ENERGY ANALYSIS OF HEAT PUMP WITH SUBCOOLER

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Abstract: The paper presents a model based analysis for ground source heat pump with use of a subcooler to increase efficiency of domestic hot water preparation. A new mathematical model for ground source heat pumps with subcooler has been built for the design of such a heat pump and for the heat pump system simulations in TRNSYS environment. It has been shown that heat pumps with subcooler could theoretically increase the heating capacity by 10 to 25 % without the additional electric consumption. The simulation analysis has been made to compare the reference heat pump with standard design and an identical heat pump but with a subcooler as additional heat exchanger in DHW system. The real annual energy savings of the DHW system with a heat pump with subcooler are 7.5 %, the seasonal performance factor increases by 8.3 %.

Key Words: heat pump, subcooler, domestic hot water, system performance

1 INTRODUCTION

Heat pumps become very popular for space heating and domestic hot water preparation in Europe. The number of soled heat pumps at EU-14 was rising rapidly till 2009 and since than it has stagnated at 700 thousand pieces (see Table 1).

Table 1: Number of sold heat pumps in EU-14 (Nowak and Jaganjacova 2013)

Year	2005	2006	2007	2008	2009	2010	2011	2012
Sales	419 620	536 031	606 161	799 902	726 608	714 560	718 134	679 302

Heat pumps achieve high efficiency in low temperature space heating systems. Condensing temperature in those applications is relatively low. However a high condensing temperature is needed for domestic hot water (DHW) preparation and the seasonal performance factor of the heat pump system could be significantly reduced, often under value of primary energy efficiency limit (SPF = 2.67) set by European directive (EC Directive 2009). The heating season in central Europe lasts usually from October to May but hot water is needed all over the year. The trend of reduction of the space heating demand in new buildings results in a high share of DHW preparation in energy balance of buildings. An effective operation of the heat pumps in DHW application is a key for further development of the technology. One of the promising ways how to improve heat pump performance in domestic hot water applications is the use of the subcooler.

This paper describes the difference between the standard design of a heat pump and a heat pump with the subcooler. To derive the operational characteristic the numerical steady state model for both heat pump designs has been developed for use in TRNSYS simulation environment. Simulation analysis of DHW system with both heat pump alternatives (with and without subcooler) has been performed to reveal the benefit of subcooler.

2 SUBCOOLER

The majority of common heat pumps systems are based on the vapour Rankin cycle with compressor, evaporator, condenser and expansion valve. The subcooler is a heat exchanger placed at the outlet of the condenser. It cools liquid refrigerant further to lower temperature and increases the heat capacity of the whole system if suitable conditions are provided (see Figure 1).

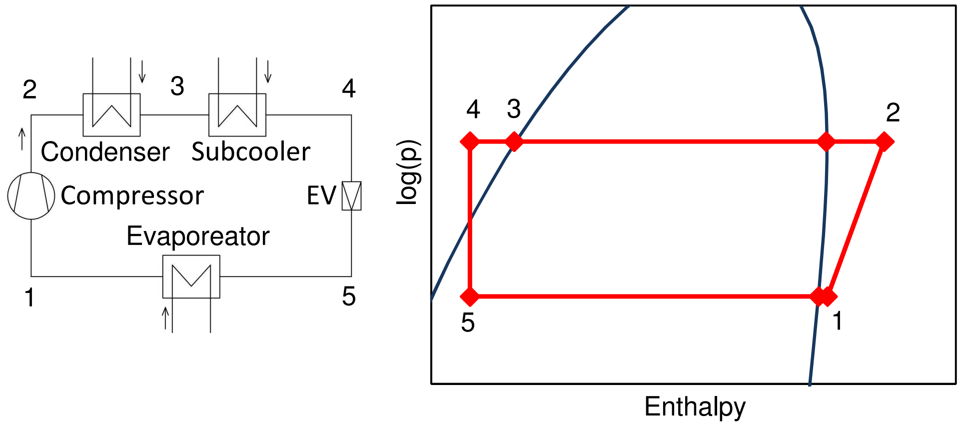

Figure 1: Pressure-enthalpy diagram and connection scheme of heat pump with subcooler

