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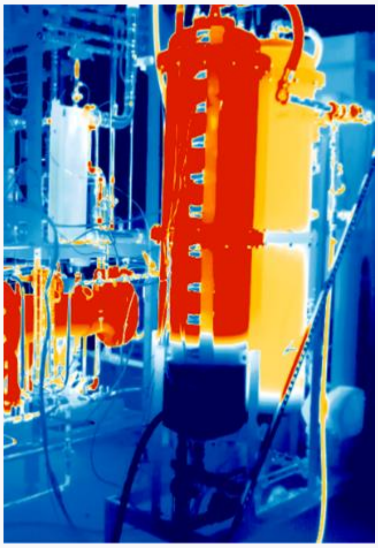
Highly Efficient Thermodynamic Cycle with Isolated System Energy Charging (ISEC)

- Final report



Title:
Highly Efficient Thermodynamic Cycle with Isolated System Energy Charging (ISEC)

Prepared for:
Project consortium

Svedan Industri Køleanlæg
Innotek
Alfa Laval
Arla Foods
Bjerringbro Varmeværk
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March 2016

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1. Preface

This report is the final report of the study: "Highly Efficient Thermodynamic Cycle with **I**solated **S**ystem **E**nergy **C**harging" (the ISEC-concept). The objective of the project is to demonstrate an improvement in the energy efficiency of heat pumps with up to 50 % by using a novel technology where heat pumps are operated together with optimum storage usage which reduces the average temperature level in the heat pump.

This research project is supported financially by the Danish Energy Agency's EUDP programme (Energy Technology Development and Demonstration). Project number: J.nr. 64013-0110.

The project is carried out in cooperation with the Technical University of Denmark and the following industrial cooperating partners: Svedan Industri Køleanlæg, Innotek, Alfa Laval, Arla Foods, Bjerringbro Varmeværk and Metro Therm.

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The project team would like to thank the following persons:

Povl Frich from the EUPD programme (Danish Energy Agency), who has supported the project with valuable inspiration.

Ville Gaunaa, who during his bachelor project at the Technical University of Denmark has contributed to the work.

Frederik Bramsen, who during a specialized study at the Technical University of Denmark has contributed to the work.

1.1. Project details

Project title	Highly Efficient Thermodynamic Cycle with Isolated System Energy Charging (the ISEC-concept)
Project identification (program abbrev. and file)	Område: Energieffektivitet, EUDP 13-I
Name of the programme which has funded the project	Energiteknologisk Udviklings- og Demonstrationsprogram (EUDP),
Project managing company/institution (name and address)	Danish Technological Institute, Gregersensvej 1, DK-2630 Taastrup
Project partners	Svedan Industri Køleanlæg Innotek Alfa Laval Arla Foods Bjerringbro Varmeværk Metro Therm Technical University of Denmark
CVR (central business register)	56976116
Date for submission	31. March 2016

1.2. Short description of objectives and results

English version

The objective of the project is to demonstrate an improvement in the energy efficiency of heat pumps with up to 50 % by using a novel technology where heat pumps are operated together with optimum storage usage which thereby reduces the average temperature level of the heat pumps.

The results show that it is a feasible technology to operate and control both the storage system and the heat pump together. Calculations, based on R717, show that it is possible to improve the performance by 25 % when heating water from 40 to 80 °C and at the same time cooling water from 30 to 25 °C. Furthermore, a 50 % increase in the performance is measured, based on R134a and without sub-cooling, in cases where water is heated from 15 to 60 °C with an outlet temperature of 13 °C from the evaporator.

Danish version

Formålet med projektet er at demonstrere en forbedring af energieffektiviteten for varmepumper med op til 50 % ved hjælp af en ny teknologi, hvor varmepumper anvendes sammen med en optimal brug af lagerbeholdere, som derved vil reducere gennemsnitstemperaturen i varmepumperne.

Resultaterne viser, at ISEC konceptet er en operationel teknologi til at drive og styre det sammenkoblede system af lagerbeholdere og varmepumpe. Beregninger baseret på R717 viser, at det er muligt at forbedre effektiviteten med 25 % ved opvarmning af vand fra 40 til 80 °C samtidig med afkøling af vand fra 30 til 25 °C. Endvidere er der målt en stigning af ydeevnen (COP) på 50 %, baseret på R134a og uden underkøling, i tilfælde af opvarmning af vand fra 15 til 60 °C med en udløbstemperatur på 13 °C fra fordamperen.

1.3. Executive summary

English

Today, the thermodynamic cycle process of a heat pump is process optimized, and components, which form part of this, only reflect a minor optimization potential. Therefore, the largest optimization potential is achieved by focusing on how the heat pump technology can be integrated into a heating system most optimally.

This is the focus of the project which aims at developing and demonstrating how a high-efficiency heating unit based on the traditional thermodynamic cycle process in a heat pump can achieve an energy saving potential up to 50 % by using the newly developed ISEC concept. The short name ISEC is an abbreviation of "Isolated System Energy Charging".

The ISEC concept is based on known technologies used in a new and revolutionary, yet simple way. Thus, the focus of the project is the adaption of components and management to achieve the best interaction between the heat pump, the heat storage and the heat consumption. In fact, it will be possible to construct certain essential components, for example the compressor, in a simpler way by means of the ISEC concept.

Today, the typical use of heat pumps is for heating with a relatively large temperature difference between the refrigerant and the media to be heated. Normally, the media will be heated in one pass of the condenser. With the ISEC concept, the media to be heated will pass the condenser several times. Thus, the media will be heated gradually and the temperature difference between the refrigerant and the media will be smaller.

By heating one tank at a time, the condensing temperature (or evaporating temperature) of the heat pump can be made to vary. This means that a condensing temperature, which is only slightly higher than the medium temperature of the liquid during the heating process, is obtained during the heating process. This is as such not possible during a traditional continuous heating of fluid, but the situation can be realized when using the ISEC concept.

The ISEC concept consists of two (or more) tanks. One of the tanks is heated (charged) while the other tank, which previously has been charged, is tapped for heat (discharged). When the second tank has been discharged, the first tank is fully charged with heat, and the system switches to discharge the first tank while the second tank is being charged. Seen from the perspective of the heat source and the heat sink, the introduction of the ISEC concept does not change the conditions.

In this project, focus is on heating by means of heat pumps, but the concept might also be applied for cooling purposes.

Project activities range from theoretical calculations, design and construction of individual components, to overall management, and construction during the experimental stage. The focus of the project covers many technological aspects which no individual company, institute or university possesses. Therefore, Danish Technological Institute has undertaken the role as the coordinating and governing body.

This concept has not been implemented earlier because heat pumps are normally designed to produce the needed amount of heat immediately when it is required. By contrast, the ISEC concept optimizes the production of heat over a period of time by separating the production and the consumption.

The project results show that it is possible to operate and control a heat pump system when complying with the developed ISEC concept. The obtained results show that it is a technology which holds the largest benefits in cases with long operating periods and relatively large temperature differences. It is expected that the concept can be used for industrial production purposes and district heating in the future. Even though the concept has been tested in the laboratory, it is necessary to conduct further studies before the concept is introduced in large-scale applications due to the inherent complexity and the many parameters which have to act together to achieve the full benefits of the concept.

The project objective has been achieved by showing it. In some cases, it will be possible to obtain an improvement of 50 % in the efficiency.

Danish

I dag er den termodynamiske kredsproces i en varmepumpe procesoptimeret og komponenterne, der indgår heri, efterlader kun et mindre optimeringspotentiale. Fokus er derfor rettet mod, hvorledes varmepumpeteknologien mest optimalt kan integreres i et opvarmningssystem.

Dette er fokusset for projektet, som har til formål at udvikle og demonstrere, hvorledes en højeffektiv opvarmningsenhed baseret på den traditionelle termodynamiske kredsproces i en varmepumpe vil kunne opnå et energibesparelsespotentiale på op mod 50 % ved anvendelse af det nyudviklede ISEC-koncept.

ISEC-konceptet er baseret på kendte teknologier anvendt på en ny og revolutionerende, men stadig enkel måde. Fokus for projektet er altså tilpasning af komponenter og styring, så der opnås et optimalt samspil mellem varmepumpe, varmelager og varmeforbrug. Det vil faktisk være muligt at konstruere visse essentielle komponenter, som f.eks. kompressoren, på en enkel måde via ISEC-konceptet.

Den typiske anvendelse af varmepumper i dag er til opvarmning med en relativ stor temperaturforskel mellem kølemidlet og mediet, der skal opvarmes. Mediet vil normalt blive opvarmet ved ét gennemløb af kondensatoren. Med ISEC konceptet vil mediet, der skal opvarmes, passere kondensatoren flere gange og vil derfor blive opvarmet gradvist, således at temperaturforskellen mellem kølemiddel og mediet bliver mindre.

Ved opvarmning af én beholder ad gangen vil varmepumpens kondenseringstemperatur (og ligeledes fordampningstemperatur) kunne bringes til at variere, således at der i opvarmningsforløbet opnås en kondenseringstemperatur, der kun er lidt højere end væskens middeltemperatur under opvarmningsforløbet. Ved en traditionel kontinuerlig væskeopvarmning er dette ikke umiddelbart muligt, men processen kan realiseres ved ISEC-konceptet.

ISEC-konceptet består af et system indeholdende to eller flere beholdere. Én beholder vil da blive opvarmet (opladet) samtidig med, at den anden beholder, som tidligere er opladet, tappes for varme (aflades). Når nummer to beholder er blevet afladt, er den første beholder fuldt opladt med varme, og systemet skifter til at aflade den første beholder samtidig med, at nummer to beholder oplades. Set fra varmekildens og varmeaftagerens side ændrer forholdene sig ikke ved introduktion af ISEC-konceptet.

Hovedudfordringerne i projektet har været opbygning af ISEC-varmepumpesystemet og styring heraf. Udfordringerne har også omfattet varmepumpedelen, hvor varmeveksler-koblingen til varmepumpen er optimeret. Endvidere er varmepumpekredsen optimeret med fokus på at opnå en enkel opbygning.

Projektets aktiviteter spænder fra teoretiske beregninger, design og konstruktion af enkelte delkomponenter til overordnet styring og konstruktion i forsøgsfasen. Ved at samarbejde mellem virksomheder, som alle besidder en bred faglighed, er der i projektet opnået resultater som intet firma, institut eller universitet havde kunnet opnå alene. Derfor har Teknologisk Institut påtaget sig rollen som koordinerende og styrende partner.

Årsagen til at dette koncept ikke er blevet anvendt tidligere er, at varmepumper normalt anvendes til at producere den nødvendige mængde varme, når der er behov for det. Det nye ISEC koncept optimerer i stedet varmeproduktionen over en tidsperiode ved at separere produktion og forbrug.

1.4. Project objectives

The main objective of the project is to demonstrate an improvement in the energy efficiency of heat pumps with up to 50 % by using a novel technology where heat pumps are operated together with an optimal use of thermal storages, which reduces the average temperature level in the heat pump.

The objective is achieved by carrying out research on how the different elements are combined most optimally through gained experience from the testing of the concept, and the selection of control concept as well as by calculating the expected performance.

1.5. Project results, dissemination and utilization of results

In the project, the ISEC concept is elaborated by carrying out research encompassing both experimental work and calculation of the performance. Moreover, the control scheme of the system has been established and analyzed.

The commercial aspect has not yet been realized, but the project partners expect that the results and the participation in the project have been fruitful for the development of new products.

The objectives stated in the project proposal have been obtained by the research done showing that the technology is feasible for the purposes intended. The work done showed more challenges than foreseen in the proposal. A number of issues related to the technology has to be considered in order to obtain an optimal solution. The outcome of the project will also be a valuable foundation for further research in other technologies, e.g. cooling applications and cooling in combination with heating at the same time.

The project results are disseminated in terms of reports and a number of papers presented at conferences as well as an extended know-how information for the participants involved in the project. The project has also provided input for a bachelor thesis project and a specialized study at the Technical University of Denmark.

The results of the project are expected to be incorporated in future products and will thereby increase the turnover, export and employment of the companies involved. The companies have gained experience by the project participation and will use this experience commercially both in relation to the utilization of the ISEC-concept as well as in other projects.

The project results will contribute to the energy policy objectives by improving the performance of heat pumps combined with energy storage and thereby increase the possibilities of achieving flexibility in the electric grid and of increasing the share of electricity used in the Danish energy consumption.

1.6. Project conclusion and perspective

The concept has been tested and found operational. A number of possible improvements of the concept has been identified, e.g. the control of both the heat pump and the storage tanks.

The improvement in efficiency by using the ISEC concept is expected to be in the area between 10 and 50 % depending of the actual design, the reference set up and the operating conditions. The largest improvement can be expected when there is a large temperature difference (lift) of the media to be heated and a large operating time. The improvements in the lower end of this interval are expected when the temperature lift is small and the number of operating hours limited.

Detailed conclusions are given in the different parts of this report.

Some of the challenges of using the ISEC technology are the additional investments compared to the savings obtained. The need for design of the single applications will also be a barrier for an extended implementation of the concept.

However, the project shows that the ISEC concept is feasible in practice and that it is able to provide a significant improvement of heat pump efficiency in comparison with traditional single-pass water heating heat pump systems. The project will also be a valuable basis for further research and development of the technology.

2. Introduction

The ISEC concept consists of two (or more) storage tanks connected to a heat pump through a valve system. One tank is heated (charged), while the other tank, which previously has been charged, is tapped for heat (discharged). When the second tank has been discharged, the first tank is fully charged with hot water, and the system switches to discharging the first tank, while the second tank is being charged, and so on.

Seen from the heat sink perspective, the introduction of the ISEC concept does not change the conditions. The connected heat sink will receive a continuous flow of hot water as if connected directly to the condenser of the heat pump. By heating one tank at a time, the condensing temperature of the heat pump can be made to vary, starting at a low temperature and increasing as the charge-tank heats up. This results in an average COP of the ISEC system, which is significantly higher than the COP of a traditional heat pump with the same capacity.

The report is structured in three main parts. Part I describes the main content of the scientific research in the project. Part II provides the experience from the test setup of the storage tanks where the heat source is an electric heat resistance and a heat pump, respectively. Part III is an evaluation of how the ISEC concept might be utilized in combination with a district heating system.

3. PART I; Final scientific dissemination of the research in the ISEC project

Abstract

Part I of this report presents the results of the scientific research conducted in the ISEC project. The aim of this work was to clarify how the ISEC concept performs compared to the traditional heat pump.

The purpose of the ISEC concept is to provide a high-efficient heat pump system for hot water production. The ISEC concept uses two storage tanks for the water, one discharged and one charged. Hot water for the industrial process is tapped from the charged tank, while the other tank is charging. Charging is done by circulating the water in the tank through the condenser of a heat pump several times and thereby gradually heating the water. The charging is done with a higher mass flow rate than the discharging to reach several circulations of the water during the time frame of one discharging. This results in a lower condensing temperature than if the water was heated in one step. Two test setups were built, one to test the performance of the heat pump gradually heating the water and one to investigate the stratification in the storage tanks. Furthermore, a dynamic model of the system was implemented in Dymola, and validated by the use of test data from the two experimental setups. It is shown that there is a good consistency between the model and the experimental tests. The ISEC concept approaches the efficiency of a number of heat pumps in series and the COP of the system may reach 6.8, which is up to 25 % higher than a conventional heat pump heating water in one step.

The final dissemination of the scientific research of the ISEC project is presented in five sections. Presenting the overall concept, the mathematical model used, the test setups, a validation of the system, theoretical calculations using the model and a conclusion of the results.

