CHALMERS
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Internal Heat Exchanger Application in Heat Pumps

Evaluation and testing of different internal heat exchangers for efficiency improvement

Master's thesis in Sustainable Energy Systems

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Department of Architecture and Civil Engineering
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Cover: An illustration of 7000i AW heat pump.

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Abstract

A heat pump is a device which is an effective means of producing heat to satisfy heating and cooling demands for a building following the refrigeration cycle through transferring thermal energy from a colder region to a warmer region. This thesis is limited to explore options available in efficiency improvements of heat pumps focusing on cost reduction. Achieved through conducting an extensive study on efficiency improvement collecting relevant information within the scope of heat pumps.

Internal heat exchanger has been selected from the conducted literature review diving deep into this due to its adaptability to different product variants. Various options for internal heat exchanger have been formulated which has been then designed and verified. The options included are plate heat exchangers, parallel contact and concentric tube arrangement of pipes within the refrigeration circuit where there is a high possibility of heat exchange.

Parallel contact and concentric tube heat exchangers are evaluated by assessing their potential of heat transfer between the media by performing a steady state thermal analysis. The changes observed in the performance of heat pump among these options are compared by developing a simulation model. Furthermore, a conceptual design of components has been carried out covering all operating conditions and simulating it for an existing system. A sensitivity analysis has been carried out to obtain in-depth understanding of the model for their reliability. Testing of parallel contact heat exchanger is performed on a real product for the same operating conditions followed in the theoretical simulations to conform the improvements in coefficient of performance.
The results from simulations show that among the options evaluated for internal heat exchangers, concentric tube heat exchangers have the highest potential for heat transfer. Whereas the alternative of plate heat exchanger is the best it is not preferrable due to the costs. Practically a maximum of 5% increase in coefficient of performance can be observed from testing and theoretically a 4.5% increase in coefficient of performance from theoretical simulations.

Keywords: Heat Pumps, COP improvement, Internal Heat Exchangers, Plate Heat Exchanger, Concentric Tube Heat Exchanger, Parallel Contact Heat Exchanger.

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I have been fortunate to have colleagues like Abhay Hervatte who helped me formulate thermal analysis which was a crucial part of my thesis, the entire Engineering Refrigeration Module (ERM) team at Bosch Thermoteknik AB who helped me at various stages of my thesis providing me good insights about the working of heat pumps and technical nuances related to refrigeration systems in general.

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Hemanth Kumar Amperayani, Gothenburg, September 2021

Contents

Preface

In this study, the options for improving heat pump design from a refrigeration circuit point of view is investigated. The project is carried out Engineering Refrigeration Management (ERM) department at Bosch Thermoteknik AB. This department works mainly on modelling of heat pump from design perspective. The options for improving the coefficient of performance (COP) are screened for being a fit solution for the upcoming and also currently existing products of Bosch Heat Pumps developed in Tranås plant in Sweden. The modifications from this study constitute both Air water and Liquid water heat pumps. The thesis work aims to help in development process for heat pumps in the future.

Tranås, September 2021
Hemanth Kumar Amperayani

Notations

P_{COMP}	Power required by the compressor (kW)
Q_{COND}	Heat delivered by the condenser (kW)
Q_{EVAP}	Heat absorbed by the evaporator (kW)
Q_{IHX}	Heat exchanged in the internal heat exchanger (kW)
U	Overall heat transfer coefficient ($W/m^2.K$)
A	Contact area for each fluid side (m^2)
H / h	Enthalpy (kJ/kg)
T	Temperature (K)
P	Pressure (bar)
X	Quality of vapour
m_dot	Mass Flow (kg/s)
DT_SUB	Subcooling temperature difference over the condenser (K)
DT_SH	Superheat temperature difference over the evaporator (K)
VR0	Receiver Valve
VR1	Expansion Valve
LW	Liquid Water Heat Pump
AW	Air Water Heat Pump
EEV	Electronic Expansion Valve
Comp	Compressor
Cond	Condenser
Evap	Evaporator
COP	Coefficient of Performance
SCOP	Seasonal Coefficient of Performance
LMTD	Logarithmic Mean Temperature Difference
IHE	Internal Heat Exchanger

Table of Figures

List of Tables

1 Introduction

This chapter introduces the background of this thesis, there after the aim & scope, research question along with the scope of the study and the structure of the thesis.

1.1 Background

The energy used to heat the spaces we live and work in is one of the highest contributors to our individual carbon footprints. Building and construction sector account for 37% of energy demand and 38% of greenhouse gas emissions. Significant energy use in buildings are for human well-being purposes such as comfortable temperatures, air quality, lighting and cooking. Reducing the impact of heating through greenhouse gas emissions is one of the highest needs of the moment to curb global warming effects from anthropogenic emissions (Swing Gustafsson et al. 2016).

There are many approaches to reduce anthropogenic emissions, when it comes to specific product categories the best prospects are in terms of increasing energy efficiency of products that are the reason for emissions. Energy efficiency is the most cost-effective way of reducing energy consumption so reducing the emissions that result from heating buildings can make a difference in the climate impact (Heat Pump Market Size & Share, Growth | Industry Report, 2019-2025 n.d.). The trend in decarbonization and variation in energy prices leads consumer to use sustainable and cost-effective heating solution alternative instead of conventional sources of space heating technologies.

Heat pumps are energy efficient alternative for air conditioners or furnaces that are used to heat up. It uses electricity to transfer heat from thermal reservoir in the opposite direction, by absorbing heat from cold space and releasing it to a warmer one. A heat pump is a device used to heat or cool buildings using the refrigeration cycle principles using only quarter of energy needed for a gas boiler, it could also use electricity generated from solar energy to operate. The emissions from heat pumps are dependent on the energy sources that is supplied through the electricity grid. The performance of heat pumps depends on the refrigerant, seasonal performance factor and the control system (Mahmoud et al. 2021).

The climate benefits of electrification are due to (1) the superior performance of heat pumps, which use at least three to four times less energy than gas appliances, and (2) the fact that electricity grids are already using significant amounts of renewable energy and will continue to get cleaner over the lifetime of a new electric appliance (How to cut carbon out of your heating - BBC Future n.d.).

The applications of heat pumps are in domestic and industrial purposes. Due to increase in requirement of maintaining indoor environment and limitations in energy consumption, stimulated the development of heat pump technology. With expansion in domestic hot water and space heating demand in North America and European regions; energy efficient solutions are expected to gain prominence in the future by electrification of domestic heat provision. Favourable government policies related to energy consumption and tax concession offered for installing energy saving products lead to higher demand for heat pumps in residential sector. (Heat Pump Association n.d.)

Heat pumps can be sorted into two main categories based on the prime mover in use, they are compression and absorption heat pump. The former uses a compressor operated by mechanical power to increase the pressure and temperature of the refrigerant. It is suitable for small scale usage such as domestic heating or cooling systems. In absorption heat pumps an absorber and generator is installed instead of a compressor, also in addition to the refrigerant, absorbent fluid is also added which helps to increase the pressure and temperature of the refrigerant by changing the concentration of the solution. Heat pumps can be classified according to their heat sources it is illustrated in the table below.

Table 1.1: Classification of heat pumps

Heat Pump Type	**Heat source**	**Heat Sink**
Air source heat pump	Outdoor air	Indoor area / water tank
Water source heat pump	Lake/ Sewage water / Wells	Indoor area / water tank
Ground Source heat pump	Ground	Indoor area / water tank

There is space for improvements for every product, especially for a technology that is growing right now. Heat pumps are designed based on refrigeration circuit and they work both ways, to deliver heating in the winter and cooling in the summer but the focus of the product is to provide heating. This thesis aims to suggest efficiency or coefficient of performance improvement measures for heat pumps that are compatible for most types of heat pumps, thereby employing these measures can be beneficial to the operation of the heat pump.

1.2 Aim and scope

The aim of this thesis work is to understand existing heat pump technology and suggest coefficient of performance (COP) improvement measures based on a literature survey on existing technologies. The solutions that result in COP improvement should be applicable to all range of products and not just one, hence a generic solution needs to be provided to adapt the ideology into other products. Based on the literature review, one technology – an internal heat exchanger - is investigated and validated for design and performance for its application in the heat pumps in more detail. The aim is to develop and evaluate potential solutions for an internal heat exchanger that could potentially benefit the performance of the heat pump in terms of increasing the COP and higher working temperatures. These results are then presented in this report with explanations as to why these changes are observed and the reasons related to it.

1.3 Research Question

There are many methods and solutions to increase the COP of a heat pump, implementing these solutions could increase the price of the product which may or may not justify the benefits it yields. This thesis aims to address the following research questions:

- What are the proven methods and solutions that can increase the COP of a heat pump without significantly increasing the cost of the product?
- Within the scope of internal heat exchangers, which is the best method that can possibly be beneficial to implement to achieve the best performance of the heat pump?

1.4 Structure of the thesis

The structure of the thesis is as follows and the relevant information can be found the respective segments for which explanations are given below;

Chapter 1 – Introduction

This chapter introduces the background of this thesis, there after the aim & scope, research question along with the scope of the study and the structure of the thesis.

Chapter 2 – Literature Review

This chapter discusses about the relevant concepts that helped to develop a hypothetical means of heat transfer within the refrigeration circuit from practically proven examples that can be directly applied with modifications for the use case related to the thesis.

Chapter 3 – Methodology

This chapter explains the steps taken to model each solution type and the parameters that were taken into consideration for modelling the heat pump and the input data set for internal heat exchanger dimensioning.

Chapter 4 – Results & Discussions

The results from theoretical simulation of various types of internal heat exchangers and results from testing are presented here and discusses about why the results are the way they are.

Chapter 5 – Conclusion

The conclusions of the thesis work is presented which answers the question whether the application of internal heat exchangers is beneficial to the heat pump or not.

2 Literature Review

This chapter consists working principle of heat pump refrigeration cycle and literature review of potential measures of efficiency improvement for heat pumps based on the refrigeration cycle, addition of new components and refrigeration circuit concept level.

2.1 Basic Heat Pump Working Cycle

A common vapour compression heat pump is mainly composed of four components such as: evaporator, condenser, compressor and expansion valve. When the heat pump is running, saturated (or even super-heated) refrigerant vapour is sucked into the compressor Figure 3.1 in which the temperature and pressure are raised; afterwards, it passes through the condenser to exchange heat with the heating medium, which is in the secondary loop that is not depicted in the figure. The refrigerant is cooled at the same time until being saturated (or even sub cooled to the liquid region) then it comes to the evaporator by passing through the expansion valve, where expansion occurs ideally without any enthalpy change. This low-pressure liquid/vapour mixture refrigerant is then evaporated by absorbing heat from the surrounding cooling media and becomes saturated (or even super-heated vapour), when it is followed by another cycle. The difference between heating mode and cooling mode lies in whether heat supply (from the condenser side) or heat removal (from the evaporator side) is required. This will determine whether the indoor unit is the evaporator or the condenser.

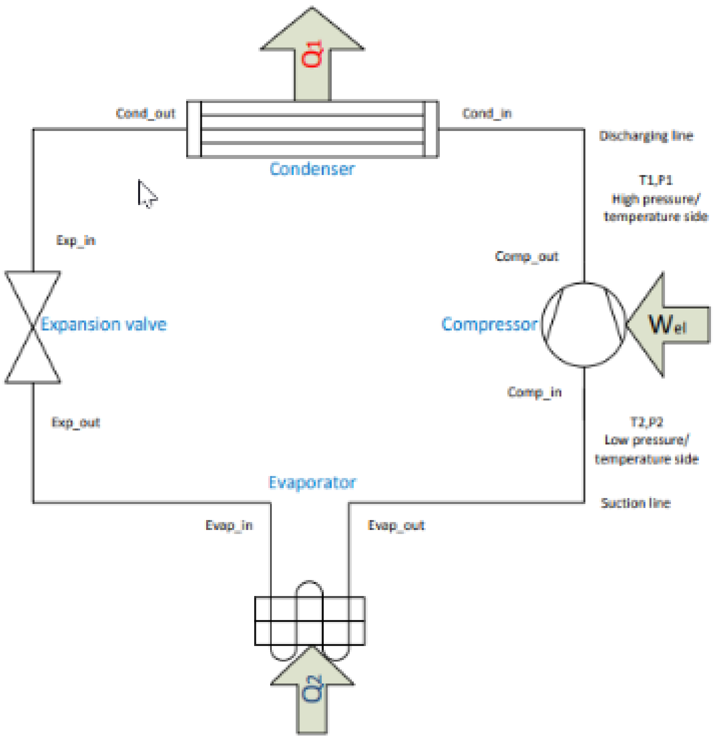

Figure 2.1: Basic Heat Pump Schematic Diagram.

The basic vapour compression process assumes an adiabatic expansion and an isentropic compression as well as neither superheating nor sub cooling. This whole process can be plotted in the p-h and T-s diagram as can be seen in below figures 3.2 and figure 3.3 respectively.

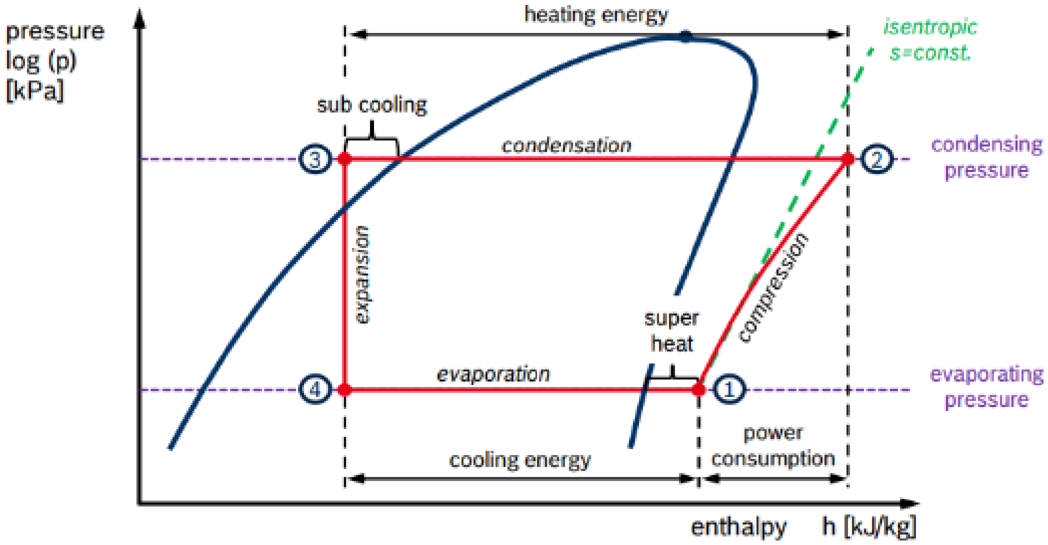

Figure 2.2: P-h Diagram.

