Experimental Analysis of Variable Capacity Heat Pump Systems equipped with a liquid-cooled frequency inverter

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Master of Science Thesis
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Energy Technology EGI-2013-006MSC
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<th>Examiner</th>
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<tr>
<td>2013-02-09</td>
<td>Joachim Claesson</td>
<td>Hatef Madani</td>
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Commissioner: Contact person
Abstract

Using an inverter-driven compressor in variable capacity heat pump systems has a main drawback, which is the extra loss in the inverter. The present experimental study aims to recover the inverter losses by using brine-cooled and water-cooled inverters, thereby improving the total efficiency of the heat pump system. In order to achieve this goal, a test rig with the air-cooled, water-cooled and brine-cooled inverters is designed and built, and a comparative analysis of the recovered heat, inverter losses and system performance is conducted when the compressor is driven by the water-cooled, brine-cooled and air-cooled inverters at three different switching frequencies for each inverter.

The experimental results show that the inverter losses as a magnitude and as a ratio of the total consumed power are lowest in the brine-cooled inverter and highest in the air-cooled one at all the compressor speeds and all the inverter switching frequencies. Moreover, the recovered energy varies between 45 and 125 (W) in the water-cooled inverter, which corresponds to 63 and 69 (%) of the inverter losses; while it varies between 61 and 139 (W) in the brine-cooled inverter, which corresponds to 79 and 90 (%) of the inverter losses. It is also proved that the improvement of the system coefficient of performance (COP$_{sys}$) is almost the same when the water-cooled or the brine-cooled inverter is used and varies between 0.54 and 3 (%) in comparison with using the air-cooled one. Indeed, the total isentropic efficiency of the compressor is improved slightly when using the water-cooled inverter and little more when using the brine-cooled one at the same running conditions. In addition, the total isentropic efficiency of the compressor is improved by increasing the inverter switching frequency when any of the inverters is used.

The experimental results also show that cooling the inverter by the water, which comes out from the condenser, increases the maximum temperature of the base plate of the inverter about 10 °C which could cause a two-fold deterioration in the inverter median life in comparison with cooling the inverter by air. On the contrary, using the brine for cooling the inverter decreases the maximum temperature of the base plate of the inverter about 30 °C which could cause about a six-fold improvement in the inverter median life.
Acknowledgements

The smallest act of kindness is worth more than the greatest intention.

Khalil Gibran (Lebanese writer 1883 –1931)

Apart from my own efforts, the success of this project depends largely on the encouragement and guidelines given by many others. Therefore I consider my work as a practical culmination of a collective cooperation par excellence; and I take this opportunity to express my gratitude to the people who have been instrumental in the successful completion of this project.

First, I take immense pleasure in thanking my examiner Associate Prof. Joachim Claesson for his support and guidance in the project, as well as, in many courses in which he was the first lecturer during my journey at KTH. I am also deeply grateful to my supervisor Dr. Hatef Madani who gave me the opportunity to do my thesis in the field that I really wanted and liked. I also would like to thank him for his support and guidance during project.

I am highly indebted to (CTC Enertech AB), who financed my project, for their constant support as well as for providing necessary information regarding the project. Special thanks to Kent Karlsson (Engineering in CTC Enertech AB) for his unlimited support, interest, valuable discussions and quick response even during his vacation time.

Two people have made their distinctive mark on my work by encouraging me during the project period: Dr. Samer Sawalha and the lab manager Peter Hill. I am deeply grateful to Samer for his kindness, friendship and for all his discussions and concern about what is happening in Syria nowadays, which eased my sorrow and pain. Thanks must also go to Peter, who in addition to his constant practical support, has sustained me during many long working days with his kind and encouraging words (Va duktig du är! Bra! bra jobbat!........).

The technicians Benny Sjöbery, Karl Åke and Anders Eklund also deserve my thanks for their approach to building the test rig, despite their busy schedule. Many thanks for the staff of the division of Applied Thermodynamic and Refrigeration for their kindness, gentleness and the friendly atmosphere that they have created in the division.

I will never forget the gentleness of Behzad Monfared who gave me some of his precious time to help me with the software of the acquisition system. Thanks also to, Sad Jaral, Monika Ignatowicz, Zahid Anwar Nabil Kasem and Omar Abuelnaga who helped me with their useful discussions in solving some of the problems that appeared during the building and the operation of the test rig.

Wisdom ceases to be wisdom when it becomes too proud to weep, too grave to laugh, and too selfish to seek other than itself.

Khalil Gibr
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<td>( \dot{E}_p )</td>
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<tr>
<td>br, out</td>
<td>Brine Outlet</td>
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<tr>
<td>br, inv</td>
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<td>comp, loss</td>
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<tr>
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<td>Inverter Losses</td>
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<tr>
<td>is, tot</td>
<td>Isentropic Total</td>
<td></td>
</tr>
<tr>
<td>ref</td>
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</tr>
<tr>
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<td>Refrigerant Inlet</td>
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<tr>
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<tr>
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<td>Refrigerant Condenser Inlet</td>
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<td>Refrigerant Condenser Outlet</td>
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<tr>
<td>ref, evap, in</td>
<td>Refrigerant Evaporator Inlet</td>
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</tr>
<tr>
<td>ref, evap, out</td>
<td>Refrigerant Evaporator Outlet</td>
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<tr>
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<tr>
<td>tot, loss</td>
<td>Total Losses</td>
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<tr>
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<tr>
<td>wt, in</td>
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</tbody>
</table>
wt, out  Water Outlet
wt,inv  Water Inverter
wt,inv,in  Water Inverter Inlet
wt,inv,out  Water Inverter Outlet

Abbreviations

BPHE  Brazed plate heat exchanger
COP  Coefficient Of Performance
EMI  Electromagnetic Interference
GHG  Greenhouse gas
GSHP\(_S\)  ground source heat pump systems
LMTD  Logarithmic Mean temperature Difference
PWM  Pulse-Width Modulation
RPS  Revolution Per Second
SPF  Seasonal Performance Factor
VSD  Variable Speed Drive

Greek Symbols

\(\eta\)  Efficiency
1. Introduction

Using Ground Source Heat Pumps (GSHPs) for heating commercial and residential buildings has been increasing dramatically the last decade. The estimated number of these systems in Europe is 1.25 million, and one third of them are concentrated in Sweden alone (Bayer et al, 2011). In 2008, GSHPs saved 3.7 million tons of CO\textsubscript{2}, which corresponds to 0.7\% greenhouse gases (GHGs) emission savings. The estimated saving potential by using GSHP\textsubscript{s} in Europe is about 30\% of the total GHGs. (Bayer et al, 2011)

Any improvement in the efficiency of the GSHP\textsubscript{s} systems will increase their Seasonal Performance Factor (SPF), thereby leading to a remarkable reduction in the energy consumption and increasing the saving potential of the GHGs. Matching the fluctuated heating load in commercial and residential buildings with the produced energy of the GSHP\textsubscript{s} by using variable capacity control systems is one of the possible methods to approach this improvement.

Two ways, for controlling the capacity of the GSHP\textsubscript{s}, are available and cost-effective: the on/off controlling method and the one that uses variable-speed capacity control technology with an inverter-driven compressor (Zhao et al, 2003). In 2010 Madani et al found out that the SPF of a system using one of these methods depends directly on the percentage of the peak load that a GSHP covers. Thus, if this percentage is lower than 65\%, then using the inverter-driven compressor for controlling the capacity of the GSHP will be more economical than the on/off controlling method; and in this condition the opportunity of reducing the energy consumption is high not only in the newly installed systems but also in the already existing ones by retrofitting them with inverter-driven compressors (Madani et al, 2010a).

Additionally, Madani et al (2010b) conducted an experimental analysis of a variable capacity heat pump equipped with an inverter-driven compressor and found out that the inverter losses increase when the compressor speed increases, despite the fact that it’s percentage of the total compressor power decreases from almost 8\% at compressor speed 30 (RPS) to 4\% at compressor speed 90 (RPS). Moreover, Madani et al (2010b) detected that when the inverter switching frequency is increased the total losses of the compressor and the inverter remain almost constant while the inverter losses themselves increase. Consequently, it has been concluded that there is a trade-off between the inverter losses and the compressor losses (Madani et al, 2010b).

Within this context of the inverter losses, recovering the inverter losses could improve the heat pump efficiency. This gave rise to this experimental analysis in which the performance of a heat pump, equipped with air-cooled inverter, is evaluated at different compressor speeds and inverter switching frequencies; and compared with the performance of the same heat pump when this heat pump is fitted with the same kind of inverter but cooled by liquid.
1.1. Background

The inverter is an electronic device that converts the magnitude and the frequency of an electric voltage. The main task for the inverter cooling system is to maintain the temperature of the inverter below the failure temperature. At junction temperatures of 125 °C, the silicon-based power electronic devices begin to lose reliability and at 150 °C these devices begin to break down (Ayers et al, 2006). Therefore, manufacturers try to push down the maximum temperature of these electronic devices at least to be in the domain of 60-85.

Bhunia et al (2007) conducted an experimental comparison of three thermal management techniques: 1) forced convection air-cooling over a finned heat sink; 2) liquid flow in a multi-pass cold plate; and 3) liquid micro-jet array impingement. In this experimental comparison, Bhunia et al (2007) found that the higher the temperature of the inverter is, the lower the power conversion efficiency. In the same study, Bhunia et al (2007) introduced how in transistor inverters when the device temperature increases the electron mobility decreases while the channel resistance increases, which in turn reduces the maximum collector current and thereby the maximum possible output power. Indeed, the device temperature is governed by the base plate temperature, which in turn is governed by the cooling mechanism and the temperature of the cooling fluid (Bhunia et al, 2007). In a word, a better cooling mechanism and lower base plate temperature of the inverter would enhance the power conversion efficiency and increase the maximum possible output power.

Moreover, reducing the base plate temperature will reduce the junction temperature of the silicon-based devices; and every 10°C reduction in the device junction temperature corresponds to a two-fold improvement in device median life (Bhunia et al, 2007). Equally important, increasing the cooling effectiveness could double the output current of an inverter while using the same quantity of silicon which leads to a significant cost saving in the electronic device (Meysenc et al, 2005).

However, a detailed study about the possibility of using a liquid-cooled inverter to drive the compressor and recovering the inverter losses in a variable capacity heat pump that can be used as baseline for a trade-off study is not available. On the other hand and in order to improve the reliability, increase the power density and reduce the cost of the automotive power electronics, many attempts has been conducted aiming to improve the efficiency of the thermal management techniques for compact power conversion applications.

Micro-jet array impingement cooling is one of the technologies that has been tested, developed and proved a significant improvement in the cooling effectiveness. Specifically, jet-based heat exchanger can provide up to 45 % lower thermal resistance, 79 % increment in power density and 118% increment in specific power with respect to the baseline channel-flow heat exchanger. Unfortunately, this technology has a main drawback that it requires high pumping power as a result of the high-pressure drop across the micro-jet arrays (Narumanchi et al, 2012).
A forced convection liquid microchannels is also an effective cooling technology that can be used for high power density electronic devices. This technology can enhance the maximum output power that the module can deliver and reduce the device temperature; thus increase the reliability and lifetime of the module. Likewise, this technology reduces the non-uniformity of the device temperature. However, Forced convection liquid microchannels technology keeps the pumping power high (Meysenc et al, 2005).

Two-phase heat exchanger with boiling condensing loop is a promising alternative that yields high dissipation rates and increases the heat exchange coefficient without increasing the pressure drop in the cycle; thus maintains low pumping power. Nevertheless, this technology is the least developed compared with other available cooling technologies. (Agostini et al, 2007)

As a matter of fact, forced convection air cooling over a finned heat sink still the most used cooling technique for cooling electronic devices. In spite of the fact that it is the most secure technology, the air-cooling technology has many drawbacks like high thermal resistance, low power density, high junction temperatures, large heat sink and noise in comparison with the above mentioned technologies (Bhunia et al, 2007).
2. Objectives

The primary focus of this project is to recover the heat losses of the inverter by cooling it by the water in the load cycle, or by the brine in the source cycle, in order to improve the total efficiency of the heat pump system. Moreover, the project aims to conduct a comparative analysis of the performance of the system, the compressor losses behavior and the inverter losses behavior at different compressor speeds and inverter switching frequencies when the heat pump is equipped by air-cooled, water-cooled and brine-cooled inverter. Furthermore, the comparative analysis aims also to cover the behavior of the temperature of the base plate in the three aforementioned inverters during the operation under the different kinds of the running conditions.
3. Methodology

In order to achieve the forward mentioned objects, mainly evaluate the possibility of recovering the inverter heat loss and its effect on the heat pump overall performance, a test rig which accomplish three different alternative configurations is built (see fig. 1). In the first two configurations a liquid-cooled inverter is used to drive a hermetic scroll compressor and cooled one time by the water in the load cycle and another time by the brine in the source cycle (see fig. 2 & fig. 3). While in the third configuration an air-cooled inverter is used (see fig. 4).

![Fig. 1. Schematic of the test rig with three different configurations](image)

The test rig can accomplish the first two different configurations by closing and opening the already installed valves without any need for a new construction work. In the three configurations a closed loop has been implemented in which the heat in the source cycle is obtained from the load cycle by a heat exchanger placed after the condenser while the surplus produced heat is stored in the tank (see fig. 1).
When the liquid-cooled inverter is used the main flow of the cooling fluid, in the load and the source cycles, is divided before the inverter cooling cycle by a manual three-way valve. Indeed, in order to get a measurable temperature difference over the inverter cooling cycle, a small portion of the cooling fluid is passed through this cooling cycle (see fig. 1).

On the first hand, the recovered heat losses in the inverter are obtained by measuring the volume flow of the fluid in the inverter cooling cycle and the temperatures at the inlet and the outlet of this cycle. On the other hand, the inverter losses are measured by installing two digital power meters, one before the inverter and another one after it (see fig. 1).

