	0.6984
Volume flow rate (m^3 / s)	0.000276
Volume flow rate (cm^3 / s)	276.2092
Displacement Fixed (cc)	15.7834

3.4 Water Tank Heating Time

As a domestic hot water system, the amount of time it will take to heat the tank of water is very important. It is assumed that the water starts at 10° and is heated in two stages to the desired temperature of 65°C. It is first heated until it reaches the refrigerant saturated liquid state at 57.88°C and then it is further heater to 65°C. The total time taken to heat the full tank of water in the two ambient air temperatures cases are shown in tables 6 and 7, with the full calculations in appendix 3.

Table 6 Time Taken to Heat Tank with Outside Air Temperature 10°C

In the Tank		
Energy required to heat the tank	460886.173	kJ
Heat Input to Tank <57.88°C	3.3627	kW
Heat Input to Tank >57.88°C	0.8877	kW
Heat Loss from Tank	0.126	kW
Time taken to Heat the Tank <57.88°C	34.43	hours
Time taken to Heat the Tank >57.88°C	21.76	hours
Total Heating Time	56.19	hours

Table 7 Time Taken to Heat Tank with Outside Air Temperature of -15°C

In the Tank		
Energy required to heat the tank	460886.173	kJ
Heat Input to Tank <57.88°C	3.2275	kW
Heat Input to Tank >57.88°C	0.4724	kW
Heat Loss from Tank	0.18327	kW
Time taken to Heat the Tank <57.88°C	36.61	hours
Time taken to Heat the Tank >57.88°C	57.33	hours
Total Heating Time	93.94	hours

Once the water has reached 65°C it can be used within the household. The amounts of water available at this temperature over a 24 hour period are found to be 285.58kg and 99.35kg at 10°C and -15°C respectively.

3.5 Loss of Pressure in Pipes due to Bending

When fluid travels around a corner, due to the nature of the path that the fluid takes, there is always a pressure drop due to separation on the inside of the bend. These losses must be minimised in order for the pressure in the pipes to remain high enough that there is sufficient pressure for the refrigerant to travel back up the pipes towards the expansion valve, evaporator vane and the compressor. This is imperative as for the system to run continually, there must be no excess build of liquid state refrigerant in the lower part of the heat exchanger due to pressure loss.

Pressure losses in a piped system follow an inversely proportional relationship with the radius of the bend in the pipe. Therefore, in order to minimise the pressure losses due to bending, the radius of the heat exchanger must be kept relatively large. As a result, the radius of bend for the spiralled heat exchanger was designed to be 200mm. This then gave a total pressure loss due to the bending and friction of the pipe of approximately 6422.644Pa. This calculation was made assuming there would be no drop in velocity due to the loss in pressure whilst the fluid was travelling through the heat exchanger, which would have negligible effect on the overall pressure comparisons.

Due to the design of the system, the compressed fluid must travel through 3 right angle turns before reaching the spiralled heat exchanger. These turns must be kept tight in order for the pipes to navigate the complex design of the tower. Therefore, the radius of bend must be 9mm. As the pipes are of a small diameter however, this only results in a pressure loss due to bending and friction of 546.1986Pa. Unfortunately, due to the current design of the tower, and the efficiency of the heat exchanger design, these losses are both unavoidable and necessary. They do not however, affect the ability of the refrigerant to be propelled back up the tower as once these losses have been partially offset by the change in head, the overall loss in pressure is approximately 2000kPa.

3.6 Piping Lengths

The length of piping required for the design was calculated based on values given by the nacelle team for the height of the tower, and average lengths of piping currently used in car AC condensers and evaporators. It was therefore required that 4.5m of piping would be used for the condenser, 4m would be used in the evaporator, and 18m of piping would be needed for transporting the refrigerant up and down the 9m tower. A further 3.5m of piping was to be accounted for to ensure there is sufficient piping lengths for all the necessary connections. This lead to a total piping length required for the final design of 30m.

In the feasibility study a much smaller tower was being used, and so the length of piping required for the study was greatly reduced. This led to the total length of piping required for the feasibility design to be 10m.

4. Control

4.1 Compressor Displacement Control

A control system in the compressor is required to achieve the desired water temperature over a range of different wind speeds. The simplest way to do this is to fix the displacement of the compressor. Our compressor was originally designed for use in a car's air conditioning system which requires a constant cooling effect and so the same volume of refrigerant must be displaced regardless of the input angular velocity. This constant volume displacement was carried out by connecting the input shaft of the compressor to a wobble plate, whereby at low velocities the input shaft is tilted leading to maximum piston displacement and at high velocities the inertial effects on the plate centralise it thus reducing the piston displacement.

The aim of our design was to find a value for the piston displacement that would result in suitable heat transfer over a wide range of wind speeds to provide a water temperature near to the specified 65°C. Using interpolation the length of spacer required was calculated to be approximately 10mm, and so it was manufactured by hand using some leftover piping in the laboratory and a hacksaw. It was then fitted to the input shaft between the spring and securing circlip to hold the wobble plate in place, thereby fixing the piston displacement.

4.2 Rejected Displacement Control Methods

Screw Fixing the Displacement

One possible design option for fixing the compressors displacement would have been to drill a hole through both the wobble plate and the body below it and screw it into a fixed position. This idea was discarded, after examination, however, as it was apparent that this would require a much more skill and precision than simply using a spacer.

Use of a Governor

Although a governor would provide a more accurate means of fixing the compressors displacement it would also require a good degree more technical skills when it comes to its manufacturing, thereby adding to the overall cost, and so it was considered unnecessary for the scale of our design.

5. Bill of Materials

Table 8 - Bill of Materials

Part Name	Serial/Model #	Description	Qty	Unit Cost (£)	Other Cost (£)	Cost (£)
Compressor	Serial: 6180007324	Compresses refrigerant from atmospheric pressure to 16bar, driven by rotor	1	25	0	25
Reducing Coupling 15mm - 10mm	Model: 51238	Reducing Coupler to reduce pipe diameter from 15mm to 10mm; Compression fitted	2	1.99	5	8.98
Copper Pipe	Model: 17717	10m of 10mm diameter copper pipe to transport r134a refrigerant	10	£59.99/25m	0	24
10mm Mini Ball Valve	Model: 79038	Reduces pressure in 10mm diameter pipe from high to atmospheric; Compression Fitted	1	6.09	5	11.09
Jubilee Clip	Model: 69166	Securing mechanism used to attach flexible hosing to rigid pipe, ranges between 20mm-50mm	2	0.82	5	6.64
Compression Elbow 10mm	Model: 24019	Changes direction of 10mm diameter pipe through a right angle; Compression fitted	10	1.88	10	28.8

5.1 Overall Cost

The overall cost for parts is **£104.51**. This is under the original budget of £120, leaving a surplus of £10.49 approximately 10%. As a prototype this is very promising as it allows further, more refined prototypes freedom to use slightly more expensive materials or methods of production in order to create a higher performing and more efficient heat pump for the consumer. By having a budget surplus, it very much shows that the design is feasible and should be taken forward with confidence.

6. Prototype Testing

The first prototype of our heat pump design started by working the copper pipe into the desired spiral shape. The pipe between the compressor and expansion valve was designed as one piece to avoid unnecessary joints which increase the chance of a leak and incur energy losses due to viscous friction. The downside to this was that particular attention was required to avoid kinking the pipe when sharp turns were required. The desired lengths of pipe were marked onto the pipe using a marker pen by measuring out pieces of string the required lengths and comparing them to the pre-coiled pipe. This allowed us to clearly see where the pipe should be cut and where the pipe should be coiled to form the heat exchanger segment. The heat exchanger was formed by wrapping the piping around an old gas canister to achieve a uniform helix shape. The remaining length of pipe was then curved back up through the centre of the coil where the expansion valve was fitted.

The expansion valve used was a basic ball valve which uses a manually operated lever to alter the orifice size of the valve and therefore outlet pressure. Linked to the expansion valve, the evaporator was produced by a similar method of coiling a length of copper wire around a cylinder. The compressor was slightly modified to allow us to fit the micro-bore copper pipe to both the inlet and outlet by removing the rubber piping and using an adaptor fitting. The piping in and around the compressor housing was connected using elbow pipe fittings, [18] as appropriate bending facilities were not readily available. These elbow joints allow for connections between the piping around the nacelle at right angles. As outlined in section 2.3.2, leaks would likely be an issue here and the joints proved to be quite expensive even for the scaled-down prototype. This reinforced the previous decision to use cold bending as the manufacturing process.

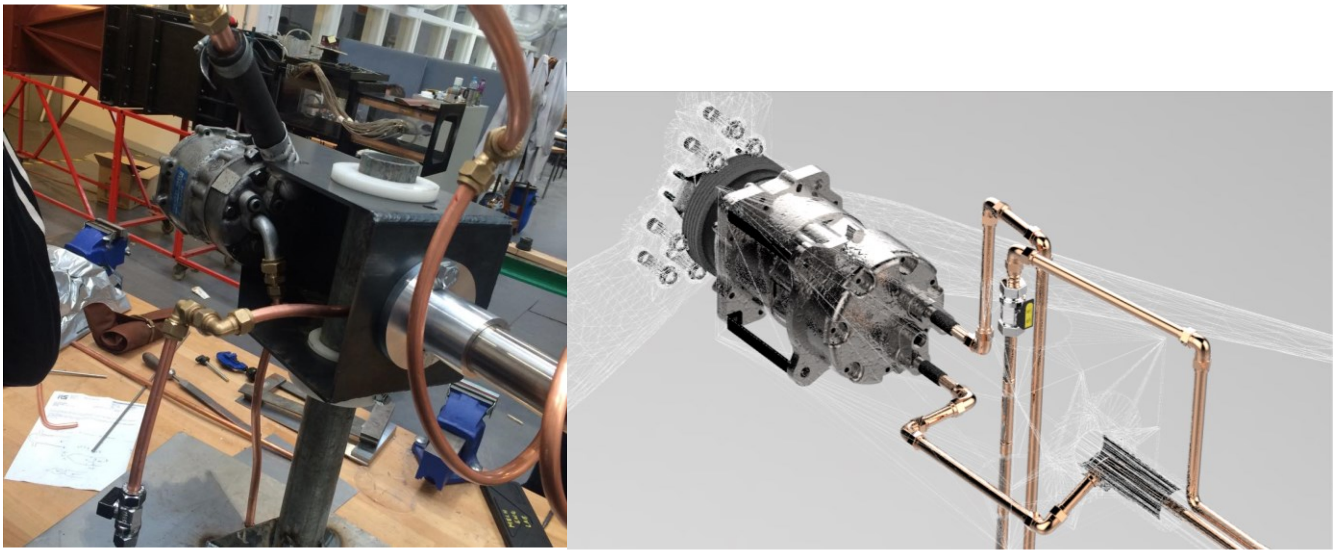

Figure 9 - Comparison between Prototype and Final Design Model

7. Conclusions

This project was found to be highly feasible, efficient and sustainable. The calculations suggest that the water could be heated to a suitable temperature in a reasonable amount of time, even in extreme weather. Hot water will always therefore be available especially considering there would certainly be a secondary heating element before the water reached the end user. The use of an automotive compressor is possible however care would have to be taken to find one which runs at a sufficiently low power input. The refrigerant is legal for use however it is important to follow standard when emptying the system as it is illegal to release the gas into the atmosphere. Careful testing of the system is therefore also necessary before filling to ensure that there are no leaks, minimising the risk to the environment.

