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Evaluation of an economizer solution for a heat pump

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Abstract

This thesis evaluates the performance of a heat pump intended to be used as an economizer. The idea is to use the heat from still hot refrigerant after the condenser in a larger heat pump, running on propane, as the heat source for the economizer pump. So, at the same time as this increases the subcooling for the larger pump, the necessary heat is provided to the smaller heat pump. Though, in this report only the small pump was considered.

Components for the pump were already decided upon, and when the components were all available the layout of the pump was planned. One of the main interests for the project was to maintain the refrigerant charge as small as possible, since it is using the flammable refrigerant isobutane. Also, the measurement points and devices were decided upon, what, where and how the desired values should be obtained.

When the heat pump was laid out and the rest of the test facility was constructed test were run in order to evaluate the performance of the heat pump. For the tests water was used as both heat sink and source, even though the future heat source will be propane. The tests included different combinations of three parameters; evaporating temperature, water temperature at the condenser outlet and compressor speed.

The results show that the heat pump is overall well-functioning and the coefficient of performance is relatively high for the operating conditions studied. As expected, the heating capacity increases with the increased speed of the compressor. Heating capacity, cooling capacity and COP all three increases with an increased evaporating temperature while maintaining the water temperature from the condenser constant. The COP values are considered quite high, the economizer could be economically beneficial though more economic evaluation must be made.

Sammanfattning

Den här rapporten utvärderar en separat värmepumps funktionalitet med syfte att användas som en "economizer". Idén är att använda värme från varmt köldmedium efter kondensorn från en större värmepump, med propan, som värmekälla till den mindre "economizer" pumpen. Samtidigt som underkylningen ökar för den större värmepumpen används den värmen som värmekälla i förångaren till den lilla värmepumpen. I denna rapport är endast den mindre pumpen utvärderad.

De komponenter som skulle användas var redan bestämda och när alla komponenter fanns på plats startade planeringen av hur pumpen skulle sättas ihop. En viktig aspekt inom projektet var att mängden köldmedium skulle hållas liten. Detta på grund av att köldmediet är isobutan som är ett brandfarligt köldmedium. Även planeringen av vad som skulle mätas var en del av förarbetet.

När pumpen var byggd kördes tester för att kunna utvärdera hur väl fungerande värmepumpen är. För dessa tester användes bara vatten i både förångaren och kondensorn, men för fortsatt arbete kommer propan att användas som värmekälla. Testerna inkluderade olika kombinationer av tre parametrar; förångningstemperatur, vattentemperatur ut från kondensorn och varvtal för kompressorn.

Resultaten visar att pumpen fungerar som den ska och COP värdena är relativt höga för många av de testade fallen. Som väntat ökar producerad värme med ett ökat varvtal. Producerad värme, kyla och COP ökar med en högre förångningstemperatur om vattentemperaturen ut från kondensorn är densamma. Värmepumpen är väl fungerande men mer utvärdering, såsom exempelvis ekonomiska beräkningar, måste göras för att fastställa att det är en bra lösning.

Foreword

This thesis is the last part of my master program Sustainable Energy Utilization at the Royal Institute of Technology, and also the final work for my engineering degree in Mechanical Engineering. I would like to thank my supervisor Jan-Erik Nowacki for all interesting discussions and guidance. Furthermore, I would also like to thank the rest of the participants in the research project. Lastly, I would like to thank Peter Hill, lab responsible, and Benny Sjöberg, lab technician, for all their help and patience during this experimental thesis.

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1. Introduction

In this chapter the background to this project is presented, along with the purpose and aim of this project. Also, the research questions to be answered and the used methodology are described.

1.1 Background

Almost all buildings in Sweden need hot water for some purpose. It could be for heating tap water, for space heating (radiators or floor heating) or for heating air in the ventilation system. In spite of population increase and an increase in the surface to be heated, the heating demand for all buildings has been reduced by 1.4% per year in Sweden since 1995. This is mainly due to heat pumps, now saving 25 TWh per year, but also due to better insulation of buildings. The electricity demand for heating has simultaneously been reduced with 1.4% per year (Grahn 2019).

Though there are several options for Domestic Hot Water (DHW) production, heat pumps are today commonly used in Sweden, and it is one of the most environmentally friendly options. The installation of heat pumps in single family houses is increasing. In total 37 TWh of heat was produced from all heat pumps in Sweden, for all types of buildings, in 2019 (Grahn 2019). The heat pump share is increasing, since the new houses built today are extensively equipped with heat pumps from the beginning and also older buildings are installing heat pumps. For multifamily buildings, district heating is the largest source for heat, but today there are also other options like larger heat pumps. Though, important to remember is that parts of the district heating is also produced by heat pumps. Using an economizer is a way to increase both the efficiency and power of a heat pump economically. A few more components must though be added to the heat pump.

1.2 Purpose and aim

There are many economizer solutions, utilizing the hot liquid after the condenser in a heat pump or refrigeration machine. The purpose of this project is to evaluate the implementation of a certain type of economizer solution for a heat pump. How will the economizer affect the system, i.e. how will the amount of heat produced and electricity input vary during different conditions. The COP of the system will be a function of this heat output and the electric input and a good indicator of the efficiency of the system. A heat pump will be built and first tested with water as heat source. It will later in the research project be evaluated for an economizer solution as a part of a larger propane heat pump.

In the future, thus, this could be a possible solution for an economizer system. The aim is to be able to produce additional heat with the economizer at e.g. higher temperature, than the main condenser. This extra heat, at a higher temperature, could be used for other purposes such as tap water heating.

Also, the small intended economizer heat pump can be used as a standalone heat pump in place of a substation in a district heating context. Such a heat pump substation would hypothetically enable lowering of the temperatures in the whole district heating network. The heat pump could increase the secondary temperatures to buildings with high-temperature radiators and for DHW production, while using district heating water as a heat source. Another usage in that context would be to use the district heating return water as a source in individual buildings outside the main district heating network. Distribution of that heat to e.g. single family houses could then be made using cheaper system of plastic pipes. The heat pumps would be placed in the individual buildings.

1.3 Research questions

The purpose of this project is to evaluate the economizer system:

- How well functioning is the tested economizer solution during different operating conditions?
    - How well do the components in the system function?
- How are the experimental results compared to the simulated ones?
- What amount of refrigerant is needed in the system in order for it to function properly?

1.4 Methodology

First a literature review was conducted, studying the existing solutions and the theory behind these. There are several thus different systems using “economizers” in different ways and there are advantages and drawbacks for each of them. The literature review was performed to get a deeper understanding of “economizers”.

An excel model, created by Professor Eric Granryd, was used for simulations of the economizer system setup in this project. The simulations were made in order to be able to compare the test runs with calculated values, to evaluate the functionality of the system. This way the model used could also be verified converging measurements and calculations.

Originally two different economizer heat pumps were going to be tested. One would use a Sanden compressor and the other would use a Bitzer compressor. However, due to an outbreak of the Corona virus and other causes, the Bitzer economizer heat pump was delayed. Thus, only the Sanden economizer heat pump was tested. To test both economizer heat pumps however, an existing test rig was rebuilt and pre-tested in the laboratory. It had to be decided how this test rig would function, how the components should be connected to each other and which data that had to be collected in order to be able to evaluate the system.

When the system had been designed and all components were gathered, the economizer heat pump was assembled. Subsequently, the system was run under different conditions, to provide good results for analysis. Following the test runs, an evaluation of the system took place where the results were obtained.

1.5 Scope and limitations

The scope of this project consisted of the construction and test runs of the economizer heat pump system. Both test runs and calculations were done for the system, to be able to answer the research questions and properly evaluate the system.

The main limitation of this project is economical. Since this project is a part of a larger research project, there has also been a budget to keep in mind. That affects both the economizer heat pump system and the test rig. Also, time has been a limiting factor. Therefore this evaluation does not describe the final implementation of the economizer system. A real economizer heat pump will e.g. likely use propane as a heat source, as compared to water as evaluated here. Future adjustments can, and likely also will, be made.

2. Literature review

Below a literature review of different systems involving economizers is presented.

2.1 Various types of economizers

An economizer is used in a heat pump or refrigeration cycle in order to increase the heating or cooling capacity. According to Granryd et al. (2011) there are two common solutions for economizer systems using the same refrigerant circuit. One of the solutions includes using two compressors, a flash tank and two expansion devices, see figure 1. After the first expansion device the liquid tends to collect at the bottom and gas at the top of a flash tank. The vapor in the flash tank, at an intermediate pressure, is compressed and mixed into the high pressure refrigerant flow from the main compressor. The liquid from the flash tank, is led to a low pressure expansion valve, passed through the evaporator and the main compressor. After the two compressors, the refrigerant flows are mixed.

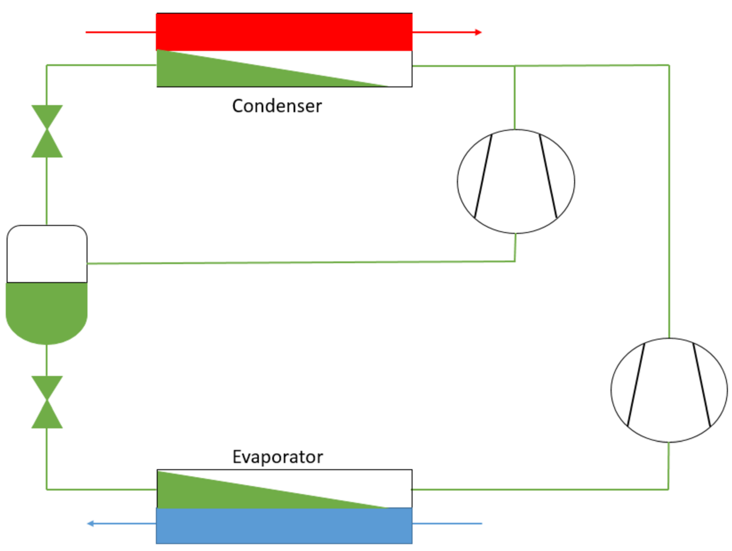

Figure 1: Economizer solution, with flash tank

The other solution includes a heat exchanger, see figure 2. In this case a smaller fraction of the refrigerant after the condenser is passed through an expansion device to an intermediate pressure and then through a counter flow heat exchanger/evaporator. In this heat exchanger heat is absorbed from the main part of the refrigerant. This subcools the main part of the refrigerant significantly. After the heat exchanger, the rest of the now subcooled refrigerant passes through the next expansion device. The temperature of the liquid before that main expansion is then much lower than it would have been without the economizer. Thus more heat is taken up in the evaporator. The evaporated gas from the economizer in figure 2 is shown leading into the intermediate pressure port on the compressor.

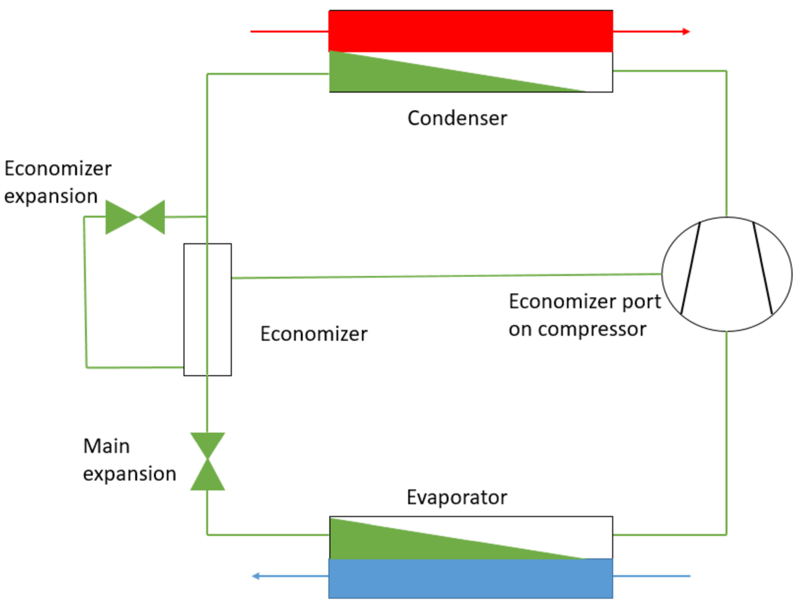

Figure 2: Economizer solution, with heat exchanger and compressor with economizer port

The main advantage of a scheme according to figure 2 is that the oil-return to the compressor is guaranteed, whereas special arrangements for that might be required, using a connection according to figure 1.

The same heat extraction, resulting in subcooling, could however also be done using an independent heat exchanger, using the heat for another purpose, e.g. producing lukewarm water for heating the cellar of a house defrosting e.g. the entrance.

In this project, however, a heat exchanger after the condenser is subcooling the refrigerant in the larger cycle, while being the evaporator of a smaller heat pump, see figure 3. The two are thus connected through the heat exchanger. This results in high evaporating temperatures for the smaller pump.

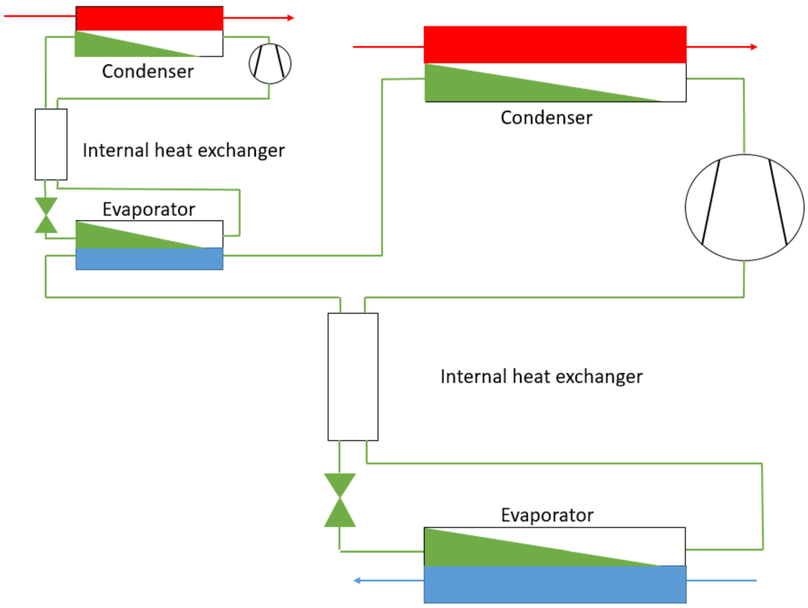

Figure 3: Economizer solution evaluated in this project

2.2 Advantages and drawbacks of different systems

There are advantages and drawbacks with each economizer type. The two common different systems are presented above in figures 1 and 2. But, for each of them there are many different possible variant configurations. The two internal heat exchangers are mainly used to avoid refrigerant entrainment in the oil, but also have marginal impacts of the COP.

Either two compressors or a single compressor with two stage compression can be used. These alternatives can be combined with either a flash tank or a subcooling heat exchanger. For example see figures 4 and 5. For this system you can either mix the two circuits at middle pressure level, two compression in series, or have two parallel compressors for the two circuits.

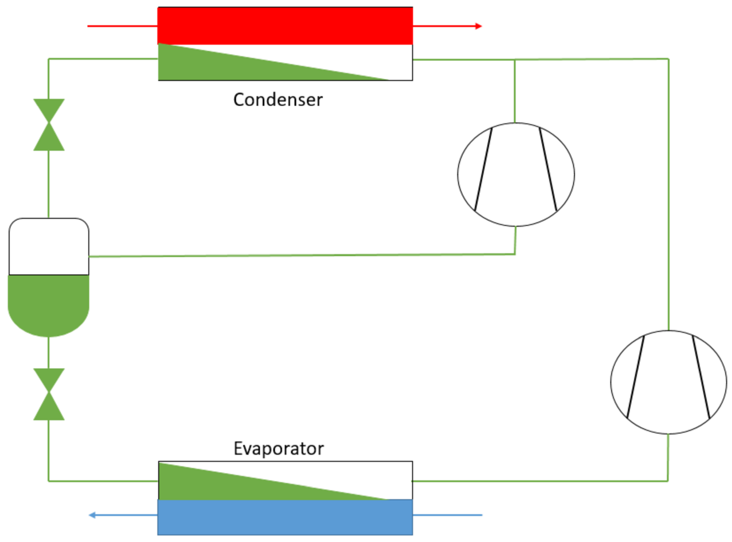

Figure 4: Systems with two compressors

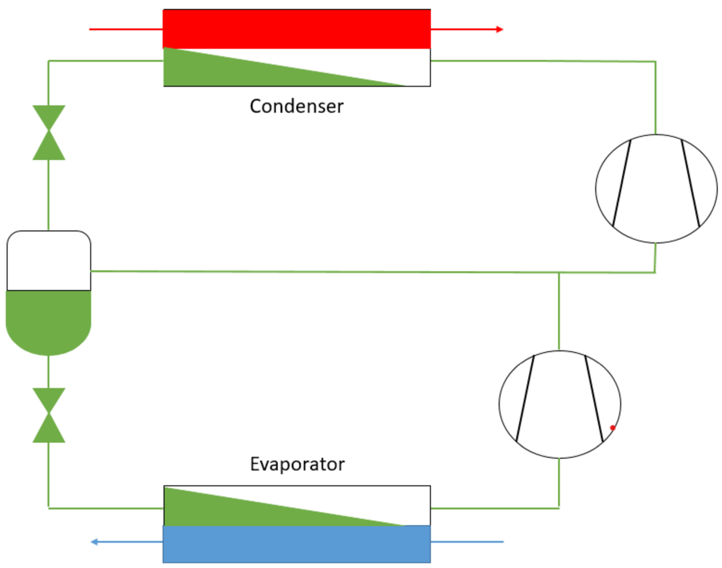

Figure 5: Systems with two compressors

One of the disadvantages with having two compressors in series, mixing the liquid and vapor circuits, is the fact that the inlet temperature to the second compressor is relatively high (Granryd et al. 2011). Though, this temperature can be reduced with a "bubble through" intercooling flash tank, figure 6.

