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Heat pump for water preparation in block of flats

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Abstract

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

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Keywords: subcooler; desuperheater; DHW;

1. Introduction

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

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

This paper discusses the possibility of increasing efficiency of the system with 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 building over one year period. The paper includes basic HP with desuperheater and subcooler model and its validation. Verified model of HP was used in TRNSYS [5] simulation software with a model of DHW preparation.

2. The mathematical model of heat pump

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

2.1. Model assumptions

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

- the *UA* value of each heat (HX) is stable and constant
- the stable degree of superheating in the evaporator
- the stable degree of subcooling in the condenser
- pressure losses are negligible
- there are no heat losses from refrigerant cycle
- expansion in expansion valve is adiabatic
- oil do not influence refrigerant cycle and heat transfer

The assumption of stable *UA* value in each HX is problematic. *UA* value varies with flow rates, temperatures and other properties on both refrigerant and water/brine side of HX. However, small change does not influence significantly overall results. Model of heat transfer in every HX would make a simulation of a system with HP much slower due to an iterative process of evaluation of heat transfer rates in two-phase regions in the evaporator, condenser, and desuperheater.

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

2.2. Model of compressor

A simplified model of the real compressor is derived from the theory of piston compressors. It describes the mass flow of refrigerant and power input of compressor in every working condition and can be derived from manufacturer's data sheet. The refrigerant mass flow rate $\dot{m}_{ref}$ is described as follows:

$$\dot{m}_{ref} = V_{sw} \cdot \lambda_v \cdot \rho_{ref,s} \cdot n \tag{1}$$

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

$$\lambda_v = 1 - C \cdot (\sigma - 1) \tag{2}$$

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

$$\eta_{ie} = D_1 + D_2 \cdot \phi + D_3 \cdot \phi^2 + D_4 \cdot \phi^3 + D_5 \cdot p_{con} \tag{3}$$

$$\phi = \sigma^{1/n_{pol}} \tag{4}$$

In above equation, p_{con} is condensing pressure, D_1 to D_5 are constants for parametrization, and ϕ is defined in equation (4), where n_{pol} is an average exponent of isentropic compression. Model of compressor calculates pressure ratio, isentropic and volumetric efficiency accordingly to pressure at suction and discharge.

2.3. Model of heat exchangers

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

$$\dot{Q}_{hx,1} = \dot{m}_{ref} \cdot abs\left(h_{ref,in} - h_{ref,out}\right) \tag{5}$$

$$\dot{Q}_{hx,2} = \dot{m}_{liq} \cdot c_{p,liq} \cdot abs\left(t_{liq,out} - t_{liq,in}\right) \tag{6}$$

$$\dot{Q}_{hx,3} = UA \cdot \delta_{\ln} \tag{7}$$

In above equations, $\dot{Q}_{hx,1}$, $\dot{Q}_{hx,2}$ and $\dot{Q}_{hx,3}$ are heat exchange rates on refrigerant side, secondary side and heat transfer through heat exchanger surface area rate respectively. $h_{ref,in}$ and $h_{ref,out}$ are specific enthalpies of refrigerant at inlet and outlet of section, $c_{p,liq}$ is specific capacity of secondary liquid, $\dot{m}_{liq}$ is mass flow rate of secondary liquid, $t_{lig,in}$ and $t_{liq,out}$ are temperatures of secondary side liquid at inlet and outlet of section, δ_{ln} is logarithmic mean temperature difference, U is heat transfer coefficient, is heat transfer area. The value of *UA* can be found in data sheets of HX. *UA* value does not change in simulations. Heat pump model consists 4 HX–condenser, evaporator, desuperheater, and subcooler. XH are modeled in sections in accordance with expected phase condition on each side. Model of each HX is divided into:

- evaporator - two sections
- condenser - 3 or 2 sections (depends on desuperheater)
- desuperheater - 1 or 2 sections
- subcooler - 1 section

2.4. The operating parameters of heat pump

Every heat pump works with some refrigerant. Thermodynamics properties are calculated in every time step with a mathematical model of refrigerant. Type of refrigerant is given at beginning of simulation and doesn't change, therefore it is the first parameter. Other parameters are set at beginning of simulation as well:

- The degree of superheating in evaporator
- The degree of subcooling in condenser

- The superheating in compressor suction piping
- The pressure loss in compressor suction
- An operating envelope of compressor

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

2.5. Cycle convergence algorithm

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

2.6. Model validation

The model of heat pump was tested on heat pump prototype. Schematic of the refrigerant cycle and test bench is in Fig. 1. The prototype contains scroll compressor, brazed plate HX and electronic expansion valve. List of main components and properties for the model validation is in Table 1. The prototype contains also the superheater which was implemented solely for testing and it is not included to validation. The prototype works with refrigerant R410a.

