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HIGH EFFICIENT HEAT PUMP SYSTEM USING STORAGE TANKS TO INCREASE COP BY MEANS OF THE ISEC CONCEPT – PART I: MODEL VALIDATION

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ABSTRACT

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 result 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. This paper shows that there is a good consistency between the model and the experimental tests.

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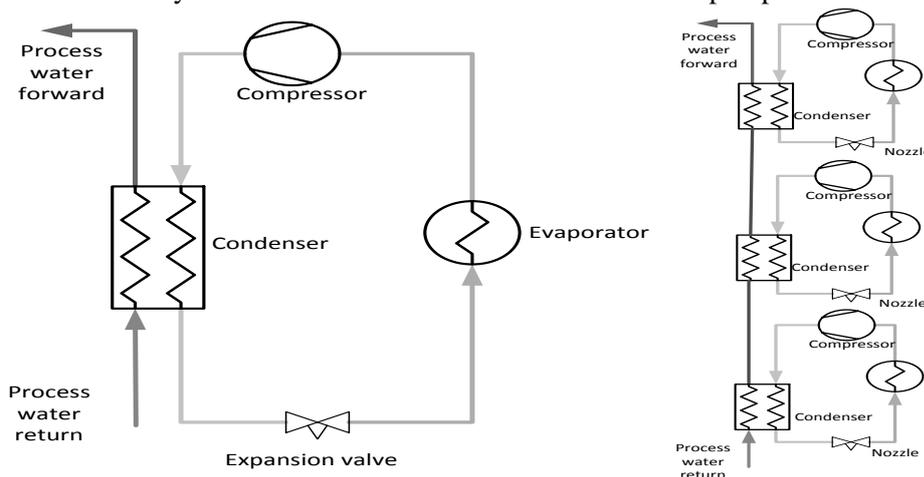
Figure 1 – Left: Conventional heat pump system for water heating for industrial processes. Right: Three heat pumps in series.

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 each condenser is equal. 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.
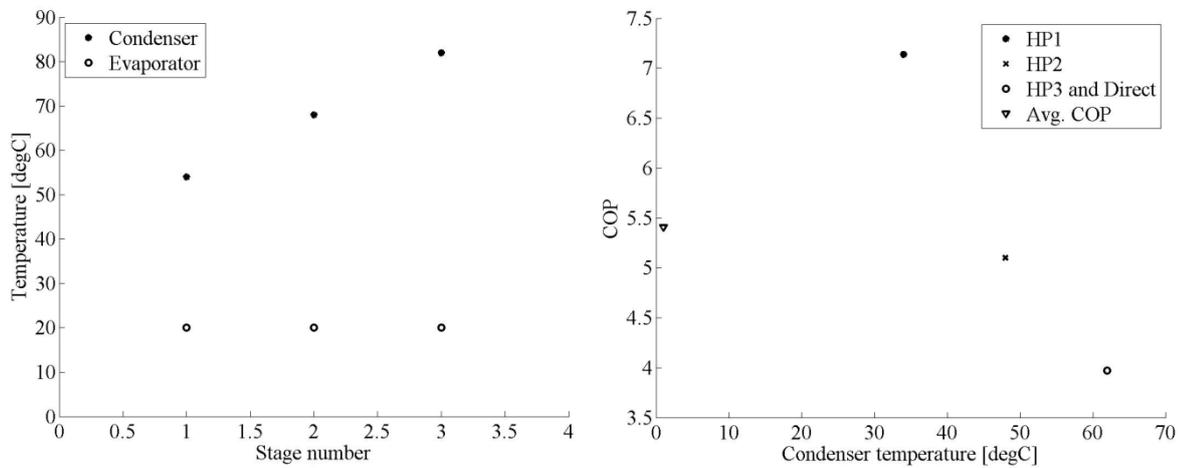

Figure 2 – Comparison of water heating from 40°C to 80°C using one direct heat pump and three heat pumps in series. Left: Evaporating and condensing temperatures for the three heat pumps in series. Right: COP for each of the three heat pumps in series compared to the direct heating heat pump and the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