As shown also in fig. 1, different temperature sensors and pressure sensors are installed on the test rig in order to acquire the temperature and the pressure in certain positions; and thereby evaluate and analyze the performance of the heat pump system at different configurations (see fig. 1). The base plate in the liquid-cooled and the air-cooled inverters are fitted with temperature sensors in order to measure the temperature of the base plate of these inverters during the different kinds of running conditions.
4. Experiment set up

4.1. Configurations

In the first alternative configuration, the liquid-cooled inverter is cooled by the water in the load cycle (see fig. 2). In this case the water inter the cooling cycle of the inverter after it leaves the condenser.

![Diagram of test rig](image)

*Fig. 2. Schematic of the test rig when the liquid-cooled inverter is cooled by the water in the load cycle after neglecting the inactive valves and connections*

The other possible configuration is to cool the liquid-cooled inverter by the brine in the source cycle before it enters the evaporator (see fig. 3). In order to ensure the purity of the cooling liquid in the inverter cooling cycle, a faucet is installed for draining this cycle before switching from one configuration to another. In order to facilitate the understanding of these two configurations,
Fig. 2 and fig. 3 have been drawn after neglecting the inactive valves and connections in each configuration.

In addition to the liquid-cooled inverter, the test rig is equipped with an air-cooled inverter and can be run by any one of these inverters by changing the electrical connections. Fig. 4 represents the test rig when it runs by the air-cooled inverter and after neglecting all the inactive valves and connections in this configuration.
4.2. The test rig components

The test facility consists of a heat pump unit, two inverters, two liquid pumps, a storage tank, two plate heat exchangers, two electronic controlled three-way valves, two manual controlled three-way valves and a data acquisition system (see fig. 1). As it can be seen in table 1, the specifications of the main components of the heat pump unit are presented. The heat pump is equipped with a variable speed hermetic scroll compressor, an electronic expansion valve, Power+ inverter and a Combo Drive software installed on µPC programmable board in order to maintain the SIAM compressor working conditions inside the operating envelop specified by the manufacturer (see Appendix A).

The load and source cycles are equipped with two pumps that produce constant flow which could be changed manually by bypass constructions mounted after the pumps. This gives the opportunity of controlling the mass flow in both cycles and makes the test rig flexible. However,
during the operation at each configuration, the mass flow in the load and source cycles should be constant. Thus, the openings of the valves in the bypass constructions remain also constant.

Table 1. The heat pump unit components

<table>
<thead>
<tr>
<th>Component</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>Mitsubishi Electronic group (model. SIAM ANE33FPFMT) 6PH 180Hz 78-400V</td>
</tr>
<tr>
<td>Condenser</td>
<td>Counter-current BPHE (SWEP B25TH*70/1P-SC-M)</td>
</tr>
<tr>
<td>Evaporator</td>
<td>Counter-current BPHE (SWEP QA80H*56/1P-SC-S)</td>
</tr>
<tr>
<td>Combo driver</td>
<td>(CAREL, 2012b)</td>
</tr>
<tr>
<td>Combo controller</td>
<td>µPC Small (UPCB001BS0 - UPCB001DS0), 230 V power supply, embedded valve driver</td>
</tr>
<tr>
<td>Expansion Valve</td>
<td>Unipolar electronic expansion valve (EVD evolution model. E2V24USF10)</td>
</tr>
<tr>
<td>Inverter</td>
<td>Power + (model. PSD0014400/PSD00144A0)</td>
</tr>
<tr>
<td>PGD1 display</td>
<td>pGD1 (PGD1000FW0), panel or wall-mounted + telephone cable</td>
</tr>
<tr>
<td>Pressure sensor</td>
<td>Suction pressure transducer 0-17,3 bar</td>
</tr>
<tr>
<td>Pressure sensor</td>
<td>Discharge pressure transducer .0-5V 0/45 BAR</td>
</tr>
<tr>
<td>Temperature sensor</td>
<td>Discharge temperature probe, 3 m length</td>
</tr>
<tr>
<td>Temperature sensor</td>
<td>Suction temperature probe, 3 m length</td>
</tr>
</tbody>
</table>

As the main object of the experiment is to measure the recovered heat in the inverter cooling cycle, two PT100 sensors (type A) with high precision have been mounted at the inlet and the outlet of the inverter cooling cycle in order to measure the temperature difference over the inverter cooling cycle (see fig. 1). In addition, the cycle is equipped with an electromagnetic flow meter with high precision at very low flow fluid (0.05-1 L/min) in order to measure the volumetric flow in this cycle. Moreover, the air-cooled and liquid-cooled inverters are fitted with two PT100 sensors (type A) on the back side for measuring the temperature of the base plate during the operation.

Furthermore, the test rig is equipped with two YOKOGAWA WT130 digital power meters for measuring the power consumption before and after the inverters. All the signals from the power meters, flow meter, temperature sensors and pressure sensors are received by a data acquisition system. The acquired data are transferred online and can be downloaded simultaneously.

4.3. The liquid-cooled and air-cooled inverters

As mentioned, the test rig is equipped with two different inverters, one of them is cooled by air and the other is designed to be cooled by liquid (see fig. 5 & fig. 6). As shown in fig. 5, the air cooled inverter is fitted with a finned heat sink that absorbs the inverter heat losses and dissipates it to the ambient. In order to ensure a sufficient dissipation of this heat, the inverter is fitted with a fan that produces sufficient air flow (CAREL, 2012a).

Fig. 5. The air-cooled inverter (CAREL, 2012a)
As seen in fig. 6 and fig. 7, the liquid cooled inverter has been delivered without any cooling heat exchanger. Therefore, a cooling heat exchanger has been designed and manufactured locally for the purpose of this experiment. However, the heat exchanger is not optimized from the economical point of view but from the practical and technical ones that satisfy the testing conditions and ensure the capability of measuring the recovered heat in the inverter.

As shown in fig. 8, the designed heat exchanger consists of two aluminum plates and a copper pipes structure. The aluminum plates works as a heat sink for the inverter dissipated heat and transfers this heat to the copper pipes. After that, the dissipated heat is transferred to the cooling fluid inside the copper pipes. Before running any test, the heat

**Fig. 6. The liquid-cooled inverter (CAREL, 2012a)**

**Fig. 7. The liquid-cooled inverter as it delivered**

**Fig. 8. An assembly of the inverter heat exchanger**
exchanger is well insulated in order to prevent any heat dissipation to the ambient.

It is also worth mentioning that the heat exchanger design takes into account that the hottest spot of the base plate of the inverter that should be cooled is concentrated in the middle of the inverter (see fig. 6) (CAREL, 2012a). Furthermore, according to the manufacturer, the temperature of the base plate should not exceed 70°C during the operation; and the cooling system should not cause any condensation on the internal plate surfaces of the inverter (CAREL, 2012a).

Fig. 9 shows the liquid-cooled inverter after it has been fitted with the heat exchanger and installed in the test rig. A picture of the test rig including the heat pump unit, data acquisition system, pumps and power measurement system is shown in fig. 10.
4.4. Testing procedures and measured parameters

The system is tested with the air-cooled, water-cooled and brine-cooled inverters in sequence for the same set-point source/load side temperatures (5/35). For each inverter, the compressor rotation speed is changed by the inverter in the range of 20 Rotation Per Second (RPS) to 110 (RPS) by increasing the compressor speed 10 (RPS) each step (see table 2). All the inverters have three different switching frequencies (4 kHz, 6 kHz, and 8 kHz) and the test is repeated at all of them and in the same conditions (see table 2). This switching frequency is the frequency of closing and opening the transistors in the inverter in order to create sine-like current waves at the output of the inverter.

While the compressor speed is changing during the test, the water and the brine mass flow is held constant, as well as, the temperature of the water at the inlet of the condenser ($T_{w,con,in}$) in the three different configurations. This temperature ($T_{w,con,in}$) represents the load temperature and equals to 35 °C (see table 2).

Meanwhile, the temperature of the brine at the inlet of the evaporator ($T_{br,eva,in}$) is held constant when the water-cooled and the air-cooled inverters are used in the first two configurations; and in this case this temperature ($T_{br,eva,in}$) represents the source temperature and equals to 5°C (see fig. 2 and fig. 4). In contrast, when the brine-cooled inverter is used in the third configuration the test rig maintains constant temperature of the brine at the inlet of the cooling cycle of the inverter ($T_{br,inv,in}$) while the temperature of the brine at the inlet of the evaporator fluctuates slightly above 5 °C depending on the recovered heat in the cooling cycle of the inverter (see fig. 3). Thus, the source temperature in the last configuration is represented by the brine temperature at the inlet of the cooling cycle of the inverter ($T_{br,inv,in}$) and also equals to 5 °C.

Table 2. The different test points for the test rig when the air-cooled, water-cooled and brine-cooled inverters are used at three different switching frequencies and ten successive compressor speeds

<table>
<thead>
<tr>
<th>Cooling types</th>
<th>Water-cooled inverter</th>
<th>Brine-cooled inverter</th>
<th>Air-cooled inverter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water inlet temperature</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brine inlet temperature</td>
<td>5</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Switching frequency (kHz)</td>
<td>4 kHz</td>
<td>6 kHz</td>
<td>8 kHz</td>
</tr>
<tr>
<td>Rotation speed (RPS)</td>
<td>20</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>30</td>
<td></td>
<td></td>
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<td>40</td>
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<td>50</td>
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<td>60</td>
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<td>70</td>
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<td>80</td>
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<td>90</td>
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<tr>
<td></td>
<td>100</td>
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</tr>
<tr>
<td></td>
<td>110</td>
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</tbody>
</table>
In order to maintain the above mentioned running conditions, both of them the load and source cycles are equipped with two electronic-controlled three-way valves which leads to a more stable and reliable performance of the test rig and reduce the consumption of the tap water.

During the test the following parameters are measured (see fig. 1):

- The volumetric flow in the inverter cooling cycle
- The temperatures at the inlet and the outlet of the inverter cooling cycle
- The temperature of the base plate on the back side of the inverters
- The power before and after the inverter
- The condensation and evaporation pressure of the refrigerant
- The temperatures of the water at the inlet and the outlet of the condenser
- The temperatures of the brine at the inlet and the outlet of the evaporator
- The temperatures of the refrigerant at the inlet of the compressor and the expansion valve; the temperatures of the refrigerant at the outlet of the compressor, the condenser and the evaporator (see fig. 1)

All the other needed parameters such as enthalpies, mass flow, superheat, and sub cooling temperatures are calculated based on the measured parameters or by using the REFPROP software.

4.5. Measurements uncertainty, Chauvenet’s criterion and temperature sensors and power meters calibration

In any experiment, the reliability and the validity of the acquired data should be determined. Therefore, an uncertainty analysis for the targeted measurements is of high importance even at the early stage of designing the test rig (see Appendix C). Accordingly, detailed uncertainty analyses for the inverter losses and the recovered energy measurements are done at the designing stage of the test rig. This gave the possibility of determining the most important parameters, which are used for calculating the inverter losses and the recovered energy, from the uncertainty point of view. Indeed, facilitated the choosing process of the measurement equipment by building it on the base of a very solid uncertainty analysis from the very beginning.

As result, the uncertainty analysis of the inverter losses measurements highlighted the importance of the power meters precision. Consequently, two digital power meters of reasonable quality (YOKOGAWA WT130) are chosen and calibrated directly before running the test (see Appendix C.IV C.IV). On the other hand, the uncertainty analysis of measuring the recovered heat in the inverter showed clearly the importance of using very precise flow meter and temperature sensors (see Appendix C.V).
Despite the fact that the temperature sensors are chosen of reasonable quality (PT100, Type A), the two temperature sensors that are used for measuring the temperatures at the inlet and the outlet of the cooling cycle have been calibrated to each other’s in order to enhance the measuring precision (see Appendix C.V.2)

In the uncertainty analysis of both of them, the inverter losses and the recovered heat measurements, the uncertainty of type A and type B are taken into account (see Appendix C.I). Moreover and in both of them also, Chauvenet’s criterion is used for data reduction (see Appendix C.III).
5. Results

5.1. The inverter losses behavior

5.1.1. The measured inverter losses and its estimated uncertainty in the brine-cooled, water-cooled and Air-cooled inverters at three different inverter switching frequencies

Fig. 11, fig. 12 and fig. 13 show the losses behavior of each of the air-cooled, water-cooled and brine-cooled inverters respectively when the compressor rotation speed increases from 20 to 110 (RPS) at the three different switching frequencies (4 kHz, 6 kHz and 8 kHz) of the inverters. Furthermore, these figures represent the uncertainty of the measured inverter losses with a confidence level of 95 % (see Appendix C.IV). The vertical error bars indicate the amount of uncertainty associated with each point in these three figures.

A. Air-cooled inverter

Fig. 11 shows that the losses of the air-cooled inverter at 4 (kHz) switching frequency of the inverter increase from about 74 at 20 (RPS) compressor rotation speed to about 147 (W) at 110 (RPS) compressor speed. Moreover, at the same rotation speed these losses increase between 15-26 (W) by increasing the switching frequency depending on the rotation speed and the switching frequency. Meanwhile, the uncertainty of the inverter losses measurements increases from about 0.6 (W) to about 4 (W) in parallel with the compressor speed increasing from 20 (RPS) to 110 (RPS) with some exceptions at low speeds, like 20, 30 and 40 (RPS), and certain switching frequencies where the uncertainty is relatively high (see Appendix C.IV, Table 3).

![Air-cooled inverter graph](image)

*Fig. 11. The measured losses of the air-cooled inverter $P_{\text{inv,loss}}$ (W) and its estimated uncertainty with a confidence level of 95 (%) versus compressor rotation speed $N$ (RPS) when the switching frequency of the inverter is 4, 6 and 8 kHz. (see Appendix C.IV, Table 3)*
B. Water-cooled inverter

As seen in fig. 12 the water-cooled inverter losses at 4 (kHz) switching frequency of the inverter increase from about 71 at 20 (RPS) compressor rotation speed to about 149 (W) at 110 (RPS) compressor speed. In addition, when the switching frequency of the inverter increases at the same rotation speed, the water-cooled inverter losses increase in the range of 13-20 (W) depending on the compressor speed and the switching frequency. So, in the water-cooled inverter, the higher the speed of the compressor or the switching frequency of the inverter is, the higher the inverter losses. Also, fig. 12 shows that the uncertainty of the inverter losses measurements increases from about 0.4 (W) to about 3.5 (W) in parallel with the compressor speed increasing from 20 (RPS) to 110 (RPS) (see Appendix C.IV, Table 4).