The system should have a long life span as the copper pipes are corrosion resistant to both the water and the refrigerant. The compressor may require some maintenance throughout however as a standard part this should be simple. As stated though care must be taken as the refrigerant is a pollutant. The amount of disturbance due to noise was a concern for a domestic system however the prototype showed that this will not be a problem.

The overall cost for the prototype parts is £104.51. This is under the original budget of £120, leaving a surplus of approximately 10%. As a prototype this is very promising as it allows further, more refined prototypes freedom to use slightly more expensive materials or methods of production in order to create a higher performing and more efficient heat pump for the consumer. By having a budget surplus it very much shows that the design is feasible and should be taken forward with confidence.

8. Recommendations for Further Study

The design discussed in this report is a viable option but potential improvements are possible to increase the efficiency of the system.

Smaller Compressor

In order to refine the system for future improvements, a smaller compressor could be used. The current compressor's 7-piston assembly, with a potential displacement of between 8cc - 160cc, is unnecessary and inefficient. The compressor is required to operate at a displacement of 14cc and therefore is only required to work within approximately 5% of its potential range. Moving forward, a greater emphasis should be placed upon using a smaller compressor that would utilize its range of displacements to greater effect, allowing for an increase in efficiency. This could be achieved by reducing the properties of the rotational motion that drives the compressor; a decrease in rpm would allow for an increase in torque, which would then facilitate a compressor with a smaller displacement.

Piping around Nacelle

The design of the piping at the top of the tower is inefficient as it requires three elbows joints in order to navigate the pipe down the tower towards the heat exchanger. As well as this being wasteful of the copper, the many elbows lead to greater pressure losses in the pipes, as shown in Appendix 4. A more efficient design would result in less energy losses so saving the client money by being more sustainable.

Tank Thermometer

Incorporating a wax thermostat, such as one commonly found in radiators, would allow for the heating of the water to be monitored. These devices use a chamber full of wax which expands upon melting to drive a piston. In radiators this piston is used to restrict the flowrate of the heating fluid until the temperature drops below the given value. Our system would employ this type of thermostat in a similar fashion. A simple digital thermometer should be included in the design to allow the user to check the water temperature within the tank to check that it lies within a reasonable range from the optimum value.

Piping Length

The values for piping length were taken as estimates from current compressor and evaporator lengths used in car AC systems, and are therefore not guaranteed to allow the full heat transfer quoted in the calculations. A study into the exact length of piping required for full heat transfer to and from the refrigerant would allow for minimal piping waste in the design, as well as a potential reduction in the pressure drop in the piping should it prove to be less than is originally planned in these calculations.

9. Project Management

The team was split into three rough skills groups: two project managers, four analysts and two CAD modellers. Initially the group met to brainstorm ideas and establish sections that would be followed to complete the project, and the managers outlined a projection of deadlines for these areas of work, as shown in appendix 13. They were careful to ensure that everyone had an important role to play and that the whole group was aware at all times what stage the project was at. Regular, weekly group meetings were held with minutes taken. Each member could present their work, and others could contribute with ideas and queries to help ensure that the final project would be as high quality as possible. A new set of tasks would then be outlined, with the managers considering what people enjoyed working on and where each member's talents lay, and a deadline would be agreed. Often, tasks were given to groups of people to work on together which was considered beneficial both for the project and for the dynamics of the group.

The managers would liaise with the managers of the other sub groups of nacelle and rotor to ensure that the project as a whole was well linked. As questions about the details of the brief arose, managers were quick to correspond with the client and confirm that what the group was designing was in fact what was desired. This safeguarded both the continuing development of the design and the confidence of the members in their work. It was considered that the group had a very encouraging and positive dynamic throughout the project which allowed for all members to do their best work.

For the production of the prototype, the managers became very involved in agreeing a bill of materials to be ordered with the lab technicians. Agreeing a time that the group may go to the workshop was also very important. Managers sourced and ordered some materials directly for the group to ensure that the production could be swiftly continued. The team enjoyed the workshop time, with some more experienced members taking a leading role in production.

Due to the nature of the report requiring the information to be readily accessible and easily understood, the team has created a brief introduction to the heat pump part of the design with all the key information on the project website, found at http://windturbinedesign.wix.com/sustainableenergy . It is hoped that any prospective customers would be able to find any basic information they require for the design on the website, without the need of having to look through the feasibility report.

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Appendix 1 – Temperature of Refrigerant at Compressor Outlet

Table 9 – Input Data for Calculations

Input Data			
Data	**Symbol**	**Value**	**Units**
Power Input	P_T	1.55	kW
Isentropic Efficiency	η_I	0.88	
Discharge Pressure	P	16	Bar
Mechanical Efficiency	η_M	0.82	
RPS of Compressor		17.5	RPS
Compressor cylinder Outer Diameter	d_c	2.9	cm
Compressor cylinder area	A_c	6.605192975	cm^2
Volume of Water in the Tank	V_w	2	m^3
Specific Heat of Water	c_w	4.19	kJ/kgK
Density of Water	ρ_w	999.97	kg/m^3
Mass of Water	m_w	1999.94	kg
Heat Loss from Tank at outside temp 10°C	Q_{Lost10}	0.126	kW
Heat Loss from Tank at outside temp -15°C	$Q_{Lost-15}$	0.18327	kW
Temperature of Water	T_w	65	°C

The calculations for the average UK air temperature (10°C) can be seen below. At the designated entry temperature and pressure the refrigerant exists as a superheated gas, and its corresponding entropy and enthalpy can be found from reading the refrigerant tables.

h_{1avT} = 412.2 kJ/kg s_{1avT} = 1.8581 kJ/kgK

The Isentropic expansion can then be calculated using an ideal isentropic process for the compression of the refrigerant. This means that the ideal $s_{2avT} = s_{1avT}$

The refrigerant tables for the outlet compressor pressure of 1600kPa could then be examined and the nearest entropy values above and below the s_{1avT} value taken, along with their corresponding temperature and enthalpy values. Interpolation could then be carried out in order to find the enthalpy and temperature that corresponds to this ideal entropy value, as seen in the Interpolation 1 section of table 10 below.

The enthalpy change can then be calculated for this ideal process to be:

$$\Delta h_{ideal} = h_{2avT} - h_{1avT} = 68.2846 \text{ kJ/kg}$$

The Isentropic efficiency of the compressor is then used to calculate the actual enthalpy increase in the compressor:

$$\Delta h_{Actual} = \frac{\Delta h_{Ideal}}{Isentropic\ Efficiency} = 77.5961 \text{ kJ/kg}$$

This actual enthalpy change can then be added to h_{1avT} to find the actual h_{2avT} value. This can then be used to calculate the actual temperature and entropy of the refrigerant at the compressor outlet using a second interpolation seen in table 10 below. This gives a final value of 112.18°C for the refrigerant leaving the compressor.

Table 10 Interpolations to Calculate Output Enthalpies and Temperatures for 10°C

Temperature (°C)	**Pressure (kPa)**	**Enthalpy (kJ/kg)**	**Entropy (kJ/kgK)**
Interpolation 1 for Ideal Isentropic Compression			
100	1600	476.2	1.8467
103.8255	1600	480.4846	1.8581
105	1600	481.8	1.8616
Interpolation 2 for Real Isentropic Compression			
110	1600	487.4	1.8761
112.1783	1600	489.7961	1.8824
115	1600	492.9	1.8905

The same process can be carried out for the other ambient air temperature of -15°C, giving the values seen in tables 11 & 12 below:

Table 11 Interpolations to Calculate Output Enthalpies and Temperatures for -15°C

Temperature (°C)	**Pressure (kPa)**	**Enthalpy (kJ/kg)**	**Entropy (kJ/kgK)**
Interpolation 1 for Ideal Isentropic Compression			
75	1600	447.7	1.7675
79.6108	1600	453.1407	1.7829
80	1600	453.6	1.7842
Interpolation 2 for Real Isentropic Compression			
85	1600	459.3	1.8467
86.9345	1600	461.5054	1.8525
90	1600	465	1.8616

Table 12 Output from Interpolation of Results for -15°C

Output			
Ideal Enthalpy Change at Outlet	Δh_{ideal}	61.3407	kJ/kg
Actual Enthalpy Change at Outlet	Δh_{Actual}	69.7054	kJ/kg
Actual h2 at Outlet	h_2	461.5054	kJ/kg
Actual Temperature at Outlet	T_2	86.9345	°C