The heat capacity of the subcooler additional to the condenser heat capacity almost doesn't increase the compressor power input. The evaporator cooling capacity increases and due to that evaporating temperature slightly lowers (the change depends on the evaporators design), which leads to slightly higher compressor power input. However, the change in compressor power input is almost negligible. The coefficient of performance for heat pump with subcooler can be written as

$$COP=\frac{\dot{Q}_C+\dot{Q}_S}{P_{Comp}} \quad (1)$$

The subcooler heat is released at lower temperature level than the heat from condenser. Effective use of subcooler requires the low operation temperature, therefore it could be used e.g. for tempering rooms with low temperatures, preheating cold water in DHW applications, pool heating, etc.

The benefit from the subcooler is a function of several parameters. The most important are the condensing temperature, the temperature of the fluid entering the subcooler, specific heat transfer rate of heat exchangers (UA values) and the type of refrigerant. For the theoretical calculations of the heat capacity and the COP for the heat pump it is necessary to make complex calculations of the whole cycle.

3 HEAT PUMP MODEL DESCRIPTION

The heat pump model consists of a subroutines for modelling the compressor, condenser, evaporator and subcooler. The presented numerical model has been developed for ground source heat pumps. However it can be used with some changes or simplifications also for air source units. Every component is described by a sub-model within the main heat pump model. Only steady state conditions have been assumed to simplify the modelling. An idealized expansion is assumed in the expansion valve. It accurately models steady state condition if the valve is positioned at the evaporator inlet. The refrigerant mass flow rate through the expansion valve is exact to achieve the required superheating at the evaporator outlet. The enthalpy does not change during expansion. The pressure differences on heat exchangers and in piping connections have been neglected in the model.

3.1 Compressor sub-model

The compressor represents the principle element of a heat pump. It takes the evaporated refrigerant and presses it to the condensing pressure. The compressor consumes electric energy and its energy performance can be described by volumetric and isentropic efficiency characteristics.

The compressor sub-model is used to calculate refrigerant mass flow rate and outlet temperature for a given inlet pressure, inlet temperature and outlet pressure. At first refrigerant mass flow rate has to be calculated according equation 2.

$$\dot{m}_{ref}=\dot{V}_{sw} \cdot \lambda_v \cdot \rho_{su} \tag{2}$$

The volume flow rate is given by the geometry of the compression chamber and the compressor shaft speed. The volumetric efficiency is calculated by empirical equation 3 (Duprez et.al. 2007).

$$\lambda_v=C_1 + C_2 \cdot \sigma \tag{3}$$

A linear substitution is accurate enough for most spiral compressors without variable speed drives. Constants can be evaluated from manufacturer datasheets as best fitting substitution and can be found in the appendix.

An ideal gas compression is either isothermal or adiabatic. The real gas compression can be described as ideal gas compression with deviation. The deviation is equal to the isentropic efficiency (equation 4).

$$\eta_{ie}=\frac{P_{ie}}{P_r} \tag{4}$$

Isentropic efficiency is calculated by empirical equation 5 (Duprez et.al., 2007) in the sub-model.

$$\eta_{ie}=D_1+D_2 \cdot \sigma+D_3 \cdot \sigma^2+D_4 \cdot \sigma^3+D_5 \cdot \sigma^4 \tag{5}$$

Constants can be evaluated from manufacturer datasheets as best fitting substitution. Then the outlet enthalpy is given by equation 6.

$$h_{out}=h_{in}+\frac{h_{ie}-h_{in}}{\eta_{ie}} \tag{6}$$

3.2 Heat exchangers sub-model

The condenser and evaporator are divided into subsections. The condenser model contains desuperheating, condensing and subcooling regions. The evaporator is divided into evaporating and superheating regions. The steady state heat exchanger model uses common energy balance equations 7 to 9 for every region and side of the heat exchanger (refrigerant, heat transfer medium).

