Experimental Analysis of Variable Capacity Heat Pump System Equipped with Vapour Injection and Permanent Magnet Motor

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Experimental Analysis of Variable Capacity Heat Pump System Equipped with Vapour Injection and Permanent Magnet Motor

Masters of Science Thesis

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ABSTRACT

This study analyzes the performance of variable capacity heat pump scroll compressor which is equipped with vapour injection and permanent magnet motor. Refrigerant used in the system is R410A. The study is divided in two phases. In first phase, tests are carried out for heat pump without vapour injection. Heat pump's performance including COPs, heating/cooling capacities, inverter losses, heat transfer behaviour in condenser/evaporator are analyzed.

Inverter losses increase but the ratio of inverter losses to the total compressor power decreases with increase in compressor speed. Electromechanical losses of compressor are much higher than the inverter losses and so make most part of the total compressor losses (summation of inverter and electromechanical losses).

In second phase benefits of vapour injection are analyzed. For vapour injection, heat pump’s performance is evaluated for two different refrigerant charges: 1.15kg and 1.28kg. It is noted that heat pump performs better for refrigerant charge 1.15kg even at lower compressor speeds as compared to refrigerant charge 1.28kg. For refrigerant charge 1.15kg, heat pump COP cool with vapour injection increases by an average of 10.66%, while COP heat increases by an average of 9.4%, at each compressor speed except for 30Hz, as compared to conventional heat pump cycle with no vapour injection. Similarly refrigerant temperature at outlet of compressor also reduces with vapour injection which leads to the better performance of heat pump.
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<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>A</td>
<td>Heat transfer area</td>
<td>m²</td>
</tr>
<tr>
<td>B</td>
<td>Constant</td>
<td>-</td>
</tr>
<tr>
<td>C</td>
<td>Constant</td>
<td>-</td>
</tr>
<tr>
<td>dₑ</td>
<td>Equivalent diameter</td>
<td>m²</td>
</tr>
<tr>
<td>E_comp</td>
<td>Actual compressor power</td>
<td>kW</td>
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<tr>
<td>f</td>
<td>Frequency (compressor speed)</td>
<td>Hz</td>
</tr>
<tr>
<td>g</td>
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<tr>
<td>h</td>
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<tr>
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<td>q</td>
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<tr>
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<td>K or °C</td>
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<td>Theoretical compressor power</td>
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<td>$\dot{w}_{\text{isen}}$</td>
<td>Isentropic compression power</td>
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<td>$\alpha$</td>
<td>Heat transfer coefficient</td>
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<td>$\mu$</td>
<td>Dynamic viscosity</td>
<td>Pa.s</td>
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<tr>
<td>$\lambda$</td>
<td>Thermal conductivity</td>
<td>W/m.K</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
<td>kg/m³</td>
</tr>
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</table>

**Index**

AC

AC

BPHE

BPHE

BPM

BPM

br

br

comp

Comp

cond

Cond

COP

COP

desup

desup

EES

EES

EEV

EEV

EM

EM

evap

evap

exp

exp

f, l

f, l

g

g

GSHP

GSHP

HP

HP

HX

HX
i
in
isen
Ln
MS
NOP
out
PA
PH
PHE
Pt
R or ref
RHP
RLP
sat
SC, sub
SH
SPF
SPM
tot
VI
wt

Greek symbols

\( \eta \)  
Efficiency

\( \xi \)  
Variable part of electromechanical losses

\( \nu \)  
Built-in volume ratio

\( \pi \)  
Built-in pressure ratio
Chapter 1

Introduction

Ground source heat pumps, GSHPs (sometimes referred to as geothermal, GeoExchange, or water-source heat pumps), exchange heat to and/or from ground source to provide heating or cooling. GSHPs make use of the relatively constant temperature of ground throughout the year as compared to ambient air and a circulating fluid (refrigerant, water, brine etc.) to exchange heat. Temperature gradients in ground at about 3-ft depth or lower are less variable than ambient air (Oak Ridge National, 2011). Like a cave, the ground temperature is warmer than the air above it in winters and cooler than air in summers. The earth acts as a heat source in winters and heat sink in summers. As with any other heat pumps, GSHPs are able to heat, cool and supply hot water when needed. Compared to air-source heat pumps, they do not depend on the outside air temperature.

There are four basic types of ground loop systems. Three of these - horizontal, vertical and pond/lake - are closed loop systems and the fourth type is open loop system. Close loop systems circulate heat transfer fluid (water or antifreeze, e.g. brine etc) through a closed loop that is buried in ground or submerged in water (in case of pond/lake). The circulating fluid extracts heat from ground water or rock and exchanges this heat with refrigerant in a heat exchanger (HX). In another variant for this configuration, refrigerant is circulated through the pipes buried underground to exchange heat with the earth’s rock or soil, thus eliminating the need for water pumps and water-to-refrigerant heat exchangers. However, this increases the refrigerant charge in the loop and there is a risk of refrigerant leakage into the ground which creates a barrier in its frequent use.

In horizontal closed loop configuration, pipes run horizontally in the ground. The trenches are dug a few feet deep, at least 4 feet into the ground and U-shaped pipes are laid horizontally. This configuration is particularly suitable in places where adequate land is available. Vertical loop field’s pipes run vertically into the ground in the well dug 100-400 feet deep. The vertical pipes are connected at the bottom with a U-bend to form a loop. They are connected with the horizontal pipes in the manifold, placed in trenches, and connected to heat pump in building. A pond loop system consists of coils of pipe attached to frame and located at the bottom of a pond or water at least 8 feet deep under the surface to prevent freezing (United States Department of Energy). Open loop systems require an underground well for heat exchange. Warm water from a well is pumped directly through the loop, it exchanges heat with the refrigerant in a heat exchanger and once it has circulated through the system it is again recharged into the ground in another well.

Brine to water ground source heat pumps are one of the fastest growing applications being used for indoor comfort in USA and European countries. The world has seen a sharp increase in their capacity and installations over the last years. As of 2005, there are over a million units installed worldwide providing 15,000MW thermal capacity (Rybach, 2005) with more than 10% annual global increase in about 30 countries over the last 10 years (Lund & Sanner, 2004). In Sweden alone the most common type of heat pump is ground source heat pump, which extracts heat from a borehole, the ground or seawater. Systems using the vertical borehole are the most common type (Karlsson, Axel, & Fahlén, 2003). According to (Montagnud & Corberán, 2012) the use of GSHPs accounts for 40% savings in annual electricity consumption compared to air heat pumps. Given their huge potential and use all
over the world, it is important that consideration be given to their efficiency improvement to save energy and reduce green house gases.

1.1 Background

To increase efficiency of a given heating system, efficient components and control techniques should be employed for the system to operate at optimum point. Components which have shown the most potential in this regard are capacity-controlled compressors, pumps, fans and electronic expansion valves (Karlsson F., 2003). It has been found that capacity control of GSHPs to match the demand and supply load is one of the promising methods to reduce energy consumption inside buildings and improve their overall efficiency. These systems improve comfort and efficiency in areas where heating and cooling loads are changing by varying the capacity of compressor to match the demand. In this way the compressor adjusts its capacity to maintain the constant room temperature. Compressor runs on reduced capacity in time of lower energy demand but ramps up its capacity to match the load in time of higher energy demand.

Heat pumps for space and water heating are subjected to continuously varying heating loads and they must be able to control their capacity to meet the load. Different methods to control the capacity of GSHPs are: compressor ON/OFF control, hot gas bypass, evaporator temperature control, clearance volume control, cylinder unloading and variable-speed controlled compressor. The most common type of capacity control for commercial GSHP is the intermittent control or on/off control for the compressor, where the compressor is switched on and off (called cycling) by a thermostat (Montagnud & Corberán, 2012). A frequency converter, often called inverter, is used to accomplish the purpose of variable speed of the compressor. The inverter controls the frequency supplied to the compressor motor in order to change the speed of compressor. Changing frequency also changes the mass flow rate of refrigerant passing through compressor and its thermal capacity (Karlsson F., 2003).

On/off control operates effectively at design load conditions, but under part load conditions inefficiencies can increase due to compressor cycling. These inefficiencies are attributed to the refrigerant migration and cooling down of the compressor cycle during off period (Finn & Killan, 2010). The efficiency of heat pumps controlled by variable capacity can be improved due to better performance at part load, reduced need for supplementary heating and defrosting and fewer on/off cycles. While running at part loads, compressor with intermittent control is switched on and off to match the load, which causes the heat pump to operate at high condensation and low evaporation pressures than it would do if it were capacity controlled. Capacity control also allows heat pump to decrease the on/off cycles which cause wear on compressor. This reduced cycling frequency leads to reduced losses and longer life expectancy (Karlsson F., 2003).

(Zhao, Zhao, Zhang, & Ding, 2003) made a comparison among variable speed capacity control and three other controls for GSHPs and concluded that changing the compressor speed is the preferred method. In another study conducted by (Madani, Claesson, & Lundqvist, 2011) on the comparison between on/off controlled and variable capacity heat pump system, they found that variable speed capacity control yields better performance compared to the constant speed on/off control when the ambient temperature is below the balance point and the auxiliary heater operates. The annual modelling of both on/off and variable speed systems done by (Madani, Claesson, & Lundqvist, 2011)
showed that when the on/off controlled heat pump system is designed to cover 90% of annual energy demand with auxiliary heater taking care of the rest, seasonal performance factor (SPF) of the system may be improved 10% by switching to a variable speed heat pump system.

