DESIGN AND CONSTRUCTION OF A NOVEL DIRECT EXPANSION SOLAR ASSISTED HEAT PUMP

TEAM 2

Project Team:	1. Charuka Marasinghe	30023058
	2. Dhruvi Patel	30037478
	3. Edwin Pulikkottil	30024426
	4. Jessica Sambi	30016532
	5. Kerwin Virtusio	30037455
	6. Nadia Elgamal	30017358

Course Instructors: Dr. Alex Rameriez-Serrano

Teaching Assistant: Danny Wong

Project Sponsors:
1. Dr. Aggrey Mwesigye
2. Dr. Simon Li

Department of Mechanical and Manufacturing Engineering

ENME 501

December 10, 2022

University of Calgary

Executive Summary

This report will explore the research, calculations, and engineering design as the preliminary work required before constructing a Direct Expansion Solar Assisted Heat Pump. These systems combine the benefits of solar energy and heat pump technology to provide hot water for domestic use with high efficiencies. With Calgary being the sunniest city in Canada [1], there is a vast under-utilization of solar energy. This report outlines a heat pump design that utilizes solar energy to heat water for residential use. A parametric study was performed to determine the optimum solar collector configuration. Subsequently, a component matching study was conducted to find compatible compressor, condenser, and expansion valve components. The results from these studies are presented in this report with plans for producing a working model next semester.

Acknowledgements

The Direct Expansion Solar Assisted Heat Pump project is supported by the Department of Mechanical and Manufacturing Engineering at the University of Calgary. The authors would like to acknowledge Dr. Aggrey Mwesigye, Dr. Simon Li, and Dr. Ron Hugo for providing their technical expertise in support of the development of this project. The authors are also grateful to industry professional David Barret of Klass Mechanical for providing insights crucial to the successful manufacturing of The Direct Expansion Solar Assisted Heat Pump system.

Table of Contents

List of Figures

List of Tables

List of Symbols and Variables

Greek Letters

α	absorbance
α_L	linear expansion coefficient $[1/K]$
β	tilt angle of collector
δ	absorber plate thickness [m]
ε	emissivity
μ	viscosity $[Pa \cdot s]$
ν	kinematic viscosity $[m^2/s]$
ρ	density $[kg/m^3]$
σ	Boltzmann's constant
τ	transmittance

Capital Letters

A	collector area $[m^2]$
C_b	bond conductance $[W/mK]$
C_p	thermal capacitance of circulating fluid
D	outside pipe diameter $[m]$
D_i	inside pipe diameter $[m]$
F'	collector efficiency factor
FR	collector heat removal factor
$h_{c,p-c}$	convection heat transfer coefficient between absorber plate and cover
h_{fi}	heat transfer coefficient between fluid and tube wall
h_w	convection heat transfer coefficient between the cover and the ambient
I	irradiance $[W/m^2]$
K_L	loss coefficient
L_c	collector length $[m]$
L_p	length of pipe $[m]$
P_e	Pressure drop due to elbows $[Pa]$
P_f	pressure drop due to friction $[Pa]$
Q	heat transfer $[W]$

Re	Reynolds number
T	temperature $[K]$
U	heat loss coefficient $[W/m^2K]$
V_w	wind speed $[m/s]$
W	tube centre to centre distance $[m]$

Lowercase Letters

e	absolute pipe roughness [m]
f	friction factor
g	gravity $[m/s^2]$
h	heat loss coefficient
k	conductivity
$\dot{m}$	mass flow rate of refrigerant $[kg/s]$
w	velocity of fluid $[m/s]$
z	height $[m]$

Subscripts

a	ambient
c	collector
g	glazing
I	insulation
i	inlet of refrigerant
k	conductivity
L	heat out
p	plate
pm	plate mean
U	heat in

Project Overview

1.1 Background and Motivation

Climate change is a growing concern due to the effects that global warming has on the environment. As the population of the world increases, the energy demand will increase in turn. Due to the reliance on fossil fuels, this will only increase the amount of greenhouse gases that is produced into the atmosphere. This is why it is important to explore alternative energies that are relatively "greener" in comparison to the standard fossil fuels that are mainly used.

A major sector where energy can be saved is domestic hot water heating. 19.3% of the residential energy usage in Canada is from water heating [2]. This statistic can be improved with the implementation of technology that uses other sources of energy to provide water heating. This project focuses on a Direct Expansion Solar Assisted Heat Pump (DX-SAHP), which is an example of a technology that can be used in place of natural gas for heating. The DX-SAHP combines the benefits of a heat pump with the abundance of solar energy that we receive.

Unlike an Air Source Heat Pump (ASHP), the DX-SAHP uses a combination of solar energy and the ambient temperature to evaporate the refrigerant, improving the performance and making it viable for areas with a colder climate. The use of solar energy in conjunction with the heat pump reduces the amount of electricity that is needed for a standard heat pump to generate hot water.

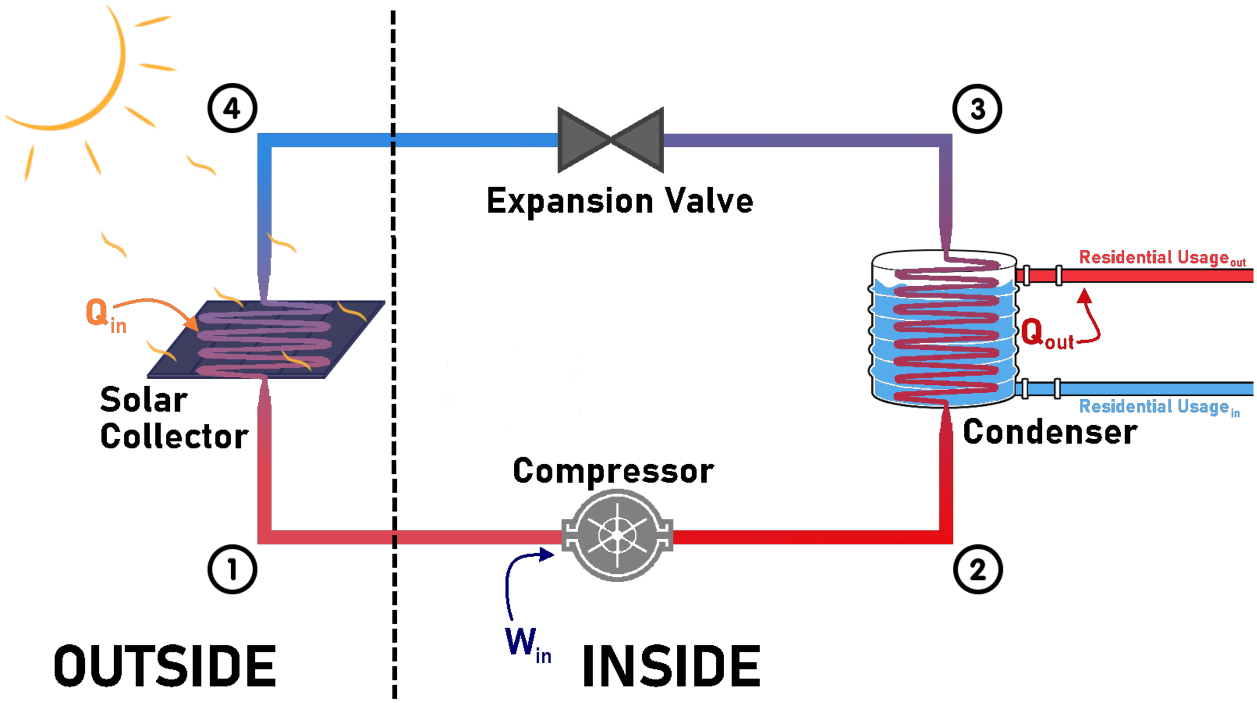


Figure 1.1: DX-SAHP Schematic

The solar collector is what will be used instead of a traditional evaporator in the DX-SAHP. The collector is heated up, due to it being in direct sunlight, and thus it heats up and evaporates the refrigerant running below it. Ambient temperatures play a critical role in the performance of a heat pump with a solar collector. This is due to the solar collector directly being affected by how much irradiance, the amount of light striking a surface, is available. To design a collector for colder climates, where perhaps less irradiance is available, there must be key considerations that are considered. Additionally, other required components must be matched to the demand and to the solar collector size.

1.2 Problem Statement

The current state of heating in Alberta is overly reliant on natural gas and the technology that is currently being used to heat domestic water is contributing to the increase in global temperatures via greenhouse gas emissions. DX-SAHP can be used to phase out natural gas heaters by using the sun as an energy source instead. Therefore, a DX-SAHP can be designed that can heat an average household of 3 people within the Calgary region.

1.3 Design Requirements

The goal of this capstone is to design a DX-SAHP that can heat 225L of domestic hot water to a household of 3 people. This number was calculated from the information from the Government of Canada stating that the average Canadian uses 75L of hot water a day [2]. Knowing this, a DX-SAHP will be designed and manufactured. The effort will be mainly focused on optimizing the area of the collector while reducing the heat losses from convection and radiation. Component matching analysis and selecting a compatible compressor, condenser, and electronic expansion valve is necessary so that it is system functions properly. This heat pump must be equipped with all the necessary controls and data acquisition systems. Afterwards a working model will be made in order to be functional within Calgary and an economic and environmental analysis will be conducted to justify the proposed system.

To determine the sizing of the DX-SAHP initial parameters must be set.

Statistics Canada states that the average Canadian household occupies 2.47 members, and the average Canadian uses 75L of hot water per day [3]. For the purposes of the DX-SAHP, it will be assumed that a domestic household of 3 people will require 225L of hot water per day. The hot water will be required to exit the system at a temperature of 50°C for residential use to avoid bacteria buildup in the water tank such as legionella.

The municipal water supply differs in temperature depending on the time of year. When it is winter, municipal line comes in at 10°C unlike in the summer when it comes in at 20°C. This is an important parameter to be considered when quantifying what heat load the heat pump will have to supply.

To validate and maintain quantifiable design goals, the project will be considered successful if it is able to sustain 225L of water at 50°C while maintaining a coefficient of performance (COP) greater than 2.3. This means the DX-SAHP must be able to provide the following heat loads.

$$Q_L = 225L \times \frac{1000g}{1L} \times 4.18\frac{J}{g^{\circ}\mathrm{C}} \times (50\,^{\circ}\mathrm{C} - 10\,^{\circ}\mathrm{C}) = 37,620kJ \tag{1.1}$$

Heat Load Required during Winter Conditions

$$Q_L = 225L \times \frac{1000g}{1L} \times 4.18\frac{J}{g^{\circ}\mathrm{C}} \times (50\,^{\circ}\mathrm{C} - 20\,^{\circ}\mathrm{C}) = 28,215kJ \tag{1.2}$$

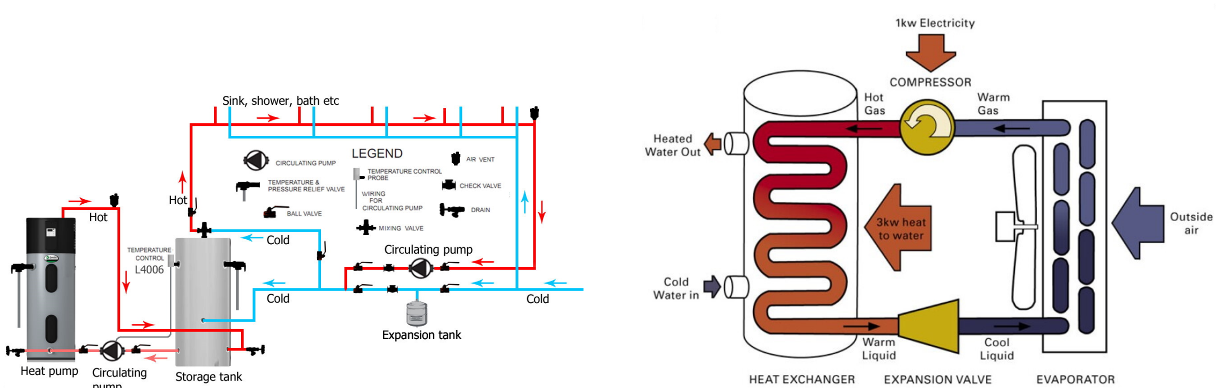
Heat Load Required during Summer Conditions

Conceptual Design

2.1 Background Research

The most common water heater is the storage tank water heater. The way it functions is by heating up water that will be stored within a storage tank. This is most conventional since it is easy and cheap to install when comparing it to other water heaters. The limiting factor is that it is limited by the size of the storage tank. It may take a while for the hot water to be heated again if it runs out.

Another type of water heater that is used is an Air Source Heat Pump (ASHP). What makes this heat pump unique is that it uses the heat within the air to heat the water. All it needs is a little bit of electricity to power the system. This makes the ASHP two to three times more efficient than most water heaters [4]. The major drawback is that it needs to take heat from the air, therefore it is not suitable for basements or colder climates.



(a) Storage Tank Water Heater [5]

(b) Air Source Heat Pump Water Heater [6]

Figure 2.1: Conventional Water Heaters

Unlike the ASHP, the DX-SAHP uses solar energy to provide the heat needed to heat the water. When comparing Figure 2.2 with Figure 1.1, a solar collector is used instead of an evaporator. This gives all the benefit of an Air Source Heat Pump with the additional benefit of working in colder climates since the sun can provide enough energy to heat the water.

When designing a DX-SAHP, it is important to consider the irradiance available in the area. Irradiance is the amount of energy received from the sun within an area. As seen in Table 1, the amount of sun Calgary receives is drastically different between the winter and summer data sets.

Winter months are defined from October to March in Table 2.1 while summer is between April to September. The justification behind choosing those date ranges is that it makes use of the entire year, and it is split between when Calgary has the most daylight hours versus the least. It also happens to line up with the solstices.

Table 2.1: Calgary Irradiance Data

Hour	Average Winter Irradiance Data (W/m^2)	Average Summer Irradiance Data (W/m^2)
1	0	0
2	0	0
3	0	0
[4]	0	0
5	0	1.398907104
6	0	12.99453552
7	0.510989011	76.3715847
8	7.736263736	182.0710383
9	54.40659341	327.442623
10	149.6538462	445.6065574
11	229.7857143	557.2896175
12	289	612.0601093
13	316.7087912	629.9289617
14	301.6868132	587.7923497
15	252.4285714	539.8196721
16	178.3241758	474.3715847
17	86.85714286	374.3879781
18	29.04395604	252.4043716
19	3.708791209	142.9016393
20	0	44.86885246
21	0	4.256830601
22	0	0
23	0	0
24	0	0

2.2 Collector Design Alternatives

There are three types of solar collectors that are commonly used in the market; they are flat plate collectors, evacuated tube collectors, and concentrating collectors. Each comes with their own benefits and drawbacks.

Flat plate collectors consist of a flat absorber plate that is orientated towards the sun. They use both direct and diffuse solar radiation and normally do not require a tracking system. Their main applications are solar water heating, heating for buildings, air conditioning and heat for industrial processes [7].

In contrast, evacuated tube collectors consist of several rows of parallel transparent glass tubes which have the working fluid flowing within it. The glass tubes are cylindrical in shape which results in the sunlight always being perpendicular to the heat absorbing tubes. This is a major benefit since this collector can be used when the sun is low in the sky or on cloudy days and they are particularly useful in colder climates.

Concentrating collectors are more suited for systems that require higher temperatures than what is achievable with a flat collector. The concentrating collector can be optimized by decreasing the area of heat loss when comparing it to flat plate collectors. This is done by placing an optical device between the source of radiation and the surface. A disadvantage that this technology has is that a sun tracking system is required to always maximize the incident radiation. This tracking system increases the overall cost of the collector and leads to additional maintenance which is why it was not considered at all for this project [7].

When comparing flat plate collectors with evacuated tubes, there are several categories that can be considered. When comparing costs, evacuated tubes are around 10-15% more expensive than flat plate. This was important to consider due to the limited budget that this project has [8]. Another important area that needs to be looked at would be how the collector can handle snow. Since there will be no tracking system for our design, there will be no way to shake off the snow, so the collector must remove it passively. The benefit of a flat plate collector is that snow can shed easily unlike with evacuated tubes in which it can get stuck due to the tubes creating a strong vacuum [8].

Both flat plate and solar collectors are excellent at heating water. The main question that needs to be asked when selecting the collector type is how much water needs to be heated to the desired temperature. Evacuated tubes are great for colder climates since they can heat water up to 121°C but it has the tendency to overheat. Therefore, evacuated tubes are more commonly used for commercial rather than domestic purposes. Unlike evacuated tubes, flat plate collectors can heat water up to 82°C which means that it is has smaller chance of overheating. This temperature range is suitable for domestic hot water usage [8].