Water-cooled inverter

![Graph](image)

* Fig. 12. The measured losses of the water-cooled inverter $P_{inv,loss}$ (W) and its estimated uncertainty with a confidence level of 95 (%) versus compressor rotation speed $N$ (RPS) when the switching frequency of the inverter is 4, 6 and 8 kHz. (see Appendix C.IV, Table 4)

C. Brine-cooled inverter

In the brine-cooled inverter the losses increase also when the compressor speed increases. As example, at 4 (kHz) switching frequency of the inverter the losses of the inverter increase from about 68 to about 141 (W) when the compressor rotation speed increases from 20 to 110 (RPS). While the increment in the losses, according to the inverter switching frequency increasing, varies between 11 and 17 (W). On the other hand, the uncertainty of the inverter losses measurements increases gradually from about 0.5 (W) to 3.4 (W) in parallel with the compressor speed increasing from 20 (RPS) to 110 (RPS) except at rotation speed 50 (RPS) where the uncertainty has a relative high value (See fig. 13) (see Appendix C.IV, Table 5).
5.1.2. The measured inverter losses in the brine-cooled, water-cooled and Air-cooled inverters at the same switching frequencies of the inverter

Fig. 14, fig. 15 and fig. 16 represent the losses behavior of the air-cooled, water-cooled and brine-cooled inverters when the compressor rotation speed increases from 20 to 110 (RPS) at the three different switching frequencies of the inverters 4, 6 and 8 (kHz) respectively.

As shown in these three figures, almost at each rotation speed of the compressor the air-cooled inverter has the highest losses while the brine-cooled inverter has the lowest one; equally important, this difference in the inverters losses at each compressor speed increases by increasing the switching frequency (see fig. 14, fig. 15 and fig. 16).
5.2. The recovered heat losses behavior

5.2.1. The recovered heat losses and the estimated uncertainty

Fig. 17 and fig. 18 show the recovered heat behavior in the water-cooled and brine-cooled inverters respectively when the compressor rotation speed increases from 20 to 110 (RPS) and the switching frequency of the inverters varies between 4 kHz, 6 kHz and 8 kHz. Furthermore, these figures represent the total uncertainty of the calculated recovered heat according to the measured parameters with a confidence level of 95% (see Appendix C.V). The measured parameters are the volumetric flow of the fluid in the inverter cooling cycle and the fluid temperatures at the inlet and the outlet of this cycle. The vertical error bars indicate the amount of uncertainty associated with each point in these two figures.

**A. The recovered heat in the water-cooled inverter and the estimated uncertainty at the three different switching frequencies**

Fig. 17 shows how the recovered heat and its estimated uncertainty in the water-cooled inverter increase gradually as a result of the compressor rotation speed or the inverter switching frequency increasing. At 4 (kHz) switching frequency of the inverter, this recovered heat increases from about 45 (W) at 20 (RPS) compressor speed and to about 94 (W) at 110 (RPS) compressor speed (see fig. 17). Increasing the inverter switching frequency leads to an increment in the recovered heat in the range of 10 to 17 (W). Equally important, fig. 17 shows how the estimated uncertainty
of the recovered heat in the water-cooled inverter increases from about 1.9 (W) at 20 (RPS) compressor speed and 4 (kHz) switching frequency of the inverter to about 5 (W) at 110 (RPS) compressor speed and 8 (kHz) switching frequency of the inverter (see appendix D, Table 6).

**Water-cooled inverter**

* Switching frequency 4 KHz  * Switching frequency 6 KHz  * Switching frequency 8 KHz

![Graph showing recovered heat and uncertainty vs. compressor rotation speed](image)

*Fig. 17. The recovered heat $Q_{inv}$ (W) and its estimated uncertainty with a confidence level of 95 (%) in the water-cooled inverter versus compressor rotation speed $N$ (RPS) when the switching frequency of the inverter is 4, 6 and 8 kHz (see Appendix D, Table 6)*

**B. The recovered heat losses in the Brine-cooled inverter at the three different switching frequencies**

As shown in fig. 18, the recovered heat and its estimated uncertainty in the brine-cooled inverter also increase gradually as a result of the compressor speed or the inverter switching frequency increasing. In the brine cooled-inverter, the recovered heat is higher than the one in the water-cooled inverter. At 4 (kHz) switching frequency of the inverter, this recovered heat starts from about 62 (W) at 20 (RPS) compressor speed to about 114 (W) at 110 (RPS) compressor speed (see fig. 18). As in the water-cooled inverter, increasing the switching frequency of the brine-cooled inverter will increase the recovered heat by 9 to 15 (W) (see Appendix D, Table 7).

Meanwhile, the estimated uncertainty of the recovered heat in the brine-cooled inverter is higher in comparison with the water-cooled one. This uncertainty increases from about 2.9 (W) at 20 (RPS) compressor rotation speed and 4 (kHz) switching frequency of the inverter to about 6.4 (W) at 110 (RPS) compressor rotation speed and 8 (kHz) switching frequency of the inverter. (see Appendix D, Table 7)
5.2.2. A comparison between the recovered heat losses in the brine-cooled and water-cooled inverters at the different switching frequencies

In order to simplify the comparison between the two different liquid cooling methods, three addition diagrams have been implemented (see fig. 19, fig. 20 and fig. 21). In each diagram the recovered heat in the water-cooled and brine-cooled inverters has represented at the same inverter switching frequency. As shown in these diagrams, the recovered heat is higher when the inverter is cooled by the brine at all the compressor speeds and all the inverter switching frequencies. Moreover, fig. 19 shows how the difference between the recovered heat in the brine-cooled inverter and the recovered heat in the water-cooled inverter, at 4 (kHz) switching frequency, varies between about 14 to about 21 (W) depending on the compressor speed, which represents the highest difference at the three

![Diagram showing the recovered heat in the brine-cooled inverter versus compressor rotation speed at different switching frequencies.]

**Fig. 18.** The recovered heat $Q_{\text{inv}}$ (W) and its estimated uncertainty with a confidence level of 95 (%) in the brine-cooled inverter versus compressor rotation speed $N$ (RPS) when the switching frequency of the inverter is 4, 6 and 8 kHz (see Appendix D, Table 7)

![Diagram comparing the recovered heat in the brine-cooled and water-cooled inverters at 4 kHz switching frequency.]

**Fig. 19.** The recovered heat $Q_{\text{inv}}$ (W) of the brine-cooled and water-cooled inverters versus compressor rotation speed $N$ (RPS) at switching frequency 4 (kHz) of the inverter
different switching frequency. On the contrary, fig. 21 highlights the fact that at switching frequency 8 kHz of the inverters the difference in the recovered heat in the water-cooled and the brine-cooled inverters is the lowest between the differences at the three different switching frequencies; and varies from about 11 to 16 (w) depending on the compressor speed. Finally, this difference varies between 13 and 20 (W) depending on the compressor speed at inverter switching frequency of 6 (kHz) (see fig. 20).

![Switching frequency 6 kHz](image1)

![Switching frequency 8 kHz](image2)

**Fig. 20.** The recovered heat $Q_{inv}$ (w) of the brine-cooled and water-cooled inverters versus compressor rotation speed $N$ (RPS) at switching frequency 6 (kHz) of the inverter

**Fig. 21.** The recovered heat $Q_{inv}$ (w) of the brine-cooled and water-cooled inverters versus compressor rotation speed $N$ (RPS) at switching frequency 8 (kHz) of the inverter

### 5.2.3. The recovered heat as ratio of the inverter losses

In order to get better idea about which is the most effective recovering method of the inverter heat losses (brine or water cooling), the recovered heat is related to the losses in the inverters and illustrated in fig. 22 when the compressor speed is increased from 20 to 110 (RPS) and at the three different switching frequencies 4, 6 and 8 (kHz). As it can be easily concluded from fig. 22-a, just about 63 to 69 (%) of the losses in the water-cooled inverter is recovered depending on the compressor speed and the inverter switching frequency. While, the recovered amount of the heat losses in the brine-cooled inverter under the same conditions of compressor speed and inverter switching frequency varies between 79 to 90 (%) (see fig. 22-b). Moreover, in the water-cooled inverter the lower the inverter switching frequency is, the lower the recovered percentage of the inverter losses (see fig. 22). In contrast, in the brine-cooled inverter at 4 (kHz) switching frequency the recovered percentage of the inverter losses is relatively higher than the recovered one at the 6 and 8 (kHz) switching frequency.
Fig. 22. The recovered heat $Q_{\text{inv}}$ as percentage (%) of the inverter losses $P_{\text{inv,loss}}$ in the water-cooled inverter (a) and the brine-cooled one (b) versus compressor rotation speed $N$ (RPS) when the switching frequency of the inverter is 4, 6 and 8 kHz.

5.3. The inverter losses and the recovered heat related to the inverter input power

Fig. 23, fig. 24 and fig. 25 represent the losses and the recovered heat in the inverters as percentage of the inverter input power. As it can be observed from these three diagrams, the ratio of the inverter losses to the inverter input power and also the ratio of the recovered heat to the inverter input power decrease when the compressor rotation speed increases or when the inverter switching frequency decreases (see Appendix B).

5.3.1 Air-cooled inverter

As it can be observed from fig. 23, the ratio of the losses to the input power of the air-cooled inverter decreases when the compressor rotation speed increases or when the inverter switching frequency decreases. However, at all the switching frequencies this ratio decreases obviously when the compressor speed increases from 20 to 60 (RPS). On the contrary, the effect of the compressor speed reduction on this ratio is much lower when the compressor speed decreases.
increases from 60 to 110 (RPS). As example, at 4 (kHz) switching frequency of the inverter this ratio decreases from about 9.55 (%) to about 3.46 (%) when the compressor speed increases from 20 to 60 (RPS); while when the compressor speed increases from 60 to 120 (RPS) this ratio decreases just from about 3.46 (%) to about 2.67 (%). On the other hand, at the same compressor speed the ratio of the losses to the input power of the air-cooled inverter increases by about 0.2 to 1.8 (%) when the inverter switching frequency increases (see fig. 23).

5.3.2 Water-cooled inverter

Fig. 24 shows how the ratios of the losses to the inverter input power and the recovered heat to the inverter input power in the water-cooled inverter decrease by increasing the compressor rotation speed or decreasing the inverter switching frequency. Both ratios, decrease obviously when the speed of the compressor increases from 20 to 60 (RPS) but both of them are not much affected by changing the compressor speed when the speed is higher than 60 (RPS) (see fig. 24).

![Diagram showing the inverter losses and recovered heat in water-cooled inverter](image)

*Fig. 24. The inverter losses $P_{inv,loss}$ (a) and the recovered heat (b) as percentage (%) of the inverter input power $P_{inv,in}$ versus compressor rotation speed $N$ (RPS) when the inverter is cooled by water and the switching frequency of the inverter is 4, 6 and 8 kHz.*

At 4 (kHz) inverter switching frequency when the compressor speed increases from 20 to 60 (RPS), the ratio of the losses to the input power of the water-cooled inverter decreases from about 9.15 (%) to about 3.34 (%) (see fig. 24-a); While the ratio of the recovered heat to the inverter input power decreases from about 5.78 (%) to about 2.19 (%) (see fig. 24-b). On the other hand, When the compressor speed increases from 60 to 110 (RPS) and at 4 (kHz) inverter switching frequency, the ratio of the losses to the input power of the water-cooled inverter decreases from about 3.34 (%) to about 2.28 (%) (see fig. 24-a); While the ratio of the recovered heat to the inverter input power decreases from about 2.19 (%) to about 1.43 (%) (see fig. 24-b).
5.3.3 Brine-cooled inverter

As in the water-cooled inverter, the ratios of the losses to the inverter input power and the recovered heat to the inverter input power in the brine-cooled inverter decrease by increasing the compressor rotation speed or decreasing the inverter switching frequency. Both ratios, decrease obviously when the speed of the compressor increases from 20 to 60 (RPS) but both of them are not much affected by changing the compressor speed when the speed increases from 60 to 110 (RPS) (see fig. 25).

![Graphs showing the ratios of losses and recovered heat to the inverter input power versus compressor rotation speed for brine-cooled inverter with different inverter switching frequencies.](image)

**Fig. 25.** The inverter losses $P_{\text{inv},\text{loss}}$ (a) and the recovered heat (b) as percentage (%) of the inverter input power $P_{\text{inv,in}}$ versus compressor rotation speed $N$ (RPS) when the inverter is cooled by brine and the switching frequency of the inverter is 4, 6 and 8 kHz

For example and at 4 (kHz) inverter switching frequency when the compressor speed increases from 20 to 60 (RPS), the ratio of the losses to the input power of the brine-cooled inverter decreases from about 8.79 (%) to about 3.13 (%) (see fig. 25-a); While the ratio of the recovered heat to the inverter input power decreases from about 7.93 (%) to about 2.8 (%) (see fig. 25-b). On the other hand, When the compressor speed increases from 60 to 110 (RPS) and at 4 (kHz) inverter switching frequency, the ratio of the losses to the input power of the water-cooled inverter decreases from about 3.13 (%) to about 2.16 (%) (see fig. 25-a); While the ratio of the recovered heat to the inverter input power decreases from about 2.8 (%) to about 1.74 (%) (see fig. 25-b).

Indeed, at the same compressor speed and when the inverter switching frequency increases, the ratio of the losses to the input power of the brine-cooled inverter increases by about 0.25 to 1.5 (%) (see fig. 25-a); while the ratio of the recovered heat to the input power of the brine-cooled inverter increases by about 0.2 to 1 (%) (see fig. 25-b)
5.4. The heat pump unit performance

5.4.1. The system coefficient of performance

Fig. 26, fig. 27 and fig. 28 represent the system coefficient of performance \( \text{COP}_{sys} \) versus compressor rotation speed at each inverter switching frequency when the heat pump is equipped with air-cooled, water-cooled and brine-cooled inverters. The \( \text{COP}_{sys} \) is defined as the ratio of the heat capacity \( Q_{\text{cond}} \) to the inverter input power \( P_{\text{inv, in}} \) when the heat pump is equipped with the air-cooled or the brine-cooled inverter. In contrast, when the heat pump is equipped with the water-cooled inverter the \( \text{COP}_{sys} \) is defined as the ratio of the heat capacity \( Q_{\text{cond}} \) plus the recovered heat in the inverter \( Q_{\text{inv}} \) to the inverter input power \( P_{\text{inv, in}} \). It is worth mentioning that the extra added pumping power due to the pressure drop in the inverter cooling cycle is too low and neglected because the velocity of the fluid in the inverter cooling cycle is too low (see Appendix B). The \( \text{COP}_{sys} \) has the same decreasing trend if the compressor speed increases at all the inverter switching frequencies and when the inverter is cooled by air, water or brine except at speeds from 100 to 110 (RPS) where the \( \text{COP}_{sys} \) remains almost constant for all the inverter cooling methods (see fig. 26, fig. 27 and fig. 28).