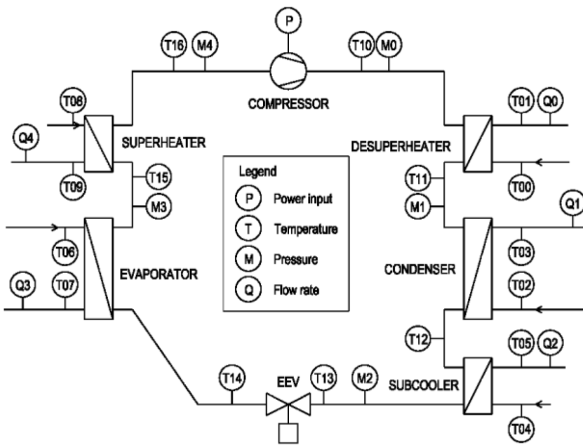


Fig. 1. Scheme of tested heat pump and tested values

Table 1. List of component used in tested prototype

Component		Value	Unit
Scroll compressor	V_{sw}	21.7	cm^3
	C	$2.22\cdot10^{-2}$	-
	D_1	$-2.31\cdot10^{-2}$	-
	D_2	$8.31\cdot10^{-1}$	-
	D_3	$-2.48\cdot10^{-1}$	-
	D_4	$2.22\cdot10^{-2}$	kPa^{-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 degree of superheating controlled by expansion valve was set on 4 K. average subcooling degree in condenser was 2 K. Results of model validation are in Table 2 and in Fig. 2, 3, 4 and 5. $\dot{Q}_{heat,total}$ is total heat transfer on the high-pressure side of HP. The average error of *COP* is less than 5 % if its value. The model fits measured data with good accuracy.

Table 2. List of errors of model from measured data

Working arrangement	*P* (%)		$\dot{Q}_{heat,total}$ (%)		*COP* (%)	
(on high-pressure side)	Average	Maximal	Average	Maximal	Average	Maximal
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
Condensed, 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

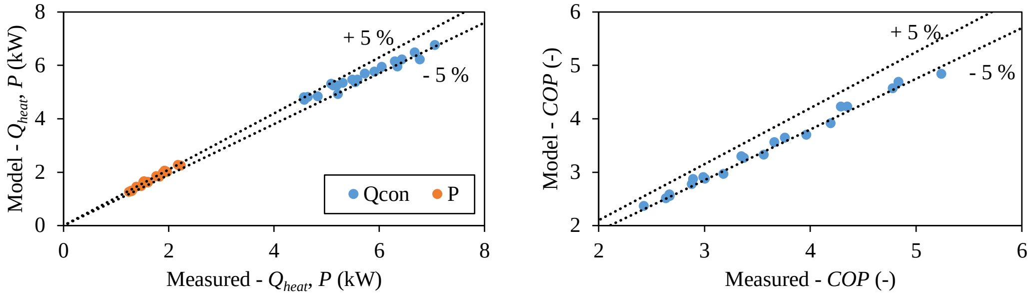


Fig. 2. Deviation between model and measured data for heating capacity, power input and *COP* in condenser arrangement

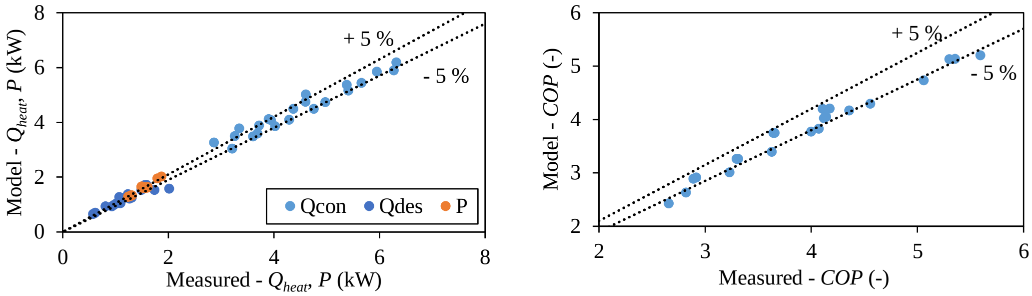


Fig. 3. Deviation between model and measured data for heating capacity, power input and *COP* in condenser + desuperheater arrangement