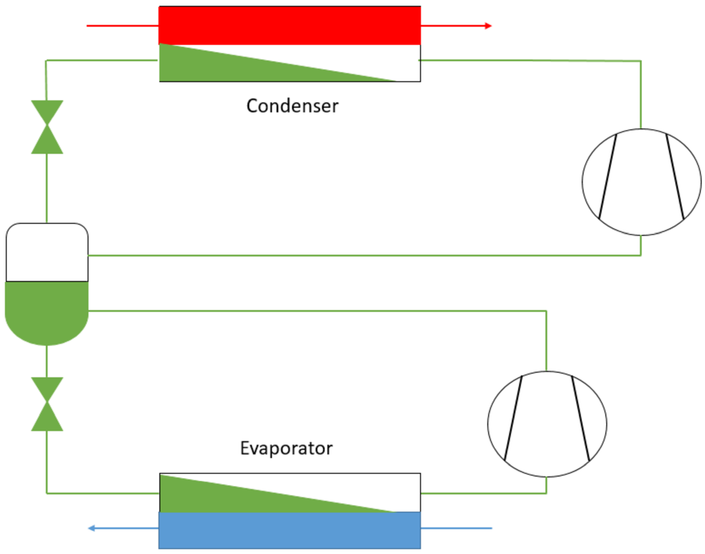

Figure 6: "Bubble through" intercooler

Two stage compression is combined with two-stage throttling. The main advantages of using two-stage throttling is the reduction in throttling losses. The higher the specific heat of the liquid after the condenser is, the larger the gain is with economizers.

Another option is to use one compressor, but still have an intermediate pressure. Using a compressor with two suction lines will also reduce the work from the compressor. This could, as mentioned earlier, be combined with either a flash tank or a heat exchanger for subcooling. Such compressor solutions have in studies shown to increase heating capacity, cooling capacity and COP of the system (Guo-Yuan and Quin-Hu 2003). The oil return problems can also be solved that way.

Whenever using an economizer the compressor power decreases, compared to the one step heat pump cycle, as the smaller flow is no longer compressed from the lowest pressure level. It is compressed from the intermediate pressure, which requires less compressor power. This increases the COP of the system (SWEP n.d).

Using a layout of the system like in figure 3, it is possible also to use one or two different refrigerants. Using one refrigerant makes the system less complex, while using two has some other advantages. The economizer refrigerant could e.g. reach a higher temperature than the main refrigerant.

3. Theory

In this chapter the theory behind the project is presented.

3.1 General

In this section, general theory useful in this project, is presented.

3.1.1 Carnot

The Carnot cycle consists of two isentropic and two isothermal processes and it is assumed to operate with infinite heat exchanger areas. In theory it is a completely reversible cycle. It is the "ideal" cycle. In the Carnot cycle the entropy difference over the two heat exchangers is equal, see figure 7 (Claesson 2004). When running clockwise in a temperature entropy (Ts) diagram, the cycle produces work while conducting heat from a source at a high temperature to a sink at a low temperature. The reverse Carnot cycle runs counterclockwise in a Ts diagram absorbing work while pumping heat from a low temperature source to a high temperature sink.

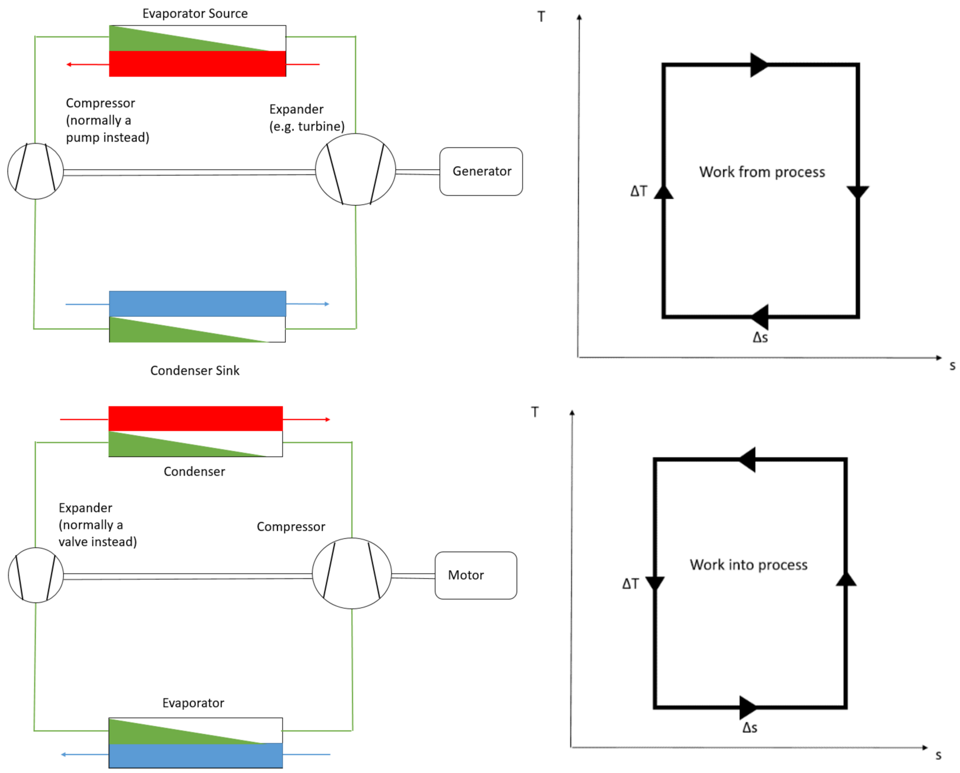

Figure 7: A Carnot cycle on the upper row and a reverse Carnot cycle on the lower

Evaluating the operation of a system, the Coefficient Of Performance (COP) is one of the most interesting factors. There is thus a theoretical limitation of the COP, due to the thermodynamic properties (Granryd et al. 2011). As shown below the theoretical limit, for a reversed Carnot cycle, depends on the condensing and the evaporation temperatures in the system (Claesson 2004) (Granryd et al. 2011).

$$COP_{1C} = \frac{T_1}{T_1 - T_2} \quad \textit{Heat ejected to sink/work in} \qquad (1)$$

$$COP_{2C} = \frac{T_2}{T_1 - T_2} \quad \textit{Heat absorbed from source/work in} \qquad (2)$$

Looking at the equations, it can be concluded that the temperature difference in the heat pump (T_1 - T_2) should be kept small in order to receive a high efficiency. But these temperatures are defined by the source and sink temperatures and can thus not be chosen arbitrarily to achieve a high COP (Claesson 2004). T_1 and T_2 are suitable for a process using an evaporating and condensing refrigerant.

3.1.2 Refrigeration process losses

The Carnot cycle is a cycle without any losses, that never occurs in reality. There are always irreversibilities of some kind, affecting the overall efficiency of the system. The Carnot efficiency is comparing the actual cycle with the Carnot cycle, it thus measures how efficient the system is compared to the Carnot cycle, see equations 3-5 (Granryd et al. 2011).

$$COP_1 = \frac{\dot{Q}_1}{\dot{E}_k} \qquad (3)$$

$$COP_2 = \frac{\dot{Q}_2}{\dot{E}_k} \qquad (4)$$

$$\eta_C = \frac{COP_2}{COP_{2C}} \qquad (5)$$

Then, which are the main losses in the system? As described above the Carnot cycle consists of two isentropic and two isothermal processes. One of the losses appear due to the temperature after the compressor is higher than the temperature would be as a result of an isentropic compression process. More compressor power is thus needed just to heat the gas. The larger power results in an extra temperature lift and a lower efficiency of the whole system. Also, since the hot gas temperature in the system is higher that the condensing temperature, the condensing process is not isothermal.

The other large loss is due to the throttling in the expansion device. Because of the throttling losses, the entropy increases over the expansion device, while instead the enthalpy remains constant. The increased entropy results in a lower system efficiency. An expander, recovering the available work there, could increase the efficiency significantly. Such expanders are however not available yet. How large the throttling loss will be depends on the refrigerant used in the system. Using two expansion devices or a perfectly functioning economizer will decrease the throttling losses and increase the efficiency of the system (Nickl et al. 2005). But, as explained in detail later in this chapter there are also other losses to consider as well in the system.

Another loss stems from the temperature differences in the heat exchangers, where entropy is generated. The last significant loss is motor and transmission losses. However, when using a hermetic compressor, as in the case, they are usually lumped together with the compressor losses.

3.1.3 Refrigerants

In the project two refrigerants are considered, though only one is used for the closest upcoming experiments. The final idea is to combine a larger and a smaller heat pump to a new solution of an economizer according to figure 3. In the large pump the refrigerant is assumed to be propane, R290, and in the small system the refrigerant is isobutane, R600a. Using propane and isobutane does however not come without complications.

Propane is a sustainable refrigerant due to its low environmental impact, and the refrigerant has very good thermodynamic properties. The refrigerant is non-toxic and has zero ozone depletion potential. The difficulty with propane is the fact that the refrigerant is highly flammable (Linde n.d.a). Therefore there are regulations and restrictions when using propane. The same goes for isobutane, R600a, as it is also flammable. It is a natural refrigerant as well, it has a low environmental impact and it is non-toxic with zero ozone depletion potential (Linde n.d.b). They are two good options, but they need to be used with caution.

3.1.4 Integrating into a building

Today utilizing a heat pump in a single family house is very common in Sweden. Heat pumps are also used for heating in larger buildings, though using district heating is still much more common. The heat pumps normally produce hot water as well as heat and sometimes for a multitude of other purposes in buildings. It depends on the system configuration in the individual building.

The hot water in our taps need to be between 50 and 60 °C, to avoid both microbial growth and scalding (BBR 2015). Generally hot water is therefore mixed with cold water in the tap, to adjust the temperature. The water temperature to the radiators is also adjusted, depending on the outdoor temperature. Cold winter days (at “DVUT”) the temperature to the radiators normally stays below 55 °C, though a few buildings need 70 °C (Stockholm Exergi n.d.). Hot water can sometimes be provided to a heating coil for ventilation, heating the incoming outdoor air. Logically, the more heat produced by the heat pump for heating the more hot water can also be produced. Implementing this economizer solution will increase the hot water production. In summertime the heat pumps normally only produces hot tap water, while sometimes also providing cooling for AC systems. Older heat pump installations, could not achieve the maximum power needed for both heating and hot water thus needing assistance from backup heaters midwinter. Modern installations through, with speed regulated heat pumps, normally avoid backup heaters for other than safety reasons.

Implementing the system solution studied in this project, figure 3, the available temperature from the small economizer system is higher than for the stand alone heat pump itself. As the system will operate with a high evaporating temperature, it is also possible to reach a high condensing temperature. The high condensation temperature in turn providing facilities with warmer tap water, at least from the economizer part of the pump. Having two cycles, each with a lower temperature lift, instead of having one with a higher temperature lift will increase the COP for the system.

3.1.5 Integration into the world

Heat pumps can be considered a rather environmentally friendly solution, depending on the source for electricity production. A smaller amount of electricity input results in a larger amount of heat. The electricity production in Sweden has a rather low environmental impact, 13 g CO_2 per kWh (Energiföretagen 2017). In other countries this number is usually higher.

Some advocates a “marginal electricity” approach. They claim that an increased amount of heat pumps electric consumption will always lead to a higher electricity demand, thus assuming that direct electricity heating is not replaced. It is then very important to consider how this further demand will be produced in the world. Will the electricity come from installed renewable power plants or will the marginal new production stem from not as environmentally friendly sources? However other marginal effects should also be taken into account. If heat pumps are not used, what other sources will replace them and what marginal effect will that result in.

The main factors affecting the future success of the economizer technology is the economy. The technology must be economically sound to compete, it must increase both the capacity and the total efficiency. There must however be enough to gain from the technology, when comparing to the investment and operational costs. E.g. the payback time for a heat pump with the economizer solution should not be larger, as compared to the original heat pump without economizer. Heat pumps with economizer are generally not intended for single family houses, but for larger buildings such as apartment buildings or office buildings.

3.2 Calculations

Below the theory used in this project is presented, along with tools used for the models.

3.2.1 Compressor model

The input to the refrigerant system is the amount of electricity consumed by the compressor, which is estimated as the mass flow multiplied with the enthalpy difference over the compressor (Granryd et al. 2011). Apart from that there are inputs to e.g. pumps and the control system. The compressor power will depend on the refrigerant used in the system and the states before and after the compressor.

$$\dot{E}_k = \dot{m_{ref}} \cdot \Delta h \tag{6}$$

For the compressor there are two important efficiencies, the isentropic and the volumetric. The volumetric efficiency is the achieved volume flow divided by the displacement volume flow. The isentropic efficiency is the ratio between the ideal power needed to compress the refrigerant between the two pressures and the actual power used. There are several losses, which is why the electricity consumption always is greater than the compressor power needed. Apart from the friction losses in the compressor and refrigerant there are e.g. electrical resistance losses that have to be included when dealing with hermetic compressors, as in this case. Below in equation 7-9 it is shown how the two efficiencies are used for calculations (Granryd et al. 2011).

$$\dot{V}_{2k} = \eta_v \cdot \dot{V}_s \tag{7}$$

$$\dot{m}_{ref} = \dot{V}_{2k} \cdot \rho_{2k} \tag{8}$$

$$\dot{E}_k = \dot{m_{ref}} \cdot \frac{\Delta h_{is}}{\eta_{is}} \tag{9}$$

3.2.2. Heat exchanger models

The main purpose of a heat exchanger is to transfer heat from one fluid to another as efficiently as possible. In the refrigeration system there are a number of heat exchangers, at least a condenser and an evaporator, but other heat exchangers might be included too. The refrigerant passing through the heat exchangers is either giving off heat to or absorbing heat from the other side. Below in equation 10-13 are three options to calculate the exchanged heat. Inputs are the refrigerant mass flow, the enthalpy difference, the temperature difference and the specific heat capacity. The equations can be applied to either of the fluids in the heat exchanger (Havtun, Bohdanowicz and Claesson 2018). Though, equation 11 does not apply to the evaporating and condensing processes in the refrigeration cycle, since the heat transferred results in phase change and not temperature difference.

$$\dot{Q} = \dot{m_{ref}} \cdot \Delta h \tag{10}$$

$$\dot{Q} = \dot{m_{ref}} \cdot c_p \cdot \Delta t \tag{11}$$

$$\dot{Q} = UA \cdot LMTD \tag{12}$$

$$LMTD = \frac{(T_{11} - T_{22}) - (T_{12} - T_{21})}{\ln \frac{T_{11} - T_{22}}{T_{12} - T_{21}}} \tag{13}$$

The first index denotes the fluid hot (1) or cold (2) and the second inlet (1) or outlet (2). A design of a suitable exchanger that should be used, how large areas that are needed and what heat transfer coefficient that can be achieved, must be calculated. The epsilon-NTU method, presented in equations 14-19 below, is one way to estimate the required UA value in a certain case, it is a way to estimate the needed size of the heat exchanger. Epsilon is a measure of how much heat will be transferred compared to the maximum heat available (Havtun, Bohdanowicz and Claesson 2018). Equation 18 is accurate for counter flow heat exchangers, and equation 19 when Cr=0, when phase change is occurring in the heat exchanger.

$$C_{min} = \min\,(\dot{m}_c \cdot c_{p,c}, \dot{m}_h \cdot c_{p,h}) \tag{14}$$

$$C_{max} = \max\,(\dot{m}_c \cdot c_{p,c}, \dot{m}_h \cdot c_{p,h}) \tag{15}$$

$$C_r = \frac{C_{min}}{C_{max}} \tag{16}$$

$$NTU = \frac{UA}{C_{min}} \tag{17}$$

$$\varepsilon = \frac{1 - \exp\,(-NTU(1 - C_r))}{1 - C_r \cdot \exp\,(-NTU(1 - C_r))} \tag{18}$$

$$\varepsilon = 1 - e^{NTU} \tag{19}$$

$$\Delta T = \varepsilon \cdot \Delta T_{in} \tag{20}$$

The epsilon-NTU method is a good tool during the planning of the system, for selection of components. It is important that the heat exchanger has the capacity to transfer the requested amount of heat from one fluid to another. If e.g. the heat exchanger area is too small or the heat transfer coefficient is too low, problems might occur.

3.2.3 Refrigerant program and tools used

Tools that have been used in this project are CoolProp with add-ins into Excel. CoolProp evaluates the state properties of a refrigerant. With two inputs you can get a third. For example, knowing temperature and pressure the enthalpy can be obtained. The program is very useful in this project and for any calculation including refrigerants. Excel has been used for calculations and simulation of models. All measured data have been imported into excel, for further evaluation.

Another program used in this project, useful for a rather quick estimation of the performance of a cycle, is the program Refrigeration Utilities. In this program cycles can quickly be laid out, with temperatures and refrigerant, and e.g. the COP is calculated.

3.2.4 Efficiencies using different refrigerants in economizer

The efficiency of the system is depends on several factors, two examples being temperatures and refrigerant type in the system. In order to see the difference between different refrigerants and temperatures the program Refrigeration Utilities was used, to draw cycles and conduct a quick evaluation. Only simple one stage cycles were then considered. From such a rough estimation it can e.g. be concluded that the higher the temperature lift is the lower the COP, for the same refrigerant, exactly as the inverse Carnot formula predicts.

Also, a comparison between different refrigerants was thus evaluated. The operating conditions were then held the same using different refrigerants. It is clear that the refrigerant itself has an important role for the losses in the cycle.

The two refrigerants propane and isobutane are used in this project and therefore of especial interest. They were chosen as they both had a very low GWP combined with a good efficiency. Different refrigerants are more or less suited for a certain application considering not only the efficiency, but also e.g. stability, toxicity, flammability, cost, GWP, ODP and compatibility with components in the loop.

3.2.5 Internal heat exchanger

Using an internal heat exchanger in the heat pump, between the gas stream from the evaporator and the liquid from the condenser, will lead to increased superheating and subcooling in the system. The warm fluid from the outlet of the condenser will give off heat to the colder gas from the outlet of the evaporator, see figure 9. In the figure the effect of the internal heat exchanger are marked red. The temperature at the inlet to the expansion valve will therefore be lower than it would be without the internal heat exchanger and the temperature at the inlet to the compressor will also be higher. This will also result in a higher outlet temperature from the compressor.