$$\dot{Q}_{hx,ref}=\dot{m}_{ref}\cdot\left(h_{ref,in}-h_{ref,\ out}\right) \quad (7)$$

$$\dot{Q}_{hx,liq}=\dot{m}_{liq}\cdot c_{p,liq}\left(t_{liq,out}-t_{liq,in}\right) \quad (8)$$

$$\dot{Q}_{hx}=UA\cdot\frac{\delta_1-\delta_2}{\ln\frac{\delta_1}{\delta_2}} \quad (9)$$

The overall heat transfer coefficient U is given by the manufacturer for given conditions (considered flow rates) and assumed to be constant to simplify the model. The fouling factor has been neglected.

The heat pump model iterates the condensing temperature, evaporating temperature and temperature after the subcooler in a loop until it reaches sufficient accuracy. The accuracy is given by the difference between the calculated UA value and the UA value given by the manufacturer. The input conditions are considered unchanged.

4 COMPARISON OF A HEAT PUMP WITH AND WITHOUT THE SUBCOOLER

Energy saving buildings have significantly reduced space heating demand and domestic hot water preparation becomes the dominant heat demand. The use of subcooler could increase the effectiveness of a high temperature hot water preparation. To reveal the benefit of the subcooler use the performance of particular a ground source heat pump with defined refrigerant R410A, compressor swept volume 4 m^3/h and heat exchangers have been modelled in two alternatives: with and without subcooler. A detailed characterization of components is shown in the appendix. The heat capacity of the reference heat pump with standard design is 5.5 kW at nominal conditions B0/W35 and a COP of 4.45.

4.1 Heat capacity at constant evaporating temperature

The condenser heat capacity is dependent on the evaporating temperature. Heat pumps are typically operated with low condensing temperatures below 40 °C in space heating mode and achieve high COP. The requirement of comfort hot water preparation (above 55 °C) leads to high condensing temperature above 60 °C and significant reduction of COP. Figure 2 shows the condenser heating capacity and subcooler heating capacity characteristics for a brine inlet temperature of 0 °C. The condenser inlet temperature $t_{c,in}$ and subcooler inlet temperature $t_{s,in}$ are considered as variables.

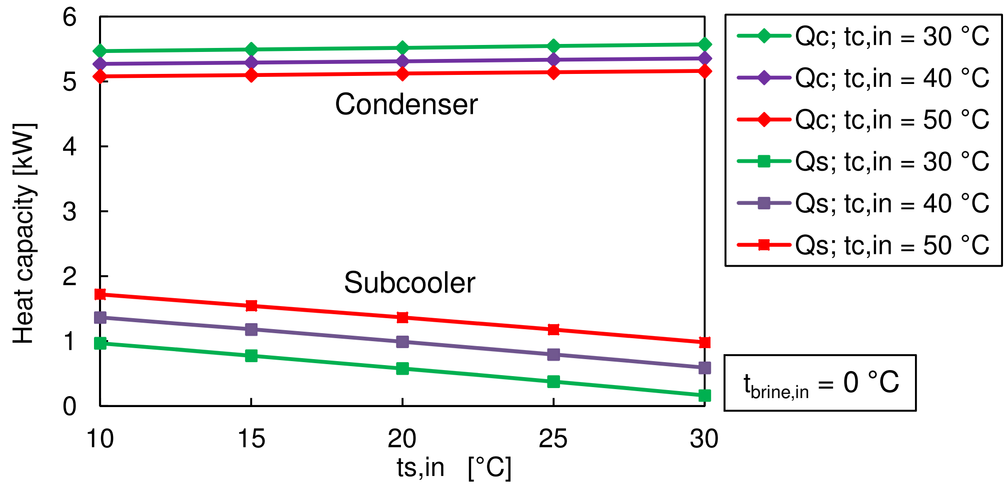

Figure 2: Condenser and subcooler heat capacity

The condenser heat capacity slightly decreases with rising condensing temperature. The subcooler heating capacity strongly depends on the temperature difference between the condensing temperature and the temperature of the water at the inlet of the subcooler.