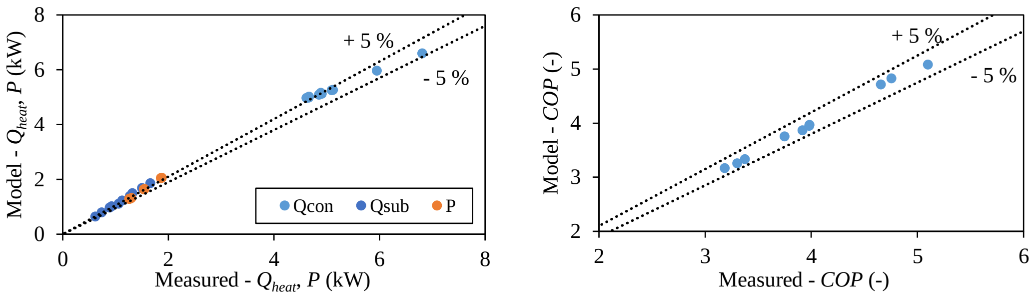


Fig. 4. Deviation between model and measured data for heating capacity, power input and *COP* in condenser + subcooler arrangement

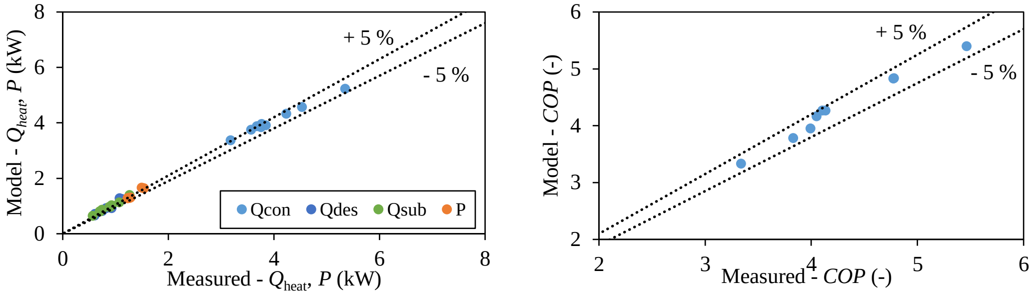


Fig. 5. Deviation between model and measured data for heating capacity, power input and *COP* in condenser + desuperheater + subcooler arrangement

3. Hot water preparation system in apartment building

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

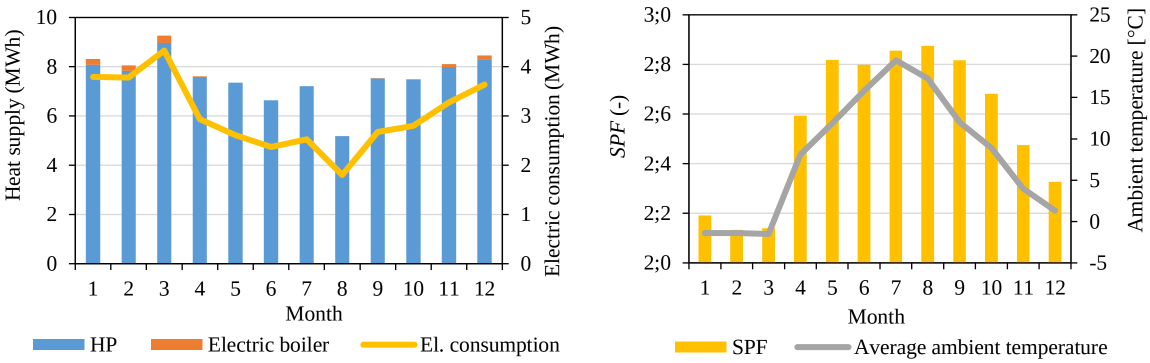


Fig. 6. Measured data of heat supply from heat pump and electric consumption (on left), calculated *SPF* of system and average ambient temperature (on right)

The measured system is in western part of the Czech Republic and is assembled with an air source HP. The nominal heating capacity of HP (A7/W35) is 20.2 kW. The backup heater is an electric boiler. Heat pump regulation turns off the unit when the ambient temperature is lower than -10 °C and whole energy consumption is then covered by an electric boiler. The block of flats has 48 housing units. Demanded temperature of DHW to circulation system is 55 °C. DHW circulation system is needed to ensure comfort for residents and works nonstop, however it is estimated that 40 % of energy is lost there. Accumulation of hot water is provided by 6 m^3 water tank. Hot water tapping profile was modeled with respect to standard EN 16147 [3] and a 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 function describing monthly profile was implemented to simulation.