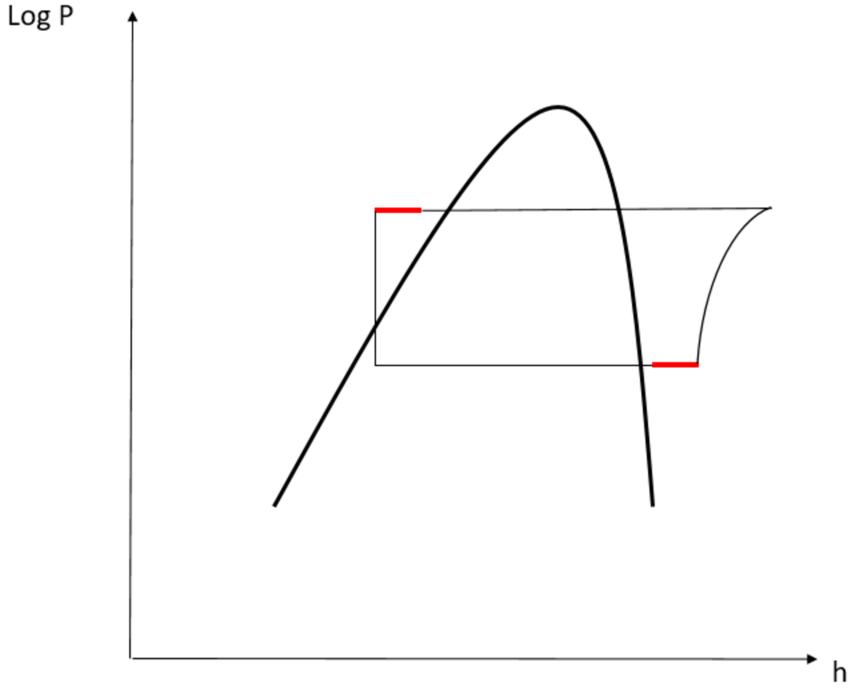

Figure 9: Effects of an internal heat exchanger

Why is the internal heat exchanger beneficial in this case? The internal heat exchanger will help make certain that sufficient superheat is reached. This is important especially for cycles using isobutane, as the outlet from the compressor must not condense during compression. Also, the COP of the system increases with an increase in suction gas temperature (Danfoss 2000). But, one of the main reasons why the internal heat exchanger is needed is to avoid dilution of the oil. If the temperature of the incoming refrigerant is too low it will dilute the oil. The viscosity of the oil would then lower, endangering both the bearings and the sealing of the compressor.

3.2.6 Expected results for economizer loop

System temperatures are expected to be relatively high, because of the high evaporating temperatures. The water loop through the condenser is also expected to often run with high temperatures, as one of the goals is to produce hot DHW. The economizer thus economizes generation of DHW at high temperatures while still achieving a good COP for the main loop and the capacity is linearly dependent on the mass flow. As the compressor is speed regulated the mass flow can be easily varied. Another expected result is that the system will have a small refrigerant charge, one of the main goals in the project. The specific weight of isobutane liquid is much lower than most refrigerants. A low mass of flammable refrigerant, below 150g, will enable the installation of this economizer heat pump in an arbitrary location.

4. Practice

In this chapter a description of the system tested in this project is presented. This includes the components for the heat pump, the measuring devices and the final layout of the whole test system.

4.1 Excel model

In order to evaluate the possible benefits of the economizer system before the construction of the heat pump simulations were made in an excel model, created by Professor Eric Granryd. In this model the economizer heat pump was connected to a main propane heat pump. After that, the economizer heat pump was analyzed as disconnected from the main heat pump. The test runs in the lab of the economizer heat pump is only connected to two water circuits. Important assumptions used for the simulations are stated in table A1 in appendix A. Simulations were made for the same operating conditions as described below in table 1. The values from the simulations were helpful deciding e.g. pressure gauges and the amount of refrigerant to be used. Suitable mass flows for the two water circuits could also be determined as the heat exchange of the condenser and the evaporator was included in the simulation. The same amount of heat can e.g. warm up a lot of water just a few degrees or alternatively heat less water with a larger increase in temperature.

As previously mentioned the excel model already existed. In the model the parameters set during each simulation was the speed of the compressor, water outlet temperature from the condenser and the evaporating temperature. Looking at the condenser, the temperature difference between water inlet and outlet was set to either 5 degrees, as asked for by project members. The UA value was assumed to be 1.5 kW/K. Using the epsilon NTU method presented above the condensing temperature was calculated.

Worth noticing for the evaporator, the superheat was set to 3 K. This turned out to not be possible for the experiments due to fluctuations. Therefore the superheat was set to 4 K during measurements. For the evaporator the temperature difference between water inlet and outlet was set to either 5 or 20 degrees, resulting in a certain possible mass flow of water. Using a temperature difference at the outlet of 1 degree, and the LMTD value the UA value for the evaporator could be evaluated.

For the internal heat exchanger the epsilon NTU method was used, together with an assumed UA value of 40 W/K. The isentropic efficiency was assumed to be 65% and the volumetric 95%.

4.2 Description of heat pump system to be tested

As described above, the evaporator in the small economizer system is intended to subcool the refrigerant after the condenser in a much larger heat pump. The only system considered experimentally in this project is though the small economizer heat pump. It is just tested with water in both the condenser and the evaporator loop in order to get a general understanding of the system behavior during different operating conditions. Table 1 below states the operating conditions tested for the system. Totally 27 cases are considered, combining the different parameters. For three desired water outlet temperature from the condenser three different evaporating temperatures were tested, and each of them at three different compressor speeds.

The design of the heat pump mockup also falls in line with the project goal to keep the refrigerant charge low. After some deliberation and cooperation the layout in figure 10 was decided. The components are placed so that all liquid pipes get short. That reduces the amount of refrigerant. More pictures of the test facility can be found in appendix B. The layout of the test rig is presented in figure 11.

Table 1: Operating conditions for economizer system

Water condenser outlet [°C]	**T_2 [°C]**	**Compressor speed [rpm]**
50	15	2000
60	25	4000
70	35	6000

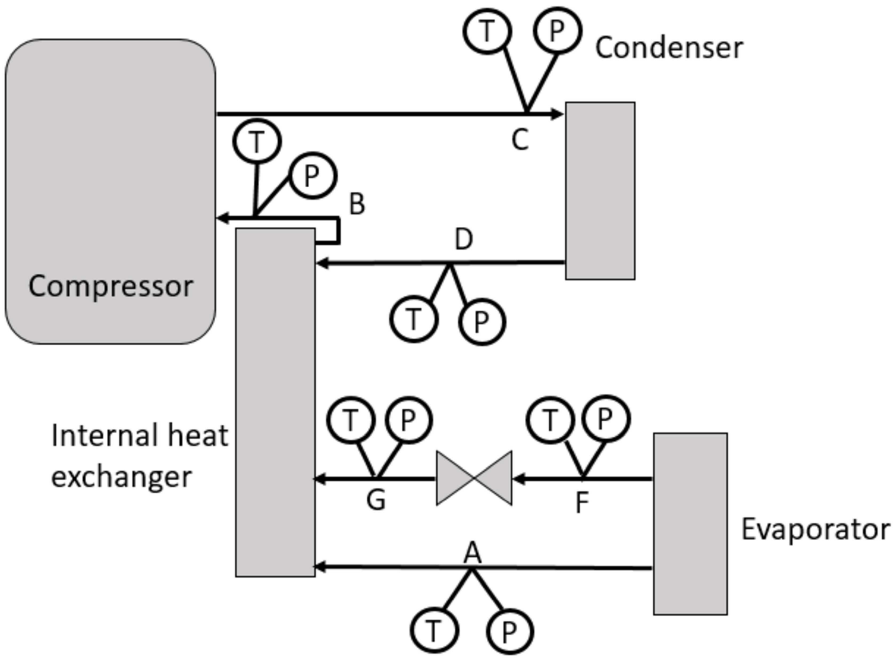

Figure 10: Heat pump layout

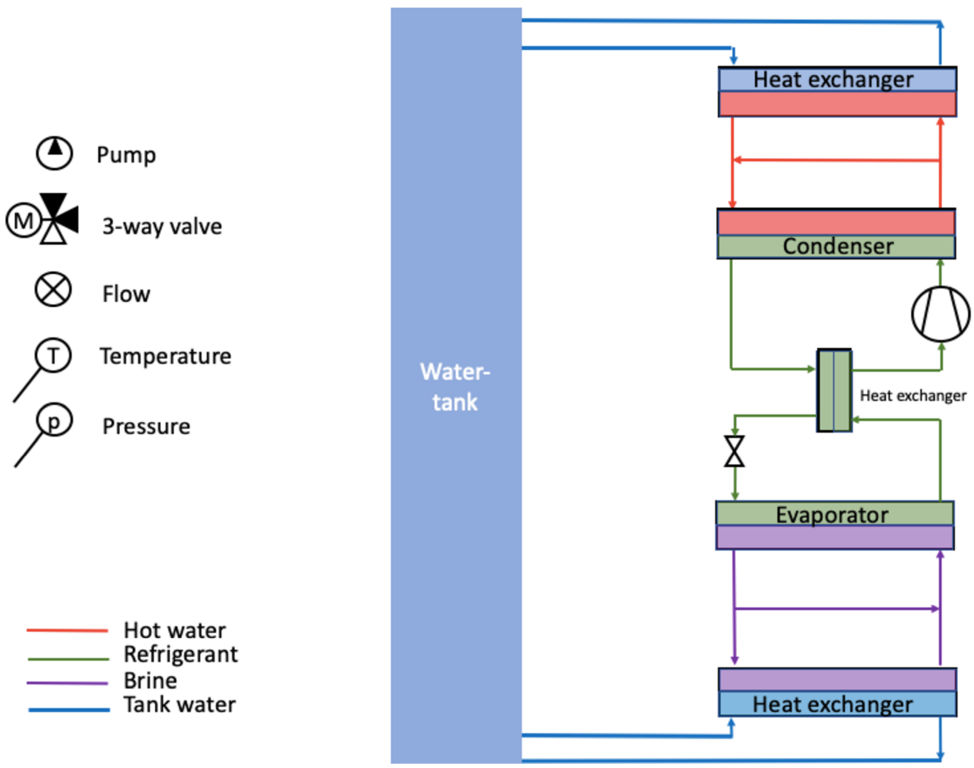

Figure 11: System layout

4.3 Compressor with speed control

The heat pump designed in the lab contains a compressor from Sanden. It runs between up to 8500 rpm, in this project it was used as described in table 1. It is a semi hermetic scroll compressor, primarily designed for electrical cars. It has a built in volume ratio of approximately 1.7 and the displaced volume is 33 cm^3/rev. The isentropic and volumetric efficiencies determined, varies with speed, pressure ratio and temperature of the suction gas. Important for the mounting was also to make sure the compressor had the ability to vibrate, therefore it was placed on rubber feet. Flexible hoses for the refrigerant were used on both suction and discharge side of the compressor to avid copper hardening of the piping. Oil used for the heat pump is Emkarate ester oil RL 32H.

4.4 Heat exchangers

The heat exchangers used for the heat pump are presented below.

4.4.1 Condenser and Evaporator

Both condenser and evaporator were produced by Alfa Laval, the two heat exchanger are prototypes. Information from the company included the internal volume of the two heat exchangers. The internal volume of the condenser is 340 cm^3 and of the evaporator 380 cm^3. Both were mounted vertically. For the condenser the condensing process is downwards, and the evaporation in the evaporator process upwards.

4.4.2 Internal Heat Exchanger

The internal heat exchanger in this project is a very important component. The internal liquid side volume has been kept small to minimize the refrigerant charge. For this project a heat exchanger constructed by one of the project members is used. It is a symmetric counter flow heat exchanger.

4.5 Expansion valve

The expansion device used is the E2V-Z from Carel. It is an electronic expansion valve with a stepper motor, controlled by a regulator to regulate the superheat after the evaporator. The device includes temperature- and pressure sensors.

4.6 Measuring devices

Below, the measuring devices used during the project are presented. An attempt to measure the compressor power was made, but since the results were inaccurate the measurements and devices used for the measurements concerning the compressor were excluded from this thesis.

4.6.1 Flow

The water flow to the condenser and evaporator is controlled by two pumps. As the minimum flows for the pumps are too large, the flow is also controlled by a throttle valve. Two flow meters were used, for the flow of the water to both the condenser and the evaporator. The flow meters are magnetic flow meters and configured to measure flows in the span 0-6 m^3/h. Both meters are made by YOKOGAWA, model RFX025G.

4.6.2 Temperature

Thermocouples were used for temperatures, put inside the pipes in order to measure the temperature accurately. For some pipes the diameter was very small and the meter would occupy a rather large part of the cross section area, therefore the thinnest thermocouples available in the lab were used for these measurement points.

4.6.3 Pressure

For the absolute pressure, absolute pressure transmitters from YOKOGAWA were used. Since there will be two pressure levels, one higher pressure in the condenser and one lower in the evaporator, two transmitters were ordered. The absolute pressure transmitters used in this project can measure pressures in the span 0.1-20 bar, though they can be programmed to measure in a smaller span. After the evaluation of the system they were set to measure in the span 0-15 bar on the high pressure side and the other 0-5 bar on the low side. The accuracy of the pressure transmitters is 0.055%, which results in an accuracy of 0.011 bars for both meters. Since the meters have been set to measure in the span 0-15 and 0-5 bar, this results in an accuracy of 0.073% and 0.22% respectively. The pressure transmitter used is EJA510E-JBS7N-014EN/KU22.

4.6.4 Speed of compressor

It is possible to regulate the compressor speed via a control panel. That is a part of the compressor solution, which came with the device. In that way the speed can be adjusted according the measurement points presented in table 1.

4.7 Description of test facility

A description of the test facility is presented below.

4.7.1 Calibration of test facility

Before the construction of the heat pump the already existing test facility was tested to make sure it was functioning properly. In the beginning of the project another heat pump, from a previous project, was removed from this test facility. The remaining parts of the already existing facility then consisted of a water tank and the two water circuits, one to the evaporator and the other to the condenser. For both circuits there were pumps, flow meters, temperature regulated three way valves and temperature sensors. The tank was used for blending the hot and cold water and for cooling the compressor power to the room air of the laboratory using an air coil.

An electric heater was connected to the circuits, for calibration. The power generated by the heater to the circuit was measured. Using the measured power and comparing it to the absorbed heat in the circuit it could be evaluated if the existing test facility was functioning properly. The absorbed heat was calculated using the volumetric flow, the density, the temperature difference in the circuit and the specific heat capacity of water, see equation 11.

After test runs, measurements and calculations for both circuits it was determined that the generated heat did not differ much from the absorbed heat in the circuit, only around 5% at most. However important to keep in mind is the fact that the pipes at this time were not insulated, likely resulting in smaller heat losses. Also, the tests showed that when the electric heater is switched on the system takes some time to stabilize. This stabilization time was kept in mind during the further testing of the heat pump. Tests were conducted and it was concluded that the system already in place functioned well, since the differences in generated and absorbed heat was limited to a few percent.

4.7.2 Piping layout

Figure 10 shows how the heat pump was built. Pipe sizes were determined after consultation with participants within the research project. Two extra important pipes are the ones connected to the compressor. Since the compressor vibrates when the pump is running, it was important to make sure that the pipes would be able to handle the vibrations. Therefore more flexible pipes were used between the compressor and the condenser and between the compressor and the evaporator. From the outlet from the condenser to the inlet to the evaporator the pipe diameter is 4 mm. From evaporator to internal heat exchanger 8 mm, internal heat exchanger to compressor 11 mm and lastly from the compressor to the condenser 7 mm.

4.7.3 Control

The control of the system consists of two PID regulators controlling the two three way valves, from ESBE, thus regulating water temperatures on both sides. The two valves control the outlet temperature from the condenser and the inlet temperature to the evaporator. Water from the heat pump heat exchangers circulates through the whole system, se figure 10. The desired outlet temperatures from the condenser are stated in table 1. As the PID regulators cannot directly control the evaporating temperature, the inlet temperature was regulated so that the evaporating temperature comes as close to the values stated in table 1 as possible. As described earlier, the compressor speed was regulated by using a control panel belonging to the compressor. Through the control devices described above the operating conditions for the heat pump were changed and the system's function was tested according to table 1.

4.7.4 Measuring devices

In figure 12 the layout of the system, including points for measurements, is presented. All thermocouples were used at the same time along with the two absolute pressure meters. So, the pressures were not measured for all measurement points laid out in figure 10 at the same time. The pressures were only measured in points B and C, before and after the compressor. The rest will be used for future measurements.