4.2 Evaporator cooling capacity at constant condensing temperature

The cooling capacity depends on condensing temperature. If the condensing temperature grows, the evaporator cooling capacity degreases. For the same condensing and evaporating temperature the heat pump with the subcooler has a higher cooling capacity. The difference is equal to the subcooler heating capacity (see Figure 3). This has to be considered when dimensioning the ground source (e.g. boreholes). The other issue is a lower outlet temperature of the brine with the same brine mass flow. The mass flow should be higher to reach the same temperature difference in the evaporator.

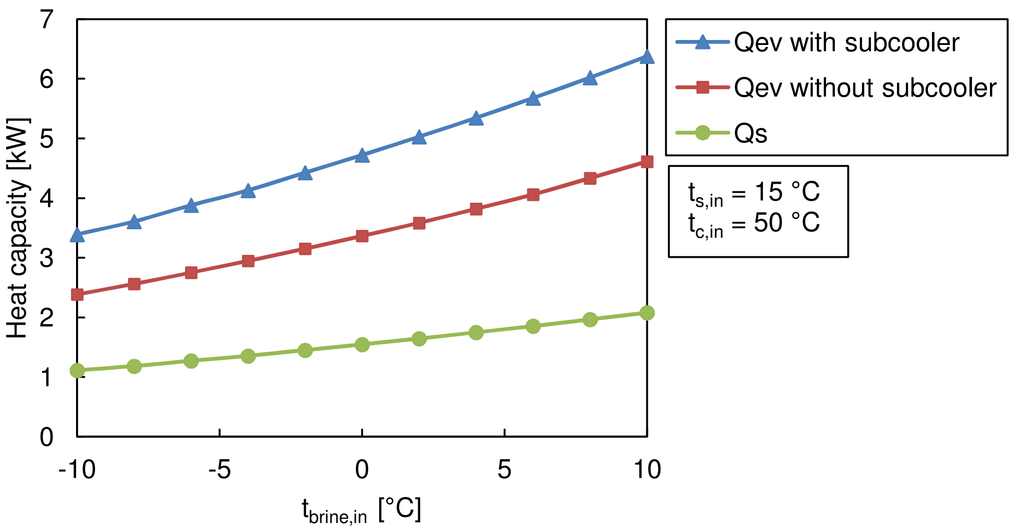

Figure 3: Evaporator cooling capacity and subcooler heat capacity

4.3 COP comparison

The COP of the heat pump strongly depends on many factors (condensing pressure, evaporating pressure, superheating and pressure drop in the suction line, compressor, etc.). Figure 4 shows the difference between the COP for the reference heat pump and the COP for the heat pump with subcooler for different brine inlet temperature and condenser inlet temperature. The subcooler inlet temperature is 15 °C (see Figure 4). However the temperatures are changing during heating up the storage in domestic hot water applications, therefore whole system simulations has to be done to approximate the real subcooler benefit (see part 5).

