Ing. Jan SEDLÁŘ
CTU in Prague,
University Centre for Energy Efficient Buildings

Heat Pump for Water Preparation in a Block of Flats

Příprava teplé vody v bytovém domě tepelným čerpadlem

Recent trends in building construction in central Europe show a growing number of well insulated residential buildings. The heat demand for domestic hot water preparation is, in those applications, similar or higher than for space heating. Regular vapour-compression heat pumps reach a relatively low SPF 2.5 in such applications. The way to increase SPF is to use a heat pump with multiple heat exchangers on the high-pressure side (desuperheater, condenser, and subcooler) and a special hot water storage tank with three internal heat exchangers for good temperature stratification. A prototype of a heat pump with a desuperheater and subcooler was developed and a mathematical model of a refrigerant cycle was tested on it. The model of the heat pump describes the heating capacity and power input with an average relative deviation lower than 5 %. The validated model was used in simulations of a system for hot water preparation in a block of flats. The water draw in the simulation was set under the European standard and measured data from real applications. The results of simulations show SPF improvement by 26 % for a heat pump with a desuperheater and subcooler in comparison with a standard heat pump.

Keywords: *subcooler; desuperheater; domestic hot water*

Poslední trendy při výstavbě budov ve střední Evropě ukazují zvyšující se počet dobře zateplených budov. Potřeba tepla na přípravu teplé vody je podobná nebo může být i vyšší než pro vytápění. Klasická tepelná čerpadla se základním jednostupňovým parním cyklem dosahují při přípravě teplé vody relativně nízkého SPF 2.5. Jednou z možností zvýšení SPF je použití více výměníků tepla na vysokotlaké straně (kondenzátor + chladič par + dochlazovač) a upraveného zásobníku teplé vody pro dobrou teplotní stratifikaci. Pro určení potenciálního přínosu zmíněného uspořádání byl vyvinut prototyp a matematický model tepelného čerpadla s chladičem par a dochlazovačem. Model dokázal popsat chování tepelného čerpadla s průměrnou odchylkou 5 %. Validovaný model byl pak použit v simulacích systému pro přípravu teplé vody v bytovém domě. Výsledky simulací ukazují zlepšení SPF až o 26 % pro tepelné čerpadlo s chladičem par a dochlazovačem v porovnání se standardním zapojením.

Klíčová slova: *chladič par, dochlazovač, teplá voda*

INTRODUCTION

Heat pumps offer the possibility to substitute traditional heat sources like gas heaters, oil boilers, etc. and to reduce the primary energy consumption. The heat pump market shows growth in a number of installed units over the last three years [1]. With an increasing number of well-insulated buildings in central Europe brings a growing share of heat for hot water preparation in the total energy consumption of a building. The average energy consumption of space heating per m^2 is less than 70 % of its value from 1990 [2]. The energy demand for space heating is almost identical to water heating. A bigger portion of domestic hot water (DHW) in the total energy consumption of a household creates a demand for an increase in the energy efficiency of heat pumps in such applications. Heat pumps designed for DHW applications are measured under the standard EN 15147 [3]. The measured coefficient of performance (*COP*) [-] of a heat pump with a tapping profile can be recalculated into the energy efficiency of the hot water preparation and can be compared with other types of water heaters.

The portion of energy for water heating in the total energy consumption in a block of flats might be greater than for space heating due to the building envelope which is usually well insulated. The energy needed for water heating is higher than that in family house applications for the same water draw due to heat losses in the distribution system. In some applications, heat losses in distribution are comparable with the heat demand.

A typical block of flats in the Czech Republic is connected to a district heating system from a central gas boiler room or a heating plant. Energy savings over the last 15 years lowered energy consumption and made district heating systems less efficient. The maintenance and operational costs of the system did not change and the downswing in traded heat forced district heating providers to raise the price of heat [4]. This action is forcing the transformation from district to local heating systems and sometimes to systems with heat pumps in the long term.

This paper discusses the possibility of increasing the efficiency of the system with a heat pump (HP) in a block of flats application for water heating. The paper is based on an efficiency measurement of an air source HP in five blocks of flats construction over a one-year period. The paper includes a basic HP with a desuperheater and a subcooler model and its validation. The verified model of the HP was used in the TRNSYS [5] simulation software with a model of the DHW preparation.

THE MATHEMATICAL MODEL OF THE HEAT PUMP

To estimate the heat capacity and a power input of a brine/water source heat pump, a mathematical model of the heat pump (HPM) was developed. The HPM is comprised of compressor, condenser, evaporator, desuperheater and subcooler models. The model of the expansion valve is not included because the process in the expansion valve is considered adiabatic, which is valid for direct expansion evaporators [6]. Other devices and pipes in the refrigeration cycle are just considered as negligible pressure losses and are not taken into account in the model. The thermodynamic properties of a refrigerant are enumerated by Peng-Robinson equation of state [7]. The model is simplified in general so that its parameters can be derived from the manufacturer's data sheets for each component.

Model assumptions

The model should approximate the performance of the HP prior to its construction. The only available data comes from the data sheets of the components, therefore, the model has to neglect any minor technical characteristics of the components. The list of assumptions and simplifications is here:

- a heat transfer area A [m^3] and an overall heat transfer coefficient U [W·m^{-2}·K^{-1}] value of each heat exchanger (HX) are stable and constant
- a stable superheat in the evaporator
- a stable subcooling in the condenser
- pressure losses are negligible
- there are no heat losses from the refrigerant cycle
- expansion in the expansion valve is adiabatic
- oil does not influence the refrigerant cycle and heat transfer

The assumption of a stable UA value in each HX is problematic. The UA value varies with flow rates, temperatures and other properties on both the refrigerant and water/brine side of the HX. However, a small change does not significantly influence the overall results. More complex model would make the simulation slower.

The stable superheat of the refrigerant in steady state conditions can be achieved by an electronic expansion valve in case of the superheat in the evaporator is higher than the minimal stable superheat [9]. The subcooling in the condenser is given by the refrigerant charge and in the case that refrigerant collector is situated between the condenser and expansion valve. Pressure losses influence the precision of the modelling [6]. However, it is difficult to estimate them without knowledge of the piping in the refrigerant cycle and the internal geometry of the HX. Heat losses can be neglected with insulated devices, especially when the compressor is insulated.