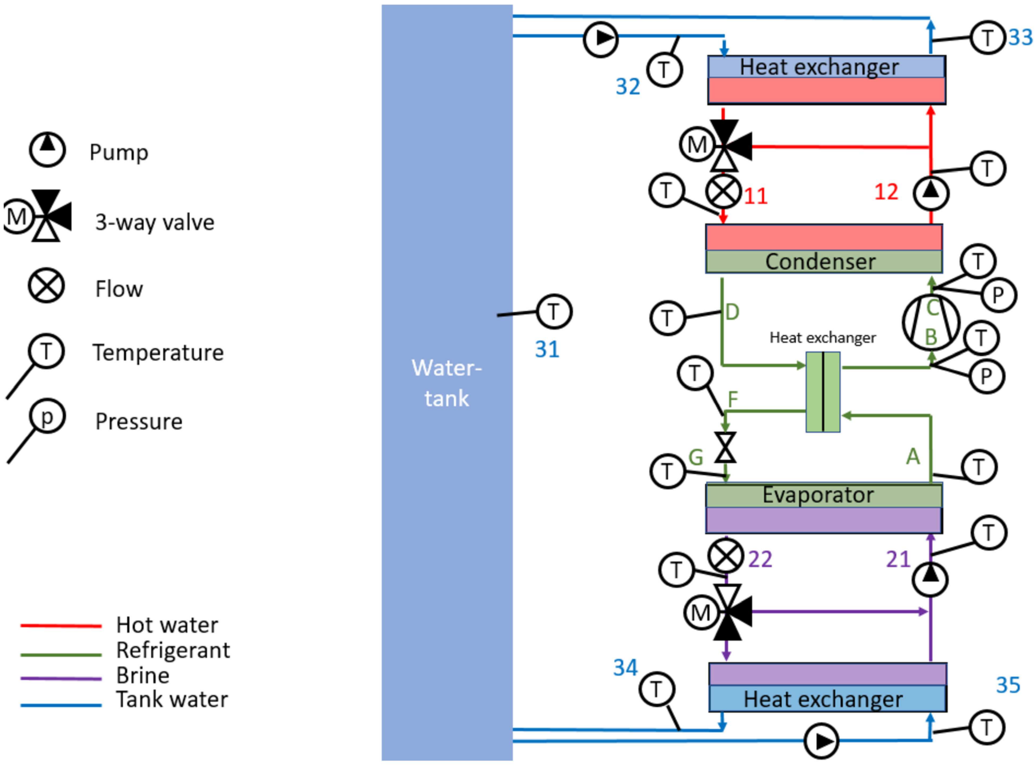

Figure 12: Measurement points

5. Results

Below the results from the experiments are presented. One thing to keep in mind is the fact that the mass flow in the system is derived from the heat balance over the evaporator to be able to evaluate the system properly. Using the heat balance over the condenser might give a different outcome, this was not evaluated further. For all measurements the water flows in the evaporator and the condenser have remained constant, around 0.23-0.25 l/s. This differs from the simulations. During measurements the water flow was kept constant resulting in different temperature differences over the heat exchanger. The higher the compressor speed, the large the change in temperature for the water circuit.

5.1 Evaluation method

Below the results from the simulations and the test runs are presented. The results are then compared in the discussion, thus evaluating the system under different operating conditions. Some of the results from the measurements were inputs in the excel model. Therefore, the comparison is performed where possible. Though, worth observing that the inputs in excel model, such as efficiencies, affect the overall results for the simulations. To start with the system was filled with 120g isobutane. It turned out to be a quite good amount of refrigerant to use for the first set of tests, resulting in minimum subcooling of around 3 degrees. All tests were run with 120 g isobutane and 50 ml oil in the compressor. As mentioned earlier the exact temperatures in table 1 were not achieved, but as close as possible. Then results were processed. Since the conclusion was made that it would ease the comparison if the two temperatures were the same for all cases, a small model was created in excel. This small model is built on a connection between the for example heating capacity and the two temperatures, for evaporation and condensing process, using either method of least squares or the trend line function for graphs in excel.

For the presentation of results below the measurements are marked with an M and the simulations marked with an E. The values 50, 60 and 70 represent the water outlet temperature from the condenser.

5.1.1 Heating power

As seen in figures 13-15 the heating capacity increases with speed of the compressor. Also, for the same water outlet temperature from the condenser the produced heat increases with a higher evaporating temperature. It can also be seen that the higher the water outlet temperature the lower the capacity. The trend is the same for all three speed for the compressor. Though, the values from simulations are somewhat different but the trend is the same. In figure 16 the measured heating capacity is plotted against the simulated. It can be seen that the values are on a somewhat straight line. Though, the higher the capacity the larger the deviation.

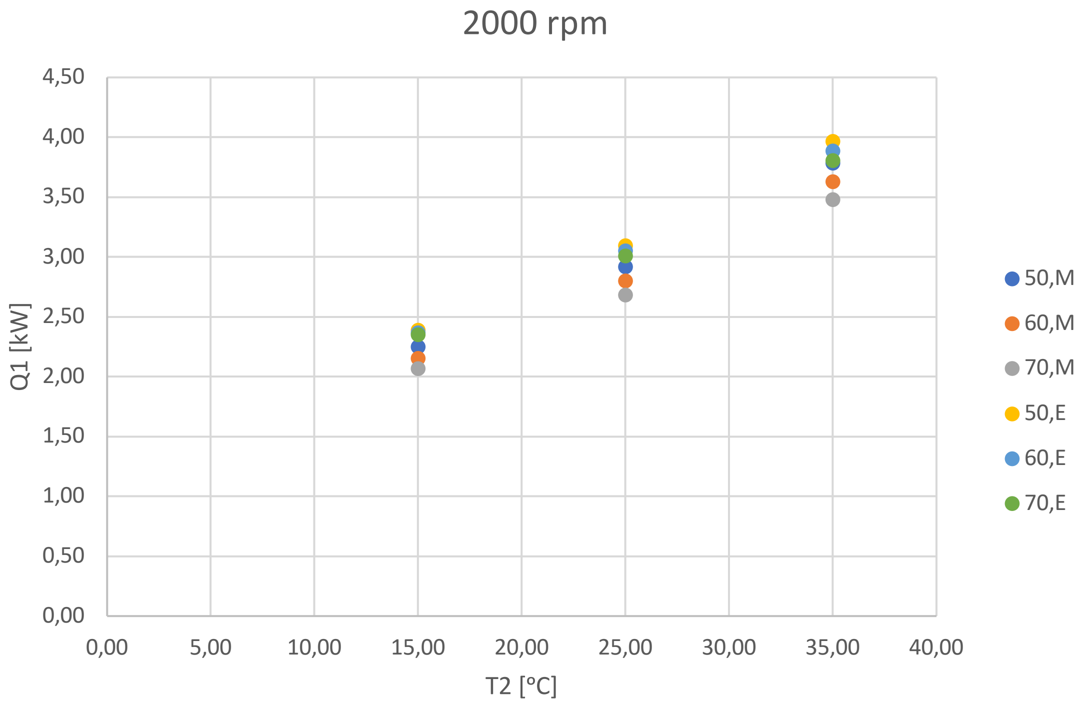

Figure 13: Heating power for 2000 rpm

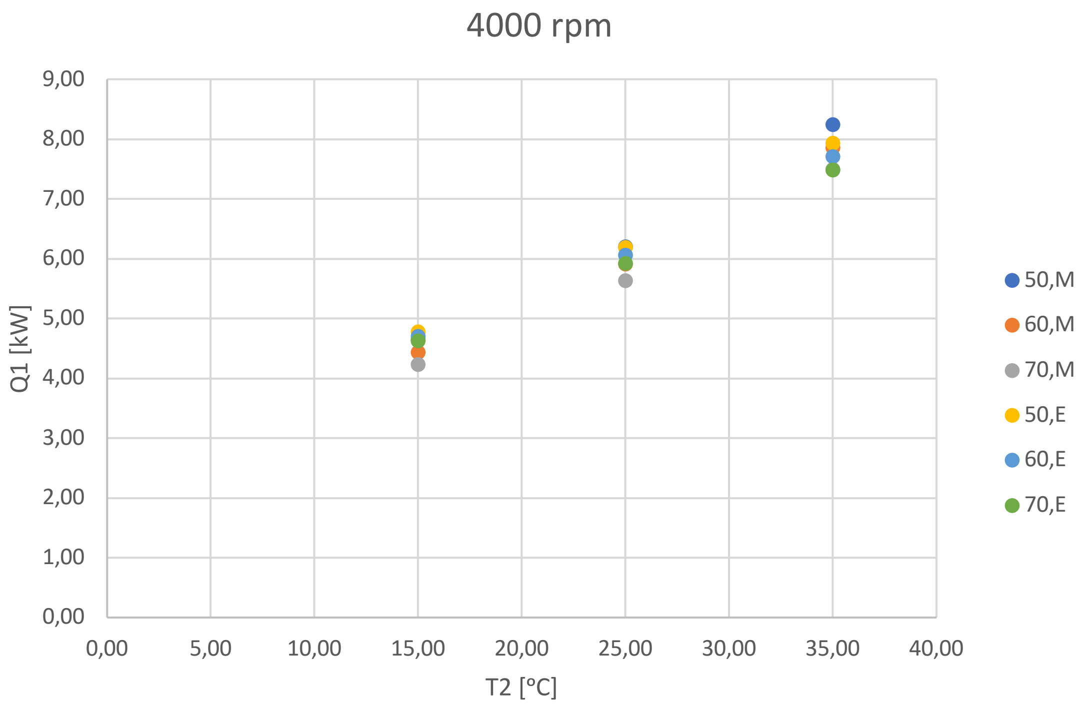

Figure 14: Heating power for 4000 rpm

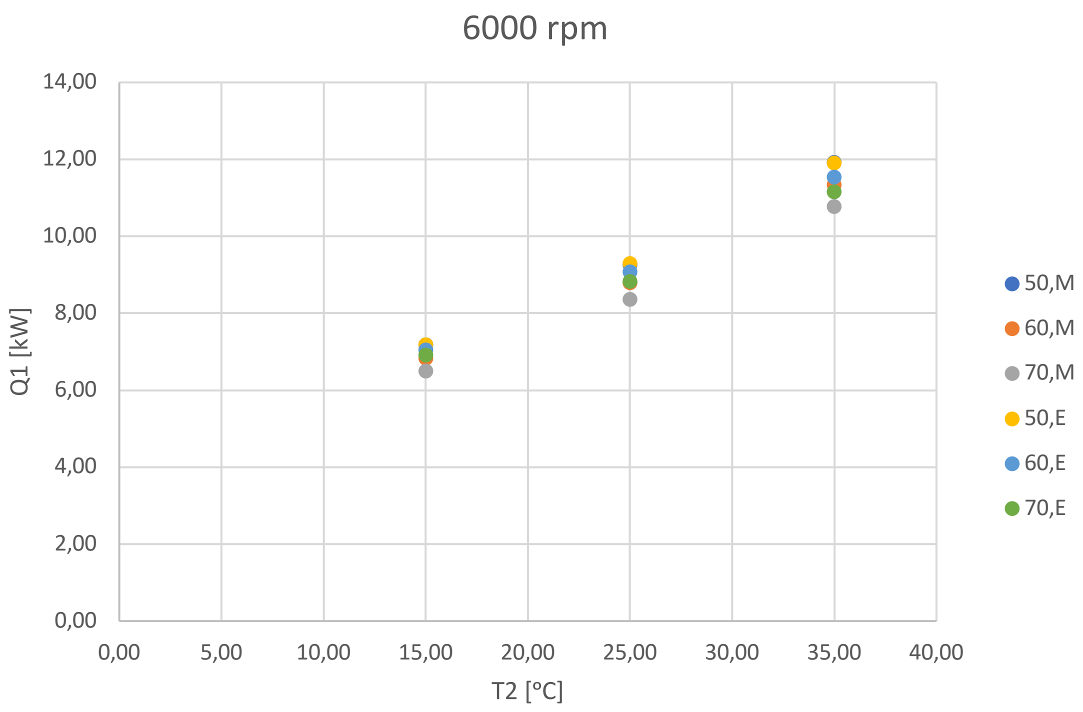

Figure 15: Heating power for 6000 rpm

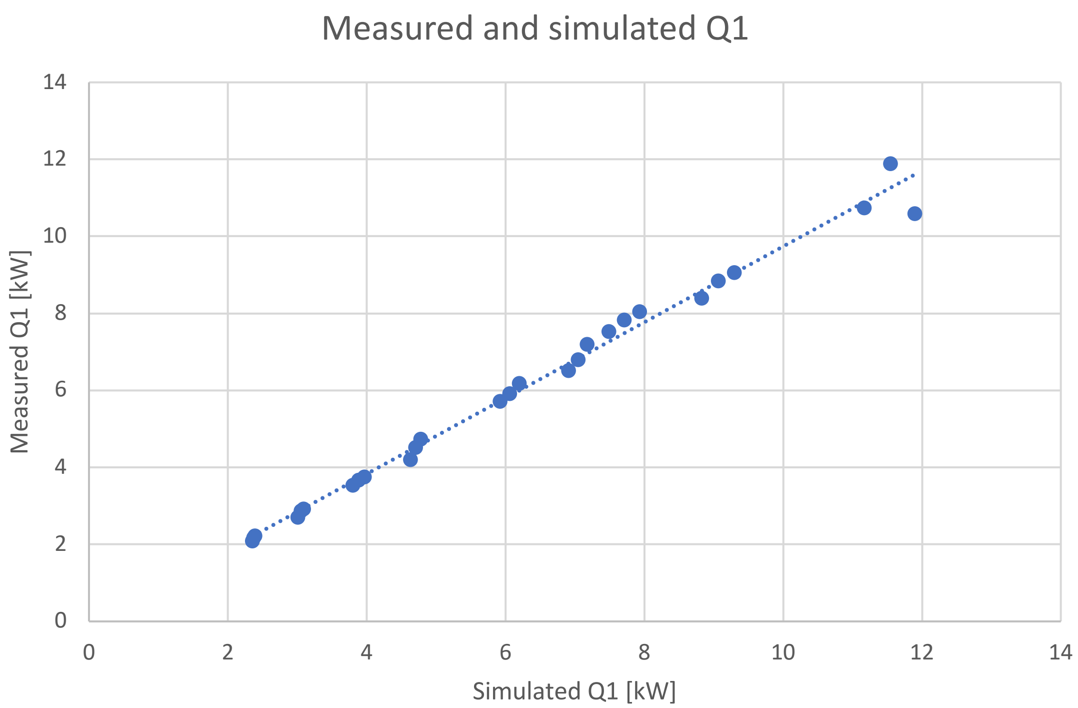

Figure 16: Measured and simulated heating capacity

5.1.2 Electric power

After evaluation of the measured compressor power and comparison to the calculated value, it clearly showed that the measured values were way too low to be accurate. Therefore the measured compressor power is excluded from this report. Since the measured compressor power was inaccurate, the results presented for the compressor are based on the mass flow and the enthalpy difference over the compressor. It can in figures 17-19 be seen that for the same evaporating temperature the compressor power is larger for a higher condensing temperature. With an increased evaporating temperature, for the same condensing temperature, the measured compressor power increases for both 4000 and 6000 rpm. For 2000 rpm the trend differs, figure 17. For these cases the compressor power decreases with a decreased temperature difference. For all speeds the trends are similar for the measurements as for the simulations, though the values differ. For the simulations the volumetric efficiency was assumed to be 95% and the isentropic efficiency 65%. One of the main differences thought is the very low compressor power for a temperature at the condenser outlet of 50 °C and an evaporating temperature of 35 °C. This applies for all speeds of the compressor. Differences between the measured and the simulated values can also be seen in figure 20. There are some larger deviations between the two sets of data.

