DTU Library

High efficient ammonia heat pump system for industrial process water using the ISEC concept. Part 2

Olesen, Martin F.; Madsen, Claus; Olsen, Lars; Paaske, Bjarke ; Rothuizen, Erasmus Damgaard

Published in:
Proceedings of 11th IIR Gustav Lorentzen Conference on Natural Refrigerants

Publication date:
2014

HIGH EFFICIENT AMMONIA HEAT PUMP SYSTEM FOR INDUSTRIAL PROCESS WATER USING THE ISEC CONCEPT - PART II

Martin F. Olesen[a], Claus Madsen[b], Lars Olsen[c], Bjarke Paaske[d] and Erasmus Rothuizen[e]

[a]*Danish Technological Institute, Taastrup, Denmark, mfo@teknologisk.dk,*
[b]*Danish Technological Institute, Taastrup, Denmark, clma@teknologisk.dk,*
[c]*Danish Technological Institute, Taastrup, Denmark, lo@teknologisk.dk,*
[d]*Danish Technological Institute, Taastrup, Denmark, bjpa@teknologisk.dk*
[e]*Technical University of Denmark, Copenhagen, Denmark, edro@mek.dtu.dk*

Abstract
The Isolated System Energy Charging (ISEC) concept allows for a high efficiency of a heat pump system for hot water production. The ISEC concept consists of two water storage tanks, one charged and one discharged. The charged tank is used for the industrial process, while the discharged tank is charging. The charging of the tank is done by recirculating water through the condenser and thereby gradually heating the water. The modelling of the system is described in Part I [1]. In this part, Part II, an experimental test setup of the tank system is reported, the results are presented and further modelling of the heat pump and tank system is performed (in continuation of Part I). The modelling is extended to include the system performance with different natural refrigerants and the influence of different types of compressors.

Keywords: Hydrocarbons, compressor, heat pump, ISEC, modelling, experimental

1. INTRODUCTION

This paper is a part II and more or less a continuation of [1]. The paper presents the preliminary results of an ongoing project regarding a method for optimizing the efficiency of heat pumps by utilizing an external tank system. The system is referred to as the ISEC system or ISEC concept. ISEC is an abbreviation of “Isolated System Energy Charging”.

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

This paper continues from part I by focusing on modelling the system with different natural refrigerants and different types of compressors. Furthermore, an experimental test setup of the tank system is reported and the preliminary results are presented.

2. CONCEPT

The principle of a conventional heat pump system used to heat water for industrial processes is seen in Figure 1. The heat pump heats the process water (could also be other fluids) from the return temperature, e.g. $T_{p,return} = 20$ °C, directly to the temperature used for the process, e.g. $T_{p,supply} = 60$ °C. For this reason, the conventional system is referred to as a direct heat pump.

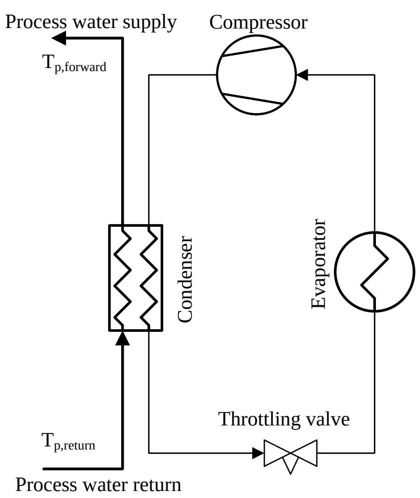

Figure 1: A conventional heat pump setup, where the process water is heated directly in the condenser. Referred to as a direct heat pump.