A. Switching frequency 4 kHz

As it can be observed from fig. 26, the \( \text{COP}_{sys} \) has almost the same value at all the compressor rotation speeds when the inverter is cooled by water or brine and the inverter switching frequency is 4 (kHz); and decreases from about 5.96 to 4.24 when the compressor speed increases from 20 to 110 (RPS).

![Switching frequency 4kHz](image)

**Fig. 26.** The system coefficient of performance \( \text{COP}_{sys} \) versus compressor rotation speed \( N \) (RPS) when the heat pump is equipped with air-cooled, water-cooled and brine-cooled inverters; and the switching frequency of the inverter is 4 kHz.
On the other hand and at the same inverter switching frequency, the COP\textsubscript{sys} has lower value when the used inverter is cooled by air and decreases from 5.8 to 4.2 if the compressor speed is increased from 20 to 110 (RPS) (see fig. 26).

**B. Switching frequency 6 kHz**

At inverter switching frequency of 6 (kHz), the COP\textsubscript{sys} has almost the same value at almost all the compressor rotation speeds when the inverter is cooled by water or brine and decreases from about 5.9 to 4.23 when the compressor rotation speed increases from 20 to 110 (RPS) (see fig. 27). Also, when the heat pump is equipped with air-cooled inverter and at inverter switching frequency of 6 (kHz) the COP\textsubscript{sys} is lower than the one when the inverter is cooled by water or brine at all the compressor speeds and decreases from 5.72 to 4.2 when the compressor rotation speed increases from 20 to 110 (RPS) (see fig. 27).

**Switching frequency 6kHz**

![Switching frequency 6kHz graph](image)

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*Fig. 27. The system coefficient of performance COP\textsubscript{sys} versus compressor rotation speed N (RPS) when the heat pump is equipped with air-cooled, water-cooled and brine-cooled inverters; and the switching frequency of the inverter is 6 kHz.*

**C. Switching frequency 8 kHz**

When the compressor speed increases from 20 to 110 (RPS) and at inverter switching frequency of 8 kHz, the COP\textsubscript{sys} has almost the same value when the heat pump is equipped with water-cooled or brine-cooled inverter and decreases from 5.8 to 4.24 (see fig. 28). Meanwhile, when the inverter is cooled by air the COP\textsubscript{sys} decreases from 5.66 to 4.2 if the compressor speed increases from 20 to 110 (RPS) but stay lower than the COP\textsubscript{sys} when the used inverter is cooled by water or brine at all the compressor speeds (see fig. 28). It is also worthy of notice that at some compressor speeds the COP\textsubscript{sys} is slightly higher when the heat pump is equipped with the brine-cooled inverter than the one when it is equipped with the water-cooled inverter (see fig. 28).
Fig. 28. The system coefficient of performance $\text{COP}_{\text{sys}}$ versus compressor rotation speed $N$ (RPS) when the heat pump is equipped with air-cooled, water-cooled and brine-cooled inverters; and the switching frequency of the inverter is 8 kHz.

5.4.2. The total isentropic efficiency of the compressor

Fig. 29, fig. 30, fig. 31, fig. 32, fig. 33 and fig. 34 introduce the total isentropic efficiency of the compressor $\eta_{\text{is}}$ versus compressor rotation speed when the heat pump is equipped with the air-cooled, water-cooled and brine-cooled inverters and at different inverter switching frequencies. The total isentropic efficiency of the compressor $\eta_{\text{is}}$ is defined as the ratio of the consumed power in the compressor at assumed isentropic compression process to the actual consumed one (see Appendix B).

A. Air-cooled inverter

As seen in fig. 29, the total isentropic efficiency of the compressor $\eta_{\text{is}}$ when the heat pump is equipped with the air-cooled inverter decreases from about 0.85 at 30 (RPS) compressor speed to about 0.8 at 100 (RPS) compressor speed; Meanwhile, when the compressor speed increases from 20 to 30 (RPS) the total total isentropic efficiency of the compressor decreases obviously from about 0.9 to about 0.85.On the contrary, by increasing the compressor rotation speed from 100 to 110 (RPS) the total total isentropic efficiency of the compressor $\eta_{\text{is}}$ increases from about 0.8 to about 0.83 (see fig. 29). Moreover, in general at each compressor rotation speed the total total isentropic efficiency of the compressor increases slightly by increasing the inverter switching frequency.
Air-cooled inverter

![Graph showing isentropic efficiency versus compressor rotation speed]

Fig. 29. The total isentropic efficiency of the compressor $\eta_{is}$ versus compressor rotation speed $N$ (RPS) when the inverter is cooled by air and the switching frequency of the inverter is 4, 6 and 8 kHz

B. Water-cooled inverter

When the heat pump is equipped with the water-cooled inverter the total isentropic efficiency of the compressor $\eta_{is}$ decreases from about 0.86 to about 0.81 when the compressor speed increases from 30 to 100 (RPS) (see fig. 30). In addition, the total isentropic efficiency of the compressor decreases obviously from about 0.91 to about 0.86 when the compressor speed increases from 20 to 30 (RPS). On the contrary, when the compressor speed increases from 100 to 110 (RPS) the total isentropic efficiency of the compressor $\eta_{is}$ increases from about 0.81 to about 0.83 (see fig.30). Also, at the same compressor speed the total isentropic efficiency of the compressor increases slightly by increasing the inverter switching frequency(see fig. 30).

C. Brine-cooled inverter

As in the other cases, when the heat pump is equipped with the brine-cooled inverter the total isentropic efficiency of the compressor $\eta_{is}$ decreases from about 0.86 to about 0.81 when the compressor speed increases from 30 to 100 (RPS) (see fig. 31). Additionally, when the compressor speed increases from 20 to 30 (RPS) the total isentropic efficiency of the compressor decreases from about 0.92 to about 0.86. on the other hand, by increasing the compressor rotation speed from 100 to 110 (RPS), the total isentropic efficiency of the compressor $\eta_{is}$ increases from about 0.81 to about 0.835 (see fig. 31). Furthermore, as it could be observed in fig. 31 at the same compressor rotation speed the total isentropic efficiency of the compressor $\eta_{is}$ increases slightly by increasing the inverter switching frequency.
Fig. 30. The total isentropic efficiency of the compressor $\eta_t$ versus compressor rotation speed $N$ (RPS) when the inverter is cooled by water and the switching frequency of the inverter is 4, 6 and 8 kHz.

Fig. 31. The total isentropic efficiency of the compressor $\eta_t$ versus compressor rotation speed $N$ (RPS) when the inverter is cooled by brine and the switching frequency of the inverter is 4, 6 and 8 kHz.
D. The total isentropic efficiency of the compressor $\eta_{is}$ at the same inverter switching frequency when the heat pump is equipped with the brine, water and Air-cooled inverters

In order to facilitate the comparison between the three different cooling methods of the inverter from the total isentropic efficiency of the compressor $\eta_{is}$ point of view, the diagrams fig. 32, fig. 33 and fig. 34 are introduced. In each of these diagrams, the total isentropic efficiency of the compressor $\eta_{is}$ versus compressor rotation speed is illustrated at the same inverter switching frequency when the heat pump is equipped with the three different inverters.

As it can be observed in fig. 32, fig. 33 and fig. 34 at almost each compressor rotation speed the total isentropic efficiency of the compressor $\eta_{is}$ has the lowest value when the air-cooled inverter is used while it has the highest value when the brine-cooled inverter is used. Furthermore, the difference in the total isentropic efficiency of the compressor $\eta_{is}$ when the three different inverters are used is very low at compressor speeds in the range of 50 to 70 (RPS) while this difference increases at speeds lower and higher than this range (see fig. 32, fig. 33 & fig. 34). Equally important, the higher the inverter switching frequency is the higher the difference in the total isentropic efficiency of the compressor $\eta_{is}$ when the different cooling methods of the inverter are used (see fig. 32, fig. 33 & fig. 34).

**Switching frequency 4kHz**

![Diagram showing isentropic efficiency versus compressor rotation speed with different inverter cooling methods](image)

*Fig. 32. The total isentropic efficiency of the compressor $\eta_{is}$ versus compressor rotation speed $N$ (RPS) when the inverter is cooled by water, air and brine; and the switching frequency of the inverter is 4 kHz.*
Switching frequency 6kHz

![Graph showing the total isentropic efficiency of the compressor \( \eta_s \) versus compressor rotation speed \( N \) (RPS) when the inverter is cooled by water, air and brine; and the switching frequency of the inverter is 6 kHz.]

**Fig. 33.** The total isentropic efficiency of the compressor \( \eta_s \) versus compressor rotation speed \( N \) (RPS) when the inverter is cooled by water, air and brine; and the switching frequency of the inverter is 6 kHz.

Switching frequency 8kHz

![Graph showing the total isentropic efficiency of the compressor \( \eta_s \) versus compressor rotation speed \( N \) (RPS) when the inverter is cooled by water, air and brine; and the switching frequency of the inverter is 8 kHz.]

**Fig. 34.** The total isentropic efficiency of the compressor \( \eta_s \) versus compressor rotation speed \( N \) (RPS) when the inverter is cooled by water, air and brine; and the switching frequency of the inverter is 8 kHz.

5.4.3. The fluctuation in the measured power before and after the inverter

The unexpected behavior of the coefficient of performance of the system COP_{sys} and the total isentropic efficiency of the compressor \( \eta_s \) at compressor speeds higher than 100 (RPS) could be attributed to the considerable reduction in the fluctuation of the power at the input and the output.
of the inverter, which in its turn could affect the compressor performance (see fig. 35). Fig. 35 shows One hundred continuous measurements of the power before and after the brine-cooled inverter at 6 (kHz) switching frequency when the compressor speed is 100 (RPS) (see fig. 35-a) and when the speed is 110 (RPS) (see fig. 35-b) during the same specific time. It can be easily observed how the fluctuation magnitude in the measured power before and after the inverter is little more than 300 (W) when the compressor speed is 100 (RPS) (see fig. 35-a); while when the compressor speed is 110 (RPS) this magnitude decreases to about 100 (W) (see fig. 35-b).

![The inverter power input and output at 100 (RPS)](image1)

![The inverter power input and output at 110 (RPS)](image2)

*Fig. 35. One hundred continuous measurements of the power before and after the brine-cooled inverter at 6 kHz switching frequency when the compressor rotation speed is 100 (RPS) (a) and when the compressor speed is 110 (RPS) (b) during the same specific time*

5.4.4. The Carnot efficiency of the system

As the losses and the recovered heat in the inverter are relatively low especially at high compressor rotation speed, any fluctuation in the measured temperatures could affect the calculated COP$_{sys}$. During the test, the maximum acceptable fluctuation in the temperatures was 0.3 °C. So in order to be sure and avoid any misleading result caused by an overlapped effect of the refrigerant evaporation and condensation temperatures fluctuation, the Carnot efficiency of the system $\eta_{Cd}$ is introduced which is defined as the ratio of the coefficient of performance of the system COP$_{sys}$ to the Carnot coefficient of performance COP$_c$.

As it has been mentioned before, The COP$_{sys}$ is defined as the ratio of the heat capacity $Q_{cond}$ to the inverter input power $P_{inv,in}$ when the heat pump is equipped with the air-cooled or the brine-cooled inverter. In contrast, when the heat pump is equipped with the water-cooled inverter the COP$_{sys}$ is defined as the ratio of the heat capacity $Q_{cond}$ plus the recovered heat in the inverter $Q_{inv}$ to the inverter input power $P_{inv,in}$. On the other hand, the Carnot coefficient of performance COP$_c$ is defined as the ratio of the refrigerant condensation temperature $T_1$ (K) to the difference between the refrigerant condensation and evaporation temperatures $T_1-T_2$ (K) (see Appendix B).
Fig. 36, fig. 37 and fig. 38 represent the Carnot efficiency of the system $\eta_{Cd}$ versus compressor rotation speed when the heat pump is fitted with the air-cooled, water-cooled and brine-cooled inverters at specific inverter switching frequency. As it could be seen in these three figures, at all the rotation speeds of the compressor the Carnot efficiency of the system $\eta_{Cd}$ has the lowest value when the heat pump uses the air-cooled inverter. In addition, the lower the compressor speed is, the higher the difference between the Carnot efficiency of the system $\eta_{Cd}$ when the heat pump is fitted with air-cooled inverter on the one hand and the Carnot efficiency of the system when the heat pump is fitted with water-cooled or brine-cooled inverter on the other hand (see fig. 36, fig. 37 and fig. 38); Moreover, the higher is the inverter switching frequency, the higher the difference.

**Switching frequency 4 kHz**

Fig. 36. Carnot efficiency of the system $\eta_{Cd}$ versus compressor rotation speed $N$ (RPS) when the heat pump is equipped with air-cooled, water-cooled and brine-cooled inverter; and the switching frequency of the inverter is 4 kHz.

**Switching frequency 6 kHz**

Fig. 37. Carnot efficiency of the system $\eta_{Cd}$ versus compressor speed $N$ (RPS) when the heat pump is equipped with air-cooled, water-cooled and brine-cooled inverter; and the switching frequency of the inverter is 6 kHz.