When looking at all the parameters, the flat plate collector was selected instead of the evacuated tube or concentrating collector. Although the evacuated tube collector works better in colder climates than the flat plate collector, it was excessive in terms of both cost and design work for domestic water heating. The flat plate satisfies many requirements for a collector in colder climates, and it is the recommended collector for domestic water heating [8]. Therefore, for the design, the flat plate collector was chosen.

2.3 Refrigerant Alternatives

A refrigerant is a working fluid used in the thermodynamic cycle of a heat-pump, where the fluid will undergo multiple phase changes from liquid to vapor and vice versa, throughout the system cycle. One of the first components in a heat pump, which must be determined early on, is the refrigerant. To determine which components (i.e., compressor, condenser, and expansion valve) will be used in the DX-SAHP, the working fluid must be selected, and calculations will be performed to find the operating points of the system. Although there are many refrigerants to choose from, the list can be drastically cut down depending on multiple criteria. These criteria will look at the environmental acceptability, safety, application, performance, and the economics associated with various refrigerants.

The Ozone Depletion Potential (ODP) is a measure of a refrigerant's ability to damage the ozone layer relative to CFC-11 with an ODP of 1. Emissions from CFC's (chlorofluorocarbons), HCFC's (hydrochlorofluorocarbons), and other synthetic chemicals which created an "ozone hole" over the South Pole [9] have led to the Montreal Protocol on Substances that deplete the Ozone layer [10] – a global agreement made to phase out ozone-depleting substances. For the DX-SAHP, only refrigerants with an ODP equal to zero will be considered.

The Global Warming Potential (GWP) is the next major criterion regarding environmental acceptability. This is a metric measuring the energy of emissions, which one ton of a specific gas will absorb relative to the emissions of one ton of carbon dioxide (CO_2) over a hundred-year period [11]. For example, a refrigerant with a GWP of 2088 will have 2088 times the global warming potential of CO_2 over 100 years. In 2016, the Kigali amendment was made to the Montreal Protocol [12], proposing a complete phase down of HFCs by 2047 due to their high global warming potential. Furthermore, the European Union [13] took action to place market prohibitions on gases with a GWP greater than 750 in air-conditioning systems by 2025. Therefore, the refrigerants in the selection will also be required to have a GWP below 750.

Refrigeration cycles have three distinct applications: high temperature (comfort conditioning), medium temperature (food refrigeration), and low temperature (transport refrigeration). Domestic water heating falls into high temperature comfort conditioning applications. The most widely used refrigerant in these applications has been R-410A. Due to the high SEER (Seasonal Energy Efficiency Ratio) ratings R-410A has provided with many heat pump systems as compared to other refrigerants, it has dominated the air conditioning market for components. However, due to R-410A's high GWP of 2088, refrigerants with similar thermodynamic properties used as replacements for the eventual phase down of R-410A were explored.

Following the ASHRAE Standard 34 refrigerant safety classification, most refrigerants in use currently pose a very low toxicity and flammability threat – giving an ASHRAE safety designation of A1. Following the ASHRAE Standard 34 [14] refrigerant safety classification, most refrigerants in use currently pose a very low toxicity and flammability threat – giving an ASHRAE safety designation of A1. ASHRAE Standard 34 assigns refrigerants to two toxicity (A or B), and four flammability classes (1, 2, 2L, 3). The safety designations for refrigerants are as follows:

- Class A (Low Toxicity)
  - Occupational exposure limit is 400ppm or greater
- Class B (High Toxicity)
  - Occupational exposure limit is less than 400ppm.
- Class 1 (No flame propagation)
  - No flame propagation at 60°C and atmospheric pressure.
- Class 2L (Low flammability)
  - Flame propagation at 60°C and atmospheric pressure.
  - Lower Flammability Limit $> 0.10 kg/m^3$ and Heat of Combustion $< 19,000 kJ/kg$
  - Burning velocity ¡= 10cm/s at 23°C
- Class 2 (Flammable)
  - Flame propagation at 60°C and atmospheric pressure.
  - Lower Flammability Limit $> 0.10 kg/m^3$ and Heat of Combustion $< 19,000 kJ/kg$
- Class 3 (High flammability)
  - Flame propagation at 60°C and atmospheric pressure.
  - Lower Flammability Limit $<= 0.10 kg/m^3$ or Heat of Combustion $< 19,000 kJ/kg$

Because the system is being designed for a residential water heating supply, due to possibility of leakage, only refrigerants designated in Class A toxicity will be used.

As with many design considerations in engineering, there is an equivalent exchange when reducing the global warming potential of refrigerants. A general trend can be observed in refrigerants where a lower GWP equates to a higher flammability designation; most new generation refrigerants with a GWP less than 750 have an ASHRAE safety designation of A2L. While refrigerants of Class 1 are the most desirable, Class 2L refrigerants can also be considered safe [15] to use in domestic heating systems as they have a high Minimum Ignition Energy and would need to be exposed to an open flame or high energy source with sufficient concentrations to ignite.

Although each refrigerant will result in different system efficiencies, by looking at the critical temperature of the refrigerant, a correlation can be made for both the coefficient of performance and the cooling capacity of the system. As the critical temperature of the refrigerant increases, the coefficient of performance of the system is found to increase, while the cooling capacity is found to decrease [16]. A higher coefficient of performance will ultimately result in a lower energy bill for the end user, while a lower cooling capacity will result in a larger system. As the project location is for a cold climate in Calgary, the refrigerant must also be chosen to have a freezing point, $T_{fp} < -50\,^{\circ}\mathrm{C}$. Furthermore, CoolProp [17] was used to conduct the thermodynamic analysis, and therefore, refrigerants of choice must be available on the database such that the system calculations can be performed.

Finally, the economics and procurement of the refrigerants will be considered where the system will require between 1-5kg of charge. After contacting vendors, many refrigerants were found to have more than 3 months lead times due to COVID-19 supply chain issues, and as a result would not satisfy the project's timeline. Many new generation refrigerants were also found to be cost-ineffective when compared to their incremental performance benefits. These refrigerants ranged from $400 to $3000 per their minimum selling quantities.

A design requirements table was then created to easily compare the refrigerants as seen below.

Table 2.2: Refrigerant Criteria

Refrigerant	ODP	GWP	Alternative To	Safety Class	$T_{critical}$ (°C)
R-410A	0	2088	R-22	A1	72.13
R-717 (NH_3)	0	0	R-22	B1	132.4
R-1234yf	0	4	R-134A	A2L	94.70
R-1234ze	0	1	R-134A	A2L	109.4
R-32	0	675	R-410A	A2L	78.40
R-454B	0	466	R-410A	A2L	78.10
R-454C	0	148	R-410A	A2L	82.40
R-455A	0	146	R-410A	A2L	86.60
R-466A	0	733	R-410A	A1	76.50
R-515B	0	299	R-134A	A1	108.7
R-290	0	3	R-410A	A3	97.00

After assessing the criteria of each refrigerant, and contacting vendors of the remaining options, it was determined that R-32 was to be the refrigerant of choice in the DX-SAHP. The decision was contingent upon the refrigerant's suitability with the team's initial selection criteria. With a zero ODP, a GWP less than 750, a critical temperature of 78.4°C, but most importantly, a procurement time of 2 months, the R-32 was determined to be the appropriate working fluid.

Upon determination of R-32 as the working fluid for the system, the component matching phase began. Although some components were found for this refrigerant (e.g., expansion valve), when looking for a compressor compatible with R-32, and which could be procured within a reasonable time frame, component matching proved to be difficult.

After consulting with a subject matter expert working in the heating ventilation and cooling (HVAC) sector for over 15 years, the team determined the reason for the difficulties to be that R-32 and all other similar new generation refrigerants are still considered novel to the industry. As R-410A is still dominating the air conditioning industry in North America, components for comfort cooling heat pump applications are designed around R-410A being the working fluid. However, the research was not in vain; since R-32 is the widespread refrigerant of choice in Asia and is slowly being phased in around parts of Europe as the next replacement for R-410A, it is beneficial to have considered it as a potential working fluid for the DX-SAHP.

Finally, although the GWP was far too high, it was still suggested to design the system using R-410A as North America has not yet caught up with the refrigerant phase down plan. Furthermore, the use of R-410A as a system refrigerant is not detrimental to the project's environmental considerations as the final solution to avoid the use of a high GWP refrigerant is through sourcing drop-in replacements. A drop-in replacement for R-410A (such as R-470A) is a refrigerant which can simply swap R-410A while maintaining the same system components. This allows for the system calculations, and thus, the system components to be modeled and procured based upon R-410A as the working fluid. The advantage of taking the approach of using drop-in refrigerants is that, as new refrigerants become available, supply chain delays would not hinder the progress of the build assembly as R-410A could always be used to complete the project.

2.4 Component Design Alternatives

2.4.1 Compressor Type Selection

For the purposes of the compressor in the DX-SAHP, the hermetic sealed variable speed compressor has been selected as the primary choice. These types of compressors are positive displacement and can achieve higher compression ratios per single stage of compression [18]. They are more compact and less prone to vibration issues.

Hermetic compressors are widely used in domestic refrigeration systems where continuous maintenance cannot be ensured by the user. A hermetic compressor consists of the compressor being mounted directly on the shaft of the motor. The compressor and motor are confined together within an outer shell, reducing the potential for dust particles to enter as easily, and thereby reducing any impact on the operation of the compressor [19]. With both the compressor and motor being directly coupled on the same shaft and confined within a common casing, the potential for leakage of R-410A is reduced and essentially eliminated [20]. The investigated hermetic compressors for the purposes of system analysis are of the reciprocating and rotary types.

The variable speed aspect of the compressor works by adjusting the speed of the motor to meet required demands while conserving energy in the process. For the purposes of this analysis, having the ability to control the speed of the compressor allows for the control of the pressure differential of refrigerant, R-410A. Therefore, it is essential to control the varying inlet conditions entering the compressor to obtain a condensation temperature of 60°C.

The advantages of a hermetic variable speed compressor are less noise, faster pull-downs to our established condensation temperature, more consistent temperature, less wear on system components, less vibrations, and decreased energy bills [21]. The disadvantage of using this type is that the motor drive cannot be maintained in its place and if the motor were to fail, the whole compressor would need to be replaced.

Out of the many available compressors, the following design alternatives for a hermetic variable speed compressor have been taken into consideration, allowing for use in domestic water heating applications [22] [23].

1. Scroll variable speed compressor [24]
2. Reciprocating variable speed compressor [25]
3. Rotary screw variable speed compressor
4. Centrifugal variable speed compressor
5. Open motor hermetic speed compressor

The following figure shows common compressors and their classification.

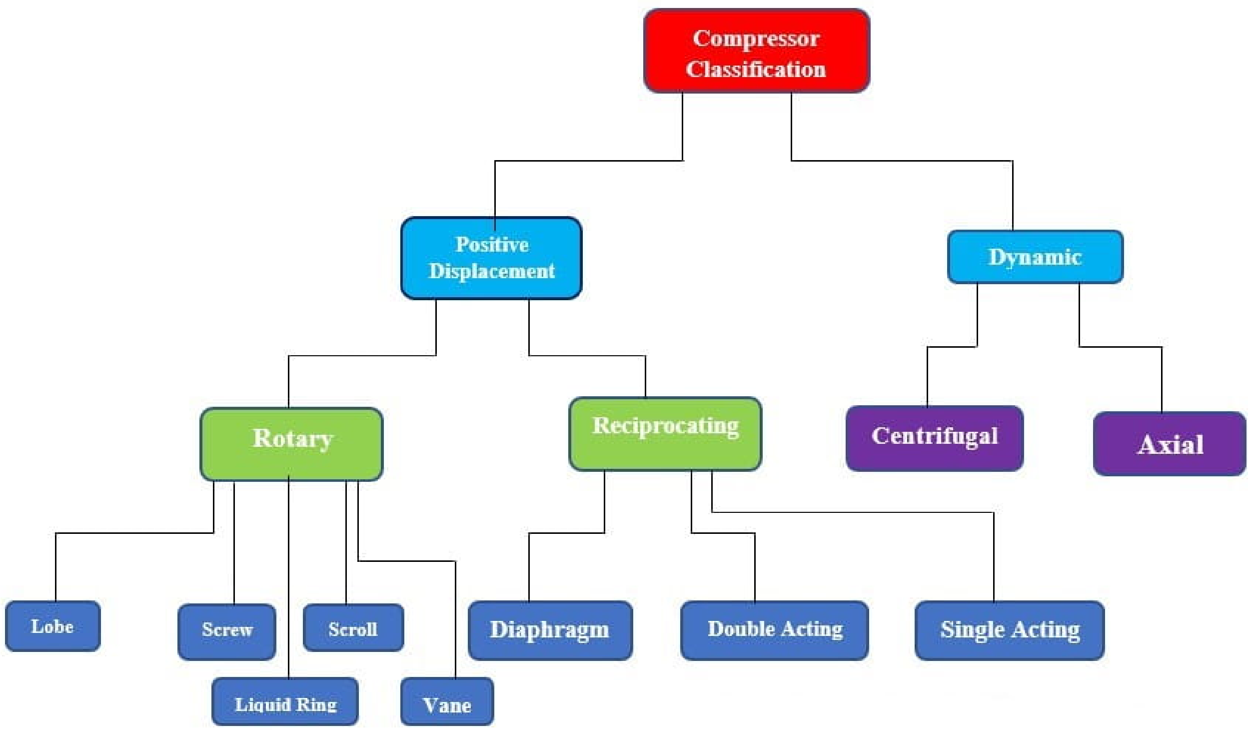


Figure 2.2: Compressor Classification [26]

Out of the above five listed compressors, the primary design alternative would be the scroll variable speed compressor. The rotary screw and reciprocating positive displacement variable speed compressors may also be used for the purposes of the DX-SAHP. However, since rotary screw variable speed compressors are more commonly used in commercial and industrial applications, they will not be considered as the primary design alternative. The reciprocating variable speed compressors are commonly used in residential applications except that this type does not operate as type rotary which is preferred for the analysis of DX-SAHP. Scroll variable speed compressors are available for lower power ratings, are compatible with R-410A, come in a single phase, and can operate within the operating pressures of the system. They exhibit energy saving capabilities [27] [28].

The reasons for not selecting the remaining as design alternatives is because while all the above can be used in similar applications, the centrifugal type compressors are often used in applications such as, large chillers, refineries, plants, and industrial applications [29]. Therefore, they operate with higher horsepower and for higher operating pressures than required for the purpose of the DX-SAHP. Additionally, centrifugal compressors consist of dynamic applications whereas for the purposes of the system, a positive displacement compressor shall be used. The open motor variable speed hermetic compressor is not suitable as it has a higher potential of refrigerant leakage than a sealed hermetic variable speed compressor would.

2.4.2 Condenser Type Selection

Storage tank water heaters are by far the most prevalent configuration of water heaters available on the market today; however, tankless or "On-Demand" water heaters are slowly acquiring some of that market share due to their reputation of running more efficiently. For the purposes of selecting a condenser/water storage tank for the Direct Expansion Solar Assisted Heat Pump, the team considered the following factors:

i. The operational time span of the evaporator/collector.

ii. The stability of meteorological conditions of the design locale.

iii. The temperature of the inlet municipal water supplied for domestic use.

By relying on weather station data from the Government of Canada [30], we determined the average, daily "bright sunshine hours", to be 4.68 hours for Calgary during the winter season. These hours would support peak operation of the Solar Thermal Collector, and outside of which, the system performance may decline, and in extreme conditions, stagnate, ceasing the supply of hot domestic water in the absence of a suitable thermal mass. Assessing the stability or consistency of the meteorological conditions, such as ambient temperatures and average irradiance, the team concluded that the short operational time frame, paired with the instability of meteorological conditions and potential for inclement winter weather conditions, such as extreme subzero temperatures and collector shading due to snowfall, the DX-SAHP cannot support a tankless water heater module as a steady supply of hot water would not be guaranteed outside of optimal operational conditions. Furthermore, considering that the temperature of the supplied groundwater determines the length of the heating period in a tankless water heater, and that inlet municipal water is supplied at approximately 10°C during the winter, adopting a tankless water heater is not a feasible option for this application due to prolonged heating times. In conclusion, the team opted to adopt a shell and coil condenser wherein the shell or "storage tank" would serve as the water reservoir or thermal mass from which emergency supply of hot water could be provisioned. Further selection considerations are highlighted in section 3.3.2 below.