The condensing temperature must be higher than the outlet temperature of the water assuming the desuperheater can't be used to heat the water above 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 are 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 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 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 concepts have previously been described Rothuizen and Olesen (Rothuizen, et al., 2014) (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 °C 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 a relatively few degrees. 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.
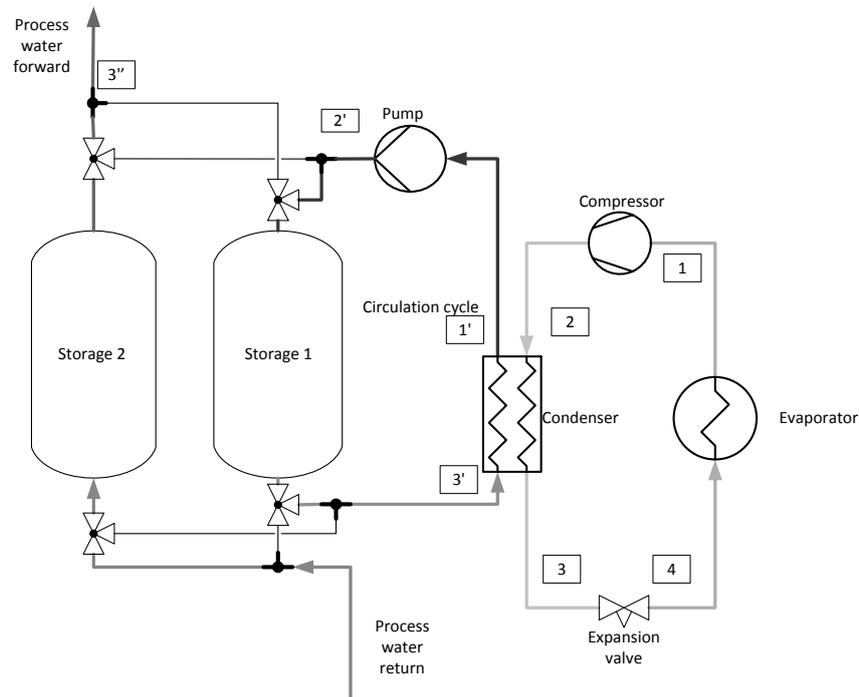

Figure 3 - A sketch of the ISEC system. Water is charged by circulating from tank one through the condenser while the water tapped from tank two is used to cover demand. Water is circulated multiple times through the condenser and tank one. When tank two is fully discharged, the system switches so tank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

2. 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 made. Dymola enables models to be reused and adds a graphical interface to the Modelica language. 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 reduction valve and the pumps were based on simple models, which will be described in the following. Fluid properties were found using the fluid property library CoolProp with an interface to Dymola. 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 forces the temperature in the condenser to increase.

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 transferring area as input. 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. 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.

$$\dot{Q}_0 = C_0 + C_1 T_0 + C_2 T_c + C_3 T_0^2 + C_4 T_0 T_c + C_5 T_c^2 + C_6 T_0^3 + C_7 T_c T_0^2 + C_8 T_0 T_c^2 + C_9 T_c^3 \tag{1}$$

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

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

Tank

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

$$M_i * Cp * \frac{dT_i}{dt} = \frac{(k+\Delta k)A_{C,i}}{\Delta x_{i+1 \to i}}(T_{i-1} + T_i) + \frac{(k+\Delta k)A_{C,i}}{\Delta x_{i-1 \to i}}(T_{i-1} - T_i) + U_i A_{s,i}(T_{env} - T_i) + \dot{m}_{down} C_p (T_{i-1}) - \dot{m}_{up} C_p (T_i) - \dot{m}_{down} C_p (T_i) + \dot{m}_{up} C_p (T_{i+1}) + \dot{m}_{in} C_p T_{in} - \dot{m}_{out} C_p T_i \tag{3}$$

The change in mass and temperature in each node, i, is calculated by taking into account: the conduction between adjacent layers, given by $\frac{(k+\Delta k)A_{C,i}}{\Delta x_{i+1 \to i}}(T_{i\pm1} \pm T_i)$, the heat transfer through the wall to the surroundings, given by $U_i A_{s,i}(T_{env} - T_i)$, 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 $\dot{m}_{in,out} C_p T_{in,out}$. Units: $T\ [K]$, $M\ [kg/s]$, $C_p [J/kgK]$, $k [\ W/mK]$, $A\ [m^2]$, $U\ [W/m^2K]$, $x\ [m]$

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 an isentropic efficiency.