Model of the compressor

A simplified model of the real compressor is derived from the theory of piston compressors. It describes the mass flow of the refrigerant and the power input of the compressor in every working condition and can be derived from the manufacturer's data sheet. The refrigerant mass flow rate m_{ref} [kg·s^{-1}] is described as follows:

$$m_{ref} = V_{sw} \lambda_v \rho_{ref,s} n \quad [\text{kg·s}^{-1}] \tag{1}$$

In the above equation, V_{sw} is the swept volume of compressor [m^3], $\rho_{ref,s}$ is the refrigerant density [kg·m^{-3}] at the compressor suction, λ_v is the volumetric efficiency [-] and n is the compressor rotational speed [s^{-1}]. The swept volume V_{sw} is provided by the manufacturer of the compressor. The rotational speed n is given by the local electric network frequency. λ_v is changing with the different pressure and temperature state of the refrigerant in the suction and discharge of the compressor. It depends on the pressure ratio. In foregone applications of vastly used piston compressors, the volumetric efficiency was a function of the clearance volume when the piston was at top dead centre. Today's mostly used compressors in heat pumps applications are without the mentioned issue, λ_v is close to 1 and can be described as follows:

$$\lambda_v = 1 - C(\sigma - 1) \quad [-] \tag{2}$$

In the above equation, σ is the pressure ratio [-] and C is a constant [-]. The isentropic efficiency η_{ie} [-] describes the compressor's efficiency energy. The real compression process can be defined as polytropic with a varying exponent of compression. A simplified description of the real compression is undertaken through the difference between real and isentropic compression. η_{ie} in the model is described as follows [9]:

$$\eta_{ie} = D_1 + D_2\varphi + D_3\varphi^2 + D_4\varphi^3 + D_5 p_{con} \quad [-] \tag{3}$$

$$\varphi = \sigma^{1/n_{pol}} \quad [-] \tag{4}$$

In the above equation, p_{con} is the condensing pressure [Pa], D_1 [-] to D_4 [-] and D_5 [Pa^{-1}] are constants [-] for the parametrisation, and φ is defined in Equation (4), where n_{pol} is the average exponent of the isentropic compression [-]. The model of the compressor calculates the pressure ratio, the isentropic and volumetric efficiency according to the pressure at the suction and discharge.

Model of the heat exchangers

The model of the HX calculates the heat exchange rates from the calorimetric equations and the heat transfer equation. For steady state conditions, the heat flows of liquid on both sides are equal to the heat flow transferred through the working surface. It can be described as follows:

$$Q_{hx,1} = m_{ref}\, abs\left(h_{ref,in} - h_{ref,out}\right) \quad [\text{W}] \tag{5}$$

$$Q_{hx,2} = m_{liq} c_{p,liq}\, abs\left(t_{liq,out} - t_{liq,in}\right) \quad [\text{W}] \tag{6}$$

$$Q_{hx,3} = UA\delta_{\ln} \quad [\text{W}] \tag{7}$$

In the above equations, $Q_{hx,1}$, $Q_{hx,2}$ and $Q_{hx,3}$ are the heat exchange rates [W] on the refrigerant side, the secondary side and the heat transfer through the heat exchanger surface area rate, respectively. $h_{ref,in}$ and $h_{ref,out}$ are specific enthalpies [J·kg^{-1}] of the refrigerant at the inlet and outlet of the section, $c_{p,liq}$ is the specific capacity [J·kg^{-1}·K^{-1}] of the secondary liquid, m_{liq} is the mass flow rate [kg·s^{-1}] of the secondary liquid, $t_{lig,in}$ and $t_{liq,out}$ are the temperatures [°C] of the secondary side liquid at the inlet and outlet of the section, δ_{ln} is the logarithmic mean temperature difference [K]. The value of UA [W·K^{-1}] can be found in the data sheets of the HX. The heat pump model consists of four HX – condenser, evaporator, desuperheater, and subcooler. The heat exchangers are modelled in sections in accordance with the expected phase condition on each side:

- an evaporator - two sections,
- a condenser - two or three sections,
- a desuperheater - one or two sections,
- a subcooler - one section.

The operating parameters of the heat pump

Every heat pump works with some refrigerant. The thermodynamic properties are calculated with a mathematical model of the refrigerant in every time step. The type of refrigerant is given at the beginning of the simulation and does not change, therefore, it is the first parameter. Other parameters are set at the beginning of the simulation are:

- the superheating temperature difference in the evaporator,
- the subcooling temperature difference in the condenser,
- superheating in the compressor suction piping,
- the pressure loss in the compressor suction,
- the operating envelope of the compressor.

The operating envelope is necessary for system simulations. It limits the compressor working conditions. The model checks the maximum and minimum operational pressure, the pressure ratio and the outlet temperature of the refrigerant. If the limited boundary is crossed, the model responds accordingly (e.g., when the temperature at the compressor discharge is over the maximum temperature model, the compressor turns off).

Cycle convergence algorithm

A special algorithm was developed in order to solve the HP model with three successive HXs on the high-pressure side of the refrigerant cycle. The algorithm can be divided into two parts. The first part searches for the optimal condensing temperature t_{con}, the evaporating temperature t_{ev}, the specific enthalpy at the outlet of the desuperheater $h_{des,out}$ and the temperature of the refrigerant at the outlet of the subcooler $t_{s,out}$ in sequential order to find a balance of energy between Eq. 5, 6 and 7 for every used HX. The second part calculates the model of the refrigerant system and the HX.

MODEL VALIDATION

The model of the heat pump was tested on a heat pump prototype. The schematic of the refrigerant cycle and test bench is shown in Fig. 1. The prototype contains a scroll compressor, a brazed plate HX and an electronic expansion valve. The list of the main components and properties for the model validation is shown in Table 1. The prototype also contains the superheater which was implemented solely for testing and it is not included in the validation. The prototype works with refrigerant R410A.

