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Integration of a compact two fluid PCM heat exchanger into the hot superheated section of an air source heat pump cycle for optimized DHW generation

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ABSTRACT

This contribution presents a concept for the direct integration of a refrigerant/water Heat Exchanger (HEX) with a Phase Change Material (PCM) in the hot superheated section of an R32 - air source compression Heat Pump (HP) cycle, for optimized Domestic Hot Water (DHW) generation in multi-family houses. The concept takes advantage of the PCMs high thermal storage capacity integrated into a high performance compact enhanced plate-and-fin aluminium HEX. On the refrigerant side, it works as a de-superheater for DHW generation during heating and cooling operation whereas the process water is connected to decentralized DHW storages located in single apartments of a low energy building. We present results from simulations at a system level for typical operating conditions and corresponding seasonal and annual performances. Compared to conventional systems, the results indicate savings up to 11% of electric energy over the year for DHW generation in average climate.

Keywords: domestic hot water, energy efficiency, heat pump, heating, hot gas, phase change material.

1. INTRODUCTION

Today's commercially available air source HPs work highly efficient and make a valuable contribution to reach climatic goals as the reduction of CO_2 in our atmosphere. To further increase the efficiency of HPs many different concepts using latent heat thermal energy storage technologies were proposed recently. The aim is to decrease the size of buffer tanks and to avoid the oversizing of HPs reducing thermal peak loads by shifting heating and cooling demands in time (Pardiñas *et al.*, 2017), to optimize integration of solar thermal collectors (Kapsalis *et al.*, 2016), to enhance the storage density of DHW storages (Zou *et al.*, 2017) or to enhanced the defrosting performance of air source HPs (Song *et al.*, 2018). Recent studies showed further that a wide spectrum of PCMs are available to be used in such storages at different temperature levels (Cabeza *et al.*, 2011).

This contribution employs latent heat thermal energy storage technology to increase the efficiency of air source HP systems in multi-family houses for DHW generation. The key idea is to optimally use the available exergy of the refrigerant during operation all over the year. This means that thermal energy at a high temperature level in the superheated section of the HP cycle is stored in a PCM to be used later for high temperature DHW generation rather than being used at intermediate temperatures for heating or even being wasted in the case of cooling operation. The optimal assignment of high temperature heat during conventional operation is realized by the direct integration of a refrigerant (R)/water (W) heat exchanger (HEX) with phase (P) change material – in short: RPW-HEX . The compact RPW-HEX is integrated in the hot refrigerant flow exiting the compressor of a compression HP on the primary side and in the process water flow on the secondary side. From the secondary side of the RPW-HEX, the process water delivers the thermal energy to decentralized DHW storages located in the individual apartments of a multi-family house. The RPW-HEX also manages flexible DHW demands and by this minimizes the operation of the air source HP in the highly non-efficient direct DHW operating mode. Furthermore, it adds the necessary flexibility by decoupling the high temperature high power energy streams (for DHW generation) and the low temperature and low power energy stream (for heating). To guarantee high heat transfer coefficients between the hot gaseous refrigerant and the PCM, between the PCM and the process water and between the refrigerant and the process water, with a minimum of refrigerant mass, a plate and fin aluminium heat exchanger design is used. The present study answers the question, how much of the heat can be stored in the RPW-HEX and therefore delivered to the decentralized DHW storages in a typical application. Furthermore, possible savings in electric energy are quantified and compared to a reference system without the RPW-HEX. For doing so, simulation studies at selected operating points were carried out. Annual efficiencies for the proposed system and the reference system without RPW-HEX were calculated by summing up steady states over one year on an hourly basis.

2. CASE STUDY AND OPERATION MODES

The case study considers an application with three apartments located in average climate conditions in Europe and an air source HP. The main purpose is heating in winter and DHW generation all over the year. Through switching into reverse mode, cooling during summer is also possible.