However, (Karlsson & Fahlen, 2007) concluded from their research that the variable capacity heat pump systems did not improve the annual efficiency compared to on/off controlled heat pump despite improved performance at part load. (Karlsson F., 2007) suggested that this is mainly due to the inefficiencies of inverter, electric motor of compressor and the need for control of pumps used in the heating and ground collector systems. In another experimental study conducted by (Cuevas & Lebrun, 2009), they found that inverter efficiency varies between 95% and 98% for variable speed scroll compressor when its electric power varies between 1.5kW and 6.5kW.

However, in inverter-driven compressor, additional losses are incurred in the system due to the losses in inverter itself and in the motor caused by the non-sinusoidal waveform which influence the overall coefficient of performance (COP) of a heat pump. Experimental investigations carried out by (Qureshi & Tassou, 1994) indicate that efficiency of variable speed refrigeration systems at part loads is severely affected by the poor performance of induction motors.

For compressors to run at different speeds (from low to high) for wide application range, compressor motors should operate at high efficiency. When compressors operate at part loads their efficiency decreases due to the decrease of motor efficiency at low rotating speed. This is due to increase in motor copper loss (Kazuhiro Matsukawa, 2008). With the conventional induction motors used for inverter controlled compressors, there is a limitation in reduction of copper loss (called the exciting loss) created by the current generated through the aluminium conductor. In order to improve the efficiency, exciting losses should be minimized. On the other hand, since the ferrite permanent magnets mounted on the surface permanent magnet (SPM) synchronous motor generates magnetic flux which is required for torque, the motor gives high efficiency without the exciting loss. However, since there is a limitation in magnetic flux density, large size motors cannot output high efficiency performance. In addition, the efficiency of the SPM motor reaches the highest point at the maximum speed and is unsatisfactory in the range of low to medium speed (Obitani, Ozawa, Taniwa, & Kajiwara, 2000).

For heating systems to operate at high efficiency dedicated compressors are developed to suit the needs of the climatic conditions, to achieve the required COPs and operating ranges. The refrigeration cycles equipped with vapour injection (VI) port in scroll compressors use an economizer and allow heat pumps to operate at high condensing and low evaporating temperatures (operating map). Vapour injected scroll compressors make use of higher pressure ratios which deliver higher benefits. The main advantages are a higher COP, increased operating map and higher heating capacity at low evaporation than a conventional cycle (Liegois & Winandy, 2008).

In a study (Liegois & Winandy, 2008) make a comparative analysis between scroll compressors for air-source heat pumps (ASHPs), GSHPs with vapour injection and GSHPs without vapour injection, and find that vapour injection gives a noticeable increase of COP with respect to heat pump without vapour injection. (Wang, Hwang, & Radermacher, 2008) conducted a comparative study between the vapour-injected R410A heat pump system and the conventional system. Their study concluded that there is an increase in cooling capacity of around 14% and improvement in COP of 4% at ambient temperature 46.1°C. The heating capacity increased by 30% with 20% improvement in COP
gain at -17.8°C. They also concluded that vapour injection system performs better in high ambient temperatures for cooling mode and low ambient temperatures for the heating mode.

1.2 Working principle of the heat pump system used in experimentation

In this study a vapour-injected scroll compressor equipped with permanent magnet motor is being used. The general working principle of heat pump with this arrangement is the same as any other heat pump except that an additional economizer is added into the system.

In this section a general working principle of a heat pump system is described. Some explanation about the vapour injection configuration into the heat pump and its possible advantages is also given.

How heat pump works:

Heat pump is a device that transfers energy from a heat source at low temperature to the heat sink that is relatively at a higher temperature than the heat source. It works on the principle of refrigeration cycle (also called vapour compression cycle). It can be used for both heating and cooling depending on the application.

![Figure 1-1: Schematic of a heat pump system](image)

For adiabatic system the heat balance can be written as

\[ Q_1 = Q_2 + W \]  

The cooling COP is defined as

\[ \text{COP}_{\text{cool}} = \frac{q_2}{W} \]

Where \( q_2 \) is the evaporating capacity, also called the refrigerating capacity.
Whereas heating COP is given by the following correlation

\[
\text{COP}_{\text{heat}} = \frac{\dot{Q}_\text{v}}{W}
\]

\(\dot{Q}_\text{v}\) is the condensing capacity.

In case of vapour injection a portion of the condensed liquid is expanded through an expansion valve into the economizer (brazed plate counter flow heat exchanger) which acts as a subcooler for the condensed liquid. The expanded refrigerant is superheated in this section and is injected in the intermediate vapour injection port in scroll compressor. The additional subcooling increases the evaporating capacity by reducing the enthalpy of refrigerant entering the evaporator. Heating capacity also increases due to the additional mass flow through the condenser. The vapour injected in the intermediate port is compressed from higher inter stage pressure than the suction pressure. The COPs are also higher with vapour injection scroll compressors than conventional scroll compressors delivering the same capacity because the added capacity is achieved with less power.

![Diagram of vapour injection with upstream liquid extraction](image)

*Figure 1-2: Vapour injection with upstream liquid extraction*

Liquid is usually extracted upstream for the economizer expansion device as shown in Fig.1-3. Downstream extraction refers to taking the liquid for heat exchanger expansion device from HX liquid exit as shown in Fig.1-4 below. The overall heat gain or loss for downstream extraction is negligible compared to upstream extraction. The injected mass flow, i, passes through HX twice and causes additional pressure drop on liquid cooling side, which may result in need for a larger HX. For these reasons upstream liquid extraction is usually preferred.
Figure 1-3: Vapour injection with downstream liquid extraction
Chapter 2

Objectives

The objective of this study is to experimentally analyze the performance of variable capacity heat pump scroll compressor with and without vapour injection. Experiments are carried out in the study with and without vapour injection to analyze the impact of variable capacity compressor on:

1. The overall performance of heat pump unit (heating capacity, cooling capacity, COP cool, COP heat, compressor power, isentropic efficiency, Carnot efficiency)
2. The loss behaviour of variable speed compressor
3. The loss behaviour of inverter
4. Heat transfer behaviour in condenser
5. Heat transfer behaviour in evaporator

As explained earlier, compressors with permanent magnet motors can reduce the electromechanical\(^1\) losses incurred during the operation of heat pump, this study will also aim to determine the electromechanical losses incurred in scroll compressor used in experimentation. A compressor model is built using EES (Engineering Equations Solver), which gives constant and variable part of electromechanical losses of compressor. Inverter losses are estimated by measuring the power before and after the inverter.

This study also aims to determine the impact of the vapour injection on the overall performance of heat pump unit. For this purpose an economizer, expansion valve and a sight glass are added in the system. Experiments are run for vapour injection with different refrigerant charge and the comparative analysis is made.

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\(^1\) In this study electromechanical losses refer to the combined losses incurred in motor and due to non-isentropic compression work of compressor. Since compressor is a scroll-compressor, the losses cannot be segregated into motor and compression losses.
Chapter 3

Experimental setup

An experimental test rig used to analyze the variable capacity compressor, as shown in Fig.3-1. To carry out experiments without vapour injection, test rig is equipped with heat pump unit with brazed plate evaporator and condenser, inverter-driven variable speed compressor, water tank, brine pump, water pump, two external plate heat exchangers, valves, power meter and data acquisition system. Electronic expansion valve is used to maintain the degree of superheat at 5°C before the compressor inlet. To carry out experiments with vapour injection, additional heat exchanger, called economizer and an electronic expansion valve are added in the test rig. Refrigerant R410A is used in the heat pump unit. Power is measured before and after the inverter using the power meter.

3.1 Test facility

The experimental facility consists of four separate loops: refrigerant flow loop, brine loop and two water loops.

3.1.1 Refrigerant flow loop

The refrigerant loop contains a variable speed compressor that circulates refrigerant through condenser, economizer (in case of vapour injection), expansion valve and evaporator at variable flow rate. In case of heat pump system without VI, refrigerant is pumped into the condenser by the inverter-driven variable capacity compressor where it exchanges its heat with water in the secondary circuit. Refrigerant is then expanded through an electronic expansion valve into the evaporator, it is superheated here and pumped again through compressor.

![Schematic showing the heat pump system without vapour injection and other components of test rig](image_url)
For vapour injection system, some of the condensed liquid is extracted from the main liquid line and expanded into an economizer where it exchanges heat with the condensed liquid flowing in the pure liquid line. Economizer acts a subcooler for the already condensed liquid. The extracted vapour is superheated in the economizer and injected into the compressor at intermediate pressure through a vapour injection port.