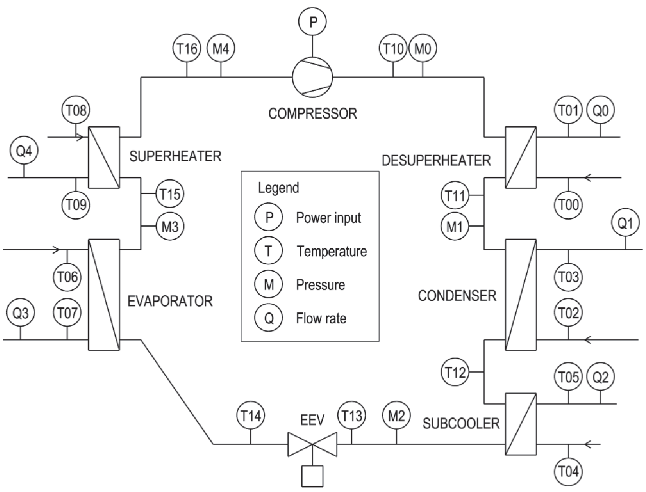

Figure 1 Diagram of the tested heat pump and measured values

Table 1 List of components used in the tested prototype

Component		Value	Unit
Scroll compressor	V_{sw}	21.7	cm^3
	C	$2.22 \cdot 10^{-2}$	-
	D_1	$-2.31 \cdot 10^{-2}$	-
	D_2	$8.31 \cdot 10^{-1}$	-
	D_3	$-2.48 \cdot 10^{-1}$	-
	D_4	$2.22 \cdot 10^{-2}$	-
	D_5	$1.36 \cdot 10^{-8}$	Pa^{-1}
Evaporator	UA_{con}	1494	$W \cdot K^{-1}$
Condenser	UA_{ev}	1306	$W \cdot K^{-1}$
Desuperheater	UA_{des}	100	$W \cdot K^{-1}$
Subcooler	UA_{sub}	300	$W \cdot K^{-1}$

The prototype was measured in various working conditions. The superheating controlled by the expansion valve was set at 4 K. The average subcooling in the condenser was 2 K. The results of the model validation are shown in Table 2 and in Fig. 2, 3, 4 and 5. $Q_{heat,total}$ is the total heat transfer on the high-pressure side of the HP. The average error of the *COP* is less than 5 % of its value. The model fits the measured data with good accuracy.

Table 2 List of errors of the model from the measured data

Working arrangement (on the high-pressure side)	*P* (%)		$Q_{heat,total}$ (%)		*COP* (%)	
	Avg.	Max.	Avg.	Max.	Avg.	Max.
Condenser	3.1	8.2	3.0	8.2	4.1	8.2
Condenser, desuperheater	4.0	10.2	3.7	10.0	4.2	9.6
Condenser, subcooler	5.6	10.0	5.3	8.9	1.2	4.0
Condenser, desuperheater, subcooler	4.3	10.4	5.3	10.0	1.8	3.8

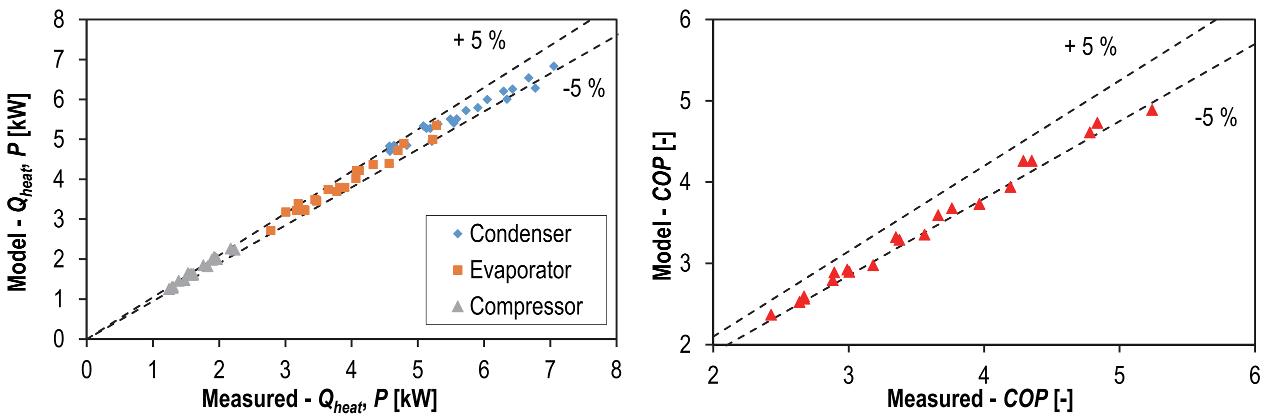

Figure 2 The deviation between the model and the measured data for the heating capacity, the power input (on the left) and the COP (on the right) in a condenser arrangement

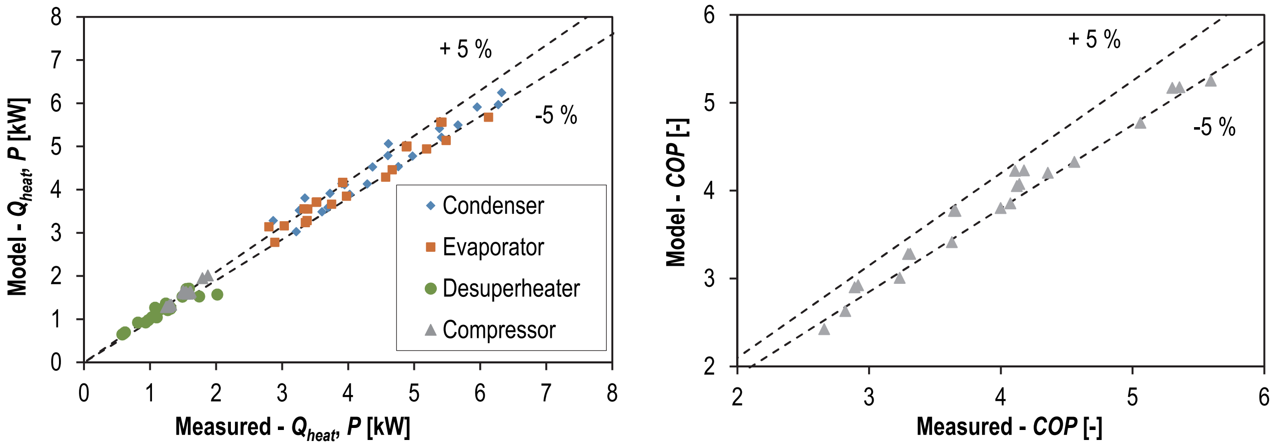

Figure 3 The deviation between the model and the measured data for the heating capacity, the power input (on the left) and the COP (on the right) in a condenser + desuperheater arrangement