Part I of this report includes the following parts:

1. Introduction
2. Methods
    a. Description of dynamic model developed
    b. Description of test setups of the R134a refrigeration system and tank test setup
3. Results
    a. Validation of model using test results.
    b. Numerical calculations and optimization of the system using ammonia.
4. Discussion
5. Conclusion

3.1. Introduction

3.1.1. Introduction to the ISEC concept

A conventional heat pump system used to heat water for industrial processes is shown in figure 1. The heat pump heats the water from the return temperature directly to the temperature used for the process. Therefore, the conventional system will also be referred to as a direct heat pump.

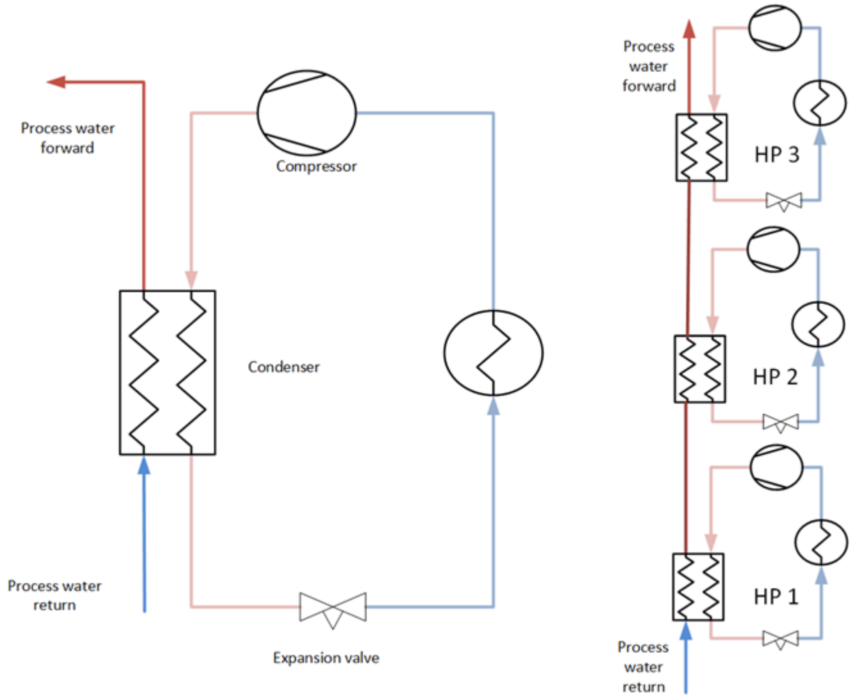

Figure 1 – Left: Conventional heat pump system for water heating for industrial processes. Right: Three heat pumps in series.

The production of hot water may be continuous, as it delivers directly to the water circuit. The idea of the ISEC concept comes from considering several heat pumps in series as shown in figure 1 Right. Considering the two systems in figure 1, both heats the water from process return temperature to the process water forward temperature. However, the three heat pumps in series heat the water gradually as it passes each condenser (Wang, et al., 2010). The temperature increase of the water in each condenser is similar. The performance difference between one heat pump and three in series is shown in figure 2, for the case of heating water from 40 °C to 80 °C. The efficiency of the compressors was disregarded; hence, the performance difference between the two systems occurs solely due to the differences in the condensing temperatures.

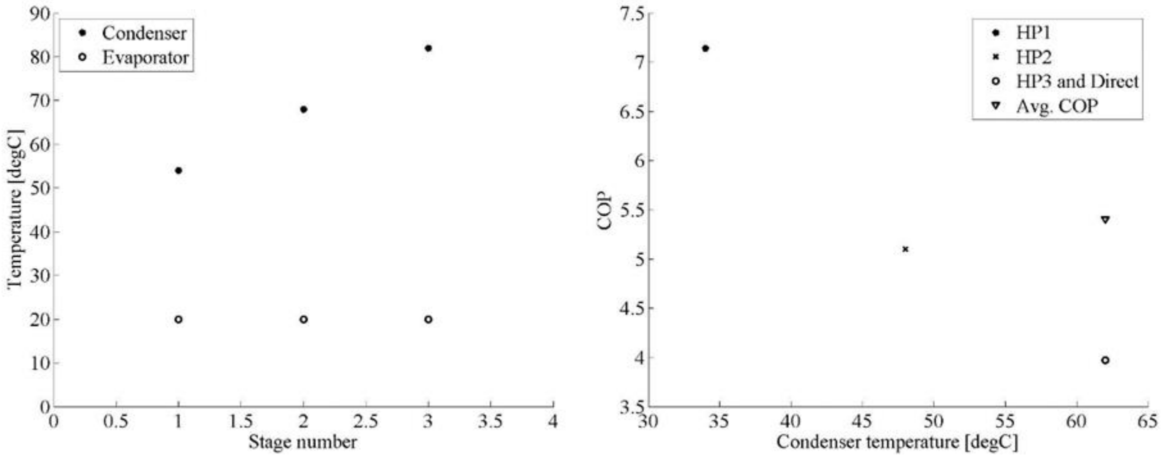

Figure 2 – Comparison of water heating from 40 °C to 80 °C using one direct heat pump and three heat pumps in series. Left: Evaporating and condensing temperatures for the three heat pumps in series. Right: COP for each of the three heat pumps in series compared to the direct heating heat pump and the average COP of the heat pump in series.

The condensing temperature must be higher than the outlet temperature of the water assuming the desuperheater cannot be used to heat the water above the condensing temperature. This results in a large mean temperature difference between the water and the refrigerant, which results in entropy generation, and accordingly, a lower COP. If heating is performed stepwise, the condenser of each step only needs to operate at the water outlet temperature of this step. The condensing and evaporating temperatures of each heat pump are shown in figure 2 (left) and the COP of each stage is shown in figure 2 (right) together with the average COP of the series of heat pumps and the COP of the conventional, direct heat pump. For this simple analysis, the COP for the direct heating system is 4.0 and the average COP for the serial system is 5.4, thus an improvement of almost 25 %. This improvement is closely related to savings of operating expenses. The ISEC concept may reach the same performance as a series of heat pumps, using a single heat pump only. The ISEC-based heat pump system delivers a continuous supply of hot water, but by means of a storage tank system instead of several heat pumps in series. In the ISEC based heat pump system, there is a gradual heating of water in a tank with an increasing condensation temperature in the thermodynamic cycle until the desired temperature is reached. This is achieved by recirculating the water through the condenser. Subsequently, the process is repeated by heating the water in the second tank (charging), while the water in the first tank is used for industrial process heating (discharging). A principal sketch of the ISEC system is shown in figure 3. The discharged water is used for heating, while the tank is filled with cold water, but maintaining temperature stratification in the tank. The cold water will be in the bottom of tank and heated water in the top. The principle described above is mainly relevant if there is a relatively large temperature difference between the inlet temperature of the cold water and the outlet temperature of the hot water for the industrial process. Traditionally, as shown in figure 1, a direct heating from the inlet temperature to the outlet temperature is carried out simultaneously by means of the heat pump. In principle, this can be done with or without storage of water. The ISEC concept has previously been described by Lôffler (Lôffler, 2014) (Lôffler, 2015), Rothuizen et al (Rothuizen, et al., 2014) and Olesen et al (Olesen, et al., 2014).

The ISEC heat pump system involves heating of water in a tank with a gradually increasing temperature e.g. heating the water 4 K per circulation of the water. The ISEC concept requires several recirculations of the charging water to obtain the final temperature of process water. The mass flow rate of the recirculation stream is higher than for the discharging tank in order for the tank to be charged when the discharging tank is empty assuring continuous availability of process water.

This method results in the achievement of a substantial improvement in heat pump efficiency as it only needs to raise the temperature relatively few Kelvin. Initially, the heat pump operates at a low temperature while it operates at a substantially higher temperature at the end of the process. This increase in temperature affects the heat pump performance; the mean COP over the period of a complete charging process approaches the COP of a cycle operating at the thermodynamic average temperature of the water. To ensure low losses in the process during discharge, a good stratification of the water in the tanks should be maintained so that, in principle, there is an infinitely small layer of separation between hot and cold water.

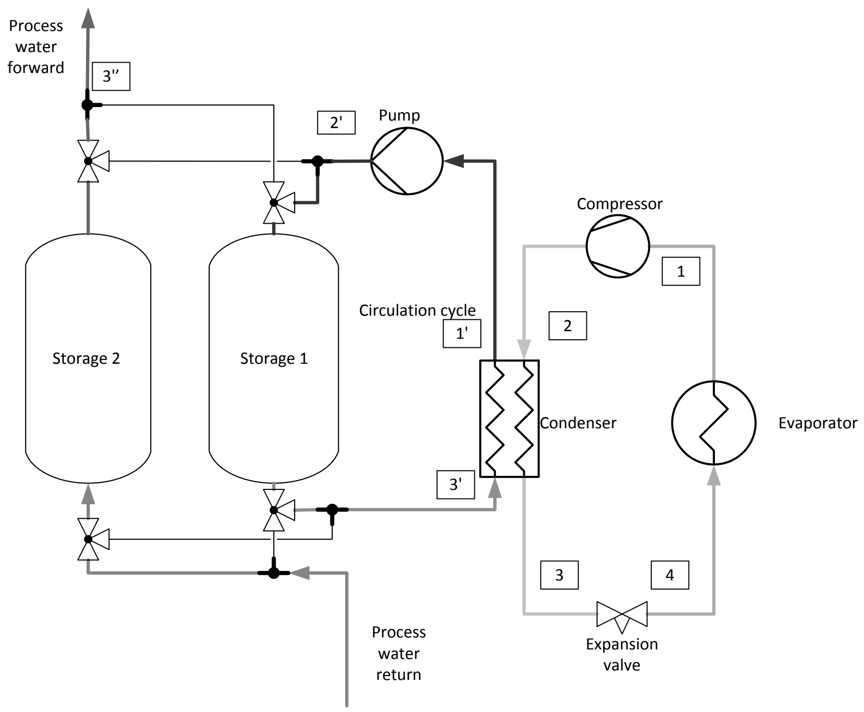

Figure 3 - A sketch of the ISEC system. Water is charged by circulating from tank one through the condenser while the water tapped from tank two is used to cover demand. Water is circulated multiple times through the condenser and tank one. When tank two is fully discharged, the system switches so tank one is tapped while tank two is charged by circulating water multiple times through the condenser.

The stratification comes naturally due to buoyancy. However, a number of factors cause a degradation of the stratification. The two main concerns are 1) The inlet of water in the tanks can cause disturbance and mixing of the two water volumes and 2) Heat conduction between the water layers and in the tank wall results in temperature equalization. By proper design of the inlets and the tanks, it is assumed to be possible to maintain a good stratification.

3.2. Methods

3.2.1. Dynamic model of the ISEC system

A model of the ISEC system was implemented in Dymola, a software specifically designed for dynamic simulations, and system simulations were carried out. Dymola enables models to be reused and adds a graphical interface to the Modelica language (Dassault Systèmes AB, 2010). The model was implemented including all major components and a simple control of the system based on water temperature, deciding when to change tanks for charging and discharging. Mass and energy balances were applied for all components. The key components such as the condenser, the compressor and the tanks were modelled in greater detail, while the evaporator, the throttling valve and the pumps were based on simpler models. The models are described in the following sections. Fluid properties were found using the fluid property library CoolProp (Bell, et al., 2014). All the component models except for the tank were quasi static models, while the tank model was dynamic; hence the change of water temperature in the condenser eventually affected the water temperature out of the tank, which then forced the temperature in the condenser to increase during the charging cycle.

Condenser and evaporator

The condenser was modelled as one component, but with both a desuperheating and a condensing part. In order to model the heat transfer the log mean temperature difference was applied for each section (Incropera, et al., 2007). The condenser was considered to be a plate heat exchanger. The heat transfer coefficients were found from empirical correlations summarized in table 1. The calculated heat transfer coefficients were together with the inlet temperature of the water and the outlet temperature of the compressor used to find the condensation temperature for a given heat transferring area.

The evaporator was modelled using a constant overall heat transfer coefficient and heat transfer area as inputs. The evaporation temperature was found from the temperature difference and the constant heat transfer coefficient and area.

Flow	Heat exchanger	Correlation
Single phase flow Water and refrigerant	Condenser	Local heat transfer coefficient (α) (Martin, 2010) (Martin, 1996)
Two phase flow Refrigerant	Condenser	Local heat transfer coefficient (α) Yan (Yan, et al., 1999)
Single and two phase flow Water and refrigerant	Condenser and Evaporator	Log mean temperature difference T_{LMTD} (Incropera, et al., 2007)

Table 1 - The correlations used in the condenser and evaporator models.

Compressor

Multiple compressor simulation models were used depending on the refrigerant. For ammonia calculations at constant isentropic efficiency was given, while for the R134a calculations used to compare the model with test results eq. 1 and 2 was used. One with a given isentropic efficiency and one using the compressor polynomials given in eq. 1 and eq. 2. The isentropic efficiency of the compressor was found from compressor polynomials. The cooling capacity, $\dot{Q}_0$, was calculated from eq. 1 and the compressor power, $\dot{W}$, was calculated from equation eq. 2.

$$\dot{Q}_0 = C_0 + C_1T_0 + C_2T_c + C_3T_0^2 + C_4T_0T_c + C_5T_c^2 + C_6T_0^3 + C_7T_cT_0^2 + C_8T_0T_c^2 + C_9T_c^3 \text{ [W]} \quad (1)$$

$$\dot{W}_{comp} = C_{10} + C_{11}T_0 + C_{12}T_c + C_{13}T_0^2 + C_{14}T_0T_c + C_{15}T_c^2 + C_{16}T_0^3 + C_{17}T_cT_0^2 + C_{18}T_0T_c^2 + C_{19}T_c^3 \text{ [W]} \quad (2)$$

T_0 [°C] is the evaporation temperature, T_c [°C] is the condensing temperature and $C_{0..19}$ are compressor specific constants supplied by the manufacturer.

Tank

The tanks were modelled according to the methods presented by Cruickshank (Cruickshank, 2009). The tank was discretized in vertical direction by dividing it into a specified number of layers. The energy balance for each layer was defined as shown in eq. 3:

$$M_ic_p\frac{dT_i}{dt} = \frac{(k+\Delta k)A_{c,i}}{\Delta x_{i+1\to i}}(T_{i+1} - T_i) + \frac{(k+\Delta k)A_{c,i}}{\Delta x_{i-1\to i}}(T_{i-1} - T_i) + U_iA_{s,i}(T_{env} - T_i) + \dot{m}_{down}c_p - \dot{m}_{up}c_pT_i - \dot{m}_{down}c_pT_i + \dot{m}_{up}c_pT_{i+1} + \dot{m}_{in}c_pT_{in} - \dot{m}_{out}c_pT_i \text{ [J]} \quad (3)$$

The change in mass and temperature in each node, *i*, is calculated by taking into account: the conduction between adjacent layers, given by $\frac{(k+\Delta k)A_{c,i}}{\Delta x_{i\pm1\to i}}(T_{i\pm1} - T_i)$, the heat transfer through the wall to the surroundings, given by $U_iA_{s,i}(T_{env} - T_i)$, mass that enters and/or exits to/from the adjacent layers: $\dot{m}_{down,up}C_p(T_{i\pm1})$ and finally at the boundary conditions: the mass into and out of the tank at the boundary $\dot{m}_{in,out}C_pT_{in,out}$.

Reduction valve and pump

The reduction valve was modelled as an isenthalpic process where the pressure drop causes a change in temperature. The pump was considered to be adiabatic and its energy consumption was found from the pressure losses in the system and a given isentropic efficiency.