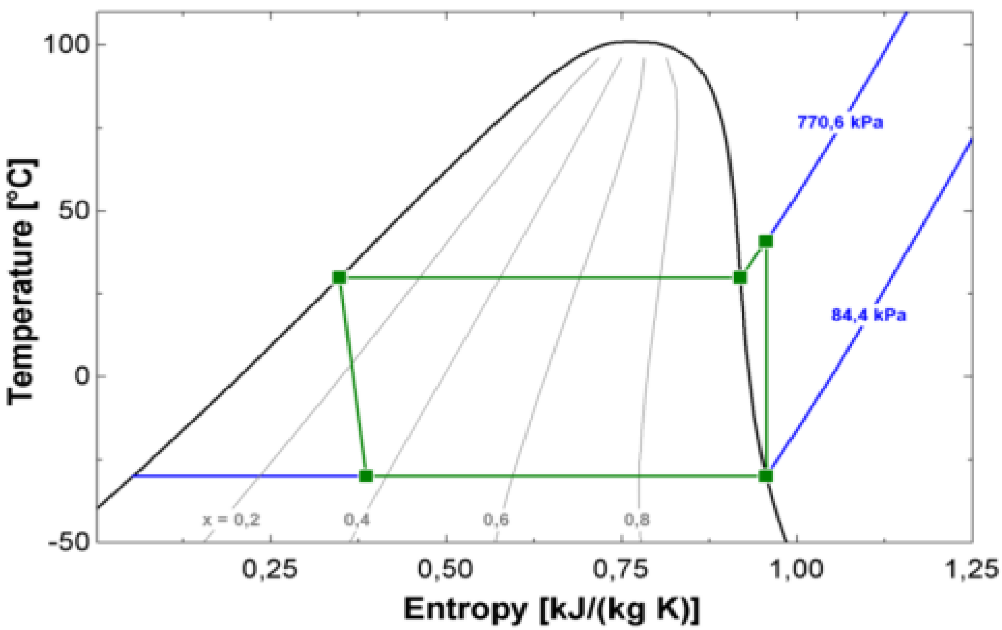

Figure 2.3: T-s Diagram

In an ideal vapour compression refrigeration cycle, the vapour refrigerant at low pressure and temperature from evaporator is drawn into the compressor, where the refrigerant is compressed isentropically and discharged with high pressure and temperature. The refrigerant then goes through the condenser, releasing heat during the condensation process. The vapour refrigerant turns into liquid phase and the subcooled refrigerant goes through the expansion valve where its pressure and temperature decreases resulting in a mixture of liquid and vapour form of the refrigerant. The liquid-vapour refrigerant evaporates and changes into vapour phase in the evaporator which acts as the heat sink. And the process repeats again form where the refrigerant enters the compressor and undergoes the cycle.

2.2 Improvement measures

This section consists of literature review of concepts that can theoretically and in few cases practically proven to improve the COP of heat pumps by bringing about changes in refrigeration circuit design, addition of components and component dimensioning.

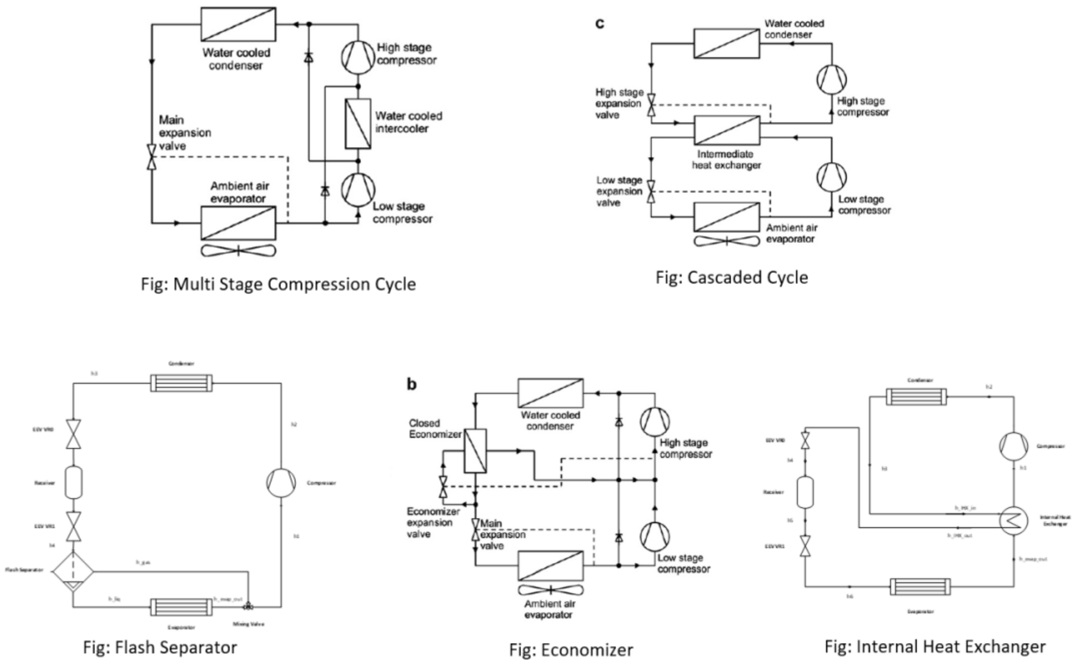

Figure 2.4: Circuit diagram of various improvement measures for heat pumps

Above figure 2.4 depicts a few samples of the refrigeration circuit diagram consisting of modifications that can help improve the performance of the heat pumps. They are explained in detail in the following subsections.

2.2.1 Cascaded Compressor

Air source heat pumps at low outdoor temperatures as low as -30 °C, a decrease in heat output can be observed. In this paper, (Bertsch and Groll 2008) an air source heat pump using R410A as refrigerant was tested at low temperatures for heating mode.

What:
A multi-stage vapour compression refrigeration cycle with two compressor stages

How:
Using a low stage compressor and high stage compressor maybe also with an intercooler in between the compressor stages.

This is achieved by fitting the heat pump with two compressors an intercooler between the compressor stages facilitating larger operating range of the heat pump in terms of outdoor temperatures.

Pros:
This method improves the COP of the system by 8% at lowest performance operating range for the use case.

Cons:
Although there are potential benefits to this method, this solution follows with drawbacks such as additional compressor and usage of oil separator to ensure appropriate oil distribution between the compressors that increases the price of the product.

2.2.2 Cascaded Condenser

Two stage compression cascade refrigeration system is widely used in air conditioning where the evaporation temperature constantly decreases, the single stage compression system will be limited by low evaporating pressure. This system consists of at least two stage compression refrigeration system that is cascaded by a heat exchanger with the other refrigeration cycle (Pan et al. 2020).

These two circuits (Figure 2.5) are differentiated as low temperature circuit and high temperature circuit in which the refrigerants are CO_2 and R404a respectively. The low temperature circuit provides space cooling.

What:
A refrigeration system with two refrigeration cycles consisting of different refrigerants in each of the cycles.

How:
The refrigeration cycles are differentiated according to low temperature circuit and high temperature circuit. The low temperature circuit provides ultimate cooling to the space. The refrigerant used are CO_2 and R404a respectively.

Pros:
The pros of using a cascaded cycle over a single stage cycle is that high-pressure ratio, high discharge and oil temperatures with low volumetric efficiencies can be avoided. Using a single refrigerant for wide range of temperatures results in high pressures in condenser, low evaporator pressure and low volumetric efficiencies.

Cons:
Not suitable for small scale heat pumps.

Table 2.1 shows the increase in COP that is observed with different combinations of refrigerants from high temperature circuit and low temperature circuit.

Table 2.1: Cascaded condenser COP variation with different refrigerant combination

High Temperature Refrigerant	Low Temperature Refrigerant	COP increase
R404a	CO_2	N/A
R717, R290, R1270, R404a (R717 ODP = 0 ; GWP = 1)	R744 (ODP = 0 ; GWP = 1)	By 53%
R152a	CO2	2.831 vs (1.901 for CO2/CO2)

Where GWP stands for "Global Warming Potential" which is an equivalent of heat absorbed by any greenhouse gas as a multiple of heat that would be absorbed by the same mass of CO_2.

The diagram below is a sample of a cascaded condenser setup, where the cascaded condenser transfers heat from the high temperature circuit to low temperature circuit.

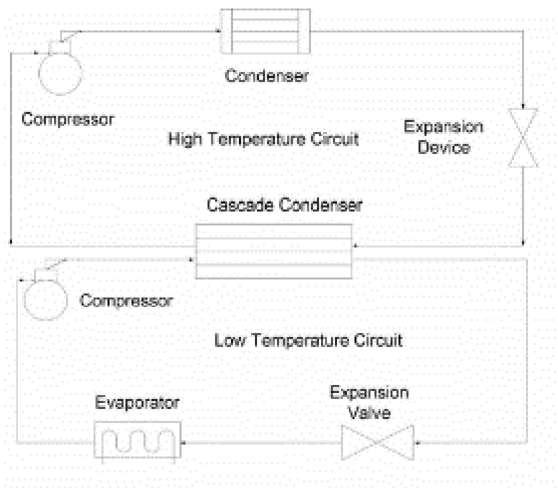

Figure 2.5: Two circuit refrigeration system circuit design (Jain n.d.)

2.2.3 Flash Separator

A flash separator is a device used to separate a fluid from different phases, usually vapour phase from liquid phase. Flash gas separators are generally used in oil refineries, refrigeration systems, power plants, chemical plants and so on. To prevent the flash gas separator from being overloaded, a sight glass or a liquid level sensing machine is used to control and monitor the flow. Figure 2.6 shows a schematic circuit diagram of a refrigeration circuit consisting of a flash separator.

What:
Two stage compression with flash separator before the evaporator. (Mbarek, Tahar, and Ammar 2016)

How:
The 2-phase refrigerant exiting from the condenser enters the flash separator before entering the evaporator, where in the gas & liquid phases are extracted from different outlet points, therefore only the liquid state refrigerant is supplied to the evaporator and the refrigerant that is already in gaseous state is supplied to the second stage compressor.

Pros:
Cooling of refrigerant exiting condenser prevents flash gas at EV and ensures superheating of suction gas, therefore liquid refrigerant entering into the compressor can be avoided.
Flash separator is used to reduce flow rate of steam to the evaporator and helps optimize its size.

Cons:
Flash distillation is not effective in separating components of comparable volatility. It is not suitable for two components systems. It is not an efficient distillation when nearly pure components are required, because the condensed vapour and residual liquid are far from pure.

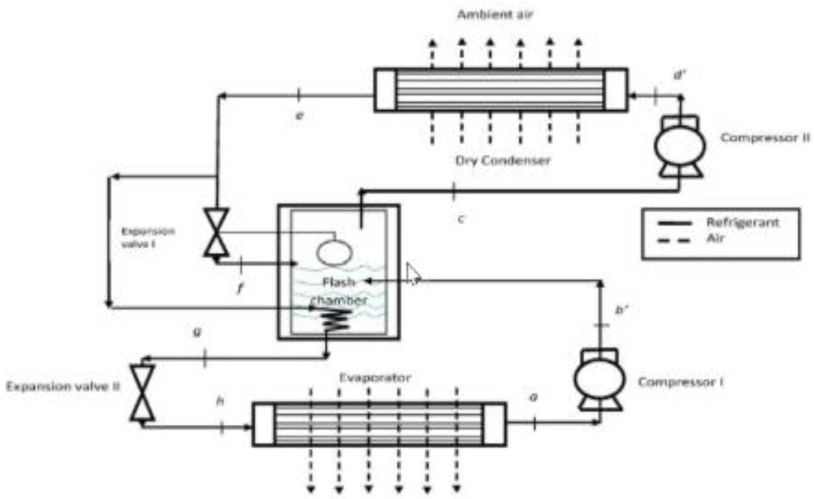

Figure 2.6: Flash separator before evaporator

2.2.4 Economizer

The economizer is an alternative for intercooler if the refrigeration circuit consists of two stage compression process. For a heat pump with single stage compression, the compressor has a supplementary inlet through which the vapourised refrigerant from the economizer flows through (G. Ma and Li 2007).

What:
The economizer delivers a certain amount of two-phase refrigerant to a mixing chamber in the suction line of the compressor where it is mixed with hot discharge fluid from the first stage compressor after which the refrigerant flows through high stage compressor (G. Y. Ma and Chai 2004).

How:
The refrigerant exists in two phase after the condenser, this two phase refrigerant in vapour phase is circulated back into the compressor meanwhile the refrigerant in liquid phase is sent to the evaporator.

Pros:
Up to 14% increase in coefficient of performance and higher flow temperatures, improved heat pump cycle in cold regions and high flow temperatures at low ambient temperatures.

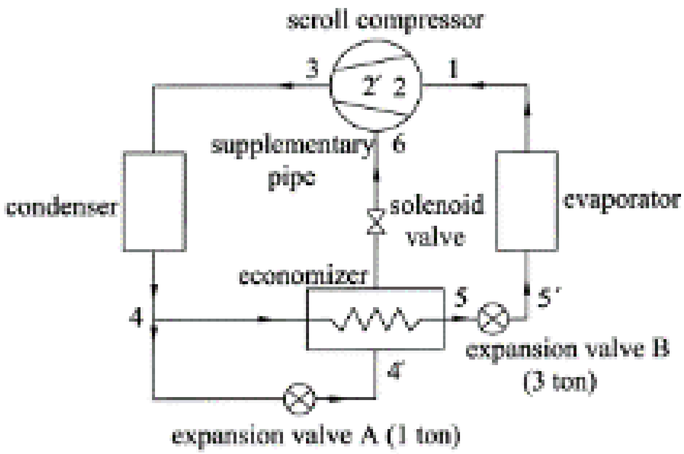

Figure 2.7: Refrigeration circuit with economizer

Cons:

The disadvantage of using an economizer is high power consumption by the compressor, due to pressure drop in the economizer heat exchanger it significantly increases the compressor power and due to superheated refrigerant vapour entering the compressor.

2.2.5 Refrigeration Circuit Configuration:

The circuits in this section consist of combination of measures that could improve the working cycle and ultimately the COP of the heat pump system.

2.2.5.1 Two stage compressors with intercooler

What: Two stage heat pump with intercooler

How: 2 compressors in arranged in series with a HX in between the outlet of 1st compressor and inlet of 2nd compressor (Bertsch and Groll 2008)

Benefits: Decreases the refrigerant inlet temperature for 2nd stage compression, extending the operation of heat pump to at very low outdoor temperatures.