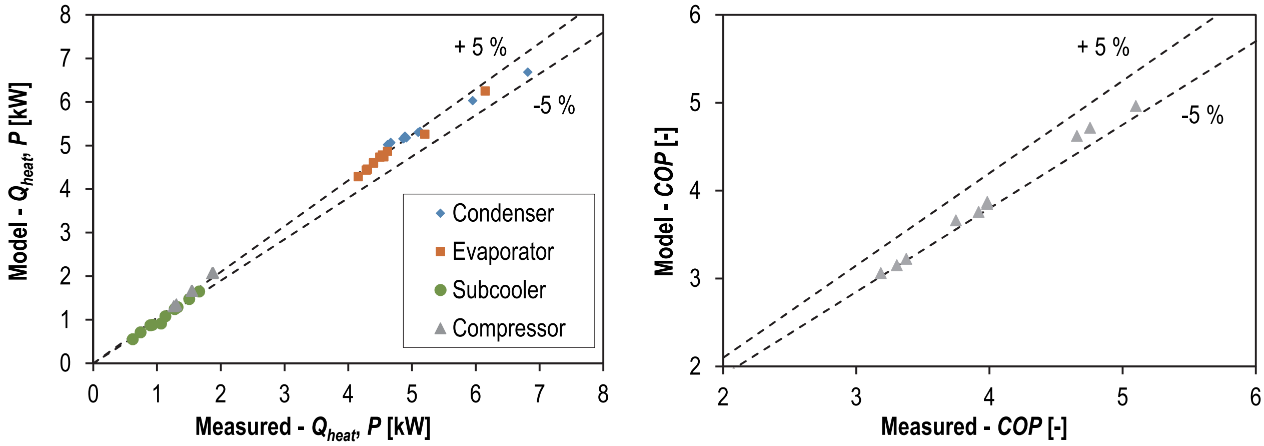

Figure 4 The deviation between the model and the measured data for the heating capacity, the power input (on the left) and the COP (on the right) in a condenser + subcooler arrangement

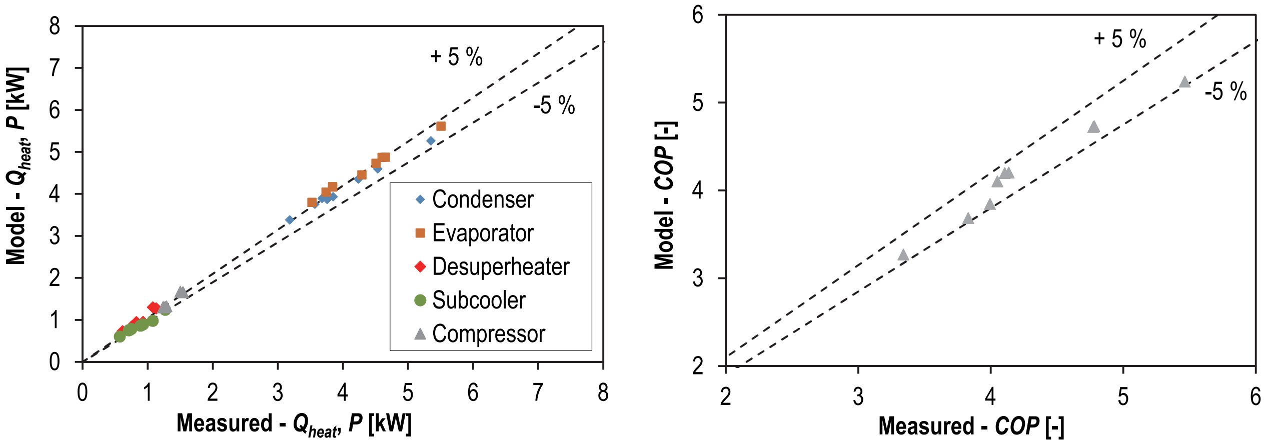

Figure 5 The deviation between the model and the measured data for the heating capacity, the power input (on the left) and the COP (on the right) in a condenser + desuperheater + subcooler arrangement

HOT WATER PREPARATION SYSTEM IN APARTMENT BUILDING

The validated model of the heat pump was used in the simulation of domestic hot water (DHW) preparation system in a block of flats application. The DHW consumption is modeled in TRNSYS simulation software and was dimensioned in accordance with the measured data from the real HP application. Fig. 6 shows the main characteristics of the measured system - supplied heat, electric energy consumption and measured SPF. The total electricity consumption is 36.5 MWh·a^{-1}, the total heat provided by the HP and the electric boiler is 91.3 MWh·a^{-1} and the SPF of the whole DHW system is 2.5.

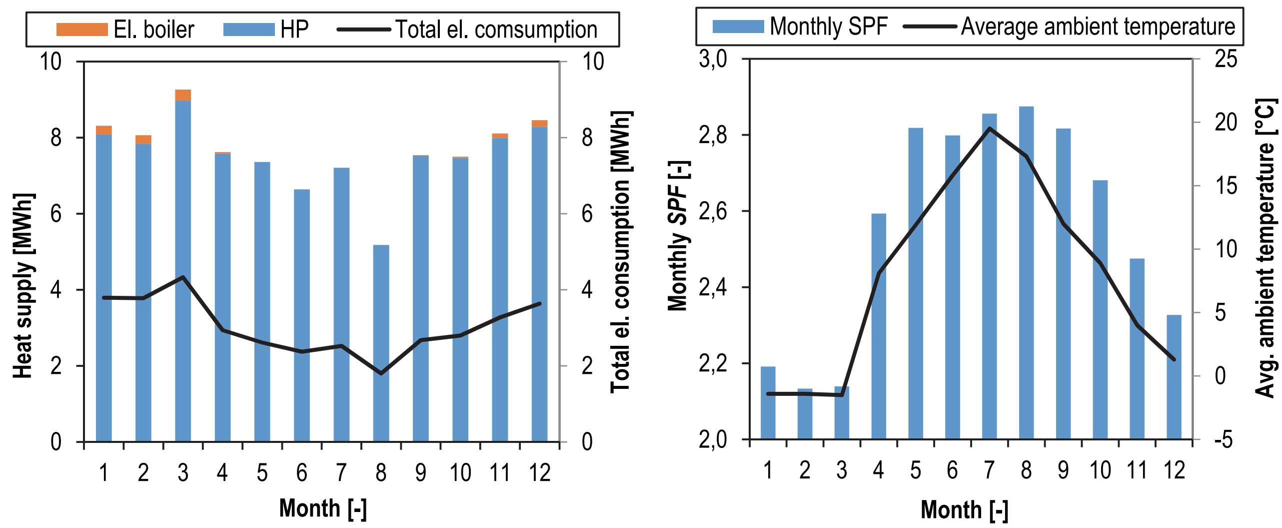

Figure 6 The measured data of the heat supply from the heat pump and the electric consumption (on the left), the calculated SPF of the system and the average ambient temperature (on the right)