3.1.2 Brine and water loops

As shown in Fig. 3-2, secondary circuits to the evaporator and condenser side are: brine loop and water loop 1 respectively. A fixed speed pump in water loop 1 circulates water through the secondary circuit on condenser side. Water takes away heat from the refrigerant in condenser through water loop 1 and exchanges some of the heat with brine in an external plate heat exchanger (plate heat exchanger, brine/water) to maintain brine’s temperature entering the evaporator. Heating load to the brine is provided by water loop 1. Brine is circulated in a secondary loop through the evaporator where it gives off its heat to vaporize the refrigerant in evaporator. Temperature at the inlet of brine is controlled by a three-way manual valve, which is positioned at different incremental points from time-to-time. Water temperature at inlet or outlet of condenser is controlled via a tap water valve in water loop 2. Tap water circulates through water loop 2 to maintain temperature of the water in water loop 1 at inlet or outlet of condenser.
3.2 Heat pump system’s components

The variable capacity compressor and frequency inverter, manufactured by Emerson Copeland, are used in the test rig. The compressor is heat pump dedicated R410A variable speed and vapour injection scroll type. The frequency inverter is a single phase input and three phase output device that sends three phases current to the compressor motor. The inverter is controlled by a user interface on a remote computer that is used to run compressor on variable speed. The communication between computer and inverter is via Wireless/Modbus router and the receiver antenna connected with the inverter (see appendix C).

![Experimental test rig with data acquisition system](image)

Figure 3-3: Experimental test rig with data acquisition system

Following table summarizes the specifications of some of the main components in the test rig.

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>Emerson Copeland (ZHW16K1P), AC 3PH</td>
</tr>
<tr>
<td>Evaporator</td>
<td>Prototype counter-current BPHE (526*119 – NOP = 30, assumed SWEP B25)</td>
</tr>
<tr>
<td>Condenser</td>
<td>Prototype counter-current BPHE (526*119 – NOP = 40, assumed SWEP V80)</td>
</tr>
<tr>
<td>Expansion valves</td>
<td>CAREL electronic expansion valve (EVD evolution model, E2V24BSFØØ and E2V24BRB00)</td>
</tr>
<tr>
<td>Inverter</td>
<td>Emerson (model, EV1081A-C1), 1PH Input, 3PH Output</td>
</tr>
<tr>
<td>T sensors</td>
<td>Pt1000⁰ → 7 sensors for HP without VI, 9 sensors for HP with VI</td>
</tr>
<tr>
<td>High P transducer</td>
<td>Clima check, Range 0-50 Bar(g), Vout = 1-5V</td>
</tr>
<tr>
<td>Low &amp; intermediate P transducers</td>
<td>Clima check, Range 0-35 Bar(g), Vout = 1-5V</td>
</tr>
</tbody>
</table>

² Due to electromagnetic interference between inverter & Pt1000 sensors, there should be a considerable distance between Pt1000 T sensors’ cables and inverter.
For data acquisition, ClimaCheck system, called Performance analyzer 8:7 (PA 8:7), is used which comes with its software. If pressure and temperature sensors are to be used other than the ones specified in table 1, it is important to configure them in the data acquisition system software ClimaCheck Performance Analyzer 8:7. For configuring P sensors following parameters are set:

<table>
<thead>
<tr>
<th>Range</th>
<th>Output</th>
<th>Zero</th>
<th>Full scale</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-35 Bar (g)</td>
<td>1-5 V</td>
<td>-775</td>
<td>3600</td>
</tr>
<tr>
<td>0-50 Bar(g)</td>
<td>1-5 V</td>
<td>-1150</td>
<td>5100</td>
</tr>
</tbody>
</table>

Data logger is connected directly to the computer using a serial cable. The program in Climachek software package, called ClimaCheck Standard, is used to analyze the data. If connections of the sensors are done according to table 3, all the sensors will automatically be connected to the correct variables in ClimaCheck.

<table>
<thead>
<tr>
<th>Description</th>
<th>Variable</th>
<th>Sensor transducer</th>
<th>Channel no.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant temperature compressor out</td>
<td>TT_RComp_out = °C</td>
<td>Pt1000</td>
<td>TT1</td>
</tr>
<tr>
<td>Refrigerant temperature compressor in</td>
<td>TT_RComp_in = °C</td>
<td>Pt1000</td>
<td>TT2</td>
</tr>
<tr>
<td>Refrigerant temperature Expansion valve in</td>
<td>TT_RExp_in =°C</td>
<td>Pt1000</td>
<td>TT3</td>
</tr>
<tr>
<td>Evaporator Secondary Temperature in</td>
<td>TT_SecC_in = °C</td>
<td>Pt1000</td>
<td>TT4</td>
</tr>
<tr>
<td>Evaporator Secondary Temperature out</td>
<td>TT_SecC_out = °C</td>
<td>Pt1000</td>
<td>TT5</td>
</tr>
<tr>
<td>Condenser secondary temperature in</td>
<td>TT_SecW_in = °C</td>
<td>Pt1000</td>
<td>TT6</td>
</tr>
<tr>
<td>Condenser secondary temperature out</td>
<td>TT_SecW_out = °C</td>
<td>Pt1000</td>
<td>TT7</td>
</tr>
<tr>
<td>Free temperature</td>
<td>TT_X8</td>
<td>Pt1000</td>
<td>TT8</td>
</tr>
<tr>
<td>High pressure refrigerant</td>
<td>PT_RHP = kPa(a)</td>
<td>0-50 Bar(g)</td>
<td>AI9</td>
</tr>
<tr>
<td>Low pressure refrigerant</td>
<td>PT_RLP = kPa(a)</td>
<td>0-35 Bar(g)</td>
<td>AI10</td>
</tr>
</tbody>
</table>

It is important to change the refrigerant in this program to R410Amix before starting the scan. The scan interval is set to 1s and logging is enabled. Scanning is allowed to continue for duration of 10 minutes and data is collected. This data is then exported to the excel file and analyzed in MS Excel. However, if data is collected by selecting the wrong refrigerant, it is possible to re-calculate the data. In such case save this data in excel format, change the data source to Excel inside ClimaCheck Standard program and import the required excel file. Go to the data source tab, connect the required variables and re-calculate the file.
Description of the components which are added in the system for vapour injection is given below.

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brazed plate heat exchanger(^3)</td>
<td>SWEP – B8T x 10/M-pressure</td>
</tr>
<tr>
<td>Electronic expansion valve</td>
<td>Carel’s E2V24BSFOO</td>
</tr>
<tr>
<td>Expansion valve drive(^4)</td>
<td>Carel’s EVD Twin Evolution</td>
</tr>
</tbody>
</table>

For data acquisition, ClimaCheck’s new template and new data source for the economizer system is used. It is important that the variables are connected according to the following table for the correct measurements in the data source of ClimaCheck software. For more understanding the software’s manual can be used.

<table>
<thead>
<tr>
<th>Description</th>
<th>Variable</th>
<th>Sensor transducer</th>
<th>Channel no.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant temperature compressor out</td>
<td>TT_RComp_out = °C</td>
<td>Pt1000</td>
<td>TT1</td>
</tr>
<tr>
<td>Refrigerant temperature compressor in</td>
<td>TT_RComp_in = °C</td>
<td>Pt1000</td>
<td>TT2</td>
</tr>
<tr>
<td>Refrigerant temperature Expansion valve in</td>
<td>TT_RExp_in = °C</td>
<td>Pt1000</td>
<td>TT3</td>
</tr>
<tr>
<td>Evaporator Secondary Temperature in</td>
<td>TT_SecC_in = °C</td>
<td>Pt1000</td>
<td>TT4</td>
</tr>
<tr>
<td>Evaporator Secondary Temperature out</td>
<td>TT_SecC_out = °C</td>
<td>Pt1000</td>
<td>TT5</td>
</tr>
<tr>
<td>Condenser secondary temperature in</td>
<td>TT_SecW_in = °C</td>
<td>Pt1000</td>
<td>TT6</td>
</tr>
<tr>
<td>Condenser secondary temperature out</td>
<td>TT_SecW_out = °C</td>
<td>Pt1000</td>
<td>TT7</td>
</tr>
<tr>
<td>Condenser refrigerant temperature out</td>
<td>TT_Rcond_out</td>
<td>Pt1000</td>
<td>TT8</td>
</tr>
<tr>
<td>High pressure refrigerant</td>
<td>PT_RHP = kPa(a)</td>
<td>0-50 Bar(g)</td>
<td>AI9</td>
</tr>
<tr>
<td>Low pressure refrigerant</td>
<td>PT_RLP = kPa(a)</td>
<td>0-35 Bar(g)</td>
<td>AI10</td>
</tr>
<tr>
<td>Intermediate pressure refrigerant (P in economizer)</td>
<td>PT_RMP = kPa(a)</td>
<td>0-35 Bar (g)</td>
<td>AI11</td>
</tr>
<tr>
<td>Intermediate temperature refrigerant (T at the inlet of economizer port after SH)</td>
<td>TT_Rcomp_in_MP</td>
<td>Pt1000</td>
<td>AI13</td>
</tr>
</tbody>
</table>

\(^3\) Used as an economizer.

\(^4\) Expansion valve drive EVD Twin Evolution can drive two expansion valves and it replaces the old EVD Evolution drive in vapour injection experimentations.
Chapter 4
Methodology

This study is carried out in two phases:

I. In the first phase experiments are run without vapour injection
II. In the second phase vapour injection components are added and experiments are run again for different refrigerant charges.

Loss behaviour of variable speed compressor, frequency inverter and total isentropic efficiency are analyzed only in the first phase of study. While the overall heat pump performance, heat transfer behaviour in condenser and evaporator is analyzed in both the first and second phase.