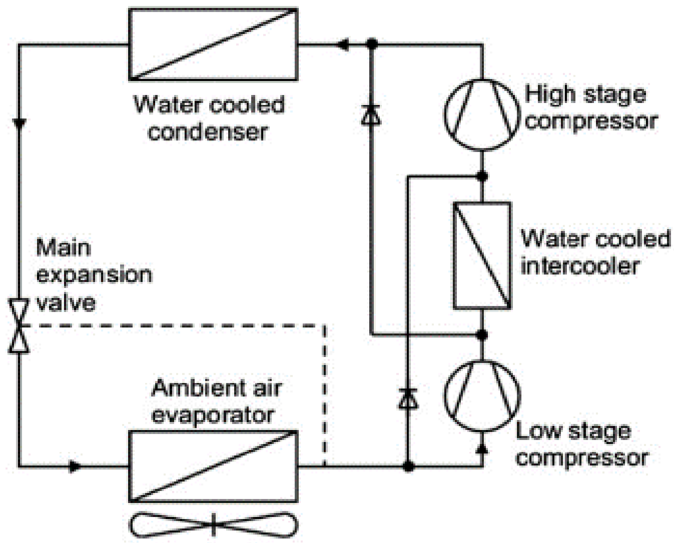

Figure 2.8: 2 Stage Compressor with intercooler

2.2.5.2 Two stage compression with economizer

What: Two stage HP with ECONOMIZER

How: 2 stage compressor cycle to be operated with economizer after the condenser.

The economizer delivers certain amount of refrigerant (2 phase) to a mixing chamber in the suction line of the HP compressor stage, where it is mixed with hot discharge gas from LP compressor stage.

Benefits: The refrigerant from the economizer sub-cools the remaining refrigerant before entering the EV, improving the system COP.

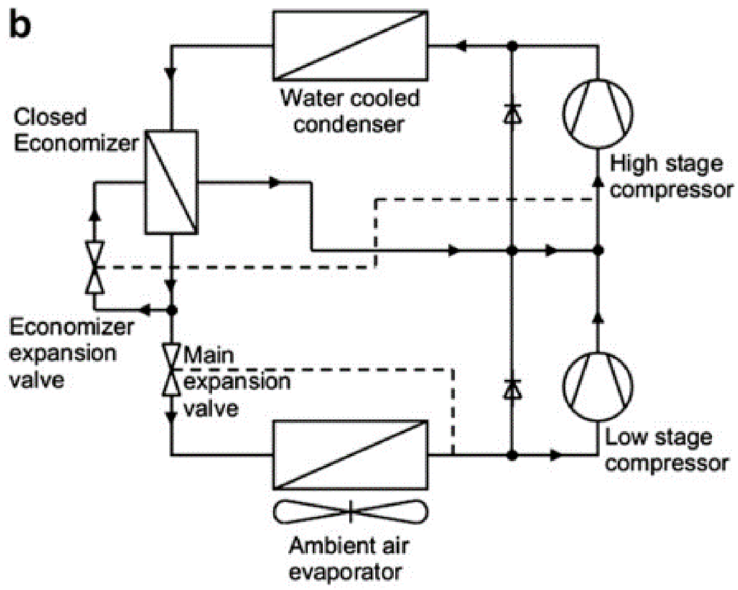

Figure 2.9: Two stage compressors with economizer

2.2.6 Internal Heat Exchanger

Internal Heat Exchangers (IHE) are used to increase the performance of per unit flow of refrigerant by transferring heat between low and high pressure flows within the circuit. Similar to economizer heat exchanging principle, of transferring heat from liquid line to the suction line. Where the liquid line has high temperature, liquid refrigerant exchanging heat with low temperature vapour phase refrigerant in the suction line to the compressor. (FJC Air Conditioning & Products ,Tools n.d.)
It has no moving parts and works by counter current movement of hot liquid refrigerant flowing through the outer tube while at the same time the refrigerant in vapour phase flows through the inner tube in a concentric pipe (Karampour, Karampour, and Sawalha 2014).

The internal heat exchanger (IHE) is often used in refrigeration systems. In general, cooling the refrigerant leaving the gas cooler prevents expansion gas at the expansion valve, and overheating the suction gas prevents liquid refrigerant from the evaporator from entering the compressor. (Aprea and Maiorino 2008)

What:
Heat transfer from high pressure hot liquid phase refrigerant to low pressure cold vapour phase refrigerant.

How:
Using plate heat exchangers or concentric pipes with a counter flow arrangement.

Pros:
Increased discharge temperatures & COP and does not require additional amount of refrigerant to be used.

Cons:
Leakages and pressure drop in the system are typical problems that arise in an internal heat exchanger. Weld defects can cause leakages and corrosion and vibrations can cause failures to the component.

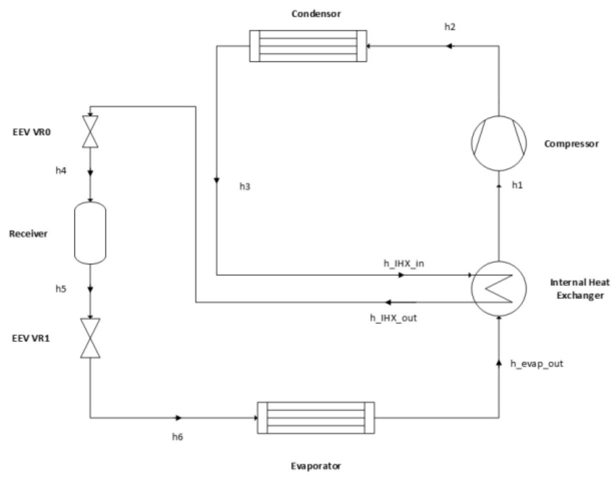

Figure 2.10: Internal heat exchanger circuit diagram

2.3 Summary of Literature Review

Below is the table which consists of summary of measures that are considered for improving the efficiency of heat pumps. While the above sections explain about the working the principles, the table in this section describes the pros and cons of implementing the changes in terms of percentage increase in change.

Table 2.2: Summary of measures from literature review

Component	Change	Pros	Cons	COP
2.2.1 Cascade Compressor	Multistage compressor with intercooler	Possible to obtain higher discharge temperature	Additional components increasing the price of the product	Approximately 8% increase
2.2.2 Cascade Condenser	2 circuits with 2 different refrigerants	Higher operating temperature ranges	Additional ref circuit	Up to 7.5% increase
2.2.3 Flash Separator	Placing flash separator before evaporator after EEV and receiver	Reduces energy of compressor (or LP compressor if multistage)	Will increase the volume of the system, thereby reducing the volumetric efficiency. But yields high discharge temperatures	From 5 – 10% increase
2.2.4 Economizer	Installing economiser after receiver and before 2nd stage compressor (if exists)	Reduces quality of vapour into the evaporator increasing the quality of refrigerant	Additional EV and piping	Up to 14% increase in COP
2.2.6 Internal HX	Heat exchange between liquid line and suction line	Less compressor speed, improves COP, higher discharge temperatures	High pressure drops across plate heat exchanger for the same mass flow rates	Up to 3.4% increase in COP

Refrigeration circuit configurations is not included in the above table because it consists of combination of individual measures, and the percentage increase in efficiency cannot be directly with the addition of measures as there are many more complexities surrounding it.

3 Methodology

This chapter explains the steps taken to model each solution type and the parameters that were taken into consideration for modelling the heat pump and the input data set for internal heat exchanger dimensioning.

3.1 Overview

To give a comprehensive overview of the methodology section and how I arrived to the solution;

Firstly, a code was written on EES for understanding the functioning of heat pump which yields the results such as COP, temperature, enthalpies, mass flow and quality of liquid phase of refrigerant at different components. This code is then modified to include an internal heat exchanger and the same output values are generated. The inclusion of internal heat exchanger involves dimensioning the heat exchanger size and its successful operating conditions at simulating condition stated in section 3.3. After which the data is characterised as a function and used in the code. Furthermore, the mass balances and energy balances are carried out at the internal heat exchanger to avoid redundancies during the simulation. The results from this simulation are checked for improvements in performance of the heat pump. Once that is confirmed, other types of internal heat exchangers are evaluated for their usability in the heat pump. A sensitivity analysis is performed to examine the change in performance of the heat pumps for varying amount of heat transfer in the evaporator and the internal heat exchanger which constitute to be superheat zone.

The alternative internal heat exchanger designs are modelled on CAD for performing a thermal analysis by taking measurements of pipes used in the refrigeration circuit and other components that are additionally used for the modifications and their material properties. The solution is developed for different designs by performing steady state thermal analysis using ANSYS and COMSOL to understand and analyse if there was any heat transfer using this method. From which the best alternative among those can be finalised. A sensitivity analysis is performed to check the amount of effective heat transfer for varying length of these type of heat exchangers.

3.2 Basic Heat Pump Circuit

The basic heat pump circuit consists of four basic components namely compressor, condenser, receiver, expansion valve and evaporator as depicted in figure 3.1. This circuit serves as basis of comparison for a heat pump circuit with internal heat exchanger.

Engineering Equations Solver is a general equation solving program that is used to solve differential, integral, non-linear algebraic equation constituting of thermodynamic database of refrigerant properties. To get familiarised with this software a theoretical code was developed to understand the working of this software. After which the simulation is performed on a pre-existing code of an already existing heat pump system.

The thermodynamic equations required for modelling of a heat pump are entered in this program to find out their properties from which the power required by the compressor, condenser heat output, evaporator heat output and the mass flow can be determined (see Appendix for a detailed description).

These are defined by assuming the common operating conditions such as condensing temperature, evaporating temperature, super heat in the evaporator and sub cooling in the condenser. The corresponding enthalpies at the inlet and outlet of each component is found creating a simulation model of the heat pump.
After which the assumed conditions are changed to possible operating conditions and data is obtained for all operating conditions, such as data required for COP calculations and temperature at different conditions. Where the coefficient of performance (COP) is determined by;

$$COP = \frac{Condenser\ Heat\ Output}{Compressor\ Power} \quad (1)$$

With the addition of components, changes will be made to this code in different versions and the corresponding output is compared to the performance of a basic circuit to assess their impacts. Such as equating the heat lost on the liquid line (represented by blue, in figure 3.1) with heat gained on the vapour line (represented by brown colour, in figure 3.1). The equations for heat lost or gained is determined by the following equation:

$$Q = \dot{m}.\ C_P.\ \Delta T \quad (2)$$

$$Q = \dot{m}.\ \Delta H \quad (3)$$

$$Q_{\ Liquid\ Line} = \dot{m}.\ \Delta H \quad (4)$$

$$Q_{\ Vapour\ Line} = \dot{m}.\ \Delta H \quad (5)$$

$$P_{\ Compressor} = \dot{m}.\ \Delta H \quad (6)$$

Δ H is the enthalpy difference between IHE on the liquid line and vapour line respectively. Moreover, the logarithmic mean temperature difference at the evaporator needs to be modified as well for maintaining the minimum temperature difference with the heat sink. A minimum 3 K temperature difference is maintained with the refrigerant and the brine in the heat sink.

The mass flow is first assumed to be of a certain value with which the initial code was developed. Further on the code used for actual simulation contains libraries of other sub codes developed by the company which constitutes of individual components and their operational limits for various conditions which helps to determine the functioning of the heat system more accurately. Such as the compressor, condenser, expansion valve, evaporator, internal heat exchanger, refrigerant characteristics like mass flow.

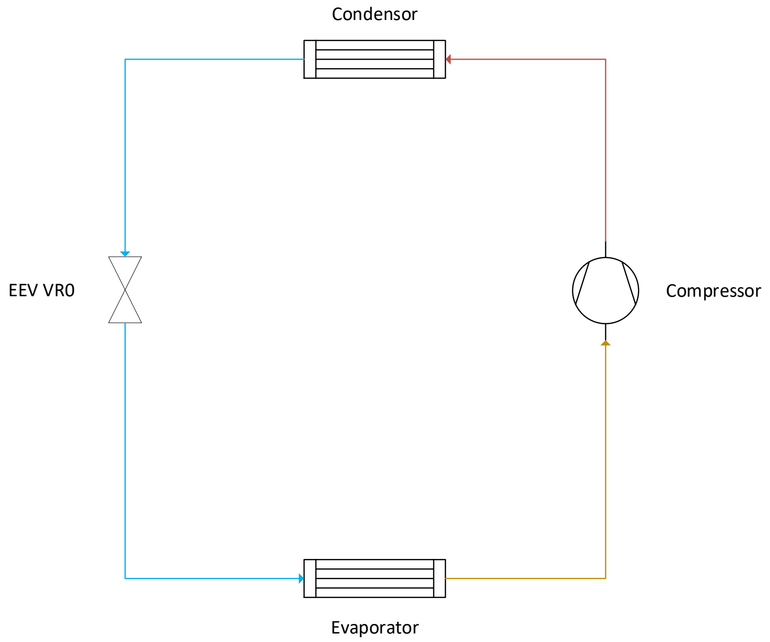

Figure 3.1: Basic Heat Pump Circuit Diagram

Parameters such as number of plates in the plate heat exchangers like the condenser and the evaporator are fixed from initial design conditions of the existing product. The product is a liquid water heat pump that is under development for future release which uses a brine solution as heat sink in the evaporator.
The minimum temperature difference from domestic hot water line and the hot refrigerant in refrigeration circuit is maintained constant at 5 K at the condenser and 3 K at the evaporator between the cold refrigerant in liquid phase and the brine solution.

3.3 Simulation Conditions

The simulations performed in the thesis follows some constant operating conditions and parameters that are fixed. Such as the refrigerant used for all the cases is R410a. The heat pump is operated in heating mode only and not in cooling mode. The maximum pressure drop across the heat exchanger is limited around 1 bar. Flow rate of the refrigerant is maintained constant at 0.04 kg/s for initial simulations. Subcooling in the condenser and superheat over the evaporator is maintained at difference of 5 K from inlet to outlet. Whereas the temperature difference over the internal heat exchanger is considered to be within the limits of subcooling on the liquid side and superheat on the suction side.

The heat pump modelling is only concerned with the refrigeration circuit and does not take into account the brine side circuit or the domestic water circuit of the heat pump and water circulation system.

3.4 Internal Heat Exchanger

This section discusses about various alternatives implemented to the heat pump circuit as internal heat exchangers to observe changes in performance. The list below represents the types of internal heat exchangers being considered for this thesis. They are considered based on availability of already existing product and possibility to modify the refrigeration circuit with least effort.

1. Plate heat exchanger (PHE)
   A PHE is used to achieve the heat transfer between the fluids.

2. Concentric Tube Heat Exchanger
   A tube within a tube wherein the fluids flow and the heat transfer is achieved by

3. Parallel Pipe Heat Exchanger
   a. Brazed pipes in parallel
      Heat transfer is achieved by brazing two pipes together on the side facilitating contact for heat transfer.

   b. Clamping pipes in parallel
      Pipes are clamped together to maintain contact between them to facilitate heat transfer.