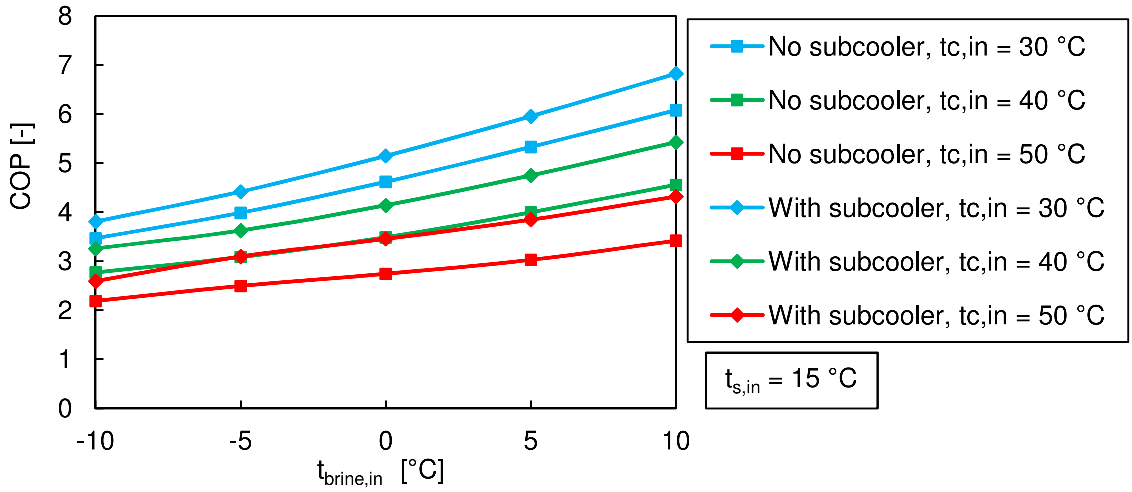

Figure 4: COP characteristics for both heat pump alternatives

The COP increases with increasing evaporating temperatures and decreases with higher condensing temperatures. An improvement in COP for heat pump with subcooler is higher for larger temperature difference between the condensing temperature and water inlet to subcooler.

5 USE OF SUBCOOLER FOR DOMESTIC HOT WATER PREPARATION

The domestic water preparation system with a ground source heat pump has been built in the simulation software TRNSYS (Klein 2010) for seasonal system simulations. A new TRNSYS component model for heat pumps has been used to model both the heat pump with standard design and the heat pump with subcooler. Both heat pumps use identical components (compressor, heat exchangers, etc.) except the subcooler which is used as additional heat exchanger. The heat capacity characteristics of the heat pumps alternative have been shown above. The water storage tank has been modelled with the component type 340 (Drueck 2006), the ground borehole with a depth of 110 m has been modelled with the type 451 (Wetter and Huber 1997).

Both DHW preparation systems use a hot water tank with internal tube heat exchanger(s). The hot water tank has a water volume of 300 l. The temperature of cold water is 10 °C assumed as constant all over the year. The required temperature of hot water has been considered 55 °C (according to Czech legislation). The daily hot water consumption of 160 l/day has been used as typical value for house with 4 persons. The hot water daily load profile has been used in accordance with the European standards (EN 15450:2007) for a household with bath.

5.1 System without subcooler

The reference heat pump DHW system alternative considers a standard heat pump connected to one internal heat exchanger in a hot water tank. The surface area of the internal heat exchanger is 4.8 m^2. The tank volume is charged by the condenser of the heat pump according to the temperature sensor located in the lower part of water tank. If the temperature at the outlet from the hot water tank is lower than the required temperature, auxiliary heater heats up the outlet water to the required temperature. The hydraulic layout is shown in Figure 5.