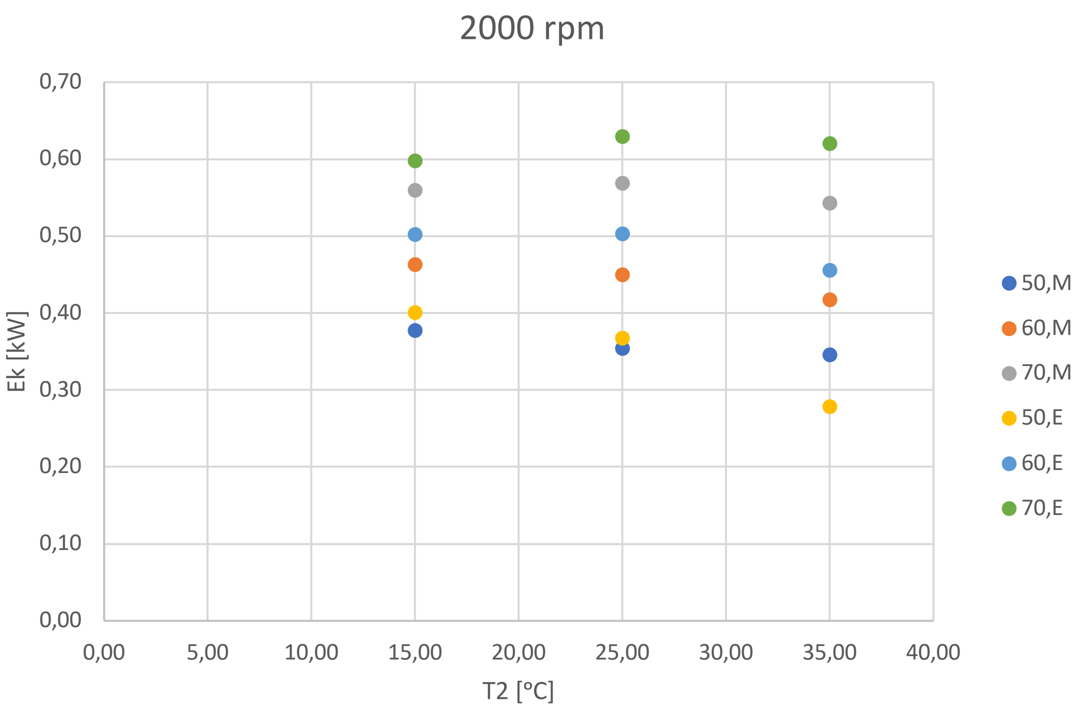

Figure 17: Compressor power for 2000 rpm

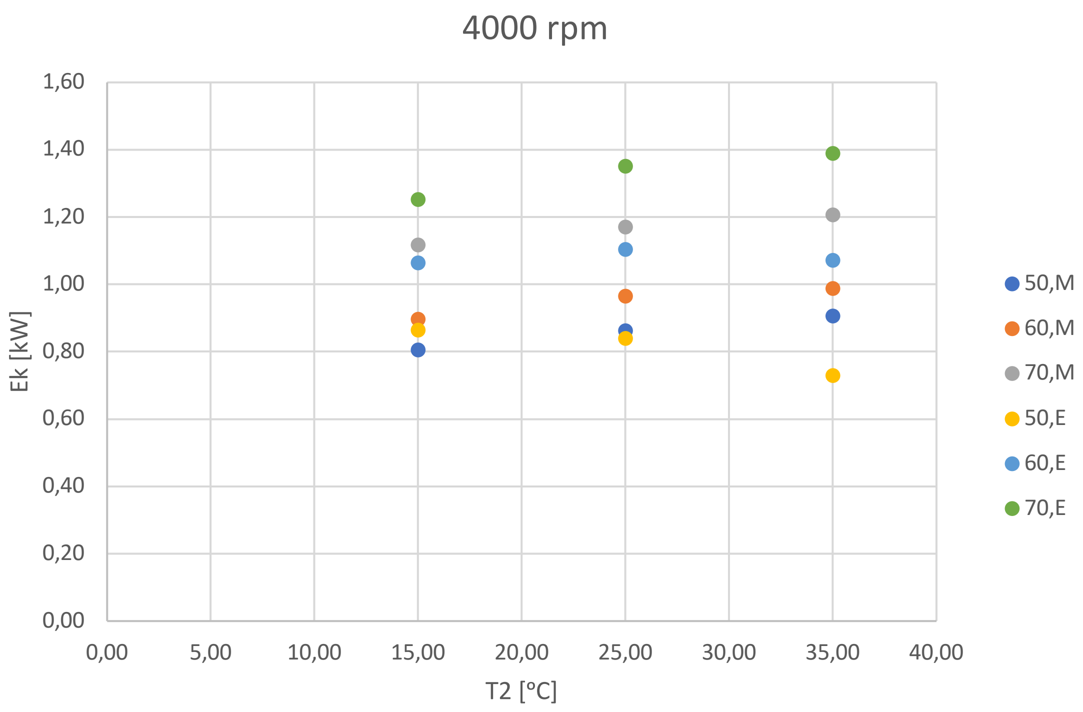

Figure 18: Compressor power for 4000 rpm

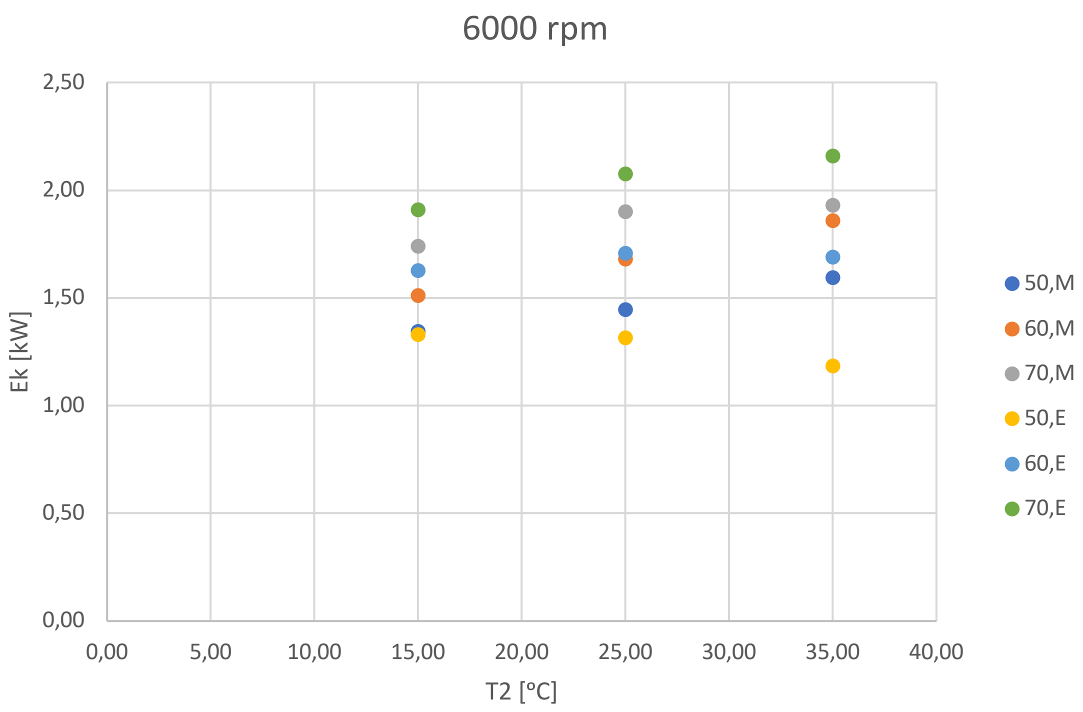

Figure 19: Compressor power for 6000 rpm

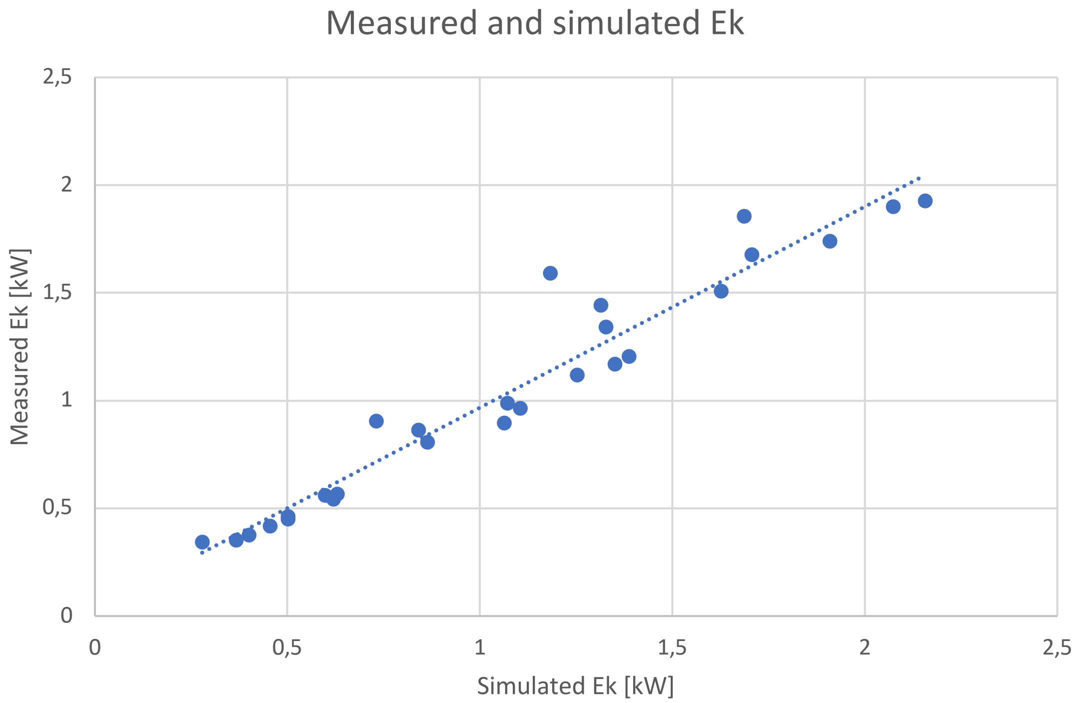

Figure 20: Measured and simulated compressor power

5.1.3 Cooling power

For the cooling capacity the trend is the same as for the heating power. The capacity increases with speed as well as a higher evaporating temperature. Also, the capacity is larger for the lower condensing temperature at the same evaporating temperature. As for the heating capacity, the trend of the simulations is the same as for the measurements for the cooling capacity. Results for cooling capacity can be seen in figures 21-23. In figure 24 the measured cooling capacity is plotted against the simulated. As for the heating capacity and the compressor power there are some deviations, but they are quite small.

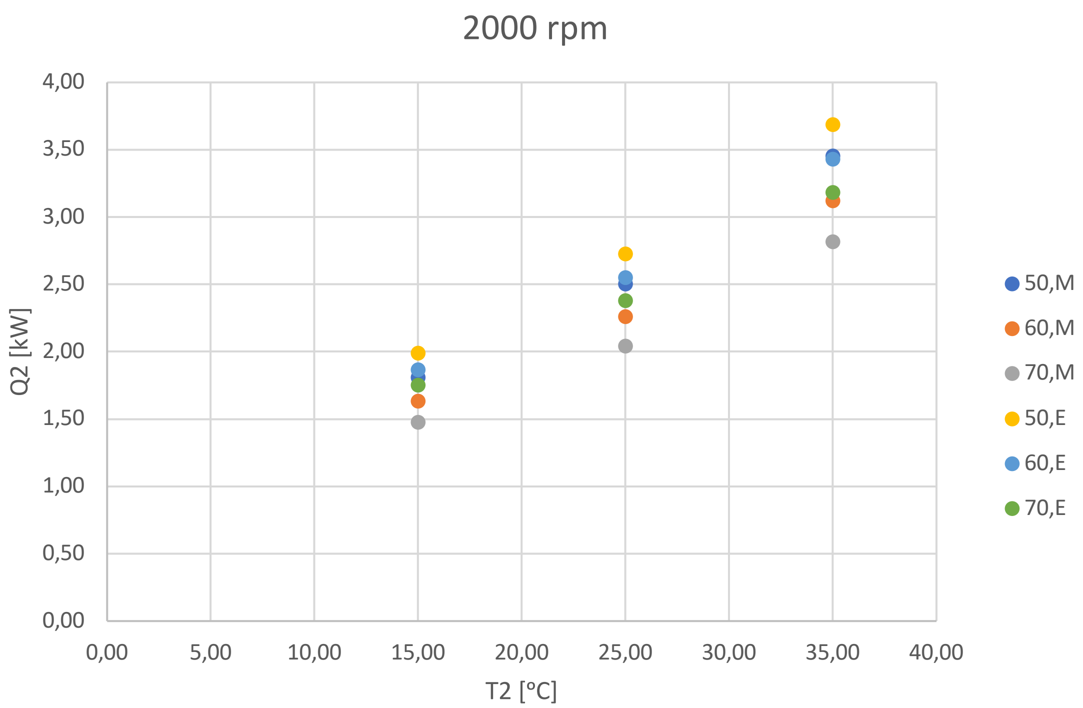

Figure 21: Cooling power for 2000 rpm

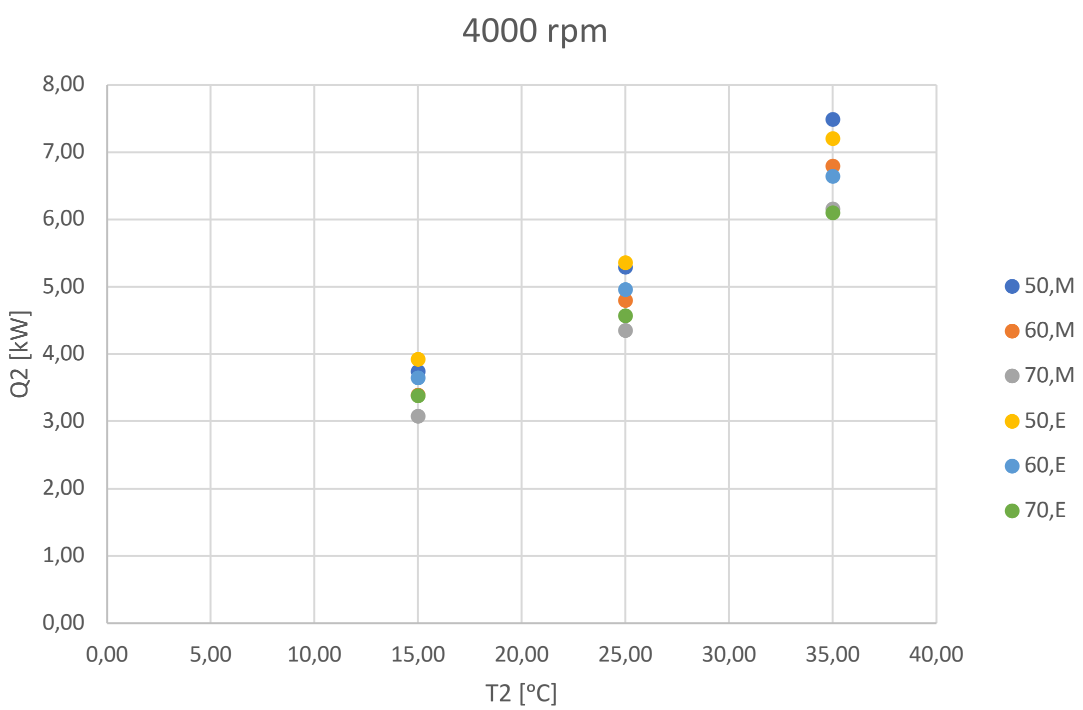

Figure 22: Cooling power for 4000 rpm

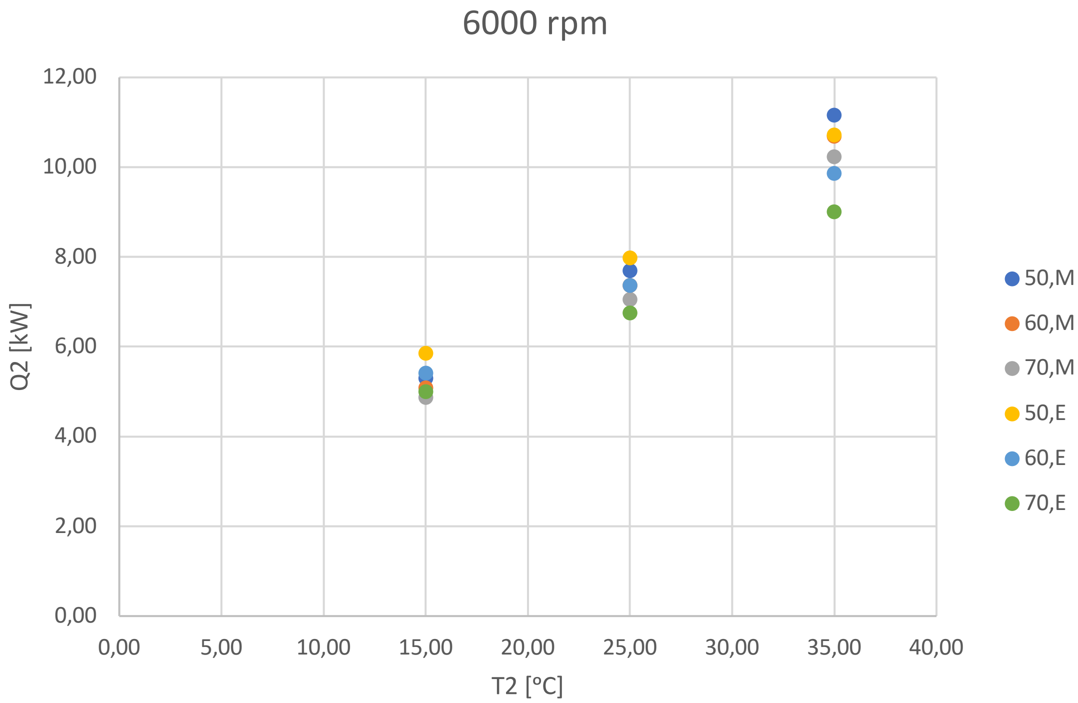

Figure 23: Cooling power for 6000 rpm

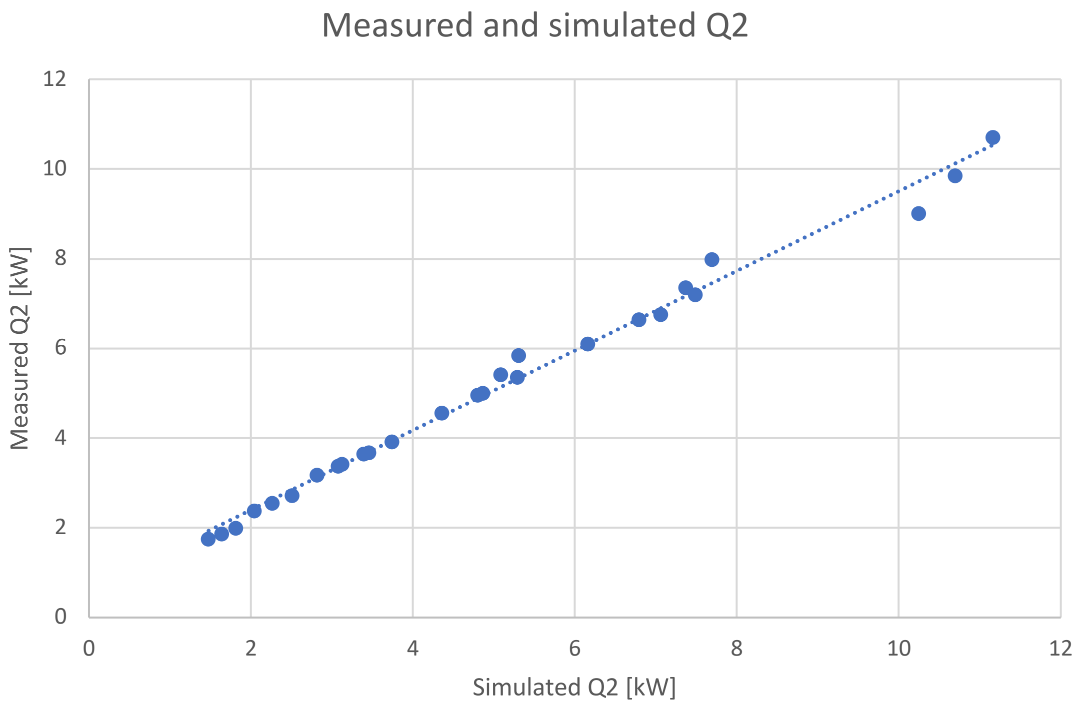

Figure 24: Measured and simulated cooling capacity

5.1.4 Heat balance check

A heat balance for the system shows the result below in figure 25, using Q_1, Q_2 and E_k from the isobutane system. The mass flow was calculated from the heat balance over evaporator, and then used for calculating Q_1 and E_k. The deviation decreases with higher heat capacity. The produced heat minus the cooling and the compressor power should in the "ideal case" be zero. Though, for all real cases it will not be zero, because of losses. The results show that for most cases the produced heat is even larger than the sum of the cooling capacity and the compressor power. In figure 25 it can be seen that the deviation is largest for the lowest speeds. In figure 25 the result from the heat balance is divided by the compressor power.

