LARGE-CAPACITY PROPANE HEAT PUMPS FOR DHW PRODUCTION IN RESIDENTIAL BUILDINGS

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ABSTRACT

Using heat pump technology to provide Space Heating (SH) and to produce Domestic Hot Water (DHW) for residential buildings has been widely applied during past decades. In this study, two scenarios adopting large-capacity propane heat pumps are defined and evaluated. These two scenarios, which are named after Scenario A and Scenario B respectively, provide SH and DHW either separately by two units or integrally by one unit. The COP_1s of two scenarios are compared based on the simulation results from experimentally validated models. The results show that two scenarios have almost equal efficiency; the relative difference is within 6%. In the optimization analysis of Scenario B, varying DHW heating capacity produced by the desuperheater in the heat pump is modelled. The DHW demand ratio varies from approximately 9% to 20% with no detectable influences on the COP_1. The corresponding COP_1s and temperature profiles in the heat exchangers are demonstrated. The simulation results indicate that increasing DHW capacity in Scenario B can narrow down the temperature approach in the condenser and insignificantly improves the overall COP_1s.

Keywords: Propane, Heat Pump, Space Heating, Domestic Hot Water, COP

1. INTRODUCTION

The heating demand in residences and commercial buildings can be divided mainly into two categories: Space Heating (SH) and Domestic Hot Water (DHW). SH normally has a variable temperature level of inlet/outlet hot water to the heat distributing system depending on its type and the outdoor temperature, while DHW requires the hot water storage temperature to be above 60 ^{o}C to prevent the presence of legionella according to EU legislation (Pitarch, 2017). The ratio of DHW heating demand to the sum of DHW and SH heating demand can differ significantly over the month of a year, different climate conditions and types of building. For instance, single-family houses in Sweden annually demand 150 kWh/m^2 of SH as a mean value in 2012 (Sköldberg, H., *et al.*, 2014); the annual DHW heating demand in a typical house with 120 m^2 and 3 occupants is estimated to be around 3300 kWh (Janson, U., 2010), taking up approximately 15% of the total heating demand. However, the monthly ratio varies a lot, ranging from an estimated value of 7% in the coldest month to 100% during non-heating seasons in this case. On the other hand, this ratio can be vastly increased in low-energy or passive houses due to significant reductions in required heating demand of SH. Another study shows that the ratio is up to 50% to 85% in these high-performance residences in Scandinavia (Dokka and Hermstad, 2006). Therefore, when using heat pump systems to provide heating, the technology to fulfill both SH and DHW demands and sizing issues to cover the varying ratio should be properly considered.

In general, DHW in typical residential buildings can be produced by heat pumps in two ways: An individual unit with a relatively low capacity to cover the whole demand or a separate water heater in an integrated unit simultaneously providing SH. In European markets, the sales of individual DHW heat pump with average capacity of 2 to 3 kW has been significantly developing during the past decade. On the other hand, several large-capacity ground-source heat pump units (100 to 400 kW) using HFCs as the refrigerant have been successfully implemented featured with desuperheaters in the discharge line, which can provide DHW simultaneously with space heating or cooling (EHPA, 2016). Both solutions are promising replacement for direct electrical heating in providing DHW, which will save energy in buildings.

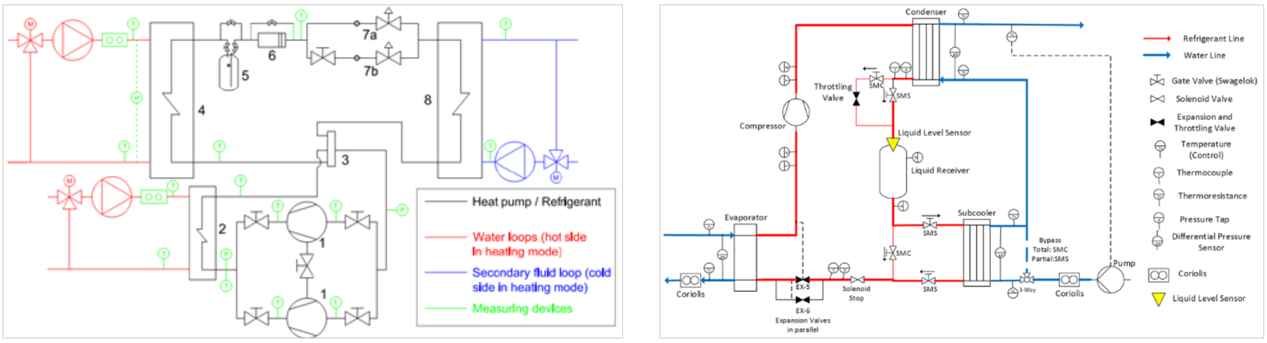
The working fluid of a heat pump featured with DHW production will influence its performance, since DHW should be heated up to at least 60 °C, which is typically higher than the temperature level needed for only SH. Propane (R290) has higher critical temperature than common-used HFCs such as R404A, R407C and R410A. It also theoretically gives lower compressor discharge temperature compared to these HFCs in typical levels of condensation pressure in such heat pumps (Palm, B., 2008). Another thermodynamic advantage of propane is that it is a suitable refrigerant for subcooling for its high specific heat in liquid states. In a recent research project about large-capacity heat pumps using natural refrigerants, three prototypes using R290 were experimentally investigated. Two of the prototypes are further studied in the present paper: a geothermal heat pump configured with a desuperheater covering demands for DHW (CASE 2) and a heat pump booster dedicated for DHW production (CASE 3) respectively. Referring to the previous results, two scenarios are defined and compared: one scenario uses a single heat pump (CASE 2) to cover all the hot water demand (SH and DHW) and the other scenario uses both heat pumps for DHW and SH separately (i.e. CASE 2 for space heating only and CASE 3 for DHW only). Based on the experimentally validated models, the objective of this study is to compare the COP_1s of these two scenarios and to carry out the optimization of CASE 2 heat pump to fulfill the residential SH and DHW heating demands with changing DHW demand ratios.