3.2.2. Test setup to demonstrate the ISEC concept

Two test setups were built in order to prove the concept and validate the dynamic models. The first setup is the tank system, which was used for examining the stratification of the water in the tanks (Olsen, et al., 2015). The second system is a heat pump test setup, which has the ability to recirculate water through the condenser. The two systems were built separately and tested independently of each other. The tank test setup is shown in figure 4 (left) and consisted of two tanks of 0.110 m³ each, which were filled with water. During test, one of the tanks was filled with water, which was recirculated at a mass flow rate of 0.12 kg/s through an electrical heater to heat up the water. The other tank was discharged into a sink at a mass flow rate of 0.11 kg/s. Upon complete charging of one tank, the valve settings were switched in order to simulate the ISEC system with tank changes and multiple charges and discharges.

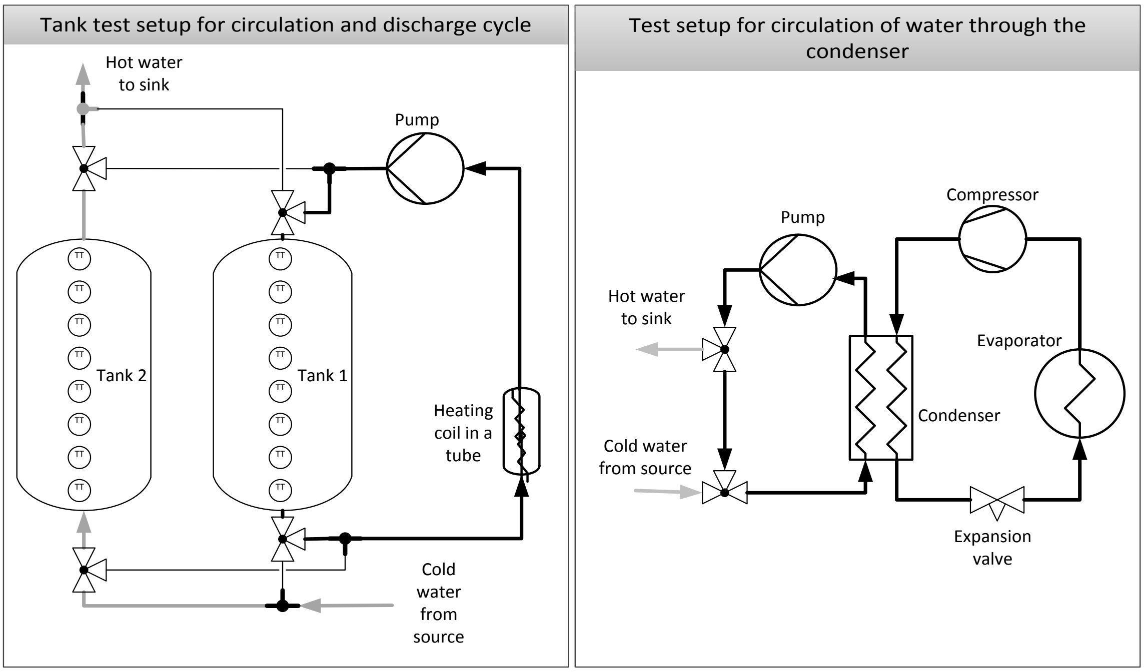

Figure 4 - The two test setups made to investigate prove the ISEC concept. Left: The test setup for the tank system. Right: The heat pump test system.

The heat pump test system used R134a as refrigerant and was capable of recirculating water in a pipe system through the condenser. The water loop was charged with cold water. During the run, the water was recirculated until a given temperature was reached. Then the system was flushed, which resulted in the hot water being discharged, while the system was charged with cold water. The heat pump test system is shown in figure 4 (right). The results were obtained from experiments made using the following procedure. First the pipes and the condenser were filled with cold water at $20\ °\mathrm{C}$ from the source and the heat pump was turned on. The water was then recirculated through the system and the condenser with a mass flow rate of app. 1.43 kg/s. When the water reached a temperature of 60 °C the water was flushed into the sink, new cold water from the source was tapped and the heating cycle was repeated. For the tests the condenser had an area of $3.48\ \mathrm{m}^2$, the nominated capacity of the compressor was $9.4\ kW$ with at volume flow rate of $40\ \mathrm{m}^3/\mathrm{h}$, the UA value per area of the evaporator was $5740\ \mathrm{W}/(\mathrm{m}^2\mathrm{K})$, based on data supplied by the manufacturer.

Figure 5 shows 3 consecutive heating cycles in the heat pump test system. It is shown that the heating of the water happens gradually over each cycle. The temperature of the water was measured on the outside of the pipes and a time delay due to the conduction through the pipes must be expected as the charging cycle was fast compared to the speed of the heat transfer through the tube walls, this is an observation based on visual inspections and it is obvious from the results. When changing the water from hot to cold water in the cycle shown in figure 5, it may be seen that the outlet temperature of the valve drops rapidly to below -10 °C. This is due to the dynamics of the system and the influence of changing from a condensing temperature of 60 °C to 20 °C.

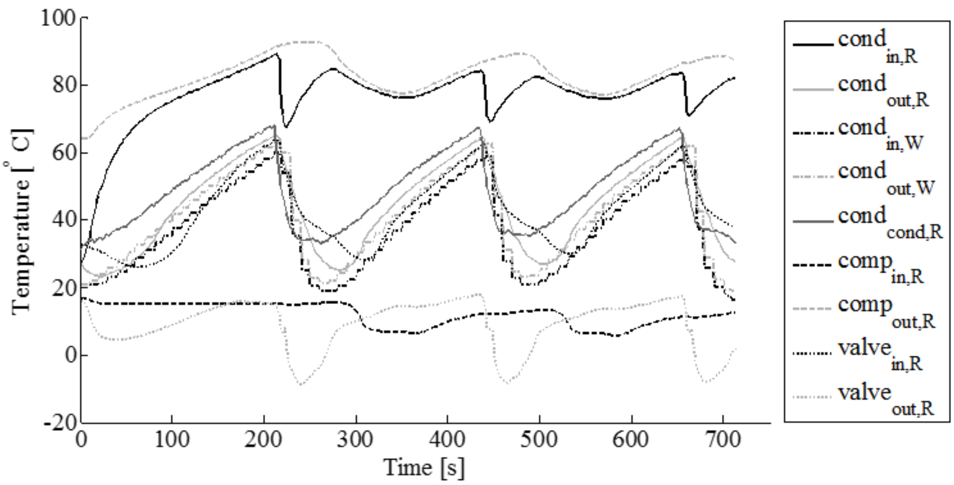

Figure 5 – Three consecutive heating cycles in the heat pump following the gradual heating principle of the ISEC concept. The temperature of the water (W) and refrigerant (R) is shown for the inlet (in) and outlet (out) of the condenser (cond), the compressor (comp) and the valve.

Figure 5 shows 3 consecutive heating cycles in the heat pump test system. It is shown that the heating of the water happens gradually over each cycle. The temperature of the water was measured on the outside of the pipes and a time delay due to the conduction through the pipes must be expected as the charging cycle was fast compared to the speed of the heat transfer through the tube walls, this is an observation based on visual inspections and it is obvious from the results. When changing the water from hot to cold water in the cycle shown in figure 5, it may be seen that the outlet temperature of the valve drops rapidly to below -10 °C. This is due to the dynamics of the system and the influence of changing from a condensing temperature of 60 °C to 20 °C.

The test of the charging and discharging of the tanks was carried out with the following procedure: Both tanks were filled with cold water. Then tank 1 in figure 4 (right) was charged by circulating the water twice past the heater. The water temperature raised approximately 20 K per circulation, which means that the water was heated from 20 °C to 60 °C in two circulations. When the first tank was charged, the tanks were shifted and charging of the second tank was started, while the first tank was discharged. This process proceeded a number of times, while the temperature was measured on the outside of the tank at different heights. For further information on the tests and test setup for the tanks, see Olsen et al. (Olsen, et al., 2015).

3.3. Results

3.3.1. Model validation

The heat pump model

The heat pump model was compared to test data from the test setup and the temperature development in the modelled tank was compared to the results from the test stand with the tanks. For comparison of the heat pump, only one heating process was used for validation. The heating process used is the second in figure 5, i.e. the heating process from approximately 250 seconds to 450 seconds.

Figure 6 (left) compares the pressures at the inlets and outlets of the condenser and evaporator of the heat pump tests to the model. Figure 6 (right) shows a comparison between the measured inlet and outlet temperatures, the condensing temperature, and the calculated temperatures. It may be seen that the measured and modelled pressures in the condenser are very similar. The model assumes constant pressure in the heat exchangers and therefore the inlet and outlet pressure are the same while a small pressure drop can be observed in the measured values. The evaporating pressure is higher for the model than for the measurements. This difference could be explained by the constant UA-value used as input to the evaporator model. The value obtained from the manufacturer was found from a different set of temperatures and at different mass flow. The calculated condensing temperature is very close to the one measured. However, the measured temperature out of the condenser is lower than the calculated temperature. This can be explained by the pressure loss in the heat exchanger and a little sub-cooling of the refrigerant, which occurs before the refrigerant leaves the condenser. The model assumed no pressure loss and no sub-cooling in the condenser. A significant difference between the measured and calculated inlet temperature to the condenser is a result of the dynamics of the tested system.

The heat capacity of the system is not taken into account in the model. This explains the higher temperature of the refrigerant in the beginning of the charging period and the lower temperature at the end observed from the measurements compared to the model. Figure 7 (left) shows the measured and modelled temperature of the water at the inlet and outlet of the condenser. The measured water temperature is a little lower than the modelled temperature. Though, the temperature increase of the water in the condenser is almost the same. The difference in temperatures could be explained by the time delay in the temperature measurement equipment. This could also explain the small decrease in temperature initially in the tests.

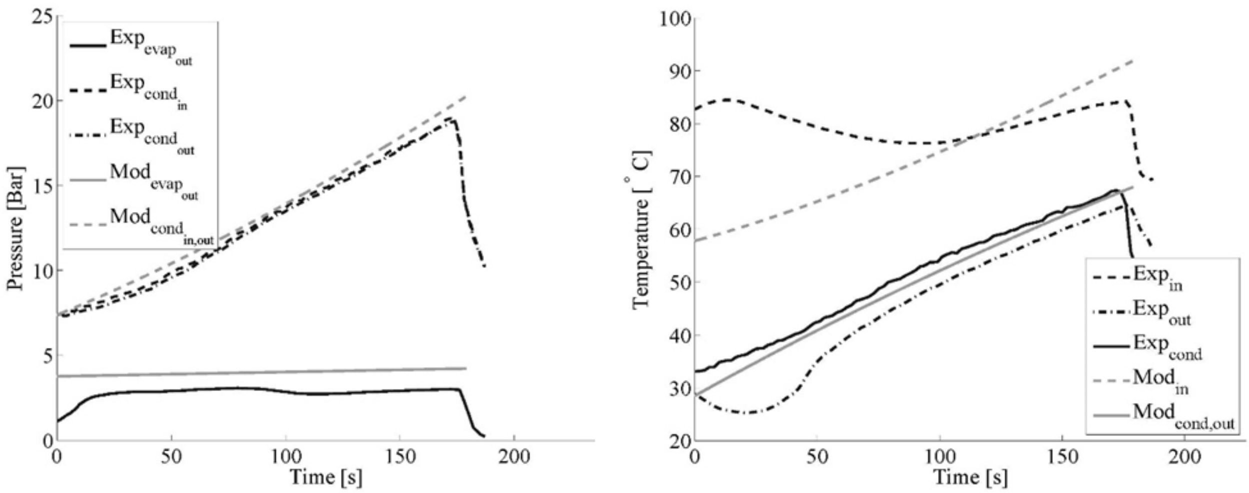

Figure 6 – Left: The measured and modelled pressures of the refrigerant at the inlet and outlet of the condenser and evaporator. Right: Comparison of the inlet, outlet and condensing temperatures of the refrigerant of measurements and model.

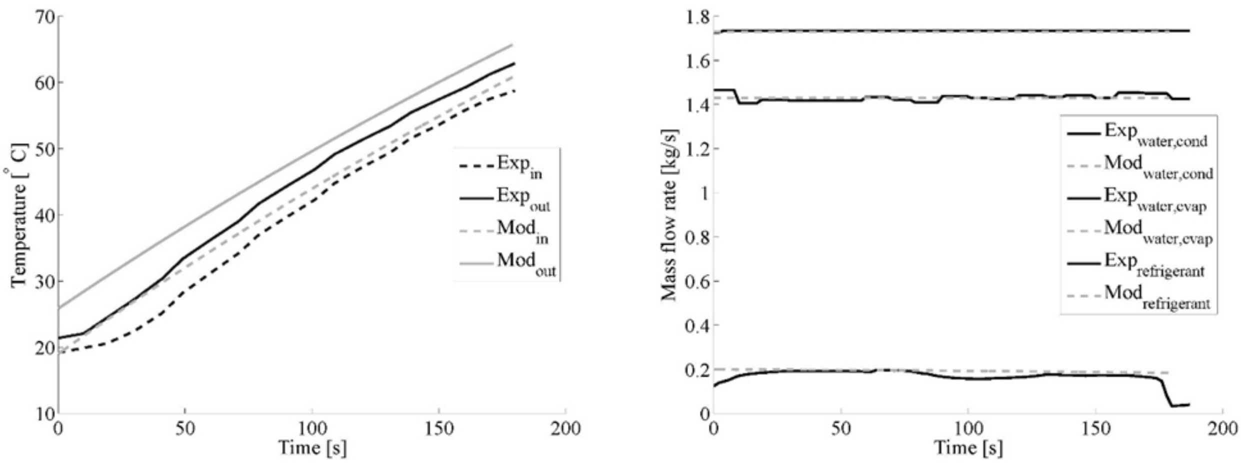

Figure 7 – Left: The water temperature into and out of the condenser for both the model and test. Right: The Mass flow rates of the water on the secondary side of the condenser and evaporator and the refrigerant mass flow rate in the heat pump.

A comparison of the measured and modelled mass flow rates is shown in figure 7 (right). The mass flow rate of the water in the experiments is set by the pump and the refrigerant flow by the compressor. In the model, the water mass flow rates were set to match the flow of the test and the refrigerant mass flow rate was calculated using the compressor polynomials. It is seen that the water mass flow rates are almost identical, while the mass flow rate of the refrigerant obtained using the model is a little higher than in the measured one. This could be due to the lower evaporation temperature, see figure 6 (left,) in the test, causing a higher pressure difference across the compressor affecting the volumetric efficiency as the density is lower at the inlet. Furthermore, it has been observed that the overall heat transfer coefficient given by the manufacturer is $2014\ \mathrm{W/m^2K}$ and the calculated overall heat transfer coefficient has a deviation between $2.3\ \%$ to $15.8\ \%$ of the given value.

The tank model

The following shows a comparison between the tank model in Dymola and tests done on the tank setup shown in figure 4 (left). Figure 8 shows a comparison of the temperature distribution in the tanks for both discharging and charging.

For the test, 10 thermocouples placed evenly spaced on the outside of the tank in vertical direction were used to measure the temperature, see figure 4 (left). For the tank model 100 control volumes of equal size, distributed vertically was used. In the figure, the temperature of every tenth volume is shown. Comparing the test with the model of tanks charging and discharging, it may be seen that the change in temperature is delayed in the test. Since the temperature measurements are made on the outside of the tank, a time delay due to conduction through the tank wall is expected.