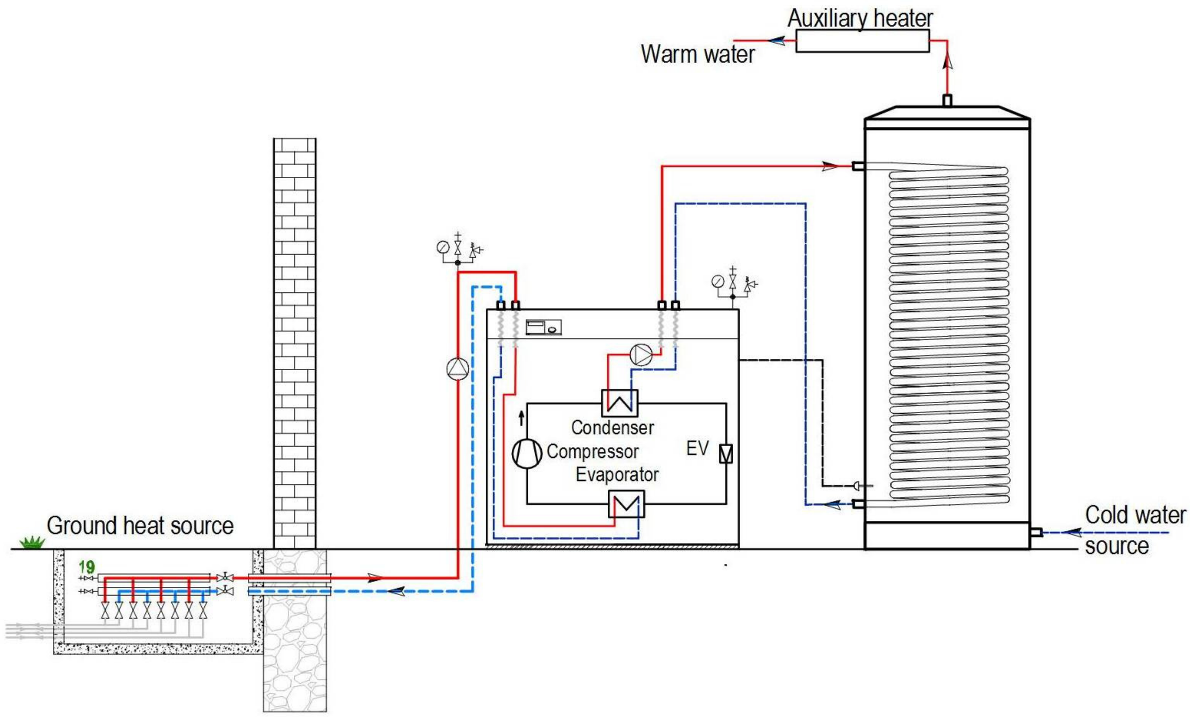

Figure 5: Reference DHW system with a standard heat pump

5.2 System with subcooler

An alternative heat pump DHW system with subcooler considers a heat pump condenser connected to an upper internal heat exchanger and a subcooler connected to a lower internal heat exchanger (preheating cold zone) in a hot water tank. The surface area of upper internal heat exchanger is 3.0 m^2, the surface area of the lower internal heat exchanger is 1.8 m^2. The upper tank volume is charged by the condenser of the heat pump according to the temperature sensor located in the upper part of water tank. The hydraulic layout is shown in Figure 6.