**Switching frequency 8 kHz**

Fig. 38. Carnot efficiency of the system $\eta_{Cd}$ versus compressor speed $N$ (RPS) when the heat pump is equipped with air-cooled, water-cooled and brine-cooled inverter; and the switching frequency of the inverter is 8 kHz.
As it can be seen obviously in fig. 36, fig. 37 and fig. 38, when the compressor speed increases from 30 to 100 (RPS) Carnot efficiency of the system $\eta_{Cd}$ decreases smoothly but from 20 to 30 (RPS) compressor speed it decreases obviously; on the contrary, when the compressor speed increases from 100 to 110 (RPS) this efficiency increases unexpectedly.

5.4.5. The total losses of the compressor and the inverter

A. The total losses of the compressor and the inverter at different switching frequency when the heat pump is equipped with the air-cooled, water-cooled and brine-cooled inverters

The total losses of the compressor and the inverter $P_{\text{loss,tot}}$ is the sum of the losses in the inverter $P_{\text{inv,loss}}$ and the losses in the compressor $P_{\text{com,loss}}$ (see Appendix B). Fig. 39, fig. 40 and fig. 41 represent the total losses $P_{\text{loss,tot}}$ versus compressor rotation speed at the three different inverter switching frequencies when the used inverter is cooled by air, water or brine.

As it can be seen in these three figures, the total losses of the compressor and the inverter $P_{\text{com,loss}}$ increases gradually from about 100 to about 600 (W) by increasing the compressor speed from 20 to 110 (RPS) whatever the used inverter is.

Meanwhile and despite the kind of the used inverter, at each compressor rotation speed the higher the inverter switching frequency is, the higher the total losses $P_{\text{com,loss}}$.

The variation in the total losses of the compressor and the inverter $P_{\text{com,loss}}$ according to the inverter switching frequency changing is highest when the used inverter is cooled by air and lowest when the inverter is cooled by brine (see fig. 39, fig. 40 and fig. 41)
B. The total losses of the compressor and the inverter when the used inverter is cooled by air, water and brine at the same inverter switching frequency

Fig. 42, fig. 43, and fig. 44 show the total losses of the compressor and the inverter $P_{\text{loss, tot}}$ versus compressor rotation speed when the used inverter is cooled by air, water and brine but at the same inverter switching frequency in each diagram. Fig. 42 shows how the total losses of the compressor and the inverter $P_{\text{loss, tot}}$ along with the different compressor speeds at 4 (kHz) inverter switching frequency is almost the same when the heat pump uses the three kinds of inverters. On the other hand, as it could be observed from fig. 43 and fig. 44, at inverter switching frequencies of 6 and 8 (kHz) the total losses of the compressor and the inverter $P_{\text{loss, tot}}$ could be slightly lower when the used inverter is cooled by water and lowest when the inverter is cooled by brine.
5.5. The temperature of the base plate

The median lifetime and the investments costs of the inverter are related with the temperature of the base plate during the operation (Bhunia et al, 2007) (Meysenc et al, 2005). Consequently, the temperature of the base plate of the air-cooled, water-cooled and brine-cooled inverters along with the compressor rotation speed increasing is measured and illustrated in three different diagrams at the three different inverter switching frequencies (see fig. 45, fig. 47 and fig. 48).

Fig. 45 shows the temperature of the base plate of the three inverters (air-cooled, water-cooled and brine-cooled inverters) versus compressor rotation speed at 4 (kHz) inverter switching frequency. The figure highlights how the maximum temperature of the base plate of the water-cooled inverter at 4 (kHz) inverter switching frequency is higher than the one of the base plate of the air-cooled inverter by about 11 (°C). On the contrary, the maximum temperature of the base plate of the brine-cooled inverter is lower than the one of the base plate of the air-cooled inverter by about 26 (°C) (see fig. 45).

As it can be also seen in this diagram, increasing the compressor speed from 20 to 110 (RPS) leads to an obvious increasing in the temperature of the base plate of the water-cooled inverter from about 43 to about 60 (°C) while the temperature of the base plate of the brine-cooled inverter increases just from about 15 to about 23 (°C) (see fig. 45). In contrast, when the compressor speed increases from 20 to 60 (RPS) the temperature of the base plate of the air-cooled inverter has an oscillating behavior at each compressor speed and the mean value of this
temperature remains almost constant and around 42°C (see fig. 46); while from 60 to 110 (RPS) compressor speed the base plate temperature of the air-cooled inverter increases from about 42 to about 49°C (see fig. 45).

Aiming to clarify the above mentioned behavior of the base plate temperature of the air-cooled inverter at compressor speeds lower than 60 (RPS) and 4 (kHz) inverter switching frequency, a diagram of 300 continuous measurements of the base plate temperature of the air-cooled and water-cooled inverters at 40 (RPS) compressor speed and 4 (kHz) inverter switching frequency and during the same time is illustrated (see fig. 46).

As it can be easily observed in this figure, the temperature of the base plate in the air-cooled inverter oscillates between 40 and 45.5°C (see fig. 46-a) while in the water-cooled inverter the temperature of the base plate remains almost constant and around 46.4°C (see fig. 46-a).

In fig. 47 it can be seen how at 6 (kHz) inverter switching frequency when the compressor speed increases from 20 to 110 (RPS), the base plate temperature of the inverter increases from about 16 to about 24°C the brine-cooled inverter, from about 40 to about 53°C in the air-cooled inverter and from about 44 to about 62°C in the water-cooled inverter. Moreover, fig. 47 shows that the maximum temperature of the base plate of the water-cooled inverter in these running conditions is higher than the one of the base plate of the air-cooled inverter by about 10°C while the maximum temperature of the base plate of the brine-cooled inverter is lower by about 28°C.

Finally, at 8 (kHz) inverter switching frequency, the temperature of the base plate of the inverter increases from about 17 to about 27°C in the brine-cooled inverter, from about 43 to about 57°C in the air-cooled inverter and from about 46 to about 63°C in the water-cooled inverter (see fig. 48). Also, it can be observed that the maximum temperature of the base plate of the water-cooled inverter in the above mentioned running conditions is higher than the one of the base plate of the air-cooled inverter by about 6°C while the maximum temperature of the base plate of the brine-cooled inverter is lower by about 31°C (see fig. 48).
Fig. 46. The base plate temperature $T_{bp}$ (°C) of the air-cooled inverter (a) and the water-cooled inverter (b) versus the number of readings at compressor speed of 40 (RPS) and inverter switching frequency of 4 (kHz)

Fig. 47. The base plate temperature $T_{bp}$ (°C) of the air-cooled water-cooled and brine-cooled inverters versus compressor rotation speed $N$ (RPS) at 6 (kHz) inverter switching frequency

Fig. 48. The base plate temperature $T_{bp}$ (°C) of the air-cooled water-cooled and brine-cooled inverters versus compressor rotation speed $N$ (RPS) at 8 (kHz) inverter switching frequency
6. Conclusions

6.1. The inverters losses

In general, increasing the compressor speed or the switching frequency of the inverter will increase the losses in the air-cooled, water-cooled and brine-cooled inverters. Also, the increment magnitude in the inverter losses along with the compressor speed increasing has higher values at lower inverter switching frequencies.

At the same running conditions of compressor speed and inverter switching frequency, the magnitude of the losses in the water-cooled inverter is lower than the one in the air-cooled inverter while this magnitude in the brine-cooled inverter is the lowest. Indeed, the difference between the losses in these three inverters increases by increasing the switching frequencies.

The ratio of the inverter losses to the inverter input power is highest in the air-cooled inverter (2.2 – 13 %) and lowest in the brine-cooled inverter (2 -12 %). In the three kinds of inverters (air-cooled, water-cooled and brine-cooled), this ratio decreases by increasing the compressor speed or decreasing the switching frequency of the inverter. However, when the compressor speed increases from 20 to 60 (RPS) the aforementioned ratio decreases remarkably while the effect of increasing the compressor speed from 60 to 110 (RPS) is much lower.

6.2. The recovered heat

The recovered heat in the water-cooled and brine-cooled inverters increases by raising the compressor speed or the inverter switching frequency. Yet, the recovered heat in the brine-cooled inverter (61-139 W) is higher than the recovered heat in the water-cooled inverter (45-125 W). Also, the higher the inverter switching frequency is, the lower the difference in the recovered heat in the water-cooled and brine-cooled inverters.

The ratio of the recovered heat to the inverter input power in the brine-cooled inverter (1.7 - 10 %) is higher than the one in the water-cooled inverter (1.4 - 8.5 %) under all the running conditions. In both of them the water-cooled and the brine-cooled inverters and at compressor speeds lower than 60 (RPS), this ratio shows high sensitivity for the compressor speed and decreases remarkably by increasing the compressor speed from 20 to 60 (RPS); while this sensitivity decreases at speeds higher than 60 (RPS).

The recovered heat percentage of the losses in the water-cooled inverter varies between 63 and 69 (%) depending on the compressor speed and the inverter switching frequency. On the other hand, this percentage is much higher in the brine-cooled inverter and varies between 79 and 90 (%) which means that the brine-cooled inverter is more effective than the water-cooled one for recovering the heat losses in the inverter.
6.3. The coefficient of performance of the system

The coefficient of performance of the system COP_{sys} has the same decreasing trend along with the compressor speed increasing from 20 to 100 (RPS) at all the switching frequencies of the inverter when the heat pump is equipped with the air-cooled, water-cooled or brine-cooled inverter. In contrast, when the compressor speed increases from 100 to 110 (RPS) the COP_{sys} remains almost constant when any of the three inverters is used.

Despite the fact that the brine-cooled inverter is more efficient than the water-cooled one for recovering the heat losses in the inverter, using any one of them has almost the same effect on the COP_{sys} at all the switching frequencies and all the compressor speeds. Since, the recovered heat in the brine-cooled inverter should be transferred through the heat pump (evaporator, refrigerant cycle and the condenser) and then delivered to the customer heating net while in the water-cooled inverter the recovered heat is delivered directly to the customer heating net.

The improvement in the COP_{sys} by using the water-cooled or brine-cooled inverter varies between 0.54 and 3 (%) of the COP_{sys} when the conventional air-cooled inverter is used. In general, the lower the compressor speed is, the higher this improvement in the COP_{sys} but this can be noticed more at rotation speeds lower than 60 (RPS).

Equally important, the behavior of the COP_{sys} matches the behavior of the carnot efficiency of the system when the heat pump runs by the air-cooled, water-cooled and brine-cooled inverters and at the different switching frequencies; which means that the aforementioned COP_{sys} results are not affected by the fluctuation of the refrigerant evaporation and condensation temperatures.

6.4. The total isentropic efficiency of the compressor

At all the switching frequencies and when the heat pump is equipped with the air-cooled, water-cooled or brine-cooled inverter, the total isentropic efficiency of the compressor of the compressor \( \eta_{is} \) decreases gradually when the compressor speed increases from 30 to 100 (RPS). In contrast, by increasing the compressor speed from 20 to 30 (RPS), the total isentropic efficiency of the compressor \( \eta_{is} \) decreases obviously. While on the contrary, increasing the speed of the compressor from 100 to 110 leads to a sudden increasing in the isentropic efficiency.

At the same inverter switching frequency and compressor rotation speed the total isentropic efficiency of the compressor \( \eta_{is} \) increases slightly by using the water-cooled inverter and a little more by using the brine-cooled inverter in comparison with the standard state when the conventional air-cooled inverter is used. Furthermore, the difference in the total isentropic efficiency of the compressor \( \eta_{is} \) when the three different inverters are used is very low at compressor speeds in the range of 50 to 70 (RPS) while this difference increases at compressor speeds lower and higher than this range. Indeed, this difference increases by increasing the inverter switching frequency at each compressor speed. As the used compressor is a hermetic one and designed to be cooled down by the discharge gas, the higher the compressor losses are, the
lower the total isentropic efficiency of the compressor $\eta_{\text{is}}$. In a word and according to the above mentioned results about the total isentropic efficiency of the compressor $\eta_{\text{is}}$ in this experiment, the losses of the compressor decrease by increasing the switching frequency or by using a water-cooled or brine-cooled inverter.

It is noticed in this experiment that the fluctuation in the input and the output power of the inverter affects the compressor behavior, where, the higher the fluctuation of the power before and after the inverter is, the higher the losses of the compressor. This explains the unexpected behavior of the COP$_{\text{sys}}$ and the $\eta_{\text{is}}$ at compressor speeds higher than 100 (RPS).

### 6.5. The total losses of the compressor and the inverter

The total losses of the compressor and the inverter $P_{\text{loss, tot}}$ increases by increasing the inverter switching frequency when the used inverter is cooled by air, water or brine. On the other hand, at the same switching frequency the total losses $P_{\text{loss, tot}}$ decreases slightly when the heat pump uses the water-cooled inverter and little more by using the brine-cooled inverter in comparison with the standard state when the heat pump uses the conventional air-cooled inverter.

In this study, the compressor losses are calculated depending on the assumption that it corresponds to a certain constant percentage of the inverter power output at each compressor speed. Therefore, the calculated compressor losses stay constant at the same compressor speed when the switching frequency or the cooling method of the inverter is changed. Accordingly, the total losses $P_{\text{loss, tot}}$ results reflect the effect of the inverter losses’ behavior on the total losses’ behavior while the effect of the compressor losses is suppressed when the switching frequency or cooling method of the inverter is changed. In summary, in order to avoid any misleading conclusion about the total losses $P_{\text{loss, tot}}$ further studies should be done in which the compressor losses is directly calculated or measured; and the effect of changing the switching frequency or the cooling method of the inverter on the compressor losses behavior is well analyzed and evaluated.

### 6.6. The temperature of the cold plate

Increasing the compressor speed raises the temperature of the base plate of the three different inverters (air-cooled, water-cooled and brine-cooled inverters) but the water-cooled inverter is more affected than the air-cooled and brine-cooled inverter; since the water in the load cycle that is used for cooling the inverter comes out from the condenser of the heat pump and the temperature of it increases by increasing the compressor speed. Consequently, in the case of using water-cooled inverter, it is recommended to take the cooling water of the inverter from the load cycle before it enters the condenser and return it after the condenser (in another word, bypassing the condenser).