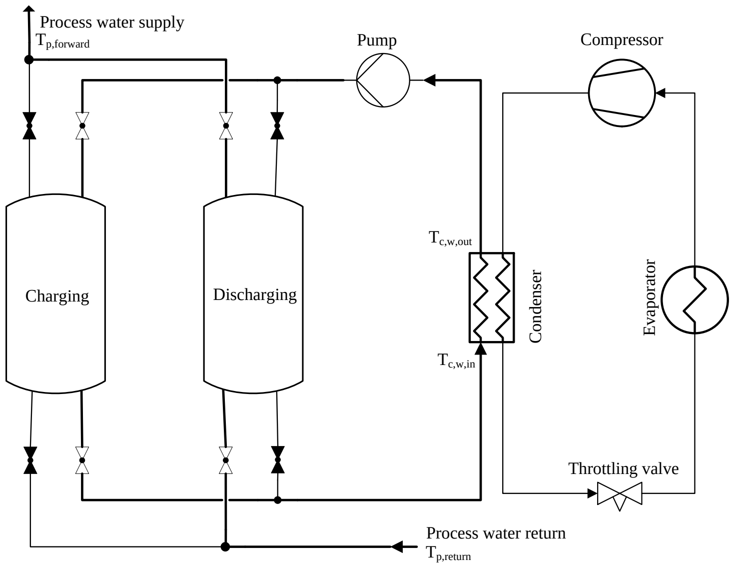

Figure 2: A principle sketch of the ISEC system. One tank is charging, while the other tank is discharging.

In principle, the heat pump system based on the ISEC concept produces a continuous production of hot water by means of a tank system instead of e.g. several heat pumps in series. 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.

The ISEC system is seen in Figure 2; 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).

Initially, both tanks are cold (discharged). When starting to charge, the water recirculates a number of times in a closed loop through the condenser and back to the tank. This 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 (discharging). It is essential that the tapped hot water and the cold return water filling the tank is separated by good stratification in order to supply a constant temperature ($T_{p,supply}$) to the industrial process. Stratification during charging is not required in order for the system to be beneficial. However, the efficiency will increase, if this is achieved. In part I [1], the charging tank was modelled as fully mixed, while in this paper the charging tank is modelled as fully stratified.

The principle described above is most suitable, if there is a relatively large temperature difference between the cold return water ($T_{p,return}$) and the hot forward water ($T_{p,supply}$). Traditionally, as seen in Figure 1, a direct heating from $T_{p,return}$ to $T_{p,supply}$ is done within the condenser unit of the heat pump. This results in a rather high condensing temperature (T_c), which traditionally is equal or above the water outlet temperature ($T_{p,supply}$).

The ISEC concept results in an improvement of the heat pump efficiency, as it only needs to raise the temperature 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. Initially, the heat pump has a high COP, which decreases as the temperature increase.

3. SIMULATION

3.1 MODEL

The modelling is performed with the EES (Engineering Equation Solver) software, which is characterized by the fact that it is capable of numerically solving thousands of coupled non-linear algebraic equations. Moreover, the software has access to a large build-in database of fluid/solid property values, e.g. enthalpy, pressure, conductivity etc.

Even though this paper is a continuation of part I [1], the modelling in this part is performed in another software and some of the assumptions differs.

3.1.1 Tank model

The tank is modelled as an ideal tank with full stratification and no losses. This means that the tank essentially works like a buffer or time delay. The water that enters the tank at the top and exists at the outlet at the bottom, when the whole tank is tapped, will maintain its state throughout the process. Equation (1) describes this process mathematically.

$$\begin{gathered} T_{out}(t_1) = T_{in}(t_0) \\ \int_{t_0}^{t_1} \dot{m}(t)\, dt = m_{tank} \end{gathered} \tag{1}$$

Where $\dot{m}(t)$ is the mass flow in and out of the tank, and m_{tank} is the total mass of the water in the tank. For simplicity's sake, the water density is assumed constant.

It should be emphasized that this ideal tank model is applied to both charging and discharging of the tank.

3.1.2 Heat pump

The heat pump is modelled as a simple heat pump in that it only contains four components: compressor, condenser, throttling valve and evaporator.

In general, pressure loss and heat loss are neglected in both the piping and the heat exchangers.

Compressor

Multiple compressor models are used depending on the refrigerant. However, all models are based on two compressor polynomials. The cooling capacity, $\dot{Q}_0$, is calculated by equation (2) and the compressor power, $\dot{W}$, is calculated in terms of equation (3).

Equations:

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

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

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

The isentropic efficiency, η_{is}, of the compressors is derived from the cooling capacity and the compressor power, and it is used for the cycle calculations.