2.4.3 Expansion Valve Selection

For the purposes of the throttling valve in the DX-SAHP system, an electronic expansion valve was selected. Throttling valves allow control of the amount of mass flow rate by adjusting the size of the flow path through the valve. The two types that were initially compared were the thermostatic valve (TXV) and the electronic expansion valve (EXV). The following explanation tends to the operation of both [31], differences [32], and final selection choice.

TXV's typically use sensing bulbs to sense the temperature of the suction line. These bulbs are slightly warmer than the saturation temperature of the refrigerant and have an increase in pressure when the suction line temperature exceeds the saturation temperature. The increased pressure in the bulb indicates that more refrigerant is required to manage the system's evaporating heat load. The opening of the valve occurs by the internal connections from the bulb to the power element. The power element consists of a diaphragm and with increasing pressure, the diaphragm is bent downwards to open the valve.

EXV's use an electronic controller to calculate the superheat based on the temperature and pressure at the suction line, and the outlet of the solar flat plate collector. For the controller to read the pressure and temperatures, respectively, pressure transducers and thermistors may be used. The programming of the controller controls the valve movement by either opening or closing the valve, based on the inputs read by the sensors.

The main disadvantages of using a TXV is that if the pressure differential between the sensing bulb, combined pressure below the diaphragm, and the spring are significantly reduced, the opening and closing of the valve will be affected. Ultimately, this creates a problem for the system to operate as efficiently, mainly on the release of the mass flow rate as required for the heating load.

Based on this, the EXV was selected as the throttling valve to be placed in the system. EXV's offer more flexibility for system design requirements by having the ability to use the controller without needing to physically adjust the valve.

Conceptual Design

3.1 Collector Analysis

3.1.1 Heat Transfer Analysis

To model the heat transfer, depicted in Figure 3.1, occurring on the surfaces of a flat plate collector, the following simplifications can be made:

- Performance is steady state.
- Heat losses through the front and back are to the same ambient temperature.
- Construction is of the sheet and serpentine manifold type.
- Uniform flow exists within the tubes.
- Absorption of solar energy by the cover is insignificant insofar as it affects losses from the collector.
- Temperature drop through the cover is negligible.
- Heat flow through the cover is one-dimensional.
- The cover is opaque to infrared radiation.
- Heat flow through the back insulation is one-dimensional.
- The sky is considered a black body for long-wavelength radiation at the equivalent sky temperature.
- Temperature gradients around the tubes are negligible.
- Properties of the collector are independent of temperature.
- Dust, dirt, and snow buildup on the collector are negligible.
- Shading of the collector absorber plate is negligible.

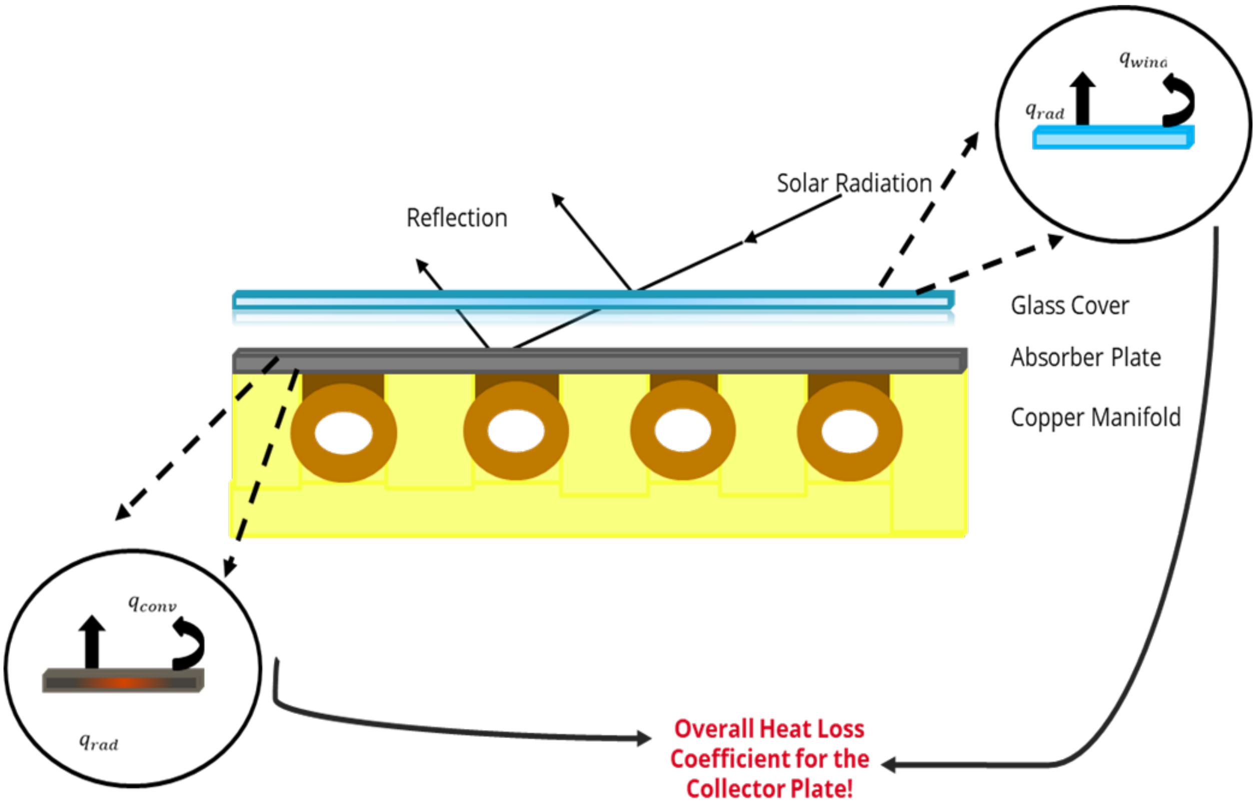
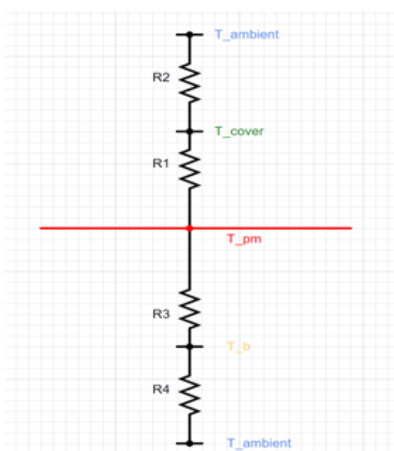
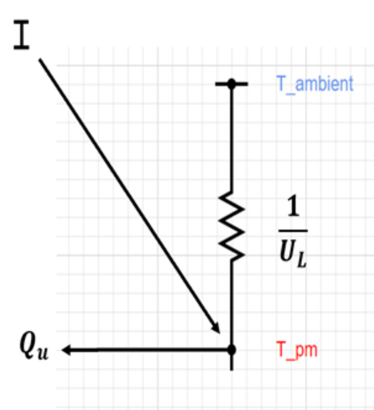


Figure 3.1: Heat Transfer from a Flat Plate Collector

The heat losses can be analytically simplified by representing the heat transfer from the collector's absorber plate through collector's cover at the top, and insulation at the back and subsequently, to the ambient as the thermal network depicted in Figure 3.2a. An equivalent thermal network, as shown in Figure 3.2b, can then be deduced to encompass the overall steady-state heat transfer occurring across the collector. The heat transfer analysis is derived in full detail below.



(a) One-Cover Flat-Plate Collector



(b) Equivalent Thermal Network

Figure 3.2: Thermal Network Diagrams

First, the top heat losses, both convective and radiative, from the absorber plate to the cover can be evaluated as follows to determine the first thermal resistance, R1:

$$Q_{loss,top} = h_{c,p-c}(T_{pm} - T_c) + \frac{\sigma(T_{pm}^4 - T_c^4)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1} \tag{3.1}$$

$$h_{r,p-c} = \frac{\sigma(T_{pm} - T_c)(T_{pm}^2 - T_c^2)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1} \tag{3.2}$$

$$R1 = \frac{1}{h_{c,p-c} + h_{r,p-c}} \tag{3.3}$$

Similarly, the top heat loss, both convective and radiative, from the cover to the ambient can be evaluated as follows to determine the second thermal resistance, R2:

$$h_{r,c-a} = \frac{\sigma\varepsilon_c(T_c + T_{sky})(T_c^2 + T_{sky}^2)(T_c - T_{sky})}{(T_c - T_a)} \tag{3.4}$$

$$R2 = \frac{1}{h_w + h_{r,c-a}} \tag{3.5}$$

Finally, the total top heat loss coefficient, U_{top}, is found to be the inverse of the summation of R1 and R2 as follows:

$$U_{top} = \frac{1}{R1 + R2} \tag{3.6}$$

A useful empirical equation for U_{top} was developed by Klein (1979) following the basic procedure of Hottel and Woertz (1942) and Klein (1975). This relationship fits the graphs for U_{top} for mean plate temperatures between ambient and 200°C to within $\pm 0.3 W/m^2K$ and is represented below:

$$U_{top} = U_{tC} + U_{tR} \tag{3.7}$$

The heat loss through convective effects, U_{tC}, can be quantified as:

$$U_{tC} = \left[\frac{M}{\left(\frac{c}{T_{pm}}\right)\left(\frac{T_{pm}-T_a}{M+f}\right)^e} + \frac{1}{h_w} \right]^{-1} \tag{3.8}$$

Where:

$$f = (1 + 0.089h_w - 0.116h_w\varepsilon_p)(1 + 0.07866N) \tag{3.9}$$

$$e = 0.43\left(1 - \frac{100}{T_{pm}}\right) \tag{3.10}$$

$$c = 520(1 - 0.000051\beta^2) \tag{3.11}$$

$$h_w = 5.7 + 3.8V_w \tag{3.12}$$

The heat loss through radiative effects, U_{tR}, can be quantified as:

$$U_{tR} = \frac{\sigma(T_{pm}^2 + T_a^2)(T_{pm} + T_a)}{(\varepsilon_p + 0.059Mh_w)^{-1} + \frac{2M + f - 1 + 0.133\varepsilon_p}{\varepsilon_g} - M} \tag{3.13}$$

Therefore:

$$U_{top} = \left[\frac{M}{\left(\frac{c}{T_{pm}}\right)\left(\frac{T_{pm}-T_a}{M+f}\right)^e} + \frac{1}{h_w} \right]^{-1} + \frac{\sigma(T_{pm}^2 + T_a^2)(T_{pm} + T_a)}{(\varepsilon_p + 0.059Mh_w)^{-1} + \frac{2M + f - 1 + 0.133\varepsilon_p}{\varepsilon_g} - M} \tag{3.14}$$

R3 represents the resistance to heat flow through the insulation while R4 represents the convection and radiation resistance to the environment. With appropriate back insulation, it is usually possible to assume R4 is zero and all resistance to heat flow is due to the insulation.

The heat loss through the bottom, U_b, of the collector can be defined as:

$$U_{bottom} = \frac{1}{R} = \frac{\delta_1}{k_1} \tag{3.15}$$

The heat loss through the sides, U_{edge}, of the collector can be defined as:

$$U_{edge} = \frac{Q_{edge}}{A(T_{pm} - T_a)} \tag{3.16}$$

Where:

$$Q_{edge} = A_p(T_{pm} - T_a) \tag{3.17}$$

The total heat loss coefficient is the sum of the heat loss coefficients for the top, bottom, and sides of the collector. It can be defined as:

$$U_L = U_{top} + U_{bottom} + U_{edge} \tag{3.18}$$

$$U_L = \left[\frac{M}{\left(\frac{c}{T_{pm}}\right)\left(\frac{T_{pm} - T_a}{M + f}\right)^e} + \frac{1}{h_w} \right]^{-1} + \frac{\sigma(T_{pm}^2 + T_a^2)(T_{pm} + T_a)}{(\varepsilon_p + 0.059Mh_w)^{-1} + \frac{2M + f - 1 + 0.133\varepsilon_p}{\varepsilon_g} - M} + \frac{\delta_1}{k_1} + \frac{Q_{edge}}{A(T_{pm} - T_a)} \tag{3.19}$$

Finally, the total useful heat gain of the collector can be quantified as:

$$Q_u = F' A_c \left[I(\tau_c \alpha_c) - U_L(T_{fi} - T_a) \right] \tag{3.20}$$

The fin efficiency represents the represents the efficacy with which energy absorbed by the absorber plate and the tube spacing (conceptualized as fins) is collected on the sides of the tubes for subsequent heat transfer into the working fluid:

$$F = \frac{tanh\left[\frac{m(W - D)}{2}\right]}{\left[\frac{m(W - D}{2}\right]} \tag{3.21}$$

Where:

$$m = \sqrt{\frac{U_L}{k_p \delta_p}} \tag{3.22}$$

Physically, F', the collector efficiency factor, represents the ratio of the actual useful energy gain to the useful gain that would result if the collector absorbing surface had been at the local fluid temperature. It is essentially a constant for any collector design and fluid flow rate.

$$F' = \frac{\frac{1}{U_L}}{W\left[\frac{1}{U_L(D+(W-D)F)}\right] + \frac{1}{C_b} + \frac{1}{\pi D_i h_{fi}}} \tag{3.23}$$

The collector heat removal factor is a quantity that relate the actual useful energy gain of the collector to the useful energy gain has the entire collector surface were at the fluid inlet temperature.

$$FR = \frac{mC_p}{A_c U_L}\left[1 - \exp\left(\frac{-A_c U_L F'}{mC_p}\right)\right] \tag{3.24}$$

It's important to note that, as the mass flow rate through the collector increases, the temperature rise through the collector decreases. This corresponds to lower losses as the average collector temperature is lower, leading to an increase in the useful energy gain. This increase is reflected by an increase in the collector heat removal factor FR when the mass flow rate increases.

3.1.2 Collector Efficiency

The collector's instantaneous efficiency is defined as the ratio of useful heat energy gain to total energy incident on the collector's surface:

$$\eta = \frac{Q_u}{IA_c} \tag{3.25}$$

The day-long collector efficiency is the summation of instantaneous efficiencies at known time steps, in our case, on an hourly basis:

$$\eta_{day} = \frac{\sum Q_u}{\sum IA_c} \tag{3.26}$$

As seen in the equations above, the absorber plate's mean temperature is important in determining the values evaluated by the previous governing equations. With many unknowns, this value can only be determined through an iterative solution approach using an initial guess for the plate mean's temperature. For our purposes, an initial guess of $T_{pm} = T_{fi} + 5$ is reasonable [33]. Following the iterative algorithm described in the 'Code Logic' below, and summarized in Figure 3.4, the equation below can be used to determine a convergent solution for the final mean temperature of the plate:

$$T_{pm} = T_{fi} + \frac{\frac{Q_u}{A_c}}{FRU_L}(1 - FR) \tag{3.27}$$

3.1.3 Thermodynamic Cycle Analysis & Collector Efficiency Optimization

In thermodynamics, heat pump cycles are bound by two reservoir temperatures, namely, the evaporation and the condensation temperatures. For the purposes of the system, the condensation temperature is regarded as a set point: since the system is designed to support an outlet water temperature of 55°C for domestic use, T_{cond} is constrained to be approximately 60°C. Determining the optimal, steady state evaporation temperature on which to base the collector design is key, not only to optimizing the flat plate collector's area, but also to minimizing the radiative and convective heat losses emanating from its surfaces. Closely tied to the ambient temperatures, the evaporation temperatures of the working fluid circulating within the collector's manifold dictate the useful heat gain of the collector or, more specifically, the efficiency of the collector, and correspondingly, the COP of the overall system. Referring to ASHRAE's heat pump & air conditioning design conditions for Calgary, an initial range of design evaporation temperatures between -10°C and 10°C was selected.

Additionally, meteorological data sets encapsulating average, hourly, winter-day temperatures and Irradiance values were loaded into the MATLAB [34] file. Using a C++ Fluid Properties', MATLAB-accessible library, CoolProp [17], the thermodynamic states of the Refrigerant R410A, including temperatures, pressures, enthalpy, and entropy, were determined for points 1 through 4 of the thermodynamic cycle. As depicted in Figure 3.3, the isobar between on which states 2-3 lie represents the set, saturation pressure corresponding to the design condensation temperature of 60°C. The collection of dashed isobars on which states 4-1 lie correspond to the saturation pressures of the chosen range of evaporation temperatures to undergo analysis. To simplify the analysis, the following assumptions of the thermodynamic cycle were made:

i. Constant pressure heat addition occurs in the collector.

ii. Constant pressure heat rejection occurs in the condenser.

iii. Isentropic compression occurs between states 1-2 in the compressor.

iv. Isenthalpic expansion occurs between states 3-4 in the expansion valve.

v. The refrigerant enters the compressor at a quality of 1 or in a saturated vapor state.