2. METHODOLOGY

This work aims at comparing two scenarios regarding the COP_1 and optimize the operation of CASE 2 on the conditions with varying DHW demand ratios. However, the experimental data obtained from previous studies are not directly comparable due to their different testing evaporation temperatures. Therefore, the experimental results were used to validate the IMST-ART models. The model outputs will be used for comparisons and optimization in the results section.

2.1 Experimental campaign of two heat pump units

Two large-capacity heat pump prototypes using propane were designed and experimentally characterized in terms of their performance of producing DHW on different working conditions. In Fig.1, the experimental set-ups of these two prototypes are illustrated. The detailed experimental set-ups, test matrices and results have been presented in previous studies by Piscopiello *et al.*, (2016) and Pitarch *et al.*, (2017). DHW is produced in different capacities and by different heat exchangers in two prototypes. In CASE 2, the condenser, which is a Brazed Plate Heat Exchanger (BPHE) marked as 4 in Fig. 1 (a), produces the majority of heating capacity for SH, while the desuperheater, which is a smaller BPHE marked as 2 in the same figure, produces the rest of the heating capacity for DHW. The unit can work with or without the desuperheater activated depending on the needs for DHW; it can provide solely the SH through the condenser. On the other hand, CASE 3 is a heat pump optimized to work at high subcooling in such a way that is able to work with high COPs when the needed water temperature lift in the condenser is high. This fact made this prototype specifically adapted for DHW production.



(a) CASE 2 (b) CASE 3

Figure 1. Experimental set-up of the two heat pump prototypes

2.2 Performance indicators of the two scenarios

Based on the configuration of these two heat pumps, SH and DHW can be provided either integrally in one unit or individually by two units, two scenarios are defined in the following approach:

- Scenario A: SH and DHW are separately provided by two individual units, CASE 2 and CASE 3 heat pumps respectively. I.e. CASE 2 heat pump works with only condenser activated for SH production only, and CASE 3 works on the condition with optimum subcooling in the condenser for DHW production;
- Scenario B: SH and DHW are simultaneously provided by the condenser and the desuperheater from CASE 2 heat pump.

As a result, the performance indicators of these two scenarios are defined in Eq. (1) and Eq. (2) in terms of the COP_1. The ratio of heating capacity of DHW to the sum of DHW and SH is defined as β in Eq. (3).

$$COP_{1,A} = (\dot{Q}_{SH,CASE\ 2} + \dot{Q}_{DHW,\ CASE\ 3}) \cdot (\dot{E}_{CASE\ 2} + \dot{E}_{CASE\ 3})^{-1} \quad (1)$$

$$COP_{1,B} = (\dot{Q}_{SH,\ CASE\ 2} + \dot{Q}_{DHW,\ CASE\ 2}) \cdot \dot{E}_{CASE\ 2}^{\ -1} \quad (2)$$

$$\beta = \dot{Q}_{DHW} \cdot (\dot{Q}_{SH} + \dot{Q}_{DHW})^{-1} \quad (3)$$

In these three equations, $\dot{Q}$ and $\dot{E}$ denote the heating capacity and compressor power input respectively in the unit of kW. Referring to the subscripts, $\dot{Q}_{SH,CASE\ 2}$ means the heating capacity for SH coming from CASE 2 heat pump, for example.