1. To analyze the loss behaviour of variable speed compressor, frequency inverter, total isentropic efficiency, following parameters are kept constant:

   Brine temperature to the inlet of evaporator (called the source temperature) = \( T_{\text{br,in}} \) and water temperature to the outlet of condenser (called the sink temperature) = \( T_{\text{wt,out}} \)

   The following set points are used:
   
   a. 2C/40C as the source/load side temperature respectively
   b. 2C/45C as the source/load side temperature respectively
   c. 5C/45C as the source/load side temperature respectively

   Where 2C, 5C are the brine inlet temperatures to the evaporator and 40C, 45C are the water outlet temperatures of the condenser.

2. To analyze the overall heat pump performance (as described in Chapter 2), heat transfer behaviour in condenser and evaporator, the following parameters are kept constant:

   Brine temperature to the inlet of evaporator (called the source temperature) = \( T_{\text{br,in}} \), water temperature to the inlet of condenser (called the sink temperature) = \( T_{\text{wt,in}} \)

   The following set points are used:
   
   a. 2C/25C as the source/load side temperature respectively
   b. 5C/25C as the source/load side temperature respectively
   c. 5C/30C as the source/load side temperature respectively
   d. 5C/35C as the source/load side temperature respectively

   Where 2C, 5C are the brine inlet temperatures to the evaporator and 25C, 30C, 35C are the water inlet temperatures to the condenser.
Figure 4-1: Schematic showing brine and water temperature controls in heat pump

The source and sink temperatures (set points) are chosen because of the stability of system within these temperature lifts. The source side temperature is not allowed to go below 2°C because of the sudden drop in temperature owing to the speed change of compressor. It causes the brine temperatures to drop suddenly especially at higher speeds and the system may have to shut down before proceeding further.

The brine temperature to the inlet of evaporator is controlled using three-way valve. The tap water valve controls the water temperature to the inlet and outlet of condenser. It is difficult to maintain temperatures at the set points for a long period. The measurements are, therefore, recorded for the time period of ten minutes approximately at each compressor speed for each set point.

Due to the manual control of temperatures on brine and water side using three-way valve and tap water valve respectively, it is very difficult to maintain the exact temperatures of \( T_{\text{br, in}} \), \( T_{\text{wt, in}} \) and \( T_{\text{wt, out}} \). Therefore, the variation of ±0.4°C is allowed in the measurement of these temperatures.

The data collected via ClimaCheck Standard software program, is exported to MS Excel, where invalid values (outside the brine and water set point temperature limits) are filtered out.

With and without the vapour injection, compressor is run from 30Hz to 90Hz, because as the frequency (f) is increased beyond 90Hz compressor gets heated up and trips off.

The following parameters are measured during the experiments:

1. Compressor speed
2. The compressor power before and after the inverter
3. Condensation and evaporation pressures
4. Water inlet and outlet temperatures to the condenser, brine inlet and outlet temperatures to the evaporator, refrigerant temperature in and out of compressor, refrigerant temperature inlet of expansion valve.

Compressor speed is changed using the computer interface. Compressor power is measured using the power meter. All pressures and temperatures are measured using the respective sensors and data acquisition system.

Compressor power is measured before and after the inverter using Yokogawa power meter.

Once all the data is acquired, EES code is generated to study the overall heat pump performance, heat transfer behaviour in condenser and evaporator. In a separate EES code for the compressor model, built-in volume ratio, constant and variable parts of electromechanical losses of compressor are determined. The built-in volume ratio obtained from modelling is then used to calculate the compressor power in compressor and to evaluate losses due to mismatch between actual and built-in pressure ratio.

4.1 Limitations

The limitations which were faced during experimentations are summarized below.

1. Compressor speed cannot be increased beyond 90Hz because of protective T sensors at outlet of compressor, which trip off compressor at speeds greater than 90Hz.
2. The source side temperature is not allowed to go below 2°C because of the sudden drop in temperature owing to the speed change of compressor. It causes the brine temperatures to drop suddenly especially at higher speeds and the system may have to shut down before proceeding further.
3. Manual T control valves limit the accuracy to control brine and water temperatures at inlet/outlet of evaporator and condenser.

4.2 Assumptions

Following major assumptions are made during the calculations:

1. Heating losses in compressor are assumed to vary from 5% to 8% of total compressor power depending on compressor speed.
Heat pump performance without vapour injection

As explained before, for measuring the inverter losses, the results are generated for set points (2°C, 40°C), (2°C, 45°C) and (5°C, 45°C). In these set points 2°C and 5°C are the brine inlet temperatures to the evaporator, whereas 40°C and 45°C are the water outlet temperatures from the condenser. For measuring the performance of heat pump unit, heat transfer behaviour in condenser and evaporator, the results are generated for set points (2°C, 25°C), (5°C, 25°C), (5°C, 30°C) and (5°C, 35°C). In these set points 2°C and 5°C are the brine inlet temperatures to the evaporator, whereas 25°C, 30°C and 35°C are the water inlet temperatures to the condenser.

Fig. 5-1 presents the heat pump COP cool for three different set points where the source side temperature is kept constant and load side temperature is allowed to vary and the compressor speed is changed from 30Hz to 90Hz. The source/load side temperatures are 5°C/25°C, 5°C/30°C and 5°C/35°C respectively. Fig. 5-2 presents the heat pump COP cool for two different set points for the same range of compressor speeds where the load side temperature is kept constant and source side temperature is allowed to vary. The source/load side temperatures are 2°C/25°C and 5°C/25°C respectively.

![Figure 5-1: Heat pump COP cool when load side temperature is changed](image1)

![Figure 5-2: Heat pump COP cool when source side temperature is changed](image2)

Fig.5-1 and Fig.5-2 both show the decreasing trend in COP values of heat pump as compressor speed is increased. From fig. 5-1, it can be seen that at each compressor speed COP values are the highest when load side temperature is set at the lowest point. The COP Cool values of heat pump decrease from almost 5 at 30Hz to 2.56 at 90Hz. For each set point COP values decrease as compressor speed changes from 30Hz to 90Hz because of increase in pressure ratios. In fig. 5-2, it can be seen that the COP Cool values of heat pump decrease from almost 5 at 30Hz to 3 at 90Hz. Also as the source side temperature decreases so does the COP value at each compressor speed.
Fig. 5-3 presents the heat pump COP heat for three different set points where the source side temperature is kept constant and load side temperature is allowed to vary with an increase in the compressor speed from 30Hz to 90Hz. The source/load side temperatures are 5C/25C, 5C/30C and 5C/35C respectively. Fig. 5-4 presents the heat pump COP heat for two different set points, where the load side temperature is kept constant and source side temperature is allowed to vary and the compressor speed is changed from 30Hz to 90Hz. The source/load side temperatures are 2C/25C and 5C/25C respectively.

Fig. 5-3 shows that COP values decrease with an increase in the compressor speed. At each compressor speed COP values are the highest when load side temperature is kept at the lowest set point. Fig. 5-4 shows that COP value at each compressor speed decreases as the source side temperature decreases from 5C to 2C.

Fig. 5-5 below presents the heat pump heating capacity for three different set points where the source side temperature is kept constant and load side temperature is allowed to vary with an increase in the compressor speed from 30Hz to 90Hz. The source/load side temperatures are 5C/25C, 5C/30C and 5C/35C respectively. Fig. 5-6 presents the heat pump heating capacity for two different set points where the load side temperature is kept constant and source side temperature is allowed to vary with an increase in the compressor speed from 30Hz to 90Hz. The source/load side temperatures are 2C/25C and 5C/25C respectively.
Fig. 5-5 depicts that the heating capacity of heat pump increases from almost 5.6 kW to 12.9 kW with an increase in the compressor speed. When the set point temperatures are changed, heating capacity remains almost constant at each compressor speed. Fig. 5-6 shows that the heating capacity of heat pump increases from almost 5.5 kW to 13 kW with an increase in compressor speed. At each compressor speed heat pump has high heating capacity when source side temperature is high.

Fig. 5-7 below present the heat pump cooling capacity for three different set points where the source side temperature is kept constant and load side temperature is allowed to vary with an increase in the compressor speed from 30Hz to 90Hz. The source/load side temperatures are 5C/25C, 5C/30C and 5C/35C respectively. Fig. 5-8 present the heat pump cooling capacity for two different set points where the load side temperature is kept constant and source side temperature is allowed to vary with an increase in the compressor speed from 30Hz to 90Hz. The source/load side temperatures are 2C/25C and 5C/25C respectively.
Fig. 5-7 shows that the cooling capacity of heat pump increases from almost 4.7 kW to 9.75 kW with an increase in the compressor speed. The change in set point temperatures has a minimal effect on cooling capacities of heat pump at each compressor speed. Fig. 5-8 shows the similar trend with cooling capacity increasing from almost 4 kW to 9.7 kW with an increase in compressor speed. At each compressor speed heat pump has high cooling capacity when source side temperature is high.