These assumptions will later be corrected for thorough accounting for sub-component efficiencies as well as pressure drop in the collector.

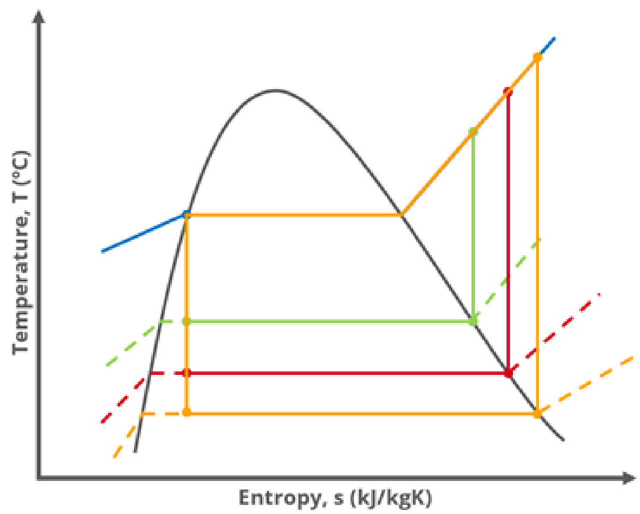


Figure 3.3: T-s Diagram for Probable Design Evaporation Temperatures

Using CoolProp [18], the team determined the performance parameters of the isolated heat pump cycle, namely, Q_L, Q_H, W_{comp} and COP. With W_{comp} or theoretical compressor work in mind, appropriate sizing for the compressor was determined. Next, code was developed which amalgamated the flat plate collector's governing equations and, through iteration, allowed for the determination of the flat plate's mean temperature. Figure 3.4 below depicts the complete iteration algorithm utilized in the MATLAB code.

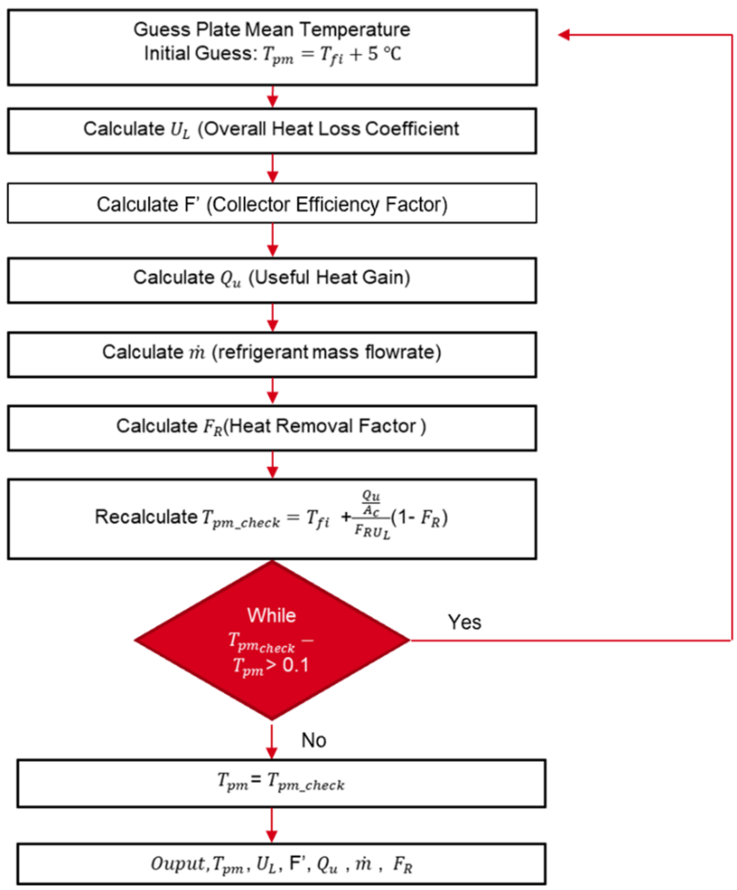


Figure 3.4: Iterative Solution for Flat Plate Mean Temperature

Once the flat plate's mean temperature was determined, the collector's useful heat gain, and, subsequently, the collector's efficiency was evaluated for every data point in the evaporation temperature range. The evaporation temperature's impact on the isolated heat pump cycle COP is diametrically opposed to its impact on the useful heat gain of the collector: on one hand, the COP of the heat pump cycle increases as the gap between the evaporation and condensation reservoir temperature is minimized, or when the chosen evaporation design temperature is elevated. On the other hand, the collector's efficiency declines with the elevation of evaporation temperature as a result of increased heat losses from its surfaces. Noting this inverse relationship, it's deducible that a plot of the product of COP and Collector Efficiency (a quantity defined as the overall system COP) versus evaporation temperature would exhibit a characteristic inflection point at the evaporation temperature that maximizes both these inversely related parameters. For the DX-SAHP, the inflection point was seen to occur at -2°C. Knowing the design evaporation temperature, the team was able to subsequently determine the predicted collector efficiency, and the predicted useful, net collected heat. These values will later be leveraged to evaluate the theoretical performance of the DX-SAHP against the logged experimental performance. Using the code, the team also determined the system's necessary flow rate, which supplemented the selection process of the remaining sub-components of the heat pump cycle.

3.2 Collector Final Design

3.2.1 Insulation Selection

Insulation is one of the most efficient ways to save energy by reducing heat loss during winter and thus lowering energy bills [35]. Reducing heat loss in the collector means the compressor will have to do less work to meet the hot water requirements. For the flat plate collector, it was essential to investigate the concepts pertaining to location of any heat losses to the surroundings, type, thickness, and cost of insulation.

In the solar flat plate collector, heat losses occur through the absorber plate by top losses. As the plate heats up, some of this heat is then transferred to R-410A (within the copper tubing of the collector that is brazed beneath and to the aluminum absorber plate), while some of the heat being lost to its surroundings. The heat losses occurring through the back, and sides of the collector are respectively known as bottom and edge losses [36]. From heat transfer and thermodynamic contexts, it is understood that these heat losses occur in the form of conduction, convection, and radiation [36].

Based on the engineering analysis and design of the collector, the bottom and sides require insulation as to minimize any heat losses and the consideration of an insulation cover being required in case of probable exposure area that is responsible for the occurrence of any heat losses.

The insulation materials representative of some of the materials commonly used in solar flat plate collectors and in the industry are as follows:

- Fiberglass wool.
- Rigid polyurethane foam.
- Mineral wool.
- Expanded polystyrene.
- Extruded polystyrene.

The following table represents the range of thermal conductivity values, temperature, and R-values [37] for the mentioned types of insulation materials.

Table 3.1: Range for Thermal Conductivity, Temperature, and R-Value for Insulation

Insulation Type	Thermal Conductivity, k $[W/mK]$	Temperature Range	R Value [per inch of thickness]
Fiberglass Wool	0.023 - 0.040	-195°C to 230°C	R-3.7 to R-4.2
Rigid Polyurethane Foam	0.020 - 0.035	62°C to 93°C	R-3.4 to R-6.7
Mineral Wool	0.033 - 0.040	Maximum: 649°C	R-3.7 to R-4.3
Expanded Polystyrene	0.030 - 0.040	Maximum: 75°C	R-3.9 to R-4.7
Extruded Polystyrene	0.025 - 0.040	Maximum: 74°C	R-5.0 to R-5.6

The following table identifies the American Society for Testing and Materials (ASTM) specification, material type, and/or grade for some of the insulation materials that are commonly used in the industry [38].

Table 3.2: Common Types of Insulation - Based on ASTM

Material	Insulation Standard
Cellular Glass	ASTM C 552 Type II
Elastomeric	ASTM C 534 Type I, Gr 1
Fiberglass	ASTM C 547 Type I
Flexible Aerogel	ASTM C 1728 Type I, Gr 1B
Phenolic	ASTM C 1126 Type III
Polyethylene	ASTM C 1427 Type I, Gr1
Polyisocyanurate	ASTM C 591 Type IV
Polystyrene	ASTM C 578 Type XIII

Based on the above analysis, mineral wool was selected as the insulating material to be used for the solar flat plate collector due to its excellent thermal properties. Mineral wool has low thermal conductivity values, allowing for less heat to be passed through and lost to the surroundings. The suitable temperature range allows for use up to 649°C as this material will not melt until temperatures reach beyond 1,000°C. The R-values are within a range of R-3.7-R-4.3, allowing for it to suitably resist heat flow. In addition, mineral wool is naturally moisture resistant [39].

3.2.2 Glazing Selection

Glazing refers to the top cover of the solar collector. It has three main purposes:

i. Protect the internal components from the outside environment.

ii. Minimize heat loss due to convection and radiation from the absorber plate.

iii. Allow as much solar radiation through as possible.

The two main materials used for solar collector glazing are glass and polycarbonate.

The main parameter to consider when choosing the glazing is the transmittance. Transmissivity is a measure of how much light passes through the object for a given wavelength. For a solar collector, the glazing should let through as much sunlight as possible but be opaque to the infrared radiation emitted by the absorber plate. This will allow for the most heat gain possible. The secondary parameters which should be minimized are the absorptance and reflectance of the glazing. The absorptivity is a measure of how much radiation is absorbed for a given wavelength and reflectivity is how much is reflected [40].

Another important factor to consider is the solar heat gain coefficient (SHGC). The SHGC is a measure of how much solar radiation is admitted. A high SHGC rating indicates that the materials are more effective at collecting solar heat, which is better for a solar collector [41].

Table 3.3: Properties of Various Glazing Materials

Glazing Type	Temperature Range	Transmissivity	SHGC [42][43]	Thermal Expansion Coefficient ($in/in/F$)	Density (kg/m^3)
Low Iron Tempered Glass [44]	-50°C to 240°C	91.5%	$\sim$ 0.91	4.9E-6	2530
Polycarbonate (Standard) [45]	-50°C to 120°C	86%	$\sim$ 0.80	3.75E-5	1197
Sun-Lite [46]	-50°C to 120°C	86%	$\sim$ 0.80	3.6E-5	1200
Lexan 9034 [47]	-40°C to 100°C	88%	$\sim$ 0.80	3.75E-5	1197
SunTuf [48]	-40°C to 100°C	90%	$\sim$ 0.80	3.6E-5	1200

As seen from Table 4 above, all the materials found met the temperature requirement of -30°C to 30°C. Low iron tempered glass was found to have the highest transmissivity, highest solar heat gain coefficient, and lowest coefficient of thermal expansion. Glass was found to be more opaque to the long wave radiation emitted by the absorber plate, and therefore better at trapping heat [49]. Whereas polycarbonate was found to transmit more IR radiation [50]. Additionally, polycarbonate will yellow over time from exposure to UV rays [51]; this will reduce the amount of light transmitted by it. However, glass is more than two times heavier than the polycarbonate sheets and more prone to breaking. So extra care will need to be taken when installing it in the collector.

3.2.3 Plate Material Selection

The absorber plate is the component which absorbs solar radiation and emits it as infrared radiation. This heat is then absorbed by the copper piping and then the refrigerant. For this purpose, the plate must have high absorptivity, heat conductivity, and emissivity. The most common materials for the absorber plate are copper, aluminum, and steel; the thermal conductivities of these metals are $398 W/mK$, $247 W/mK$, and $45 W/mK$ respectively [52] [53]. Steel was found to have too low thermal conductivity for this application. The price of copper was $9.55/kg [54] and aluminum was $6.33/kg [55]. Aluminum was selected for our application because it was more readily available.

A selective coating will be applied to the absorber plate to increase the amount of sunlight absorbed. The coating should have high absorbance and low emissivity, so all the absorbed heat will be transferred to the aluminum plate. It should also be able to withstand high temperatures and be UV resistant. Thurmalox 250 was identified as a coating specifically suited for solar thermal collector application [56].

3.2.4 Manifold Design

To determine the copper tube design beneath the absorber plate of the solar flat plate collector, it was convenient to create a two-dimensional drawing to determine the layout. The design of the tubing helped determine the overall length of tubing that would be required. A serpentine tube design was selected as it maximizes the amount of surface area, and for R-410A, for heat transfer to occur within a limited amount of space [57].

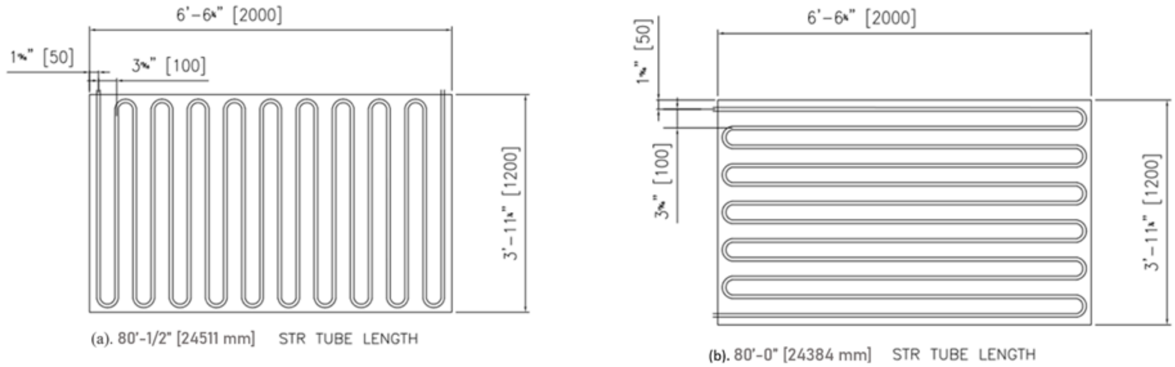
The serpentine copper tube design for the flat plate collector was completed based upon the following criteria:

i. Tube pitch of 3/4 inches [19.05 mm].

ii. Tube bend diameter of 3 15/16" [100 mm].

iii. Leave 1 15/16" [50mm] on each side of absorber plate.

In the following figure, two designs were created, (a) and (b). Both designs have an equal manifold spacing. The design for (a) was selected as this design leads to a greater surface area allowing for more heat transfer to occur. With the design for (a) having more U-bends in the tubes, this allows for more time for heat transfer to take place with R-410A. A tube bending tool may be used to force the tube to conform to a bend diameter of 3 15/16" [100 mm]. The overall straight length between both designs differs by 5 inches [127 mm] in which the cost between both tube designs will be similar.