Appendix 2 - Fixed Compressor Displacement and Heat Addition to Water

The actual power input to the compressor can be found by taking the power supplied by the rotor and combining it with the mechanical efficiency:

$$P_C = P_T \times \eta_M$$

This can then be used to calculate the Mass Flowrate of the system when combined with the actual enthalpy change calculated above:

$$\dot{m} = \frac{P_C}{\Delta h_{Actual}}$$

Once the mass flow rate is obtained the heat input into the refrigerant in the evaporator, as well as the heat output into the water during the heating process in the condenser can be calculated. As it has been stated previously the refrigerant is assumed to be a saturated liquid upon leaving the condenser up until the water temperature reaches 57.88°C. After this point it has been modelled as leaving the condenser as a saturated vapour at 65°C for simplicity. The calculations for heat addition to the water can then be split into two parts, taking the values from the refrigerant tables as follows: For a water temperature below 57.88°C, $h_3 = 284.5\ kJ/kg$, and for a water temperature passed 57.88°C it is assumed $h_3 = 435.6\ kJ/kg$. It is also assumed that the expansion in the expansion valve is Isentropic, and therefore $h_3 = h_4$:

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

$$\dot{Q}_{Cond} = \dot{m} \times (h_2 - h_3)$$

These values could then be used in conjunction with the Power input to the compressor in order to calculate the overall COP of the Heat Pump:

$$COP_{HP} = \frac{\dot{Q}_{Cond}}{P_C}$$

Going back to the mass flow rate calculated for the refrigerant as it leaves the compressor, this can be converted into a volume flowrate using the density of the refrigerant. The density was calculated by putting the output pressure and temperature of the refrigerant for both ambient air temperature cases into an online calculator for refrigerant R-134a[2] These densities were found to be 67.3922kg/m^3 for the system being used when the outside air temperature is -15°C, and 59.3018 kg/m^3 when the outside temperature is 10°C. The volume flowrates were calculated by:

$$\dot{V} = \frac{\dot{m}}{\rho}$$

The volume flow rate could then be converted into the volume of fluid pumped by each piston every revolution using:

$$V_{stroke} = \frac{\dot{V}}{RPS_{comp}}$$

Tables 13 and 14 show these values for the two different scenarios.

Table 13 Results for Heat Pump Operating at -15°C

Output	
Power input (kW)	1.271
Mass flow rate (kg/s)	0.0182
Heat Input to Evaporator with Tank T< 57.88°C (kW)	1.9565
Heat Input to Evaporator with Tank T> 57.88°C (kW)	-0.7986
Heat Output in Condenser with Tank T< 57.88°C (kW)	3.2275
Heat Output in Condenser with Tank T> 57.88°C (kW)	0.4724
COPHP < 57.88°C	2.5393
COPHP > 57.88°C	0.3716
Volume flow rate (m^3 / s)	0.000271
Volume flow rate (cm^3 / s)	270.5637
Displacement Fixed (cc)	15.4608

Table 14 Results for Heat Pump Operating at 10°C

Output	
Power input (kW)	1.271
Mass flow rate (kg/s)	0.0164
Heat Input to Evaporator with Tank T< 57.88°C (kW)	2.0917
Heat Input to Evaporator with Tank T> 57.88°C (kW)	-0.3833
Heat Output in Condenser with Tank T< 57.88°C (kW)	3.3627
Heat Output in Condenser with Tank T> 57.88°C (kW)	0.8877
COPHP < 57.88°C	2.6457
COPHP > 57.88°C	0.6984
Volume flow rate (m^3 / s)	0.000276
Volume flow rate (cm^3 / s)	276.2092
Displacement Fixed (cc)	15.7834

Appendix 3 - Water Tank Heating Time

The energy required to heat the tank was found using the data seen in table 9, appendix 1 above:

$$Q = m_w \times c_w \times \Delta T$$

For the original calculation of the length of time it takes to heat the water, its starting temperature is 10°C which is then heated to our desired water temperature of 65°C in 2 steps:

The length of time to heat the tank can then be calculated using the heat inputs to the tank from the condenser, and the heat losses from the tank:

$$t = \left(\frac{Q}{\dot{Q}_{Cond<57.88} - \dot{Q}_{Lost}} \times 60 \times 60 \right) + \left(\frac{Q}{\dot{Q}_{Cond>57.88} - \dot{Q}_{Lost}} \times 60 \times 60 \right)$$

The results of these calculations can be seen in tables 15 and 16 below:

Table 15 Time Taken to Heat Tank with Outside Air Temperature 10°C

In the Tank		
Energy required to heat the tank	460886.173	kJ
Heat Input to Tank <57.88°C	3.3627	kW
Heat Input to Tank >57.88°C	0.8877	kW
Heat Loss from Tank	0.126	kW
Time taken to Heat the Tank <57.88°C	34.43	hours
Time taken to Heat the Tank >57.88°C	21.76	hours
Total Heating Time	56.19	hours

Table 16 Time Taken to Heat Tank with Outside Air Temperature of -15°C

In the Tank		
Energy required to heat the tank	460886.173	kJ
Heat Input to Tank <57.88°C	3.2275	kW
Heat Input to Tank >57.88°C	0.4724	kW
Heat Loss from Tank	0.18327	kW
Time taken to Heat the Tank <57.88°C	36.61	hours
Time taken to Heat the Tank >57.88°C	57.33	hours
Total Heating Time	93.94	hours

Once the water has been heated up to its desired 65°C it can then be used for general consumption within the household. The amount of water available for use by the owners of the heat pump over a 24hr period can be calculated, based on a desire to keep the water temperature at 65°C.

$$Q_{24hrs} = (\dot{Q}_{Cond>57.88} - \dot{Q}_{Lost}) \times 60 \times 60 \times 24$$

$$m_{used} = \frac{Q_{24hrs}}{c_w \times \Delta T}$$

For the purposes of the model being used it is assumed that the incoming water will not be passing through quickly enough to cause the tank water to dip below the 57.88°C threshold, and so the value for the heat added to the water during the condenser has been taken to be when the refrigerant leaves the condenser at 65°C.

These give the values below for the mass that can be used in a 24 hr period:

Table 17 Mass of Water Available at Outside Air Temperature -15°C

Mass of Water Available for use with an Outside Air Temperature -15°C		
Heat Added to the Water over 24hrs	24976.99	kJ
Mass of Water 'Used over 24 hrs'	99.35	kg

Table 178 Mass of Water Available at Outside Air Temperature 10°C

Mass of Water Available for use with an Outside Air Temperature 10°C		
Heat Added to the Water over 24hrs	65812.2	kJ
Mass of Water 'Used over 24 hrs'	285.58	kg

Appendix 4 – Pressure Losses due to Bending in Pipes

$$z_2 + \frac{p_2}{\rho g} + \frac{v_2^2}{2g} + h_f = z_1 + \frac{p_1}{\rho g} + \frac{v_1^2}{2g} \quad (1)$$

Equation (1) describes the energy of a fluid flowing in a pipe, where

v_1=the initial velocity,

v_2=the final velocity,

z_1=the initial height,

z_2=the final height,

p_1=the initial pressure (=160000 Pa),

p_2=the final pressure,

h_f=the head loss due to friction,

ρ=density of fluid,

g=gravitational constant (=9.81).

It can be assumed that the difference in height is zero for the fluid travelling around an elbow as the loss in pressure due to the miniscule change in height is negligible compared to pressure loss from navigating a corner and friction in the pipes. It can also be assumed that the fluid maintains a constant velocity throughout. Therefore, equation (1) can be simplified to equation (2) below:

$$\frac{p_2}{\rho g} + h_f = \frac{p_1}{\rho g} \quad (2)$$

h_f can be calculated using equation (3):

$$h_f = f.\frac{L}{D}.\frac{v^2}{2g} \quad (3)$$

Where L/D is the ratio between the diameter of the pipe and the radius of the bend, and f the friction factor.

For a 90° pipe elbow, this L/D=30 [20]. f can be found by using a moody chart, knowing the roughness of a copper pipe (0.0014m) allows the relative roughness to be calculated.

To use a moody chart to calculate the friction factor, the Reynold's number associated with the flow is required. This can be calculated using Equation (4):

$$R_e = \frac{vD\rho}{\mu} \quad (4)$$

Where μ=dynamic viscosity of the fluid (=15.36x10-6 kg/ms).

Using the previously found volume flow rate of 0.000301m^3/s, and the cross sectional area of 7.855x10-5 m^2, the velocity of the fluid is 3.84m/s.

The Reynold's number can be calculated to be: 78336.7, giving a friction factor value of 0.00197 and a h_f of 0.887.

The outlet pressure can now be calculated for the pipe elbows by using Equation (3) to find the head loss and rearranging Equation (2) to find P_2.

The inlet pressure will be 160 kPa, which results in an outlet pressure of 159.5kPa after the 3 elbow turns, meaning a pressure loss of 546 Pa due to the elbow turns.