The proposed system distinguishes between six operating modes (a-f) whilst a conventional system distinguishes between mainly three operating modes, namely heating, cooling and DHW generation which have similar efficiencies as (d,e,f) in the proposed system,

(a) heating operation and charging the RPW-HEX ($0 < SoC \uparrow \leq 1$)
(b) cooling operation and charging the RPW-HEX ($0 < SoC \uparrow \leq 1$)
(c) energy efficient DHW generation by discharging the RPW-HEX ($0 \leq SoC \downarrow \leq 1$)
(d) conventional (inefficient) direct DHW generation ($SoC = 0$)
(e) heating operation when the RPW-HEX is fully charged ($SoC = 1$)
(f) cooling operation when the RPW-HEX is fully charged ($SoC = 1$)

where SoC refers to the state of charge of the RPW-HEX (Barz *et al.*,2018). Figure 1 (a)-(d) show the operation modes (a)-(d) in a system sketch, where the conventional operation modes (e), (f) are omitted for the sake of brevity. The HP is connected to the apartments with two hydraulic lines and the switching between charging the decentralized DHW storages and heating/cooling takes place in the apartments. During heating mode (a), most of the sensible energy in the superheated hot gas charges the RPW-HEX (R). The amount of transferred heat depends on the hot gas temperature, the refrigerant mass flow, the phase transition temperature of the (solid/liquid) PCM, and the overall heat transfer coefficients of the RPW-HEX. Hot gas temperature and refrigerant mass flow result from the operating point whereas the switching temperature range of the PCM and the heat transfer capabilities of the RPW-HEX are design inherent (design parameters). For the proposed concept, R32 as a refrigerant was used and Rubitherm PCM RT64HC as PCM with a phase change temperature of 64℃ and a narrow phase transition temperature range was selected (Rubitherm, 2019). R32 has a rather high hot gas temperature when compared to other refrigerants. To limit the compressor discharge temperature at low temperatues, liquid refrigerant from the condenser exit is injected into the compressor (which is able to handle a small amount of liquid refrigerant) entrance by means of a liquid injection valve (B). In doing so, the hot gas temperature can be limited to ≈115℃. Furthermore, an additional HEX (F) is introduced to ensure that the refrigerant is in the liquid phase at the expansion valve (X) entry. Via the condenser (C), the heat is delivered to the apartments by the heating distribution system (H). In cooling mode (b), the four-way valve (W) switches to reverse mode and the evaporator (E) acts as a heat sink, whereas the condenser (C) cools the building via the heating and cooling network (H). Because the four-way valve (W) is located after the RPW-HEX, the PCM is also charged by the hot gas during cooling operation. Note that contrary to heating mode, where it would also be possible to use the energy stored in the PCM for heating (of course with a low exergy efficiency), the energy stored in the RPW-HEX during cooling is usually not used in a conventional system. Once, the RPW-HEX is fully charged (SoC =1) by mode (a) or (b), the HP system switches to DHW generation mode (c). Contrary to direct DHW-generation mode (d), the condensing temperature, and therefore the COP, remains at the values for heating mode in (c). Therefore, the process water from the return line of the decentralized DHW storages (S) is pre-heated by the condenser at a beneficial COP. Subsequently the process water is boosted to the DHW set-point temperature ($\vartheta_w \approx 62$ °C) by discharging the RPW-HEX (R). During this operation mode, additionally the hot gas will transfer its sensible energy via the RPW-HEX (R) to the process water.