(a) Horizontally Spaced Manifold

(b) Vertically Spaced Manifold

Figure 3.5: Two-Dimensional Designs of Serpentine Copper Tube Manifold

3.2.5 Final Collector Model

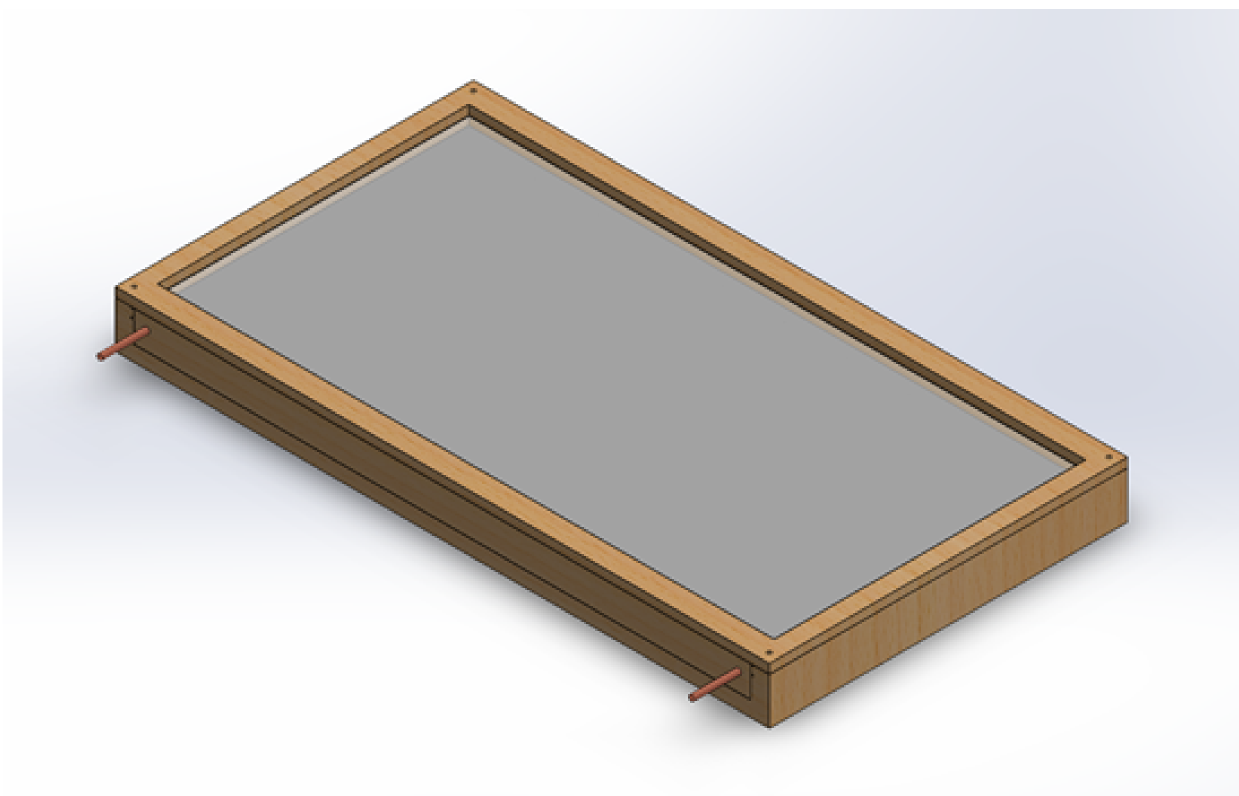


Figure 3.6: SOLIDWORKS Assembly of Solar Thermal Collector

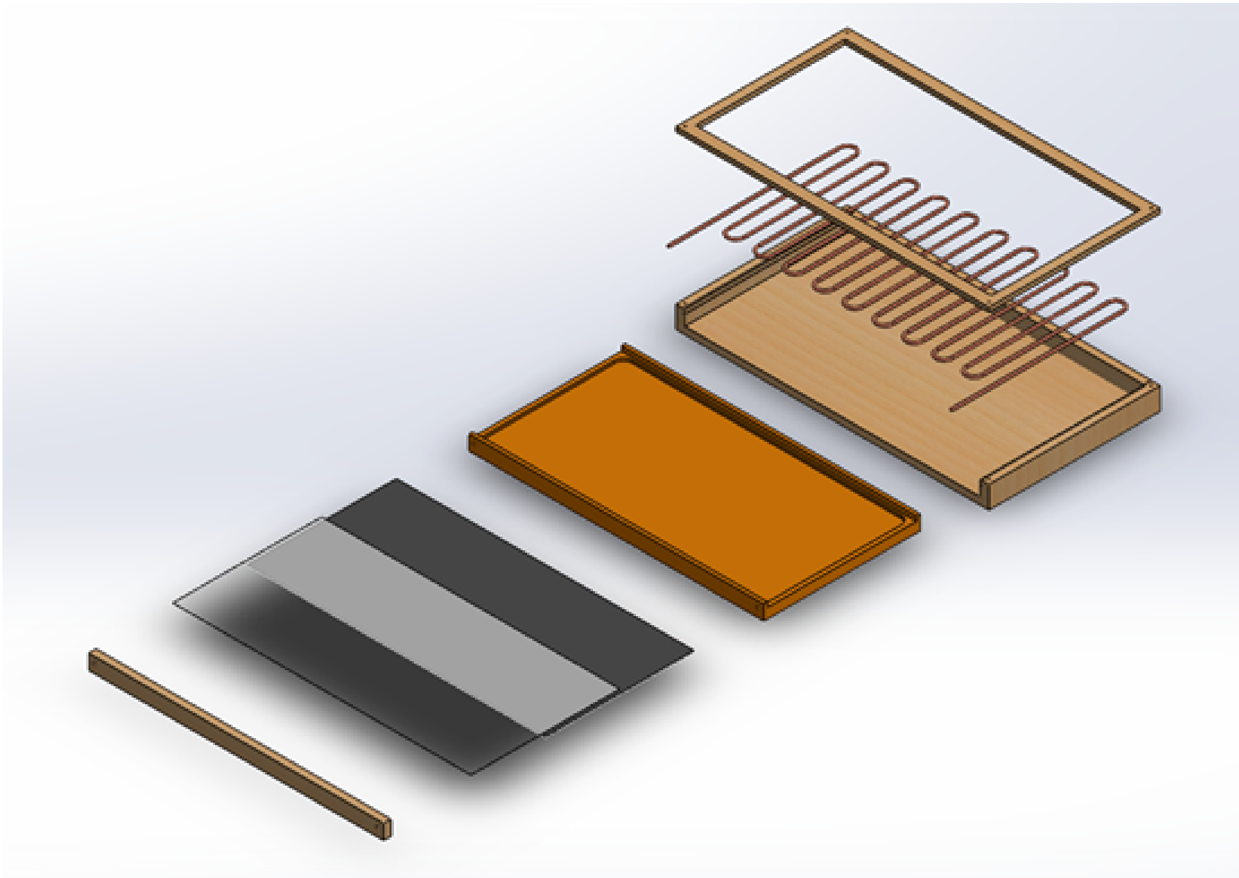


Figure 3.7: Exploded View of Solar Thermal Collector

Although the collector frame is a simple enclosure, sizing limits and assembly considerations must be understood. The wood casing will be created in multiple parts and grooves will be added to account for the thermal expansions of both the glass and absorber plates during winter thermal contractions and summer thermal expansions.

The depth of the grooves will be determined using the equation for linear thermal expansion.

$$\Delta L = \alpha_L L_c \Delta T \tag{3.28}$$

The order in which the components are assembled must be taken in consideration to avoid scenarios where components are blocked by other components. As component selections become finalized and are ready to order, fitment tolerances will be subsequently added.

3.3 Final Component Selection

For component matching, external to the solar flat plate collector, suitable components and materials for the system based on data and design parameters are being selected. The components mentioned below are the compressor, condenser, expansion valve, control system, and minor parts. The evaluation of manufacturing methods and metal-joining processes as well such as, brazing and/or welding, and use of items such as couplings, and tube adapters have been taken under consideration. The following selection of components are preliminary as of now and will be finalized in the Final Design Report.

3.3.1 Compressor Selection

A compressor is required after the collector to meet the domestic hot water requirements. A hermetically sealed compressor is ideal for domestic heat pump systems. The compressor will be a single stage, rotary vane, positive displacement compressor. The motor will be a single phase, variable speed motor. The compressor must operate within the specified voltage of approximately 110/120 V and 50-60 Hz [58]. In the case where it is greater, further in-depth investigation into transformers [59] [60] or power amplifiers is required to overcome this difference.

The compressor selection is based on the lowest evaporating temperature of -10°C. To obtain a condensation temperature of 60°C, the pressure differential needs to be controlled via the motor speed.

The compressor is also evaluated based on the criteria shown below:

i. Compatibility with R-410A.

ii. Working pressure ratings.

iii. Lubricant compatibility.

iv. Power requirement.

The power requirement for the compressor was found to be 1.138 horsepower based on the required flow rate.

The rotary vane variable speed hermetic compressor was selected for this project. These types are commonly used for appliances and for residential applications. It may be applied for the purposes of the DX-SAHP as it operates at low capacities, requiring less input power. The compressor works by rotating action of a roller inside a cylinder to compress the refrigerant.

The following figure shows what the hermetic variable speed compressor looks like for a low input power rating. However, this compressor is not a finalized selection as vendors are currently being contacted.

Figure 3.8: Hermetic Variable Speed Compressor [61]

The compatibility between the hermetic variable speed compressor and electronic expansion valve is crucial as this controls the pressure differential of R-410A through the compressor and the amount of flow rate through the valve. Vendors have been provided the design specifications, controller requirements, and refrigerant parameters to size these components. The copper piping, having a diameter of 0.75 inches, is to be connected to the suction and discharge of the compressor and valve. The determination of the diameters and materials of these lines are being communicated with the vendor as to allow the easiest route for metal-joining and installation purposes. The two choices, as of now, to connect the external piping to the suction/discharge lines of both the compressor and valve, can either be done by brazing the copper piping to the connecting tubes and/or by use of a copper adapter tube [62]. The use of an adapter tube allows for the connection of the suction/discharge connector to be connected to the external piping without the need of expanding the connectors or reducing the diameters of the connection tubes.

3.3.2 Condenser Selection

The condenser is responsible for the heat transfer between the refrigerant and the domestic hot water. It also stores and insulates the water for immediate use if required. The basic components of the condenser are a hot water tank with a copper condensing coil inside to allow for heat transfer between the refrigerant and the water. A simple heat exchanger will not suffice as hot water needs to be stored for use when there is no solar radiation available.

The selection criteria for the condenser are as follows:

i. Must be able to store at least 225L of water.

ii. Max condensing coil pressure of 3800kPa.

iii. Max condensing coil temperature of 85°C.

iv. Heat rejection of 2.5kW.

v. Compatible with R-410A.

Proper sizing of the condenser will be important in ensuring the conditions of the domestic hot water are optimal. Too small of a condensing unit may result in overheated water and vice versa. Currently vendors are being contacted with the system requirements to obtain a condenser that meets the above criteria.

Figure 3.9: Water Tank with Two Condensing Coils [63]

3.3.3 Electronic Expansion Valve Selection

Selection of electronic expansion valves (EXV) is based on the following criteria:

i. Suitable for HFC refrigerants (i.e., R-410A).

ii. Rated Capacity (kW).

Firstly, the selected valve must be compatible with the chosen refrigerant (i.e., R-410A).

Secondly, the expansion valve must be able to provide the needed pressure reduction for system parameters. This is usually given by a vendor through an EXV's capacity rating, which references the system's heat removal rate in kW. Vendors provide a capacity rating for the following refrigerant conditions, based on AHRI standards [64]:

Table 3.4: AHRI Standard Rating Conditions for EXV

Standard Rating Condition	Liquid Temperature at EXV Inlet	Condensing Temperature at EXV Inlet	Evaporating Temperature at EXV Outlet
A	37°C	38°C	4°C

If the above standard is not used, vendors must specify the operating conditions used instead for their stated rating.

Since these operating conditions differ from the ones in the DX-SAHP, a correction factor must be added to the required capacity rating, to compare it with ratings from the vendor. The following information is needed to determine the correction factor:

i. Refrigerant: R-410A.

ii. Condenser capacity: $Q_L = 2.5kW$.

iii. Evaporating temperature: $T_{evap} = -10\,°\mathrm{C}$.

iv. Condenser temperature: $T_{cond} = 60\,°\mathrm{C}$.

v. Subcooling: Assume subcooling of $\Delta T_{sub} = 4K$ at inlet of EXV.

All vendors include data sheets that specify the correction factors that must be used for their valve selection. The chosen valve was from manufacturer Danfoss who provided the following table [65] to aid in correction factor selection.

Table 3.5: Correction Factors for R-410A based on Degree of Subcooling

ΔT_{sub}	0K	4K	10K	15K	20K
Correction Factor	1.00	1.06	1.14	1.21	1.28

From Table 3.5, it can be seen that a correction factor of 1.06 must be used on Q_L to get the nominal rating.

$$COP_{corrected} = \frac{2.5kW}{1.06} = 2.36kW \tag{3.29}$$

Thus $COP_{corrected}$ can be compared with ratings provided by the vendor at the system's T_{evap} and T_{cond}, to find to find an appropriate expansion valve.

From this process it was found that the Danfoss ETS 6-10 [65] satisfied the needed capacity with a sufficient safety factor, being able to provide a capacity of $2.9kW$ at the specified T_{evap} and T_{cond}. For a full list of the vendor data sheets that were considered during EXV selection, please see the Electronic Expansion Valve selection document in the Design Binder.

3.3.4 Piping

Piping is an essential part of a heat pump; it carries the energy gained by the collector and compressor to be released in the condenser. A preliminary analysis was done to determine the drop in pressure between major components in the system. This pressure drop was considered to be from friction in the pipe, elbows, and changes in height. The variable definitions for all the equations below can be found in Appendix A.

The equation for pressure drop due to friction in a circular pipe is given as [66]:

$$\Delta P_f = \frac{f L_p \rho w^2}{2D_i} \tag{3.30}$$

The friction factor, f, is calculated separately for laminar and turbulent flows; however, in this system, only turbulent flows were found. For turbulent flow, the Colebrook White equation [66] was used to calculate the friction factor. This was solved by moving all terms to one side and using the `fzero` MATLAB [34] function to iteratively solve for the friction factor.

$$\frac{1}{\sqrt{f}} = -2.0log\left(\frac{\frac{e}{D_i}}{3.7} + \frac{2.51}{Re\sqrt{f}}\right) \tag{3.31}$$

The Reynold's number determines the type of flow (i.e., Laminar, turbulent, or transitioning). If greater than 2320, the flow was considered turbulent. The Reynold's Number was calculated as [66]:

$$Re = \frac{wD_i}{\nu} = \frac{\rho w D_i}{\mu} \tag{3.32}$$

The values of density and viscosity were found using MATLAB to access CoolProp [18]. By specifying two state parameters, temperature and quality, the density and viscosity were obtained.

The velocity of the refrigerant was calculated as:

$$w = \frac{\dot{m}}{\frac{\rho \pi D_i^2}{4}} \tag{3.33}$$

The pipe length between sections was assumed to be one meter for these calculations. Therefore, the pressure drop is shown per meter.

The pressure drops were found for each section as described below:

S1: Between 1 (Compressor) and 2 (Condenser Inlet).

S2: Between 2 (Condenser Exit) and 3 (Expansion Valve).

S3: Between 3 (Expansion valve) and 4 (Evaporator Inlet).

S4: Between 4 (Evaporator Exit) and 1 (Compressor).

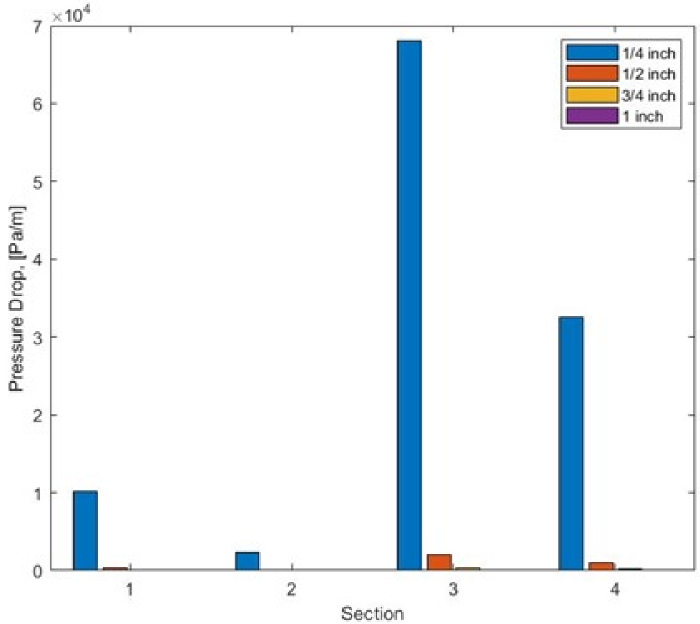


Figure 3.10: Pressure Drop for 1/4, 1/2, 3/4, and 1 inch Diameter Piping at each Section

As seen from Figure 3.10 above, the pressure drop in the 1/4 inch pipe is magnitudes greater than it is for the other diameters. Figure 3.11 below shows the pressure drops for the 1/2, 3/4, and 1-inch diameters.

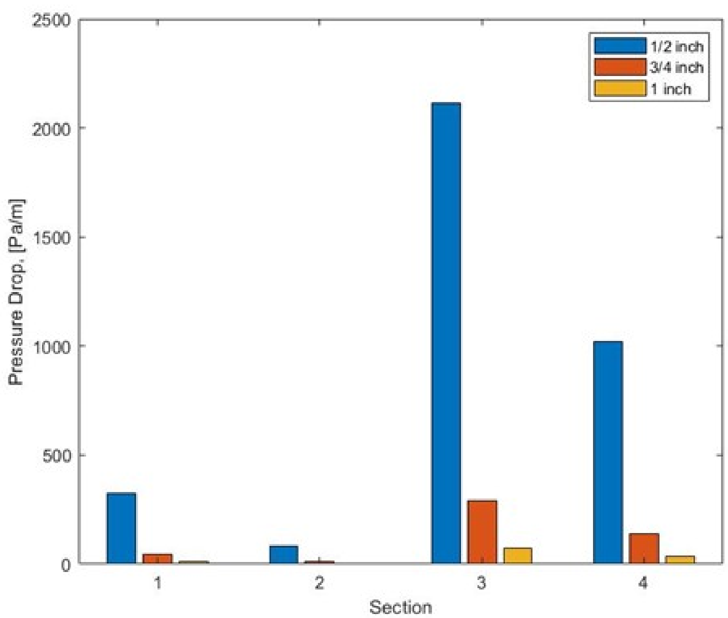


Figure 3.11: Pressure Drop for 1/2, 3/4, and 1 inch Diameter Piping at each Section

As seen from Figures 3.10 & 3.11 above, as the diameter of the piping decreased, the pressure drop per meter increased drastically. Based on these values however, the pressure drop due to friction for diameters greater than 1/4" is insignificant considering the system operating pressures that range from $750kPa - 3800kPa$.