The measured system is in the western part of the Czech Republic and is assembled with an air source HP. The nominal heating capacity of the HP (A7/W35) is 20.2 kW. The backup heater is an electric boiler. The heat pump regulation turns off the unit when the ambient temperature is lower than -10 °C and the whole energy consumption is then covered by an electric boiler. The block of flats has 48 housing units. The demanded temperature of the DHW to the circulation system is 55 °C. The DHW circulation system is needed to ensure comfort for residents and works nonstop, however it is estimated that 40 % of the energy is lost there. The accumulation of hot water is provided by a 6 m³ water tank. The hot water tapping profile was modelled with respect to the standard EN 16147 [3] and the number of the housing unit. The tapping profile is shown in Fig. 7. Hot water consumption is not same during the year (see Fig. 6), therefore, a function describing the monthly profile was implemented in the simulation.

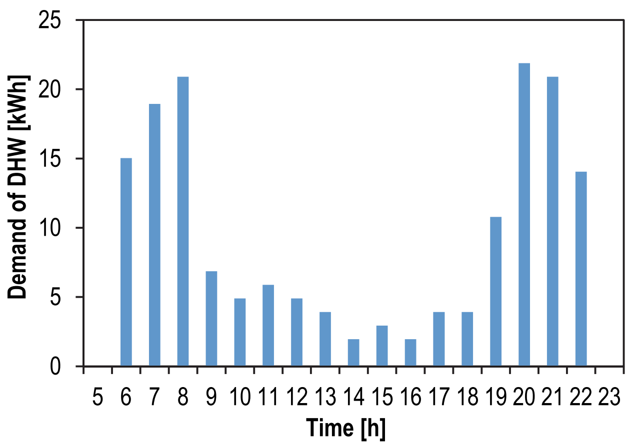

Figure 7 Daily hot water tapping profile

The model of the heat pump described in the chapter Model validation predicts the working behaviour of ground source HP. To describe the air source HP, correction of the heating capacity was undertaken. The correction is based on the ideal working behaviour of the ground source HP. The relative deviation between the ground source heat pump modelled data and the data sheet values is a function of the ambient temperature. The correction is well described by the hyperbole function of a higher order:

$$\xi = \frac{E_1}{t_{amb}^4 + E_2 \cdot t_{amb}^3 + E_3 \cdot t_{amb}^2 + E_4 \cdot t_{amb} + E_5} \quad [-] \qquad (8)$$

The function which describes heating capacity is:

$$Q_{heating} = Q_{heating,GS}\left(1-\xi\right) \quad [W] \qquad (9)$$

In the above equations, ξ is the correction factor [-], t_{amb} [°C] is the air inlet temperature (equal to the ambient temperature in the model), $Q_{heating,GS}$ is the heating capacity [W] calculated by the ground source heat pump model, Q_{heat} is the heating capacity of the HP after the correction and E_1 to E_5 are unknown constants for parametrisation.

The model of the heat pump was parameterised to describe the manufacturer's data. The data fit is shown in Fig. 8. The heating capacity is the same for the 35 °C and 45 °C outlet temperature from the condenser. The model contains some unknown constants and, therefore, the parametrisation is precise enough. The average error of the fitted values is 0.53 % for the heating capacity, 3.86 % for the power input and 3.55 % for the *COP*.

SIMULATION MODEL OF THE DHW SYSTEM

The previously described DHW system was modelled in TRNSYS simulation software. The weather was modelled with meteorological data for Prague. The list of constants used in the HP simulation is shown in Table 3 and 4.

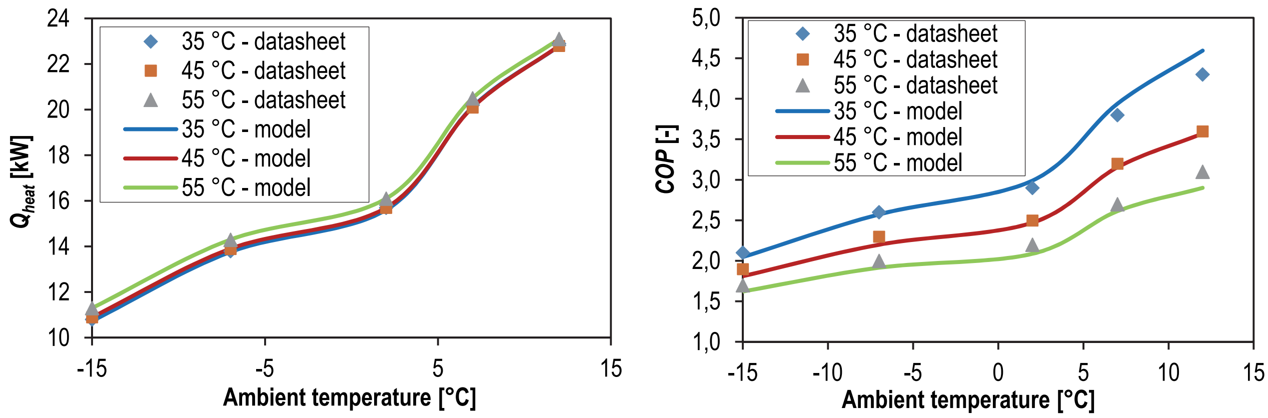

Figure 8 The heating capacity (on the left) and COP (on the right) as a function of the ambient temperature