To calculate the pressure loss due to 4 spiral turns in the heat exchanger, it can be modelled as the equivalent of 16 90° turns. For the spiral, the radius of the bend is approximately 200mm and the diameter of the pipe is 10mm. This radius to diameter ratio results in an L/D value of 50 [20], giving a new total h_f of 10.434. Again it can be assumed that the velocity of the fluid remains constant however, there will be a change in height from the top of the tower to the bottom of the heat exchanger of 8m. Rearranging equation (1), the pressure at the bottom of the heat exchanger can be calculated, using the outlet pressure of the 3 elbows, found above, as p_1 in equation (5) below:

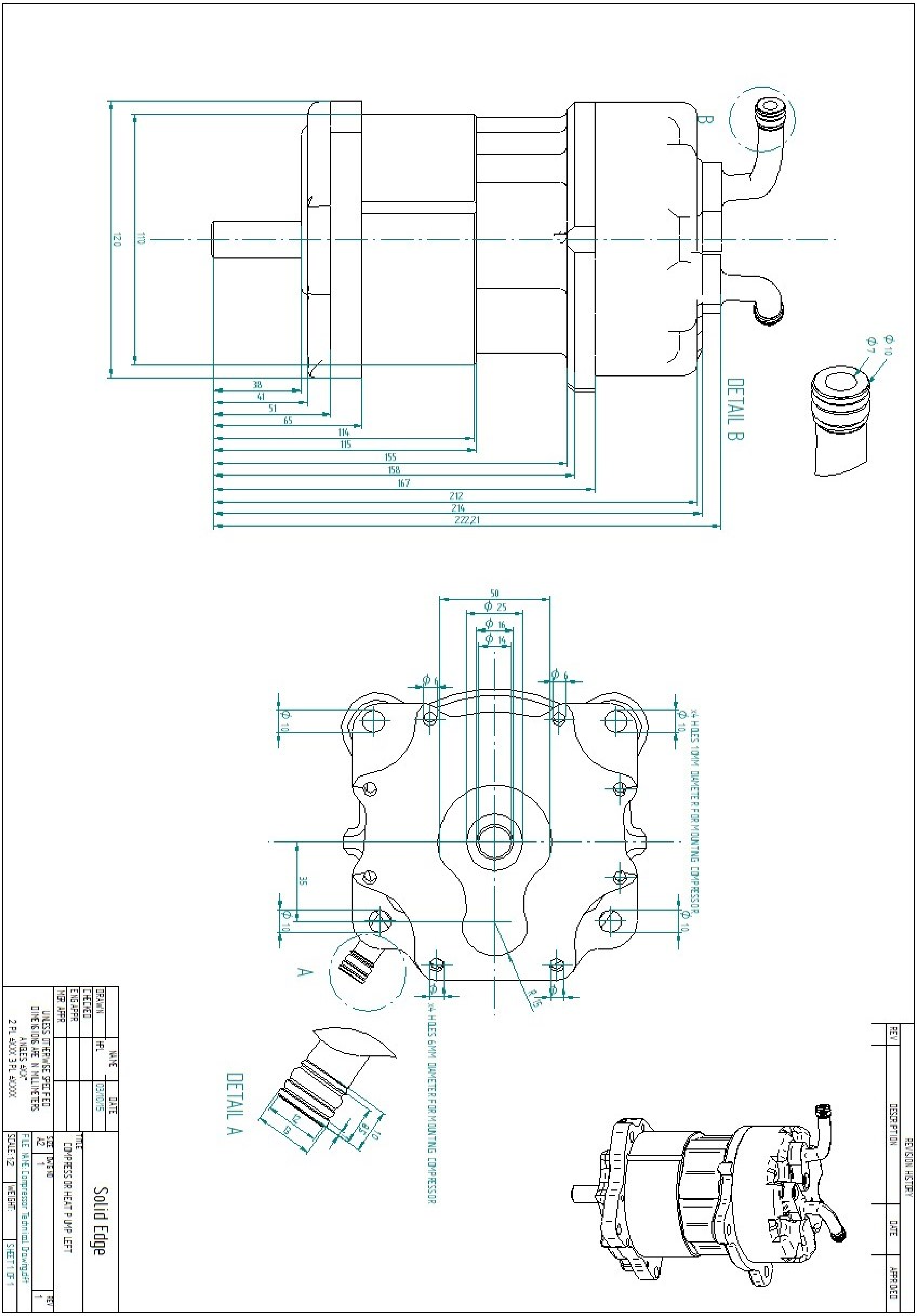
$$p_2 = \left(\frac{p_1}{\rho g} - h_f + z_1 - z_2\right) * \rho g \qquad (5)$$

Using Equation (5), the pressure at the bottom of the heat exchanger will be 156724 Pa.

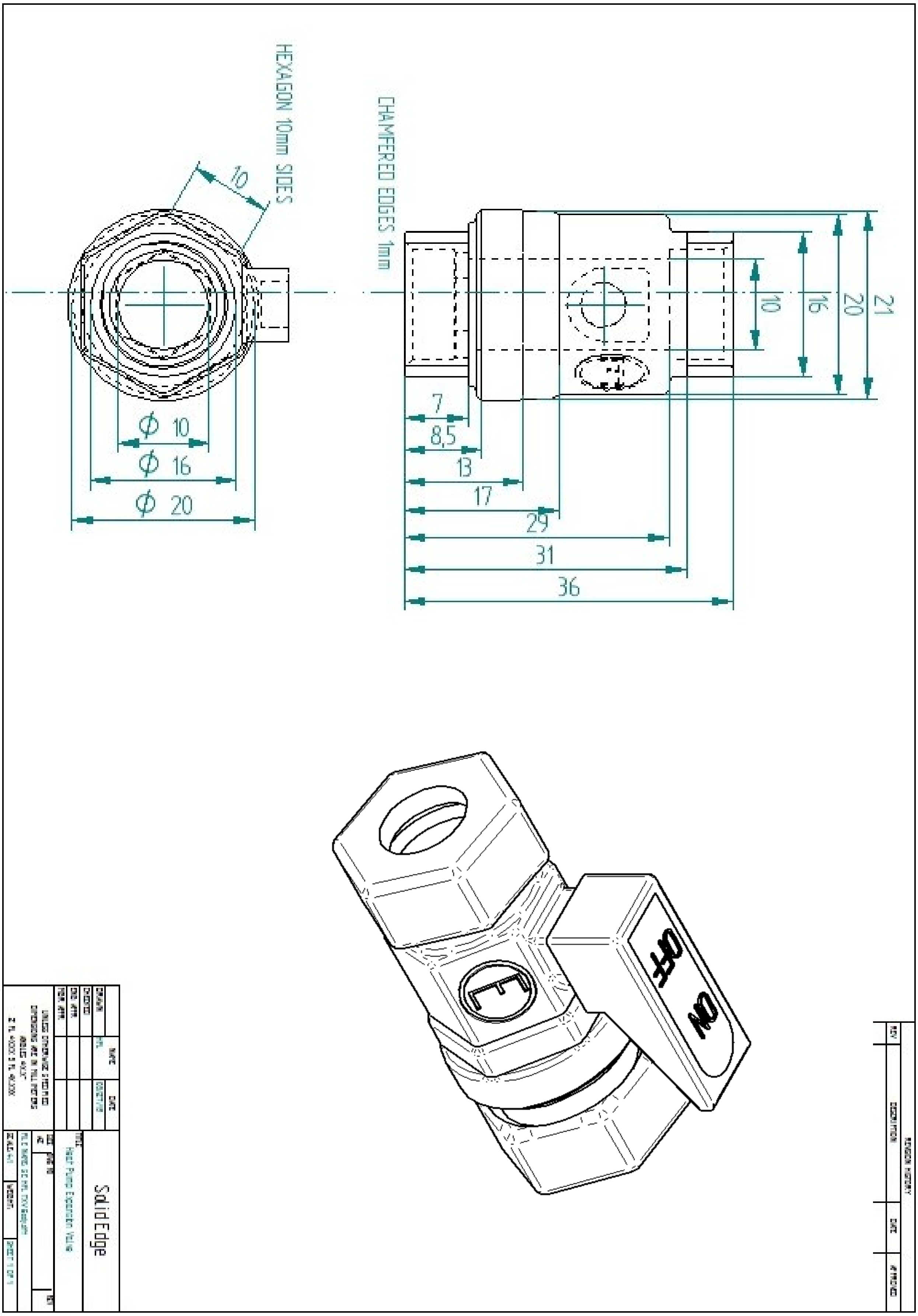
This loss in the pipes due to the spiral, 6422.644 Pa and the 3 elbows, 546.1986 Pa, is partially offset by an increase in pressure due to the change in height of 4924.306 Pa.

This will be a sufficient pressure to drive the refrigerant up the tower.

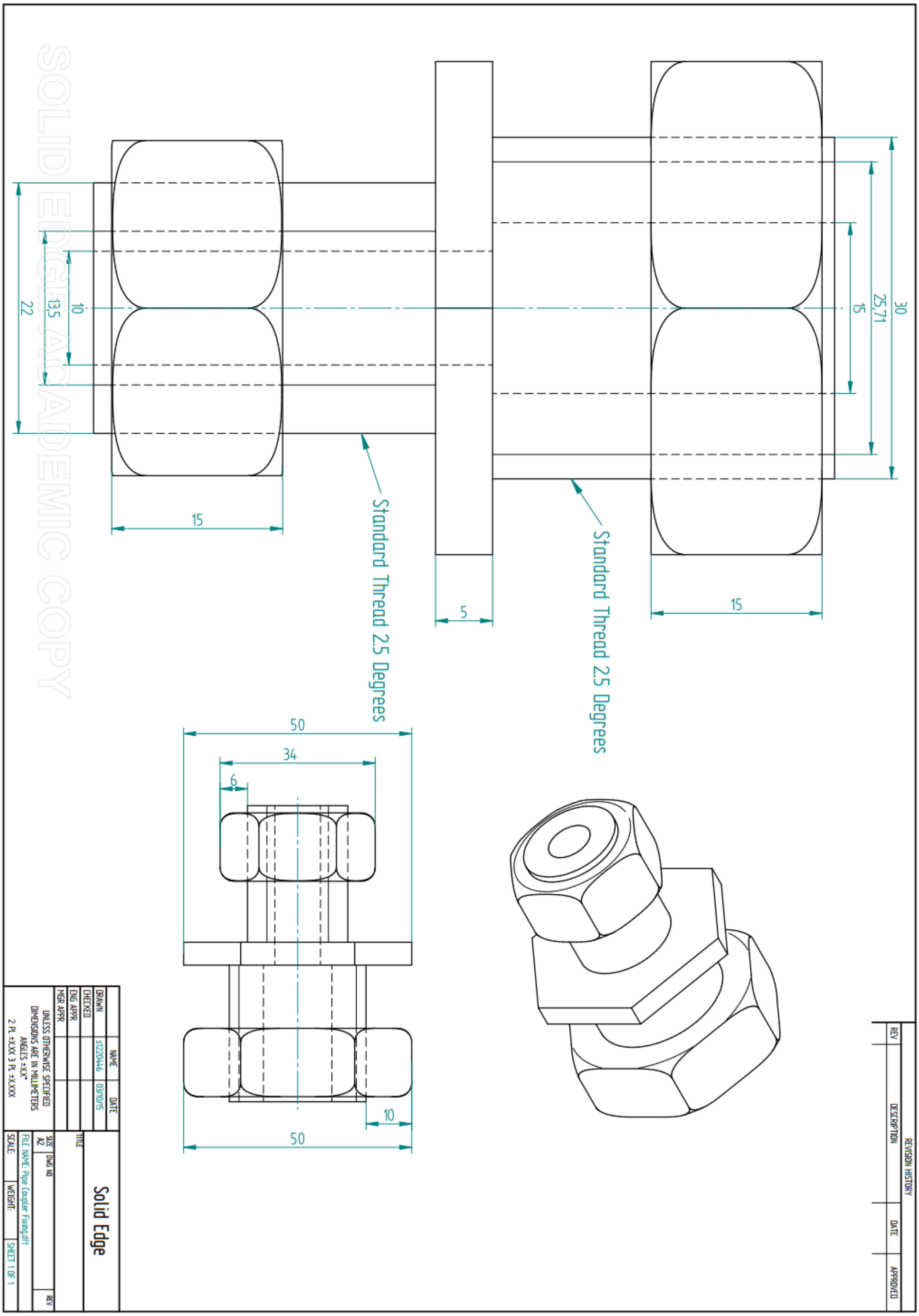
Appendix 5 – Technical Drawing of Compressor