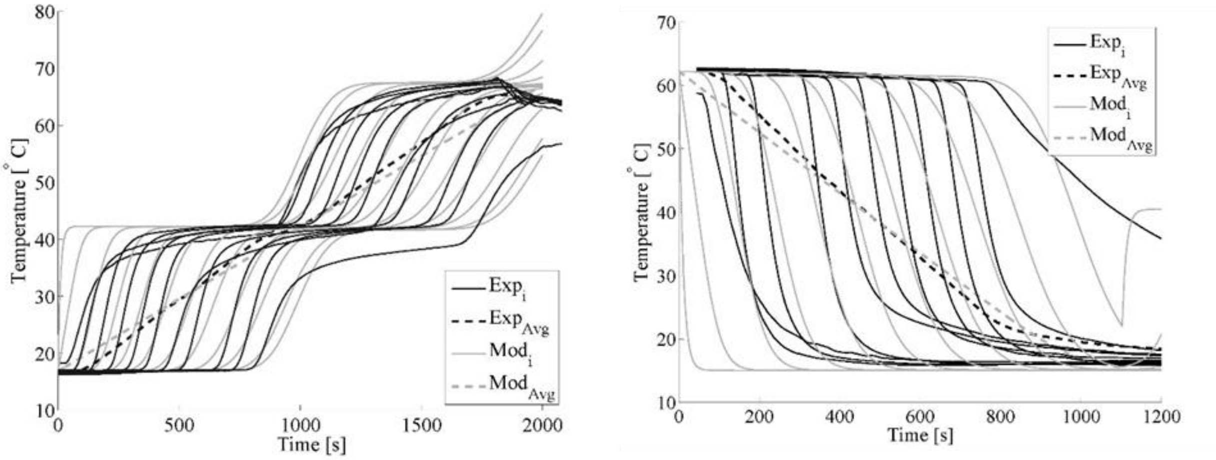

Figure 8 – Comparison of the measured and modelled temperatures in the water tank during charging and discharging shown for 10 equally distributed layers up through the tank and the average temperature of the water in the tank. Left: Charging of a tank. Right: Discharging of a tank.

Considering figure 8 (left), the modelled and tested temperatures follow the same pattern and the average water temperature calculated by the model is almost the same as the measured one. Considering the discharging tank, the model results show the same pattern as the measured results disregarding the offset in time, which partly comes from the time delay of the thermocouples. The most significant difference is the temperature of the top layer in the tank, which in the experiment cools down much slower than in the model. The explanation for this has to be found in the geometry of the outlet nozzle and the tank geometry as hot water gets trapped at the top of the tank. The calculated average temperature in the tank also fits well with the measured average temperature during discharge. A comprehensive study of the tanks is presented by Olsen (Olsen, et al., 2015).

3.3.2. Numerical calculations and optimization of the system

This section presents an analysis of the ISEC system and compares the different results to the conventional direct heating heat pump. The first parameter variation is of the number of times the water is recirculated through the heat exchanger before it reaches the desired temperature. The second parameter variation considers the heat exchanger size for a constant mass flow rate. The third analysis considers the influence of the evaporator temperature on the COP. The last analysis considers the two systems with, without subcooling, and with different isentropic efficiencies of the compressor.

Assumptions for simulations

The refrigerant for all the systems is ammonia (R717). The initial temperature of the water is 40 °C and the temperature needed for the process is 80 °C. The process needs a mass flow rate of 16.6 kg/s of hot water. The storage tanks are 10 m³ each and with the given mass flow rate the tank discharges in 10 minutes (600 seconds). The other tank therefore needs to be charged in 10 minutes. The charging tank mixes the entering water with the existing water in the tank fully, while the tank, which is discharged, is stratified with the hot water used for the process in the top and the returning water in the bottom. The ISEC system increases the water temperature with approximately 4 K for each circulation of the water. It thus recirculates the water 10 times before reaching the target temperature while the direct system heats the water in one temperature lift, hence the ISEC system has a 10 times higher mass flow rate. The pressure loss in the condenser is constant at 0.5 bar for all calculations. The evaporating temperature is 22 °C. The compressor is assumed to have a constant isentropic efficiency of 50 %, unless other is specified. The ISEC solution is compared to a conventional system. The condensing temperature for the conventional heat pump system is 81 °C, which gives a minimum temperature difference of 41 K. Except for the condenser the heat pump is assumed to be based on the same components for both a conventional system and the ISEC system. The water system will require a significantly larger mass flow for the ISEC process. It is assumed that the pump and the pipe system are dimensioned accordingly. The first part of the analysis is done without using the subcooler as indicated in Figure 3 and the last analysis is both with and without subcooling.

Thermodynamic comparison of the two systems

The conventional direct system and the ISEC system are by means of technology very similar, though the heating process of the water is different. Where the direct system heats the water by passing the condenser once, the water in the ISEC system passes the condenser a number of times and stores the water in tanks; in this case, the water is mixed inside the tank charging. The heating processes of the water for both systems are shown in figure 9 left. The right figure compares the COPs of the 2 systems; the ISEC average and the direct COP are comparable. “ISEC” is the COP at that given stage in the heating process, “ISEC average” is the average COP over the elapsed time and

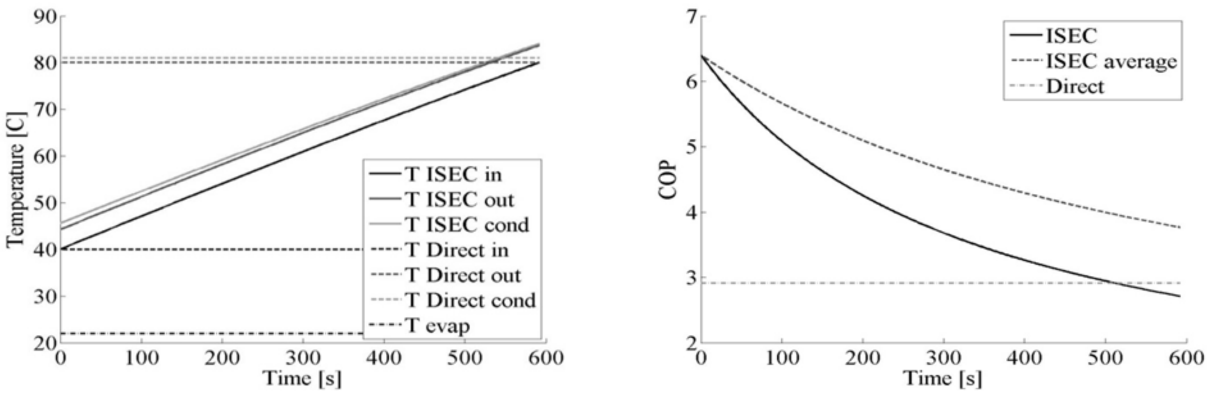

Figure 9 - The condensing temperature and the inlet and outlet temperatures of the water for the direct and ISEC systems. Right: The COP of the ISEC system at the time in the process, the average ISEC COP and the COP of the direct heating system.

“Direct” is the COP of a conventional heat pump heating in one step. It is shown that the average COP of the ISEC system is higher than the COP of the direct heat pump system and that eventually the ISEC system has a COP that is 3.8. The conventional system has a COP of 2.8. The COP of the ISEC system at a given time is lower than the average COP. For the last 60 s, it is lower than for the conventional heat pump system, as the condensing temperature is higher at this point.

The effect of recirculation of the water in the ISEC system

The ISEC system and the conventional system are compared with respect to the number of times the water is circulated in the ISEC system in 600 seconds.

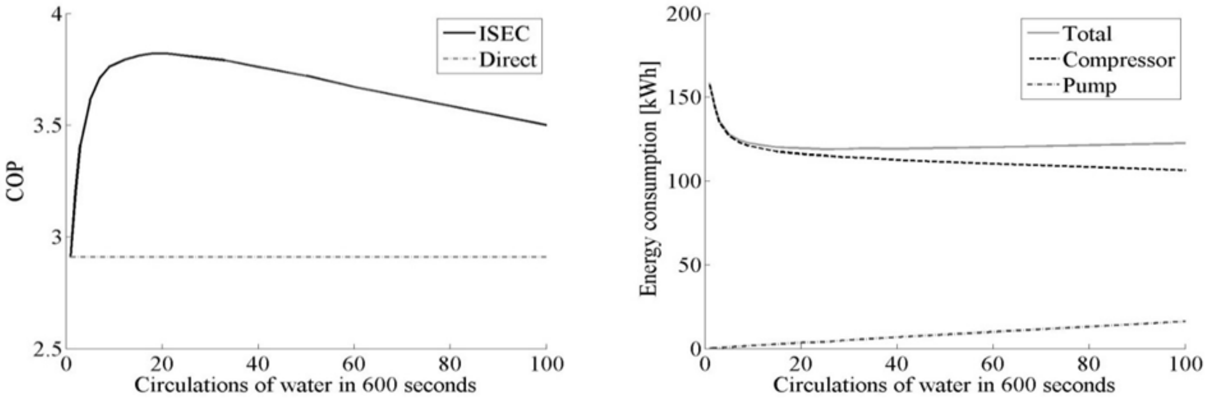

Figure 10 - Left: COP as a function of recirculation of the water through the condenser. Right: Power consumption as a function of number of recirculations.

The conventional heat pump, heats the water from 40 °C to 80 °C with only one passage through the condenser, while the ISEC system recirculates the water a number of times. Figure 10 left shows the COP as a function of circulations with reference to the conventional system. Figure 10 left shows that there are an optimum number of recirculation through the condenser. The optimum number of recirculation for this specific system is 20 and the corresponding COP is 3.8, compared to the conventional heat pump system with a COP of 2.91. This is an increase in COP of 24 %. The reason for the peak in COP is the power consumption of the system in shown in figure 10 right. The graphs shows that the compressor consumption is decreasing while the pump electricity consumption is increasing linearly as the pressure loss is 0.5 bar in all cases, as the water side is assumed to be dimensioned for the required water flow. For a number of recirculation of 20 or more the increase in pumping power becomes larger than the saving in compressor power. The reason for the decrease in compressor power consumption is that the condensing temperature decreases as the number of recirculation increases. The more times the water is circulated, the lower is the temperature increase of the water per circulation and the lower is the average condensing temperature, even though the pinch point temperature increases slightly, which can be seen in figure 11. This analysis shows that the number of circulations has an optimum for the system and that the temperature increase of the water decreases per circulation as the number of circulations increases. Furthermore, the minimum temperature difference of the condenser increases as the circulation rate increases and the condenser temperature decreases.

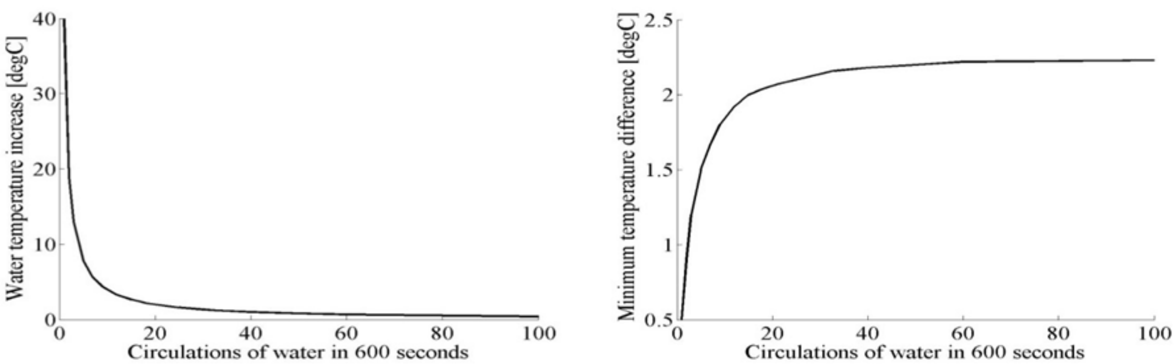

Figure 11 - Temperature difference of water across the condenser as a function of recirculation. Right: Minimum temperature difference as a function of recirculation.

Condenser area and the evaporating temperature

A parameter variation of the heat transfer area of the condenser is done for a constant mass flow rate of water. This shows how the condenser size affects the COP of the system. The mass flow rate corresponds to circulating the water 12 times through the condenser in 600 seconds, 12 circulations is chosen as the COP does not increase significantly by increasing the number of circulations as shown in figure 10 left, but the pressure drop decreases with almost a factor of two, which means the size of the heat exchanger also decreases. Figure 12 left shows the relation between condenser area and the COP. For the conventional heat pump system the heat exchanger, size does not affect the COP noticeable, as the condenser temperature has to be above 80 °C. For the ISEC concept, the condenser temperature is always above the outlet temperature of the water, and as the area increases so does the COP. This is because the condensation temperature decreases. The largest marginal increases in COP with respect to condenser size are found below 200 m2. The effect of the evaporating temperature on the ISEC system and the conventional heat pump system are shown in figure 12 right. The mass flow rate of the water side of the condenser is constant and the temperature difference between evaporation temperature and the water into the evaporator is 7 K. As the evaporator temperature approaches the inlet temperature of the water in the condenser, the COP increases for both systems as the volumetric efficiency decreases the mass flow rate of the refrigerant increases, increasing the performance of the condenser. However, the increase of the ISEC system is larger than for the conventional heat pump system.

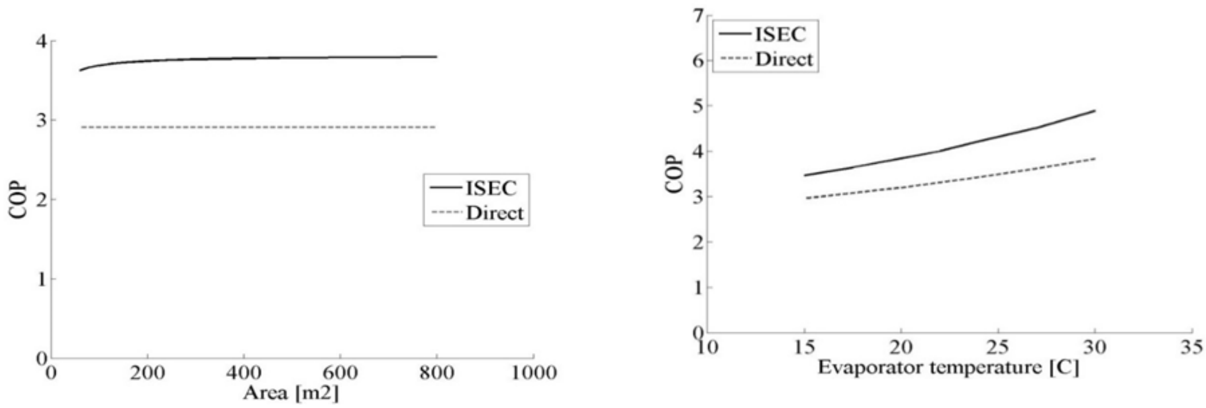

Figure 12 - Left: COP as a function of the condenser area. Right: COP as a function of evaporator temperature.