The pressure drop due to elbows was calculated as follows [66]:

$$\Delta P_e = \frac{K_L \rho w^2}{2} \tag{3.34}$$

K_L is the loss coefficient of the specified component or elbow. The loss coefficient for a long radius 90° flanged elbow is 0.2 and the loss coefficient for a regular 90° threaded elbow is 1.5 [66].

The pressure drop due to a change in height was calculated as [66]:

$$\Delta P = \rho g \Delta z \tag{3.35}$$

Since the piping is not yet finalized, the pressure drop due to elbows are shown for one elbow for each section.

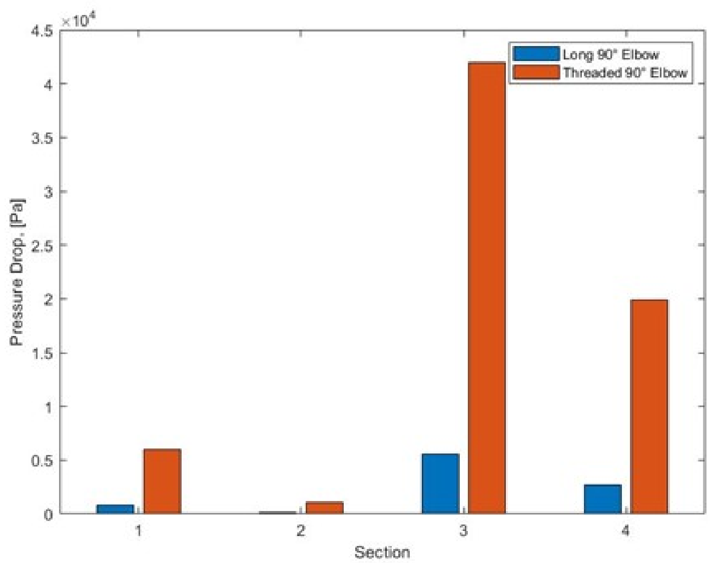


Figure 3.12: Pressure Drop from a Long Radius 90° Elbow vs. a Threaded 90° Elbow for a 1/4 inch Pipe

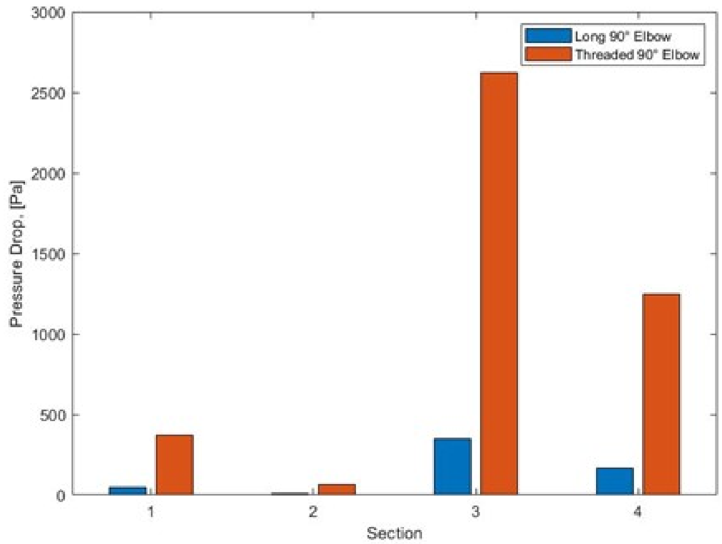


Figure 3.13: Pressure Drop from a Long Radius 90° Elbow vs. a Threaded 90° Elbow for a 1/2 inch Pipe

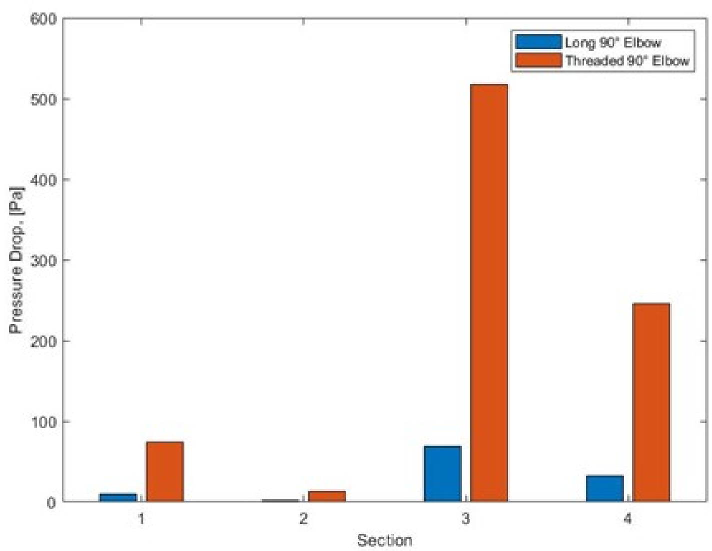


Figure 3.14: Pressure Drop from a Long Radius 90° Elbow vs. a Threaded 90° Elbow for a 3/4 inch Pipe

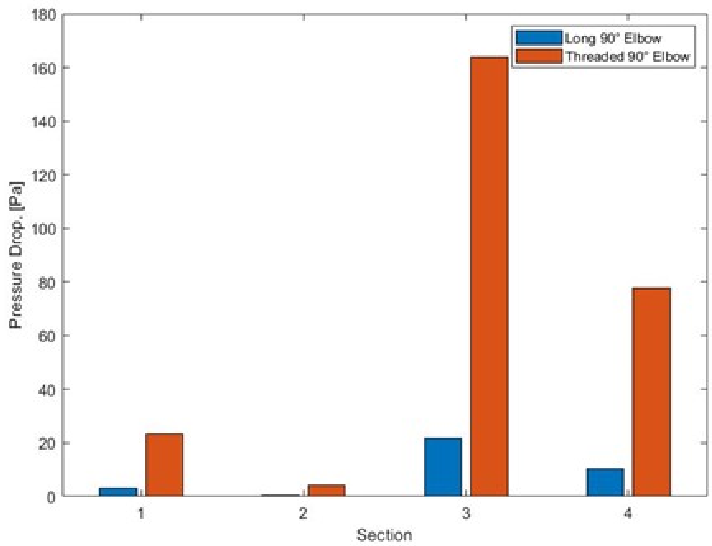


Figure 3.15: Pressure Drop from a Long Radius 90° Elbow vs. a Threaded 90° Elbow for a 1 inch Pipe

As seen from Figures 3.12 - 3.15 above, the drop in pressure follows the same trend when decreasing the pipe diameter; smaller diameter has larger pressure drops. Additionally, the threaded 90° elbow yields more drop in pressure than the pipe with a large radius elbow as expected.

Based on Figures 3.12 - 3.15, using a 1/4" copper pipe would result in too much pressure drop in our system; up to a max of $68kPa$ per meter and $19kPa$ per elbow. The 1/2" piping was found to have max pressure drops of $2.1kPa$ per meter and $2.6kPa$ per elbow. When considering installing the system for an average home, this may be too much loss in pressure. When comparing the 3/4" and 1" piping, both of their pressure drops are insignificant. For 3/4", the max pressure drop was $0.29kPa$ per meter and $0.52kPa$ per elbow. For the 1" pipe, the max pressure drop was $0.071kPa$ per meter and $0.16kPa$ per elbow. To be more cost efficient, a piping size of 3/4" was found to be acceptable in minimizing pressure drop and cost.

3.3.5 Minor Components

3.3.5.1 Accumulator

A suction line accumulator is used to prevent any liquid from entering the compressor. If all the refrigerant exiting the collector is not vaporized, there is potential for liquid refrigerant to enter the compressor. Since this system uses a vapor compressor, any liquid refrigerant entering it will damage it. Additionally, the compressor is not modular, and any damage will require a full replacement, so protecting it is a necessity. An accumulator works by using a U-tube to trap any liquid refrigerant and only allowing vapor to flow to the compressor. The accumulator must be able to contain approximately seventy percent of the total refrigerant charge [67]. Additionally, during normal compressor operation, a small amount of oil from the compressor's components will enter the piping; the accumulator also serves the purpose of collecting and returning this oil back to the compressor. To prevent any vapor from reaching the compressor, the accumulator is installed before the compressor [68].

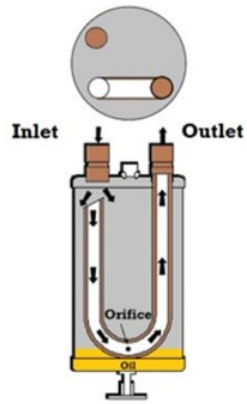


Figure 3.16: Suction Line Accumulator [69]

3.3.5.2 Filter Drier

To keep the system working at optimal conditions, it is necessary to ensure that there are as little contaminants as possible. The potential contaminants in this system are water, copper shavings, or other contaminants during installation. The filter drier ensures these contaminants are not circulated throughout the system and causing damage; it is installed in the liquid-line of the heat pump before the sight glass [70].

3.3.5.3 Sight Glass

The sight glass is needed to view the level of the refrigerant to ensure proper operation. If there are bubbles seen through the sight glass it indicates that there is not enough refrigerant in the system. Additionally, the refrigerant must be subcooled before entering the EXV; therefore, the sight glass is installed before the EXV to ensure only liquid enters it [71].

3.4 Control System

The philosophy of having a control system for this design lies in the fact that the heat pump needs to adapt to varying ambient conditions to still meet the desired heat load. For instance, when there is less available sunlight, the pump might need to be active for a longer amount of time, simply due to the lower heat input rate.

To ensure the heat load requirements are constantly being met, the system must control the following two parameters:

i. Mass flow rate of the refrigerant.

ii. RPM of the compressor.

3.4.1 Mass Flow Rate Control

Firstly, the mass flow rate of the refrigerant dictates how fast the refrigerant is flowing through the system. It is critical that the refrigerant enters the compressor as at least a saturated vapor, as any liquid-vapor mixture will damage the compressor. To ensure that full saturation is achieved, the refrigerant must spend enough time in the solar collector that it is able to fully change phase. The time spent in the collector is directly linked to mass flow rate and thus it is critical that this parameter be controlled. For example, during colder conditions where there is less available sunlight, the system needs to be able to lower the flow rate so that the refrigerant can spend more time in the collector to completely changing phase.

Mass flow rate will be controlled by the electronic expansion valve. There is a stepper motor in the valve that incrementally opens and closes the valve opening, to adjust the flow rate. This stepper motor responds to electronic signals that will be fed by an external controller. The Danfoss controller [72] was selected as it is compatible with the system's chosen electronic expansion valve. The controller input parameters include the refrigerant type, and the solar collector exit temperature and pressure. The temperature and pressure measurements will be given by an AKS model temperature sensor and an AKS 32R pressure transmitter, which are recommended from the controller's operator's manual [72]. The controller must optimize between ensuring that the refrigerant is at least a saturated vapor as it exits the collector, while still reducing the amount of superheat if possible. This is because excessive superheating at the outlet of the collector is an indicator that there is not enough refrigerant passing through and the mass flow rate should be increased. If the mass flow rate is needlessly low, the collector plate average temperature will increase, and COP of the system will fall.

Figure 3.17: EXV Controller with Compatible Temperature and Pressure Sensors

3.4.2 Compressor RPM Control

Secondly, the RPM of the compressor must also be controlled. The RPM directly correlates to the pressure differential created by the compressor. To meet the heat load, the refrigerant conditions at the compressor outlet are taken to be constant at $331K$ and $3670kPa$. The inlet conditions will be changing however depending on the evaporating temperature reached in the collector. This evaporating temperature is dictated by ambient conditions, and this leads to a situation where there are varying conditions at the inlet of the compressor and just a single outlet condition. Therefore, the compressor must change speed, depending on the inlet conditions, to create whatever pressure differential that is needed to get the refrigerant to the predefined outlet condition. Variable speed compressors are fitted with a temperature sensor at the inlet to determine refrigerant conditions. The needed outlet temperature can be set, and the controller within the compressor will adjust RPM accordingly.

Assembly Plan and Design Verification

4.1 Design and Development Plan

The design is scheduled to be completed on March 1st, 2022, to allow for a one-month testing and verification period. The design deliverables of the project will require a full-scale working model, which does not include a system prototype. The assembly of the project will be split into 5 parts:

i. Solar Thermal Collector.

ii. Piping.

iii. System Frame.

iv. Refrigerant.

v. Data Acquisition System.

The Solar Thermal Collector will consist of a glass pane, absorber plate, serpentine piping manifold, and insulation, all enclosed in a plywood frame. The assembly of the thermal collector - i.e., the brazing of the serpentine manifold onto the bottom of the absorber plate - will take place concurrently with the piping fitments into the main components (i.e., compressor, condenser, expansion valve, and collector). The piping will also be welded or brazed between the components. After the frame is constructed, and the heat pump is assembled, the system can be charged with the refrigerant. As the handling and charging of refrigerants requires qualifications, three methods will be explored:

i. Support from the maintenance team at the University of Calgary will be requested to charge the system.

ii. Support from the refrigerant training program at the Southern Alberta Institute of Technology will be requested to charge the system.

iii. If the aforementioned strategies fail, HVAC (heating, ventilation, and air conditioning) companies will be contacted through referrals to charge the refrigerant.

Finally, temperature and pressure sensors for the data acquisition system will be integrated into the piping of the system to allow for the gathering of data validation metrics such as the system coefficient of performance. The temperature probes will be inserted into brazed pockets in the piping, and the pressure sensors will be screwed into the piping through specialized T-connectors.

4.2 Design Verification

To verify the design and determine the system performance, a data acquisition system was developed.

The performance of the design can be quantified by two parameters: the COP of the system and the outlet temperature of the water.

The COP of the system can be defined as:

$$COP = \frac{Q_{out}}{W_{cycle}} = \frac{\dot{m}(h_2 - h_3)}{\dot{m}(h_2 - h_1)} = \frac{(h_2 - h_3)}{(h_2 - h_1)} \tag{4.1}$$

As evidenced by Equation 4.1, the COP of the system can be determined knowing the specific enthalpy of the refrigerant at certain points. Specific enthalpy at any location can be found through use of temperature and pressure measurements and the refrigeration table of R-410A [18]. As so, temperature sensors at 3 locations (1, 2, 3) and pressure sensors at 2 locations (1, 2) on Figure 4.1 below will be used to determine COP. The pressure sensor at 3 can be neglected under the isobaric condensation assumption. However, a temperature sensor at 3 is still necessary as it is pertinent to know and minimize the degree of subcooling at the inlet of the electronic expansion valve. Additionally, pressure transducers are approximately at least 20 times more expensive than a thermistor at any given location, so the isobaric assumption was also used for economic reasons. In the above COP equation, it was assumed that all the power input into the compressor is going into superheating the refrigerant. This in fact is not valid, as the compressor itself will hold an efficiency factor. This efficiency factor dictates how much power the compressor puts into the refrigerant, from the total power it uses.

To evaluate this efficiency factor, the following equation will also be used to determine COP:

$$COP = \frac{Q_{out}}{W_{cycle}} = \frac{\dot{m}(h_2 - h_3)}{W_{cycle}} \tag{4.2}$$

To determine COP using Equation 4.2, the mass flow rate of the refrigerant and the power consumption of the compressor must additionally be known. The mass flow rate of the refrigerant only needs to be measured at one location, as per the laws of continuity. A turbine flow meter will be used at location 1 for this purpose. Additionally, a power meter will be connected to the compressor to determine power usage.

The second parameter that will be used for design verification will be the water outlet temperature. As per the design goals, the aim is to have this temperature be 55°C. A thermistor will be paced at this location to measure this parameter.