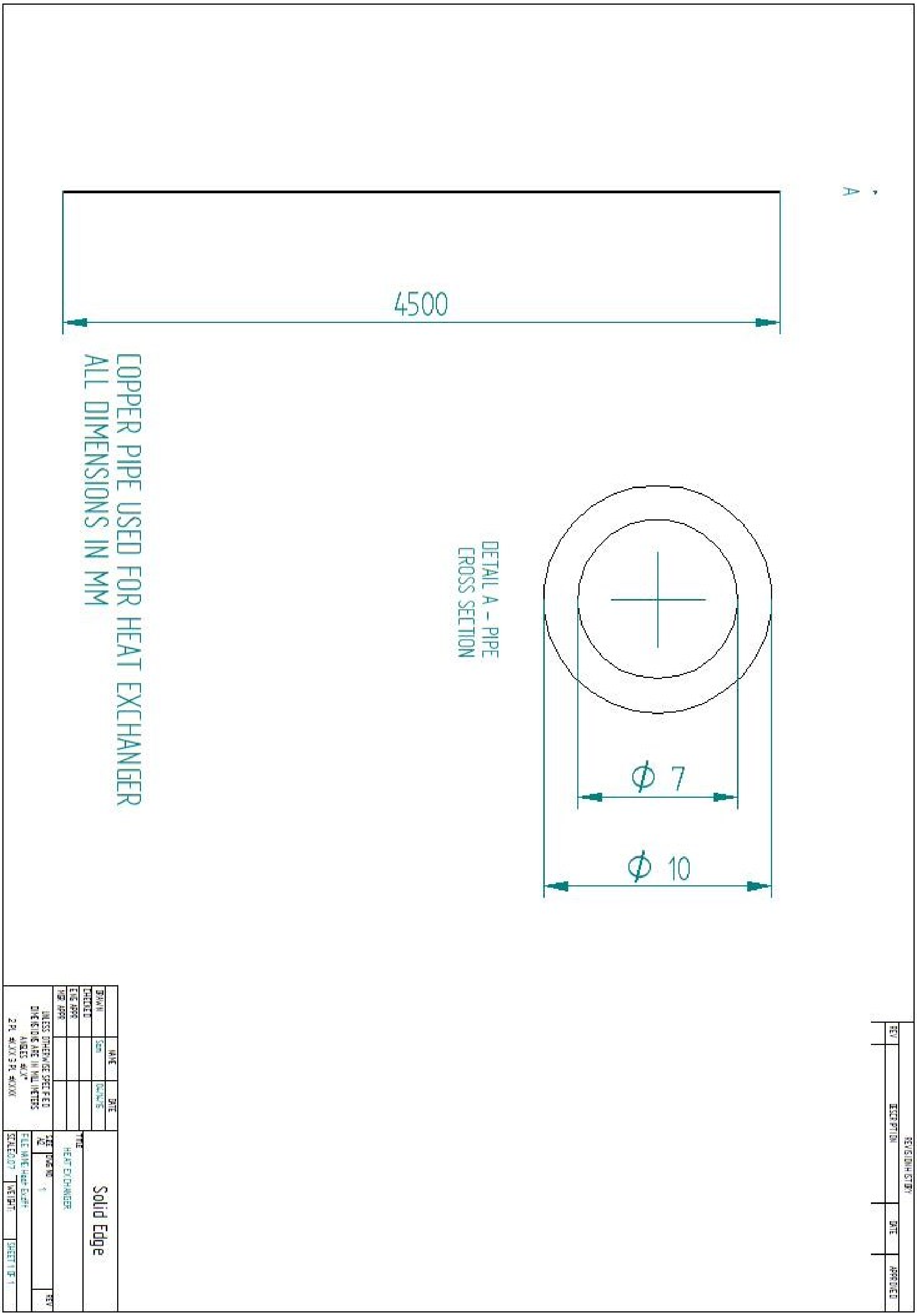
Appendix 6 – Technical Drawing of Expansion Valve



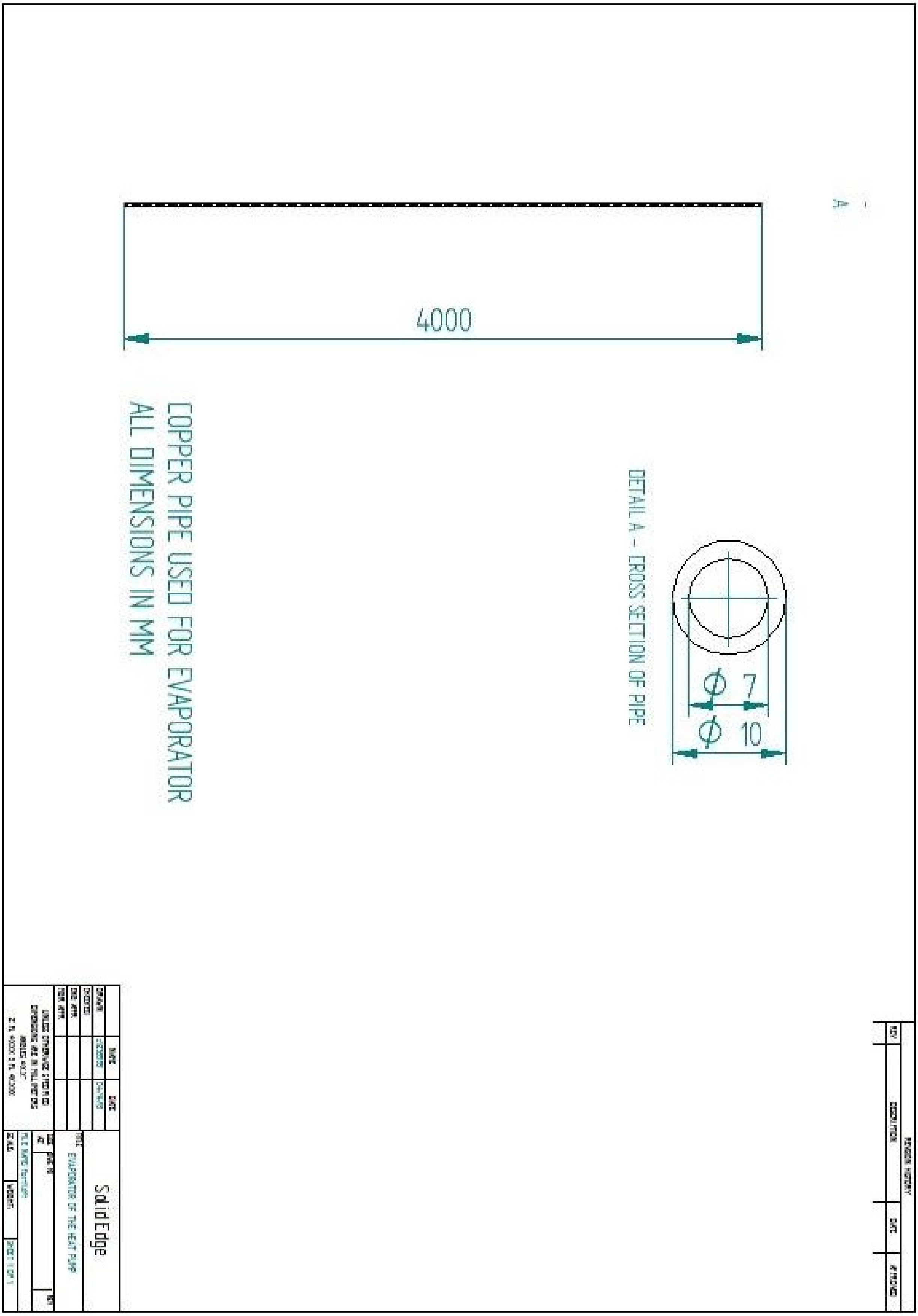
Appendix 7 – Technical Drawing of Coupler