The plate heat exchanger is dimensioned using the software developed by the manufacturer which accurately provides the operating points and necessary key performance indicators. For the other options, a static thermal analysis is performed to check for heat transfer if any that happens.

3.4.1 Plate Heat Exchanger

This circuit consists of an internal heat exchanger (IHE) on the suction line which is of plate heat exchanger type. The heat is transferred from liquid line after the condenser and before expansion valve to the suction line before the compressor and after the evaporator as shown in figure 3.2. Furthermore, the heat exchanger selection such as the product type, size and number of plates are determined by supplier software “SWEP SS G8”, which suggests appropriate heat exchanger that meets the needs. It is equipped with calculation templates that are useful for the parameters being dealt with in the simulation model.

The selection of plate exchanger is done using the software SWEP SSP G8 (SWEP calculation software - SWEP n.d.), wherein the flow conditions can be given as input for a single-phase counter flow heat exchanger and the software suggests appropriate heat exchanger for the purpose.
The heat exchanger type with the highest performance rating is chosen from the suggested alternatives from the supplier’s line of products that the software suggests. In this case “B3” type heat exchanger from SWEP is the best plate heat exchanger that can be used for internal heat exchanger.
Initially this heat exchanger needs to have 16 plates for the use case specified. Further a sensitivity analysis is performed for different flow conditions and with different number of plates.

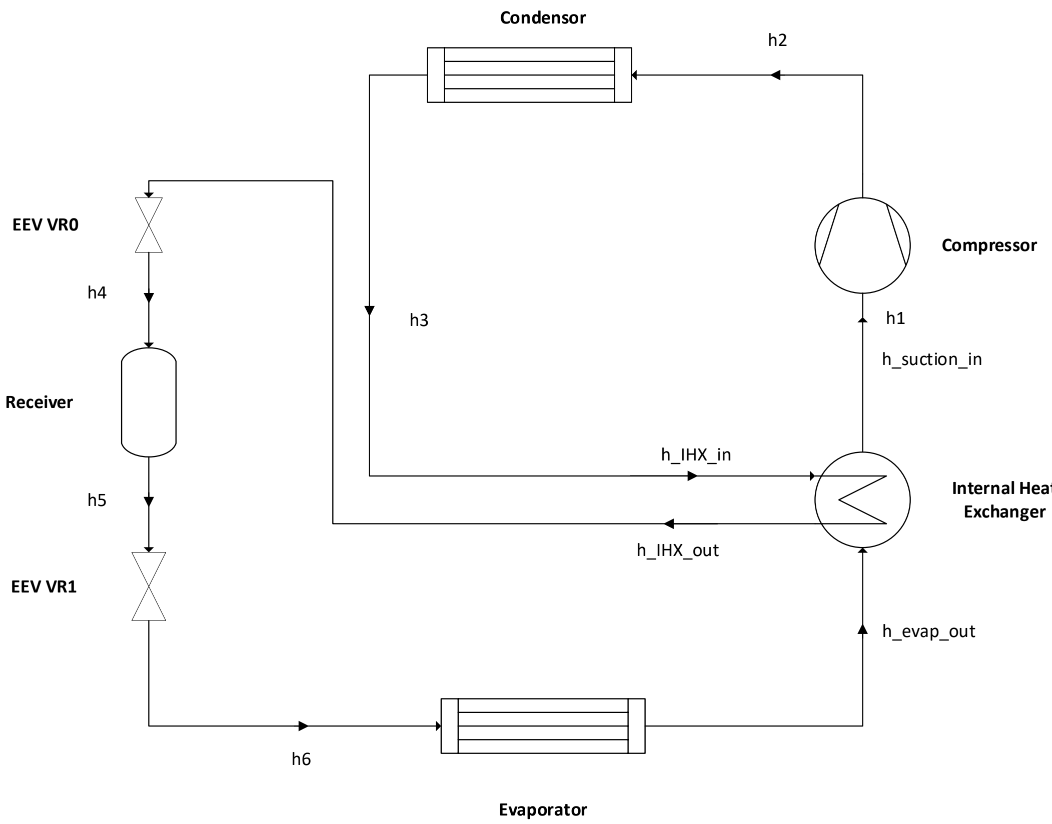

Figure 3.2: Internal Heat Exchanger Circuit Diagram

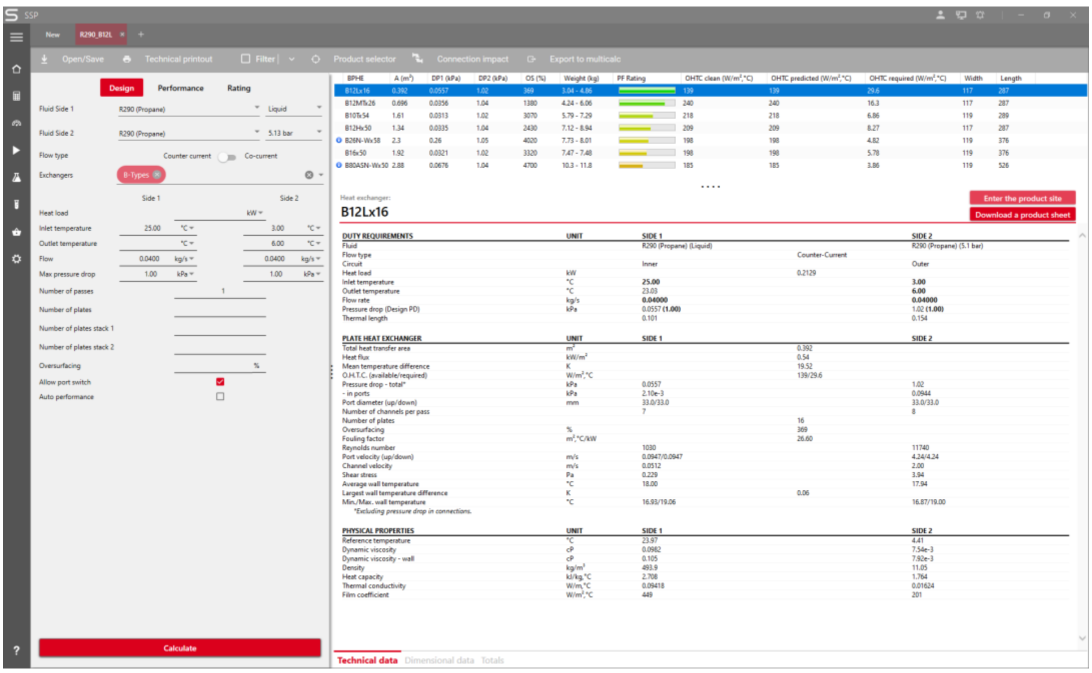

Figure 3.3: SWEP SSP G8 Heat Exchanger Evaluation

The result from this sensitivity analysis is to understand the heat transfer coefficient and outlet temperature from the internal heat exchanger on the liquid side, pressure drop across the heat exchanger, logarithmic mean temperature difference and the energy of heat transferred in Watts (W).

Using this data, a library for internal heat exchanger is created which consists of data from the heat exchanger at different operating points that is used in the heat pump model in EES, which can deliver results for different operating conditions when specified. Due to the way the code was designed, the IHE library does not automatically solve the logarithmic mean temperature difference (LMTD) that arises during the simulation. Hence energy balances and mass balances are considered at this point to avoid these error points. LMTD is the driving force for heat exchange between two fluids, higher the value of LMTD the amount of heat transfer between fluids increase. It is given by the formula;

$$Q = U.\ A.\ LMTD \tag{7}$$

Where “U” is the overall heat transfer coefficient and “A” is the area of heat transfer.

3.4.2 Concentric Pipe Heat Exchanger

In this case, instead of using a plate heat exchanger the suction line pipe engulfs the liquid line pipe because one is bigger than the other facilitating heat transfer to each other. This solution is developed on COMSOL, where the material geometry is created consisting of same dimensions as the pipes that exist in the heat pump prototype. The material of the pipe is selected as copper and refrigerant used is R410a (Lin 2014).

Dimensions of the pipe are as follows;

Length: 300 mm
Outer Dia.: 10 mm
Inner Dia.: 5 mm
Thickness of the pipe: 1 mm

The length of the pipe is assumed to be 300 mm because through visual inspection of the refrigeration circuit of the heat pump, this was approximately the space available to implement this solution.
The thermal properties of refrigerant are as follows for the initial flow conditions, these properties are obtained from EES software using the available temperature and vapour quality conditions of the refrigerant. Mass flow rate of the refrigerant is maintained constant at 0.04 kg/s.

Table 3.1: Refrigerant Properties for initial flow conditions

R410a	**Liquid Line**	**Suction Line**	**Units**
In Temp	25 (298)	3 (276)	°C (K)
Density	610.8	33.66	kg/m^3
Pressure	16	7	bar
Heat Capacity, C_p	1.705	1.158	kJ/kg.K
Kinematic Viscosity	1.15E-07	3.41E-07	m^2/s
Dynamic Viscosity	7.05E-05	1.15E-05	kg/m.s

The heat transfer is considered to be 1 dimensional steady state heat transfer, assuming turbulent flow of refrigerant in the pipes and heat loss to surrounding air is neglected assuming it is insulated in reality. After developing the model, the output that are interesting to look at are the outlet temperatures of both liquid line and suction line, LMTD, heat transferred between the refrigerant through the pipe in Watts (W), pressure drop across the pipes.

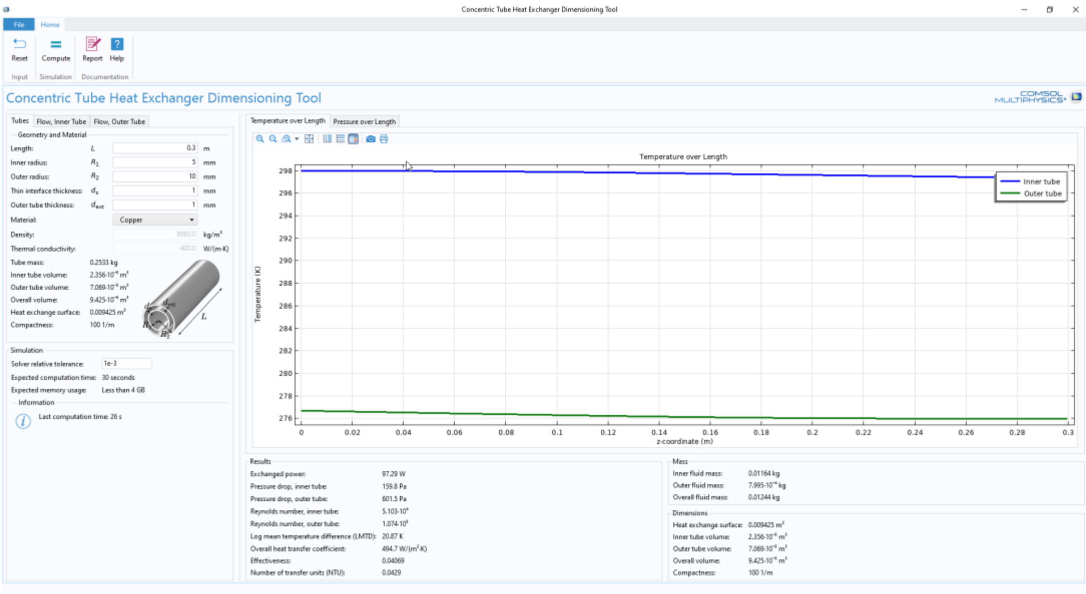

Figure 3.4: Results from COMSOL for concentric tube heat exchanger

The above image is the results that is obtained after simulation, the graph signifies the LMTD in the concentric tubes. Comparing results from other alternatives involving tubes, this method yields the highest potential for heat transfer.

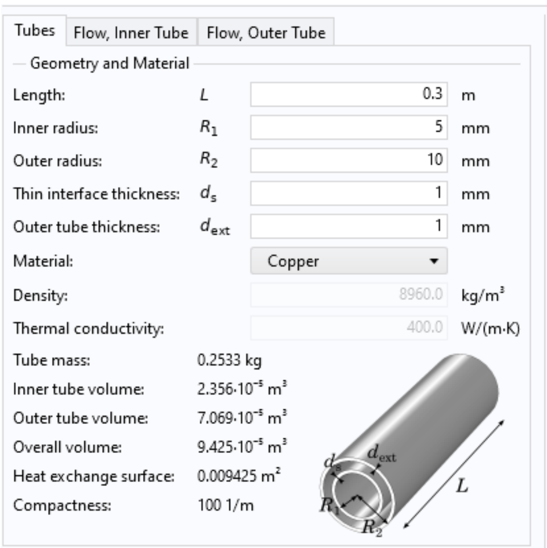

Figure 3.5: Material Properties of Concentric Tube Heat Exchanger

The material properties are manually being input for the analysis to maintain similarities within analysis for different solutions which do not have the same geometry, but they use the same material for reaching a common basis of comparison.

3.4.3 Parallel Pipe Heat Exchanger

Pipes are in parallel contact with each other to transfer heat from one pipe to the other. They are done in so 2 different ways, by brazing the pipes together and clamping the pipes onto alumina plate that is already existing in other products for heat transferring between suction line and inverter. Thermal analysis is performed for these pipes in contact on ANSYS using the same properties as before.

3.4.3.1 Brazing Pipes

Copper pipes of the same dimensions are brazed together for a length of 300 mm and thermal analysis is performed for the model. The sample in figure 3.5 is fabricated to visualise and understand the brazing profile. This lead to formulating a theoretical 1 dimensional steady state heat transfer model for evaluating the heat transfer for varying sizes of brazing area by changing the length of brazing.

Figure 3.6: Sample of pipes brazed together

Figure 3.6 shows the geometric model on which heat transfer analysis is performed. The material of the pipe and brazing area is considered to be the wall and the refrigerant. The brazing area serves has heat transfer medium in this method, therefore a sensitivity analysis is performed on varying lengths of brazing, which directly changes the area available for effective heat transfer between the pipes.

Figure 3.7 represents the CAD model developed initially for performing thermal analysis for brazing pipes in parallel solution. This CAD model does not represent the final model on which the thermal analysis was carried out on because this does not accurately represent the brazing area.