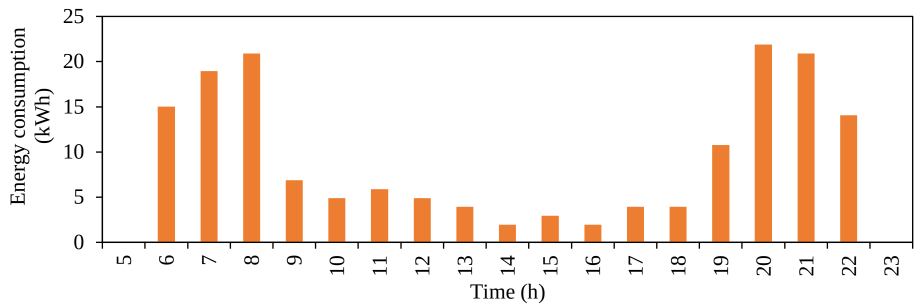


Fig. 7. Daily hot water tapping profile

The model of heat pump described in chapter 3 predicts the working behavior of ground source HP. To describe air source HP correction of heating capacity was done. Correction is based on the idealized working behavior of ground source HP. Relative deviation between ground source heat pump modeled data and data sheet values is a function of ambient temperature. Correction is well described by hyperbole function of a higher order:

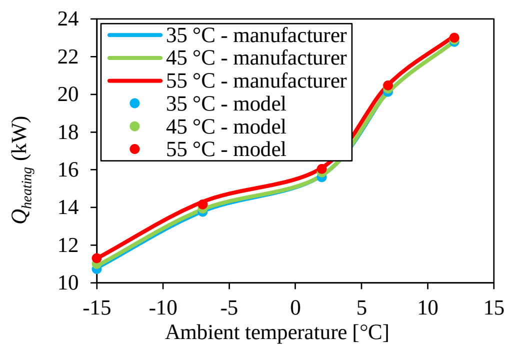
$$\xi = \frac{E_1}{t_{ev,in}^4 + E_2 \cdot t_{ev,in}^3 + E_3 \cdot t_{ev,in}^2 + E_4 \cdot t_{ev,in} + E_5} \tag{8}$$

The function which describes heating capacity is:

$$\dot{Q}_{heating} = \dot{Q}_{heating,GS} \cdot (1 - \xi) \tag{9}$$

In above equations, ξ is a correction factor, $t_{ev,in}$ is an air inlet temperature (equals to ambient temperature in the model), $\dot{Q}_{heating,GS}$ is heating capacity calculated by the ground source heat pump model, $\dot{Q}_{heating}$ is the heating capacity of HP after correction and E_1 to E_5 are unknown constants for parametrization.

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



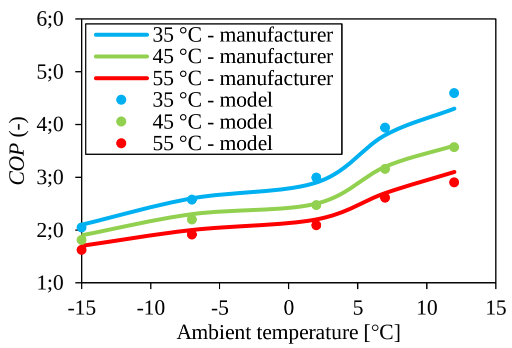


Fig. 8. Heating capacity and *COP* as function of ambient temperature

4. Simulation model of DHW system

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

Table 3. List of constants used in simulations

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

Table 4. List of *UA* values used in 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
with desuperheater	5000	6000	1000	0
with subcooler	5000	6000	0	1000
with desuperheater and subcooler	4000	6000	1000	1000

4.1. Reference system

The working scheme of reference system in on Fig. 9.a. For HWT simulation was used Type 340 [10] with volume 6 m^3. *UA* value of internal HX is 3.4 kW·K^{-1} and high 2.5 m. the temperature of the incoming cold water is 10 °C. HP is connected to the hot water tank (HWT) by inlet and outlet connection to 10 % and 90 % of its height respectively. The position of temperature sensor T1 is in 60 %. The system of measurement and regulation (MaR) controls operation of HP and circulation pump (CP1) by measuring temperatures T1 and T2. In the case of hot water overheating measured on temperature sensor T3 at the outlet of HWT, hot water is mixed with cold water. If hot water does not reach demanded temperature of DHW, electric boiler (EB) heats water on demanded 55 °C. Circulation pump CP2 works nonstop and its power input is excluded from *SPF* calculation. Heat losses of circulation system are included to heating demand because they are indifferent to the heat source. Heat losses are simulated by tube model. The length of the tube is 335 m. ambient temperature around tube varies from 10 to 40 °C during the year. Circulation return pipe direct water both to DHW pipe and HWS according to temperature T4 to improve temperature stratification in HWT.