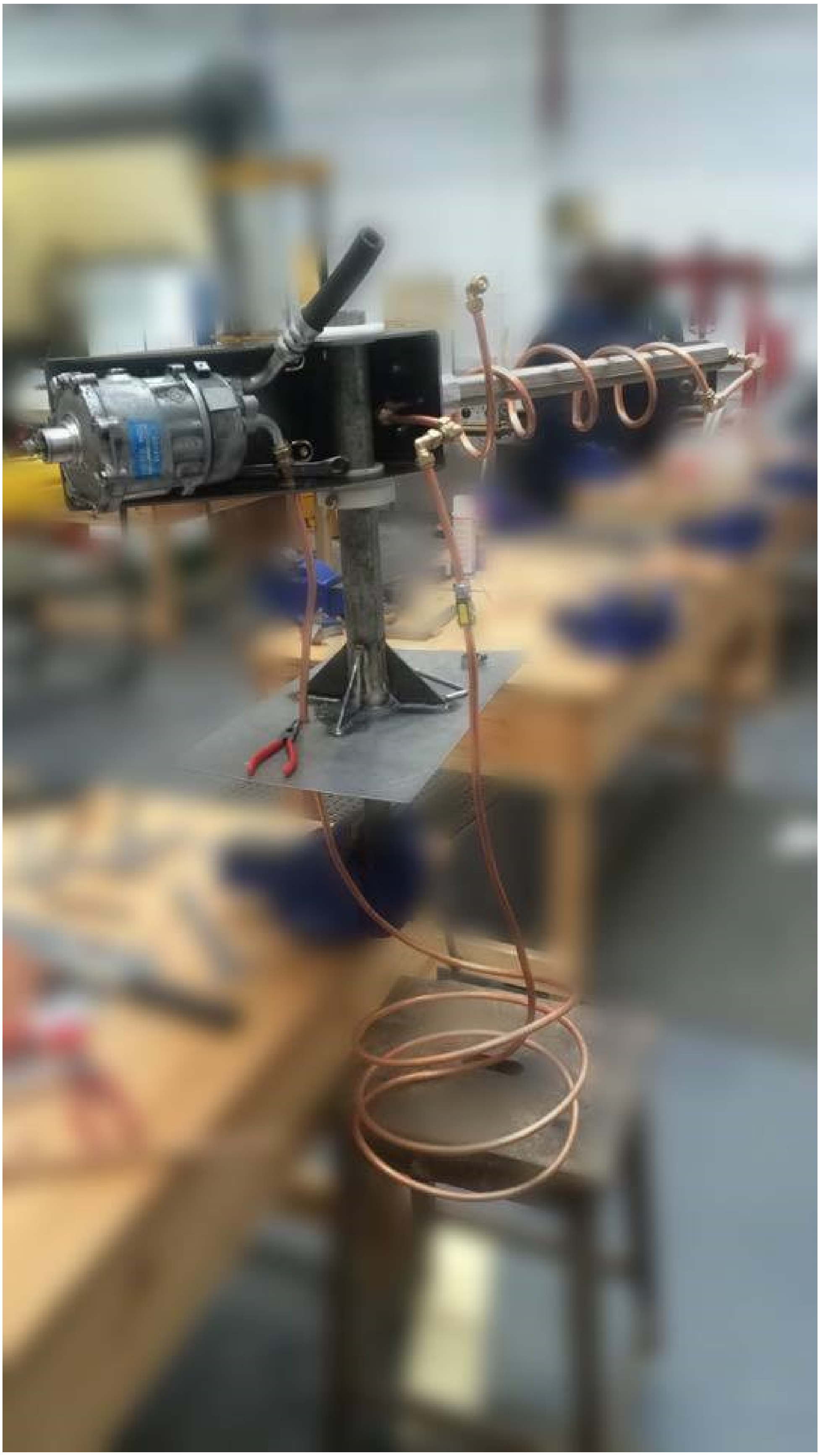
Appendix 8 – Technical Drawing of Heat Exchanger



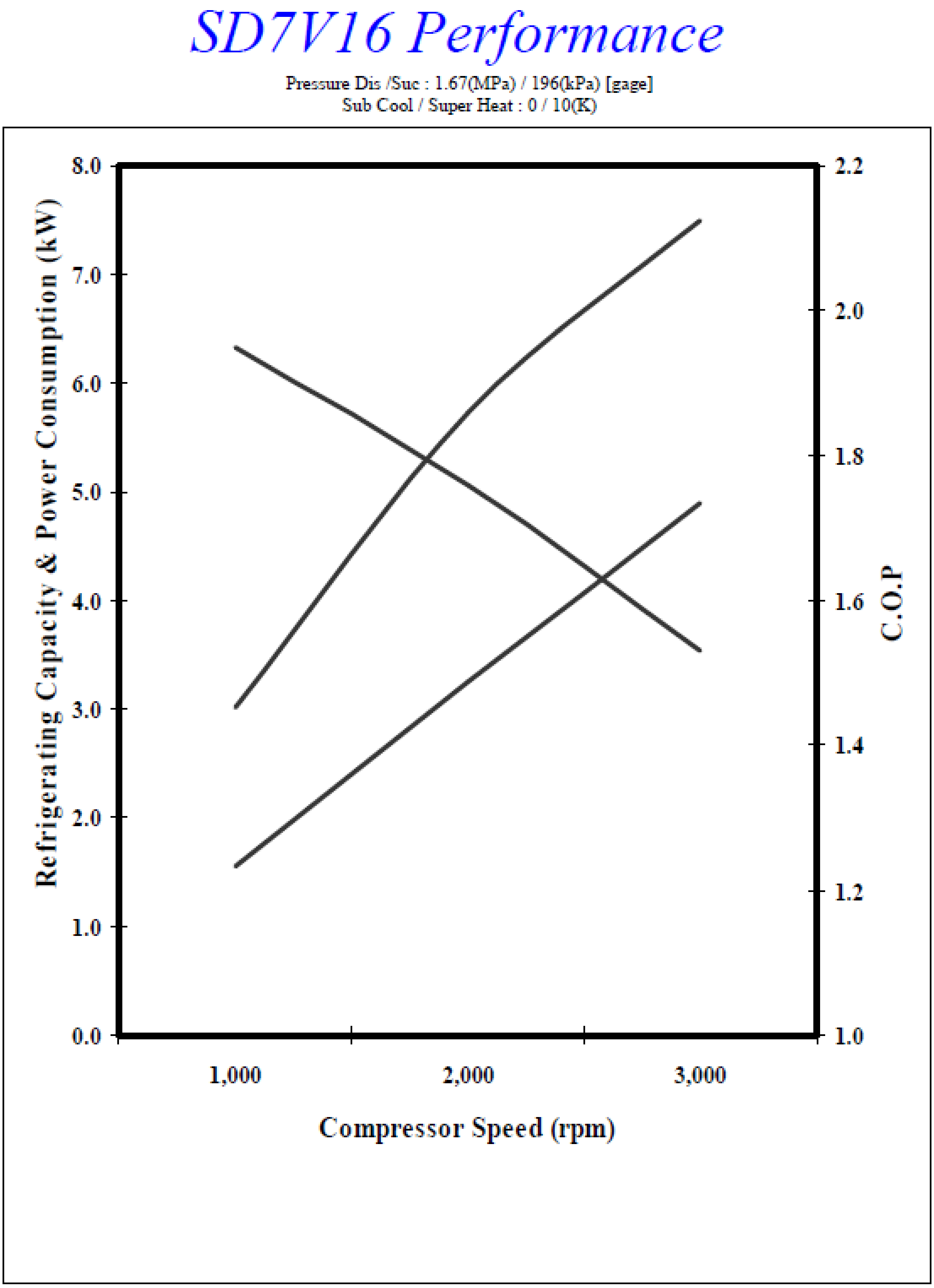
Appendix 9 – Technical Drawing of Evaporator



Appendix 10 – Photograph of Heat Pump Prototype



Appendix 11 – Compressor Performance Chart



Technical Information

T-134a—SI

DuPont™ Suva®
refrigerants

Thermodynamic Properties of HFC-134a

(1,1,1,2-tetrafluoroethane)

DuPont Product Names:

- DuPont™ Suva® 134a Refrigerant
- DuPont™ Formacel® Z-4 Blowing Agent
- DuPont™ Dymel® 134a Aerosol Propellant
- DuPont™ Dymel® 134a/P Aerosol Propellant (Pharmaceutical Grade)

The miracles of science™

TABLE 2 (continued)
HFC-134a Superheated Vapor—Constant Pressure Tables

V = Volume in m^3/kg H = Enthalpy in kJ/kg S = Entropy in kJ/(kg)(K) v_s = Velocity of Sound in m/sec
Cp = Heat Capacity at Constant Pressure in kJ/(kg)(°C) Cp/Cv = Heat Capacity Ratio (Dimensionless)