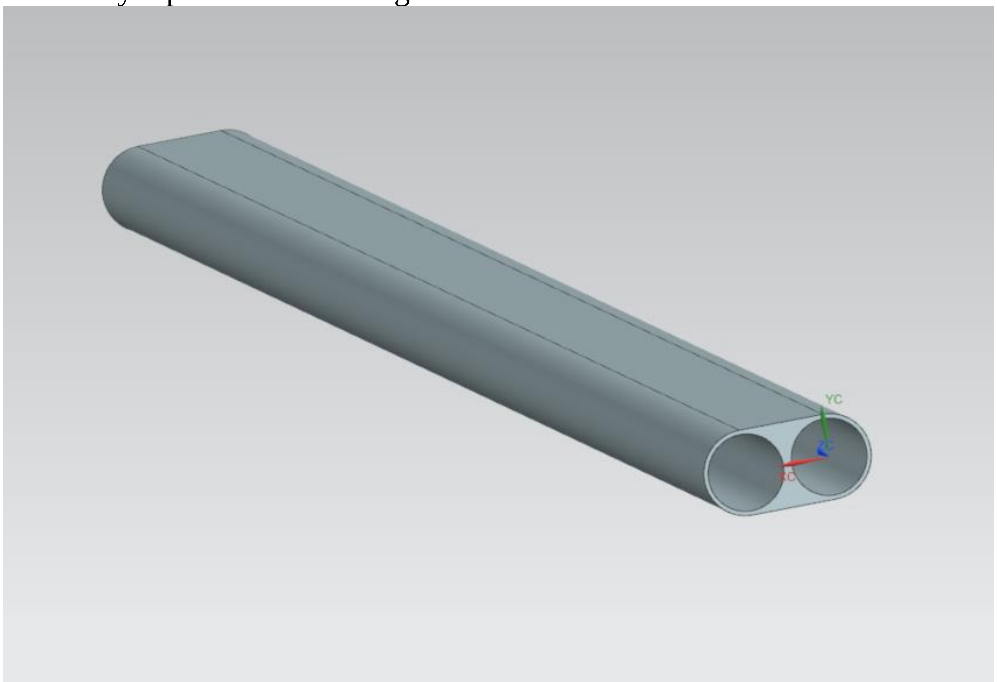

Figure 3.7: Heat Transfer in Brazed Pipes

The amount of effective heat transfer area in this model is higher than it is supposed to be. The modified geometry which has an accurate geometry that is close to reality can be found in the results section (Figure 4.8 & 4.9). Since the accurate CAD model was developed in ANSYS, a CAD model without the thermal analysis results could not be found hence the initial model is used here from which a better model is developed.

3.4.3.2 Clamping pipes using an Aluminium Plate

The copper pipes are clamped onto an Aluminium plate with circular cavity that fits the profile of the pipe. The ideology behind using an Aluminium plate is because it serves for better surface contact area between both the pipes and this sort of plate is already being used for heat transfer at a different location in the heat pump circuit.

Aluminium as a metal has very good heat transfer characteristics and this attachment is wraps the pipes with plate using an insulation material. Figure 3.7 represents the geometry of the proposed solution, where the copper pipes are mounted on to the plate made of aluminium.

This solution was interesting to look into because there already exists a method of heat transfer in heat pump used in one of the models developed by Bosch where their heat is transferred to another component for cooling purposes for which ha similar clamp in this case is used with the same dimensions. Therefore, since the components are already available, it was interesting to look into this type of parallel contact heat exchanger.

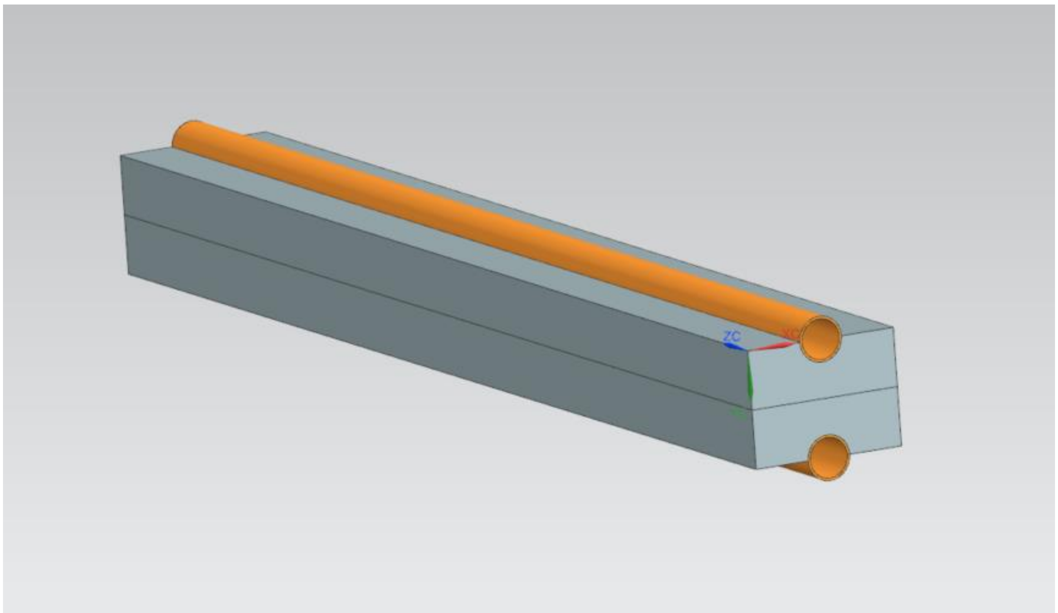

Figure 3.8: Heat Transfer through Aluminium Plates

3.5 Prototype Testing

The testing is performed in climate-controlled chamber for 2 different ambient temperature conditions of 5 °C and 15 °C. The ambient temperatures can be much lower than the chosen conditions but to reach these low ambient temperatures in the climate chamber it consumes a lot of time. The lower and upper limit of temperature represents the operation of heat pump in winter and summer conditions for heating and cooling mode. The sample is modified with a close approximation to "Clamping pipes using an Aluminium plate" alternative. An aluminium plate with a semi-circular cavity to mount pipes is already being used for another purpose in the heat pumps, therefore they are being re-purposed here for sample modification. The clamp serves as extra surface area of contact between two pipes, and aluminium is considered to have good heat transfer characteristics. A heat pump is disassembled and an appropriate spot for having two pipes in contact is identified and the sample modification is performed. The sample modification steps are explained below;

Figure 3.9: Sample Modification 1

Figure 3.9 shows that aluminium plate is being mounted on a copper pipe from suction line that is closest to the filter drier (black coloured device) from liquid line. The air gaps between the aluminium plate and the filter drier is filled up using aluminium foil after which they are clamped closer together using screw type clamps to tighten the contact and reduce any air gaps. This also helps to keep the modification firmly in place and does not move around due to vibrations.

Figure 3.10: Sample Modification 2

Finally, the whole modification setup is insulated to reduce heat losses to the surroundings as shown in figure 3.11.

The test is first conducted for the sample prototype without any modifications to the heat pump. After these tests are conducted the data from testing is extracted and analysed at stable operating points where the heat pump has reached the set point of parameters like humidity, condensing temperature, compressor speed, suction line superheat [K] at the evaporator and ambient temperature of the climate chamber. The data is extracted using MATLAB and analysed by checking the key performance indicators of the heat pump and comparing them with theoretical results.
The same procedure is continued again for the modified prototype sample.

4 Results and Discussions

The results from theoretical simulation of various types of internal heat exchangers and results from testing are presented here and discusses about why the results are the way they are.

4.1 Plate Heat Exchanger

The results from evaluating a plate heat exchanger for an internal heat exchanger is described in this section. The results are obtained from theoretical simulations of a heat pump in EES and the heat exchanger selection is performed using SWEP SSP G8 software. The results are explained by comparing the results of COP and discharge temperature from the compressor between the circuit without internal heat and with internal heat exchanger (IHE).

The graph below shows the comparison of COP between heat pump without and with IHE. The solid lines represent the COP of heat pump without IHE and the dotted lines represent the COP of heat pump with IHE. The data is extracted for compressor speeds of 30, 60 and 90 RPS. It can be observed that the COP of heat pump with IHE is higher than the heat pump without IHE. In the graph “35” represents condensing temperature of 35ºC and “35 W” means condensing temperature of heat pump 35ºC with IHE in the system.

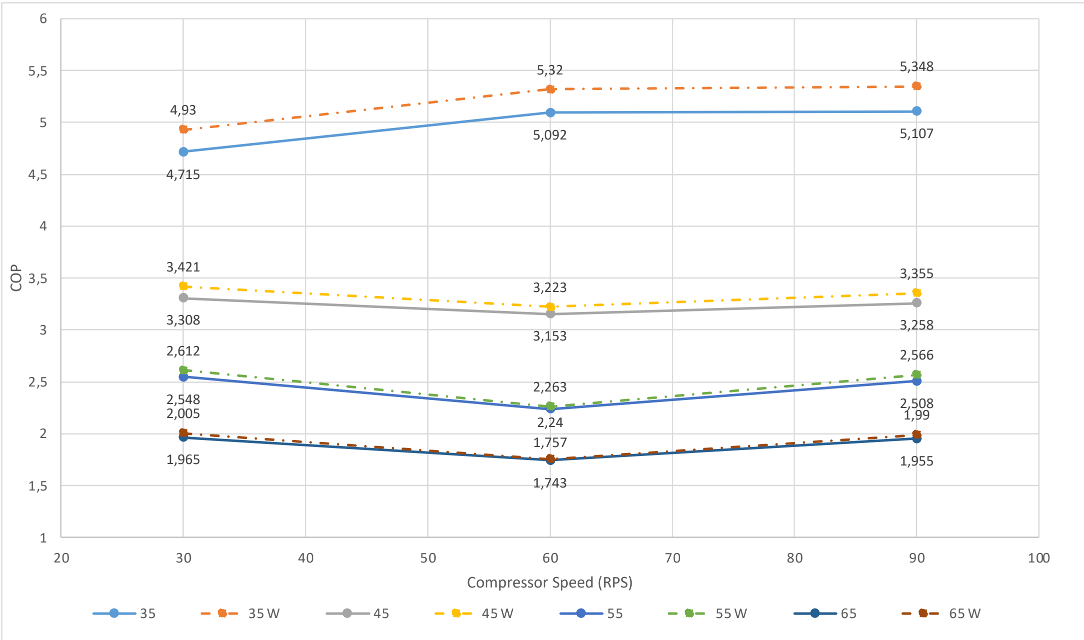

Figure 4.1: Compressor Speed vs COP

The increase in COP is due to higher heat output at the condenser, since the enthalpy is increasing due to increase in discharge temperature from the compressor after the introduction of IHE at the same compressor speed. The increase in discharge temperature from compressor outlet is shown in the figure below. The same terminology follows, where the dotted lines represent heat pump with IHE and solid represent the heat pump without IHE.

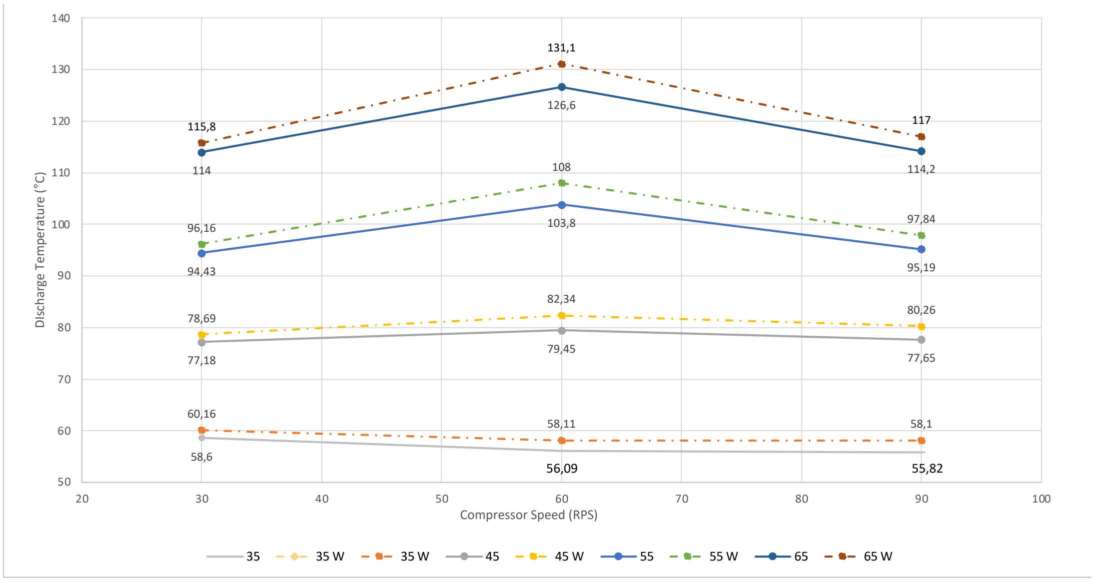

Figure 4.2: Compressor Speed vs Discharge Temperature

Higher discharge temperature directly correlates to higher heat output from the condenser. Figure 4.3 depicts the increase in heat output for different compressor speeds where the temperature is bound to increase for increase in compressor speed. For every compressor speed the heat pump with IHE represented by dotted lines are at an elevated position than solid lines which represent the basic heat pump circuit.

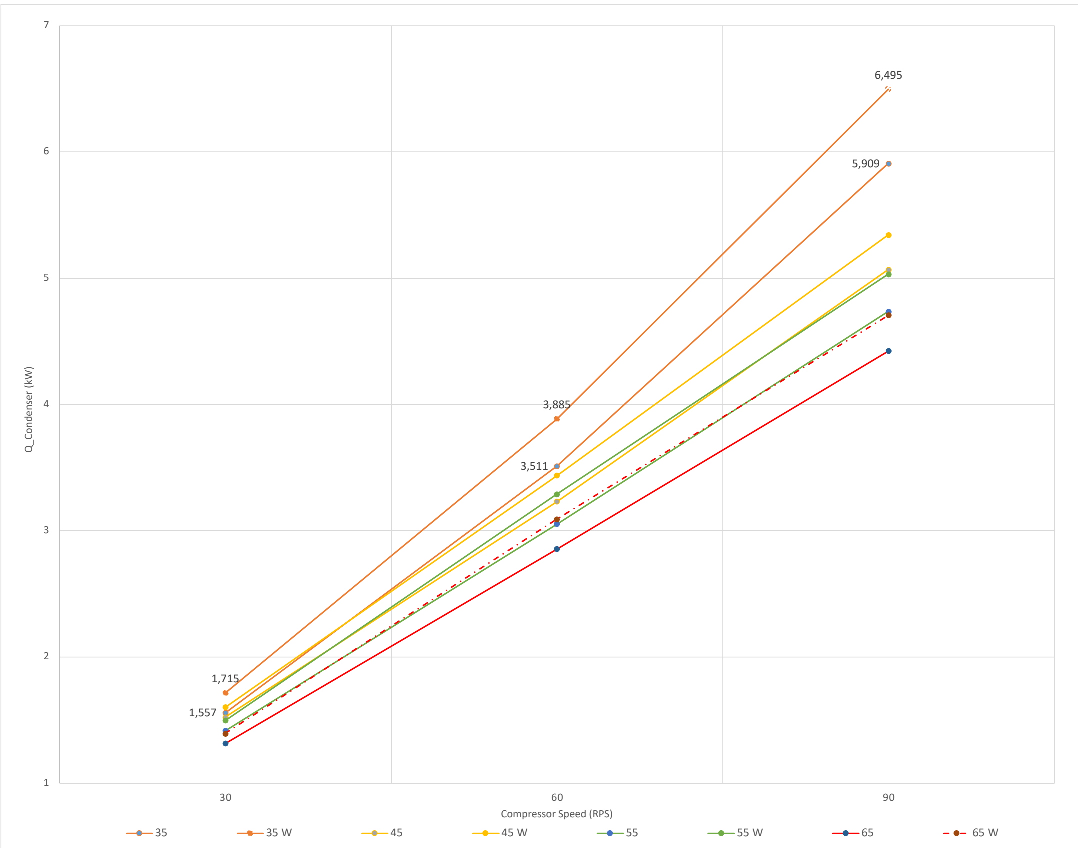

Figure 4.3: Compressor Speed vs Heat Output of Condenser

As the cold low pressure refrigerant gains temperature over the internal heat exchanger, the volume the refrigerant increases due to increase in density. This results in limitations on the compressor with limited volumetric efficiency. Hence the compressor requires more power for performing the same amount of work for given amount of refrigerant charge. Hence the heat pump with IHE has higher compressor power consumption than the heat pump without IHE.