At 4 (kHz) inverter switching frequency and compressor speeds from 20 to 60 (RPS), the temperature of the base plate has an oscillating behavior in the air-cooled inverter. In this case,
the temperature of the base plate oscillates between 40 and 45.5 (°C). This oscillating in the base plate temperature means that the temperature of the electronic components of the inverter is also oscillating. Accordingly, under the above mentioned conditions the electronic components are exposed to thermal stresses, which could shorten the inverter median lifetime.

The maximum temperature of the base plate in the air-cooled inverter is lower by about 10 °C than the one in the water-cooled inverter; while it is higher by about 28 °C than the one in the brine-cooled one. Thus, using the water that comes out from the condenser to cool the inverter under the running conditions of this experiment leads to a two-fold deterioration in the inverter median life. On the contrary, using the brine for cooling the inverter causes about a six-fold improvement in the inverter median life.

The experimental results show that the inverter losses are lowest in the brine-cooled inverter at the same running conditions of compressor rotation speed and inverter switching frequency. This could be attributed to the lower temperature of the base plate of the brine-cooled inverter in comparison with the temperature of the base plate of the water-cooled and the air-cooled inverters. In conclusion, the higher the temperature of the base plate is, the lower the conversion efficiency of the inverter.

Finally, improving the cooling effectiveness and reducing the device temperature could increase the output current and thereby the output power of the inverter while using the same quantity of silicon. Consequently, cooling the inverter by the brine in the source cycle could increase the possibility for reducing the cost of the inverter itself.

6.7. General factors and final conclusion

To summarize, using the liquid-cooled inverter in the ground source heat pump systems, improves the system efficiency, reduces the power consumption and eliminates the noise of the fan in comparison with the standard state when the conventional air-cooled inverter is used. On the other hand, the liquid cooling method could add some pumping power consumption as a result for the extra pressure drop in the inverter cooling cycle, but as long the velocity of the fluid in the inverter cooling cycle is low this pumping power is neglected. However, choosing the most suitable and profitable method for recovering the losses in the inverter does not depend just on the recovering efficiency of the heat losses in the inverter but many other factors should be taken into account like: the inverter median life, the inverter cost and all the aforementioned parameters that represent the behavior of the heat pump.

Equally important, the running conditions of the heat pump are vital to choose the suitable method for cooling the inverter. As example, the high temperature of the water in the load cycle could push the temperature of the base plate of the water-cooled inverter above the allowed limits, especially when the heat pump is used for producing hot water. Likewise, low temperatures of the brine in the source cycle could decrease the temperature of the base plate of
the brine-cooled inverter too much and cause condensation on the internal surface of the base plate of the inverter.

7. Future Work

It is assumed in this study that the compressor losses correspond to a certain constant percentage of the inverter output power at each compressor speed. Consequently, the effect of changing the inverter switching frequency or the inverter cooling method on the compressor losses is suppressed. Therefore, it is highly recommended to conduct a further study in which the compressor losses could be measured or calculated directly and thereby evaluate the effect of changing the inverter switching frequency or the inverter cooling method on it.

The temperature of the base plate is measured in this experiment by one sensor in one position in the middle of the base plate where the hottest spot in the inverter is located. It would be useful to fit the base plate with more temperature sensors in other positions and run the test under different running conditions. This would clarify how the temperature of the base plate in other positions would develop during the different running conditions and a sort of temperature map could be implemented.

All the tests in this experiment are conducted under constant conditions of water temperature at the inlet of the condenser, brine temperature at the inlet of the evaporator, mass flow of the fluid in the cooling cycle of the liquid-cooled inverter, mass flow of the water in the load cycle and mass flow of the brine in the source cycle. Therefore, more tests and further studies with different values of the these parameters are important for having a clear and solid knowledge about the reliability and profitability of using liquid-cooled inverters in the variable capacity heat pump systems.
8. Reference


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9. Appendix

A. Compressor management (Combo Drive for µPC)

Each compressor group has its own operating envelop determined by the manufacturer. This envelop is defined by the condensing and evaporating pressure and the maximum discharge gas temperature. Fig. 49 and fig. 50 show the operating envelop of the compressor used in the heat pump unit when the used refrigerant is R410A.

![Diagram showing compressor envelop](image)

**Fig. 49. The operating envelop of the compressor used in the heat pump unit when the used refrigerant is R410A. condensation pressure versus evaporation pressure (CAREL, 2012b)**

The heat pump unit that is used in the test rig is fully controlled by Combo Drive which is produced by Carel and tested and approved by the compressor manufacturer. The Combo Drive is software installed on the µPC programmable board aiming to ensure that the working conditions of the compressor will be inside the aforementioned operating envelope via Power+ inverter (CAREL, 2012b). Fig. 51 shows how the Power+ inverter, user interface pDG₁, electronic expansion valve the pressure sensors and temperature sensors are connected directly to the µPC board. On the one hand, µPC board receives signals, variables values and orders from the user interface pDG₁, the pressure and temperature sensors. While on the other hand it sends orders to both the expansion valve and the Power+ inverter and adapt their performance to the compressor operating envelop. However Combo Drive defines the working zones of the compressor by measuring the condensing pressure, evaporating pressure and the discharge gas temperature and compares this zone with the operating envelop of the compressor. When the working point of the
compressor is approaching the limits of the operating envelop of the compressor, Combo Drive stops the acceleration or deceleration of the compressor speed. On the other hand if the working point is beyond the limit of the high condensing pressure or the high compression ratio, the compressor speed is reduced. Moreover, when the working point of the compressor remains too long (default 60 seconds) in an area outside the envelope, the compressor is turned off. For advanced information see (CAREL, 2012b).

Fig. 50. The operating envelop of the compressor used in the heat pump unit when the used refrigerant is R410A. condensation temperature versus evaporation temperature (CAREL, 2012b)
Fig. 51. A schematic of the Power+ inverter, user interface pDG₁, electronic expansion valve the pressure sensors and the temperature sensors connected directly to the μPC board.
B. Calculations

The recovered heat in the water-cooled and brine-cooled inverter

\[ Q_{\text{wt,inv}} = f \left( \dot{V}_{\text{wt,inv}}, \rho_{\text{wt}}, c_{p,\text{wt}}, \Delta T_{\text{wt,inv}} \right) = \dot{V}_{\text{wt,inv}} \ast \rho_{\text{wt}} \ast c_{p,\text{wt}} \ast (T_{\text{wt,inv,out}} - T_{\text{wt,inv,in}}) \]  \( \text{(1)} \)

\[ Q_{\text{br,inv}} = f \left( \dot{V}_{\text{br,inv}}, \rho_{\text{br}}, c_{p,\text{br}}, \Delta T_{\text{br,inv}} \right) = \dot{V}_{\text{br,inv}} \ast \rho_{\text{br}} \ast c_{p,\text{br}} \ast (T_{\text{br,inv,out}} - T_{\text{br,inv,in}}) \]  \( \text{(2)} \)

\[ \dot{V}_{\text{wt,inv}} \text{ (measured)}, \quad T_{\text{wt,inv,out}} \text{ (measured)}, \quad T_{\text{wt,inv,in}} \text{ (measured)} \]

\[ \dot{V}_{\text{br,inv}} \text{ (measured)}, \quad T_{\text{br,inv,out}} \text{ (measured)}, \quad T_{\text{br,inv,in}} \text{ (measured)} \]

Compressor power before and after the inverter

\[ P_{\text{sys,inv}} = P_{\text{bef,inv}} \text{ (measured)} \quad \text{,} \quad P_{\text{aft,inv}} \text{ (measured)} \]

The inverter losses

\[ P_{\text{inv,loss}} = P_{\text{bef,inv}} - P_{\text{aft,inv}} \]  \( \text{(3)} \)

The total losses in both the inverter and the compressor

\[ P_{\text{tot,loss}} = P_{\text{bef,inv}} - W_{\text{comp}} \]  \( \text{(5)} \)

The compressor losses

\[ P_{\text{comp,loss}} = P_{aft,inv} - W_{\text{comp}} \]  \( \text{(6)} \)

\[ W_{\text{comp}} = \dot{m}_{\text{ref}} \ast \Delta h_{\text{comp}} = \dot{m}_{\text{ref}} \ast \left( h_{\text{ref,comp,out}} - h_{\text{ref,comp,in}} \right) \]  \( \text{(7)} \)

\[ T_{\text{ref,comp,out}} \text{ (measured)}, \quad T_{\text{ref,comp,in}} \text{ (measured)} = h_{\text{ref,comp,out}}, \quad h_{\text{ref,comp,in}} \]
\( \dot{m}_{\text{ref}} \) is estimated depending on the assumption that the losses in the compressor increases proportionally with compressor speed; thus, when the compressor speed increases from 30 to 90 the compressor losses increases from 3% to 6% of the compressor power input (Madani, 2012).

**The base plate temperature**

\[
T_{\text{ai, p}} \text{ (measured)} \ , \quad T_{\text{wi, p}} \text{ (measured)} \ , \quad T_{\text{br, p}} \text{ (measured)}
\]

**The recovered losses relative the total losses in the inverter**

\[
\frac{Q_{\text{vt, inv}}}{P_{\text{me, inv}}} \pm 100
\]

\[
\frac{Q_{\text{br, inv}}}{P_{\text{me, inv}}} \pm 100
\]

**Heat pump cooling capacity**

\[
Q_{\text{evap}} = \dot{m}_{\text{ref}} \Delta h_{\text{evap}} = \dot{m}_{\text{ref}} (h_{\text{ref, out}} - h_{\text{ref, in}})
\]

**Heat pump heating capacity without the recovered loss**

\[
Q_{\text{cond}} = \dot{m}_{\text{ref}} \Delta h_{\text{cond}} = \dot{m}_{\text{ref}} (h_{\text{ref, out}} - h_{\text{ref, in}})
\]

**The total isentropic efficiency**

\[
\eta_{\text{evap, inv}} = \frac{W_{\text{evap, inv}}}{P_{\text{brf, inv}}}
\]

**The total isentropic efficiency of the compressor**

\[
\eta_{\text{is}} = \frac{W_{\text{comp, is}}}{W_{\text{comp}}} = \frac{(h_{\text{ref, out, is}} - h_{\text{ref, in}})}{(h_{\text{ref, out}} - h_{\text{ref, in}})}
\]

**The heat pump COP$_{\text{sys}}$ when the air-cooled and brine-cooled inverters are used:**

\[
\text{COP}_{\text{sys}} = \frac{Q_{\text{cond}}}{P_{\text{brf, inv}}}
\]

**The heat pump COP$_{\text{sys}}$ when the water-cooled inverter is used:**
\[ \text{COP}_{\text{sys}} = \left( Q_{\text{cond}} + Q_{\text{vol,inj}} \right) / \dot{E}_{\text{vol,inj}} \] (15)

In principle, the extra pumping power owing to the extra pressure drop in the cooling cycle of the water-cooled and brine-cooled inverters should be added to the \( \text{I}_{\text{ext,inv}} \) in order to get very precise comparative of COP\(_{\text{sys}} \) values when the different inverters are used (air-cooled, water-cooled and brine-cooled inverters). Therefore, it is important to know how this extra pumping power is calculated and be aware of the magnitude of its effect on the COP\(_{\text{sys}} \). The main equations that are needed for calculating this power are presented down:

\[ \dot{E}_p = D_p f \times \dot{V} / \eta_p \] (16)

\[ \Delta p_f = f_1 \times \rho \times w^2 \times L / d \] (17)

\[ w = \frac{4 \times \pi d}{\rho \times L \times d^2} \] (18)

\[ f_1 = 0.5 + (0.79 \times \ln Re - 1.64)^2 \quad \text{where} \quad Re \geq 2.3 \times 10^5 \] (19)

\[ \text{Re} = (\rho \times d) / \nu \] (20)

\[ \dot{V} = \frac{m}{\rho} \] (21)

\[ \Rightarrow \dot{E}_p = \frac{4 \times \pi d \times L \times f_1 \times \rho}{4 \times \eta_v} \times \nu^3 \] (22)

As it is mentioned before, the mass flow in the inverter cooling cycle in this experiment is 0.011 Kg/s. Consequently, the velocity of the fluid in the inverter cooling cycle is about 0.00148 m/s for the brine and 0.00156 m/s for the water. Indeed, it can be observed from equation (22) that the pumping power is directly proportional to the velocity of the fluid cube. As example this pumping power equals to 1.65*10\(^{-3}\) (W) in the case of using brine as cooling fluid. In conclusion, the extra pumping power due to the friction in the inverter cooling cycle is too low and can be easily neglected under the running conditions of this experiment. Even, it is anticipated that the velocity of the fluid in the cooling cycle of the liquid-cooled inverter can be held low in all the different running conditions and thereby maintains very low and neglected extra pumping power.
The Carnot efficiency of the system $\eta_{cd}$

$$\eta_{cd} = \frac{COP_{sys}}{COP_c} \tag{23}$$

Where, $COP_{sys}$ is the coefficient of performance of the system and $COP_c$ is the coefficient of performance of the Carnot coefficient of performance.

$$COP_c = \frac{T_1}{T_s - T_2} \tag{24}$$

C. Uncertainty Analysis

In any experimental work the validity of the acquired data should be determined. This means that the accuracy of the appropriate measurements must be known. In addition and regardless of the care of the experimenter, it is possible that an error could happen in any experiment. Likewise, it is not acceptable to neglect acquired data points just because they don’t conform to the experimenter expectation or they fall outside the range of the expected random deviation. Thus, the elimination of any data point should be done on the basis of a consistent statistical data analysis.

C.I. Uncertainties types

There are two types of uncertainty: type A, which is evaluated by statistic methods and type B, which is determined by another means or methods. The uncertainty of type A is the random scatter in the data because of the imperfect control of the measuring process. On the other hand, the uncertainty of type B could happen because of the uncertainty of the instrumentation, which could be estimated by data books, calibration or physical property tables. Both Types A and B should be accounted for the uncertainty determination of the measurand.

C.I.1. Type A evaluation of standard uncertainty

In this paragraph, the main concepts that are used in the statistic method for uncertainty evaluation are represented in order to facilitate the understanding of the statistic method of uncertainty estimation.