Table 3 List of the constants used in the simulations

V_{sw}	C	D	E_1	E_2	E_2	E_4	E_5
(cm^3)	(-)	(-)	(-)	(°C^{-3})	(°C^{-2})	(°C^{-1})	(-)
97.2	$2.22 \cdot 10^{-2}$	See Table 1	-2.14	1.86	-2.83	-56.2	117.5

Table 4 List of the UA values used in the simulations

System	UA_{con} (W·K^{-1})	UA_{ev} (W·K^{-1})	UA_{des} (W·K^{-1})	UA_{sub} (W·K^{-1})
reference	6000	6000	0	0
+ desuperheater	5000	6000	1000	0
+ subcooler	5000	6000	0	1000
+ desuperheater and subcooler	4000	6000	1000	1000

Reference system

The working diagram of the reference system in shown in Fig. 9a. A type340 [10] with a volume of 6 m^3 was used for the hot water tank (HWT) simulation. The *UA* value of the internal HX is 3.4 kW·K^{-1} and has height of 2.5 m. The temperature of the incoming cold water is 10 °C. The HP is connected to the hot water tank by an inlet and outlet connection to 10 % and 90 % of its height, respectively. The position of the temperature sensor T1 is at 60 % of the height. The system of the measurement and regulation (MaR) controls operation of the HP and the circulation pump (P1) by measuring temperatures T1 and T2. In the case of hot water, overheating is measured on the temperature sensor T3 at the outlet of the HWT, the hot water is mixed with cold water. If the hot water does not reach the demanded temperature of the DHW, the electric boiler (EB) heats the water to the demanded 55 °C. The circulation pump (CP) works nonstop and its power input is excluded from the *SPF* calculation. Heat losses of the circulation system are included in the heating demand because they are indifferent to the heat source. Heat losses are simulated by a tube model. The length of the tube is 335 m. The ambient temperature around the tube varies from 10 °C to 40 °C during a year. The circulation return pipe leads the water both to the DHW pipe and the HWT according to temperature T4 to improve the temperature stratification in the HWT.

System with a desuperheater

The working diagram of the system is shown in Fig. 9b. The HP is connected to the HWT via two internal HXs. The condenser heats the two lower thirds of the HWT until the demanded temperature (40 °C) at T1 is reached. The desuperheater heats the upper part of the HWT until the HP is ON or the temperature at T5 is higher than 57 °C. The three-way valve V1 controls the water flow rate to the desuperheater so that temperature of water at the outlet from desuperheater is 60 °C. The piping connection between the condenser and the desuperheater ensures the minimal condensation of the refrigerant in the desuperheater. It is assumed that the water enters the desuperheater with a temperature approximately equal to the condensing temperature. The positions of T1 and T5 are at 40 % and 75 % of the HWT height, respectively. The *UA* values of the internal heat exchangers are divided in the same ratio as the *UA* values in Table 4 from the *UA* value of the reference system. The same rule is valid for the systems with the subcooler and both the subcooler and the desuperheater.

System with a subcooler

The working diagram of the system with a subcooler is shown in Fig. 9c. The subcooler preheats the lower third of the HWT. The condenser heats the upper two-thirds of the HWT. The subcooler and the circulation pump P2 always work when the HP is ON. The sensor T1 is positioned at 80 % of the HWT height.

System with a desuperheater and subcooler

A hydraulic connection combines previous cases. The subcooler heats the upper quarter of the HWT, the condenser the middle half and the subcooler the lower quarter of the HWT. The working diagram is shown in Fig. 9.d. The logic of the control is similar to the previous cases.