2.3 Simulation and model validation

To model the heat pump, the commercial IMST-ART software has been used. This is a dedicated software for modelling heat pump systems as a whole, according to the state-of-the-art. For more detailed descriptions of the software see Corberan, (2002). The model incorporates a number of sub-models representing the different parts of the heat pump: compressor (ARI polynomial), heat exchangers (detailed model based on (Corberan, 2001)) and expansion valve. All the information required by the models can be obtained directly from catalogue data from manufacturer.

The use of a detailed model for the heat exchangers allows having a temperature profile of both fluids inside the BPHE which will be useful for the result analysis. Once the models have been constructed, they have been validated in previous studies with a test matrix based on more than 15 points and in all the cases the discrepancy with the measured COP_1 has been lower than 5%, see Fig. 2 (Piscopiello *et al.*, 2016 and Pitarch *et al.*, 2017).

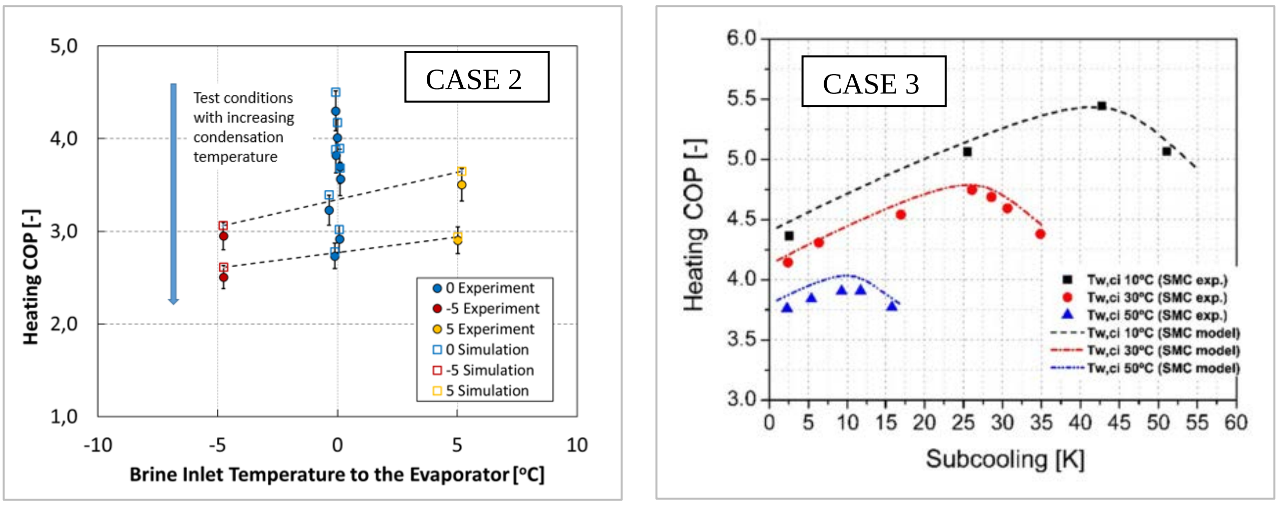


Figure 2. Comparison between experimental and model results for CASE 2 (left) and CASE 3 (right)

3. RESULTS

3.1 Comparisons of two scenarios

Once the models have been validated, the comparison between both scenarios has been made in terms of the built model. This has been motivated by the fact that the test conditions for CASE 3 were different to the test conditions for the CASE 2. The water inlet temperature for the CASE 3 was significantly higher as this prototype was developed for waste water energy recovery system. In addition, the model developed for CASE 3 has been re-scaled in order to supply the same capacity as CASE 2's DHW capacity production. The main parameters changed have been: Reduction of the compressor swept volume in order to supply the same capacity as well as heat exchangers length and number of plates in order to have the same heat transfer coefficients and temperature profiles. Therefore, in such comparisons the parameters for calculating COP_1 in Eq. (1) and Eq. (2) origin from the simulation for both cases in similar heating capacity.

The comparisons of COP_1 are illustrated in Fig. 3 on the same inlet and outlet temperature of secondary fluids in the evaporator (0/-3 ^oC) in both units, but two different levels of condensation temperature (water inlet/outlet temperature at 40/45 ^oC and 47/55 ^oC, abbreviated as W40/45 and W47/55) of CASE 2 specifically. The variable is the water inlet temperature to the BPHE that produces the DHW, and it is raised up to 60 ^oC for all the operational conditions. Firstly, it can be observed that the $COP_{1,A}$ decreases from 3.23 to 3.04 on W40/45 operational conditions with increasing DHW inlet temperature from 15 ^oC to 45 ^oC. Such reduction is due to the increasing condensation temperature of CASE 3. The same trend is found on W47/55 operational conditions, where $COP_{1,A}$ decreases from 2.85 to 2.68. However, $COP_{1,\,B}$ remains almost constant on both water temperature levels, because the condensation temperature is determined by the water temperature of the condenser instead of the desuperheater in CASE 2. In terms of the total heating capacity, the values of that for Scenario A range within around 53 kW to 58 kW, to which $\dot{Q}_{DHW,\,CASE\,3}$ contributes approximately 13 kW to 15 kW. On the other hand, Scenario B has lower total heat capacity at around 40 kW to 43 kW.