Evaporator
The evaporator model is based on equation (4).

$$\dot{Q}_0 = UA \times \text{LMTD} = \text{UA} \times \frac{T_{0,w,in} - T_{0,w,out}}{ln\left(\frac{T_{0,w,in} - T_0}{T_{0,w,out} - T_0}\right)} \quad (4)$$

Where $T_{0,w,in}$ is the inlet water/brine temperature and $T_{0,w,out}$ is the outlet temperature. The UA-value is assumed constant and chosen to fit $\Delta T = T_{0,w,in} - T_{0,w,out} = 5$ K at the low charging temperature (in the start of the charging cycle) as well as a pinch temperature of 2 K. The UA-value is constant.

Condenser
The condenser model is based on equation (5).

$$\dot{Q}_c = UA \times \text{LMTD} = \text{UA} \times \frac{T_{c,w,out} - T_{c,w,in}}{ln\left(\frac{T_c - T_{c,w,in}}{T_c - T_{c,w,out}}\right)} \quad (5)$$

Where $T_{c,w,in}$ is the inlet water/brine temperature and $T_{c,w,out}$ is the outlet temperature. The UA-values is again assumed constant and chosen to fit $\Delta T = T_{c,w,out} - T_{c,w,in} = 5$ K at the low charging temperature (in the start of the charging cycle) as well as a pinch temperature of 2 K. Assuming condensation at the whole condenser surface.

The mass flow on the water/brine side is constant during a simulation/charging, which results in a slightly decreasing ΔT through the charging cycle. Since the charging is done with a stratified tank, the mass flow on the water side must be solved so that the recirculation number N is an integer.

Where N is defined as:

$$N = \frac{\dot{m}_{c,w} \times t_{charge}}{m_{tank}} \quad (6)$$

Where $\dot{m}_{c,w}$ is the water mass flow, m_{tank} is the mass of water in the tank and t_{charge} is the total time, i.e. the amount of time which it takes to charge the tank.

Throttling valve
The throttling valve is assumed reversible and adiabatic.

General

The mean COP of the system is calculated in terms of:

$$Mean\ COP = \frac{\int \dot{Q}_c\, dt}{\int \dot{W}_{comp}\, dt} \tag{7}$$

The power consumption of the circulation pump is not included.

3.2 RESULTS

The results of the simulations covering the three refrigerants; Propane (R290), Propylene (R1270) and Isobutene (R600a) are presented. Each simulation is conducted at $N = 3$ and $N = 9$, the run time is varied in order to achieve a tank/process forward temperature of $T_{p,supply} = 60°C$. The return temperature is $T_{p,return} = 20°C$. Furthermore, on the cold side the temperature at the inlet is kept constant at $T_{0,w,in} = 10°C$.

The results for propane, propylene and isobutene are shown in Figure 3, Figure 4 and Figure 5, respectively. The curves represent the COP and mean COP development through the charge cycle. The x-axis depicts the charge level defined as:

$$Charge\ level = \frac{\int \dot{Q}_c\, dt}{\int_{t_{start}}^{t_{end}} \dot{Q}_c\, dt}$$

Which is the normalized energy content of the tank and an indicator of the progress of the charging process.

Common for all three refrigerants and the whole purpose of the ISEC concept is that the COP is high at the start of the charging cycle. As the charging proceeds, the COP decreases due to the increasing tank temperature. The staircase shape of the curves is due to the assumption of a fully stratified tank during charging. Each step of the curve indicates a full recirculation of the tank volume.

The dotted lines in the figures depict the changes in the mean COP during the charge cycle. The values at $charge\ level = 1$ is the mean COP of the heat pump with a given refrigerant. These values are also seen in Figure 6. For all three refrigerants, the mean COP is highest at $N = 9$ and slightly lower at $N = 3$. However, both COP values are considerably higher than the COP achieved with direct heating.