Figure 4.1 below provides a visual guide of sensor placements in the system.

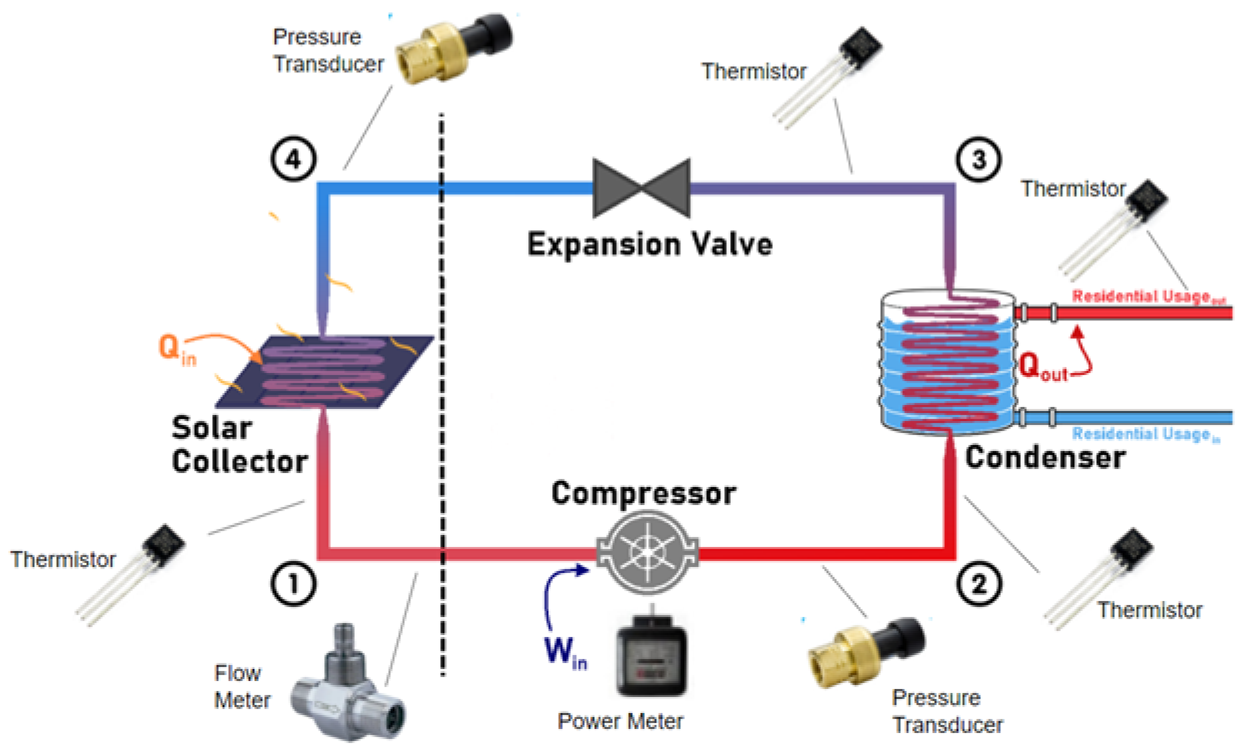


Figure 4.1: Sensor Locations

4.2.1 Data Acquisition Configuration

To get the data outputs from the sensors, several data acquisition configurations were explored. Ultimately, an Arduino Uno was decided upon as the data acquisition device as it was the most economical option. Arduino Uno's have only six analogs to digital converter (ADC) pins, whereas at least eight ADC pins would have been needed if all selected sensors gave analog outputs. To bypass this issue, digital thermistor sensors that could be connected to digital pins on the Arduino were chosen instead. Arduino Uno ADC pins have a maximum bit size of 12, which can affect resolution of the measurement picked up. For all the analog sensors chosen, this resulted in the smallest magnitude that could be measured being 1-2% of the expected value. This resolution was decided to be sufficient for the needs of the data acquisition system. Table 4.1 shows a full list of sensors used. Note that the location numbering refers to schematic in Figure 4.1.

Table 4.1: List of Sensors to be used in Data Acquisition System

Sensor Type	Location(s)	Accuracy	Operating Range	Power Supply	Output Type
Thermistor [73]	1, 2, 3, Water Outlet	$\pm 0.5\,°\mathrm{C}$	$-55\,°\mathrm{C}$ to $125\,°\mathrm{C}$	Arduino	Digital
Pressure Transducer [74]	1	0.25%	$0psi$ to $150psi$	$9V$ to $30V$ DC at $< 10mA$	Analog $0V$ to $5V$ DC
Pressure Transducer [74]	3	0.25%	$0psi$ to $1000psi$	$9V$ to $30V$ DC at $< 10mA$	Analog $0V$ to $5V$ DC
Volumetric Flow Meter [75]	2	3%	1L/min to 10L/min	3.75E-5	Amplified Square Wave
Power Meter [76]	Compressor Electrical Outlet	3%	0KWH to 9999KWH	Compressor Power Supply	Screen Display

All sensors will be tested and configured individually as per their respective manufacturers' guidelines. An external power supply will be used for all sensors that require it. For more information on the different sensors that were explored, please see the Sensor Selection excel file in the Design Binder.

To verify the design, a $COP > 2.3$ and a water outlet temperature of 55°C needs to be constantly achieved, as per design requirements. The previously described data acquisition system will assist in quantifying design verification.

Project Management

5.1 Roles and Responsibilities of Team Members

Each team member plays a vital role in the group:

- Kerwin oversees the Project Management of the group which is to ensure that everything is going according to plan and to assist in contacting vendors and setting schedules up.
- Nadia is responsible for the designing and sizing the collector. She is in charge of ensuring that the collector is functional.
- Jessica oversees component matching. Her role is important since without her, components may not work together.
- Charuka is the assembly lead. When the parts have arrived, he will be the one in the shop piecing it all together and leading us in that part of the project.
- Dhruvi will look directly into the control and data acquisition system. These include looking into all the flow meters, thermistors, and pressure transducers to list a few. It's essential to have these components for testing purposes.
- Edwin will be the testing lead when the prototype has been finished. He will oversee collecting all the data that would be needed to support our project.

5.2 Project Schedule and Deliverables

Table 5.1: Project Milestones

Milestone	Scheduled Completion Date
First Iteration of Collector Design	28-Nov-21
Final Design with all Compatible Components	1-Dec-21
Complete Bill of Materials	10-Dec-21
Order all Components	10-Dec-21
Prototype Completed	1-Mar-22

5.3 Budget for Prototype Development

The budget for the entire project is \$10,000. This will include all the fees that will be listed in the bill of materials and all the service fees that will need to be paid to access and work in machine shops.

5.4 Use of Project Resources and Contact Hours

For this project, biweekly meetings with the sponsor and meetings with the faculty advisor were scheduled as needed.

Closing Remarks

6.1 Summary

DX-SAHP shows potential in replacing natural gas heaters. The parts that have been selected for the project include a flat plate collector featuring an aluminum absorber plate, a copper serpentine manifold, and a single tempered glass glazing. The system's subcomponents, i.e., the hermetically sealed variable speed compressor, the electronic expansion valve and the condenser will be interconnected via copper tubing. Sensors have also been selected for the control of the mechanism and to enable the acquisition of data during the testing phase.

With all the components selected, the project is on track for completion. Based on the preliminary calculations, the project is anticipated to be a success. Materials are currently being sourced for the construction of the model in the new year.

6.2 Future Work

Refer to the design binder for a complete Gantt Chart as well as a list of vendors contacted for the DX-SAHP project.

The project was split into four phases:

i. Engineering Research, Calculations, and Design.

ii. Material Acquisition.

iii. Assembly.

iv. Testing.

For the Fall 2021 semester, the background research, calculations, and design required to start building a working model were completed. The bill of materials is being finalized, and sponsor approval will be sought in the coming weeks. Due to COVID-19 supply chain and procurement difficulties, materials are being ordered as soon as possible to ensure project completion within the first week of April. Therefore, the timelines on the Gantt Chart are reflected to include additional contingency to ensure component delivery is met for the March 1st, 2022, assembly milestone. Finally, the testing and validation phase will take place for the remainder of the Winter 2022 semester to ensure the set design goals have been achieved.

Bibliography

[1] O. Nag, "The Sunniest Cities in Canada," Reunion Technology Inc., 11 June 2019. [Online]. Available: https://www.worldatlas.com/articles/the-sunniest-cities-in-canada.html. [Accessed 10 December 2021].

[2] Government of Canada, "Water Heaters," Government of Canada, 4 October 2021. [Online]. Available: https://www.nrcan.gc.ca/energy-efficiency/products/product-information/water-heaters/13735. [Accessed 7 October 2021].

[3] Statistic Canada, "Appendix G Estimated number of households and average household size by domain, Canada," Statistic Canada, [Online]. Available: https://www150.statcan.gc.ca/n1/pub/62f0026m/2017002/app-ann-g-eng.htm. [Accessed 22 September 2021].

[4] Department of Energy, "Heat Pump Water Heaters," [Online]. Available: https://www.energy.gov/energysaver/heat-pump-water-heaters. [Accessed 3 December 2021].

[5] Waterheatertimer, "How to install two water heaters," [Online]. Available: http://waterheatertimer.org/two-water-heaters.html. [Accessed 6 December 2021].

[6] M. Bryan, "Heat Pump Water Heaters A Hot Commodity, But Not For Everyone — Northwest News Network," Northwest News Network, 7 November 2013. [Online]. Available: https://www.nwnewsnetwork.org/energy/2013-11-07/heat-pump-water-heaters-a-hot-commodity-but-not-for-everyone. [Accessed 6 December 2021].

[7] A. Smets, K. Jage, O. Isabella, R. V. Swaaij and M. Zeman, "Solar Thermal Energy," in Solar Energy: The Physics and Engineering of Photovotaic Conversion, Technologies and Systems, UIT Cambridge, pp. 497-518.

[8] C. Williams, "HeatSpring Magazine - Which is Better: Solar Thermal Flat Plate or Evacuated Tube Collectors?," HeatSpring Magazine, 10 May 2011. [Online]. Available: https://blog.heatspring.com/solar-thermal-flat-plate-or-evacuated-tube-collectors/. [Accessed 30 September 2021].

[9] Enivronmental Protection Agency (EPA), "epa.gov," July 2015. [Online]. Available: https://www.epa.gov/sites/default/files/2015-07/documents/phasing_out_hcfc_refrigerants_to_protect_the_ozone_layer.pdf. [Accessed 3 December 2021].

[10] United Nations Environment Programme, "Handbook for the Montreal Protocol on Substances that Deplete the Ozone Layer," United Nations Environment Programme, Nairobi, 2020.

[11] Environmental Protection Agency, "Understanding Global Warming Potentials," 18 October 2021. [Online]. Available: https://www.epa.gov/ghgemissions/understanding-global-warming-potentials. [Accessed 3 December 2021].

[12] United Nations Environment Programme, "About Montreal Protocol," [Online]. Available: https://www.unep.org/ozonaction/who-we-are/about-montreal-protocol. [Accessed 3 December 2021].

[13] European Union, "Flourinated greenhouse gases and repealing Regulation (EC) No 842/2006," REGULATION (EU) No 517/2014 OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 16 April 2014 , p. 36, 2014.

[14] American Society of Heating, Refrigerating and Air-Conditioning Engineers, "Designation and Safety Classification of Refrigerants," Standing Standard Project Committee, Atlanta, 2016.

[15] Air-Conditioning, Heating, & Refrigeration Institute (AHRI), "Lower Global Warming Potential Refrigerants: Frequently Asked Questions," Arlington, 2019.

[16] M. O. McLinden, J. S. Brown, R. Brignoli, A. F. Kazakov and P. A. Domanski, "Limited options for low-global-warming-potential refrigerants," Nature Communications, p. 9, 2017.

[17] I. H. a. W. J. a. Q. S. a. L. V. Bell, "Pure and Pseudo-pure Fluid Thermophysical Property Evaluation and the Open-Source Thermophysical Property Library CoolProp," Industrial & Engineering Chemistry Research, vol. 53, no. 6, pp. 2498-2508, 2014.

[18] K. T. A. a. K. T. Ooi, "A Review on Sliding Vane and Rolling Piston Compressors," MDPI, 21 June 2021. [Online]. Available: https://www.mdpi.com/2075-1702/9/6/125/htm. [Accessed 8 December 2021].

[19] "Hermetic Compressors," [Online]. Available: https://nptel.ac.in/content/storage2/courses/112105129/R&AC\%20Web\%20files/R&AC\%20Lecture\%202/Hyperlinks/Hermetic\%20compressors.htm.

[20] "What is Hermatically Sealed Compressor? Working, Construction & Diagram.," Electrical Workbook, 2021 July 29. [Online]. Available: https://electricalworkbook.com/hermetically-sealed-compressor/.

[21] EmersonAdmin, "Introducing Copeland Variable Speed Reciprocating Hermetic Compressors for Refrigeration," Emerson Climate Conversations, 9 March 2021. [Online]. Available: https://emersonclimateconversations.com/2021/03/09/introducing-copeland-variable-speed-reciprocating-hermetic-compressors/. [Accessed 6 December 2021].

[22] Bob, "How a Refrigeration Compressor Works," 2019 October 4. [Online]. Available: https://www.compressorsunlimited.com/blog/how-a-refrigeration-compressor-works.

[23] "Vapor-compression Refrigeration," [Online]. Available: https://en.wikipedia.org/wiki/Vapor-compression_refrigeration.

[24] Bob, "What are Scroll Compressors?," Compressors Unlimited LLC, 12 March 2019. [Online]. Available: https://www.compressorsunlimited.com/blog/what-are-scroll-compressors-).

[25] D. Langenkamp, "Copeland Variable Speed Reciprocating Hermetic Compressors for Refrigeration," Emerson Climate , February 2020. [Online]. Available: https://climate.emerson.com/documents/e360-variable-speed-reciprocating-hermetic-compressors-for-refrigeration.pdf. [Accessed 6 December 2021].

[26] E. Waqar, "Air Compressor — What are the types of compressors?," Mechanical Boost, [Online]. Available: https://mechanicalboost.com/air-compressor-types-and-applications/. [Accessed 27 September 2021].

[27] "Scroll Compressor," LENNOX, 2021. [Online]. Available: https://www.lennox.com/buyers-guide/guide-to-hvac/glossary/scroll-compressor.

[28] Bob, "What are Scroll Compressors?," Compressors Unlimited International, 12 March 2019. [Online]. Available: https://www.compressorsunlimited.com/blog/what-are-scroll-compressors. [Accessed 6 December 2021].

[29] Nevada pneumatic, 2020. [Online]. Available: https://www.nevadapneumatic.com/pro-cons-centrifugal.html.

[30] G. o. Canada, "Monthly Climate Summaries," 25 November 2021. [Online]. Available: https://climate.weather.gc.ca/prods_servs/cdn_climate_summary_e.html. [Accessed 25 September 2021].

[31] A. M. a. D. Jamie Kitchen, "EEV's vs TXV's," 30 May 2017. [Online]. Available: https://www.linkedin.com/pulse/eevs-vs-txvs-jamie-kitchen. [Accessed 8 December 2021].

[32] V. Parthan, "Thermostatic Expansion Valve— Its important concepts and 2 FAQs," LambdaGeeks, [Online]. Available: https://lambdageeks.com/thermostatic-expansion-valve-concepts-and-2-faqs/. [Accessed 8 December 2021].

[33] D, J. A. Duffie and W. A. Beckman, Solar Engineering of Thermal Processess, WIley, 2013.

[34] Mathworks, MATLAB, 2021.

[35] Energy Saver, "How Insulation Works," [Online]. Available: https://www.energy.gov/energysaver/insulation.

[36] S. H. Yoon, J.-H. Choi and K.-B. Nguyen, "Effect of working-fluid filling ratio and cooling-water flow rate on the performance of solar collector with closed-loop oscillating heat pipe.," Journal of Mechanical Science and Technology 26, January 2012. [Online]. Available: https://www.researchgate.net/figure/Heat-losses-in-a-flat-plate-solar-collector_fig1_257774731.

[37] Certified Commercial Property Inspectors Association, 2021. [Online]. Available: : https://ccpia.org/types-of-low-slope-roof-thermal-insulation/)for.

[38] National Mechanical Insulation Committee, "Condensation Control Calculator - Horizontal Pipe," 28 November 2016. [Online]. Available: https://www.wbdg.org/guides-specifications/mechanical-insulation-design-guide/simple-calculators.