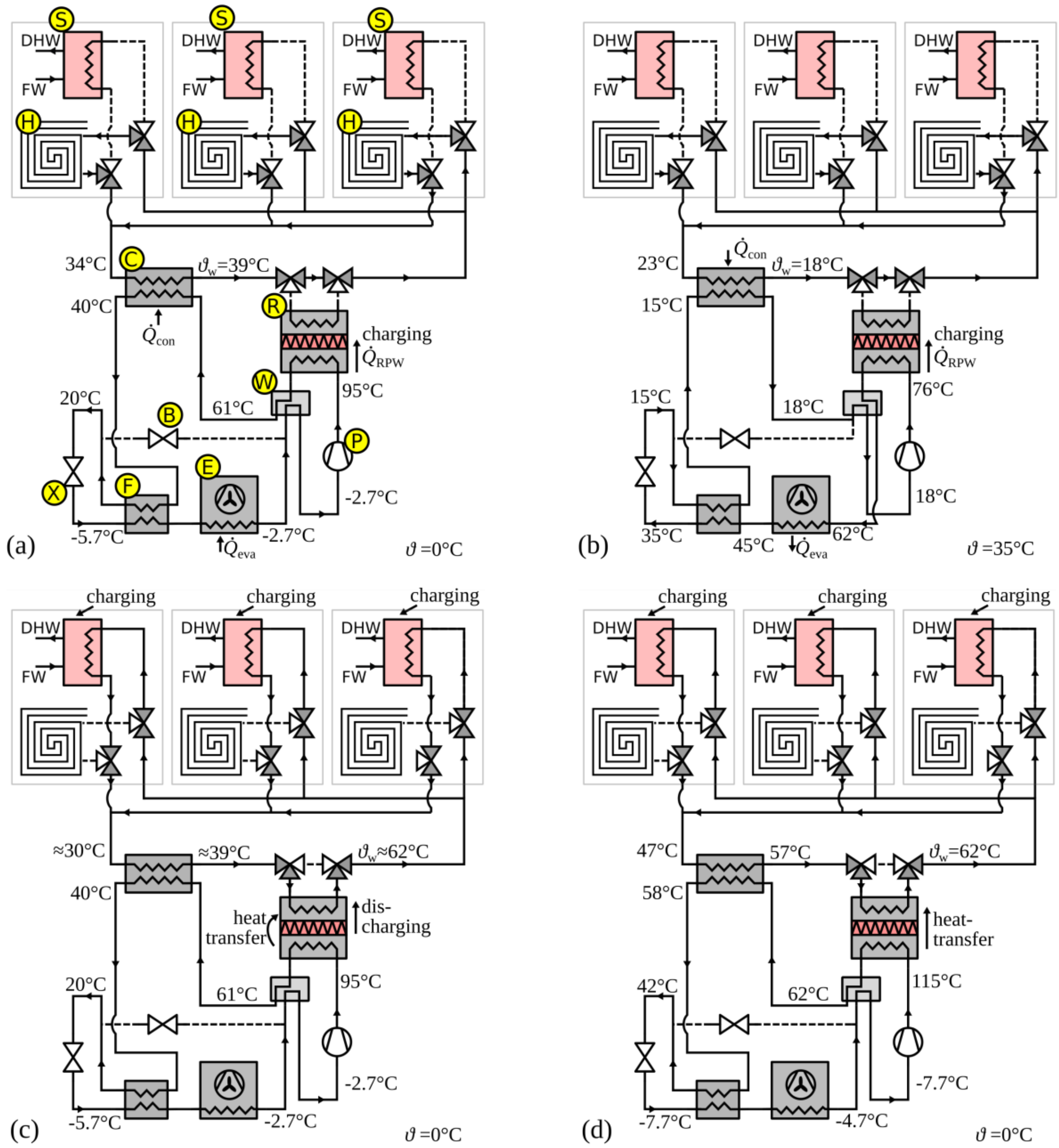

Figure 1: Concept of the proposed system for a scenario with three apartments during: (a) heating operation, (b) cooling operation, (c) energy efficient DHW generation by discharging the RPW-HEX and (d) direct DHW. (S) Decentralized sensible DHW storage, (H) low- or intermediate temperature heating system, (C) condenser, (R) RPW-HEX, (W) four way valve, (B) bypass expansion valve, (X) regular expansion valve, (F) fluid phase heat exchanger, (E) evaporator and fan, (P) compressor. The temperatures indicated in (a),(b) and (d) were taken from steady-states at an SoC of 20% and 50% for heating and cooling, respectively, whereas they were taken shortly after switching from mode (a) to (c) in (c). The ambient temperature (ϑ) was 0°C in (a,c and d) and 35°C in (b).