Fig. 5-9 presents the heat pump compressor power for three different set points. In fig. 5-9 source side temperature is kept constant and load side temperature is allowed to vary. The compressor speed is changed from 30Hz to 90Hz. The source/load side temperatures are 5C/25C, 5C/30C and 5C/35C respectively. Fig. 5-10 presents the heat pump compressor power for two different set points where the load side temperature is kept constant and source side temperature is allowed to vary with an increase in the compressor speed from 30Hz to 90Hz. The source/load side temperatures are 2C/25C and 5C/25C respectively.
Fig. 5-9 shows that the compressor power ($E_{\text{comp}}$) increases almost from 1kW to 4kW as the compressor speed is changed from 30Hz to 90Hz. Most power is consumed by compressor when the load side temperature is the highest at 35°C. Similar trend can be witnessed in fig. 5-10 which shows that the compressor power increases almost linearly from 1kW at 30Hz to nearly 3.5kW at 90Hz. Electric power consumed by compressor is almost the same for both the set points at each compressor speed.

Fig. 5-11 shows a comparison between COP actual and Carnot COP for set point (5C, 30C). Delta T is temperature difference between the condenser and evaporator side, also called temperature lift.
As temperature lift across compressor increases with an increase in compressor speed, the Carnot COP (which is a measure of ideal output a heat pump unit can deliver working between condensing and evaporating temperatures) tends to decrease. Carnot COP is maximum for 30Hz and minimum for 90Hz for the heat pump.

Fig.5-12 shows the Carnot efficiency of the heat pump unit in percentage versus the compressor speed for three different set points when the compressor is run from 30Hz to 90Hz. Carnot efficiency is defined as the ratio of actual COP Heat to the Carnot COP Heat of the heat pump working between the same condensation and evaporation temperatures.

![Carnot efficiency graph](image_url)

Figure 5-12: Carnot efficiency of heat pump without vapour injection

The Carnot efficiency reaches maximum up to 61% for the heat pump when the set point is (5C, 25C). It can be seen that for each set points maximum Carnot efficiency reaches when compressor speed is 60Hz.

5.1 Inverter loss behaviour

Inverter losses are estimated for heat pump system without vapour injection. The tests are run for three different set points (2C, 40C), (2C, 45C) and (5C, 45C), where 2C and 5C are brine inlet temperatures to evaporator; 40C and 45C are water outlet temperatures to condenser. To measure inverter losses the same power meter is connected in two different positions: before and after the inverter. Power coming from the main supply is measured with power meter connected before the inverter. Power going into the compressor is measured with power meter connected after the inverter. The difference in these readings gives the inverter loss in Watts.

Fig.5-13 shows the inverter loss for three different set points when the compressor is run from 30Hz to 90Hz. The source/load side temperatures are 2C/40C, 2C/45C and 5C/45C respectively.
**Figure 5-13:** Inverter loss in Watts versus the compressor frequency

**Figure 5-14:** Inverter loss in percentage of total compressor power versus compressor frequency
Fig. 5-13 shows inverter losses in Watts. It may be observed from Fig. 5-13 that inverter losses increase from almost 95W to 225W as the compressor speed changes from 30Hz to 90Hz with maximum losses occurring at 90Hz. However, inverter losses show little sensitivity to the source and load side temperatures, i.e. change in inverter losses is almost negligible when load or source side temperatures are changed.

Fig. 5-14 shows the ratio of inverter loss to the total compressor power (power measure before the inverter) expressed as percentage. The percentage losses decrease slightly for all the three set points as the compressor speed is changed from 30Hz to 90Hz. Inverter losses vary from nearly 10% to 6% of total compressor power depending on compressor speed.

Fig. 5-15 shows the estimated uncertainty in the measured values of inverter losses for 95% confidence interval expressed in Watts. The uncertainty values are shown for two set points (2C, 45C) and (5C, 45C). To see how the uncertainty has been estimated for compressor power and inverter losses, see appendix D.

![Figure 5-15: Measured inverter loss and estimated uncertainty for 95% confidence level expressed in Watts](image)

The vertical error bars indicate the estimated uncertainty values in the inverter loss expressed in Watts for each set point at different compressor speeds. As it can be observed from fig. 15 that the uncertainty values are relatively higher at compressor speeds 80Hz and 90Hz. This is due to the higher fluctuations in the compressor power measured by the power meter before and after the inverter.

### 5.2 Compressor loss behaviour

To estimate the compression work, constant and variable part of electromechanical losses in compressor, losses due to pressure ratio mismatch, a semi-empirical model is built as suggested by (Madani, Ahmadi, Claesson, & Lundqvist, 2010).
5.2.1 Semi-Empirical model of compressor

(Winday, 1999) proposed equation to obtain internal compression work of compressor, assuming ideal gas

\[ W_{\text{comp}} = \frac{p_{\text{sup}}}{\gamma - 1} \cdot (v_i \gamma - 1 - \gamma) + \frac{p_{\text{ex}} V_s}{v_i} \]

Where \( v_i \) is the built-in volume ratio of compressor, and is defined as the ratio of the volume of trapped gas pocket at suction to the volume of trapped gas pocket at discharge. The built-in pressure ratio is given by correlation

\[ \pi_i = v_i^Y \]

This ratio is important to determine the losses due to pressure ratio mismatch. If the built-in pressure is higher than the operating pressure ratio, over-compression occurs, which results in power loss. Similarly if the built-in pressure is lower than the operating pressure ratio, under-compression occurs, which again results in power loss in compressor. However, there is no power loss if built-in pressure ratio is equal to operating pressure ratio. This mismatch influences the compressor efficiency and the following equation 6 suggested by (Granryd, Lundqvist, & al., 2005) is used to estimate these losses.

\[ \eta_{\text{built-in}} = \left( \frac{p_{\text{ex}}}{p_{\text{sup}}} \right)^{\frac{\gamma - 1}{\gamma - 1} - 1} \left( \pi_i^\gamma - 1 \right)
\]

The swept volume flow rate is calculated using equation

\[ \dot{V}_s = \frac{V_s n}{60} \]

\( V_s \) is the swept volume of compressor and \( n \) is the compressor speed in rpm. Compressor power can be calculated by combining equation 4 and 7. The total compressor power is the summation of compression power and compressor electromechanical losses. Whereas the electromechanical losses consist of two parts: constant part, a part of losses which remain almost constant during operation of compressor and can be assumed to depend only on the characteristics of motor or compressor; and variable part of losses which depend on the operating conditions of compressor and vary when internal compression power changes due to compressor speed and pressure ratio.

\[ W = \frac{W_{\text{comp}}}{\xi} + W_{\text{const}} \]

Where \( \xi \) represents the variable part of electromechanical losses, and \( W_{\text{const}} \) represents constant part of electromechanical losses (kW).

The results from the modelling are summarized in the following table.

<table>
<thead>
<tr>
<th>Table 6: Table showing results from modelling of compressor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Built-in volume ratio</td>
</tr>
<tr>
<td>-----------------------</td>
</tr>
<tr>
<td>2.42</td>
</tr>
</tbody>
</table>
Figure 5-16 shows the losses in compressor due to the mismatch between the operating pressure ratio and built-in pressure ratio when actual pressure changes as compressor speed is changed. It can be seen that the efficiency of compressor decreases as the mismatch between the two pressure ratios increases. These losses account for the fact that operating pressure ratio does not match the built-in pressure ratio due to over-compression or under-compression, which incurs losses on compressor power. The compressor operates in a way that mismatch between built-in and operating pressure ratio is minimum at operating pressure ratio of 3.48, where the efficiency is maximum.

![Figure 5-16: Built-in efficiency of compressor showing losses due to mismatch between built-in and operating pressure ratios](image)

Total compressor losses (including inverter losses, total electromechanical losses in motor) are shown in figure 5-17 for three different set points. As explained in Chapter 2, since compressor used in this study is a scroll compressor, EM losses cannot be separated between motor and isentropic losses. Therefore, EM losses refer to the combined losses incurred in motor and due to non-isentropic compression work of compressor. Total compressor loss increases from almost 440W to 970W as compressor speed is changed. Lowest losses occur at low speeds and as speed increases, the losses also increase in compressor. For all the three set points, total compressor losses remain almost constant at each compressor speed. It is important to note that losses due to mismatch between pressure ratios are not represented by this figure.
Figure 5.17: Total compressor losses in compressor excluding built-in pressure losses when compressor speed is changed

Figure 5.18 represents the segregation between inverter and electromechanical losses for set point (5C, 45C).

Figure 5.18: Inverter and total electromechanical losses of compressor

It can be seen from figure 5.18 that majority of total losses are incurred in the electromechanical part of compressor. Electromechanical losses increase from almost 345W at 30Hz to 740W at 90Hz. As ZHW16K1P is a scroll compressor, the losses inside compressor cannot be separated into motor and compression losses. They are represented by the electromechanical losses inside compressor.
5.2.2 Compressor total isentropic efficiency

Fig. 5-19 shows the total isentropic efficiency of compressor for three different set points when compressor speed is changed from 30Hz to 90Hz. This isentropic efficiency, $\eta_{isen}$, represents all the losses in compressor comprising constant as well as variable part of electromechanical losses, isentropic compression, loss due to heating of gas by the motor during suction inside compressor and losses due to mismatch between pressure ratios. Figure represents isentropic efficiency for three different set points (2C, 40C), (2C, 45C) and (5C, 45C) for the compressor power measured before inverter.

$$\eta_{isen} = \frac{\dot{w}_{isen}}{\dot{W}}$$

Where $\dot{W}$ is the total power consumed by compressor and $\dot{w}_{isen}$ is the isentropic compression work.