3. TEST SETUP OF 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 m^3 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.
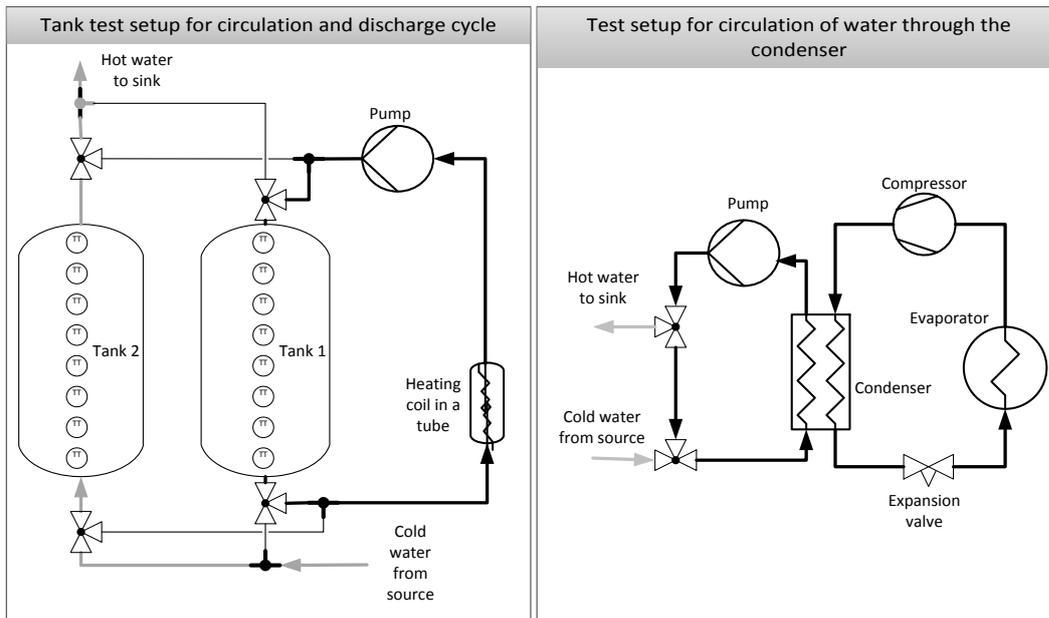

Figure 4 - The two test setups made to investigate prove the ISEC concept. Left: The test setup for the tank system. Right: The heat pump test system.

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°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 and new cold water from the source was tapped and the heating cycle was repeated. For the tests the condenser had an area of $3.48\ m^2$, the compressor was nominated to $9.4\ kW$ with at volume flow rate of $40\ m^3/h$, the UA value of the evaporator was $5740\ W/(m^2 K)$, supplied by the manufacturer.
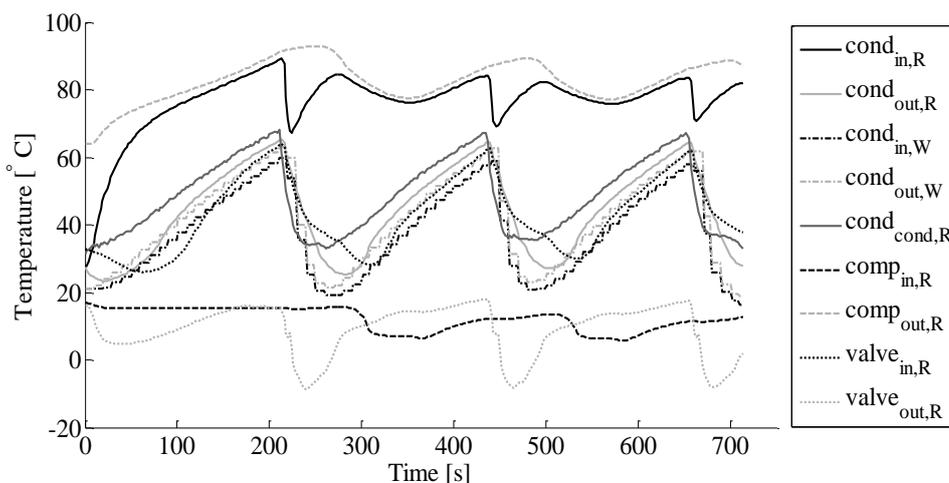

Figure 5 – Three consecutive heating cycles in the heat pump following the gradual heating principle of the ISEC concept. The temperature of the water (W) and refrigerant (R) is shown for the inlet (in) and outlet (out) of the condenser (cond), the compressor (comp) and the valve.