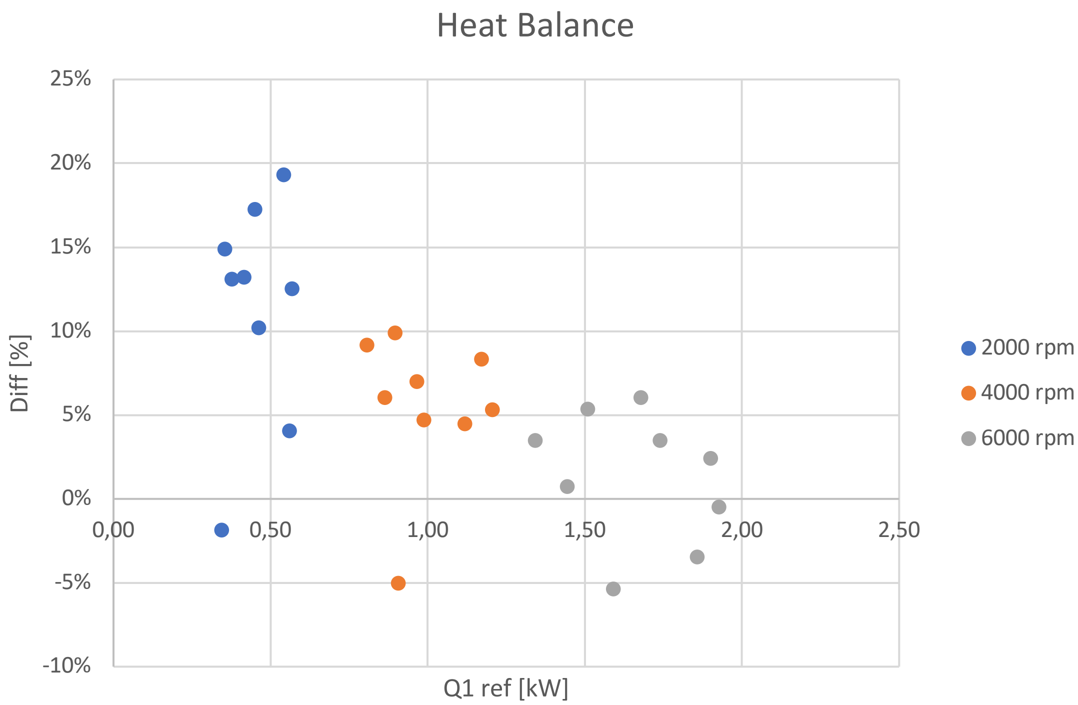

Figure 25: Heat balance for the system

For the condenser the heat produced by the refrigerant system differs from the heat absorbed by the water circuit, as seen in figure 26. Figure 26 is a result of the produced heat minus the absorbed divided by the produced, assuming that the volumetric efficiency derived from the evaporator heat balance is accurate. As seen in figure 26 the biggest deviation occurs for the lowest speed. For some the produced heat is greater and for others the absorbed.

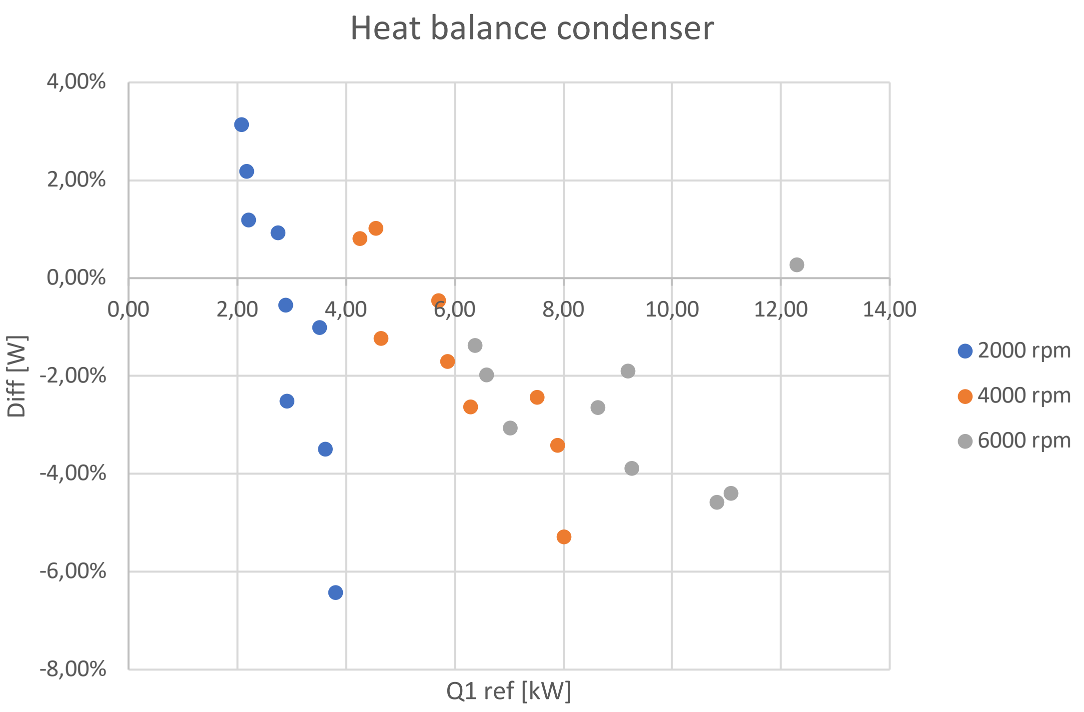

Figure 26: Heat balance for the condenser

5.1.5 Condenser UA

Looking at the UA value for the condenser the trend is that the UA value increases with heat capacity. In figure 27 all calculated UA values are presented. For the simulations the condenser UA value was set to a constant value, though this does not reflect the reality. Since the water flow was the same for measurements conducted, the difference must depend on the different operating conditions on the refrigerant side of the heat exchanger.

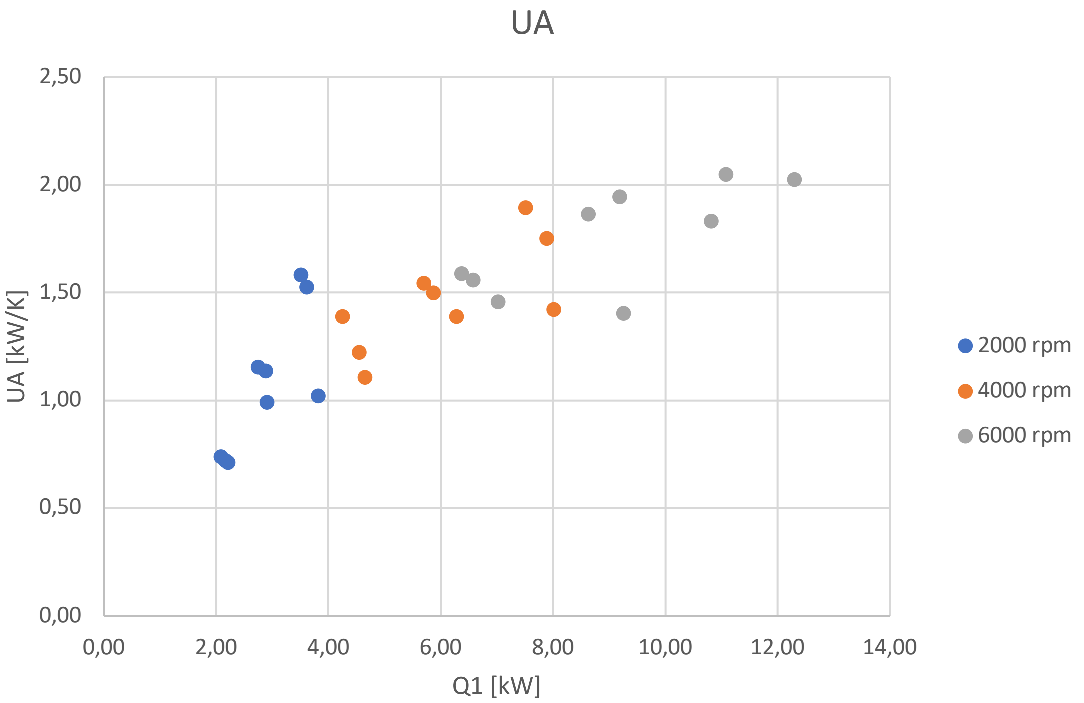

Figure 27: UA value for the condenser

5.1.6 Evaporator UA

Looking at the UA value for the evaporator the trend is that the UA value increases with heat capacity at the same speed. As seen in figure 28 the highest UA value from measurements is reached for the lowest speed. The simulated UA values are a lot higher than the measured. For the results below it should be kept in mind that the simulations and measurements are executed with different water flows. But, the trends are different for the simulations and measurements. Figure 29 shows the difference between the refrigerant going out of the evaporator and the water coming in. The higher speed the larger the difference. The difference for 2000 rpm is nonexistent and for 6000 rpm it is as high as 12 degrees.

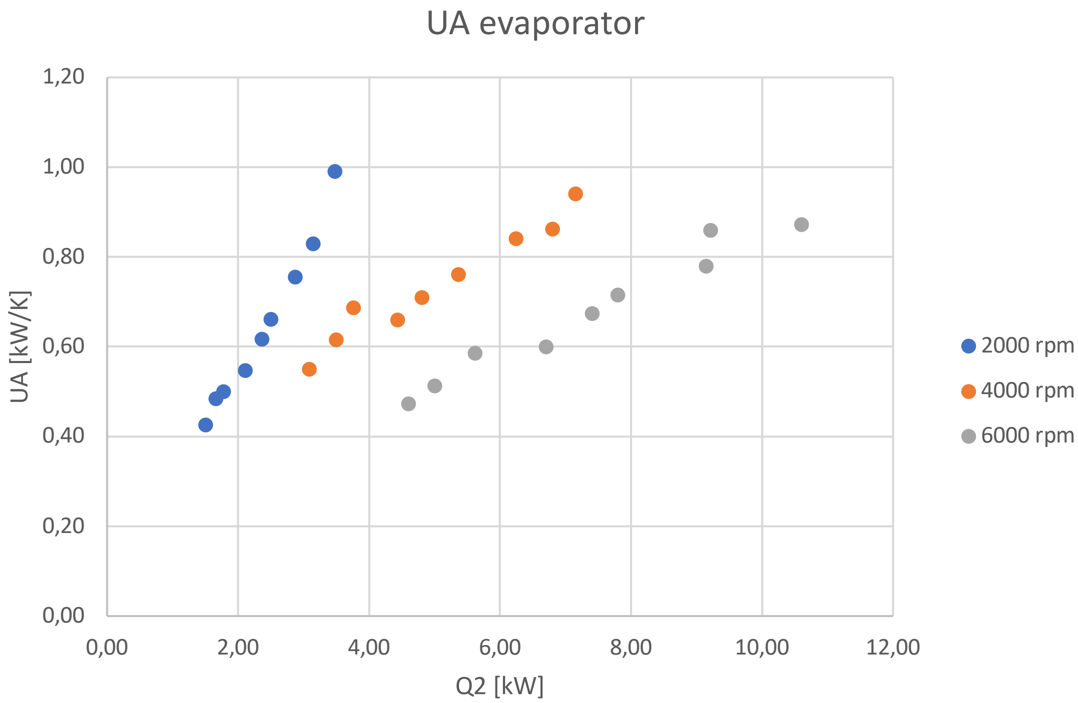

Figure 28: UA value for the evaporator

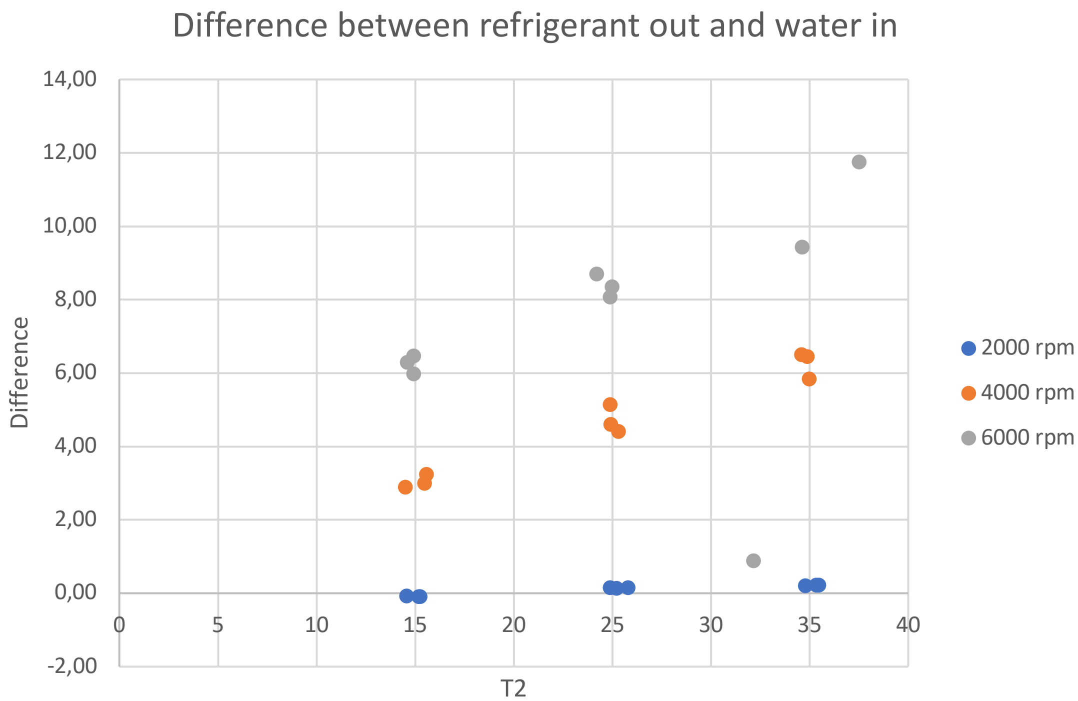

Figure 29: Temperature difference between water inlet and refrigerant outlet

5.1.7 Isentropic and volumetric efficiency

For the simulations both the isentropic and the volumetric efficiencies were set constant. Looking at the isentropic efficiency the trend is the same for all speeds. The efficiency increases, reaches a maximum around a pressure ratio of 3-3.5 and then it decreases. Figure 30 show that the higher the speed the lower the isentropic efficiency. The isentropic efficiency varies in between 50%-73%.

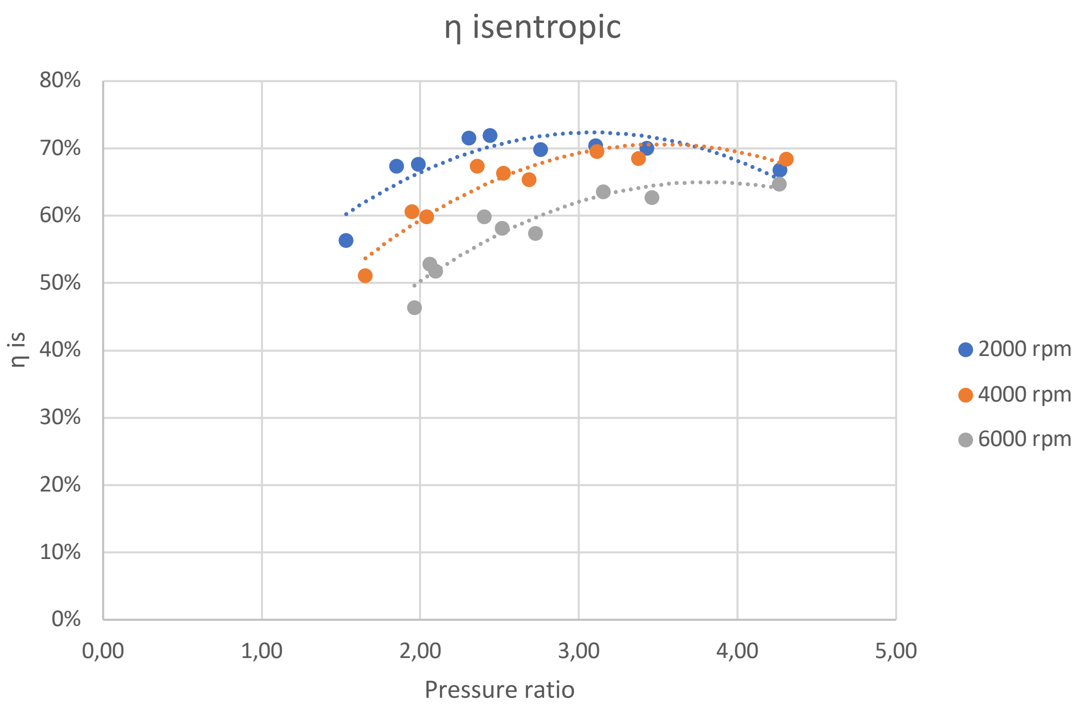

Figure 30: Isentropic efficiency

For the volumetric efficiency the trend is not as clear as for the isentropic. But, for the lowest pressure ratio the efficiency is low, it reaches a maximum around 2-2.5 and then it decreases. All efficiencies are in the span 86%-94.5%, with the highest efficiencies for the lowest compressor speed. The volumetric efficiency is presented in figure 31.