![Graph showing isentropic efficiency](image)

**Figure 5-19: Total isentropic efficiency of compressor including isentropic compression work, suction gas heating losses and electromechanical losses**

Total isentropic efficiency changes almost 8% and the maximum isentropic occurs at compressor speed between 50-60Hz for all the set points.

5.3 Heat transfer process in condenser

When compressor speed changes the refrigerant and water temperatures inlet and outlet of condenser change as well. Due to this the condensing, desuperheating and subcooling areas inside condenser also change. Fig. 5-20 shows the effect of compressor speed change on the temperatures of water and refrigerant entering and leaving the condenser for set point (5C, 30C). The refrigerant temperature entering condenser increases from nearly 65°C to above 90°C with increase in compressor speed from 30Hz to 90Hz respectively.
Figure 5-20: Change in refrigerant and water temperatures inlet and outlet of condenser for set point (5C, 30C) and heat pump

Heat transfer behaviour in condenser is analyzed in all the three sub-sections of the condenser: desuperheating, condensation and subcooling sections. To see how UA values have been calculated experimentally in each subsection, see appendix A.1 and A.2.

Fig.5-21 below shows heat transfer behaviour inside the condenser as mass flow rate of refrigerant changes with the change in compressor speed from 30Hz to 90Hz. The source/load side temperatures are 5C/30C respectively.
As it may be seen from fig.5-21 that the UA_cond value in the condensing section decreases from almost 2.46kW/K to 0.97kW/K as the mass flow rate increases, while the UA_desup values for desuperheating section increase from 0.15kW/K to 0.19kW/K. For the subcooling section the change in UA_sub value is much higher than the desuperheating section, from 0.04kW/K to 0.36kW/K.

When compressor speed is increased, mass flow rate through the compressor also increases and so does the subcooling. The condensation area decreases which results in the decrease in the UA_cond value in the condensing section. In the subcooling and desuperheating sections heat transfer takes place in a liquid phase and vapour phase respectively. As the effective condensation area decreases, desuperheating and subcooling section areas increase which results in the increase in UA_sub and UA_desup values. The increase in UA_sub value is higher than UA_desup because heat transfer takes place in the liquid phase in the subcooling section.

Fig.5-22 below shows the total UA_condenser_total values (calculated by adding UA_cond, UA_sub and UA_desup in the condensing, desuperheating and subcooling sections) inside the condenser as mass flow rate of refrigerant changes with the change in compressor speed from 30Hz to 90Hz. The source/load side temperatures are 5C/25C, 5C/30C and 5C/35C respectively.
Figure 5-22: Total UA value of condenser against refrigerant mass flow rate

Figure 5-22 shows that the total UA value of condensation decreases from 2.6kW/K to 1.53kW/K.

Fig.5-23 shows heat flux, \( q \) (kW/m\(^2\)) inside the condenser as mass flow rate of refrigerant changes with the change in compressor speed from 30Hz to 90Hz. To find heating flux, heat capacity of condenser is divided by the total heat transfer area of condenser.

Figure 5-23: Total heat flux in condenser against compressor speed

Heat flux increases in condenser from 2.3kW/m\(^2\) to 5.5kW/m\(^2\) with increase in compressor speed because of the increase in heating capacity of condenser.
Fig. 5-24 shows the total heat transfer coefficient $U$ (kW/Km²) inside the condenser as mass flow rate of refrigerant changes with the change in compressor speed from 30Hz to 90Hz. To find total $U$ value, total UA value of condenser is divided by the total heat transfer area of condenser. The source/load side temperatures are 5°C/25°C, 5°C/30°C and 5°C/35°C. The total $U$ value decreases from 1.12kW/Km² to 0.65kW/Km².

**Figure 5-24: Total $U$ value of condenser against heat flux**

### 5.4 Heat transfer process in evaporator

When compressor speed is varied the refrigerant and brine temperatures entering and leaving the evaporator also change. Fig.5-25 shows the change in brine and refrigerant inlet/outlet temperatures entering and leaving the evaporator as compressor speed is changed from 30Hz to 90Hz for set point (5°C, 30°C).
Refrigerant inlet temperature to evaporator decreases from -1°C to -8°C as compressor speed is changed from 30Hz to 90Hz.

Heat transfer behaviour is analyzed separately in both the sub-sections of evaporator: evaporating and superheating section. The overall UA\_evap and U\_evap are calculated using equations and EES code.

Logarithmic mean temperature difference (LMTD) approach is used to calculate the heat transfer inside evaporator. (Claesson, 2005) showed that although boiling heat transfer coefficient is not constant inside BPHE evaporator, LMTD approach may be used in a heat flux governed heat transfer boiling where LMTD exceeds 4-5°C, even though small deviations occur at low LMTD.
Fig. 5-26 below shows heat transfer behaviour inside the evaporator as mass flow rate of refrigerant changes with the change in compressor speed from 30Hz to 90Hz. The source/load side temperatures are 5°C/30°C respectively.

![Graph showing heat transfer behaviour](image)

**Figure 5-26: Total UA value of evaporator against refrigerant mass flow rate for set point (5°C, 30°C)**

As it may be observed from figure 5-26 that the UA value in evaporator changes only slightly and remains almost constant as compressor speed changes. However, the rate of change of UA value in the superheating section is higher and it changes from 0.07 kW/K to 0.03 kW/K. According to Cooper’s relations UA values inside the evaporating section should increase with the increase in mass flow rate of refrigerant. Because as mass flow rate increases through evaporator, effective heat transfer area inside evaporator also increases which in turn increases the UA value in evaporator.

Fig.5-27 below shows the total UA value inside the evaporator as mass flow rate of refrigerant changes with the change in compressor speed from 30Hz to 90Hz. The source/load side temperatures are 5°C/25°C, 5°C/30°C and 5°C/35°C respectively.
For all the set points the total UA value inside the evaporator remains almost constant. The total UA value in evaporator is calculated by summing the UA values in evaporating and superheating sections.

Fig.5-28 below shows the heat flux value inside the evaporator as mass flow rate of refrigerant changes with the change in compressor speed from 30Hz to 90Hz.

Figure 5-27: Variation in total U values in evaporator against refrigerant mass flow rate

Figure 5-28: Change in heat flux against compressor speed
The heat flux values inside evaporator increase from almost 2.5 (kW/m$^2$) to 5.7 (kW/m$^2$) as compressor speed is changed from 30Hz to 90Hz. At each compressor speed as the load side temperature is increased the heat flux value decreases slightly.

Fig.5-29 below shows the total U value inside the evaporator as mass flow rate of refrigerant changes with the change in compressor speed from 30Hz to 90Hz. The source/load side temperatures are 5C/25C, 5C/30C and 5C/35C respectively. The total U values for evaporator remain almost constant as mass flow rate changes.

![Figure 5-29: Variation in total U value inside evaporator against heat flux](image-url)
Chapter 6

Heat pump performance with vapour injection (VI)

The vapour injection technique is employed in scroll compressors with the addition of an economizer in vapour compression cycle, as shown in figure 1-3. With this configuration it is possible to achieve higher COPs, heating and cooling capacities than conventional refrigeration cycles. Due to the increase in capacity, smaller displacement compressor can be used for the given heating load. The operating envelopes of compressors are also widened owing to the additional cooling provided by the intermediate vapour injection.

The compressor used in study is a scroll compressor equipped with vapour injection port. As claimed by manufacturers, it is possible to achieve the following benefits:

1. Cooling capacity improvement could be up to 40% with the same displacement compressor.
2. Cooling COP could go up by 15%.
3. Heating capacity and heating COP are improved due to higher refrigerant mass flow across condenser.
4. Cooling effect of vapour injection (lower refrigerant temperature at outlet of compressor) allows large operating envelopes.
5. Vapour injection effect is proportional to the pressure ratios, i.e. higher heating capacities and COP when needed.

To make modifications for vapour injection, system is emptied first by vacuuming refrigerant R410A into a separate bottle. When the components are installed along with piping, system is checked against any leakages. System is pressurized with nitrogen at 10Bars and left undisturbed for two days. If pressure of nitrogen does not drop, it ensures no leakages in the system. Vacuum is created inside using vacuum pump to ensure no air and refrigerant R410A is filled in the heat pump.

6.1 Results and discussions

The degree of superheat at the inlet of vapour injection port of compressor is set at 5°C. System is recharged with two different amounts of refrigerant, 1.15kg and 1.28kg, and comparison is made amongst them. Tests are carried out for set points (5C, 25C) and (5C, 30C) at compressor speed varying from 30Hz to 90Hz. As compressor speed is increased beyond 90Hz, system shuts down due to protective temperature sensors at outlet of compressor. The methodology for control of temperatures at evaporator and condenser is already explained in section 4.

Fig. 6-1 and fig.6-2 present heat pump’s COP cool with and without vapour injection for set point (5C, 25C) with different refrigerant charge inside system. In Fig. 6-1 with refrigerant mass = 1.15kg, it can be seen from figure that for system with VI, COP cool are higher than system without VI at all compressor speeds except for 30Hz, where COPs for VI and without VI are almost equivalent. Heat pump COP with VI increases by an average of 10.66% at each compressor speed except for 30Hz. As explained later, this is because of the same approximate refrigerant temperature at the outlet of compressor for both the configurations.
In fig. 6-2 with refrigerant mass = 1.28kg, system with VI witnesses an average increase of 8.85% in COP cool at compressor speeds beyond 50Hz. From 30Hz to 50Hz COP cool for VI is lower than heat pump without VI. This is because of high refrigerant charge inside system which causes a higher refrigerant temperature at outlet of compressor on frequencies 30Hz to 50Hz.