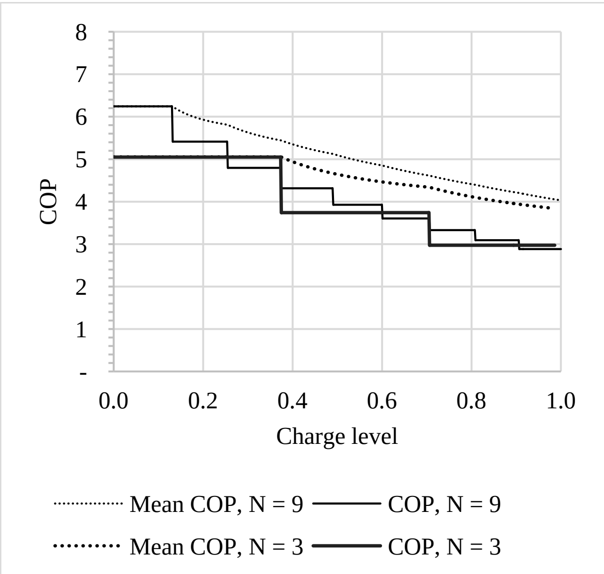

Figure 3. Propane R290

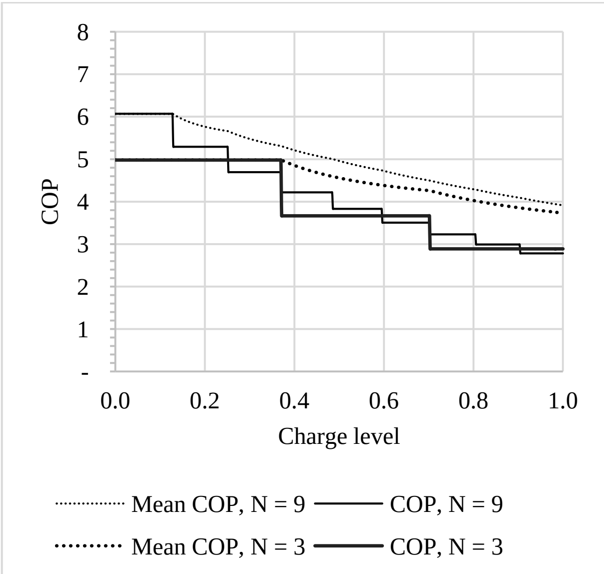

Figure 4. Propylene R1270

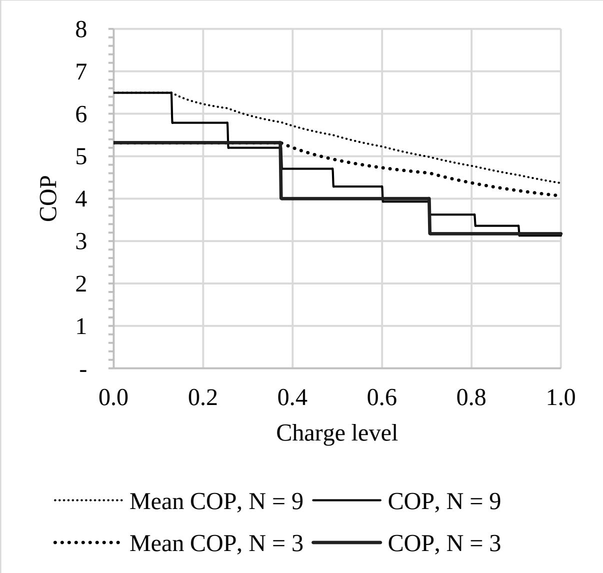

Figure 5. Isobutene R600a

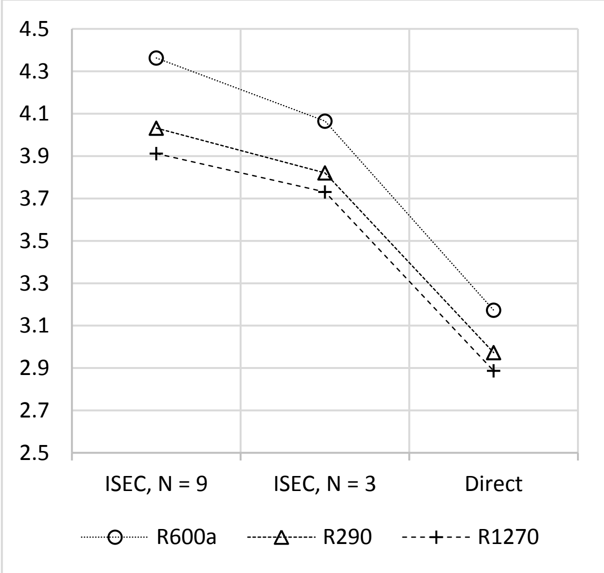

Figure 6. Mean COP for the different simulations and refrigerants

The mean COP values are compared with their relative increase in Figure 7. Where an efficiency increase of 35-37 % with nine recirculation cycles compared to direct heating is observed. Isobutene peaks with a 37 % increase. The increase is approximately 28 % at a recirculation number of three. Thus, the gain in going from three to nine circulations is only 5-7 %. It should be noticed that this comparison is made without subcooling. The gain will be smaller when including subcooling in the comparison. This was also shown for ammonia as refrigerant in [1]. Furthermore, the power consumption of the recirculation pump was not included, which would reduce the overall gain and the difference between nine and three circulations as well in that the power consumption would increase. This is expected to have a minor effect. However, there will undoubtedly be an optimal recirculation number to a given system.

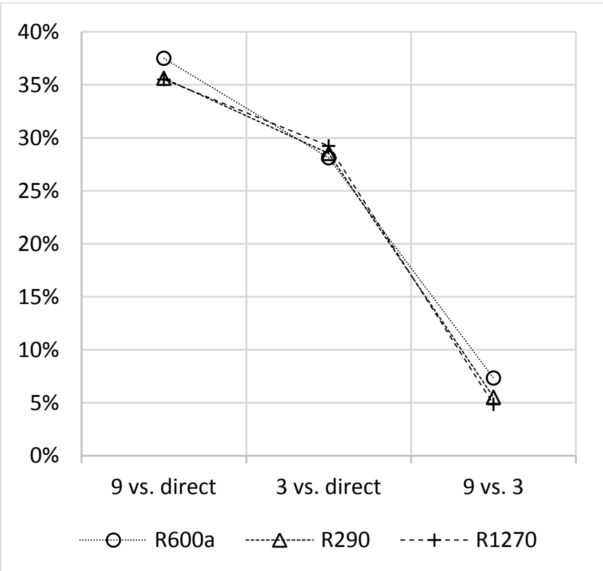