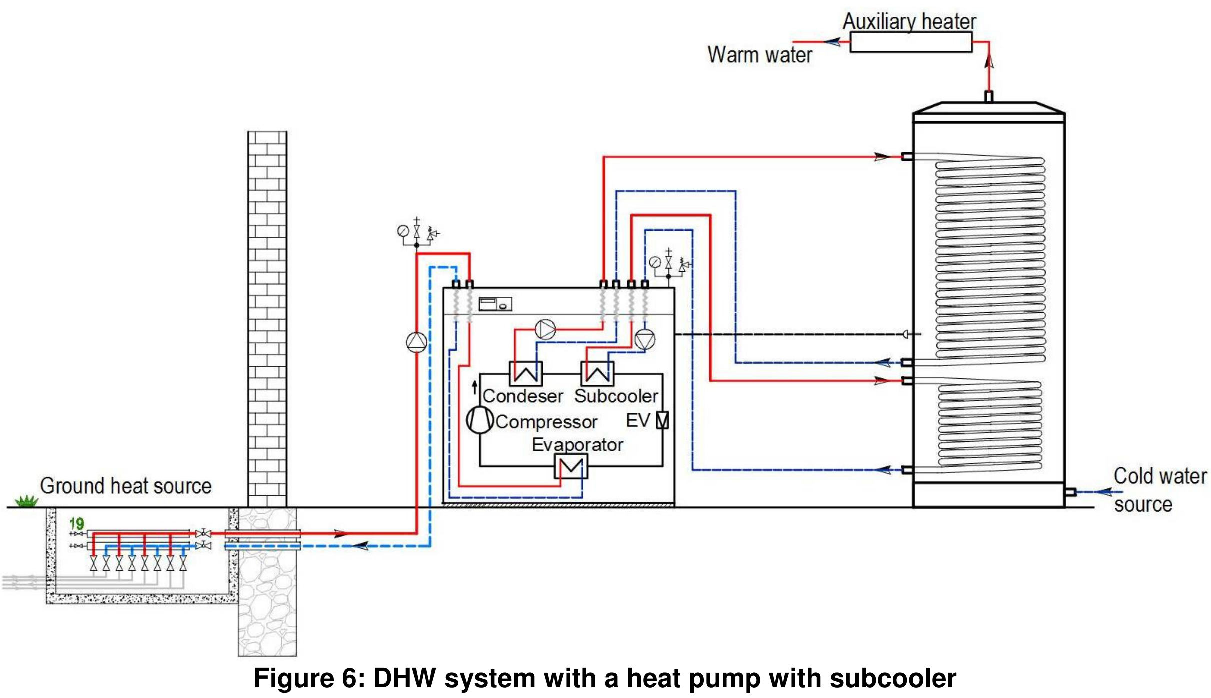

Figure 6: DHW system with a heat pump with subcooler

5 RESULTS AND DISCUSSION

Results from the whole year simulation for both DHW hot water system alternatives are shown in Table 2. Both system alternatives have identical heat demand for hot water load. The seasonal performance factor of the DHW system has been evaluated as a ratio between heat energy delivered to DHW load and total electricity consumption of the system (heat pump, auxiliary heater).

Table 2: Whole year simulations results of DHW system alternatives

Result	Reference [kWh/a]	With subcooler [kWh/a]
Energy demand for hot water preparation	2897	2897
Domestic hot water tank heat losses	141	149
Energy output of condenser	3026	2353
Energy output of subcooler	0	693
Heat pump electricity consumption	943	871
Auxiliary heater heat supply	12	0
Circulation pumps electricity consumption	47	56
Seasonal performance factor (SPF)	2.89	3.13

The electricity demand of the DHW system based on the heat pump with subcooler is 7.5 % lower. The seasonal performance factor is approximately 8.3 % higher. This is significantly lower than the potential shown in static characteristics (around 22 %) with the heat pump model for fixed temperature into the subcooler e.g. 15 °C. It is mainly caused by realistic fluctuation of temperature difference between condensing and subcooling temperature. When the inlet temperature to the subcooler reaches 30 °C its benefit is very low. The benefit from the subcooler could be improved after optimizations of the tank size (volume of cold water zone), positions of inlets and outlets for tanks heat exchangers and also sensors position. The basis for further optimization can be daily profiles of the water temperature inside the water tank evaluated from simulations (see Figure 7).