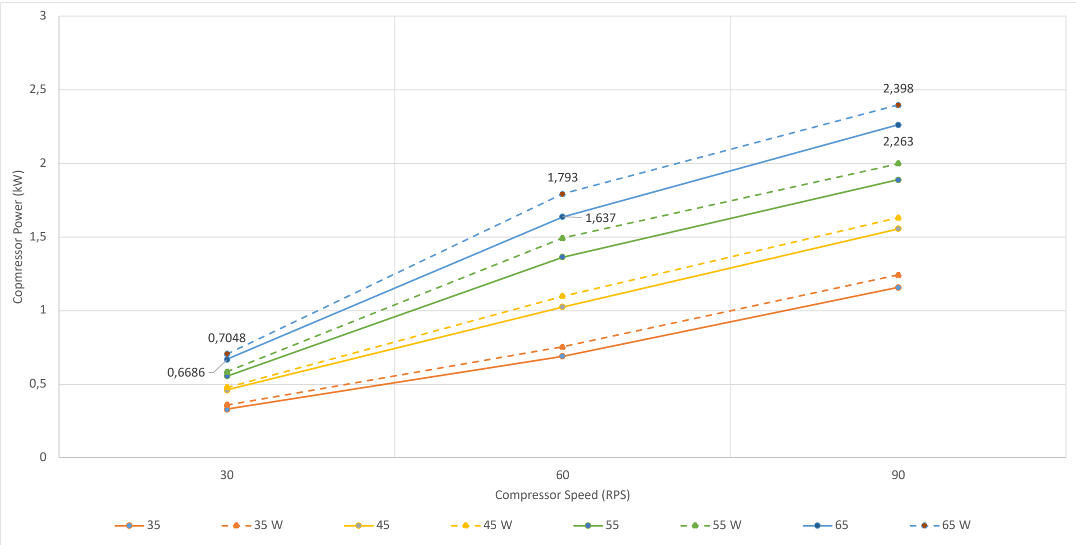

Figure 4.4: Compressor Speed vs Compressor Power Consumed

In figure 4.4, One can note that employing the internal heat exchanger increases the specific refrigerating effect, on the other hand, the specific volume of the refrigerant vapour at the beginning of the compression rises and, as a consequence, the specific compression work increases too.
The system coefficient of performance, that is the ratio of the refrigerating effect to the compression work, could be higher or lower than one of a cycle without internal heat exchanger. (Aprea and Maiorino 2008)

4.1.1 Sensitivity Analysis

The superheat zone constituting of evaporator and internal heat exchanger is 6K. It is split between 5K in the evaporator and 1K in the IHE and 1K in the evaporator and 5K in the IHE.

In figure 4.5, the iso-lines represent the trend for COP values of heat pump with IHE and the solid lines represent the values for heat pump without IHE. The heat pump with IHE has slightly better COP when majority of the heat transfer is happening at the evaporator and little of it in the IHE. Since the compressor work increases along with the volumetric mass of the refrigerant at 60% and 90% compressor speed, the COP increase saturates and does not increase much as it does when compared to increase from 30% to 60% compressor speed.

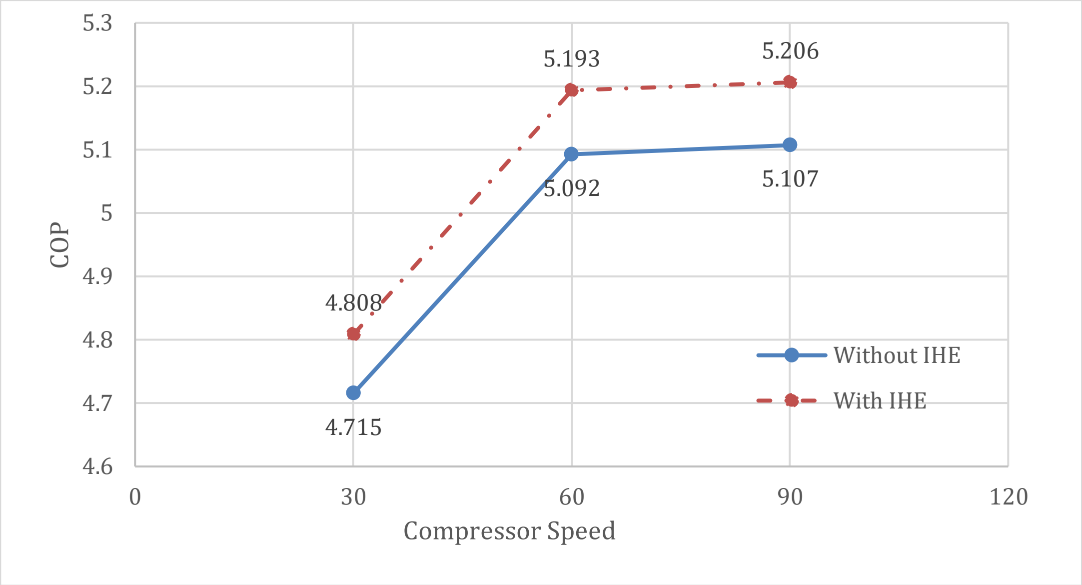

Figure 4.5: Compressor Speed vs COP for 5K ΔT in IHE

In figure 4.6, the iso-lines represent the trend for COP values of heat pump with IHE and the solid lines represent the values for heat pump without IHE. The heat pump with IHE has slightly better COP when majority of the heat transfer is happening in the IHE and little of it in the evaporator.

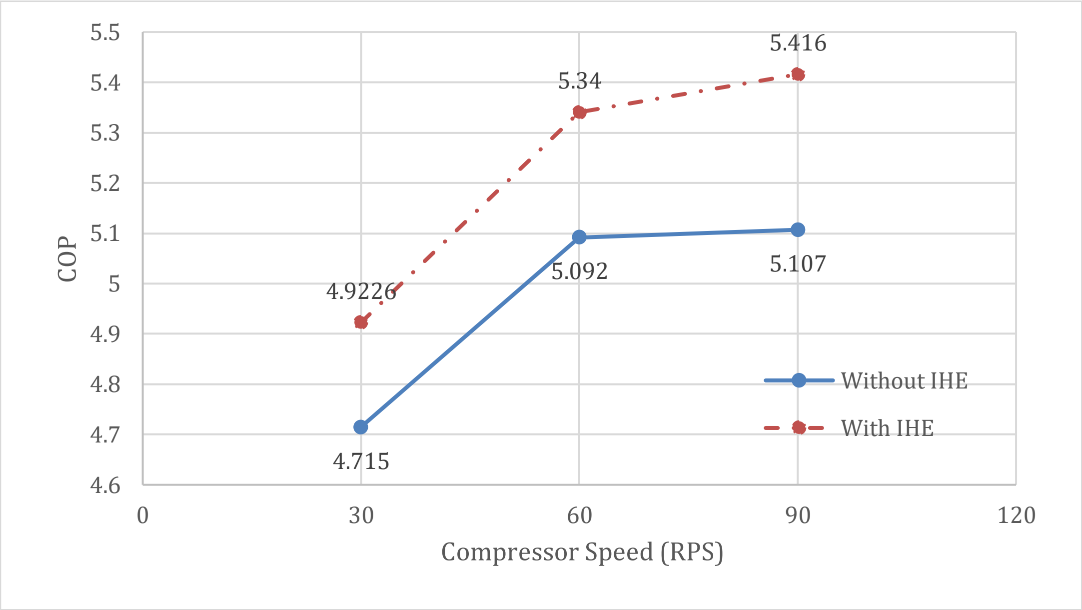

Figure 4.6: Compressor Speed vs COP for 1K ΔT in IHE

From this sensitivity analysis we understand that higher COP increase can be observed if majority of the heat transfer occurs at the IHE rather in the evaporator. This is due to the fact that at the IHE, the refrigerant in liquid phase has higher enthalpy which serves for effective heat transfer.

4.2 Concentric Pipe Heat Exchanger

The concentric pipe heat exchanger results are as shown below in Figure 4.7 which indicates results about the amount of exchanger power and pressure drops in liquid line and suction line, the overall heat transfer coefficient, effectiveness and number of transfer units.

Results

Exchanged power:	97.29 W
Pressure drop, inner tube:	159.8 Pa
Pressure drop, outer tube:	601.5 Pa
Reynolds number, inner tube:	$5.103 \cdot 10^4$
Reynolds number, outer tube:	$1.074 \cdot 10^5$
Log mean temperature difference (LMTD):	20.87 K
Overall heat transfer coefficient:	494.7 $W/(m^2 \cdot K)$
Effectiveness:	0.04069
Number of transfer units (NTU):	0.0429

Figure 4.7: Thermal Analysis Result for Concentric Pipe Heat Exchanger

After performing the analysis, the sliced view cross section of the model showed significant temperature gain on the low-pressure vapour refrigerant side, which can also be confirmed by looking at results which has better LMTD value and overall heat transfer coefficient when compared with other alternatives.

The heat driving force is determined by the factor LMTD, in this case the LMTD is about20Kwhereas in other alternatives the LMTD values are less than 10 K. The overall heat transfer coefficient increase in area of contact, when dealing with alternatives for parallel pipe heat exchangers the area of contact is only one side of the pipes. Whereas in this case the pipes are enclosed within one other therefore maintaining a higher surface area contact around the pipe which leads to better and efficient heat transfer from one resistive medium to the other. From equation (7) it can be said that the amount of heat transfer is directly proportional to overall heat transfer coefficient (U), contact area available for each fluid side (A) and the LMTD in which all the factors seems to increase for a concentric tube heat exchanger resulting in higher heat transfer potential.

4.3 Parallel Pipe Heat Exchangers

4.3.1 Brazing Pipes

The pipes are connected in parallel through brazing the pipes together by adding a filler material of the same kind (Cu) to maintain homogeneity in heat transfer characteristics. The pipe is modelled in ANSYS and the heat transfer characteristics are checked. In comparison to the concentric pipe heat exchanger, this method has very minimal heat transfer characteristics.

In Figure 4.8, the cold pipe is ranging from temperature of 276 K to 281 K. Whereas in the Figure 4.9 the hot pipe from the condenser outlet is varying from 298 K to 297 K. With a temperature drop of 1 K on the hot side the cold side is gaining about 4 K, this is due to change in heat capacity with the change in phase of the refrigerant.

It can be observed from figure 4.9 that heat transfer does not occur to the bulk of the refrigerant fluid but there is heat transferred to the walls. For the given length of brazing the heat transfer is happening just up to the walls of the pipes and for the heat to flow into the fluid and gain heat on the cold side using this method, a longer length of brazing is required to achieve the heat transfer.

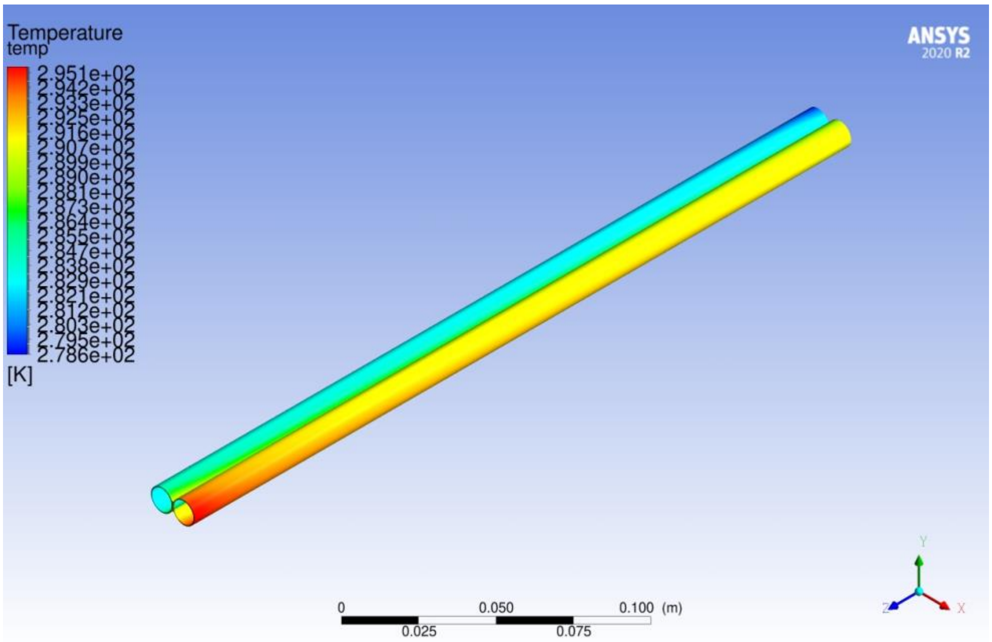

Figure 4.8: Thermal Analysis Result, Temperature Profile of Wall

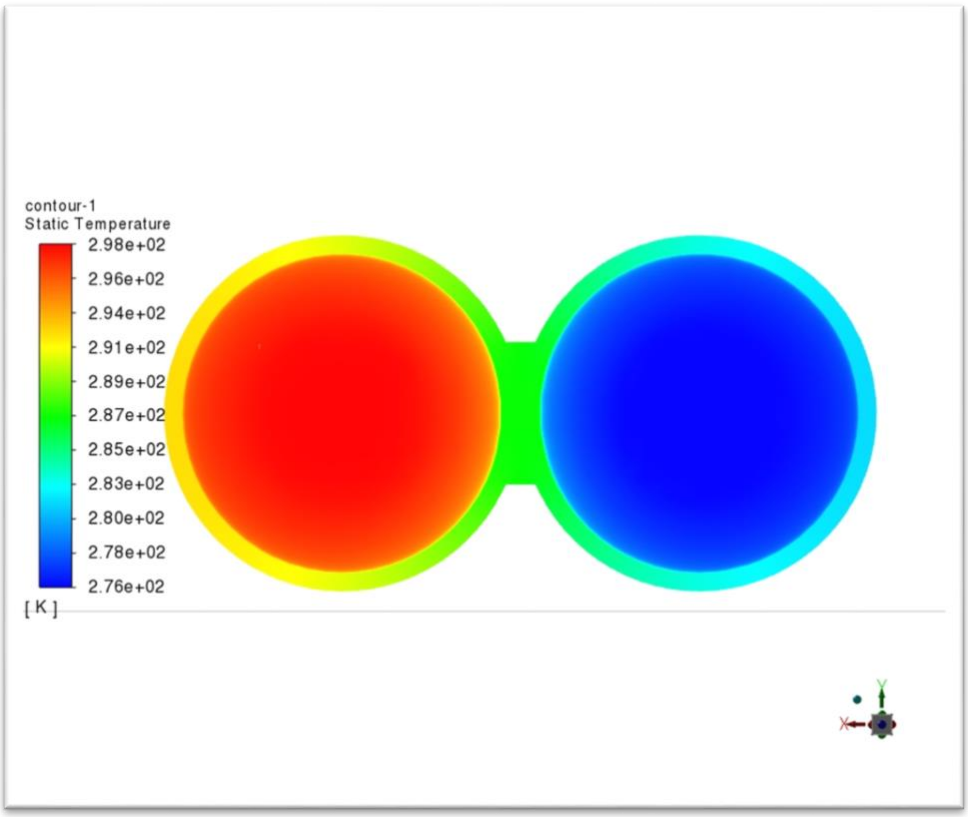

Figure 4.9: Thermal Analysis Result, Temperature profile of Refrigerant

4.3.2 Clamping pipes using an Aluminium Plate

Similar to brazing pipes together solution, placing an aluminium plate in between the pipes are evaluated for steady state thermal analysis. In this solution the aluminium plate gains heat first before transferring it on to the wall of secondary pipe consisting of cold refrigerant in it.