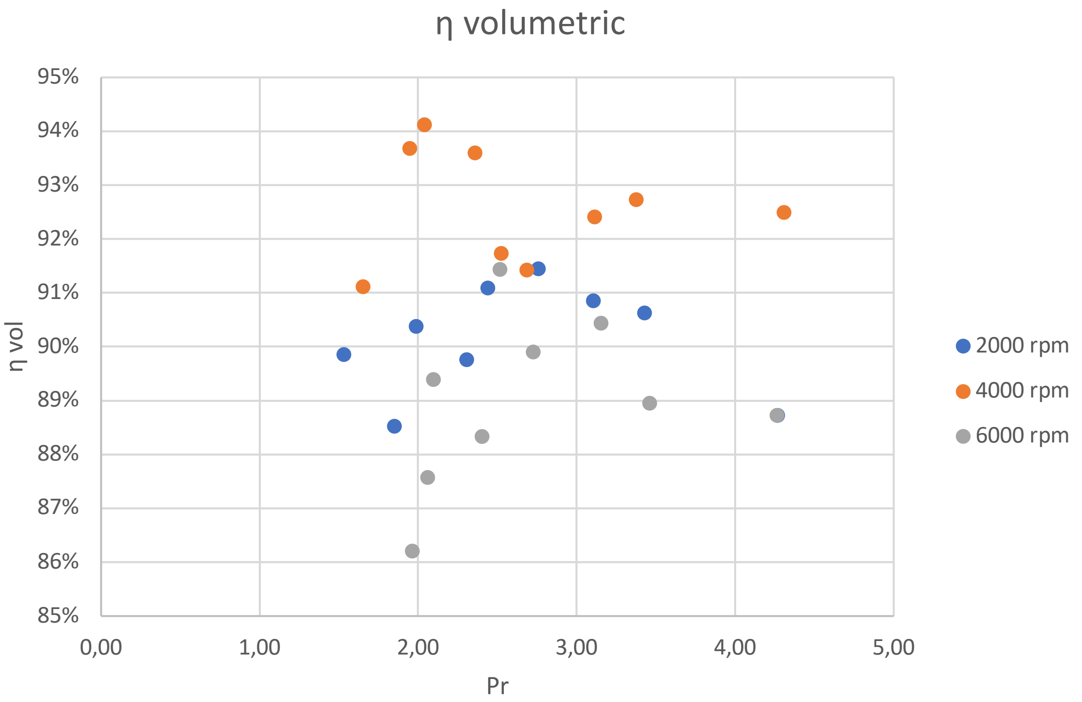

Figure 31: Volumetric efficiency

5.1.8 Coefficient of performance

Since this is a heat pump the COP1 is the one considered most interesting, produced heat divided by the compressors power consumption. The COP for the heat pump follows the same trends as the heating and cooling power. Both the highest and the lowest COP values are achieved at the lowest compressor speed of 2000 rpm. With an increased evaporating temperature the COP increases. It is higher for the lower water outlet temperature at the same evaporating temperature. For the heat pump the COP varies between 2 and 11. It can be seen in figure 32-34 that the COP_1 for the cases with 35 °C evaporating temperature and a water condenser outlet temperature of 50 the simulated COP1 is much higher than the actual. For a speed of 2000 rpm the simulated and measured values differ a lot for an outlet temperature from the condenser of 60 °C. The values differ, as expected, but the trends are the same. COP_2 can be found in appendix C. For the coefficient of performance only the compressor was considered, the power consumed for pumps and fans has been excluded. In figure 35 the measured values are plotted against the simulated ones.

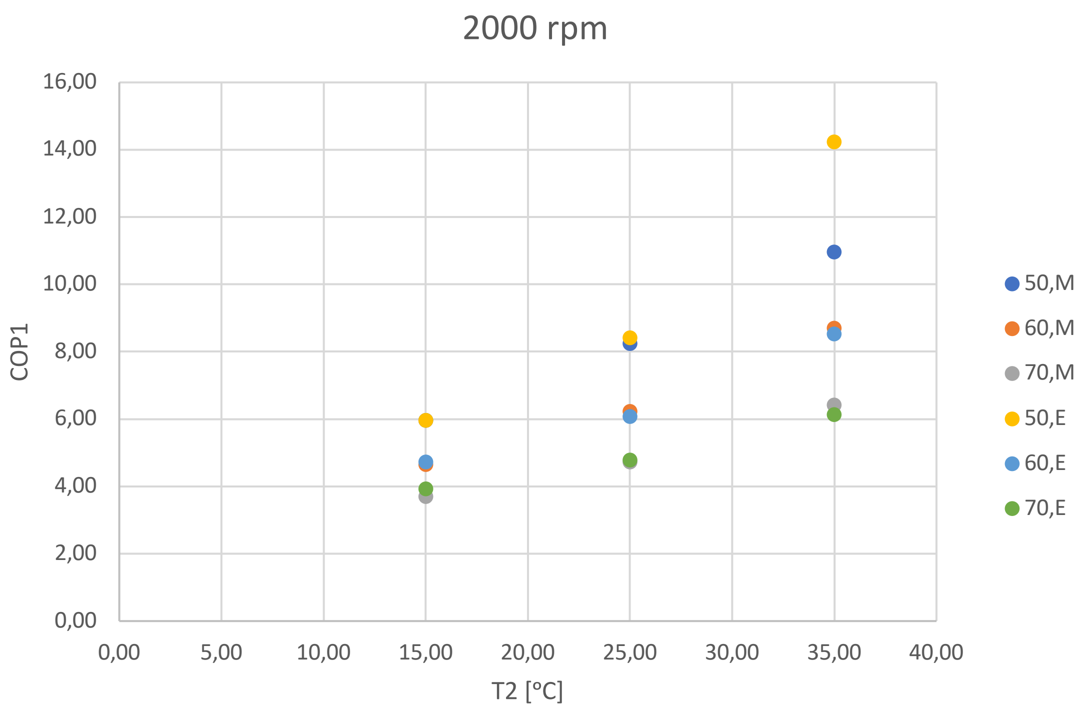

Figure 32: COP1 for 2000 rpm

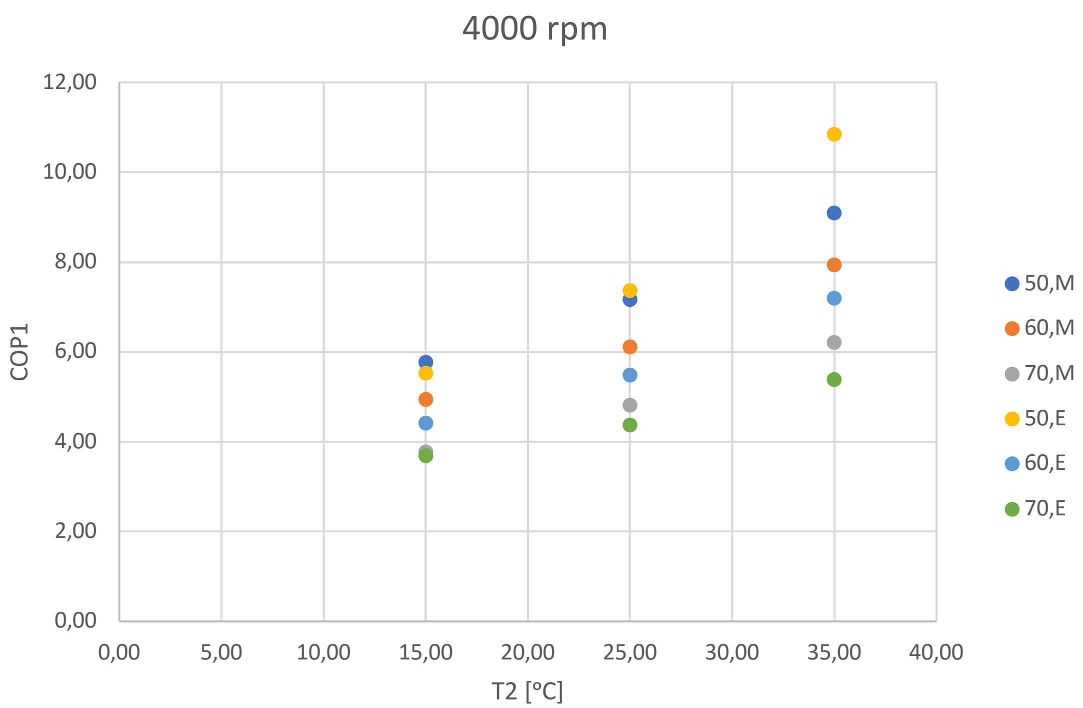

Figure 33: COP1 for 4000 rpm

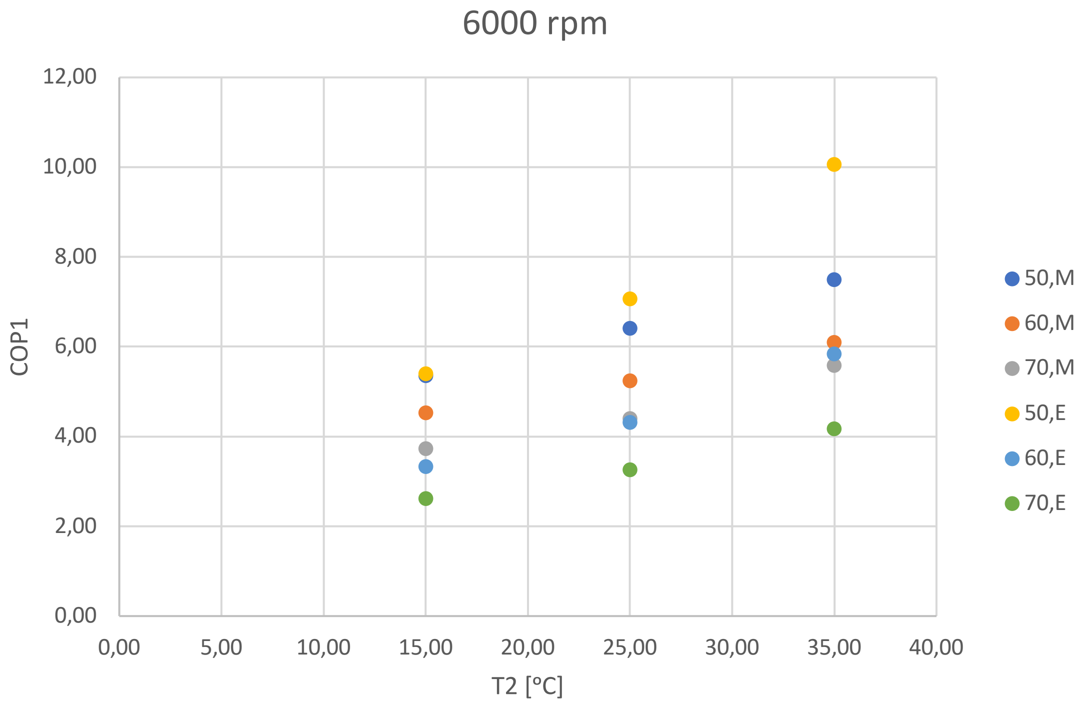

Figure 34: COP1 for 6000 rpm

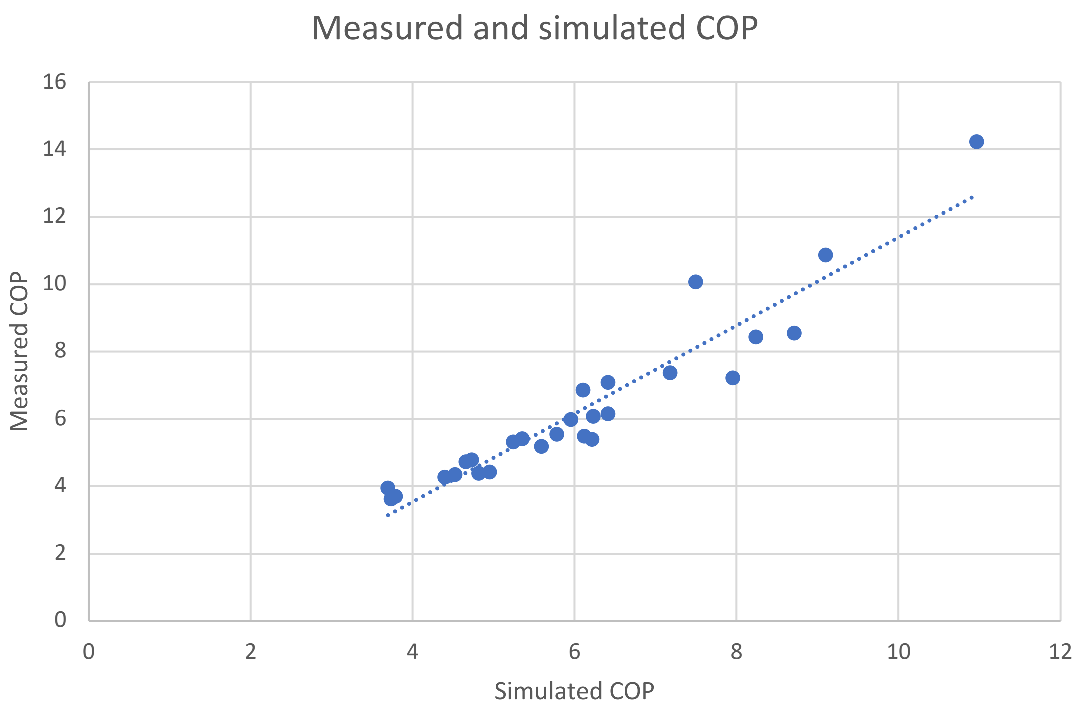

Figure 35: Measured and simulated Coefficient Of Performance

5.1.9 Carnot efficiency

Below, in figure 36, the Carnot efficiency is presented. It can be seen that the values are high, considering that the Carnot efficiency should be around 50%.

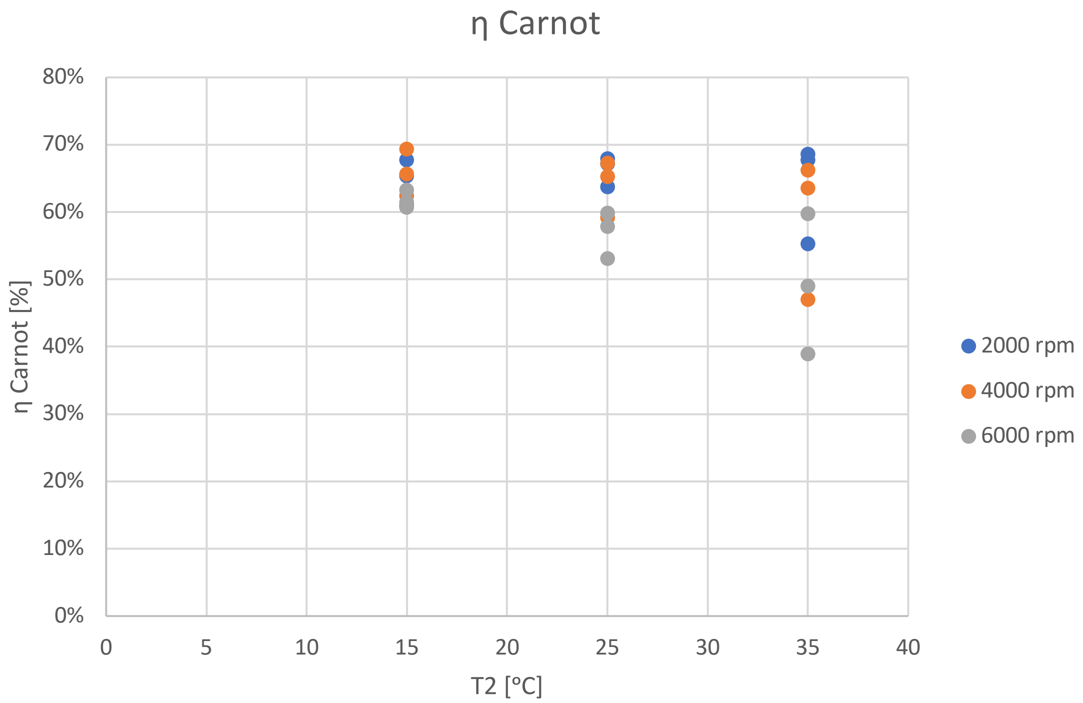

Figure 36: Carnot efficiency

6. Discussion

6.1 Experimental results

Though further investigation is needed, the heat pump is functioning well. The higher the evaporating temperature is the more heat is produced, for the same water outlet temperature from the condenser. For the cooling capacity and the COP the trends are the same. COP increases with a higher evaporating temperature, and is higher for the lowest water outlet temperature, for both COP_1 and COP_2. This result is expected as both heating and cooling capacity is higher for a water outlet temperature of 50 °C than 60 or 70 °C, but the compressor power required is for most cases lower. Experimental values for COP_1 vary between 3.7 and 11, and a majority of the values are rather high. The heat pump is well functioning, but further investigation should be able to conclude if the pump is suitable for the intended purpose. In figure 36, looking at the Carnot efficiency, it can again be seen that the COP is high for the tested system.

Looking at the compressor power the pattern is not the same for all cases. It can in figures 17-19 be seen that for the lowest water outlet temperature from the condenser the trend is that the lower the temperature lift the lower compressor power consumption for the lowest speed of 2000 rpm. For the higher speeds, 4000 and 6000 rpm, the trend is different as seen in figure 18 and 19. The compressor power consumption increases as the evaporating temperature gets higher. This is an interesting result, possibly a result of the increasing isentropic efficiency with a higher pressure ratio. It also depends on the increasing density in the suction line. Also, as seen in figure 30, the isentropic efficiency decreases with a higher speed.

As previously mentioned the isentropic efficiency increases with pressure ratio and it is highest for the low speed, with a maximum around 72%. The trend for each speed is that it increases with pressure ratio and reaches a maximum around a pressure ratio of 2-2.5, then the efficiency decreases. One of the reasons that the isentropic efficiency is lower for higher speed could be that there is more heat generated from the compressor, as a heat loss. Looking at the volumetric efficiency the trend is not as obvious, as sees in figure 31. Though it can be seen that the maximum efficiency is around 94%.

Looking at the condenser and the evaporator UA values, figures 27 and 28, both have an increasing trend as the UA value increases with a larger heat transfer. For the condenser the trend is very clear, the more heat produced the larger the UA value, independent of speed. During condensation the resistance lies within the liquid on the plate in the heat exchanger, the thinner the liquid film the larger the heat transfer coefficient is. The shear force increases with speed, resulting in a thinner liquid film, and therefore also a UA value for the condenser.

This differs from the evaporator UA value. For the evaporator the UA value increases with cooling capacity, but in this case the trend is not continuous and independent of speed. As seen in figure 28 the highest UA value is reached for the highest cooling capacity at a speed of 2000 rpm. For 4000 and 600 rpm the UA value is lower. For the same speed and water outlet temperature from the condenser the UA value increases with a higher evaporating temperature. One problem noticed with the measurements was the fact that the pressure meter to the expansion valve and the YOKOGAWA meter measuring the low pressure showed different values. The YOKOGAWA meter is measuring at the compressor inlet and the meter to the expansion device at the evaporator outlet. It could be because of a pressure drop in the internal heat exchanger, and the difference increases with speed. Since the pressure drop increases with speed the used value for the low pressure is possibly incorrect. At the same time as the expansion valve showed it regulated the opening to achieve a superheat of 4 K, the measured values never did. This depends on this pressure difference, a lower pressure results in a lower evaporating temperature and a larger superheat. Thus, since the measured values analyzed for 4000 and 6000 has a larger superheat a greater amount of the heat exchanger is used for the superheating process. This affects the UA value of the evaporator negatively. So, this should be evaluated further, with the correct evaporating pressure and system superheat.