Once the RPW-HEX is discharged, the HP switches back to heating or cooling mode, respectively. In the case that the thermal energy for DHW generation cannot be provided entirely by operating mode (c), e.g. during spring, the HP has to switch to the direct DHW generation mode (d). In this mode, similar to a conventional system, DHW has to be generated by increasing the condensing temperature to values suitable for DHW. The RPW-HEX acts in this case solely as HEX and transfers the sensible energy of the hot gas directly to process water. In operating mode (e) and (f) the RPW-HEX is fully charged and the modes are comparable to conventional heating and cooling modes.

3. METHODOLOGY

3.2 Simulation models

Simulation studies were carried out for both, the novel system and the reference system without RPW-HEX. The following performance indicators were computed to calculate the annual efficiencies: Coefficient of Performance related to the hot side of the HP for heating and cooling: COP_h, Coefficient of Performance related to the hot side of the HP for DHW: COP_{DHW} and RPW-HEX utilization factor: ε_{RPW}. The simulations were carried out in the Dymola/Modelica modelling environment using ThermoCycle library components (Quoilin *et al*., 2014). Additionally, models for the RPW-HEX, the outdoor unit and the four way valve were developed in.house (Emhofer *et al.*, 2018, Frazzica *et al.*, 2018). Thermodynamic properties were taken from the CoolProp library (Bell *et al*., 2014). The performance indicators were derived from the dynamic simulation when the system is in steady-state or when the SoC reached 20% or 50% for heating and cooling, respectively. Defrosting operation was neglected, and electric power consumption for COP calculations solely reflects the consumption of the compressor and the fan of the outdoor unit. All HP geometry/component design parameters, efficiencies and heat transfer coefficients were taken from design sheets or calculated from well-established equations and were later experimentally validated with measurements of the prototype air source HP used in the H2020 project HYBUILD (HYBUILD, 2017) without RPW-HEX.

3.2 Annual Energy Efficiency Calculations

A quasi-static approach to estimate the annual energy demand of the systems was adopted. Performance parameters at different ambient temperature levels were considered for heating, cooling and DHW generation mode (a), (b) and (d) on an hourly basis over one year or 8760 hours. Please note, that pre-heating of the process water with the condenser in operation mode (c) is not considered in the following because this process is highly dynamic and can’t be handled with this quasi-static approach. Hence, the entire energy for energy efficient DHW generation is always taken from the RPW-HEX. The assumed scenario is defined by ambient temperatures (ϑ) from Strasbourg in an hourly resolution (Meteonorm, 2016). A function for the feed water temperature to the heating distribution system (ϑ_w) see Eq. (1), was fitted to the ambient temperatures as defined in the EN14825 (2016), using variable outlet temperatures for heating and for cooling ceiling applications:

$$\vartheta_{\mathrm{w}}(\vartheta) = \begin{cases} -0.577 \times \vartheta + 39.1\,^{\circ}\mathrm{C}, & \vartheta \leq 2^{\circ}\mathrm{C} \\ -1 \times \vartheta + 40\,^{\circ}\mathrm{C}, & 2^{\circ}\mathrm{C} \leq \vartheta \leq \vartheta_{\mathrm{on,h}} = 16^{\circ}\mathrm{C} \\ 18\,^{\circ}\mathrm{C}, & \vartheta \geq \vartheta_{\mathrm{on,c}} = 20^{\circ}\mathrm{C} \end{cases} \qquad \text{Eq. (1)}$$

Note that heating starts below $\vartheta_{on,h}$ = 16 ℃ and cooling starts above $\vartheta_{on,c}$ = 20 ℃, respectively. Because the condenser is connected to the heating and cooling distribution system (c.f. Fig. 1 (a,b)), $\dot{Q}_{con}$ is used to describe the heating and cooling demand of the building which needs to be covered by the HP at all times. The design point of the HP is defined by a heating demand of $\dot{Q}_{con} = 2$ kW per apartment at ϑ=$\vartheta_{design,h}$=-10℃, typical for a low energy building. As the main purpose of the system is heating, the design cooling load ($\dot{Q}_{con}(\vartheta_{design,c})$) is calculated from the definitions from heating by considering solely heat conduction between the apartment and the ambient air while neglecting solar radiation. This approach will underestimate the cooling demand but was used to be able to work without a detailed building model. The design cooling load is given in Eq. (2):