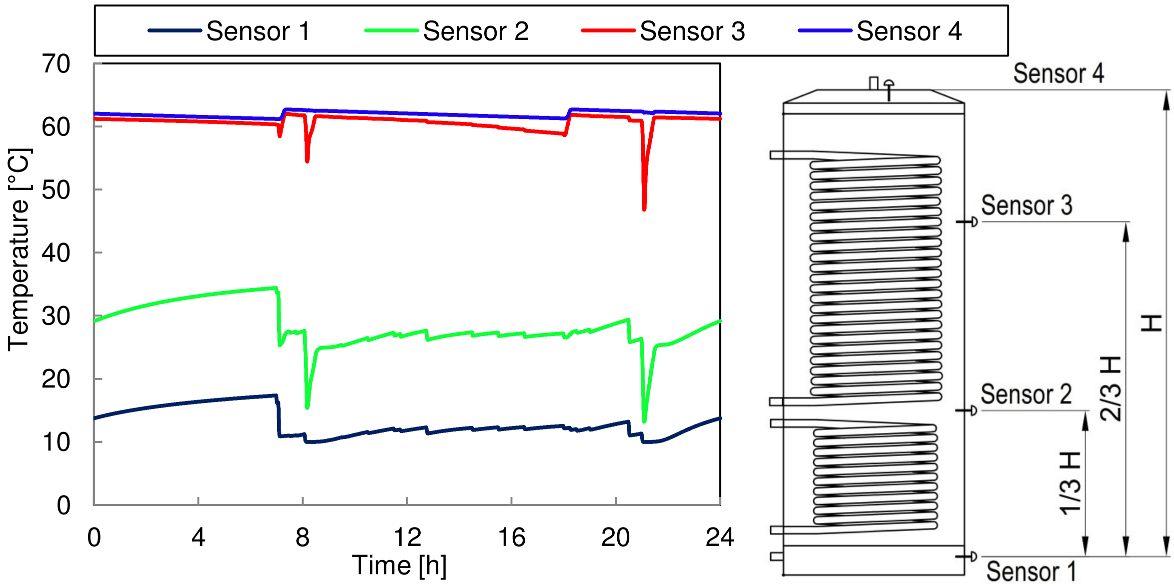

Figure 7: Daily profile of temperature inside the water tank

The heat pump with a desuperheater uses a high temperature of the refrigerant after the compression. Those high temperatures are reached even with a low condensing temperature, therefore it can be used with an advantage in parallel space heating and DHW operation. Also when it is used just for DHW applications, the upper part of storage is heated by desuperheater and condensing temperature doesn’t need to be so high. The desuperheater capacity is limited so the system can have a bigger problem to cover DHW demands peaks.

6 CONCLUSION & OUTLOOK

The model of a heat pump with a subcooler has been developed and built for the TRNSYS simulation environment. The model has been used for the determination of static characteristics of a heat pump based on compressor and heat exchanger parameters to compare the heat pump of standard design and the alternative with the subcooler. The potential of energy savings of the subcooler use and limiting conditions have been shown in a range of 10 to 25 %.

The real benefit of the subcooler application has been evaluated by a whole year energy analysis. Simulations of the domestic hot water system have been performed in TRNSYS environment to compare the performance of the reference DHW system with a standard heat pump and DHW system with a heat pump with additional subcooler. The subcooler increases the COP by 8.3 % in a typical family house DHW only application. Further analyses will focus combined space heating and DHW systems and parametric evaluation for different system conditions (temperature, tank size, heat exchangers layout).

The future work will be focused on the advanced heat pump with desuperheater for DHW, subcooler and the superheater (superheater increases the heating capacity of the desuperheater and can consume the heat from subcooler).

The subcooler increases the evaporator cooling capacity. This could be important for proper ground heat source and brine mass flow dimensioning. To fully use the subcooler potential the hot water consumption profile has to be known and hot water tank dimensioned accordingly.

7 ACKNOWLEDGMENTS

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8 NOMENCLATURE

A = area, m^2
C = constant, -
COP = coefficient of performance, -
D = constant, -
H = height, m
h = enthalpy, $J \cdot kg^{-1}$
$\dot{m}$ = mass flow, $kg \cdot s^{-1}$
p = pressure, Pa
P = power input, kW
$\dot{Q}$ = heat capacity, kW
SPF = seasonal performance factor, -
t = temperature, °C
U = overall heat transfer coefficient, $W \cdot m^{-2} \cdot K^{-1}$
$\dot{V}$ = volume flow rate, $m^3 \cdot h^{-1}$
δ = temperature difference, K
λ = efficiency, -
ρ = density, $kg \cdot m^{-3}$
η = efficiency, -
σ = pressure ratio, -

Subscripts

1,2,..,5 = numbers
brine = brine
c = condenser
ev = evaporator
hx = heat exchanger
in = inlet
is = isentropic
liq = liquid
ref = refrigerant
s = subcooler
sw = swept
su = suction
v = volumetric
out = outlet

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APPENDIX

Table 3: Compressor constants

C1	1.047	-
C2	-0.0377	-

D1	-0.7859	-
D2	1.5845	-
D3	-0.603	-
D4	0.0956	-
D5	-0.0055	-

Table 4: Heat exchangers UA

Condenser	1700	W/K
Evaporator	1100	W/K
Subcooler	200	W/K