Compressor efficiency and subcooling

The previous comparisons have been done without including subcooling in the systems and with a constant isentropic efficiency of the compressor of 50 %. This section includes subcooling for both systems, where the refrigerant is cooled down to 44 °C. Furthermore, a parameter study of the isentropic efficiency of the compressor is included. For the conventional system, the subcooling is used to heat up the water before the condenser. In the ISEC system, the subcooling is used to heat the returning water to the tank discharging to maintain a constant subcooling in the heat pump. This results in a higher initial temperature when the tank has to be charged after the discharge. Figure 13 left shows the COP as a function of the isentropic efficiency of the compressor, for both systems with and without subcooling. Increasing the isentropic efficiency of the compressor increases the COP linearly for all cases. The better performance of the ISEC system is similar for all the cases. The effect of the subcooling on the systems is significant. The increase in COP with and without subcooling is shown in figure 13 right. The "ISEC" and "Direct" show the improvement of adding subcooling to the existing system, while the other bars compare the two systems both with and without subcooling. It is worth noticing that the ISEC system without subcooling has a higher COP than the direct system with subcooling and if the isentropic efficiency of the compressor is 10 % lower in the ISEC system, the two systems reach the same COP.

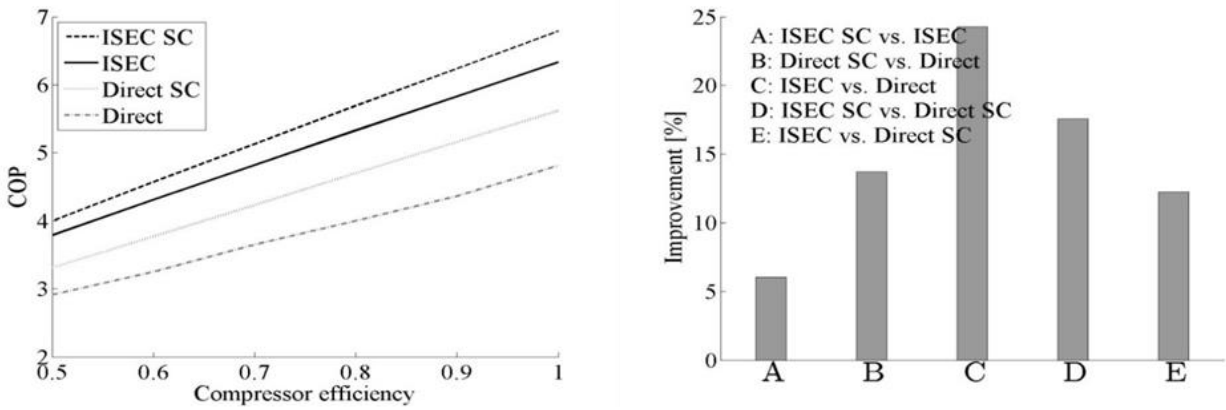

Figure 13 - Left: The COP as a function of isentropic compressor efficiency. Right: The system improvement when comparing the different heat pump sets ups between the ISEC system and the direct heat pump system.

3.4. Discussion

The ISEC system have several challenges that should be taken into account when considering the results of both the validation and the parameter variation done with the simulation model. The condenser does necessarily have to be larger in the ISEC system than in a direct heat pump system in order to minimize pressure losses and thereby the pump power as shown in figure 10 right. The condenser dimensioning should be done carefully for each specific system configuration. The size depends on the increased mass flow by circulating the water through the condenser, hence the number of recirculations. The circulation ratio influences the pressure loss and thereby the pump power needed. The tank system that needs to be added to the heat pump increases the complexity of the system. With an increased complexity there is a higher risk of malfunction and the control of the system becomes more advanced. The higher complexity of the system does not compromise the security of supply as the ISEC system can still run as a direct heat pump and heat the water directly to supply temperature without recirculation by bypassing the tank system. This could be an option if the demand suddenly increases, there is maintenance on the tank system or there is a malfunction. The tank system does also require piping and when changing between the tanks, hot water from the charge with a temperature of 80 °C would enter the cold tank before the condenser temperature is regulated down to match the cold temperature of the new charge cycle. If the tank has a good stratification this can potentially become a problem as the condenser would increase rapidly in temperature for short times, as one layer would be significantly hotter than the other layers. This is fortunately mainly a theoretical scenario, as mixing between the layers is present in the tests, but it should be considered if better technologies for stratification is used. Another risk related to the water in the piping system when changing tanks is that cold water might end up in the process stream before the hot water from the new tank run through.

The model made of the ISEC system have some discrepancies. The pressure loss across the condenser has not been investigated thoroughly and the model could be improved by including this. Although considering the results it can be seen from figure 10 right that it is possible to recirculate the water 20 times before the power consumption of the pump becomes higher than the decrease in compressor consumption due to lower condensing temperature. The evaporator temperature is calculated from a given constant heat transfer number. As it can be seen from figure 12 right the evaporator temperature and the COP of the system is correlated and a difference in evaporator temperature of 15 K makes a difference of 1.5 in COP. The evaporator temperature is therefore of significance for the COP and calculating the heat transfer number of the evaporator would improve and increase the accuracy of the model.

Another discrepancy of the model is, that it is made to optimize the thermodynamics of the system by means of the COP. The model does not consider investment costs or running costs, by including these it may give another optimum configuration.

This dissemination focus on a temperature lift of the process water of 40 K lifting the temperature from 40 °C to 80 °C. It would be interesting to investigate the ISEC concepts performance for other temperature glides of water. This should be considered in later work related to the ISEC system.

This scientific dissemination only considers new systems and compressors, which are running with similar efficiencies for the compared systems. If the ISEC system is retrofitted into an existing heat pump facility, the difference in operation COP may increase with more than 25 %, depending on the operational conditions the heat pump had before. If further work is carried out, this aspect should be investigated for a number of specific cases where an ISEC system substitutes a normal heat pump system.

3.5. Conclusion

The ISEC system is a new concept and it has previously been shown that an increase in COP of 16 % compared to a heat pump which heats without recirculation of water may be expected, when sub-cooling is present in the system (Rothuizen, et al., 2014). The comparison between the model and experiments shows a good consistency between modelled temperatures and pressures and the measured temperatures and pressures. The Dymola model seems to be capable of predicting the thermodynamics of the system during charging and discharging of the tanks, thus it can be used to predict the performance of the heat pump system and illustrate how different parameters affect the performance of the system. The heat transfer coefficient of the condenser is calculated using empirical correlations and the results show a deviation of less than 16 % from the heat transfer value supplied by the manufacturer, which is an acceptable result considering that the value from the manufacturer is given for a specific temperature interval and mass flow. The time delay of the temperature sensors has an influence on the results as the temperature measured on the water side is delayed compared to the pressure and temperature measurements of the refrigerant. For the results shown, it might explain some of the inconsistencies of the comparison between the calculated and measured results.

The COP of the ISEC system is potentially higher than for the conventional design. The case results show up to 25 % improvement, when the refrigerant is not subcooled. It is shown that there is an optimum based power consumption of the number of times the water should be recirculated through the condenser before the increase in pumping power becomes too large and the COP of the system decreases with increasing number of circulation. Increasing the size of the condenser increases the COP. When increasing the evaporator temperature, the COP of both the conventional and the ISEC system increases. This is expected as the pressure ratio across the compressor decreases, but the increase of COP is highest for the ISEC concept. The ISEC concept has proven theoretically to be of interest for heating water with energy savings of up to 17 % considering subcooling for both systems.

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4. PART II; Test of storage tanks

4.1. Test of storage tanks with electric heater

The most common method to achieve a high efficiency in case of a large temperature lift is to have a number of heat pumps connected in series. Hereby, each heat pump raises the temperature with a small share of the total temperature lift with a high system efficiency as a result. An alternative to this principle is the ISEC concept where the series of heat pumps is replaced by one single heat pump and two storage tanks. In the ISEC heat pump system, there is a gradual heating of water in a tank with an increasing condensation temperature in the thermodynamic cycle until the desired temperature has been reached.

In the first test set up, the heat pump is emulated by an electric resistance heater. The purpose is to isolate the challenges raised by operating the tank system separately instead of investigating the operation of both the heat pump and tank system simultaneously. The heat pump issues are described in the previous and the following chapters in this report.

The test setup uses an electrical resistance heater instead of a heat pump, as seen in Figure 14 and 15. One tank is "charging" (the water is being heated), while the other tank is "discharging" (the hot water is being tapped and replaced with cold process return water).

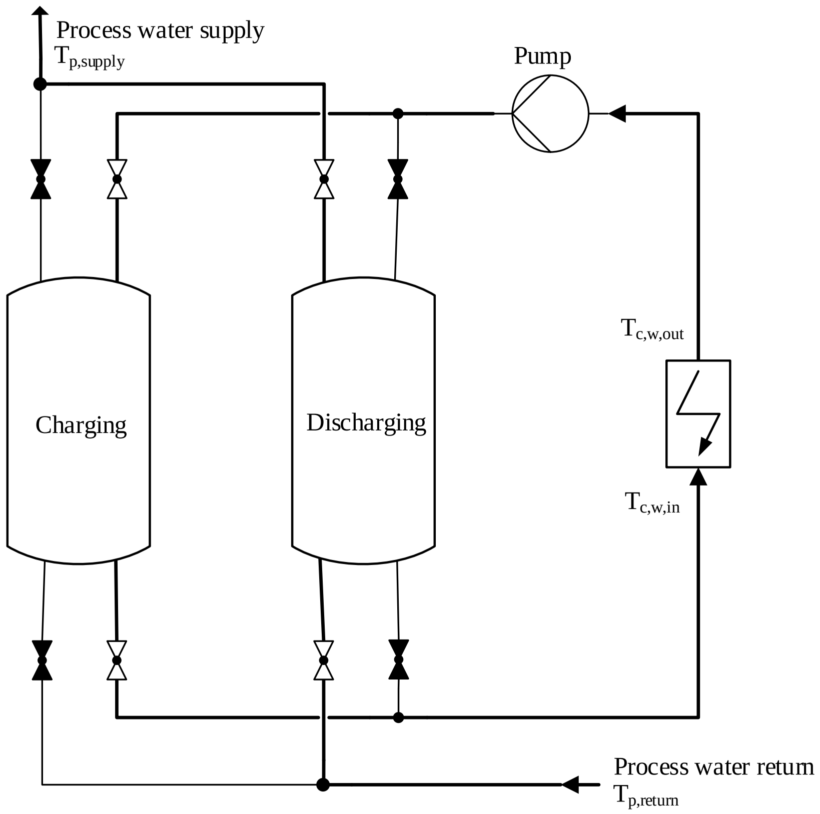

Figure 14 - A principle sketch of the tank test setup with two storage tanks, an electric resistance heater and eight valves. One tank is charging, while the other tank is discharging.

Figure 15 - Storage tank test setup with two storage tanks, an electric resistance heater and control system. The row of thermocouples is shown mounted on the storage tank.

Initially, both tanks are cold, i.e. discharged. When starting to charge, the water recirculates a number of times in a closed loop through the electric resistance heater (emulated condenser) and back to the tank. The circulation continues until the tank reaches the desired temperature, $T_{p,supply}$. Subsequently, the process is repeated by charging the second tank, while the water in the first tank is tapped for industrial process water, i.e. discharging. It is essential for the optimal efficiency that the tapped hot water and the cold return water filling the tank are separated by good stratification in order to supply a constant temperature ($T_{p,supply}$) to the industrial process. Stratification during charging is not of crucial importance in order for the system to be beneficial. However, the efficiency will increase if this is achieved, Olesen *et al.* (2014).

The ISEC concept results in an improvement of the heat pump efficiency as it only needs to raise the temperature of the water a few degrees, relatively, and thereby have a lower condensing temperature in most of the charging cycle. Initially, the heat pump operates at a low temperature, and at the end of the charging cycle, it operates at substantially higher temperatures. Moreover, the heat pump has a high COP to begin with, which decreases as the temperature increases.

A model of the heat flow in the storage tanks has been modeled. Details of the model are found in Olsen et al. 2015.

Two examples of simulated processes are shown below. In the simulation, the storage tank is divided into 10 nodes of equal size. Figure 16 shows a charging run, while Figure 17 shows a discharging run.

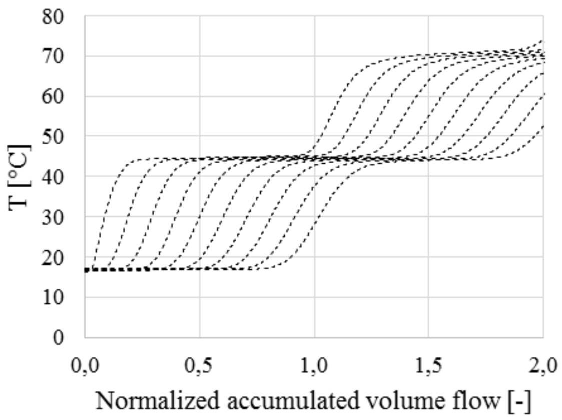

Figure 16 - Simulated node temperatures in the tank during charging at the downwards velocity of $1.36 \cdot 10^{-3}$ m/s.

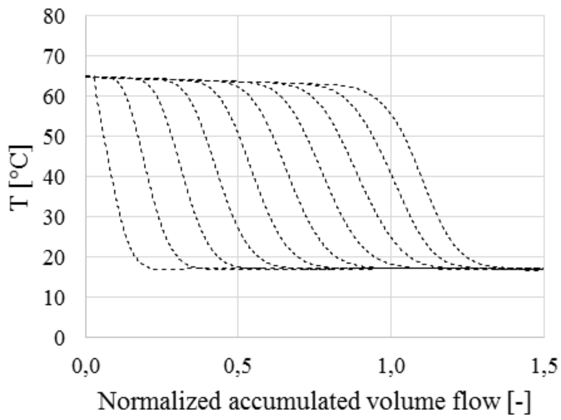

Figure 17 - Simulated node temperatures in the tank during discharging at the upwards velocity of $0.68 \cdot 10^{-3}$ m/s.

The flow velocity given in each image caption indicates the vertical velocity of the horizontal cross section of the tank. The elapsed time scale is given in normalized accumulated volume flow, which means that the value of one is the time required to charge or discharge the water in one full tank.

The charging run is characterized by a rising staircase-like structure with the node temperatures increasing sequentially from the top down (first *T1*, then *T2*, and so on). The two steps represent the heating in two stages, with a lower condensation temperature during the first stage. Increasing the flow rate will not affect the time needed to fully charge the tank (since the heater effect is fixed), but rather influence the resulting stratification in the tank by adding more stages (example measurements are included later).

The discharging run shows the reverse process with the node temperatures decreasing sequentially from the bottom up, but always in one single step, since discharging can only be done in one stage. Increasing the discharge flow rate will of course cause the tank to empty faster. However, because of the chosen normalized time scale, the plots would look very similar to the ones in Figure 17.

Experimental setup

Figure 18 shows the placement of the sensors mounted internally in the fluid stream. The flowmeters FL1 and FL2 measure the charging and discharging flow rates, respectively. The thermocouples TE1 and TE2 measure the temperatures inside the tanks, while TE9 and TE10 measure the water supply temperature and the return temperature, respectively.

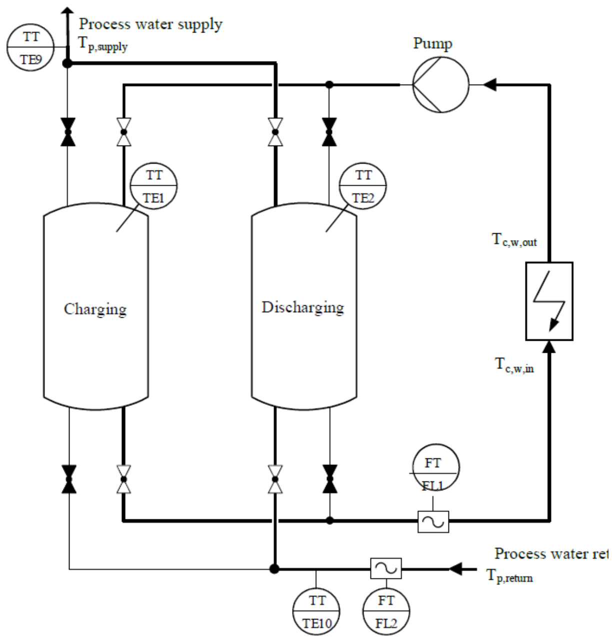

Figure 18 - Placement of the thermocouples and flowmeters mounted internally in the fluid stream.