$$\dot{Q}_{\mathrm{con}}(\vartheta_{\mathrm{design,c}}) = \mathrm{UA}\,(20^{\circ}\mathrm{C} - \vartheta_{\mathrm{design,c}}) = \frac{\dot{Q}_{\mathrm{con}}(\vartheta_{\mathrm{design,h}})}{\vartheta_{\mathrm{on,h}} - \vartheta_{\mathrm{design,h}}}\,(\vartheta_{\mathrm{on,c}} - \vartheta_{\mathrm{design,c}}) \qquad \text{Eq. (2)}$$

where $\vartheta_{design,c}$=35 °C. With Eq. (2) one finds a cooling load of 1.15 kW per apartment at 35 °C. The part load behaviour $\dot{Q}_{con}(\vartheta)$ is obtained assuming a linear dependence between $\vartheta_{on,h}$ and $\vartheta_{design,h}$, and between $\vartheta_{on,c}$ and $\vartheta_{design,c}$, respectively (EN14825, 2016), as shown in Eq. (3):

$$\dot{Q}_{con}(\vartheta) = \begin{cases} \frac{\vartheta - \vartheta_{\mathrm{on,h}}}{\vartheta_{\mathrm{design,h}} - \vartheta_{\mathrm{on,h}}} \times \dot{Q}_{\mathrm{con}}(\vartheta_{\mathrm{design,h}}), & \mathrm{heating}, \vartheta \leq 16°\mathrm{C} \\ \frac{\vartheta - \vartheta_{on,c}}{\vartheta_{\mathrm{design,c}} - \vartheta_{\mathrm{on,c}}} \times \dot{Q}_{\mathrm{con}}(\vartheta_{\mathrm{design,c}}), & \mathrm{cooling}, \vartheta \geq 20°\mathrm{C} \end{cases} \qquad \text{Eq. (3)}$$

The amount of heat transferred to the RPW-HEX $\dot{Q}_{RPW}$, is calculated using a utilization factor ε_{RPW} which relates to the total thermal energy transferred at the hot side $\dot{Q}_{hot}(\vartheta)$ as shown in Eq. (4):

$$\dot{Q}_{\mathrm{RPW}}(\vartheta) = \varepsilon_{\mathrm{RPW}}(\vartheta)\, \dot{Q}_{\mathrm{hot}}(\vartheta) \qquad \text{Eq. (4)}$$

where the heat transferred on the hot and cold side of the heat pump are given by Eq.(5) and Eq.(6):

$$\dot{Q}_{\mathrm{hot}}(\vartheta) = \begin{cases} \dot{Q}_{\mathrm{con}} + \dot{Q}_{\mathrm{RPW}}, & \text{heating, DHW generation} \\ \dot{Q}_{\mathrm{eva}} + \dot{Q}_{\mathrm{RPW}}, & \text{cooling} \end{cases} \qquad \text{Eq. (5)}$$

$$\dot{Q}_{\mathrm{cold}}(\vartheta) = \begin{cases} \dot{Q}_{\mathrm{eva}}, & \text{heating , DHW generation} \\ \dot{Q}_{\mathrm{con}}, & \text{cooling} \end{cases} \qquad \text{Eq. (6)}$$

Note that ε_{RPW} in Eq. (4) strongly depends on the temperature conditions. In the reference system ε_{RPW} is zero and therefore, the entire thermal energy on the hot side (Eq. (5)) is transferred to the condenser during heating or to the evaporator during cooling, respectively.

Furthermore a daily DHW consumption of Q^d_{DHW} =5.845 kWh per apartment is assumed, which is comparable to a medium water consumption as defined in the EN16147 (2017). The load profile is also taken from the standard, and it is assumed that the energy stored in the RPW-HEX can always be delivered to the decentralized water storages without thermal losses in space and time.