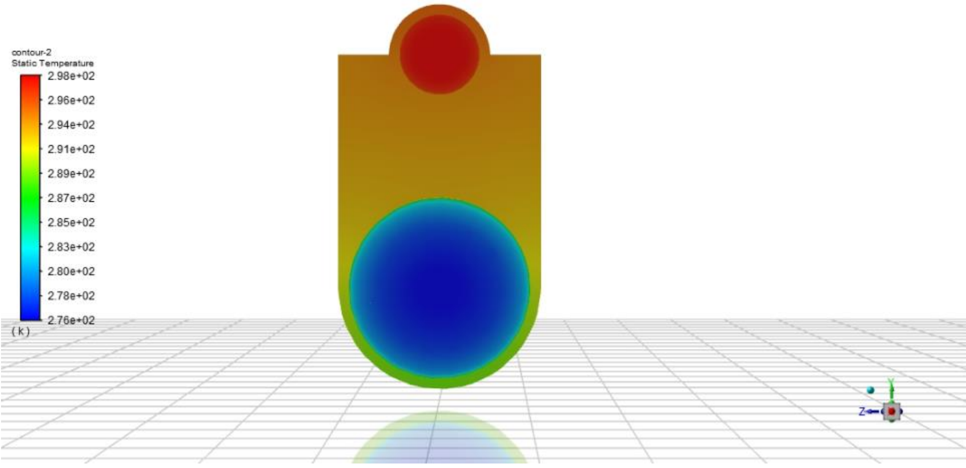

Figure 4.10: Thermal Analysis Result, Temperature profile of Refrigerant

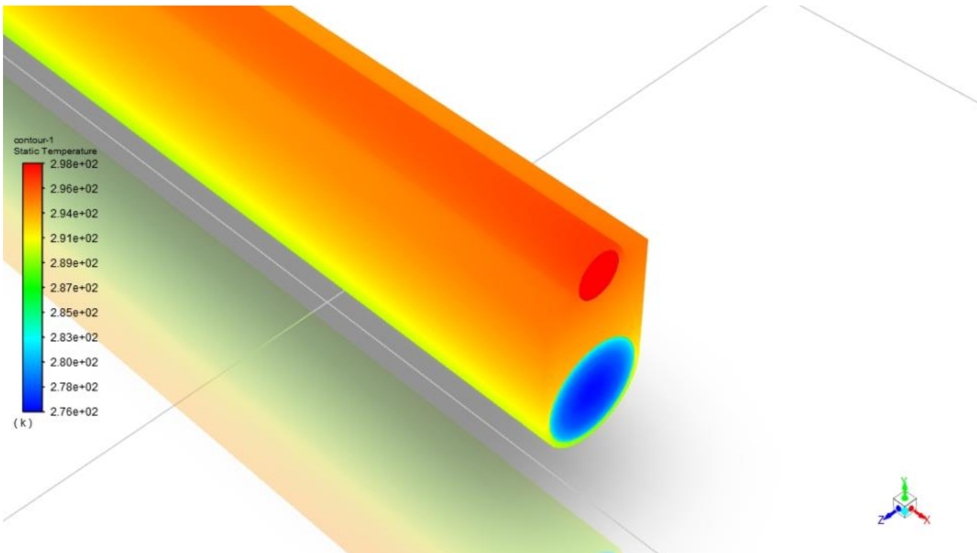

Figure 4.11: Thermal Analysis Result, Temperature porfile of wall

The difference in thermal conductivity of copper pipe and aluminium plate is not significant. From figure 4.10 and 4.11 it can be seen that for a length of contact of 300 mm, the heat transferred to the cold refrigerant (denoted blue in colour) is occurring more than it does for brazing pipes together, which is represented in green colour around the blue coloured cold vapour phase refrigerant.

A temperature difference of about 10 K is observed at the area of contact on the cold refrigerant side, the temperature gain is determined by the colour contours produced from the analysis. Due to the presence of extra surface area over a period of time better heat transfer can be realised, since this analysis is steady state and time dependent analysis could yield better results.

4.4 Results from Testing

The results from testing are used to compare the performance of heat pump prototype without and with internal heat exchanger. Parameters like COP and discharge temperature are presented although there exists data for other parameters, these are the most important indicators of performance.
The solid dots represent values from testing without internal heat exchanger and hollow dots represent value from heat pump with internal heat exchanger. In the legend, “A5W25 W” means ambient temperature for “A”, water temperature for “W” and the second “W” stands for heat pump circuit with internal heat exchanger and if it is left blank it means the value corresponds to heat pump circuit without internal heat exchanger.
In figure 4.12, the COP of heat pumps with and without internal heat exchangers are compared, it can be observed that there are certain points at compressor speed 60% and above where there is a significant increase in COP, if not the same. But at compressor speeds of 30% there lies one point which shows negative impacts of using an internal heat exchanger, this is maybe due to that is the first reading taken after modifying the prototype and the insulation material surrounding the internal heat exchanger probably did not generate and retain enough heat into the system.

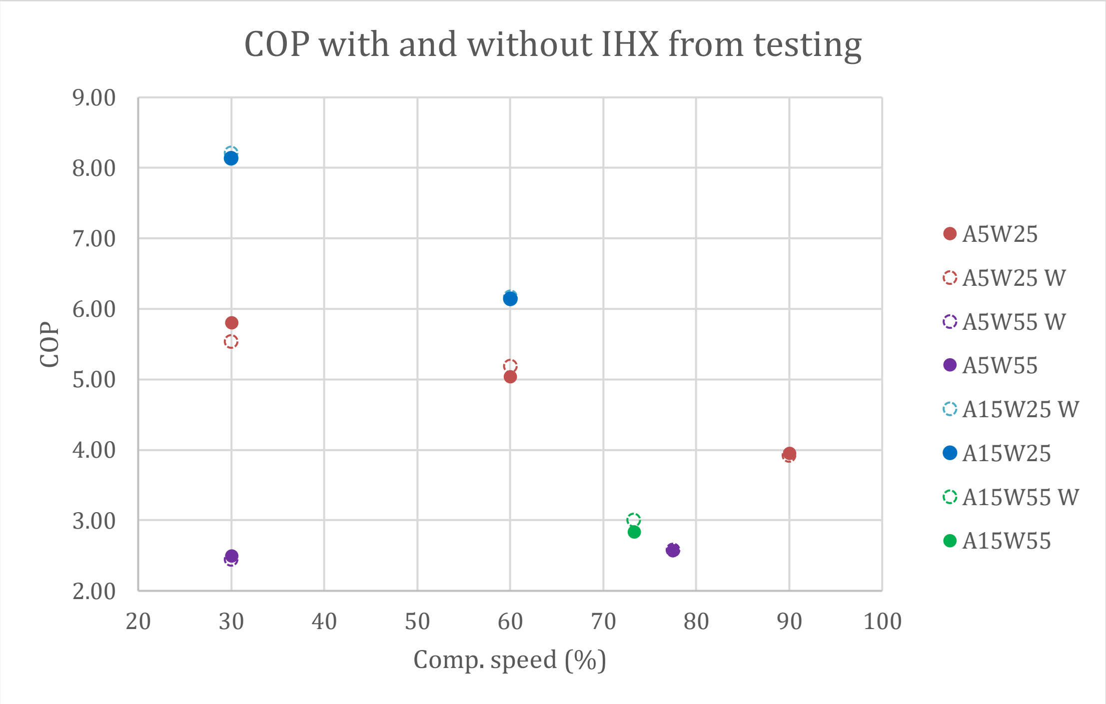

Figure 4.12: COP comparison from testing

In figure 4.13, the discharge temperature of heat pumps with and without internal heat exchangers are compared, it can be observed that there are many points in the graph where hollow dots are situated below the solid dots indicating no significant increase in temperature of discharge temperature as it was expected to increase from theoretical simulations. This can be explained due to the fact that the compressor performance is limited and optimised for the heat pump circuit without internal heat exchanger. Due to increase in volume of the refrigerant with increase in temperature and limited by the compressor's volumetric efficiency.

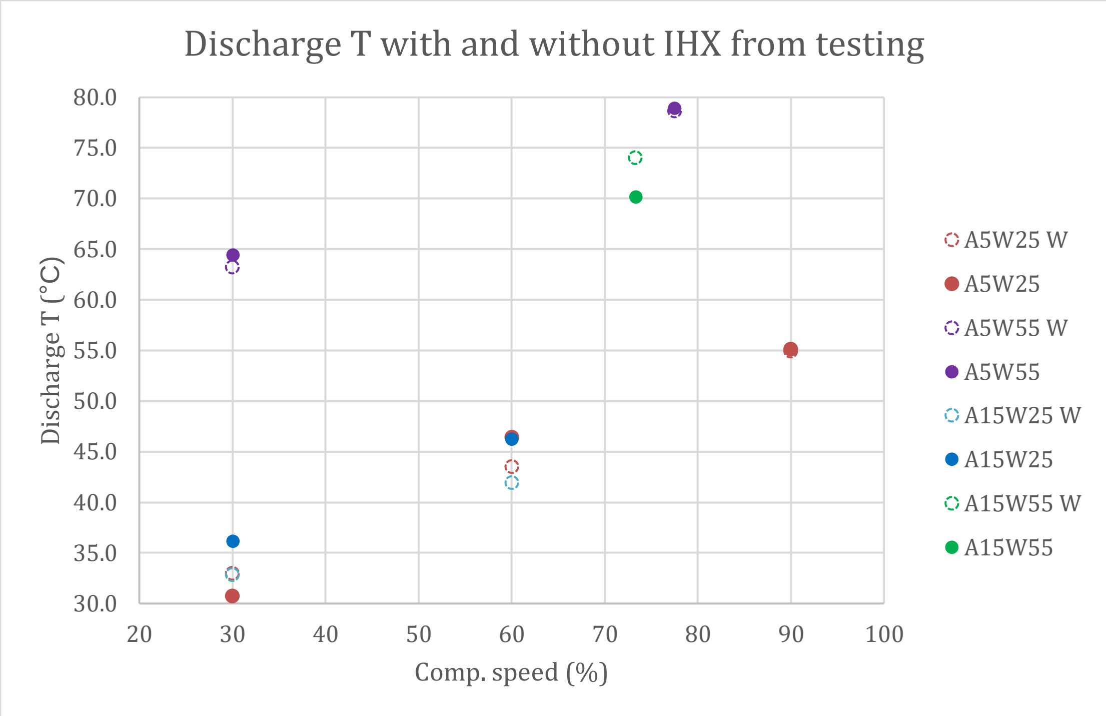

Figure 4.13: Discharge temperature comparison form testing

There are points in both the figures above where there exists a compressor speed of ~75%, this operating point was not meant to be used for the testing but it arises due to higher ambient temperature and high compressor speed, it was limited to operating at approximately 75% where as it was supposed to be 90% compressor speed according to the test matrix. This occurs due to safety limitations and reaching the operating limits of the heat pumps during testing as the refrigerant used in the prototype is highly combustible.

4.5 Overview of results

From the suggested options of transferring heat internally within a heat pump, a comparison is made among them to evaluate the length required for attaining a temperature difference of 1, 3 and 5 °C.
Below graph in figure 4.5 shows the length that is required for the pipes to be in contact to that is required for effective heat transfer. Pipes that are brazed together in parallel require the longest length to achieve a certain amount of heat transfer. Because the area of contact is the least for this method. Whereas the area of contact is higher for pipes in contact with aluminium plate and is the highest for concentric pipes. Therefore, concentric pipes require the least length to be in contact with pipes to achieve desired heat transfer.