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 and not shown in 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 $20°C$ 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).

4. MODEL VALIDATION

The heat pump model

First the heat pump model was compared to test data from the heat pump test setup and afterwards 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 app. 250 seconds to 450 seconds.

Figure 6 (left) compares the pressures at in- and outlet of the condenser and evaporator of the heat pump tests to the model and figure 6 (right) shows a comparison of the measured temperatures at the inlet and outlet as well as 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 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 subcooling of the refrigerant, which occurs before the refrigerant leaves the condenser. The model assumed no pressure loss and no subcooling 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 thermal mass of the system is not taken into account in the model. This explains the higher temperature of the refrigerant in the beginning of the charge and lower temperature at the end observed from the
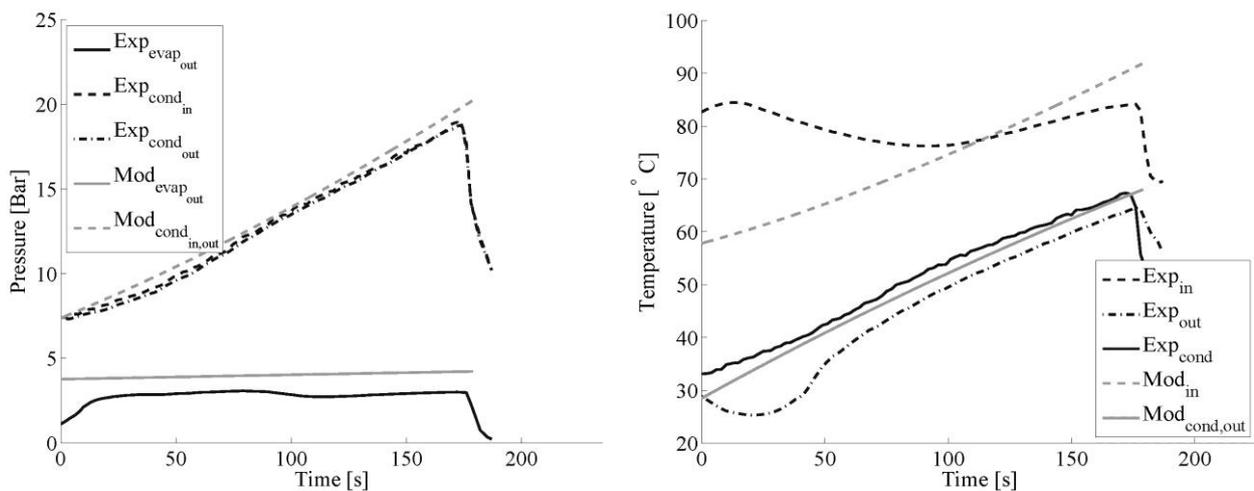

Figure 6 – Left: The measured and modelled pressures of the refrigerant at the inlet and outlet of the condenser and evaporator. Right: Comparison of the inlet, outlet and condensing temperatures of the refrigerant of measurements and model.