4.2. System with desuperheater

The working scheme of the system is in Fig. 10.b. HP is connected to HWT via two internal HX. Condenser heats two lower thirds of HWT until demanded temperature (40 °C) on T1 is reached. Desuperheater heats upper part of HWT until HP is ON or temperature at T5 is higher than 57 °C. The three-way valve V1 controls water flow rate to desuperheater so that temperature of water at the outlet from desuperheater is 60 °C. The piping connection between condenser and desuperheater ensures minimal condensation of refrigerant in desuperheater. It is assumed that water enters desuperheater with temperature approximately equal to condensing temperature. Positions of T1 and T5 are in 40 % and 75 % of HWT height respectively. *UA* values of internal heat exchangers are divided in the same ratio as *UA* values in the Table 4 from *UA* value of the reference system. The same rule is valid for systems with subcooler and both subcooler and desuperheater.

4.3. System with subcooler

The working scheme of the system with subcooler is on Fig. 11.c. subcooler preheats lower third of HWT. Condenser heats upper two-thirds of WHT. Subcooler and circulation pump CP3 work always when HP is ON. The sensor T1 is positioned at 80 % of WHT height.

4.4. System with desuperheater and subcooler

The hydraulic connection combines previous cases. Subcooler heats an upper quarter of HWT, condenser middle half and subcooler lower quarter of HWT. Working scheme is in Fig. 9.d. the logic of control is similar to previous cases.