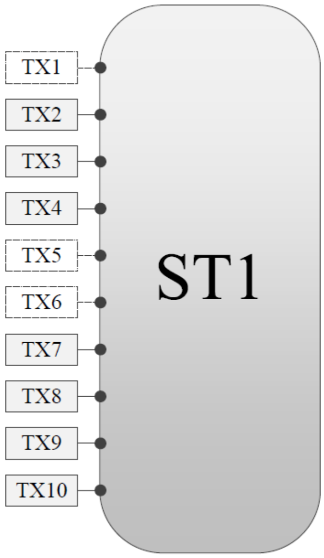

Figure 19 - Placement of the externally mounted thermocouples.

Figure 19 shows the placement of the thermocouples mounted externally down the side of the tank at regular intervals. The output data from TX1, TX5 and TX6 is disturbed by the heat capacities of adjacent flanges on the physical tank. Thus, it has been omitted from the experimental results.

Experimental Results

Figure 20 shows an example of a plot of the charging run corresponding to the simulation results in Figure 16. After charging is complete (just after two tank volumes on the time scale), the circulation is allowed to continue and to mix the contents of the tank. This is called post charge mixing. The shown temperatures are measured by use of the thermocouples TX2 through TX4 and TX7 through TX10, corresponding to the simulated temperatures T2 through T4 and T7 through T10, respectively.

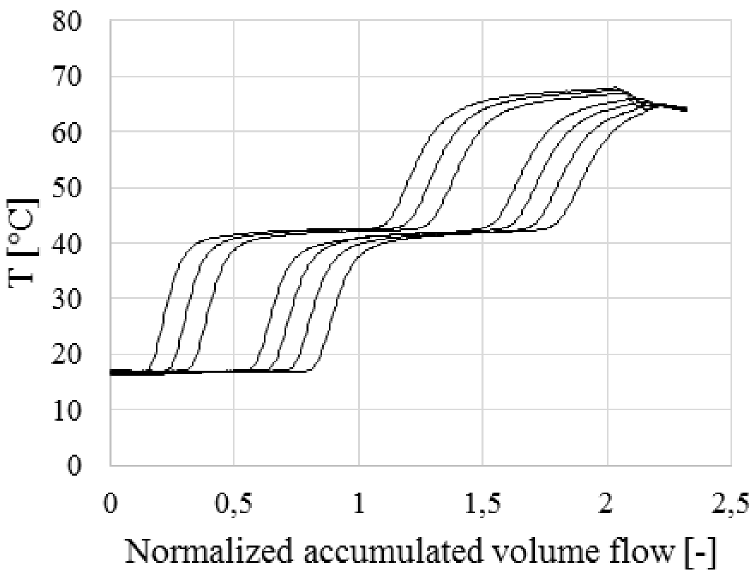

Figure 20 - Measured node temperatures in the tank during charging at a velocity of $1.36 \cdot 10^{-3}$ m/s.

When ignoring the excluded data sets, the plot strongly resembles the corresponding simulation result plot in Figure 16 with the same two-step staircase structure. Additionally, the data indicates that post-charge mixing of the contents of the tank occurs reasonably fast with only about 0.2 tank volumes of circulation being necessary in this case.

Figure 21 plots the output data from the sensors mounted internally in the fluid stream (see Figure 18), and it shows several cycles of the full, automated ISEC process. The temperatures inside both tanks are indicated, along with the supply and return temperatures as well as the circulation and discharge flow rates.

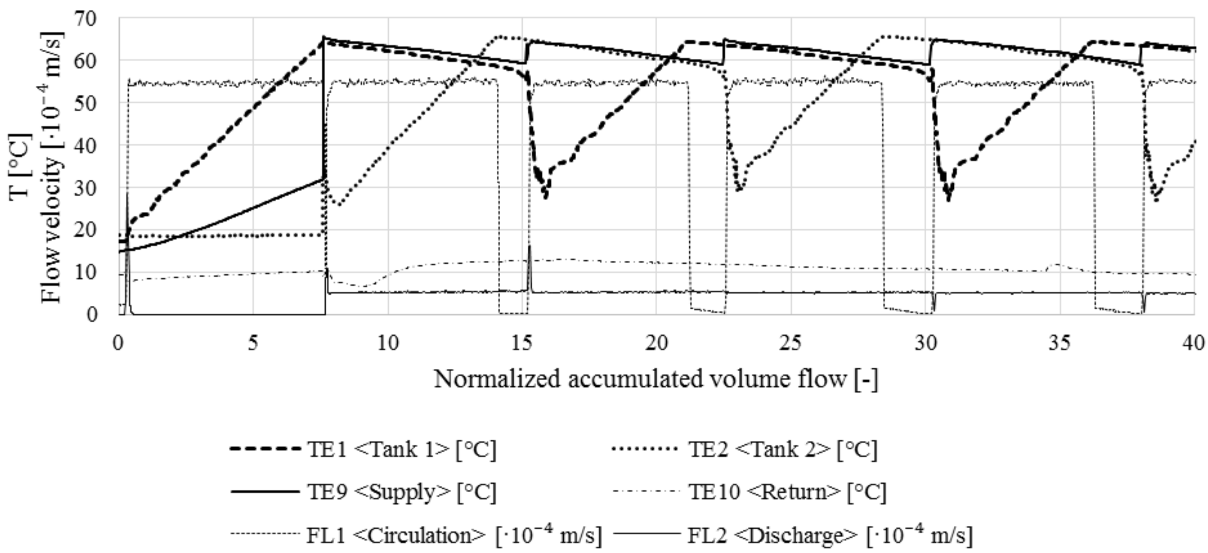

Figure 21 - Measured results from continuous cycle at a velocity of $5.42 \cdot 10^{-3}$ m/s during charging and $0.62 \cdot 10^{-3}$ m/s during discharging. The volume flow is normalized with respect to the charging flow rate. This data is measured by the internally mounted sensors (see Figure 18).

At about eight on the normalized time scale, process water starts to flow continuously at the desired return temperature range, as can be seen on the supply temperature and the discharge flow graphs. The circulation flow is intermittent, which indicates that charging is consistently complete before discharging, when this rather conservative discharge flow rate is used. Post-charge mixing is not applied in this case.

Evaluation

In all the following plots, the simulated node temperatures T1, T5, and T6 corresponding to the output from the external thermocouples TX1, TX5, and TX6 have been omitted. In all graphs, the solid lines represent measured data, while the dotted lines represent the simulation results.

A general displacement of the measured temperature levels can be seen in all the plots, with the simulated temperatures being consistently higher than the corresponding measurements. This is to be expected from the placement of the TX thermocouples (on the outside of the tank), since the model assumes that each node has a uniform temperature. However, the actual temperature on the outside of the wall will be lower than this average temperature due to the boundary layer closest to the wall, with an additional temperature drop through the wall material. In addition, there is a minor time delay due to the thermal capacity of the storage tank wall.

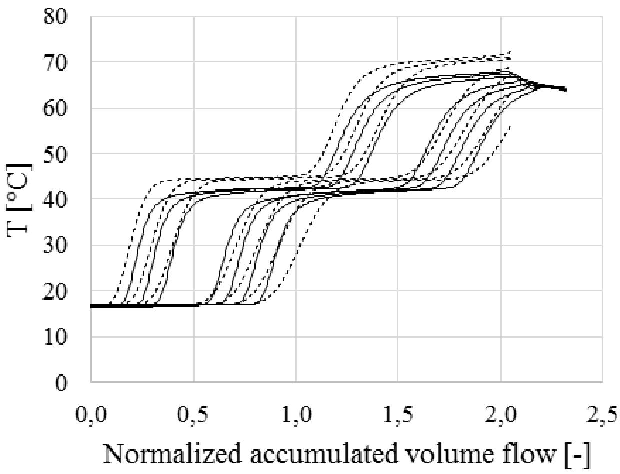

Figure 22 - Simulated and measured node temperatures in the tank during charging at a velocity of 1.36·10^{-3} m/s.

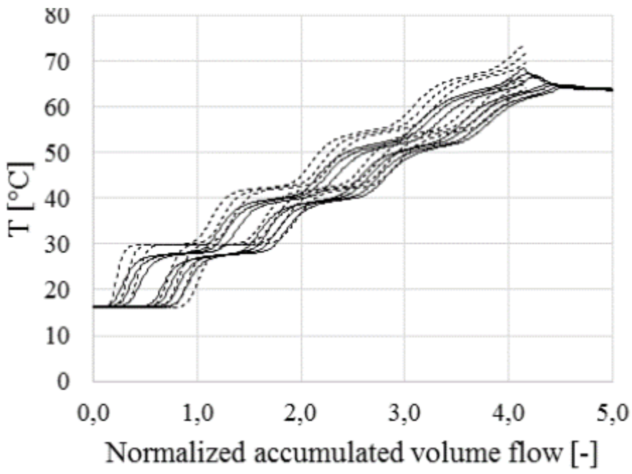

Figure 23 - Simulated and measured node temperatures in the tank during charging at a velocity of 2.71·10^{-3} m/s.

The above plots contain data from charging runs at different flow rates. Figure 22 shows the previously given two-stage charging run, while Figure 23 shows a run at twice the flow rate, yielding a four-stage run. Both plots show good correspondence between the simulation data and the experimental data. Figure 23 also indicates a gradual smoothing of the staircase as the process progresses through the stages.

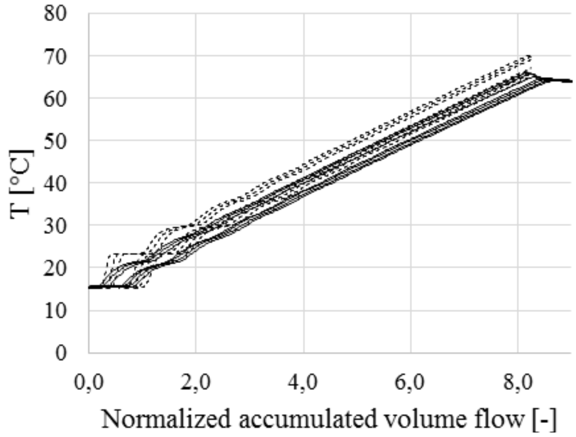

Figure 24 - Simulated and measured node temperatures in the tank during charging at the velocity of $5.42 \cdot 10^{-3}$ m/s.

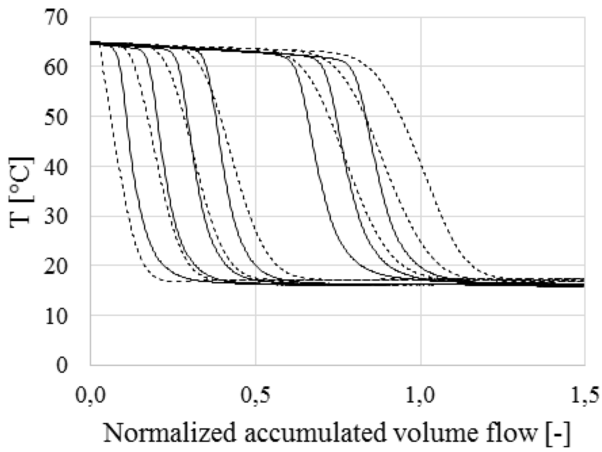

Figure 25 - Simulated and measured node temperatures in the tank during discharging at the velocity of $0.68 \cdot 10^{-3}$ m/s.

Figure 24 shows a charging run at twice the flow rate in Figure 23 (or four times the flow rate in Figure 22), yielding an eight-stage run. However, the smoothing effect is significantly stronger here than seen previously, and it has a more noticeable effect already on the second stage. The simulation results still provide an acceptable match. Please note that while the simulation results seem to match reality closely with regard to the smoothing effect, the smoothing of the simulation results is caused by numerical diffusion resulting from the previously stated compromise when implementing the model – a more fine-grained implementation would reduce this effect considerably.

Figure 25 contains the previously shown simulation data for the discharging run, along with its corresponding experimental results. Again, there is good correspondence between the simulation data and the experimental data.

Discussion

The overall purpose is to deliver process water at a uniform temperature. This can be done most efficiently by charging the tank with a minimum of mixing and discharging with a maximum of stratification with a narrow boundary layer between warm and cold water.

During charging, a low flow rate provides good stratification with relatively narrow boundary layers (minimum of mixing) between hot and cold water. However, the low flow rate results in a low COP due in a large temperature lift. Furthermore, this also demands a precise and difficult control of the volume flow rate and temperature increase ΔT in order to achieve the target temperature with a sufficient precision.

Running at a high circulation flow rate has the immediate advantage of a lowered average condensation temperature, and thus an increased COP. The stratification in the tank no longer has any readily identifiable boundary layers, and instead it takes on a more gradient-like appearance. However, the gradient distribution has the added advantage of requiring a less accurate flow control compared to low flow rate operation. The risk of overshooting the condensation temperature in order to satisfy the temperature limit will be less in this case. However, it must be remembered that a large number of cycles will increase the energy used for the pumps.

The mode of operation has to be evaluated by complete system simulations or experiments in order to find the most optimal flow rate and operation control.

During practical discharge operations, the placement of the control thermocouple inside the tank may cause the tank to be incompletely discharged, leaving a volume of unused heated water at the top of the tank, which reduces the efficiency. Proper placement of the thermocouple should alleviate this. Another option might be the placement of a separate temperature sensor in the pipe used for the discharging of water.

Conclusion

Measurements and simulations show good agreement, indicating that the utilized simulation is useable for estimating the time needed to complete a run, the type of temperature distribution in the tank (stratified or gradient), and the temperature levels at a given time.

Due to numerical diffusion stemming from the implementation of the model, the simulation is less accurate with respect to the thickness and exact location of any boundary layers. A less limited implementation of the model would noticeably reduce this effect.

A relatively high circulation flow, corresponding to a high number of passes through the heater (condenser), seems to be the best solution in that it provides a gentle temperature gradient down through the tank. By employing post-heat mixing, it will be possible to reduce this small gradient even further, and thereby decrease the temperature variance in the delivered process water. The measurements reported above show that the ISEC concept is feasible in practice in relation to the storage tank operation.

4.2. Test of storage tanks combined with a heat pump

In this test set up, the same storage tanks are applied, but the electric resistance heater is replaced by a heat pump (Figure 26 and 27). The experiences from this test is used for the operation of the tank system. In principle, the same control scheme is used for operating the storage tanks as with the electric heater.

The following test conditions are applied:

The volume flow during discharge is adjusted so that the storage tank to be discharged is emptied before the storage tank to be charged is fully charged. The compressor speed is constant. The compressor capacity is 32% of the maximum capacity. The tests are performed with a large condenser and without sub-cooling. The charging and discharging time is 28 and 22 minutes, respectively. During the charging period, the water in the storage tank passes four times. The volume flow during the charge and the discharge of the storage tanks is shown in figure 28.