<table>
<thead>
<tr>
<th>ref. mass = 1.15 kg</th>
<th>ref. mass = 1.28kg</th>
</tr>
</thead>
<tbody>
<tr>
<td>With VI</td>
<td>Without VI</td>
</tr>
</tbody>
</table>

![Figure 6-1: COP Cool of heat pump system with vapour injection for set point (5C, 25C) and ref. mass = 1.15kg](image1)

![Figure 6-2: COP Cool of heat pump system with vapour injection for set point (5C, 25C) and ref. mass = 1.28kg](image2)

Fig. 6-3 and fig.6-4 present heat pump’s COP heat with and without vapour injection for set point (5C, 25C) with different refrigerant charge inside system. In Fig. 6-3 with refrigerant mass = 1.15kg, COP heat of heat pump with VI increases by an average of 9.40%. As it can be seen from figure that for system with VI, COP heat are higher than system without VI at all compressor speeds except for 30Hz, where COPs for VI and without VI are almost equivalent.

In fig. 6-4 with refrigerant mass = 1.28kg, system with VI witnesses an average increase of 6.79% in COP heat at compressor speed beyond 50Hz. From 30Hz to 50Hz COP heat for VI is lower than heat pump without VI.

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Fig. 6-5 and Fig. 6-6 present the comparison between cooling capacities for system with VI and without VI at set point (5°C, 25°C) for different refrigerant charges. In Fig. 6-5, the system with VI witnesses an average increase of 21.82% in cooling capacity over the range of different compressor speeds as compared to system without VI.

For refrigerant charge 1.28kg inside system, average increase in cooling capacity is 25.45% for VI over different compressor frequencies.
Fig.6-7 and fig. 6-8 present the comparison between heating capacities for system with VI and without VI at set point (5C, 25C) for different refrigerant charges. In fig. 6-7, the system with VI witnesses an average increase of 19.26% in heating capacity over the range of different compressor speeds as compared to system without VI.

For refrigerant charge 1.28kg inside system, average increase in heating capacity is 24.45% for VI over different compressor frequencies.

Fig. 6-9 and fig. 6-10 show the compressor power, measured before inverter, for set point (5C, 25C) at different compressor speeds and refrigerant charges. In both the cases, the power consumption of compressor for system with VI is higher than system without VI. For refrigerant mass 1.15kg there is an average increase of 9.8% in power consumption of VI system, while for refrigerant mass 1.28kg there is an average increase of 21.7% in power consumption of VI system.
Fig. 6-11 and fig. 6-12 show the behaviour of pressure ratios (ratio of pressure at outlet of compressor to the pressure at its suction) for different refrigerant charges for set point (5C, 25C). In case of refrigerant mass 1.15kg, it is possible to achieve higher COPs, heating and cooling capacities due to the low pressure ratios for VI at each compressor speed. From fig. 6-12, it can be seen that pressure ratios are higher at frequencies 30Hz to 50Hz for heat pump with VI as compared to configuration with no VI. These higher pressure ratios lead to higher refrigerant temperatures at outlet of compressor which reduces the benefits of VI at low speeds, as explained earlier.
Fig. 6-13 and fig. 6-14 show the change in refrigerant compressor outlet temperature with the addition of vapour injection port. In case of refrigerant mass 1.28kg, refrigerant temperature at outlet of compressor is lower only at high speeds for VI, due to which higher COPs are achieved at high frequencies. From 30Hz to 50Hz, refrigerant temperature at compressor outlet is higher for VI which gives lower COPs at these compressor speeds. In case of refrigerant mass 1.15kg, higher COPs are achieved at frequencies ranging beyond 30Hz where refrigerant temperature at outlet of compressor is lower for VI.

**ref. mass = 1.15kg**

![Graph showing refrigerant temperature outlet of compressor for set point (5°C, 25°C) and ref. mass = 1.15kg](image)

**ref. mass = 1.28kg**

![Graph showing refrigerant temperature outlet of compressor for set point (5°C, 25°C) and ref. mass = 1.28kg](image)

Fig. 6-15 shows the percentage of extracted refrigerant mass that is injected in the vapour injection port of economizer for set point (5°C, 25°C) and refrigerant mass 1.15kg. The percentage of extracted mass of refrigerant for vapour injection increases from 0.67% at 30Hz to 2.27% at 90Hz. Due to more mass of total refrigerant entering vapour injection port at high speeds, higher COPs, heating and cooling capacities, low refrigerant temperature at outlet of compressor can be achieved at high compressor speeds.
6.2 Heat transfer behaviour in condenser

Heat transfer behaviour in condenser is analyzed for refrigerant mass 1.15kg. Figure 6-16 below shows the effect of compressor speed change on water and refrigerant temperature at the inlet and outlet of condenser for set point (5°C, 30°C). The refrigerant temperature at outlet of compressor increases from 64.31°C at 30Hz to 83.46°C at 90Hz.
Figure 6-16: Change in refrigerant and water temperatures inlet and outlet of condenser for set point (5°C, 30°C)

Figure 6-17 shows heat transfer behaviour in condenser in all its three sub-sections: condenser, subcooler and desuperheater.

It is seen from the figure that UA values for condenser change from 2.076kW/K to 2.629kW/K. UA values for desuperheating section increase almost linearly from 0.125kW/K to 0.269kW/K, whereas for subcooling section they remain almost constant.

Figure 6-18 shows the total UA values of condenser (summation of UA values in condenser, desuperheater and subcooler) for two set points (5°C, 25°C) and (5°C, 30°C) at different compressor speeds.
Figure 6-18: Total UA values of condenser in kW/K for two set points

It can be seen from the figure that UA values in condenser first increase as compressor speed changes from 30Hz to 50Hz and then decrease with compressor speed changing up to 90Hz.

Figure 6-19 shows the total U values in condenser for two set points (5C, 25C) and (5C, 30C).

6.3 Heat transfer behaviour in evaporator

Heat transfer behaviour in evaporator is analyzed for refrigerant mass 1.15kg. Fig. 6-20 presents the effect of change of compressor speed on brine and refrigerant temperature entering and leaving the evaporator. As compressor speed is changed from 30Hz to 90Hz, refrigerant temperature in the midst of evaporator decreases from nearly 0°C to -6°C respectively.
Fig. 6-20: Change in refrigerant and brine temperatures inlet and outlet of evaporator for set point (5C, 30C) and heat pump

Fig. 6-21 shows the UA values in evaporator in kW/K for set point (5C, 30C). UA values for evaporating section of evaporator change from 1.41kW/K at 30Hz to 2.19kW/K at 60Hz. For evaporating section the UA values first increase with compressor speed going up to 60Hz and then decrease down to 1.45kW/K. For superheating section of evaporator, the UA values increase and then become constant at higher compressor speeds.
Figure 6-21: UA values in kW/K for evaporating and superheating sections of evaporator for set point (5C, 30C)

Fig. 6-22 shows a comparison between total UA values in kW/K for two different set points at different compressor frequencies.

It can be seen from figure that the total UA value in evaporator increases as compressor speed goes up to 50Hz and then decreases. It can be seen that as load side temperature changed, the change in UA values is not so significant at each compressor speed.

Fig. 6-23 shows the total U values in evaporator in kW/Km² for two set points (5C, 25C) and (5C, 30C). To find total U value, UA value of evaporator is divided by total heat transfer area of evaporator.
It can be seen that as load side temperature changed, the change in UA values is not so significant at each compressor speed. The maximum U value in evaporator goes up to 1.37 kW/m²K at 50Hz.
Chapter 7

Conclusions

The heat pump performance is evaluated in this study. A variable capacity scroll compressor is used in heat pump which has the provision for vapour injection and is also equipped with a permanent magnet motor. The study is conducted in two phases: heat pump performance without vapour injection and heat pump performance with vapour injection. The summary of results from both the phases is presented here.

Phase 1: Heat pump performance without vapour injection

In this phase,

1. Analysis of COP heat, COP cool, heating capacity, cooling capacity, Carnot efficiency, total isentropic efficiency and compressor power is done.
2. Total electromechanical and inverter losses are analyzed.
3. Comparison of electromechanical losses is made with the EM losses incurred in compressor analyzed in a study conducted by Ahmadi, 2010.
4. Heat transfer behaviour in condenser and evaporator is analyzed

During tests it is observed that COP cool decreases by 38% while COP heat decreases by 32% with increase in compressor speed from 30Hz to 90Hz. Both heating and cooling capacities see an increase with increase in compressor speed. Carnot and total isentropic efficiency reach their maximum at compressor speed of 60Hz. It is also observed that the built-in efficiency of compressor decreases with increase in compressor speed due to increase in pressure ratios.