The annual energy efficiency ratios (EER) for heating and cooling are calculated by means of Eq. (7) and Eq. (8), similarly to the basic definitions for the $SCOP_h$ and $SCOP_c$ in the EN14825 (2016):

$$EER_{\mathrm{heating}} = \frac{Q^y_{\mathrm{heating}}}{W^{\mathrm{y}}_{\mathrm{heating}}} = \frac{\sum_{j=1}^{8760} Q_{con}(\vartheta_j)}{\sum_{j=1}^{8760} \left(\frac{Q_{con}(\vartheta_j)}{COP_h(\vartheta_j)} \right)} \qquad \text{Eq. (7)}$$

$$EER_{\mathrm{cooling}} = \frac{Q^y_{\mathrm{cooling}}}{W^{\mathrm{y}}_{\mathrm{cooling}}} = \frac{\sum_{j=1}^{8760} Q_{\mathrm{con}}(\vartheta_j)}{\sum_{j=1}^{8760} \left(\frac{Q_{\mathrm{con}}(\vartheta_j)}{COP_h(\vartheta_j) - 1} \right)} \qquad \text{Eq. (8)}$$

where the superscript y indicates the annual demand and W ist the electric energy demand of the compressor and the fan. In the reference as well as in the proposed HP system in direct heating mode (d), the COP for DHW generation COP_{DHW} will be significantly lower than the COP for heating due to the higher temperature difference between evaporator and condenser. The electric energy consumption at a certain ambient temperature ϑ_j at a certain hour of the year h_j is given in Eq. (9):

$$W_{\mathrm{DHW}}(\vartheta_j, h_j) = \frac{Q_{\mathrm{DHW}}(h_j)}{COP_{\mathrm{DHW}}(\vartheta_j)} \qquad \text{Eq. (9)}$$

With the aid of the RPW-HEX, a certain amount of thermal energy can be generated with a better with COP_h instead of COP_{DHW} leading to a reduction of the electric energy demand. The situation is different in cooling mode. In the reference system, the total energy transferred on the hot side dissipates in the evaporator and is lost. Therefore, DHW is not only generated with a better COP with the RPW-HEX, but for free, as waste heat is utilized. The reduction of electric energy demand compared to direct DHW generation is therefore given in Eq. (10):

$$\Delta W_{\mathrm{DHW,RPW}}(\vartheta_j, h_j) = \begin{cases} Q_{\mathrm{RPW}}(\vartheta_j)\left(\dfrac{1}{COP_{\mathrm{DHW}}(\vartheta_j)} - \dfrac{1}{COP_{\mathrm{h}}(\vartheta_j)}\right), & \text{heating} \\ \dfrac{Q_{\mathrm{RPW}}(\vartheta_j)}{COP_{\mathrm{DHW}}(\vartheta_j)} & \text{cooling} \end{cases} \qquad \text{Eq. (10)}$$

where $\dot{Q}_{RPW}$ follows from Eq. (4) for heating and cooling. With Eq. (9) and Eq. (10), the anual efficiency for DHW generation reads:

$$\Delta EER_{DHW} = \frac{Q^{y}_{\mathrm{DHW}}}{W^{\mathrm{y}}_{\mathrm{DHW}}} = \frac{\sum_{j=1}^{8760} Q_{\mathrm{DHW}}(\mathrm{h_j})}{\sum_{\mathrm{j=0}}^{8760}\left(W_{\mathrm{DHW}}(\vartheta_j, h_j) - \Delta W_{\mathrm{DHW,RPW}}(\vartheta_j, h_j)\right)} \qquad \text{Eq. (11)}$$