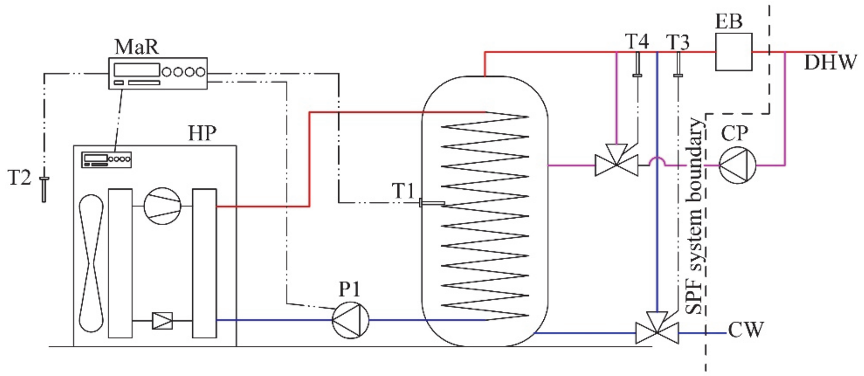

Figure 9a Reference system working diagram

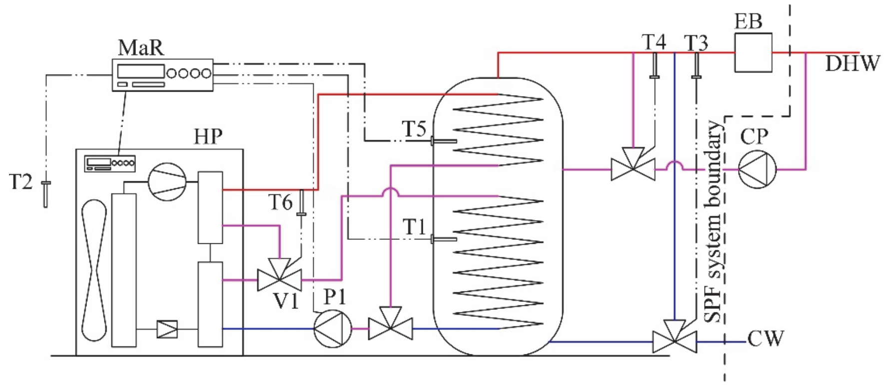

Figure 9b Working diagram of the HP with a desuperheater

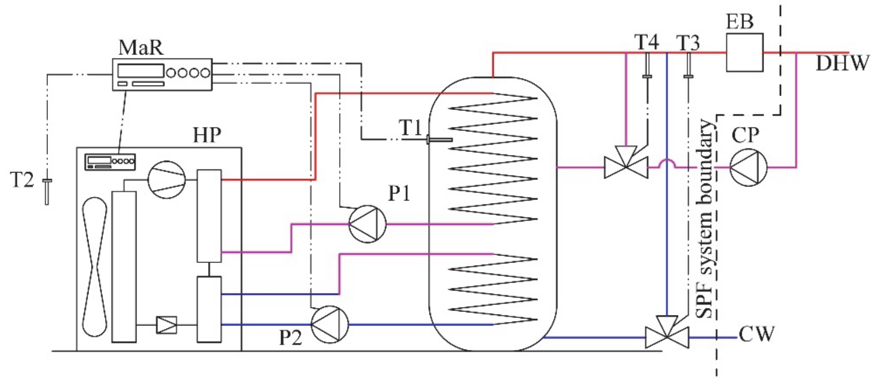

Figure 9c Working diagram of the HP with a subcooler

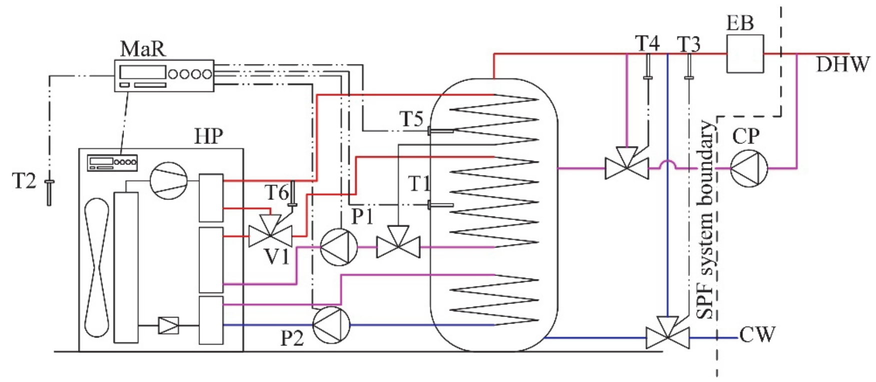

Figure 9d Working diagram of the HP with a desuperheater and a subcooler

RESULTS AND DISCUSSION

The summary of the results from the simulations for each described system is shown in Table 5. The heat consumption DHW system is 56.5 MWh and approximately 92 MWh together with circulation losses for all systems. The *SPF* of reference system is 2.55. The results of references are close to the measured data. The reference simulation, therefore, sufficiently describes the real behaviour of the system.

The *SPF* of the system with the desuperheater is 2.7, which is 6 % higher than the reference. The DHW circulation system, which is connected to the HWT, negatively influences the temperature stratification, therefore, the advantage of the desuperheater, which is in charge of the upper part of the HWT, is diminished.

The *SPF* of the system with the subcooler is 3.01, which is 20 % higher than the reference. The subcooler helps to preheat water for the condenser and the efficiency of subcooler grows with higher condensing temperatures.

The most complicated system with the desuperheater and the subcooler gives the best results. The *SPF* of the system is 3.17 which is 26 % higher than the reference. This option is comprised of the advantages of both previous systems.

The monthly results of the simulation of the system with the desuperheater and the subcooler are shown in Figure 10. The monthly *SPF* is a function of the ambient temperature and hot water consumption. In August, with a significant downswing of the DHW consumption, the *SPF* of the system decreases. The behaviour results from more significant losses in circulation. The heat supplied by the subcooler is stable during the year, contrary, the heat from the desuperheater varies more significantly with the ambient temperature.

Similar results as seen in Table 5 can be obtained with different heat exchangers. The main benefit of multiple HX heat pumps is in the better temperature distribution to the HWT. The size of the HX in the heat pump slightly influence the heat transfer which can result in a different *COP* of the HP. The *SPF* system is more dependent on the temperature distribution and the temperature stratification of the water inside the HWT.

CONCLUSION

The study mainly presented a possible increase in energy efficiency with a HP for the DHW application in a block of flats. The study is focused on the desuperheater and subcooler usage in such applications. The results proved the possibility to improve the *SPF* of the DHW system by 26 %.

The model of the air source HP for simulation is based on the ground source HP model (presented in the chapter Model validation) and the parametrisation of the data sheet values. The limitations of the air source HP model are unknown in frosting and defrosting behaviour of the HP on site. The correction function characterises ideal laboratory conditions and proved the described manufacturers data well.

The model of the ground source HP with the desuperheater and subcooler presented in the chapter The mathematical model of the heat pump describes the behaviour of the measured HP. However, the model limitation is in the assumptions (Section 4). The assumption of the constant *UA* of the HX, which is valid just in limited range of operating conditions, is also problematic.

The other limitation of the model of the DHW system as a whole is in the model of the HWT. The quality of the model and parameterisation was not tested on site. The deviation can negatively influence the error between the simulated and real *SPF* of the system. However, the same model of the HWT was used for all simulations (the only deviation was in the position of the sensors and internal HX), therefore, the ratios between the *SPF* of the systems should not vary.

Future research will focus on the connection between the HWT and HP on the impact of changing the relative positions of the sensors and the inlet and outlet port positions. The model of the HP can be improved by modelling a change of the *UA* value and implementing a compressor with capacity modulation. The system simulation should be tested on real applications via long-term measurements.

Table 5 Main results of the simulation

	Heat losses		HP					Circulation pumps	Electric boiler	*SPF*
	Circulation	HWT	Q_{con}	Q_{des}	Q_{sub}	P	*COP*			
	(MWh)	(MWh)	(MWh)	(MWh)	(MWh)	(MWh)	(-)	(MWh)	(MWh)	(-)
Reference system	35.6	0.24	90.5	-	-	35.3	2.57	0.23	0.56	2.55
HP with the desuperheater	36.0	0.25	58.5	32.9	-	32.4	2.82	0.24	1.39	2.70
HP with the subcooler	35.6	0.20	70.0	-	19.89	29.8	3.17	0.22	0.32	3.01
HP with the desuperheater and subcooler	35.9	0.25	44.15	27.15	21.48	28.75	3.23	0.22	0.19	3.18