For $n$ independent observations $q_k$ under the same conditions of measurement of a quantity $q$ that varies randomly, the arithmetic mean is the best estimation of it (JCGM, 2008).

$$\overline{q} = \frac{1}{n} \sum_{k=1}^{n} q_k \tag{25}$$
The individual observations $q_k$ differ in value because of the random variations in the quantities, or random effects. The experimental variance of the observations, which estimates the variance $S^2$ of the probability distribution of $q$, is given by (JCGM, 2008).

$$S^2_{q_k} = \frac{1}{n-1} \sum_{i=1}^{n} (q_i - \bar{q})^2$$  \hspace{1cm} (26)

This estimate of variance and its positive square root $S_{q_k}$, termed the experimental standard deviation, characterize the variability of the observed values $q_k$, or more specifically, their dispersion about their mean $\bar{q}$. However, the best estimation of the variance of the mean, termed the experimental standard deviation of the mean / the uncertainty of the mean value, is $S^2_q = S^2_{q_k} / n$ which estimate the validity of the value $\bar{q}$ and can be used to determine the uncertainty of $\bar{q}$. This uncertainty is sometimes called a type A standard uncertainty (JCGM, 2008).

$$S^2_q = \frac{S^2_{q_k}}{n} \Rightarrow S_q = \sqrt{\frac{1}{n(n-1)} \sum_{i=1}^{n} (q_i - \bar{q})^2}$$  \hspace{1cm} (27)

C.I.2. Type B evaluation of standard uncertainty

The uncertainty of type B can be estimated by scientific judgment based on all of the available information like (JCGM, 2008).

- Previous measurement data;
- Experience with or general knowledge of the behavior and properties of relevant materials and instruments;
- Manufacturer's specifications;
- Data provided in calibration and other certificates;
- Uncertainties assigned to reference data taken from handbooks.

C.II. The total uncertainty

In many cases, the targeted measurand can’t be measured directly, but determined throw N other measured quantities $(X_1, X_2, \ldots, X_n)$ which are related to each other’s by a special arithmetic relation $y = f(X_1, X_2, \ldots, X_n)$. in some cases $y$ is taken as the arithmetic mean or average of $n$ independent determinations $Y_k$ of $Y$ (JCGM, 2008).
As example:

\[ y = \bar{y} = \frac{1}{n} \sum_{i=1}^{n} Y_i = \frac{1}{n} \sum_{k=1}^{n} f(X_{1,k}, X_{2,k}, \ldots, X_{n,k}) \quad (28) \]

As example:

\[ E_{\text{inv}} = f(E_{\text{b.ref, inv}}, E_{\text{a.ref, inv}}) \quad , \quad Q_{\text{inv}} = \bar{V}_{\text{inv}} + \rho \cdot C_p \cdot (T_{\text{inv, out}} - T_{\text{inv, in}}) \]

In other cases, a \( k \) series of observations are done and the observed value of \( X_i \) is denoted by \( X_{i,k} \).

Thus, for an input quantity \( X_i \) estimated from \( n \) independent repeated observations \( X_{i,k} \) the obtained arithmetic mean \( \bar{X}_i \) is used as the input to determine the measurement result \( y \) (JCGM, 2008):

\[ y = f(\bar{X}_1, \bar{X}_2, \ldots, \bar{X}_n) \quad (29) \]

Where: \( \bar{X}_i = \frac{1}{n} \sum_{k=1}^{n} X_{i,k} \)

As example

\[ Q = f(\bar{V}_{\text{inv}}, \bar{p}, \bar{C}_p, T_{\text{inv, out}}, T_{\text{inv, in}}) = \bar{V}_{\text{inv}} \cdot \bar{p} \cdot \bar{C}_p \cdot (T_{\text{inv, out}} - T_{\text{inv, in}}) \quad (30) \]

\[ E_{\text{inv}} = f(\bar{E}_{\text{b.ref, inv}}, \bar{E}_{\text{a.ref, inv}}) = \bar{E}_{\text{b.ref, inv}} - \bar{E}_{\text{a.ref, inv}} \quad (31) \]

In the end, the uncertainty of measurement result \( y \) of two types A and B can be calculated according to the flowing equations (JCGM, 2008).

\[ S_y = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial y}{\partial X_i} \cdot S_{x_i} \right)^2} \quad (32) \]

\[ w_y = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial y}{\partial X_i} \cdot w_{x_i} \right)^2} \quad (33) \]

Where \( S_y \) is the uncertainty of the type A and \( w_y \) is the uncertainty of type B. As mentioned earlier, both types of uncertainty will be accounted in the estimation of the total uncertainty.

\[ u_y = \sqrt{S_y^2 + w_y^2} \quad (34) \]
In order to get a reliable estimation of the expected value of \( y \), the number of observations \( n \) should be large enough. In this study, 300 separate measurements were conducted for estimating the loosed and recovered power in the inverter at each set-point. Moreover, the degrees of freedom or the confidence level should be given. The aforementioned uncertainty is calculated for one standard deviation or 68.3 % confidence level. In order to obtain 95% confidence, the results should be multiplied with a factor of 1.96 (Holman, 2001).

\[
U_k = k \cdot u_y
\]  

(35)

C.III. Chauvenet’s Criterion

As mentioned earlier, it is possible to get some data points that look strange and don’t belong to the bulk data in each experimental work. These points should not be eliminated without a scientific consistent basis. Chauvenet’s criterion introduce this consistent basis throw a restrictive test applied to a bulk of data to eliminate suspicious data points. The Chauvenet’s criterion is based on the rule that the probability for the observed deviation of a certain point should not be less than \( 1/2n \) (Holman, 2001).

For the final data presentation a new mean value and standard deviation should be computed after eliminating the dubious points from the first calculations. In this study, the Chauvenet’s criterion is applied for data elimination is (Holman, 2001):

\[
\frac{|X_i - \bar{X}|}{\sqrt{\frac{1}{n-1} \cdot \sum_{i=1}^{n} (X_i - \bar{X})^2}} \leq 3.14
\]  

(36)

Where \( n \) is the number of samples, \( X_i \) (i = 1, 2, 3 …n) is the sample reading and \( \bar{X} \) is the mean value.

C.IV. The estimated uncertainty in the measurements of the inverter losses

C.IV.1. Using the uncertainty of the power before and after the inverter

In order to obtain the power consumption in the inverter, two Yokogawa power meters are used to measure the power before and after the inverter. Therefore, the inverter power consumption is calculated by:

\[
P_{\text{in},\text{loss}} = P_{\text{bef},\text{inv}} - P_{\text{aft},\text{inv}}
\]

Where \( P_{\text{bef},\text{inv}} \) and \( P_{\text{aft},\text{inv}} \) are the power consumption that measured before and after the inverter (see Eq. 3) respectively. The normal calculated uncertainty (\( S_{\text{in,inv}} \)) is expressed for one standard
deviation or 68.3 % confidence level. To express the results for 95% confidence level, the calculated values are multiplied by a factor of 1.96.

\[ U_{P_{\text{inv,loss}}} = k \cdot u_{P_{\text{inv,loss}}} \]

\[ u_{P_{\text{inv,loss}}} = \sqrt{S_{P_{\text{inv,loss}}}^2 + w_{P_{\text{inv,loss}}}^2} \]

\[ S_y = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial y}{\partial x_i} \cdot s_{x_i} \right)^2} \]

\[ S_{P_{\text{inv,loss}}} = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial P_{\text{inv,loss}}}{\partial x_i} \cdot s_{x_i} \right)^2} \]

\[ S_{P_{\text{inv,loss}}} = \sqrt{\left( \frac{\partial P_{\text{inv}}}{\partial P_{\text{bef,inv}}} \cdot S_{P_{\text{bef,inv}}} \right)^2 + \left( \frac{\partial P_{\text{inv}}}{\partial P_{\text{aft,inv}}} \cdot S_{P_{\text{aft,inv}}} \right)^2} \]

\[ \frac{\partial P_{\text{inv}}}{\partial P_{\text{bef,inv}}} = 1 \quad \& \quad \frac{\partial P_{\text{inv}}}{\partial P_{\text{aft,inv}}} = -1 \Rightarrow S_{P_{\text{inv}}} = \sqrt{\left( S_{P_{\text{bef,inv}}} \right)^2 + \left( -S_{P_{\text{aft,inv}}} \right)^2} \]

Where \( S_{P_{\text{bef,inv}}} \) and \( S_{P_{\text{aft,inv}}} \) are the estimated uncertainties of the mean value of the inverter power consumption before and after the inverter, respectively.

\[ S_{P_{\text{bef,inv}}} = \sqrt{\frac{1}{n(n-1)} \sum_{j=1}^{n} (P_{\text{bef,inv}} - \overline{P}_{\text{bef,inv}})^2} \]

\[ S_{P_{\text{aft,inv}}} = \sqrt{\frac{1}{n(n-1)} \sum_{j=1}^{n} (P_{\text{aft,inv}} - \overline{P}_{\text{aft,inv}})^2} \]

\[ w_y = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial P_{\text{inv,loss}}}{\partial x_i} \cdot w_{x_i} \right)^2} \]


\[ w_p = \sqrt{\left( \frac{\partial P_{\text{inv},\text{loss}}}{\partial P_{\text{bef},\text{inv}}} + w_{\text{bef},\text{inv}} \right)^2 + \left( \frac{\partial P_{\text{inv},\text{loss}}}{\partial P_{\text{af},\text{inv}}} + w_{\text{af},\text{inv}} \right)^2} \]

For the sake of increasing the reliability of the power meters and thereby the reliability of the test results, the power meters have been calibrated just before the test. Consequently, the uncertainty of type B for power meter before the inverter \(w_{\text{bef},\text{inv}}\) and the power meter after the inverter \(w_{\text{af},\text{inv}}\) are obtained from the calibration certificate (Andersson, 2012a) & (Andersson, 2012b).

\[ w_{\text{af},\text{inv}} = 0.001015 \times p_{\text{af},\text{inv}} \] (37)

\[ w_{\text{bef},\text{inv}} = 0.001015 \times p_{\text{bef},\text{inv}} \] (38)

C.IV.2. Using the uncertainty of the difference of the power before and after the inverter

In the case of calculating the inverter losses, the measurements of the power before and after the inverter are not interested but the difference between them. Therefore, it could be possible to calculate the uncertainty of the difference of the power before and after the inverter directly if the uncertainty of type A and type B of the power difference are available. In this case the only needed parameters are: \(S_{\text{inv},\text{loss}}\) and \(w_{\text{inv},\text{loss}}\).

\[ U_{\text{inv},\text{loss}} = k \times u_{\text{inv},\text{loss}} \]

\[ u_{\text{inv},\text{loss}} = \sqrt{S_{\text{inv},\text{loss}}^2 + w_{\text{inv},\text{loss}}^2} \]

Where \(S_{\text{inv},\text{loss}}\) is the uncertainty of the mean value of the inverter losses, which corresponds to the uncertainty of type A. \(w_{\text{inv},\text{loss}}\) is the uncertainty of type B of the inverter losses. It is reasonable to assume that the uncertainty of the power difference measurement \(w_{\text{inv},\text{loss}}\) is equal to or less than the uncertainty of the power measurements before and after the inverter \(w_{\text{af},\text{inv}}, w_{\text{af},\text{inv}}\) (Andersson, 2012c). That means that it is assumed that if the power meters have been calibrated to each other's, the resulted uncertainty of the difference between the readings of both of them \(w_{\text{inv},\text{loss}}\) will be equal or lower than the uncertainty of the reading of each of them \(w_{\text{af},\text{inv}}, w_{\text{af},\text{inv}}\) \(w_{\text{af},\text{inv}} \leq w_{\text{af},\text{inv}}\), \(w_{\text{af},\text{inv}} \leq w_{\text{af},\text{inv}}\). Therefore, it could be approved to assume that:

\[ w_{\text{inv},\text{loss}} = w_{\text{af},\text{inv}} = w_{\text{af},\text{inv}} = 0.001015 \times p_{\text{af},\text{inv}} \] (39)
On the other hand, the uncertainty of type A ($\delta_{\text{inv,less}}$) for the inverter losses can be calculated directly from the calculated differences between the measured power before and after the inverter at each reading. Taking into account the last mentioned assumption table 3, table 4 and table 5 present the measured inverter losses and the estimated uncertainty for 95% confidence level which are also illustrated in fig. 11, fig. 12 and fig. 13.

**C.V. The estimated uncertainty of the recovered heat in the inverter**

**C.V.1. Using the uncertainty of the temperatures at the inlet and the outlet of the inverter cooling cycle**

$$U_{Q_{\text{inv}}} = k \cdot u_{Q_{\text{inv}}}$$

$$u_{Q_{\text{inv}}} = \sqrt{S_{\theta_{\text{inv}}}^2 + w_{\theta_{\text{inv}}}^2}$$

$$S_{Q_{\text{inv}}} = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial Q_{\text{inv}}}{\partial X_i} \cdot s_{X_i} \right)^2}$$

$$Q_{\text{inv}} = f(\tilde{V}_{\text{in}}, \rho, \varepsilon_{\rho}, T_{\text{inv,in}}, T_{\text{inv,out}}) = \tilde{V}_{\text{in}} \times \tilde{\rho} \times \tilde{\varepsilon}_{\rho} \times (\tilde{T}_{\text{inv,out}} - \tilde{T}_{\text{inv,in}})$$

$$\frac{\partial Q_{\text{inv}}}{\partial \tilde{V}_{\text{inv}}} = \tilde{\rho} + \tilde{\varepsilon}_{\rho} \times (\tilde{T}_{\text{inv,out}} - \tilde{T}_{\text{inv,in}})$$

$$\frac{\partial Q_{\text{inv}}}{\partial \rho} = \tilde{V}_{\text{in}} + \tilde{\varepsilon}_{\rho} \times (\tilde{T}_{\text{inv,out}} - \tilde{T}_{\text{inv,in}})$$

$$\frac{\partial Q_{\text{inv}}}{\partial \varepsilon_{\rho}} = \tilde{V}_{\text{in}} \times \tilde{\rho} \times (\tilde{T}_{\text{inv,out}} - \tilde{T}_{\text{inv,in}})$$

$$\frac{\partial Q_{\text{inv}}}{\partial T_{\text{inv,out}}} = \tilde{V}_{\text{in}} \times \tilde{\rho} \times \tilde{\varepsilon}_{\rho}$$

$$\frac{\partial Q_{\text{inv}}}{\partial T_{\text{inv,in}}} = -\tilde{V}_{\text{in}} \times \tilde{\rho} \times \tilde{\varepsilon}_{\rho}$$
In order to get high precision, an electromagnetic flow meter of the VN series and of model (0.5) has been accredited (Aichi Tokei Denki, 2012). VN0.5 model is a proper flow meter for fluid flows in the range of 0.05 to 1 (L/min) (Aichi Tokei Denki, 2012). This kind of flow meters has two ways for estimating the precision of the readings depending on the volumetric flow range; where at fluid flow higher than 0.2 (L/min) the uncertainty is low and corresponds to 2% of the actual reading (Aichi Tokei Denki, 2012). Therefore, the fluid flow in the inverter cooling cycle should be chosen in the range of 0.2 to 1 (L/min) in order to get readings of high precision. Thus the approved uncertainty of type B for the used flow meter in this experiment is (Aichi Tokei Denki, 2012):

\[ w_{\bar{V}_{\text{inv}}} = 0.02 \cdot \bar{V}_{\text{inv}} \] (40)
As the dissipated heat is in the inverter is low, the temperature difference over the inverter cooling cycle is very low. Therefore, the temperature sensors that are used to measure the temperature at the inlet and the outlet of the cooling cycle of the inverter should have a very high precision. Accordingly, the used temperature sensors are PT100 of type A. The uncertainty of the measurements for this kind of sensors is calculated depending on the equation (Pentronic, 2002):

$$w_t_{\text{inlet}} = w_t_{\text{exit}} = (0.15 + 0.002 + |t|)$$

(41)

Where $|t|$ is the absolute value of the temperature in °C.