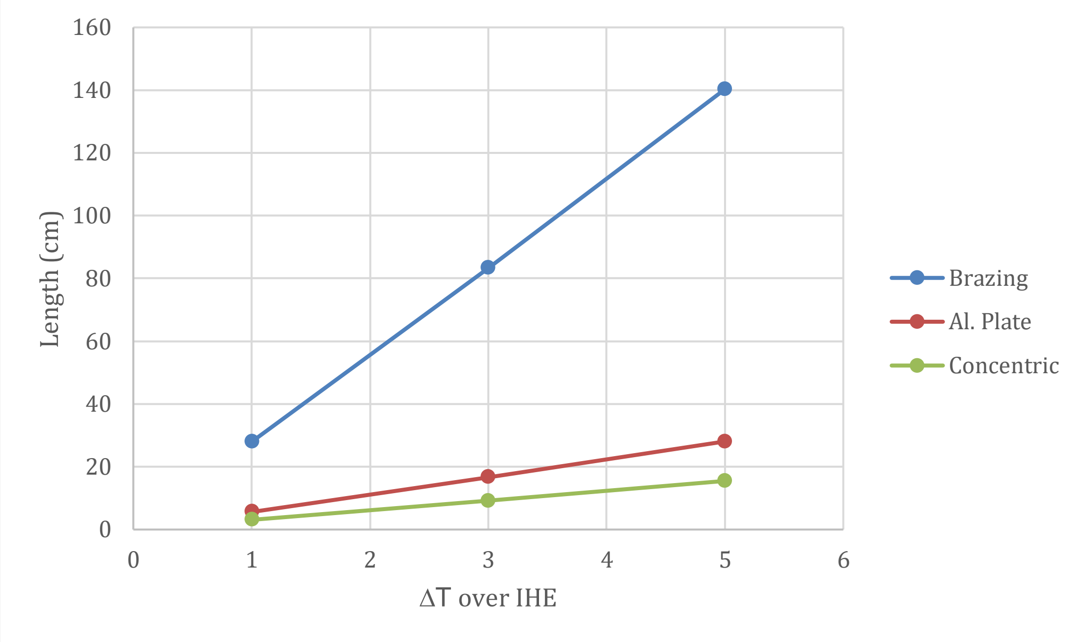

Figure 4.14: Comparison of internal heat exchanger options

Since plate heat exchangers cannot be directly compared with the heat exchangers that use pipes, a dimensional study is performed on the minimum number of plates that are required to achieve a given value of temperature gain in the internal heat exchanger. The figure 4.13 below shows that, to achieve a temperature gain of 1°C a minimum of 6 plates are required, 8 plates for temperature gain of 3°C and 10 plates for temperature for temperature gain of 5°C, over the internal heat exchanger. The number of plates increase for the increase in temperature gain values to compensate for the pressure drop on the suction line of the refrigeration circuit.

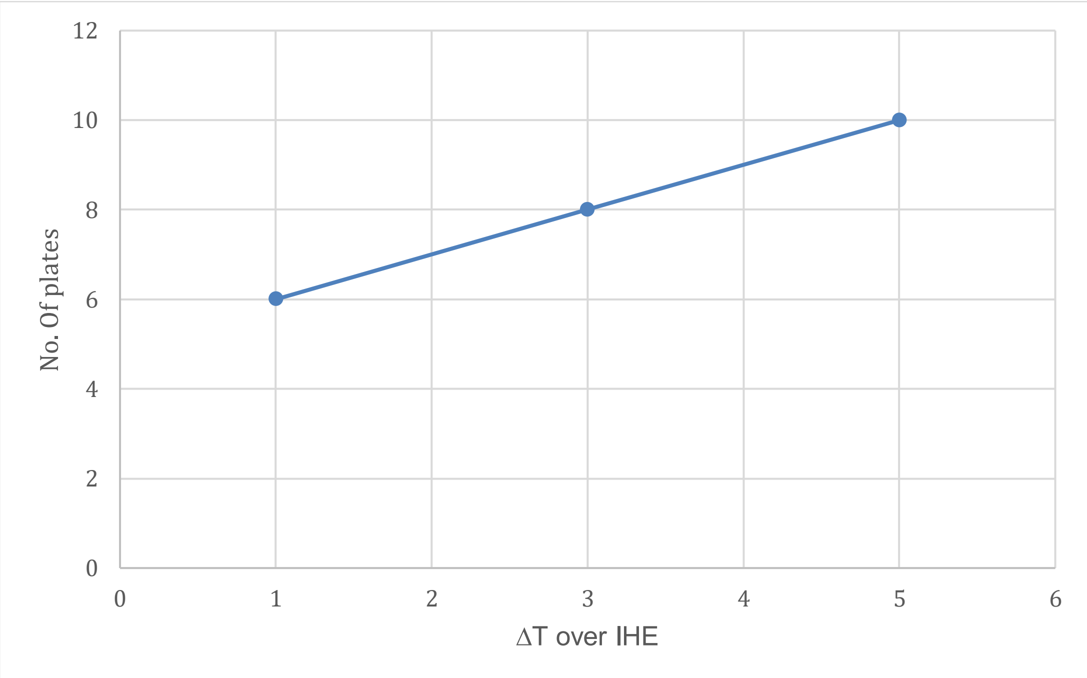

Figure 4.15: Comparison for Plate Heat Exchanger

5 Conclusion

The work presented in this thesis first explored for potential performance increasing methods for heat pumps and their prospects of improving COP is found out from the study. Apart from increasing just the COP, the effects that additional components bring within the system are also accounted for which gives a rough idea of amount of effort that goes into designing and modelling the component. From this information, internal heat exchanger (IHE) on the suction line of the heat pump was chosen to be looked into for its potential for improving the performance of heat pump.

Secondly, the internal heat exchanger was dimensioned for a plate heat exchanger to observe the benefits and it proved to increase the flow temperature, thereby increasing the heat delivered by the condenser ultimately increasing the COP of the heat pump by about 4.5%. A sensitivity analysis is performed on the internal heat exchanger by varying the amount of superheat temperature gain in the evaporator and in the IHE for which large proportion of heat transfer in the IHE proved to yield the highest performance, but it is practically difficult to achieve.

Since the benefits of plate heat exchanger type of IHE is proven to be worthy, other types of internal heat exchangers which involve making use of the pipes in the refrigeration circuit by keeping them in contact to each other is evaluated for. Among which the concentric tubes solution has the best heat transfer capacity with the least amount of length required. Whereas for the pipes that are kept in parallel contact by means of clamping and brazing had lower performance compared to concentric tubes due to lack of heat transfer area which was limited to half of it.

Tests were performed on an actual prototype my modifying the refrigeration circuit for an IHE, which involved making use of the solution clamping pipes in parallel to maintain contact in parallel for heat transfer. Although this yielded the least heat transfer in theoretical simulations, it was rather easier to develop a test model for this method. The results from testing show that up to 5% increase in COP can observed at certain operating points and there are also visible performance drops in other operating points. This proves that if other types of IHE maybe used, there might be better performance figures enhancing the operation of heat pump.

The key takeaways from implementing an IHE are that system enhancement measures like these can positively and negatively affect the system such as the refrigerant exhibiting performance shortfalls due to varied operating range for which the compressor is not dimensioned. In this case, due to higher suction line super heat in the volume of the refrigerant increases with increase in temperature of the refrigerant.

5.1 Future Work

Many different adaptations, tests, and experiments have been left for the future due to lack of time (i.e. the experiments with protype modifications and attaining stability climate chamber are usually very time consuming, requiring even days to finish). Future work concerns deeper analysis of particular mechanisms, new proposals to try different methods, or simply curiosity.

An internal heat exchanger with a helical shaped pipe surrounding the suction line could be an interesting solution because this method facilitates greater area of contact for varying pipe dimension between the two stages of refrigerants.
A time dependent thermal analysis model can be developed to speed up the process of evaluating different heat exchanger options involving pipes instead of plate heat exchangers to understand the benefits of the changes that it can bring to the system.

With stricter laws with regards to selection of refrigerants with low global warming potential, the analysis work can be adapted to new refrigerants whose results will be interesting to observe if they benefit the system or not.

Due to changes in volumetric density of the refrigerant with change in temperature which affects the compressors envelope of operational limits, in the suction line after the internal heat exchanger the diameter of the pipe can be increased to check if the compressor can operate within its envelope and the fluid can be stored before entering the compressor.

Obviously, the use of other types of individual representations and fitness functions could be investigated since they have an important influence on the results obtained at the end.

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7 Appendix

7.1 EES Code

7.1.1 Basic Heat pump Circuit

```
m_dot = 1 [kg/s]
T_Suction = T_Evaporator + DT_SH                    "Compressor Inlet Temperature"
DT_SH = 5 [C]  "delta T from evap inlet to comp inlet [K]"
T_Evaporator = -5 [C]
P_Suction=pressure(R410A,T=T_Suction,x=1)
T_Discharge = 100 [C]                               "Compressor Supplier info"
P_Discharge = P_Condensor

T_Condensor = 35 [C]
P_Condensor = pressure(R410A,T=T_Condensor ,x=0)

T_3 = T_Condensor - DT_SUB
DT_SUB = 5 [C]                                      "[K]"

"Compressor Inlet Conditions"
Power_compressor = m_dot * (h[2] - h[1])
h[1] = enthalpy(R410A,T=T_Evaporator,P=P_Suction)


"Compressor Outlet Conditions"
h[2] = enthalpy(R410A,T=T_Discharge,P=P_Discharge)

"Condensor Outlet Conditions"

h[3] = enthalpy(R410A,T=T_3,P=P_Discharge)

Q_Condensor = m_dot * (h[2] - h[3])

"Expansion Valve"                                   "Isenthaplic Expansion"
h[3] = h[4]

"Evaporator"
Q_Evaporator = m_dot * (h[4] - h[1])

"Efficiency"

COP = (Q_Condensor)/Power_compressor
```

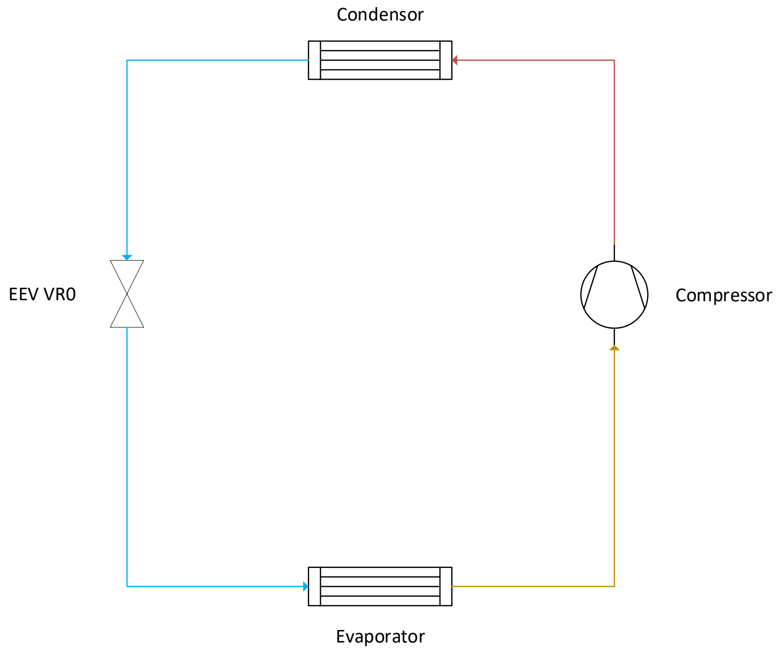

Figure 7.1: Basic Heat Pump circuit

7.1.2 Heat Pump Circuit with Internal heat exchanger

m_dot = 0.04047 [kg/s]

"h5"
T_Evaporator = -5 [C]
DT_SH = 5 [C] "delta T from evap inlet to comp inlet [K]"

"h1"
T_Evaporator_Out = T_Evaporator + DT_SH
T_Suction = (T_Evaporator_Out + DT_IHX) "Compressor Inlet Temperature"
P_Suction=**pressure**(***R410A***,***T***=T_Suction,***x***=1)

"h2"
T_Discharge = 81.66 [C] "Compressor Supplier info"
P_Discharge = P_Condensor

"h3"
T_Condensor = 35 [C]
P_Condensor = **pressure**(***R410A***,***T***=T_Condensor ,***x***=0)
T_3 = T_Condensor - DT_SUB "Condensor Outlet Temperature"
DT_SUB = 5 [C] "[K]"

"h4"
T_IHX = (T_3 - DT_IHX) "Outlet line from IHX from Condensor"
DT_IHX = 5 [C] "[K]" "IHX gain in suction"

"Compressor Inlet Conditions"
Power_compressor = m_dot * (h[2] - h[1])
h[1] = **enthalpy**(***R410A***,***T***=T_Evaporator,***P***=P_Suction)

"Compressor Outlet Conditions"
h[2] = **enthalpy**(***R410A***,***T***=T_Discharge,***P***=P_Discharge)

"Condensor Outlet Conditions"

h[3] = **enthalpy**(***R410A***,***T***=T_3,***P***=P_Discharge)

Q_Condensor = m_dot * (h[2] - h[3])

"IHX"
h[4] = **enthalpy**(***R410A***,***T***=T_IHX, P= P_EVAP_In)
P_EVAP_In = P_Suction
Q_IHX = m_dot * (h[3] - h[4])

"Expansion Valve, h[4] = h[4] ; Isenthaplic Expansion"

"Evaporator"
Q_Evaporator = m_dot * (h[5] - h[4])
h[5] =**enthalpy**(***R410A***,***T***=T_Evaporator + DT_SH , ***P***=P_Suction)
"P[5] = pressure(R410A,T=(T_Evaporator + DT_SH) ,x=1)"

"Efficiency"

COP = (Q_Condensor) / Power_compressor

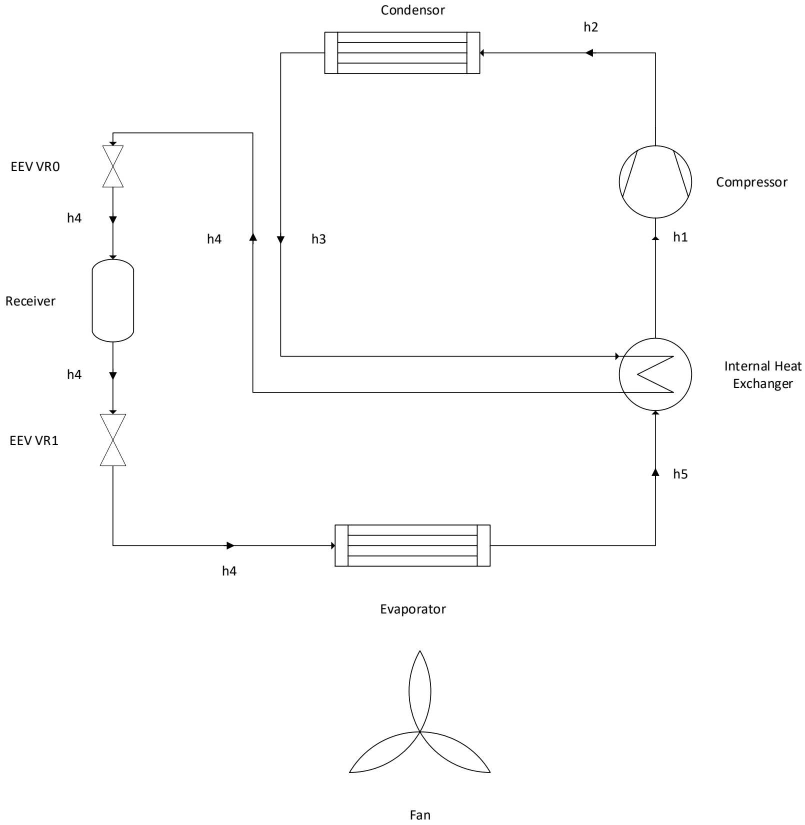

Figure 7.2: Heat Pump Circuit with Internal Heat Exchanger