Figure 7: Relative difference in COP between direct heating and ISEC with 3 or 9 cycles for the different refrigerants

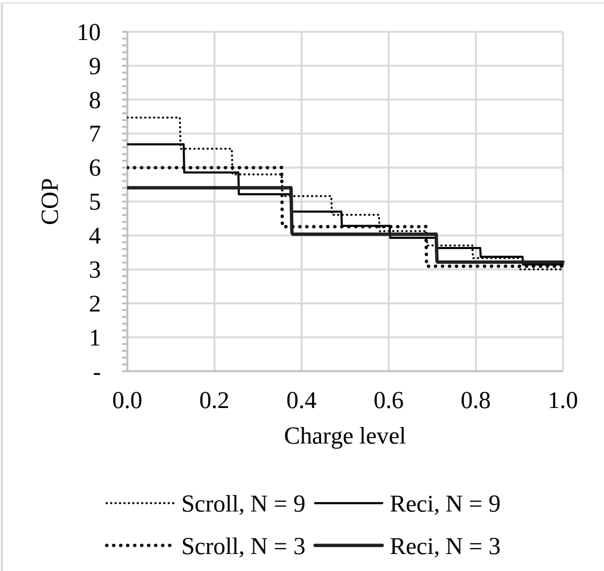

Figure 8: COP of the heat pump for a cycle number of 9 and 3 with both a reciprocating and a scroll compressor (with R134a as refrigerant)

In Figure 8, the difference between a reciprocation compressor and a scroll compressor is illustrated in terms of COP. The refrigerant is R134a, and it is seen that the heat pump with a scroll compressor yields higher COP values at the start of the charging with low condensing temperature than the heat pump with a reciprocating compressor. However, the values even out as the charging progresses and the temperature increases. This is most likely due to the isentropic efficiency of the scroll compressor peaking at a lower pressure ratio than that of the reciprocating compressor. The heat pump with a scroll compressor ends at a mean COP which is 3 % higher than the one of the reciprocating compressor at N = 9, and 2 % higher at N = 3. With direct heating, the COP for the reciprocating compressor is 4 % higher than the one for the scroll compressor. Thus, a scroll compressor seems to be preferred over a reciprocating compressor with the ISEC system, even though the gain is not overwhelming. Further studies should be conducted to see if this is a consistent tendency.