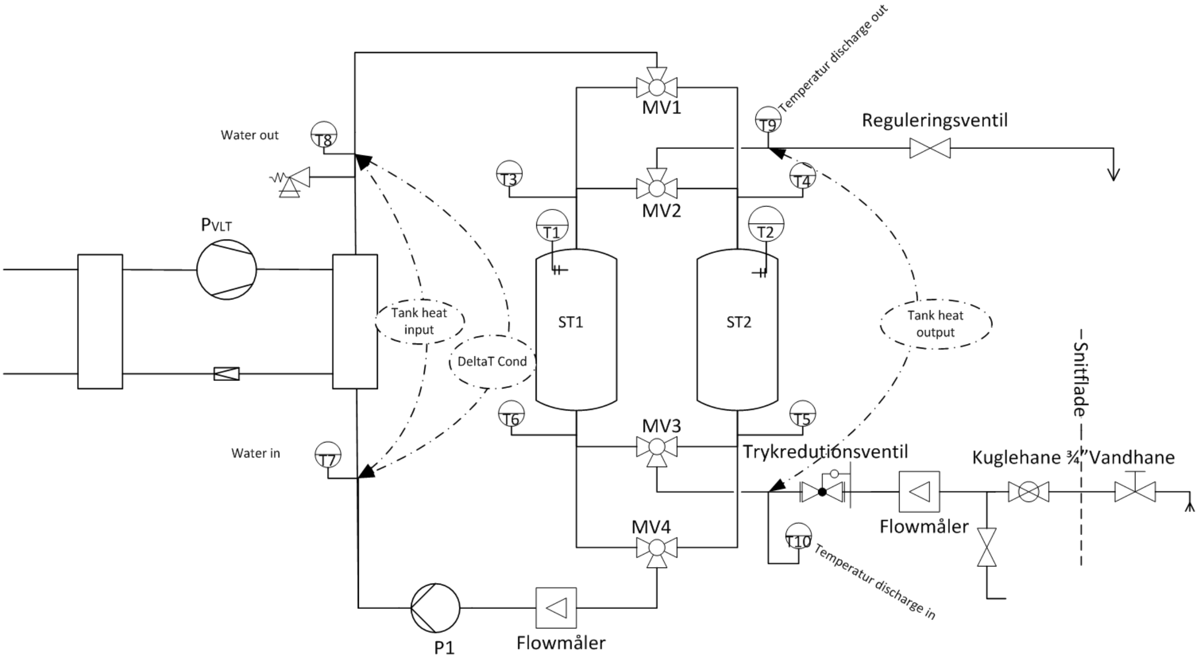

Figure 26 - A principle sketch of the tank test setup with two storage tanks combined with a heat pump and eight valves. One tank is charging, while the other tank is discharging.

Figure 27 - Storage tank test setup with two storage tanks, a heat pump and a control system.

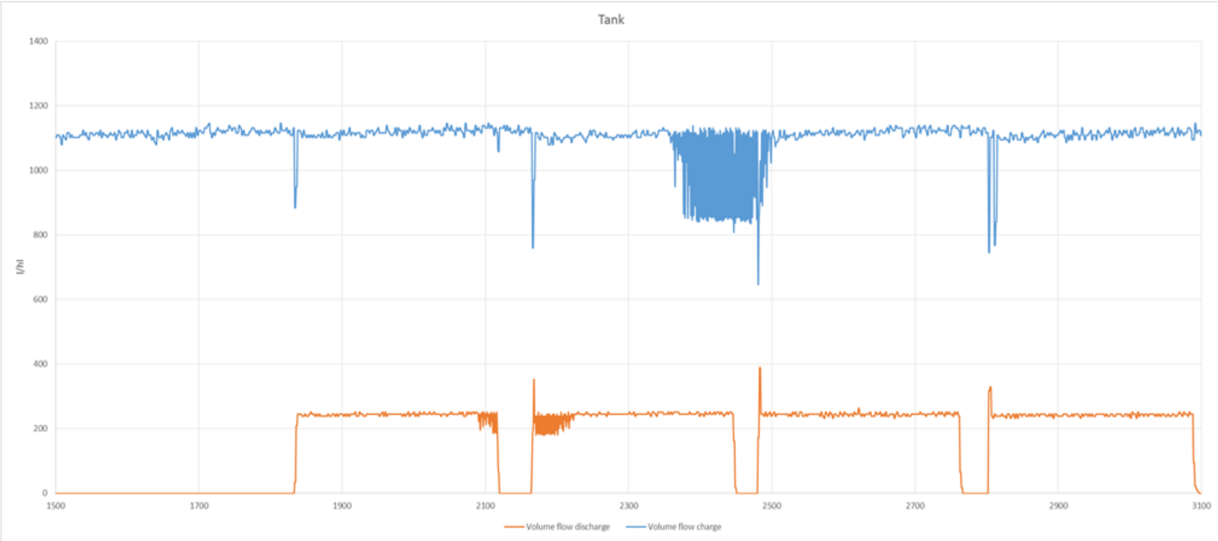

Figure 28 - The volume flow in l/h during charge (blue) and discharge (orange).

The inlet temperature of the cold water to be heated T_{10} (Temperature discharge in) is approximately 12 °C. (Figure 26 and 33).

The inlet and outlet temperatures from the condenser clearly show that there is a stratification in the storage tanks which is reduced gradually during charging (see figure 26 and 29). The maximum outlet temperature T_9 (Temperature discharge out) from the condenser is 66 °C. The condensation temperature corresponds to the outlet temperature from the heat exchanger.

The temperature differences between the inlet and outlet temperatures are shown in figure 30. A significant fluctuation is shown which can be attributed to the thermal capacity in the water and the walls in the connecting pipes. To a minor degree, it is also due to time constants in the temperature sensors.

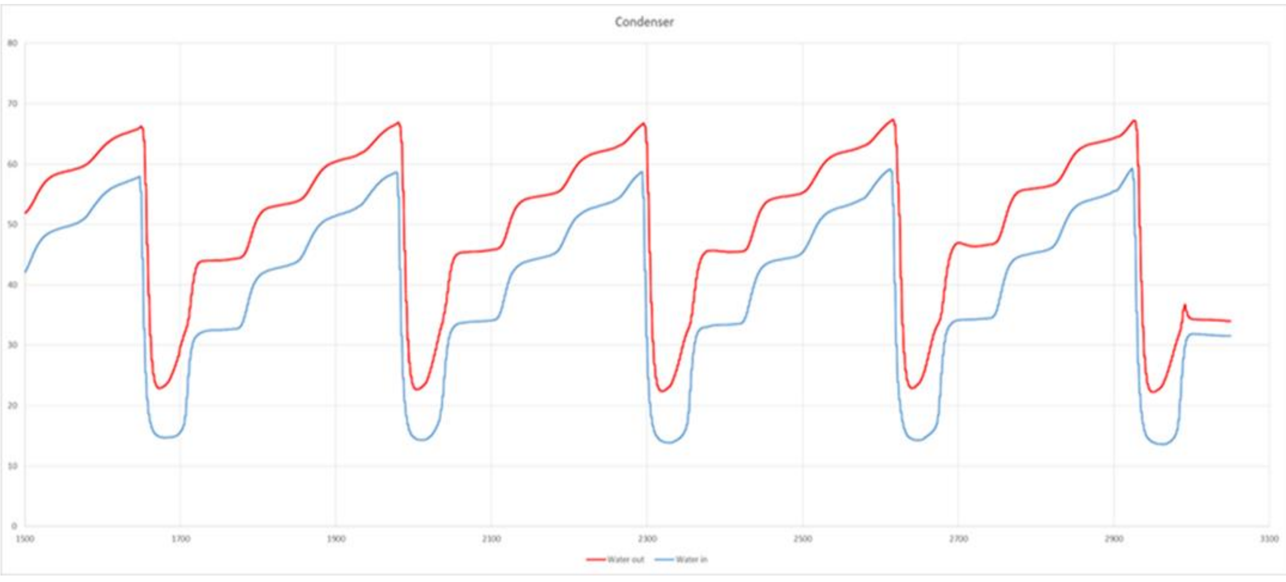

Figure 29 - Measured inlet (blue line) and outlet (red line) temperatures (°C) of the condenser.

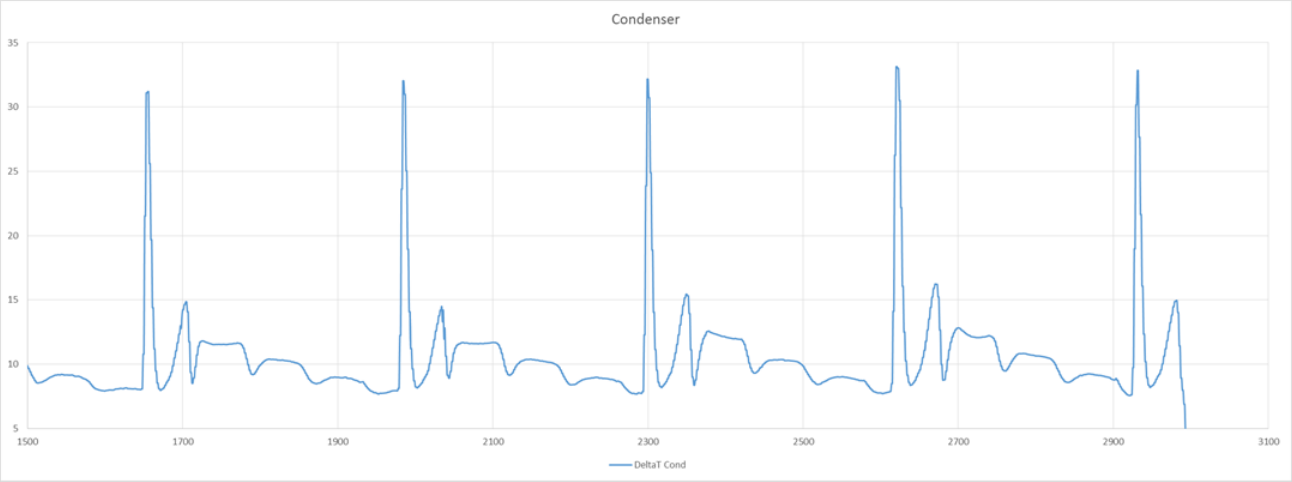

Figure 30 - Measured temperature difference across the condenser (K).

The mass flow through the evaporator is assumed to be constant.

The temperature shown in figure 31 is measured between the valve and the evaporator. The typical value is approximately 11 °C, but the temperature drops to approximately -2 °C just after the shift of storage tanks. The evaporator temperature is approximately 3 K lower than these temperatures due to the pressure loss in the evaporator. During this shift of tanks, the condensing pressure is reduced significant. This change in pressure also gives a reduction of the suction pressure, which is indicated by the temperature drops seen in figure 31. The observed reduced suction pressure is caused by two reasons; the differential pressure over the valve falls when the condensing pressure is reduced, which reduces the capacity of the valve, and flash gas is simultaneously generated in the liquid pipe before the valve, which also reduces the capacity of the valve.

The water temperature differences between the inlet and outlet of the evaporator vary between 1 and 3 K. (see figure 32). The drop in the temperature difference from 3 K to 1 K is observed at the time when the tanks are being shifted. Again, the reason is the reduced capacity of the valve, which leads to a lower refrigerant flow through the valve. Less refrigerant through the evaporator leads to a reduced cooling capacity.

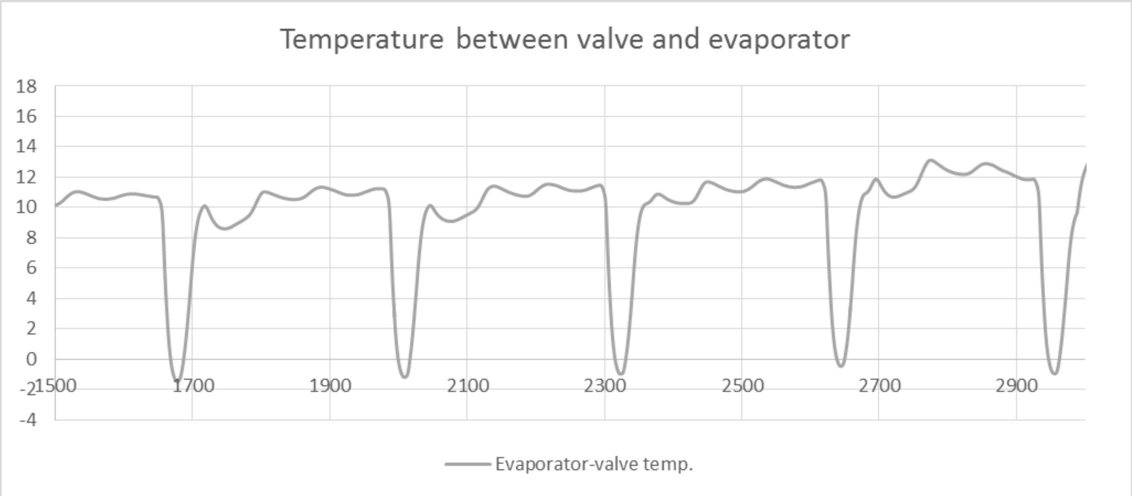

Figure 31 - Measured temperatures in pipe between valve and evaporator (°C).

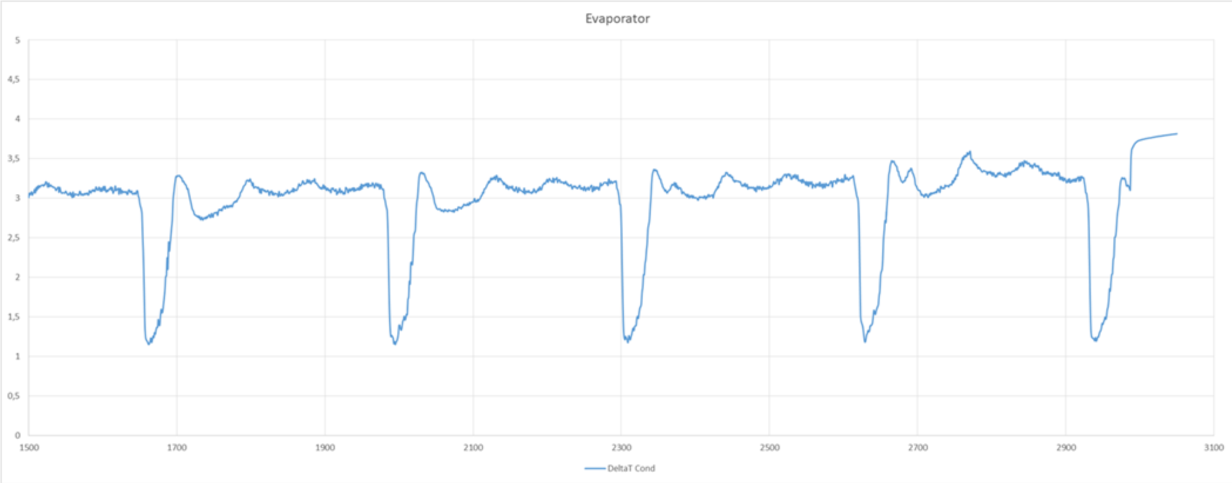

Figure 32 - Measured water temperature difference (K) across the evaporator.

The inlet and outlet temperatures from the charged storage tank are shown in figure 33. For each discharge period, the temperature is gradually decreased from approximately 66 °C to approximately 58 °C. Then, the outlet temperature drops quickly until approximately 50 °C because the separation layer between the warm and cold water in the storage tank has been reached during the discharge. The heat output can also be seen in figure 33.

The heat output decreases slightly during the discharge period corresponding to the variation of the outlet temperature. At the end of the discharge period - for a short time, there is a drop in the heat output.

Figure 33 - The inlet (orange) and outlet (blue) temperatures (°C) during discharge and the output capacity (kW) from the storage tank (red curve).