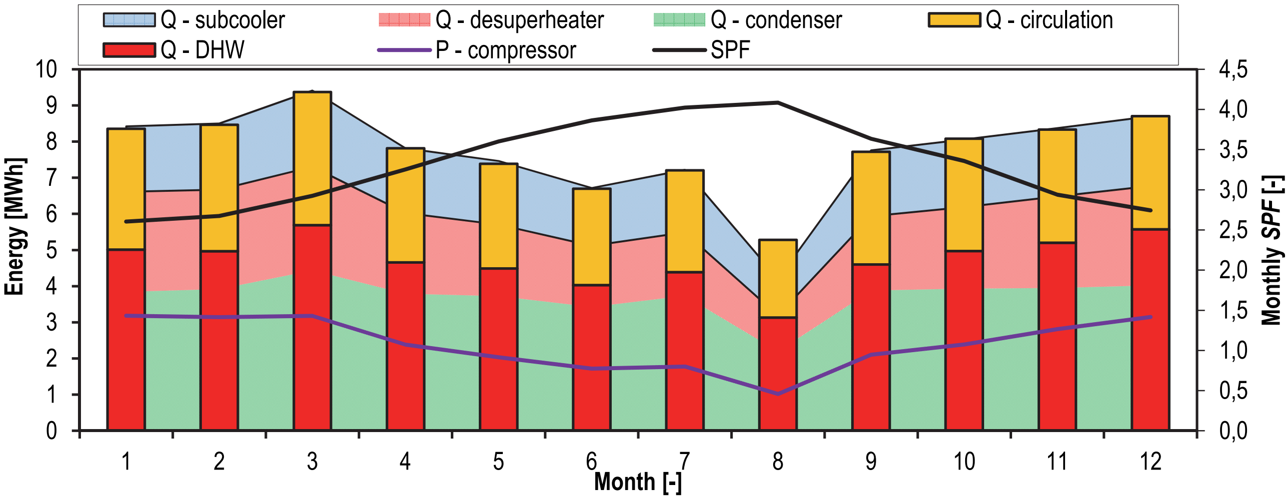

Figure 10 Monthly results of the simulation of the system with the desuperheater and the subcooler

ACKNOWLEDGMENT

This work has been supported by a project of the University Centre for Energy Efficient Buildings.

REFERENCES

[1] NOWAK, T. European Heat Pump Market and Statistics Report 2016. European Heat Pump Association (EHPA) 2016; Available at: http://www.ehpa.org/homepage/?eID¼dam_frontend_push&docID¼1930.

[2] Progress on energy efficiency in Europe. European Environment Agency (EEA) 2015; Available at http://www.eea.europa.eu/data-and-maps/indicators/progress-on-energy-efficiency-in-europe-2.

[3] EN 16147 Heat pumps with electrically driven compressors - Testing and requirements for marking of domestic hot water units. Brussel: CEN, 2011.

[4] Vyhodnocení cen tepelné energie k 1.1.2015. Energy Regulatory Office 2015; Available at: https://www.eru.cz/cs/teplo/statistika/vyhodnoceni-cen-tepelne-energie.

[5] Transient System Simulation Tool - TRNSYS 17.1 (2014), The University of Wisconsin, Madison, http://sel.me.wisc.edu/trnsys.

[6] ZAKULA, T., GAYESKI, N.T., ARMSTRONG, P.R., KEITH, L. Variable-speed heat pump model for wide range of cooling conditions and loads. HVAC&R Research 2011; 17:5, p. 670–691.

[7] PENG, D.Y., ROBINSON, D.B. New Two Constant Equation of State. Ind. Eng. Chem. 1976; 15:1, p. 59–63.

[8] SHANG, Y., WU, A., FANG, X. A study on modeling of minimal stable superheat for variable speed refrigeration system. International Journal of Refrigeration 2015; 59, p. 182–189.

[9] ERICSON, L. Rating Equations for Positive Displacement Compressors with Auxiliary Suction Ports. International Compressor Engineering Conference 1998, http://docs.lib.purdue.edu/icec/1282.

[10] DRUECK, H. Multiport Store Model for TRNSYS - Type 340. Transsolar GmbH 2006; Germany.

NOMENCLATURE

A	heat transfer area [m^2]
C	constant [-]
COP	performance factor [-]
$c_{p,liq}$	specific capacity of secondary fluid [J·kg^{-1}·K^{-1}]
D_1 - D_4	constants [-]
D_5	constant [Pa]
E_1, E_5	constants [-]
E_2	constant [°C^{-3}]
E_3	constant [°C^{-2}]
E_4	constant [°C^{-1}]
$h_{des,out}$	specific enthalpy at the outlet of desuperheater [J·kg^{-1}]
$h_{ref,in}$	specific enthalpy of refrigerant at inlet [J·kg^{-1}]
$h_{ref,out}$	specific enthalpy of refrigerant at outlet [J·kg^{-1}]
m_{liq}	mass flow rate of secondary fluid [kg·s^{-1}]
m_{ref}	refrigerant mass flow rate [kg·s^{-1}]
n	compressor rotational speed [s^{-1}]
n_{pol}	average exponent of isentropic compression [-]
p_{con}	condensing pressure [Pa]
P	electric power input [W]
Q_{con}	heat capacity of condenser [W]
Q_{des}	het capacity of desuperheater [W]
Q_{heat}	heating capacity [W]
$Q_{heat,total}$	heat capacity of high pressure side [W]
$Q_{heating,GS}$	heat capacity of ground source heat pump [W]
$Q_{hx,1}$	heat exchange rate of refrigerant [W]
$Q_{hx,2}$	heat exchange rate of secondary fluid [W]
$Q_{hx,3}$	heat transfer through HX area [W]
Q_{sub}	heat capacity of subcooler [W]
SPF	seasonal performance factor [-]
t_{con}	ambient temperature [°C]
t_{con}	condensing temperature [°C]
t_{ev}	evaporating temperature [°C]
$t_{liq,in}$	temperature of secondary fluid at inlet [°C]
$t_{liq,out}$	temperature of secondary fluid at outlet [°C]
$t_{s,out}$	temperature at the outlet of subcooler [°C]
U	overall heat transfer coefficient [W·m^{-2}·K^{-1}]
V_{sw}	swept volume [m^3]
δ_{ln}	logarithmic mean temperature difference [K]
η_{ie}	isentropic efficiency [-]
λ_v	volumetric efficiency [-]
ξ	correction value [-]
$\rho_{ref,s}$	refrigerant density in the compressors suction [kg·m^{-3}]
σ	pressure ratio [-]
Φ	polytropic correction [-]