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 lift 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 dive in temperature in the beginning of the tests.
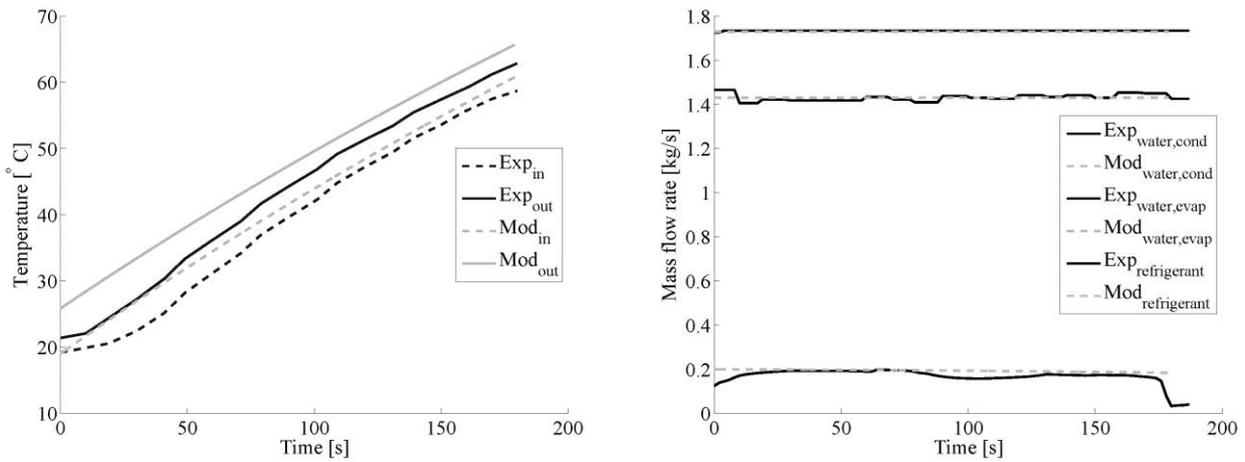

Figure 7 – Left: The water temperature into and out of the condenser for both the model and test. Right: The overall heat transfer coefficients for condenser modelled the one given for the heat exchanger of the test stand.

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 $2014\ W/m^2K$ 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.
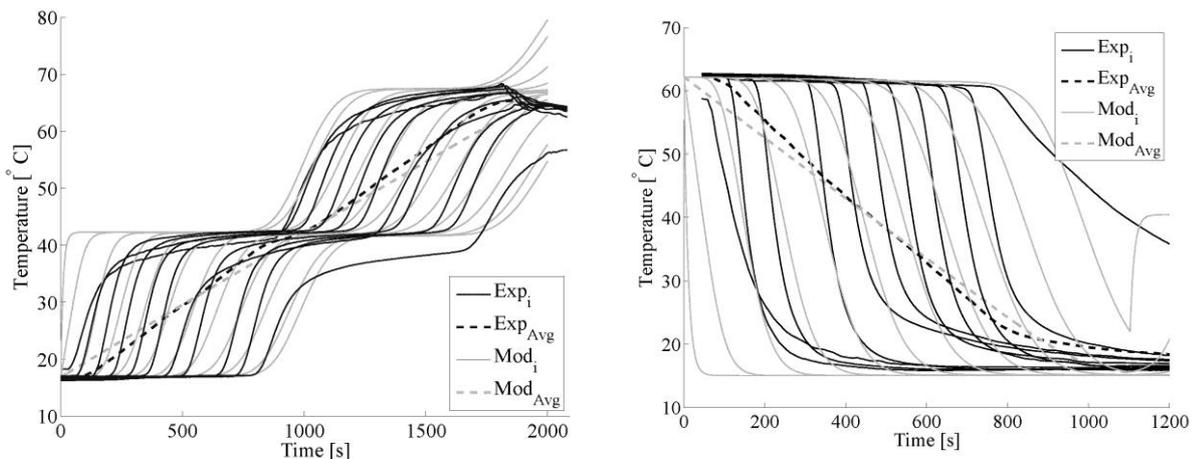

Figure 8 – Comparison of the measured and modelled temperatures in the water tank during charging and discharging shown for 10 equally distributed layers up through the tank and the average temperature of the water in the tank. Left: Charging of a tank. Right: Discharging of a tank.

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 and model of a tank charging and discharging it may be seen that the change in temperature 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. 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 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 during discharge fits well with the measured average temperature. A comprehensive study of the tanks is presented by Olsen (Olsen, et al., 2015).

5.   CONCLUSION

The ISEC system is a new concept and it has previously been shown that an increase in COP of 15 % compared to a heat pump which heat up without recirculation of water can be expected (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 see 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 model could be improved by implementing pressure loss across the condenser and evaporator and by calculating the heat transfer number in the evaporator. 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 in this paper it might explain some of the inconsistencies of the comparison between the calculated and measured results.

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