[39] "Mineral Wool Insulation Pros and Cons," Solar 365 the home to alternative energy, [Online]. Available: http://www.solar365.com/green-homes/insulation/mineral-wool-insulation-pros-cons. [Accessed 6 December 2021].

[40] Materion, "Thermal Emissivity and Radiative Heat Transfer," Materion Performance Alloys, June 2018. [Online]. Available: https://materion.com/-/media/files/alloy/newsletters/technical-tidbits/issue-no-114-thermal-emissivity-and-radiative-heat-transfer.pdf. [Accessed 27 October 2021].

[41] Energy.gov, "Energy Performance Ratings for Windows, Doors, and Skylights," US Department of Energy, [Online]. Available: https://www.energy.gov/energysaver/energy-performance-ratings-windows-doors-and-skylights#:~:text=Solar\%20heat\%20gain\%20coefficient\%20(SHGC,the\%20greater\%20its\%20shading\%20ability.. [Accessed 9 October 2021].

[42] Vitro Architectural Glass, "Starphire® Glass for Exteriors," Vitro, 2021. [Online]. Available: https://www.vitroglazings.com/products/low-iron-glass/starphire-ultra-clear-glass/exterior-starphire-glass/#downloads. [Accessed 25 October 2021].

[43] SABIC , "WLMW Polycarbonate Sheet," SABIC INNOVATIVE PLASTICS, [Online]. Available: http://www.polukarbonaat.ee/juhendid/docs/cerificate/WLMW_data_sheet.pdf. [Accessed 28 October 2021].

[44] JNS Glass & Coatings, "LOW IRON GLASS – ULTRA CLEAR, STARPHIRE, OPTIWHITE," JNS, 2021. [Online]. Available: https://jnsglass.com/low-iron-glass/. [Accessed 20 October 2021].

[45] Professional Plastics, "Polycarbonate Sheet - Standard Grade," Professional Plastics, 2021. [Online]. Available: https://www.professionalplastics.com/POLYCARBONATESHEET. [Accessed 15 October 2021].

[46] KALLITE Sales Division, "Fiberglass Solar Glazing and Greenhouse Covering," KALLITE, 2019. [Online]. Available: http://www.solar-components.com/SUN.HTM. [Accessed 10 October 2021].

[47] Professional Plastics, "LEXAN™ Sheet," Professional Plastics, 2021. [Online]. Available: https://www.professionalplastics.com/LEXANSHEET9034. [Accessed 12 October 2021].

[48] PALRAM AMERICAS, "SUNTUF® CORRUGATED POLYCARBONATE SHEET," Palram, [Online]. Available: https://www.palram.com/us/product/suntuf-diy-polycarbonate-corrugated-sheets/#1582184904168-2d0579aa-bf15. [Accessed 21 October 2021].

[49] A. Luible, "Introduction on use of glass in modern buildings," Research Gate, April 2002. [Online]. Available: https://www.researchgate.net/figure/Wave-lengths-transmission-of-regular-float-glass-and-low-iron-glass-in-comparison_fig6_37455557. [Accessed 2 November 2021].

[50] Plastic Genius, "Plexiglass Sheets, Fiberglass, UHMW, Polycarbonate & Engineering plastics," Plastic Genius, 20 April 2011. [Online]. Available: http://www.plasticgenius.com/2011/05/infrared-and-ultraviolet-transmission.html. [Accessed 2 November 2021].

[51] D. D. Priddy, "Why Do Polycarbonate Windows Sometimes Turn Yellow?," Plastic Expert Group, [Online]. Available: https://www.plasticexpert.com/wp-content/uploads/2019/01/Why-Do-Polycarbonate-Windows-Turn-Yellow.pdf. [Accessed 3 November 2021].

[52] K. Wilson, "Top 10 Thermally Conductive Materials," Thermtest, 2018. [Online]. Available: https://thermtest.com/thermal-resources/top-10-resources/top-10-thermally-conductive-materials. [Accessed 10 October 2021].

[53] K. Wilson, "Thermal Conductivity Of Steel," Thermtest, 4 May 2021. [Online]. Available: https://thermtest.com/thermal-conductivity-of-steel#:~:text=The\%20thermal\%20conductivity\%20of\%20steel,235\%20W\%2F(mK)\%20respectively.. [Accessed 11 October 2021].

[54] Macro Trends, "Copper Prices - 45 Year Historical Chart," Macro Trends, 2021. [Online]. Available: https://www.macrotrends.net/1476/copper-prices-historical-chart-data. [Accessed 2 December 2021].

[55] MetalMiner, "Aluminum," MetalMiner, 2021. [Online]. Available: https://agmetalminer.com/metal-prices/aluminum/. [Accessed 2 December 2021].

[56] Dampney Engineered Coatings, "Industrial high-temperature protective coatings," Dampney, [Online]. Available: http://www.dampney.com/Product-Line/AT/View/PID/2/Thurmalox-250. [Accessed 15 November 2021].

[57] G. Winton, "Serpintine Bending in Production," 2021. [Online]. Available: https://www.wintonmachine.com/serpentine-bending-in-production.

[58] "Power Plugs and Sockets of the World.," [Online]. Available: https://www.power-plugs-sockets.com/ca/canada.

[59] H. Chen, "What Voltage is used in Canada," [Online]. Available: https://lastfiascorun.com/canada/what-voltage-is-used-in-canada.html.

[60] Carel S.p.A, 05 August 2007. [Online]. Available: file:///C:/Users/jsamb/AppData/Local/Temp/EEV\%20Valve\%20Selection\%20Criteria.pdf.

[61] "Copeland Variable Speed Reciprocating Hermetic Compressors and Drives," Emerson Climate , [Online]. Available: https://climate.emerson.com/en-us/shop/1/copeland-variable-speed-hermetic-compressors-and-drives?fetchFacets=true#facet:&partsFacet:&facetLimit:&productBeginIndex:0&partsBeginIndex:0&orderBy:2&partsOrderBy:&pageView:list&minPrice:&maxPrice:&pageSize:&. [Accessed 9 December 2021].

[62] Carel S.p.A, "Electronic Expansion Valve Selection," 05 August 2007. [Online]. Available: file:///C:/Users/jsamb/AppData/Local/Temp/EEV\%20Valve\20Selection\%20Criteria.pdf.

[63] Hydro Solar Innovative Energy, "All in One Buffer Tank and Indirect Water Heater 250 L - Standard Diameter Lower & Upper Coil" Hydro Solar, [Online]. Available: https://hydrosolar.ca/collections/buffer-tank-indirect-water-heater-kit/products/all-in-one-buffer-tank-and-indirect-water-heater-250-l. [Accessed 26 November 2021].

[64] A. S. 1. (SI), "2017 Standard for Performance Rating of Expansion Valves," Air-Conditioning, Heating, and Refrigeration Institute, Arlington, 2017.

[65] Danfoss, "Electric expansion valves Type ETS 6 Datasheet," August 2019. [Online]. Available: https://assets.danfoss.com/documents/37229/AI227986437323en-000901.pdf. [Accessed November 20 2021].

[66] P. M.Gerhart, A. L. Gerhart and J. I. Hochstein, "Fundamentals of Fluid Mechanics Eighth Edition," WileyPlus, 2015, pp. 414 - 441.

[67] Parker Hannifin Corporation, Sporlan, February 2005. [Online]. Available: https://www.parker.com/literature/Sporlan/Sporlan\%20pdf\%20files/Sporlan\%20pdf\%20040/40-10-7.pdf. [Accessed November 2021].

[68] L. Molenda, "What is a Suction Accumulator?," HVAC School, 5 February 2019. [Online]. Available: https://hvacrschool.com/whast-is-a-suction-accumulator/. [Accessed 10 November 2021].

[69] B. Hess, "The Basics of Suction Accumulators in Home Heat Pump Systems," AC & Heating Connect - Emerson, April 2020. [Online]. Available: https://www.ac-heatingconnect.com/contractors/home-heat-pump-system-components-suction-accumulators/. [Accessed 20 November 2021].

[70] Danfoss - Engineering Tomorrow, "Filter driers and sight glasses," Danfoss, Tuesday March 01 2011. [Online]. Available: https://www.danfoss.com/en/service-and-support/case-stories/dcs/filter-driers-and-sight-glasses/. [Accessed 15 November 2021].

[71] J. Marchese, "Where To Install the Liquid Line Sight Glass," The News: Air Conditioning, Heating, Refrigeration, 29 August 2020. [Online]. Available: https://www.achrnews.com/articles/143707-where-to-install-the-liquid-line-sight-glass. [Accessed 27 November 2021].

[72] Danfoss, "Superheat controller, EKE 1D," September 2021. [Online]. Available: https://assets.danfoss.com/documents/189972/AN325741271542en-000403.pdf. [Accessed 25 November 2021].

[73] Maxim, "DS18B20 Programmable Resolution 1-Wire Digital Thermometer," REV: 042208. [Online]. Available: https://cdn-shop.adafruit.com/datasheets/DS18B20.pdf. [Accessed 15 November 2021].

[74] Omega, " All Stainless Steel Transducer/Transmitter Multimedia Compatibility PX309 Series," [Online]. Available: https://assets.omega.com/pdf/test-and-measurement-equipment/pressure/pressure-transducers/PX309.pdf. [Accessed 2021 28 November].

[75] Vision Turbine Flow Metera, "Models BV1000, BV2000 and BV3000 for Low Viscosity and Non-Aggressive Liquids NSF/ANSI Standards 61 and 372 Certified," Februrary 2019. [Online]. Available: https://assets.omega.com/spec/dynasonics_product_summary_brochure_ttm-br-01388-en_2.pdf. [Accessed 28 November 2021].

[76] "MegaPower (TM) Plug Power Meter Monitor Energy Watt Voltage Amps Meter with Electricity Usage Monitor, Reduce Your Energy Cost," [Online]. Available: https://www.bestbuy.ca/en-ca/product/megapower-megapower-tm-plug-power-meter-monitor-energy-watt-voltage-amps-meter. [Accessed 6 December 2021].

Appendix A: Gantt Chart

	TASK	ASSIGNED TO	PROGRESS	START	END
1	**Engineering Research, Calculations, & Design**		81%	07-Oct-21	24-Jan-22
1.1	Refrigerant Selection	Charuka	100%	14-Oct-21	28-Nov-21
1.2	Solar Collector Selection	Edwin, Nadia	100%	07-Oct-21	28-Nov-21
1.2.1	Glazing	Edwin	100%	07-Oct-21	05-Nov-21
1.2.2	Insulation	Jessica	100%	07-Oct-21	26-Oct-21
1.2.3	Absorber Plate	Edwin, Nadia	100%	24-Oct-21	19-Nov-21
1.2.3.1	Absorber Plate Coating	Edwin	100%	12-Nov-21	26-Nov-21
1.2.4	Piping Research and Pressure Drop	Edwin, Nadia	100%	29-Oct-21	28-Nov-21
1.2.5	Milestone: First Iteration of Collector Design	Nadia	100%	28-Nov-21	28-Nov-21
1.3	Expansion Valve Selection	Dhruvi	100%	12-Nov-21	19-Nov-21
1.4	Minor Components Selection	Charuka, Edwin	0%	10-Jan-22	24-Jan-22
1.5	Compressor Selection	Jessica	100%	05-Nov-21	01-Dec-21
1.6	Condenser Selection	Edwin, Nadia	80%	19-Nov-21	08-Dec-21
1.7	Data Acquisition System	Dhruvi	90%	10-Jan-22	24-Jan-22
1.8	Assembly Design	Charuka	75%	06-Oct-21	20-Jan-22
1.8.1	Schematic	Charuka	100%	06-Oct-21	11-Oct-21
1.8.2	Collector Model	Charuka	100%	27-Nov-21	30-Nov-21
1.8.3	Assembly Model	Charuka, Edwin	0%	10-Jan-22	20-Jan-22
1.8.4	Calculations	Dhruvi, Nadia, Edwin	100%	18-Oct-21	28-Nov-21
1.8.4.1	Thermodynamic Cycle Values	Nadia, Jessica	100%	08-Nov-21	28-Nov-21
1.8.4.2	Solar Collector Optimization	Nadia, Dhruvi	100%	18-Oct-21	26-Nov-21
1.9	Milestone: Final Design with all Compatible Components	Nadia	90%	01-Dec-21	01-Dec-21

	TASK	ASSIGNED TO	PROGRESS	START	END
2	**Material Acquistion**		0%	15-Dec-21	28-Feb-22
2.1	Refrigerant Procurement	Kerwin	0%	15-Dec-21	15-Feb-22
2.2	Solar Collector Components	Charuka, Edwin	0%	15-Dec-21	15-Feb-22
2.2.1	Glazing	Edwin	0%	26-Dec-21	15-Feb-22
2.2.2	Insulation	Jessica	0%	15-Dec-21	15-Feb-22
2.2.3	Absorber Plate	Nadia, Kerwin	0%	26-Dec-21	15-Feb-22
2.2.4	Piping	Edwin, Nadia	0%	15-Dec-21	07-Feb-22
2.3	Expansion Valve	Dhruvi	0%	15-Dec-21	15-Feb-22
2.4	Minor Components	Charuka, Edwin	0%	24-Jan-22	28-Feb-22
2.5	Compressor	Jessica, Kerwin	0%	15-Dec-21	28-Feb-22
2.6	Condenser	Edwin	0%	15-Dec-21	28-Feb-22
2.7	Data Acquisition System	Dhruvi	0%	30-Dec-21	31-Jan-22
2.7.1	Temperature Sensors	Dhruvi	0%	30-Dec-21	07-Feb-22
2.7.2	Pressure Sensors	Dhruvi	0%	30-Dec-21	28-Feb-22
2.8	Assembly Components	Kerwin	0%	24-Jan-22	28-Feb-22
2.8.1	Piping	Edwin, Nadia	0%	15-Dec-21	07-Feb-22
2.8.2	Insulation	Jessica	0%	15-Dec-21	07-Feb-22
2.8.3	Frame	Charuka, Edwin	0%	24-Jan-22	07-Feb-22
2.8.4	Specialized Tools	Kerwin	0%	24-Jan-22	28-Feb-22
2.8.5	General Assembly Components	Edwin, Kerwin	0%	24-Jan-22	28-Feb-22
2.9	Milestone: Complete Bill of Materials	All	80%	10-Dec-21	10-Dec-21
2.10	Milestone: Order all Components	Kerwin	0%	24-Jan-22	24-Jan-22

	TASK	ASSIGNED TO	PROGRESS	START	END
3	**Assembly**		0%	07-Feb-22	01-Mar-22
3.1	Solar Collector Assembly	All	0%	15-Feb-22	28-Feb-22
3.2	Frame Assembly	All	0%	15-Feb-22	28-Feb-22
3.3	Piping Assembly	All	0%	07-Feb-22	21-Feb-22
3.3.1	Thermal Collector to Accumulator	All	0%	07-Feb-22	21-Feb-22
3.3.2	Accumulator to Compressor	All	0%	07-Feb-22	21-Feb-22
3.3.3	Compressor to Condenser	All	0%	07-Feb-22	21-Feb-22
3.3.4	Condenser to Expansion Valve	All	0%	07-Feb-22	21-Feb-22
3.3.5	Expansion Valve to Thermal Collector	All	0%	07-Feb-22	21-Feb-22
3.3.6	Pipe Welding	All	0%	07-Feb-22	18-Feb-22
3.4	Refrigerant Integration	All	0%	15-Feb-22	01-Mar-22
3.5	Data Acquisition System Integration	All	0%	15-Feb-22	01-Mar-22
3.6	Milestone: Prototype Completed	Charuka	0%	01-Mar-22	01-Mar-22

	TASK	ASSIGNED TO	PROGRESS	START	END
4	**Testing**		0%	24-Jan-22	12-Apr-22
4.1	Fitment Assembly Testing	Charuka, Edwin	0%	20-Feb-22	01-Mar-22
4.2	Data Acquisition System Testing	Charuka, Dhruvi	0%	24-Jan-22	15-Feb-22
4.3	Heat Pump Cycle Testing	Nadia	0%	01-Mar-22	01-Apr-22
4.3.1	Temperature Verification	Nadia, Dhruvi	0%	01-Mar-22	01-Apr-22
4.3.2	Pressure Verification	Nadia, Dhruvi	0%	01-Mar-22	01-Apr-22
4.3.3	Coefficient of Performance Testing	Jessica	0%	01-Apr-22	12-Apr-22
4.4	Milestone: Project Closeout	Kerwin	0%	12-Apr-22	12-Apr-22