4. RESULTS AND DISCUSSION

4.1 Performance indicators obtained from simulation studies

Figure 2 shows results of the performance indicators obtained from the simulations for different constant ambient temperatures. For annual calculations missing values between the depicted points were calculated by linear interpolation. As expected the COPs significantly decrease for low (heating) and high (cooling) ambient temperatures due to the high temperature lift between evaporator and condenser. The COPs of both system are almost equal although the sensible heat of the hot gas can be used for heating in the reference heating case and doesn't need to be cooled by the outdoor air in the cooling case for the novel system. Heating, cooling and direct DHW behaviour in Fig. 2(a) corresponds to operating modes (a),(b) and (d) for the novel system. Note that if the RPW-HEX can be charged, DHW can be generated with operating mode (c) with the better COPs "heating and RPW-DHW" and "cooling and RPW-DHW", respectively, instead of "direct DHW" generation. The RPW-HEX is able to store energy for ambient temperatures below 10 °C and above 30 °C (Fig. 2(b)). Between 10 °C and 30 °C, the hot gas temperature is too low ($\lesssim 64$ °C). Below ≈ -5 °C, the liquid injection valve starts to work and therefore, the hot gas temperature is limited to ≈ 115 °C. The storable energy from superheated refrigerant then only depends on the mass flow of the refrigerant.

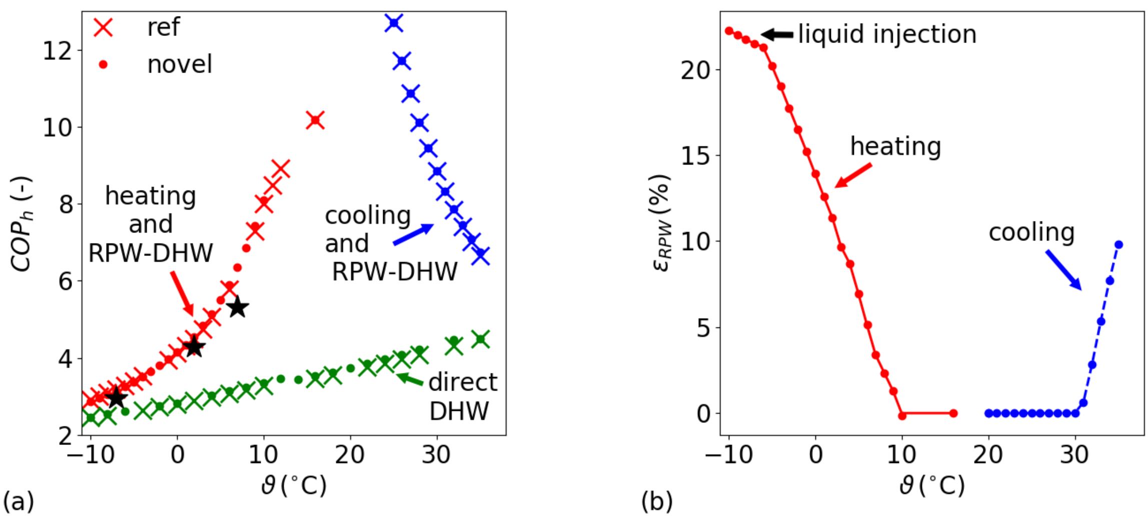

Figure 2: Performance indicators of the HP system in relation to the ambient temperature. The black stars in (a) denote the values from the measurements of the prototype HP without RPW-HEX.

4.2 Annual energy efficiency

The resulting annual heating and cooling demands are $Q_{\mathrm{heating}}^{\mathrm{y}}$=12 847 kWh and $Q_{\mathrm{cooling}}^{\mathrm{y}}$=1 093 kWh, respectively. Furthermore, the annual heating demand for DHW is $Q_{\mathrm{DHW}}^{\mathrm{y}}$ = 6 400 kWh. Figure 3 shows how much thermal energy can be provided by the RPW-HEX for DHW generation on a daily basis. During winter times in 17 days the RPW-HEX can deliver more than necessary to cover the DHW consumption. Hence, the surplus thermal energy might be shifted to the decentralized DHW storages. Due to the relatively low hot gas temperature in summer times, and the low amount of hot days (only 30 hours with temperatures above 30 °C) the RPW-HEX will not contribute significantly to the DHW generation given the climatic scenario selected here.