TEMP °C	PRESSURE = 90.00 kPa (abs) V	H	S	Cp	Cp/Cv	v_s		PRESSURE = 100.00 kPa (abs) V	H	S	Cp	Cp/Cv	v_s	TEMP °C
−28.61	0.00072	162.7	0.8564	1.2662	1.5041	756.7	SAT LIQ	0.00073	165.6	0.8681	1.2715	1.5046	746.2	−26.34
−28.61	0.21254	381.4	1.7509	0.7800	1.1527	145.4	SAT VAP	0.19246	382.8	1.7484	0.7876	1.1539	145.7	−26.34
−25	0.21622	384.3	1.7624	0.7835	1.1492	146.6		0.19372	383.9	1.7527	0.7888	1.1525	146.1	−25
−20	0.22129	388.2	1.7780	0.7888	1.1448	148.3		0.19829	387.9	1.7685	0.7935	1.1477	147.8	−20
−15	0.22624	392.1	1.7935	0.7946	1.1408	149.9		0.20284	391.8	1.7840	0.7988	1.1434	149.5	−15
−10	0.23121	396.1	1.8088	0.8007	1.1372	151.5		0.20734	395.8	1.7994	0.8045	1.1395	151.1	−10
−5	0.23613	400.2	1.8240	0.8071	1.1339	153.0		0.21182	399.9	1.8146	0.8106	1.1359	152.7	−5
0	0.24102	404.2	1.8389	0.8138	1.1308	154.5		0.21626	404.0	1.8297	0.8169	1.1327	154.2	0
5	0.24588	408.3	1.8538	0.8207	1.1279	156.0		0.22065	408.1	1.8445	0.8235	1.1296	155.7	5
10	0.25069	412.4	1.8684	0.8278	1.1253	157.5		0.22502	412.2	1.8593	0.8304	1.1269	157.2	10
15	0.25549	416.6	1.8830	0.8350	1.1229	159.0		0.22941	416.4	1.8739	0.8373	1.1243	158.7	15
20	0.26028	420.8	1.8974	0.8423	1.1206	160.4		0.23370	420.6	1.8883	0.8445	1.1218	160.1	20
25	0.26504	425.0	1.9117	0.8498	1.1184	161.8		0.23804	424.8	1.9027	0.8517	1.1196	161.6	25
30	0.26976	429.3	1.9259	0.8573	1.1164	163.2		0.24231	429.1	1.9169	0.8591	1.1175	163.0	30
35	0.27450	433.6	1.9400	0.8649	1.1145	164.6		0.24661	433.4	1.9310	0.8665	1.1155	164.3	35
40	0.27925	437.9	1.9540	0.8725	1.1127	165.9		0.25088	437.7	1.9450	0.8741	1.1136	165.7	40
45	0.28393	442.3	1.9679	0.8802	1.1110	167.3		0.25517	442.1	1.9589	0.8816	1.1118	167.1	45
50	0.28860	446.7	1.9816	0.8879	1.1093	168.6		0.25940	446.6	1.9727	0.8892	1.1101	168.4	50
55	0.29334	451.2	1.9953	0.8956	1.1078	169.9		0.26364	451.0	1.9864	0.8968	1.1085	169.7	55
60	0.29797	455.7	2.0089	0.9033	1.1063	171.2		0.26788	455.5	2.0001	0.9045	1.1070	171.0	60
65	0.30266	460.2	2.0225	0.9110	1.1049	172.5		0.27203	460.1	2.0136	0.9121	1.1055	172.3	65
70	0.30731	464.8	2.0359	0.9188	1.1036	173.8		0.27624	464.7	2.0270	0.9198	1.1042	173.6	70
75	0.31201	469.4	2.0492	0.9265	1.1023	175.0		0.28043	469.3	2.0404	0.9274	1.1028	174.9	75
80	0.31666	474.0	2.0625	0.9342	1.1010	176.3		0.28466	473.9	2.0537	0.9350	1.1016	176.1	80
85	0.32123	478.7	2.0757	0.9418	1.0999	177.5		0.28885	478.6	2.0669	0.9427	1.1004	177.4	85
90	0.32595	483.5	2.0888	0.9495	1.0987	178.7		0.29300	483.4	2.0800	0.9503	1.0992	178.6	90
95	0.33058	488.2	2.1018	0.9571	1.0976	179.9		0.29718	488.1	2.0930	0.9578	1.0981	179.8	95
100	0.33523	493.0	2.1148	0.9647	1.0966	181.1		0.30139	492.9	2.1060	0.9654	1.0970	181.0	100
105	0.33979	497.9	2.1277	0.9722	1.0955	182.3		0.30553	497.8	2.1189	0.9729	1.0959	182.2	105
110	0.34447	502.8	2.1405	0.9798	1.0946	183.5		0.30969	502.7	2.1318	0.9804	1.0949	183.4	110
115	0.34904	507.7	2.1533	0.9872	1.0936	184.7		0.31387	507.6	2.1445	0.9878	1.0940	184.6	115
120	0.35361	512.6	2.1659	0.9947	1.0927	185.9		0.31797	512.5	2.1572	0.9952	1.0930	185.8	120
125	0.35817	517.6	2.1786	1.0021	1.0918	187.0		0.32216	517.5	2.1698	1.0026	1.0921	186.9	125

TEMP °C	PRESSURE = 101.325 kPa (abs) V	H	S	Cp	Cp/Cv	v_s		PRESSURE = 110.00 kPa (abs) V	H	S	Cp	Cp/Cv	v_s	TEMP °C
−26.06	0.00073	165.9	0.8696	1.2722	1.5046	744.9	SAT LIQ	0.00073	168.2	0.8788	1.2765	1.5051	736.6	−24.25
−26.06	0.19011	383.0	1.7481	0.7886	1.1540	145.7	SAT VAP	0.17593	384.1	1.7462	0.7948	1.1551	145.9	−24.25
−25	0.19106	383.9	1.7515	0.7895	1.1530	146.1		—	—	—	—	—	—	−25
−20	0.19562	387.8	1.7673	0.7942	1.1481	147.7		0.17953	387.5	1.7597	0.7984	1.1507	147.4	−20
−15	0.20012	391.8	1.7829	0.7994	1.1438	149.4		0.18369	391.5	1.7754	0.8031	1.1461	149.0	−15
−10	0.20454	395.8	1.7982	0.8050	1.1398	151.0		0.18783	395.6	1.7908	0.8084	1.1419	150.7	−10
−5	0.20894	399.9	1.8135	0.8110	1.1362	152.6		0.19194	399.6	1.8061	0.8141	1.1381	152.3	−5
0	0.21336	403.9	1.8285	0.8173	1.1329	154.2		0.19600	403.7	1.8212	0.8201	1.1346	153.9	0
5	0.21768	408.0	1.8434	0.8239	1.1299	155.7		0.20004	407.8	1.8361	0.8264	1.1314	155.4	5
10	0.22202	412.2	1.8581	0.8307	1.1271	157.2		0.20404	412.0	1.8509	0.8330	1.1284	156.9	10
15	0.22630	416.3	1.8727	0.8377	1.1244	158.6		0.20803	416.2	1.8656	0.8397	1.1257	158.4	15
20	0.23057	420.5	1.8872	0.8448	1.1220	160.1		0.21200	420.4	1.8801	0.8467	1.1231	159.9	20
25	0.23485	424.8	1.9015	0.8520	1.1197	161.5		0.21598	424.6	1.8945	0.8538	1.1208	161.3	25
30	0.23906	429.1	1.9158	0.8594	1.1176	162.9		0.21988	428.9	1.9087	0.8610	1.1186	162.7	30
35	0.24331	433.4	1.9299	0.8668	1.1156	164.3		0.22376	433.2	1.9228	0.8683	1.1165	164.1	35
40	0.24759	437.7	1.9439	0.8743	1.1137	165.7		0.22769	437.6	1.9369	0.8756	1.1145	165.5	40
45	0.25176	442.1	1.9578	0.8818	1.1119	167.0		0.23154	442.0	1.9508	0.8831	1.1127	166.9	45
50	0.25595	446.5	1.9716	0.8894	1.1102	168.4		0.23546	446.4	1.9646	0.8905	1.1109	168.2	50
55	0.26015	451.0	1.9853	0.8970	1.1086	169.7		0.23929	450.9	1.9784	0.8981	1.1093	169.5	55
60	0.26427	455.5	1.9989	0.9046	1.1071	171.0		0.24313	455.4	1.9920	0.9056	1.1077	170.8	60
65	0.26846	460.1	2.0125	0.9123	1.1056	172.3		0.24697	459.9	2.0055	0.9132	1.1062	172.1	65
70	0.27263	464.6	2.0259	0.9199	1.1042	173.6		0.25088	464.5	2.0190	0.9208	1.1048	173.4	70
75	0.27678	469.3	2.0393	0.9275	1.1029	174.8		0.25465	469.2	2.0324	0.9283	1.1034	174.7	75
80	0.28090	473.9	2.0526	0.9352	1.1016	176.1		0.25853	473.8	2.0457	0.9359	1.1021	176.0	80
85	0.28506	478.6	2.0658	0.9428	1.1004	177.3		0.26233	478.5	2.0589	0.9435	1.1009	177.2	85
90	0.28918	483.3	2.0789	0.9504	1.0992	178.6		0.26610	483.2	2.0720	0.9510	1.0997	178.4	90
95	0.29326	488.1	2.0919	0.9579	1.0981	179.8		0.26998	488.0	2.0851	0.9586	1.0985	179.7	95
100	0.29735	492.9	2.1049	0.9655	1.0970	181.0		0.27375	492.8	2.0981	0.9661	1.0974	180.9	100
105	0.30157	497.8	2.1178	0.9730	1.0960	182.2		0.27755	497.7	2.1110	0.9735	1.0963	182.1	105
110	0.30562	502.6	2.1306	0.9804	1.0950	183.4		0.28129	502.6	2.1238	0.9810	1.0953	183.3	110
115	0.30969	507.6	2.1434	0.9879	1.0940	184.6		0.28514	507.5	2.1366	0.9884	1.0943	184.5	115
120	0.31377	512.5	2.1561	0.9953	1.0931	185.7		0.28893	512.5	2.1493	0.9958	1.0934	185.6	120
125	0.31797	517.5	2.1687	1.0026	1.0922	186.9		0.29265	517.5	2.1619	1.0031	1.0924	186.8	125
130	—	—	—	—	—	—		0.29647	522.5	2.1745	1.0104	1.0915	188.0	130

TABLE 2 (continued)
HFC-134a Superheated Vapor—Constant Pressure Tables

V = Volume in m^3/kg H = Enthalpy in kJ/kg S = Entropy in kJ/(kg)(K) v_s = Velocity of Sound in m/sec
Cp = Heat Capacity at Constant Pressure in kJ/(kg)(°C) Cp/Cv = Heat Capacity Ratio (Dimensionless)