4. EXPERIMENT

Experimentation with the system is ongoing. At the outset, the tank system is investigated without connection to a heat pump and preliminary results are presented.

4.1 SETUP

The heating of the system is carried out with an electric heating coil as a stand-in for the heat pump in that the purpose of the test is to investigate the stratification of the tank during charging and discharging.

A principle diagram of the setup is seen in Figure 9 and Figure 10 for charging and discharging, respectively. The temperature is monitored at the tank inlet, outlet and at various positions over the tank height. The volume flow is also measured.

The tank has a height-to-diameter ratio of 3.8. The volume flow during charging results in a vertical velocity in the tank of $3.3 \times 10^{-3}\ {}^{m}/_{s}$, and $0.7 \times 10^{-3}\ {}^{m}/_{s}$ during discharging.

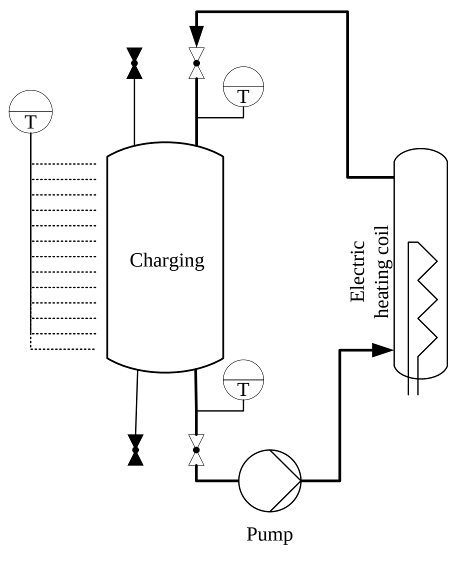

Figure 9: Experimental setup – Charging the tank

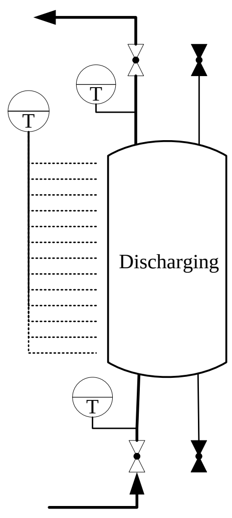

Figure 10: Experimental setup – Discharging the tank

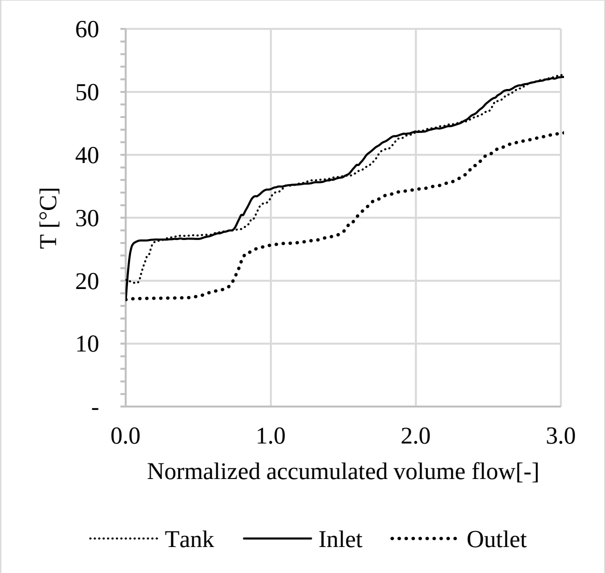

Figure 11: Temperature progress during charging, measured at inlet, outlet and inside the tank at a height of ¾ of the total tank height.