Finally, looking at a heat balance over the condenser it can be seen in figure 26 that the water seems to absorb more heat than the heat pump emits, which is thermodynamically impossible. One possible reason for this error could be the temperatures in the system, which has also been an issue for the internal heat exchanger. This is discussed more in detail below in errors and observations. Looking at the last balance, the total for the heat pump in figure 25, it does not completely add up either. The condenser emits more heat than the sum of the cooling capacity and the compressor power, meaning it must have received heat from another source or the temperatures measured are not completely accurate. As seen in figure 26 the trend for the condenser is that the discrepancy decreases with a higher heat capacity and in figure 25 it can be seen that the discrepancy for the whole pump decreases with a higher compressor speed.

Since the heat transfer in the internal heat exchanger was relatively small, the presumed error due to error in measured temperature is relatively large. Therefore the internal heat exchanger is not evaluated further in this report. The conclusion is made that more measurements must be made before the heat exchanger could be evaluated. Since the mass flow was derived from the heat transfer in the evaporator it is not possible to say if the same problem occurred, but it is a possibility.

6.2 Comparison calculation and measurements

Above in the results it can be seen that the trends are very similar for the simulations and the measurement, though the values differ. For the simulations there are some defined values which for the measurements are a part of the results. Such as the isentropic efficiency, volumetric efficiency and UA values for the condenser and the internal heat exchanger. This affects the overall result for the pump and results in differences between measured and simulated values. More information about important inputs for the simulations can be found in appendix A.

One of the larger differences between simulations and measurements were the UA values for the evaporator. As further discussed under 6.4 the superheat was very unstable for the small water flow in the evaporator, therefore the flow was increased in order to achieve stable measurement points. The simulations were set up differently, where the flow was a result of the cooling capacity divided by the specific heat capacity and the temperature difference. Presented above are the results for a temperature difference of 5 degrees. The measurements were made with a somewhat constant water flow, resulting in a varying temperature difference. Also, for the simulations the volumetric efficiency was set higher than the measured. It was set to 95%, a value the system never reached. But, overall the patterns for the measured values follow the expected from the simulations with some differences.

6.3 Is it worthwhile economically

One very important factor is that the economizer solution must be economically beneficial in order to be competitive. The extra smaller heat pump connected to the main pump must in the long run save the investor money compared to other economizer solutions or leave the pump as it is. This aspect will need more investigation and also more testing must be conducted. It must be experimentally evaluated with the propane heat pump as the heat source and economic analysis must be done for the final solution with the final components, which is not the pump tested in this report. If the economizer is not economically competitive the solution cannot be used commercially. This evaluation is not within the boundaries of this report and needs further investigation.

6.4 Errors and observations

Conducting measurements does not come without difficulties. During the experiments many problems were faced and solutions had to be invented. One result noticed was that the heat balance for the internal heat exchanger did not add up. Therefore the internal heat exchanger is excluded from the results in this report. It was investigated if the temperatures were measured correctly by adding a thermocouple to the surface of the pipe for points A, B, F and G. It was seen that the temperatures differed somewhat when compared to the values from the thermocouples in the pipes. Below, in table 2, it can be seen that the temperatures differs, but it needs to be further analyzed and more results must be obtained in order to evaluate properly. This was only done for one case only to see if a difference occurred, which it did. Therefore the results were not considered for the internal heat exchanger. All temperatures in the table below also affect the rest of the result, possible resulting in other errors. Although the temperatures do not differ much it has an noticeable effect on the heat balance over the internal heat exchanger, since the heat transferred is small.

Table 2: Temperatures for a certain case

	On pipe [°C]	**In pipe [°C]**
TA	43.2	43
TB	46.3	47
TD	57.8	59
TF	55.5	56

A second observation was that the electric compressor power measured was very low, way too low to be accurate. This could stem from that the frequency regulator was not compatible with the electric power meter used. Therefore the measured compressor power has not been used for the evaluation of data. This was noticed to late and it was not clear how to solve the problem. Therefore the measured compressor power has been excluded from this report.

Another observation made during the start of the measurements was the fact that the system was not functioning properly with a small water flow in the evaporator. It was very difficult for the expansion device to regulate the system to achieve the set superheat of 4K. One possible cause could then be sudden variations of the flow, if the flow is small even smaller variations could have a large impact. The pipes and pump were rather large for the small flow so an extra valve was installed in order to achieve the small flows. Possibly this could affect the superheat for the refrigerant in the evaporator, though this was not investigated further. After the observation the water flow was increased, achieving a much more stable system. Therefore the measurements were conducted with the larger water flow, resulting in differences between the simulations and measurements. ‘

One problem faced during the final experiments was that the tank solution would not function for all measurement points. Using the solution presented in figure 11 the temperatures in the water circuits cannot overlap each other. This problem occurred for the high compressor speed and small temperature difference between condensing and evaporating temperatures. For example, the case with a speed of 6000 rpm, an evaporating temperature of 35 °C and water outlet temperature from the condenser at 50°C. At this speed the heat transferred in the heat exchangers results in a large temperature lift, since the water flows still were quite small. This resulted in a temperature difference of around 10K, between water inlet and water outlet. This is a problem since the tank water for this case needs to provide heat to the evaporator water circuit to reach around 43 °C at the same time as it absorbs heat from the condenser circuit to cool the water to around 38 °C. This is impossible. The tank temperature must be in between the circuit temperatures. Therefore the condenser circuit was connected to a water tap near the test rig for these cases in order to be able to lower the water temperature enough to reach the desired measurement points, since the needed cooling effect could not be provided by the tank as intended.

During measurements one of the factors possible to regulate was the superheat. For all cases the set point was 4K, though this was not always achieved. The lower the compressor speed the easier the expansion valve regulated the superheat. When the speed was 2000 rpm the system was very stable and the superheat kept constant at 4K. But, when the compressor speed was increased it was more difficult for the expansions vale to maintain a superheat of 4K. For many cases the superheat was higher, up to around 7-8K. This result is from the measured temperatures, thermocouples in the system pipes. Though, the expansion device came with a display and it said otherwise. According to the display of the expansion valve regulator, the superheat was 4K for all cases. An observation from the display was that the pressure measured was for a majority of the cases higher compared to the YOKOGAWA pressure transmitter. The measured evaporating temperature for the expansion valve is dependent on the pressure, a higher pressure gives a higher evaporating temperature and therefore lower superheat. So, according to the expansion device display the valve was able to maintain 4K superheat. There is a discrepancy in the result and it can be discussed which one is correct. Most likely this difference is a result of a pressure drop in the internal heat exchanger.

One interesting factor for further investigation is an estimation of the heat losses from the heat pump. All pipes and heat exchangers were insulated but the compressor was not, which resulted in heat losses while it was running since the compressor became warmer than its surroundings. Looking at the compressor with an IR camera it was seen that parts of the compressor were very warm, above 50 °C. The majority of the compressor was around 35-40 °C. Since the temperature is warmer than the surrounding air heat losses occur. How large the losses are could be future work, or it can be tested to insulate the compressor and evaluate the total loss. For the measurements done it is possible that the temperatures are not completely accurate, this problem should first be attended to.

A simple estimation of errors were conducted for the result presented in this thesis. The assumption was made that the thermocouples had a measurement error of 0.25 degrees. By altering the measured temperatures, by adding or subtracting the measurement error, an estimation of the effect of the error was made. For the heating capacity the results were small, only around plus minus 0.5%. The compressor power was affected more, around plus minus 2.5%, since the compressor power is less than the heat capacity. The coefficient of performance was also affected, plus minus less than 2.5%. Still, this did not affect the fact that the system is both well-functioning and efficient. Though, only the temperatures were looked at for the estimation of error. Both flow meters and pressure transmitters also affect the outcome of the error. This could be investigated further, and many combinations of plus or minus measurement errors can be looked at and evaluated. Though the decision was made to look at the temperatures, since there have been issues with the measured temperatures. But as seen above, this does not affect the system that much.

6.5 Future work

One of the main aspects which should be further evaluated is the economic gain from implementing this type of economizer for a large scale heat pump. Will this solution be saving the investor money in a relatively near future or will the economizer be too expensive? This test rig evaluated will, as mentioned earlier, not be the final solution. There is another test rig being built which will actually be used, if it turns out to be well functioning. It will consist only of already commonly used components, to avoid surprises similar to the ones in this project where the components were experimental. So, the most useful economic calculation should be done on the more finished and final product, to get the result as realistic as possible.

Also, some of the results were not considered useful since were some inconsistencies. For example the temperatures around the internal heat exchanger and the compressor power. So, some of the measurements should be conducted again to achieve complete data for evaluation. Due to limited time these problems were not fixed for this report. Oher inconsistencies should also be looked at and losses should be evaluated in order to fully evaluate the functionality of the heat pump.

7. Conclusion

The experimental economizer heat pump set up, while using an automotive Sanden compressor, new condenser and evaporator from Alfa Laval and the refrigerant R600a, show very promising results. The results with 120g refrigerant show that the charge was enough for all conditions tested. Both Coefficient Of Performance and Carnot efficiency show promising results. Heat capacity is between 2 and 12 kW, depending on the operating conditions and refrigerant mass flow.

The unit is intended as an economizer for a larger propane heat pump, but it can also be used as a stand-alone high temperature heat pump. If it is decided to proceed with this scheme, future analysis is recommended, both concerning technical- but maybe mainly economic matters.

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Appendix A

Table A1: Model assumptions

Parameter	**Value**
UA condenser	2 kW/K
UA internal heat exchanger	40 W/K
Isentropic efficiency	65 %
Volumetric efficiency	95 %
Superheat	3 K
Subcooling	5 K

Appendix B

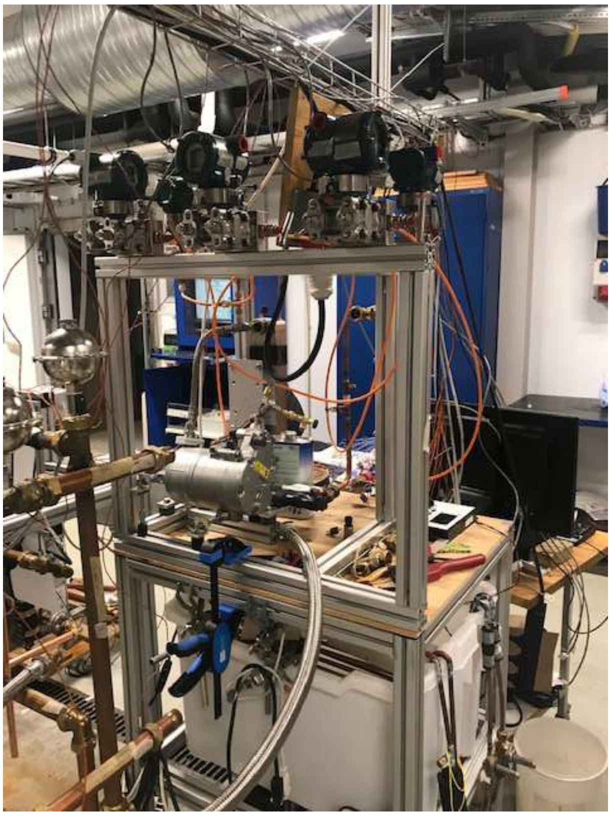

Figure B1: During construction

Figure B2: Economizer heat pump

Figure B3: Economizer heat pump

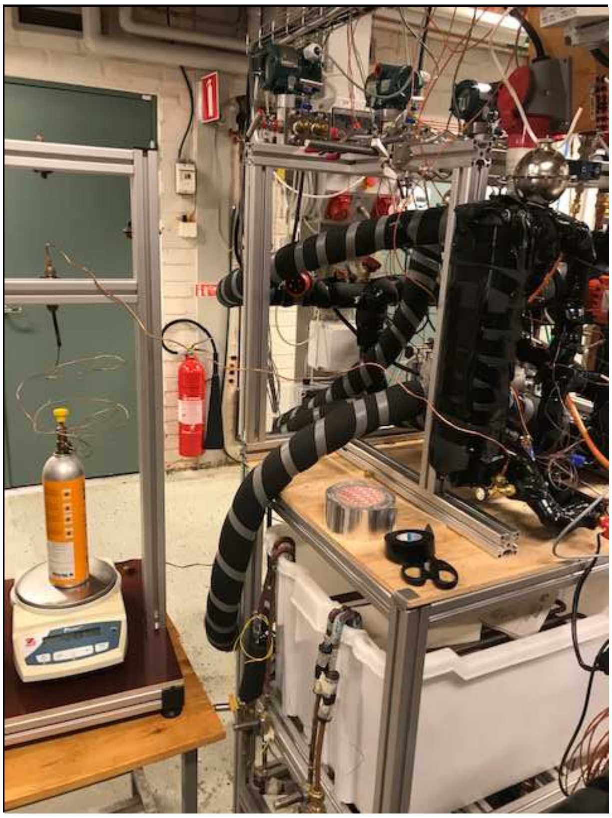

Figure B4: Filling station

Appendix C

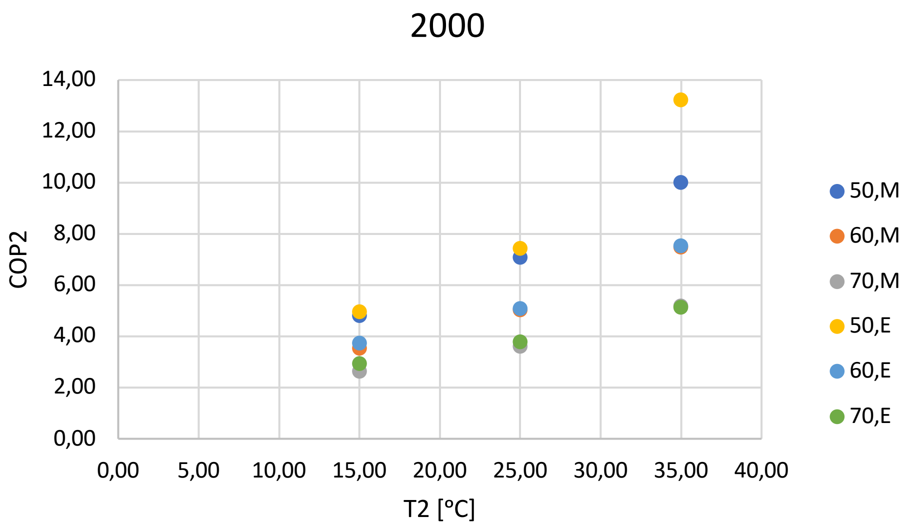

Figure C1: COP2 for 2000 rpm

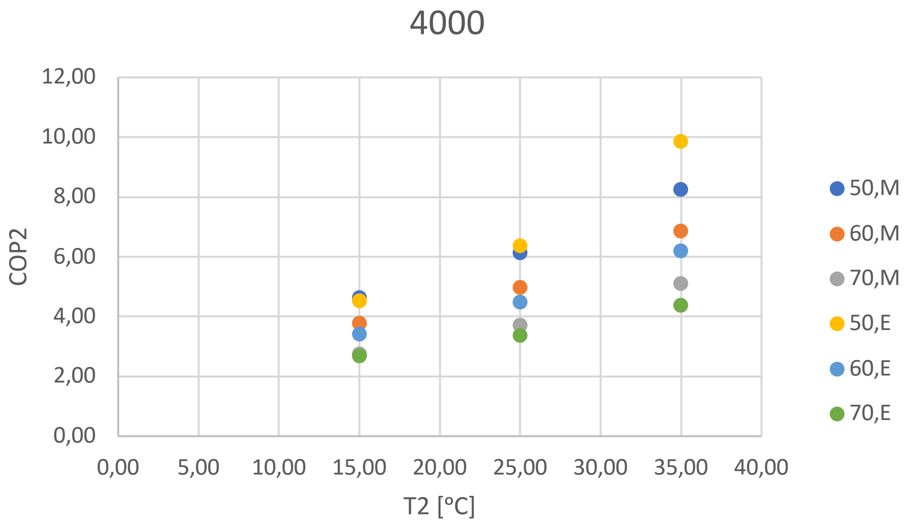

Figure C2: COP2 for 4000 rpm

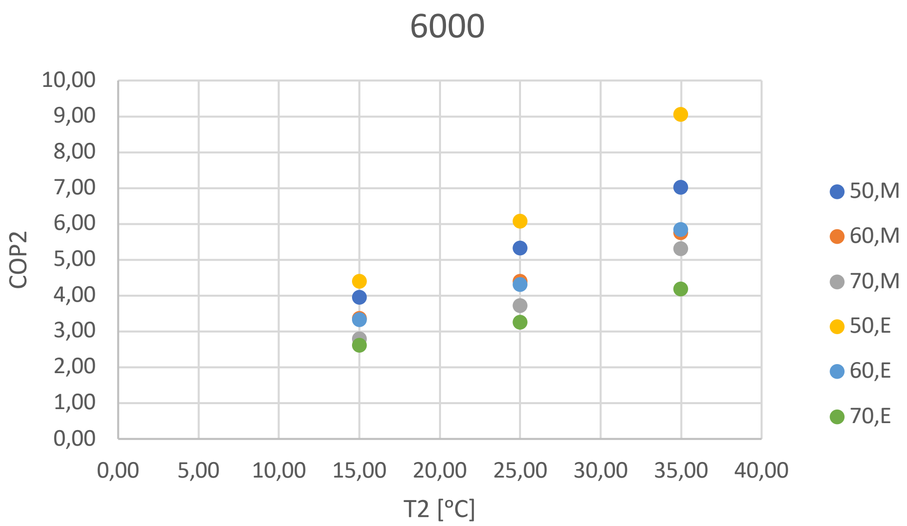

Figure C3: COP2 for 6000 rpm