TEMP °C	PRESSURE = 1500.00 kPa (abs) V	H	S	Cp	Cp/Cv	v_s		PRESSURE = 1600.00 kPa (abs) V	H	S	Cp	Cp/Cv	v_s	TEMP °C
55.2	0.00093	280.1	1.2632	1.6205	1.6778	365.0	SAT LIQ	0.00094	284.5	1.2759	1.6468	1.6959	351.6	57.88
55.2	0.01308	425.7	1.7063	1.2844	1.3876	134.2	SAT VAP	0.01215	426.5	1.7050	1.3227	1.4142	132.8	57.88
60	0.01363	431.7	1.7246	1.2358	1.3416	138.0		0.01239	429.3	1.7134	1.2953	1.3888	134.7	60
65	0.01417	437.8	1.7427	1.1989	1.3056	141.7		0.01294	435.6	1.7323	1.2448	1.3412	138.7	65
70	0.01468	443.7	1.7601	1.1715	1.2776	145.0		0.01344	441.7	1.7503	1.2083	1.3056	142.3	70
75	0.01516	449.5	1.7769	1.1507	1.2552	148.0		0.01392	447.7	1.7675	1.1811	1.2779	145.6	75
80	0.01562	455.2	1.7932	1.1349	1.2368	150.9		0.01437	453.6	1.7842	1.1604	1.2556	148.7	80
85	0.01606	460.9	1.8090	1.1228	1.2214	153.6		0.01480	459.3	1.8004	1.1447	1.2373	151.5	85
90	0.01649	466.4	1.8245	1.1136	1.2084	156.1		0.01522	465.0	1.8162	1.1325	1.2220	154.2	90
95	0.01690	472.0	1.8397	1.1067	1.1972	158.5		0.01562	470.6	1.8316	1.1233	1.2090	156.8	95
100	0.01731	477.5	1.8546	1.1016	1.1875	160.8		0.01602	476.2	1.8467	1.1163	1.1978	159.2	100
105	0.01770	483.0	1.8692	1.0982	1.1790	163.0		0.01640	481.8	1.8616	1.1112	1.1881	161.5	105
110	0.01809	488.5	1.8837	1.0959	1.1714	165.2		0.01677	487.4	1.8761	1.1077	1.1796	163.7	110
115	0.01847	494.0	1.8978	1.0948	1.1647	167.2		0.01714	492.9	1.8905	1.1054	1.1721	165.9	115
120	0.01885	499.4	1.9119	1.0945	1.1587	169.2		0.01750	498.4	1.9046	1.1042	1.1653	167.9	120
125	0.01921	504.9	1.9257	1.0950	1.1532	171.1		0.01785	503.9	1.9186	1.1038	1.1593	169.9	125
130	0.01958	510.4	1.9394	1.0962	1.1483	173.0		0.01820	509.5	1.9323	1.1042	1.1539	171.8	130
135	0.01993	515.9	1.9529	1.0979	1.1438	174.8		0.01854	515.0	1.9460	1.1053	1.1489	173.7	135
140	0.02028	521.4	1.9663	1.1001	1.1397	176.6		0.01887	520.5	1.9594	1.1069	1.1444	175.5	140
145	0.02063	526.9	1.9795	1.1027	1.1360	178.3		0.01921	526.0	1.9728	1.1090	1.1403	177.3	145
150	0.02098	532.4	1.9926	1.1057	1.1325	180.0		0.01954	531.6	1.9860	1.1115	1.1365	179.0	150
155	0.02132	537.9	2.0056	1.1090	1.1293	181.6		0.01986	537.2	1.9990	1.1144	1.1330	180.7	155
160	0.02166	543.5	2.0185	1.1125	1.1263	183.2		0.02018	542.7	2.0120	1.1176	1.1298	182.4	160
165	0.02199	549.1	2.0313	1.1163	1.1235	184.8		0.02050	548.3	2.0248	1.1211	1.1268	184.0	165
170	0.02233	554.7	2.0440	1.1203	1.1209	186.3		0.02082	554.0	2.0376	1.1248	1.1240	185.5	170
175	0.02265	560.3	2.0566	1.1244	1.1185	187.9		0.02113	559.6	2.0502	1.1287	1.1214	187.1	175
180	0.02298	565.9	2.0691	1.1288	1.1162	189.3		0.02144	565.2	2.0627	1.1328	1.1190	188.6	180
185	0.02331	571.6	2.0815	1.1332	1.1141	190.8		0.02175	570.9	2.0752	1.1370	1.1167	190.1	185
190	0.02363	577.2	2.0939	1.1378	1.1121	192.3		0.02206	576.6	2.0876	1.1414	1.1145	191.6	190
195	0.02395	582.9	2.1061	1.1425	1.1102	193.7		0.02236	582.3	2.0998	1.1458	1.1125	193.1	195
200	0.02427	588.7	2.1183	1.1472	1.1084	195.1		0.02267	588.1	2.1121	1.1504	1.1106	194.5	200
205	0.02459	594.4	2.1303	1.1520	1.1067	196.5		0.02297	593.8	2.1242	1.1551	1.1088	195.9	205
210	0.02491	600.2	2.1424	1.1569	1.1050	197.8		0.02327	599.6	2.1362	1.1598	1.1070	197.3	210

TEMP °C	PRESSURE = 1700.00 kPa (abs) V	H	S	Cp	Cp/Cv	v_s		PRESSURE = 1800.00 kPa (abs) V	H	S	Cp	Cp/Cv	v_s	TEMP °C
60.43	0.00095	288.6	1.2882	1.6746	1.7155	338.6	SAT LIQ	0.00096	292.6	1.2999	1.7040	1.7366	326.1	62.87
60.43	0.01132	427.2	1.7037	1.3635	1.4432	131.4	SAT VAP	0.01058	427.8	1.7022	1.4072	1.4749	130.0	62.87
65	0.01183	433.3	1.7217	1.3004	1.3851	135.5		0.01082	430.8	1.7110	1.3696	1.4408	132.1	65
70	0.01234	439.7	1.7405	1.2514	1.3390	139.5		0.01134	437.4	1.7306	1.3029	1.3794	136.5	70
75	0.01281	445.8	1.7583	1.2158	1.3042	143.1		0.01182	443.8	1.7491	1.2559	1.3352	140.5	75
80	0.01326	451.8	1.7754	1.1891	1.2770	146.4		0.01227	450.0	1.7667	1.2215	1.3016	144.0	80
85	0.01369	457.7	1.7920	1.1687	1.2551	149.5		0.01269	456.1	1.7837	1.1957	1.2752	147.3	85
90	0.01410	463.5	1.8081	1.1532	1.2371	152.3		0.01309	462.0	1.8001	1.1760	1.2538	150.3	90
95	0.01449	469.3	1.8238	1.1412	1.2219	155.0		0.01348	467.8	1.8161	1.1607	1.2361	153.2	95
100	0.01487	474.9	1.8391	1.1321	1.2091	157.5		0.01386	473.6	1.8317	1.1491	1.2213	155.8	100
105	0.01525	480.6	1.8541	1.1252	1.1980	159.9		0.01422	479.3	1.8469	1.1401	1.2087	158.4	105
110	0.01561	486.2	1.8689	1.1202	1.1884	162.3		0.01457	485.0	1.8618	1.1334	1.1978	160.8	110
115	0.01596	491.8	1.8834	1.1166	1.1799	164.5		0.01491	490.7	1.8765	1.1284	1.1882	163.1	115
120	0.01631	497.4	1.8976	1.1143	1.1724	166.6		0.01524	496.3	1.8909	1.1249	1.1798	165.3	120
125	0.01665	502.9	1.9117	1.1130	1.1657	168.7		0.01557	501.9	1.9051	1.1226	1.1724	167.4	125
130	0.01698	508.5	1.9256	1.1126	1.1597	170.7		0.01589	507.5	1.9191	1.1214	1.1657	169.5	130
135	0.01730	514.1	1.9393	1.1130	1.1542	172.6		0.01621	513.1	1.9329	1.1210	1.1598	171.5	135
140	0.01763	519.6	1.9529	1.1140	1.1493	174.5		0.01652	518.7	1.9466	1.1213	1.1544	173.4	140
145	0.01795	525.2	1.9663	1.1156	1.1448	176.3		0.01682	524.3	1.9601	1.1223	1.1495	175.3	145
150	0.01826	530.8	1.9796	1.1176	1.1407	178.1		0.01713	530.0	1.9734	1.1239	1.1450	177.1	150
155	0.01857	536.4	1.9927	1.1201	1.1369	179.8		0.01742	535.6	1.9866	1.1259	1.1409	178.9	155
160	0.01888	542.0	2.0057	1.1229	1.1334	181.5		0.01772	541.2	1.9997	1.1283	1.1371	180.6	160
165	0.01918	547.6	2.0186	1.1260	1.1302	183.1		0.01801	546.9	2.0127	1.1310	1.1336	182.3	165
170	0.01949	553.2	2.0314	1.1294	1.1272	184.8		0.01830	552.5	2.0256	1.1341	1.1304	184.0	170
175	0.01978	558.9	2.0441	1.1330	1.1244	186.4		0.01859	558.2	2.0383	1.1374	1.1274	185.6	175
180	0.02008	564.6	2.0567	1.1368	1.1218	187.9		0.01887	563.9	2.0509	1.1410	1.1246	187.2	180
185	0.02037	570.3	2.0692	1.1408	1.1193	189.4		0.01915	569.6	2.0635	1.1447	1.1220	188.7	185
190	0.02066	576.0	2.0816	1.1450	1.1170	190.9		0.01943	575.4	2.0759	1.1487	1.1196	190.3	190
195	0.02096	581.7	2.0939	1.1493	1.1149	192.4		0.01971	581.1	2.0883	1.1528	1.1173	191.8	195
200	0.02124	587.5	2.1062	1.1537	1.1128	193.9		0.01998	586.9	2.1005	1.1570	1.1151	193.3	200
205	0.02153	593.3	2.1183	1.1582	1.1109	195.3		0.02026	592.7	2.1127	1.1613	1.1130	194.7	205
210	0.02182	599.1	2.1304	1.1628	1.1091	196.7		0.02053	598.5	2.1248	1.1658	1.1111	196.2	210
215	0.02210	604.9	2.1424	1.1674	1.1073	198.1		0.02080	604.3	2.1369	1.1703	1.1093	197.6	215

Appendix 13 – Project Management Deadlines

12/01/2015	19/01/2015	26/01/2015	02/02/2015	09/02/2015	16/02/2015	23/02/2015	02/03/2015	09/03/2015	16/03/2015	23/03/2015	30/03/2015
					ILW						
General Research on Heat Pump theory											
Consider calculations for heat exchanger	Begin calculations for heat exchanger										
		CAD of Rotation of Pipes		Calculations for Condenser							
Compressor Research				Choose Compressor- find pressure/temp ratios etc. Required torque							
	Research Expansion/Throttling Valves		Begin TXV design- CAD and calculations								
	Research into Control System					Design Control System					
						Start making hardwear - CAD drawings finished - ideas for manufacturing			Ordering parts for Prototype	Prototype Building	
						Start Report		Justifications for manufacturing methods	Report illustrations & tables finished. BOM.	Cost estimate for project & Full references for report	Report work- formatting, proofing etc
Assign Sections & Create Timeline.	Sub Group Managers Meeting. Arrange Group Meeting. Consider next points	Arrange group meeting. Manage Project				Everything together- collab other groups & check everybody's parts work		Collect BOM for prototype	Order parts for prototype	Oversee finalising of report and production of prototype. Keep up team spirits.	Submit Report