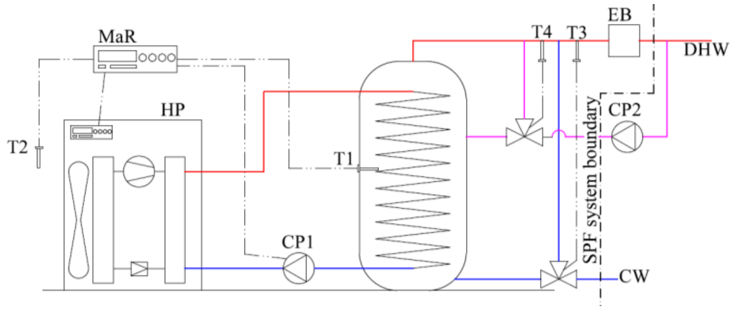


Fig. 9.a. Reference system working scheme

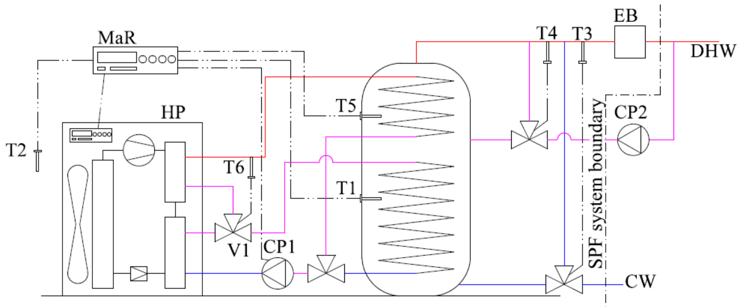


Fig. 9.b. Working scheme of HP with desuperheater

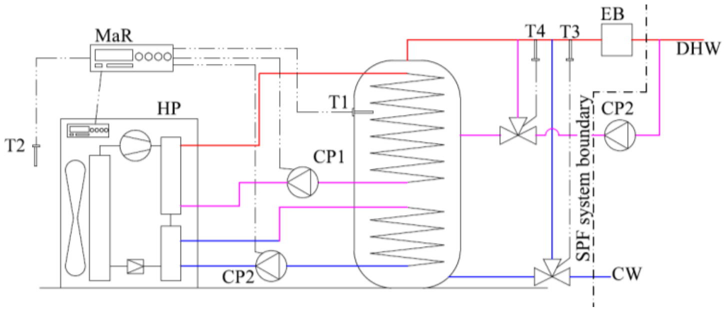


Fig. 9.c. Working scheme of HP with subcooler

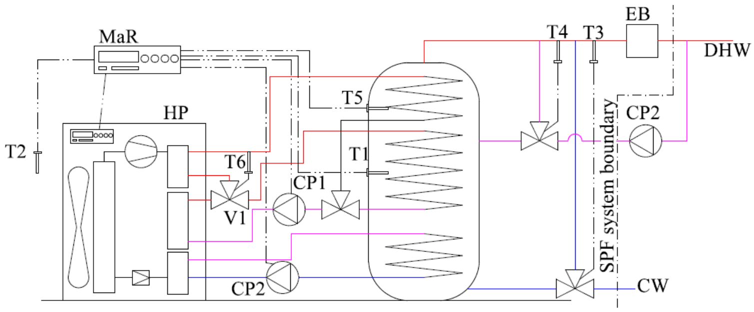


Fig. 9.d Working scheme of HP with desuperheater and subcooler

5. Results and discussion

The summary of results from simulations for each described system is in Table 5. Heat consumption of DHW system together with circulation losses is 92.3 MWh and *SPF* 2.55. Results are close to measured data. Reference simulation, therefore, describes sufficiently a real behavior of the system.

The *SPF* of the system with desuperheater is 2.64, which is 7 % higher than reference. DHW circulation system, which is connected to HWT, negatively influences temperature stratification, therefore an advantage of desuperheater, which is in charging of upper part of HWT, is diminished.

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

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

Table 5. Main results of simulation

	DHW heat consumption	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	MWh	-
Reference system	56.0	35.6	0.24	90.5	-	-	35.3	2.57	0.23	0.56	2.55
HP with desuperheater	56.6	35.8	0.26	60.7	31.4	-	32.4	2.76	0.24	1.39	2.64
HP with subcooler	55.8	35.6	0.20	70.0	-	19.89	29.8	3.17	0.22	0.32	3.01

HP with desuperheater and subcooler	56.7	35.9	0.25	44.15	27.15	21.48	28.75	3.23	0.22	0.19	3.18

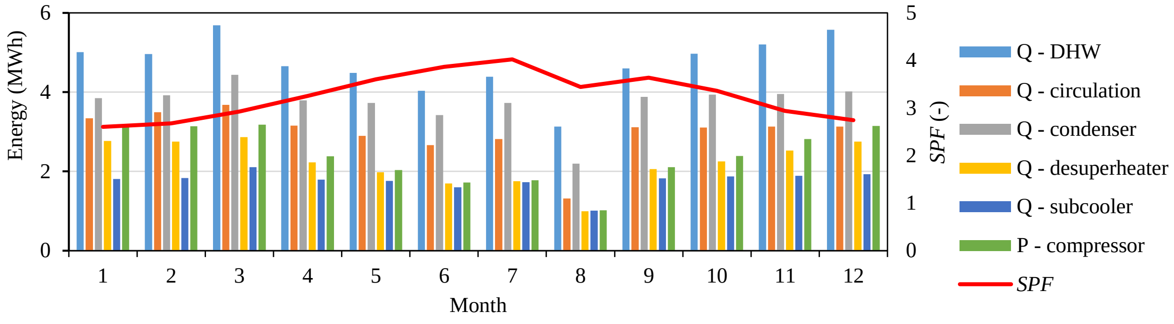


Fig. 10. Monthly results of simulation of system with desuperheater and subcooler

The monthly results of simulation of the system with desuperheater and subcooler is on Fig. 10. Monthly *SPF* is a function of ambient temperature and hot water consumption. In August with significant downswing of DHW consumption decreases *SPF* of the system. Behavior results from more significant losses of the system. Heat supplied by subcooler is stable during the year, contrary heat from desuperheater varies more significantly with ambient temperature.

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

6. Conclusion

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

The model of air source HP for simulation is based on ground source HP model (presented in chapter 3) and parametrization of data sheets values. Limitations of air source HP model are in unknown frosting and defrosting behavior of HP on site. Correction function characterizes ideal laboratory conditions and proved described well manufacturers data.

The model of ground source HP with desuperheater and subcooler presented in chapter 2 describes the behavior of measured HP. However, the model limitation is in assumptions (chapter 2.4.). Problematic is also the assumption of constant *UA* of HX, which is valid just in limited range of operating conditions.

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

The future research will focus on the connection between HWT and HP on the impact of changing relative positions of sensors and inlet and outlet ports positions. Model of HP can be improved by modeling change of *UA* value and implementing compressor with capacity modulation. System simulation should be tested or real application via long-term measurements.

Acknowledgement

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