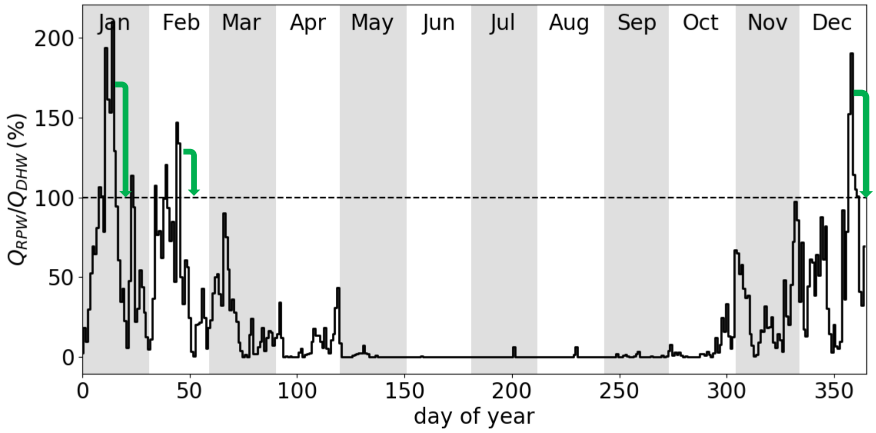

Figure 3: Coverage ratio of energy for DHW provided by the RPW-HEX. Note that, for the calculated annual performance indicators, it is assumed that surplus energy gained from the RPW-HEX can be stored for usage during the days to come (green arrows).

Table 1 summarizes the annual performance indicators and the electric energy consumption of the reference and the novel system. The annual performance for heating EER_h and cooling EER_c are almost equal whereas, the annual performance for DHW is about 11 % higher for the novel system compared to the reference system. To sum up, the novel concept saves 200 kWh electricity per year which results in 4.3 % lower annual total electric energy consumption.

Table 1: Calculated annual energy efficiencies

	$Q_{\mathrm{RPW}}^{\mathrm{y}}$(kWh)	W_{sum}(kWh)	W_{h}(kWh)	W_{c}(kWh)	W_{DHW}(kWh)	EER_{h}	EER_{c}	EER_{DHW}
Reference	0	4654	2586	105	1963	4.97	10.4	3.26
Novel	1389	4456	2583	105	1768	4.97	10.4	3.62

5. CONCLUSIONS AND OUTLOOK

Technical details and economic benefits are presented for a novel heat pump system, with a Refrigerant/PCM/Water Heat EXchanger (RPW-HEX) directly integrated in the hot gas section of an air source HP, for energy efficient DHW generation. Annual calculations indicate for average climatic conditions, intermediate heating temperatures and medium DHW consumptions, estimated savings of about 4.3 % of electric energy when compared to conventional air source HP systems. These energy savings result from a 11 % better performance for DHW generation with the novel system. The case study considers three apartments located in a low energy building, each with a heat demand of 2 kW per apartment at -10 °C and a DHW demand of 5.825 kWh per day. The absolute savings amount in this case is 200 kWh of electric energy with the proposed system per year.

The average energy price per kWh was 0.22 €/kWh for households in the Euro area in 2018 (EUROSTAT, 2018) which means a cost saving of 44 € per year. If we assume a latent storage density of 70 Wh/kg and would like to discharge the RPW-HEX after 2 hours of heating/cooling operation, we would need around 1.5 kWh storage capacity for three apartments or around 21 kg

PCM to deliver the daily DHW demand to the decentralized storages. For a typical price of 5 € per kg PCM, the PCM costs therefore about 105 €. Based on the design used for the prototype, the aluminium and manufacturing costs are around 200 € and 500 €, respectively. Summing everything up one can estimate investment costs of about 805 € for the RPW-HEX what gives a payback time of around 18 years for the given example. Next steps will include the assessment of the annual performances for other climatic conditions, building conditions and PCM materials, as the yearly performance and therefore the payback time strongly depend on these constraints. Furthermore, solar radiation needs to be included in the calculations for heating and cooling demand and experiments with the experimental air source HP and the RPW-HEX will be carried out in the months to come.

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NOMENCLATURE

COP	Coefficient of Performance (-)	W	Electric energy (kWh)
EER	Energy Efficiency Ratio (-)	ε_{RPW}	Utilization factor of the RPW-HEX (-)
Q	Thermal energy (kWh)	ϑ	Temperature (°C)
UA	Thermal conductance (kW/K)		

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