The delivered heat from the heat pump to the storage is seen in figure 34. The fluctuation corresponds to the variation seen in the temperature difference between the inlet and outlet temperatures over the condenser (see figure 30). The electric power used by the heat pump increases slightly during the charging period. The measured COP of the heat pump during charging is shown in figure 35. These values fluctuate similar to the variations of the temperature variation over the condenser and the delivered heat to the storage tank.

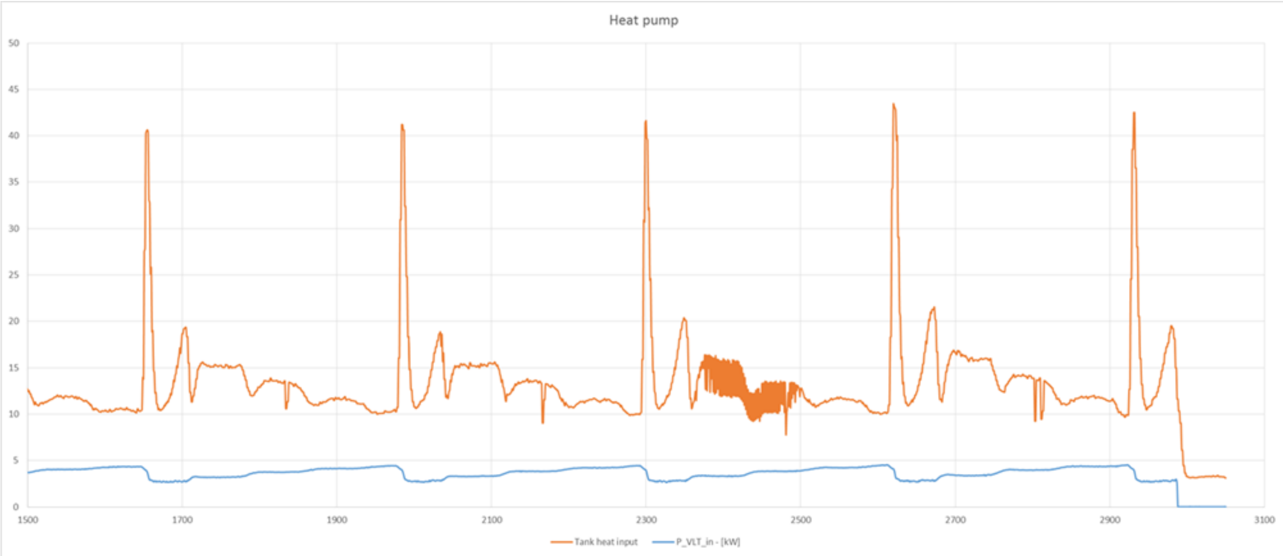

Figure 34 - The delivered heat (orange) from the heat pump to the tank and the used electric power (kW) (blue).

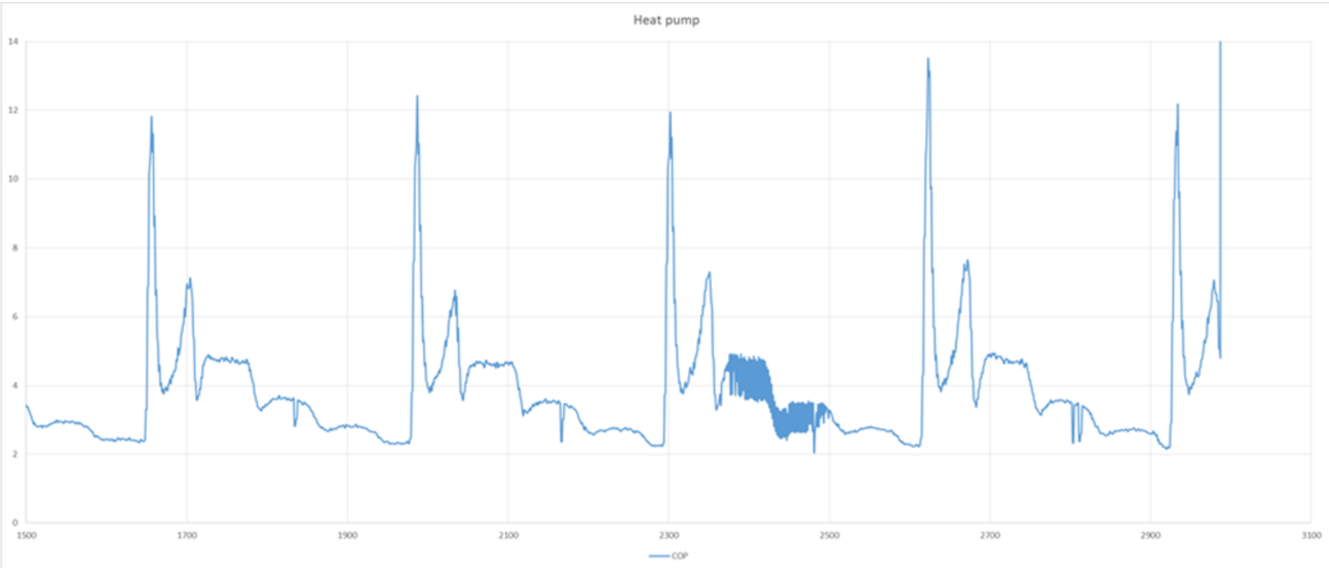

Figure 35 - Measured COP of the heat pump.

The measurements provide the following results:

The average COP using the ISEC concept with this test setup is estimated at 3.8.

The test conditions for the discharged storage tank are a maximum and a minimum temperature of 66 °C and 58 °C, respectively (see figure 33).

This result can be compared with the measured temperatures where the water is heated directly in one step. In this case, the COP is measured to 2.5.

When comparing these two cases, the increase in COP by using the ISEC concept is estimated at 52 %.

5. PART III, Case for district heating

The ISEC concept has been considered in connection with an application of a heat pump used in a district heating system where the waste heat from an industrial process is partly utilized for the heating of four villages situated between the industry and a city with a district heating system. The industry is assumed to be in operation and to deliver the necessary heat for the villages together with heat to the district heating system for about half of a year. In the other half of the year, a heat pump will supply the necessary heat for the four villages. Different solutions are investigated where heat from the return pipe of the district heating system is utilized as the heat source for the heat pump.

The first option investigated is a solution where a heat pump with a capacity of 1200 kW is located close to the city and the district heating system. Figure 36, left and right, shows this solution with and without the heat pump in operation.

In the second option, it is assumed that four heat pumps are installed, each with a capacity of 300 kW. Figure 37, left and right, shows this option at the two operating conditions.

The heat pump is assumed to deliver a supply temperature of 70 °C to the district heating system. The heat source used for the evaporator in the heat pump is the return pipe from the other part of the district heating system. The temperature of this water is assumed to be 40 °C. The water flow is adjusted to cool the water by 5 K, and the evaporator temperature is assumed to be 30 °C. The charge time when using the ISEC concept is assumed to be 70 minutes, which results in a storage size of 40 m^3.

The results are shown in table 2. The water from the storage tank is assumed to be circulated eight times during the charge period, which will give a COP of 8.1for the heat pump. By reducing the number of cycles from eight to four, the COP will decrease to 7.9. With an even further reduction to two cycles, the COP will be 7.3. If the evaporator temperature can be increased to 33 °C and the number of cycles is eight, the COP will increase to 9. In the option where four smaller heat pumps of 300 kW are applied, the storage size is assumed to be 5 m^3 and the number of charging cycles is set at four. In this case, the COP is calculated to be 9.0.

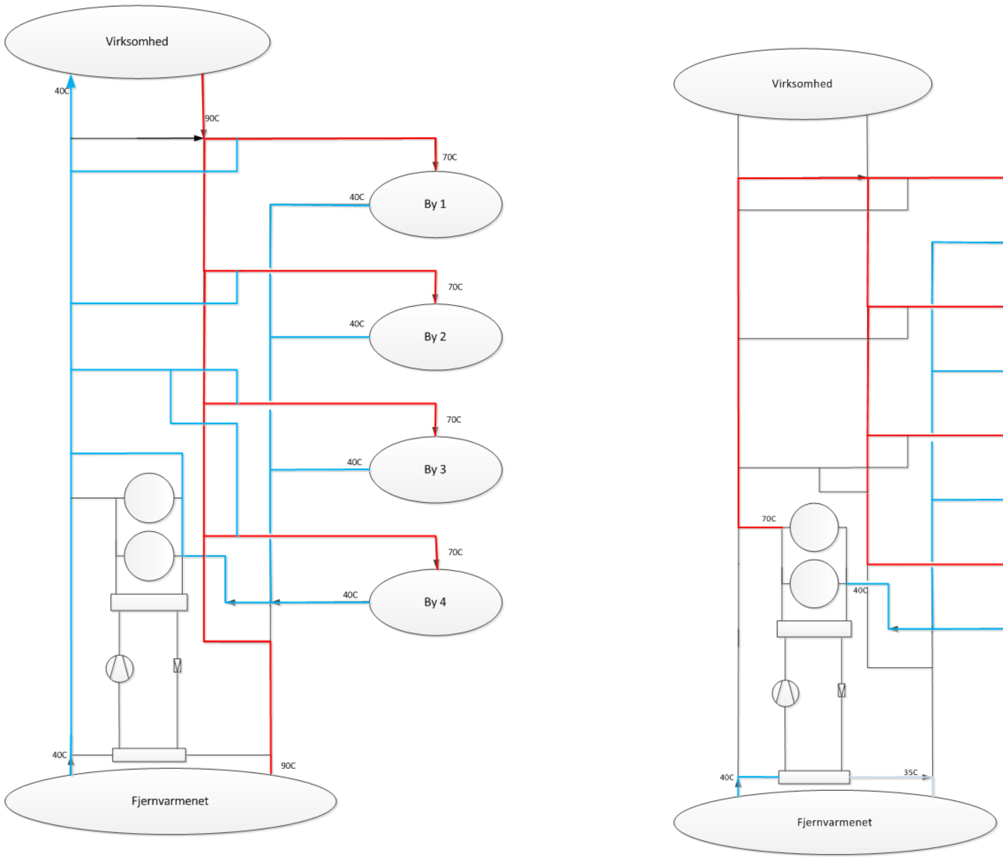

Figure 36, Left: Option 1 - Heat is supplied from the industry to the four villages and the city. The heat pump is not in operation. Right: Option 1 with one heat pump - Heat for the ISEC heat pump is supplied from the return pipe line from the district heating system.

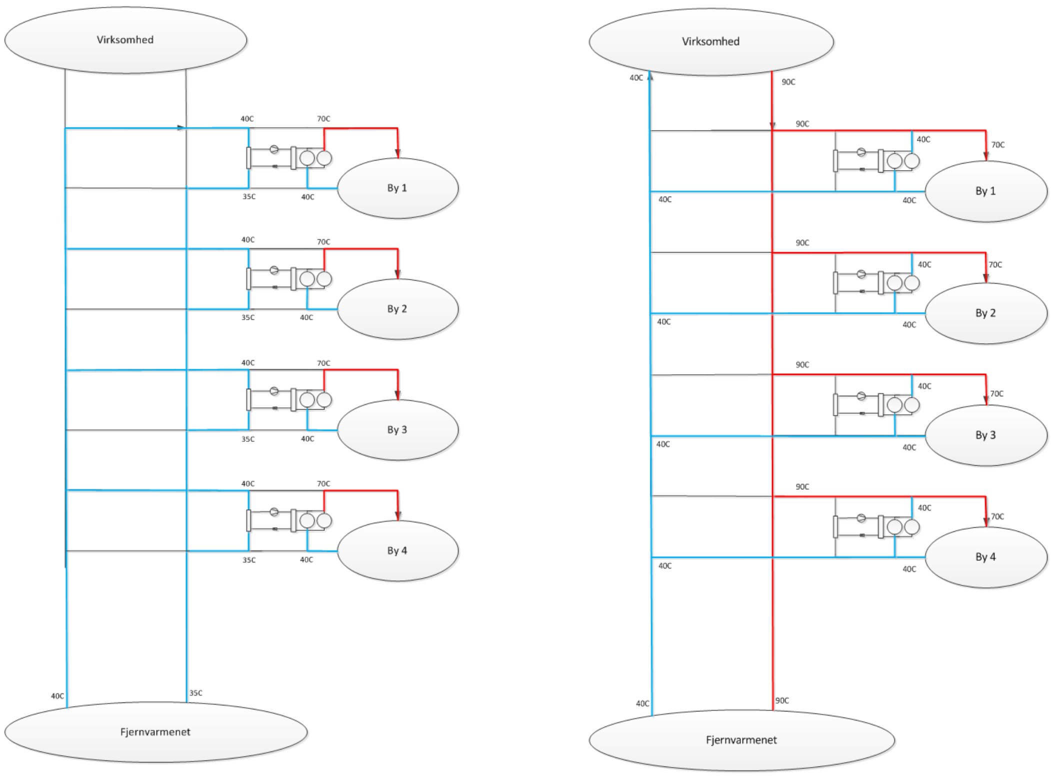

Figure 37, Left: Option 2 with four heat pumps. Heat for the ISEC heat pump is supplied from the return pipe line from the district heating system. Right: Option 2 - Heat is supplied from the industry to the four villages and the city. The heat pump is not in operation.

No	Load [kW]	Vtank [m3]	N cyc	t_charge [s]	t_cycle[s]	COP tot hot	Tsupply	Condenser side HP Treturn	Tevap [C]	Evapotator side HP Tsupply	Treturn
1	1200	40	7	4180	585	8,4	67	37	28	35	30
2	1200	40	7	4598	644	7,9	70	37	28	35	30
3	1200	40	7	4598	630	6,7	70	37	23	30	25
4	300	5	4	2299	533	6,4	70	37	23	30	25
5	300	5	4	2299	470	7,3	70	37	28	35	30

Table 2 - Results and assumptions from the calculation of COP. Load: Capacity of heat pump in kW. Vtank: Volume of storage tank in m³. N cyc: Number of cycles during charging. T_charge: Total time elapsed during the charging in s. t_cycle: Time elapsed during one cycle in s. COP tot hot: Calculated COP for heat pump. Tsupply: Supply temperature for district heating in °C. Condenser side HP: Treturn: Return temperature from district heating in °C. Tevap: Evaporation temperature in °C. Evaporator side HP: Tsupply: Supply temperature from waste heat source in °C. Treturn: Return temperature to waste heat source in °C.

The different cases are:

1. A large heat pump placed close to the district heating network. The warm side of the heat pump has supply/return temperatures of 67 °C/37 °C, respectively. The cold side of the heat pump has supply/return temperatures of 35 °C/30 °C, respectively.
2. A large heat pump placed close to the district heating network. The warm side of the heat pump has supply/return temperatures of 70 °C/37 °C, respectively. The cold side of the heat pump has supply/return temperatures of 35 °C/30 °C, respectively.
3. A large heat pump placed close to the district heating network. The warm side of the heat pump has supply/return temperatures of 70 °C/37 °C, respectively. The cold side of the heat pump has supply/return temperatures of 30 °C/25 °C, respectively.
4. A small heat pump placed local close to the buildings. The warm side of the heat pump has supply/return temperatures of 70 °C/37 °C, respectively. The cold side of the heat pump has supply/return temperatures of 30°C/25 °C, respectively.
5. A small heat pump placed local close to the buildings. The warm side of the heat pump has supply/return temperatures of 70 °C/37 °C, respectively. The cold side of the heat pump has supply/return temperatures of 35 °C/30 °C, respectively.

6. Bibliography

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