Inverter losses increase, however the ratio of inverter losses to the total compressor power decreases with increase in compressor speed. Inverter losses increase from almost 95W to 225W as the compressor speed changes from 30Hz to 90Hz with maximum losses occurring at 90Hz. However, inverter losses show little sensitivity to the source and load side temperatures, i.e. change in inverter losses is almost negligible when load or source side temperatures are changed. The percentage losses (ratio of inverter losses to the total compressor power) decrease slightly for all the set points as the compressor speed is changed from 30Hz to 90Hz. Inverter losses vary from nearly 10% to 6% of total compressor power depending on compressor speed.

Electromechanical losses of compressor are much higher than the inverter losses and so make most part of the total compressor losses (summation of inverter and electromechanical losses).

Total UA value in condenser decreases while total UA values in evaporator remain almost constant as compressor speed is increased from 30Hz to 90Hz.

Phase 2: Heat pump performance with vapour injection

For vapour injection, heat pump’s performance is evaluated for two different refrigerant charges, 1.15kg and 1.28kg. It is noted that heat pump performs better for refrigerant charge 1.15kg even at lower compressor speeds as compared to refrigerant charge 1.28kg. For refrigerant charge 1.15kg,
heat pump COP cool with vapour injection increases by an average of 10.66%, while COP heat increases by an average of 9.4%, at each compressor speed except for 30Hz, as compared to conventional heat pump cycle with no vapour injection. Similarly refrigerant temperature at outlet of compressor also reduces with vapour injection which leads to the better performance of heat pump. For refrigerant mass = 1.28kg, system with VI witnesses an average increase of 8.85% in COP cool and 6.79% in COP heat at compressor speeds beyond 50Hz.

In order for this system to operate optimally for vapour injection at almost all compressor speeds, refrigerant charge inside system should be around 1.15kg.

Experimental UA values for condenser and evaporator are estimated for refrigerant mass 1.15kg. Results show that total UA values in condenser first increase with compressor speed changing from 30Hz to 50Hz and then decrease with compressor speed going up to 90Hz. The total UA values in evaporator also show the same behaviour, increasing up to 50Hz and then decreasing. For both evaporator and condensers, as load side temperature is changed, the change in UA values is not so significant.

7.1 Future work

Manual valves have been used in the system to control brine and water temperatures at the inlet/outlet of evaporator and condenser respectively. It is difficult to maintain temperatures at the set points precisely, which affects the performance of heat pump. It is recommended to replace those valves with the automatic control valves to see how much performance is affected.

System’s performance with vapour injection has been tested at two condensing temperatures 25C and 30C. The benefits of vapour injection can be more at high condensing temperatures. It is recommended to record the performance of system with vapour injection at high condensing temperatures.

The operating envelopes are also widened for heat pump with vapour injection. The present study does not consider this, which is recommended for the future work. Compressor used in this study can also be used with wet vapour injection. It will be interesting to see the performance of system with wet vapour injection instead of vapour injection with superheat.
References


Appendix

A. Heat transfer analysis

A.1 Experimental UA values of condenser

Heat transfer area in condenser is divided into three sections: desuperheating, condensing, subcooling. Figure A.1-1 shows all the three sub-sections of a condenser.

![Figure A.1-1: Desuperheating, condensing and subcooling sections of condenser](image)

The overall heat transfer coefficient in condenser is equal to the ratio between heat flow rate, heat transfer area and the logarithmic mean temperature difference.

\[
U_{\text{cond}} = \frac{\dot{Q}_1}{A \cdot \text{LMTD}}
\]

In desuperheating section of condenser,

\[
U_{\text{A}_{\text{desup}}} = \frac{\dot{Q}_{\text{desup}}}{\text{LMTD}_{\text{desup}}}
\]

\[
\dot{Q}_{\text{desup}} = m_{\text{ref}} \cdot (h_{\text{ref, in}} - h_{g,\text{sat}})
\]

\[
\text{LMTD}_{\text{desup}} = \frac{(T_{g,\text{sat}} - T_{\text{wt},1}) - (T_{\text{ref,comp out}} - T_{\text{wt},\text{out}})}{\ln \left(\frac{T_{g,\text{sat}} - T_{\text{wt},1}}{T_{\text{ref,comp out}} - T_{\text{wt},\text{out}}}\right)}
\]

In saturated section of condenser,

\[
U_{\text{A}_{\text{cond}}} = \frac{\dot{Q}_{\text{cond}}}{\text{LMTD}_{\text{cond}}}
\]

\[
\dot{Q}_{\text{cond}} = m_{\text{ref}} \cdot (h_{g,\text{sat}} - h_{f,\text{sat}})
\]
In subcooling section of condenser,

\[ UA_{\text{subcool}} = \frac{\dot{Q}_{\text{subcool}}}{LMTD_{\text{subcool}}} \]

\[ \dot{Q}_{\text{subcool}} = \dot{m}_{\text{ref}} \cdot (h_{f,\text{sat}} - h_{\text{ref,out}}) \]

\[ LMTD_{\text{desup}} = \frac{(T_{\text{ref,condout}} - T_{\text{wt.in}}) - (T_{f,\text{sat}} - T_{\text{wt.in}})}{\ln \left( \frac{T_{\text{ref,condout}} - T_{\text{wt.in}}}{T_{f,\text{sat}} - T_{\text{wt.in}}} \right)} \]

### A.2 Experimental UA values of evaporator

Heat transfer area in evaporator is divided into two sections: superheating section and evaporating section (saturated boiling section). Figure A.2-1 shows both these sub-sections of evaporator.

![Superheating and evaporating sections of evaporator](image)

The overall heat transfer coefficient in evaporator is equal to the ratio of heat flow rate to the heat transfer area and the logarithmic mean temperature difference.

\[ U_{\text{evap}} = \frac{\dot{Q}_2}{A \cdot LMTD} \]

In superheating section,

\[ UA_{\text{sup}} = \frac{\dot{Q}_{\text{sup}}}{LMTD_{\text{sup}}} \]

\[ \dot{Q}_{\text{sup}} = \dot{m}_{\text{ref}} \cdot (h_{\text{ref,out}} - h_{g,\text{sat}}) \]
In evaporating section,

\[ U_{A_{evap}} = \frac{Q_{evap}}{LMTD_{evap}} \]  

\[ \dot{Q}_{evap} = m_{ref} \cdot (h_{g,sat} - h_{f,sat}) \]  

\[ LMTD_{evap} = \frac{(T_{br,out} - T_{ref, evapin}) - (T_{br,mid} - T_{g,sat})}{\ln \left( \frac{T_{br,out} - T_{ref, evapin}}{T_{br,mid} - T_{g,sat}} \right)} \]

\[ LMTD_{evap}^{sup} = \frac{(T_{br,in} - T_{ref, compin}) - (T_{br,mid} - T_{g,sat})}{\ln \left( \frac{T_{br,in} - T_{ref, compin}}{T_{br,mid} - T_{g,sat}} \right)} \]  

B. Switching frequency

VACON is a pulse width modulated (PWM) type inverter that uses the PWM technique to generate sine-like waves for electrical motor of compressor which run on alternating current (AC). These sine-like waves are created by increasing or decreasing the pulse width in the inverter. The smoothing of the output sine-like waveform is dependent on width and number of modulated impulses per given cycle (called switching frequency). The higher the switching frequency, the smoother the output sine-like waveform.

\[ \text{Figure B-1: Pulse width modulation} \]

C. Communication between computer and inverter drive

Serial IP redirector and VSS Gen II interface are installed on computer. Serial IP redirector is a tactical software intended for IP address to COM port conversion. The wireless router connected with the computer then sends the signal to an antenna fixed on the drive for the wireless communication. Inverter speed is controlled using VSS Gen II interface.
D. Uncertainty analysis of the power measured by power meter before and after the inverter

The uncertainty of type A in power readings, recorded by Yokogawa power meter, is estimated using the concept of standard deviation $S_x$. Power readings are recorded for a time period of 10 min and a large data set of readings is collected. However, 300 measurements are used to estimate the average value of compressor power at each compressor speed.

Standard deviation in the power readings is calculated as

$$S_x = \sqrt{\frac{\sum_{i=1}^{n} (x_i - \bar{x})^2}{n-1}}$$

Where $\bar{x}$ is the mean value of power for $n = 300$ readings and is given by relation

$$\bar{x} = \frac{\sum_{i=1}^{n} x_i}{n}$$

The concept of confidence interval is used to express the total uncertainty of a value. A confidence interval is an interval estimate of population parameter and is used to indicate the reliability of an estimate. The confidence level determines how frequently a given interval contains the estimated parameter. With normal distribution of the data for one standard deviation, confidence level is 68.3%. For the confidence equivalent to two standard deviations, confidence level is 95.4%.

To estimate uncertainty in compressor power and inverter losses a confidence level of 95.4% has been used. Uncertainty in power readings is estimated both before and after the inverter. The relation gives the total uncertainty in compressor power.
D.1 Chauvenet’s criteria

When large number of readings are recorded for a parameter, this criteria specifies that a reading may be rejected if the probability of obtaining the particular deviation from the mean is less than $1/2n$. This is a restrictive criterion to eliminate the dubious data points from the data set. After the elimination of dubious points, a new standard deviation and uncertainty should be computed for the new data set.

Since number of data points recorded for the power measurements are $n = 300$, Chauvenet’s criteria is

$$\frac{(x_i - \bar{x})}{s_x} \leq 3.14$$