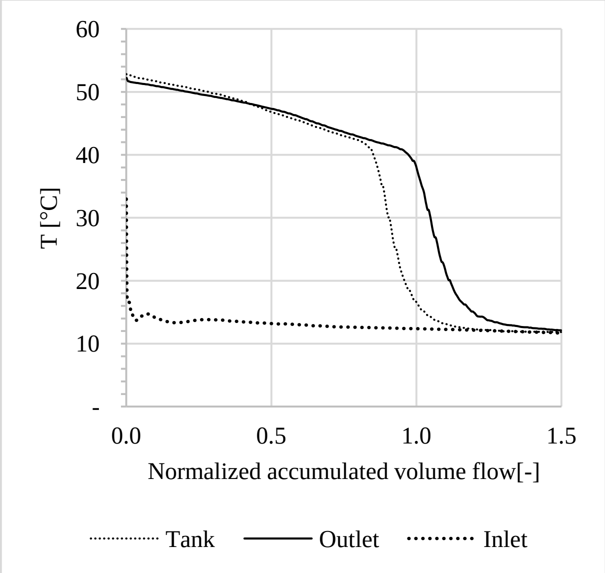

Figure 12: Temperature progress during discharging, measured at inlet, outlet and inside the tank at a height of ¾ of the total tank height.

4.2 RESULTS

The result of a single charge is seen in Figure 11, which illustrates the temperature development at inlet, outlet and inside the tank during charging. The x-axis is normalized accumulated volume flow, and it is seen that water in the tank is recirculated three times (N = 3) during the charging. The stratification is noticeable, but not ideal and it smoothens as the charging progresses.

At discharging, which is seen in a similar plot in Figure 12, a rather large temperature gradient of 10 K is observed over the tank. Good stratification is achieved, even though the transition zone is rather large. It stretches over a ¼ tank volume. Nearly a full tank volume is tapped before the transition zone reaches the outlet, which indicates that the mixing occurs at the first entrance and the stratification is maintained during discharging.

The rather large transition zone, which is left in the tank from discharging, could be problematic at the second charging in that a sudden temperature increase would occur. Further studies of the ratio between volume flow and tank dimensions will be carried out to reduce the size of the transition zone at discharging and the smoothing effect at charging. Both effects will reduce the gain of the ISEC system in terms of the simulated COP values.

5. CONCLUSION

An increase in the mean COP between 35-37 % compared to direct heating was found in the simulation of the ISEC system with an ideal stratified tank during charging and discharging. Of the three refrigerants; R600a, R1270 and R290, isobutene (R600a) yields the highest increase of 37 % at nine circulations of tank water during charging. The difference between three and nine circulations were only 5-7 %, which could be lost by increased power consumption by the recirculation pump since this was not included in the simulation. Subcooling was not included either, which would further decrease the gain.

The comparison between a heat pump with a scroll compressor and a reciprocating compressor showed a mean COP which was 3 % higher for the scroll compressor. Further studies should be conducted to see, if this is a consistent tendency.

Tests conducted on the tank system with an electric heating coil instead of a heat pump showed that stratification do occur during charging and discharging. During charging, the stratification smoothens and the transition zone increases, as the water in the tank recirculates. During the discharging, a rather large transition zone of a ¼ of the tank volume was seen. This reduces the gain of the ISEC system in terms of the simulated COP values, and further studies should be conducted.

ACKNOWLEDGEMENTS

The authors would like to thank the Danish Energy Agency for financially funding the project through the “Energy Technological Development and Demonstration” fund (EUDP). A thank you is also given to the industrial cooperating partners of the project: Svedan, Innotek, Alfa Laval, Arla Foods, Bjerringbro Varmeværk and Metro Therm.

NOMENCLATURE

T_c	Condensing temperature (℃)
T_0	Evaporating temperature (℃)
m_{tank}	Mass of water in tank (kg)
$\dot{Q}_0$	Cooling capacity (kW)
$\dot{Q}_c$	Heating capacity (kW)
$\dot{W}_{comp}$	Compressor power (kW)
$T_{p,supply}$	Supply process temperature (℃)
$T_{p,return}$	Return process temperature (℃)
$T_{0,w,in}$	Water inlet temperature at evaporator (℃)
$T_{0,w,out}$	Water outlet temperature at evaporator (℃)
$T_{c,w,in}$	Water inlet temperature at condenser (℃)
$T_{0,w,out}$	Water inlet temperature at condenser (℃)
η_{is}	Isentropic efficiency (-)
$C_{0..19}$	Compressor constants
N	Recirculation number (-)

REFERENCER

1. Rothuizen, E., Madsen, C., Elmegaard, B., Brix, W., Olsen, L., & Olesen, M. F. (2014). High efficient ammonia heat pump system for industrial process water using the isec concept - part I. *11th IIR-Gustav Lorentzen Conference*.