The density $\rho$ and the heat capacity $c_p$ of the water are obtained from the program Ref Prob; thus the uncertainty of type B for them is obtained from the program help and calculated depending on the equations:

$$w_\rho = \frac{0.0001}{100} \cdot \rho_{\text{water}}$$

(42)

$$w_{c_p} = \frac{0.1}{100} \cdot c_p_{\text{water}}$$

(43)

The density $\rho$ and the heat capacity of the brine are assumed to be constant and obtained at 30.5 (%) of Ethylene glycol-water concentration and at temperature of 5 (°C) neglecting the very small fluctuation in their values according to the temperature fluctuation in the inverter cooling cycle. The density and the heat capacity of the brine at the above mentioned conditions are 1044.1 (kg/m3), 3689.5 (J/kg.K) respectively (Melinder, 2010). However, the commercial product of Ethylene Glycol has some inhibitors; thus, the product has been tested in the lab of the university (Royal Institute of Technology, KTH) and the density value has corrected to 1043.4 (kg/m3) (Ignatowicz, 2012). According to an email communication with Åke Melinder (the author of the book: Properties of the secondary working fluids for indirect systems), the uncertainty of type B for the density $\rho$ and the heat capacity values of the Ethylene glycol-water is (Melinder, 2012):

$$w_\rho = \frac{0.5}{100} \cdot \rho_{\text{EGW}}$$

(44)

$$w_{c_p} = \frac{1}{100} \cdot c_p_{\text{EGW}}$$

(45)
C.V.2. Using the uncertainty of the difference of the temperatures at the inlet and the outlet of the inverter cooling cycle

Using the uncertainty of the temperatures at the inlet and the outlet of the inverter cooling cycle for calculating the uncertainty of the recovered energy leads to a very high value of uncertainty. This does not reflect the actual situation because the used parameter is the temperatures difference between the outlet and the inlet of the cooling cycle. Consequently, using the uncertainty of this difference for calculating the uncertainty of the recovered energy, gives a more reliable and actual evaluation of the recovered energy in the inverter cooling cycle. Therefore, this is the approved method for calculating the uncertainty of the recovered energy in this study and a detailed analysis of the needed equations are presented below.

\[
U_{\Delta T_{inv}} = k \cdot u_{\Delta T_{inv}}
\]

\[
u_{\Delta T_{inv}} = \sqrt{S_{\Delta T_{inv}}^2 + \nu_{\Delta T_{in}}^2}
\]

\[
S_{\Delta T_{inv}} = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial Q_{\Delta T_{inv}}}{\partial \chi_i} \cdot S_{\chi_i} \right)^2}
\]

\[
S_{\Delta T_{inv}} = \sqrt{\left( \frac{\partial Q_{\Delta T_{inv}}}{\partial V_{\Delta T_{inv}}} \cdot S_{V_{\Delta T_{inv}}} \right)^2 + \left( \frac{\partial Q_{\Delta T_{inv}}}{\partial \rho} \cdot S_{\rho} \right)^2 + \left( \frac{\partial Q_{\Delta T_{inv}}}{\partial c_p} \cdot S_{c_p} \right)^2 + \left( \frac{\partial Q_{\Delta T_{inv}}}{\partial T_{inv}} \cdot S_{T_{inv}} \right)^2}
\]

\[
Q_{\Delta T_{inv}} = f(V_{\Delta T_{inv}}, \rho, c_p, \Delta T_{inv}) = V_{\Delta T_{inv}} \cdot \rho \cdot c_p \cdot \Delta T_{inv}
\]

\[
\frac{\partial Q_{\Delta T_{inv}}}{\partial V_{\Delta T_{inv}}} = \frac{\partial}{\partial V_{\Delta T_{inv}}} \left( V_{\Delta T_{inv}} \cdot \rho \cdot c_p \cdot \Delta T_{inv} \right) = \rho \cdot c_p \cdot \Delta T_{inv}
\]

\[
\frac{\partial Q_{\Delta T_{inv}}}{\partial \rho} = \frac{\partial}{\partial \rho} \left( V_{\Delta T_{inv}} \cdot \rho \cdot c_p \cdot \Delta T_{inv} \right) = V_{\Delta T_{inv}} \cdot c_p \cdot \Delta T_{inv}
\]

\[
\frac{\partial Q_{\Delta T_{inv}}}{\partial c_p} = \frac{\partial}{\partial c_p} \left( V_{\Delta T_{inv}} \cdot \rho \cdot c_p \cdot \Delta T_{inv} \right) = V_{\Delta T_{inv}} \cdot \rho \cdot \Delta T_{inv}
\]

\[
\frac{\partial Q_{\Delta T_{inv}}}{\partial T_{inv}} = \frac{\partial}{\partial T_{inv}} \left( V_{\Delta T_{inv}} \cdot \rho \cdot c_p \cdot T_{inv} \right) = V_{\Delta T_{inv}} \cdot \rho \cdot c_p
\]

\[
S_{\nu_{\Delta T_{inv}}} = \sqrt{\frac{1}{n(n-1)} \sum_{i=1}^{n} (\bar{V}_{\Delta T_{inv}} - \tilde{V}_{\Delta T_{inv}})^2}
\]

62
In order to obtain the uncertainty of type B for the temperature difference \( \Delta T \), the temperature sensors at the inlet and the outlet of the cooling cycle have been connected to the acquisition systems and calibrated to each other. The calibration process conducted by:

1. Putting the temperature sensors in the same sensor bath.
2. Increasing the temperature of the liquid in the sensor bath in steps of 5 °C.
3. Taking the readings of the two sensors at the steady state of each step for ten minutes.
4. Calculating the temperature difference between the two sensors at each reading.
5. Illustrate the temperature difference versus the reference temperature (see fig. 52).
6. Extract the equation of the temperature difference correction
   \[
   y = -0.0004 \cdot x + 0.0422
   \] (46)
7. Calculate the temperature difference according to the extracted equation.
8. Calculate the difference between the measured \( \Delta T \) and the calculated one \( \Delta T_c \).
9. Finally calculate the standard deviation according to the (Holman, 2001)

\[
\sigma_{\Delta T,T} = \sqrt{\frac{1}{(n-2)} \sum_{i=1}^{n} (\Delta T_i - \Delta T_{i,c})^2}
\] (47)
The final result of the calibration process showed that the standard deviation $\sigma_{\Delta T,T}$ is equal to about 0.0058, which represents the uncertainty of type B for the temperature difference after the correlation $\nu_{\Delta T_{\text{ref}}}$.

$$y = -0.0004x + 0.0422$$
$$R^2 = 0.67322$$

**Fig. 52. The temperature difference between the readings of two PT100 versus the reference temperature during the calibration process, where the two PT100 are the temperature sensors that are used at the inlet and the outlet of the inverter cooling cycle and connected to the acquisition system.**
### D. Tables of results

**Table 3.** The measured air-cooled inverter losses $P_{\text{inv,loss}}$ (W) and its estimated uncertainty with a confidence level of 95 (%) versus compressor rotation speed $N$ (RPS) when the switching frequency of the inverter is 4, 6 and 8 kHz.

<table>
<thead>
<tr>
<th>Cooling types</th>
<th>Air-cooled inverter</th>
</tr>
</thead>
<tbody>
<tr>
<td>source/load temperatures</td>
<td>5/35 (°C)</td>
</tr>
<tr>
<td>Switching frequency (kHz)</td>
<td>4 (kHz)</td>
</tr>
<tr>
<td>$P_{\text{inv,loss}}$</td>
<td>Uncertainty</td>
</tr>
<tr>
<td>20</td>
<td>74.87</td>
</tr>
<tr>
<td>30</td>
<td>76.24</td>
</tr>
<tr>
<td>40</td>
<td>77.73</td>
</tr>
<tr>
<td>50</td>
<td>85.94</td>
</tr>
<tr>
<td>60</td>
<td>91.65</td>
</tr>
<tr>
<td>70</td>
<td>96.34</td>
</tr>
<tr>
<td>80</td>
<td>105.32</td>
</tr>
<tr>
<td>90</td>
<td>113.15</td>
</tr>
<tr>
<td>100</td>
<td>124.26</td>
</tr>
<tr>
<td>110</td>
<td>147.21</td>
</tr>
</tbody>
</table>

**Table 4.** The measured water-cooled inverter losses $P_{\text{inv,loss}}$ (W) and its estimated uncertainty with a confidence level of 95 (%) versus compressor rotation speed $N$ (RPS) when the switching frequency of the inverter is 4, 6 and 8 kHz.

<table>
<thead>
<tr>
<th>Cooling types</th>
<th>water-cooled inverter</th>
</tr>
</thead>
<tbody>
<tr>
<td>source/load temperatures</td>
<td>5/35 (°C)</td>
</tr>
<tr>
<td>Switching frequency (kHz)</td>
<td>4 (kHz)</td>
</tr>
<tr>
<td>$P_{\text{inv,loss}}$</td>
<td>Uncertainty</td>
</tr>
<tr>
<td>20</td>
<td>71.43</td>
</tr>
<tr>
<td>30</td>
<td>75.82</td>
</tr>
<tr>
<td>40</td>
<td>79.20</td>
</tr>
<tr>
<td>50</td>
<td>81.94</td>
</tr>
<tr>
<td>60</td>
<td>88.28</td>
</tr>
<tr>
<td>70</td>
<td>94.23</td>
</tr>
<tr>
<td>80</td>
<td>102.40</td>
</tr>
<tr>
<td>90</td>
<td>111.16</td>
</tr>
<tr>
<td>100</td>
<td>124.31</td>
</tr>
<tr>
<td>110</td>
<td>149.54</td>
</tr>
</tbody>
</table>
Table 5. The measured water-cooled inverter losses $P_{\text{inv,loss}}$ (W) and its estimated uncertainty with a confidence level of 95 (%) versus compressor rotation speed $N$ (RPS) when the switching frequency of the inverter is 4, 6 and 8 kHz.

<table>
<thead>
<tr>
<th>Cooling types</th>
<th>brine-cooled inverter</th>
<th>5/35 (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Switching frequency (kHz)</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>4 (kHz)</td>
<td>6 (kHz)</td>
</tr>
<tr>
<td></td>
<td>$P_{\text{inv,loss}}$</td>
<td>Uncertainty</td>
</tr>
<tr>
<td>20</td>
<td>68.24</td>
<td>0.81</td>
</tr>
<tr>
<td>30</td>
<td>73.19</td>
<td>0.63</td>
</tr>
<tr>
<td>40</td>
<td>74.88</td>
<td>0.73</td>
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<td>78.88</td>
<td>1.35</td>
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<td>82.70</td>
<td>0.94</td>
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<td>92.02</td>
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<td>80</td>
<td>97.40</td>
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<td>106.96</td>
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<tr>
<td>100</td>
<td>121.72</td>
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<tr>
<td>110</td>
<td>141.31</td>
<td>3.01</td>
</tr>
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</table>

Table 6. The recovered heat $Q_{\text{inv}}$ (W) and its estimated uncertainty with a confidence level of 95 (%) in the water-cooled inverter versus compressor rotation speed $N$ (RPS) when the switching frequency of the inverter is 4, 6 and 8 kHz.

<table>
<thead>
<tr>
<th>Cooling types</th>
<th>water-cooled inverter</th>
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</thead>
<tbody>
<tr>
<td>Switching frequency (kHz)</td>
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<td></td>
</tr>
<tr>
<td></td>
<td>4 (kHz)</td>
<td>6 (kHz)</td>
</tr>
<tr>
<td></td>
<td>$Q_{\text{inv}}$</td>
<td>Uncertainty</td>
</tr>
<tr>
<td>20</td>
<td>45.09</td>
<td>1.89</td>
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<tr>
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<td>48.08</td>
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<td>50.92</td>
<td>2.12</td>
</tr>
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<td>50</td>
<td>52.68</td>
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<td>67.10</td>
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<td>79.29</td>
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<td>110</td>
<td>93.88</td>
<td>3.80</td>
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Table 7. The recovered heat $Q_{\text{inv}}$ (W) and its estimated uncertainty with a confidence level of 95 (%) in the brine-cooled inverter versus compressor rotation speed $N$ (RPS) when the switching frequency of the inverter is 4, 6 and 8 kHz.

<table>
<thead>
<tr>
<th>Cooling types</th>
<th>brine-cooled inverter</th>
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<tbody>
<tr>
<td>source/load temperatures</td>
<td>5/35</td>
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<tr>
<td>Switching frequency (kHz)</td>
<td>4 (kHz)</td>
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