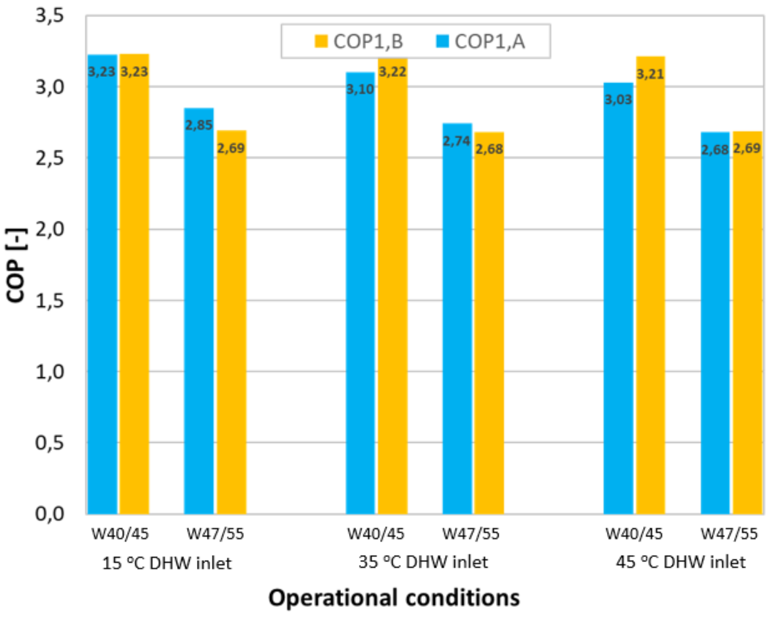


Figure 3. Comparisons of simulated total COP_1 values between Scenario A and Scenario B

When comparing these two scenarios whose COP_1 are shown in pairs in Fig. 3, the discrepancy of COP_1 is limited within about 6% among all the operational conditions. On W40/45 conditions, Scenario B can show slight advantages over Scenario A. On the contrary, Scenario A generally shows better efficiency on W47/55 conditions. Such variations are attributed to the drag or lift to $COP_{1,\,A}$ from CASE 3 comparing to $COP_{1,\,B}$, since CASE 2 shows almost equal efficiency in both scenarios. Depending on the DHW inlet temperature, CASE 3 is individually providing DHW either with higher efficiency or lower efficiency than Scenario B, leading to varied comparison results of COP_1. Generally, both scenarios almost equal in efficiency of providing SH and DHW.

With respect to the practical applicability of two scenarios, each has its own benefits and drawbacks when implemented into a building energy system. Speaking of the first investment and installation issues, Scenario A shows disadvantages because two heat pump units instead of one are supposed to be configured. Scenario B can be installed in a more compact and integral manner in the installation and the design of hydronic systems. Nevertheless, an individual unit for producing DHW as Scenario A can operate more flexibly in the situation that DHW is dominating or taking over thoroughly the heating demands during non-heating seasons or buildings with minor needs for SH. Conversely, Scenario B must meanwhile provide SH and utilize only partial heating capacity for DHW production. In addition, Scenario B can work in the reversed cooling mode; thus DHW would not affect the cooling capacity but being produced as heat recovery from the discharge gas.

3.2 Optimization of Scenario B

According to the studies mentioned in the introduction section the DHW demand ratio β varies significantly in building case to another. This optimizing analysis can be done through simulation of the vapor compression cycle by altering the water flow rate forwarded to the desuperheater and thus, varying the desuperheating degree of the hot gas of propane discharged from the compressor.

Fig. 4 shows the T-h diagram with certain presumed typical boundary conditions of a heat pump cycle working in standard rating conditions of medium and high temperature applications. In the figure, Point 1 to 2 illustrates the process of complete desuperheating of hot gas. With increasing water flow rate going through the desuperheater, its refrigerant outlet condition may vary from Point 1 to Point 2 and even further to Point 2' if partial condensation occurs in the desuperheater. The heating capacity from the desuperheater can take up at most 64% of the total capacity when water outlet temperature from it is controlled to be 60 ^{o}C, based on the experimental results by Piscopiello *et al*. (2016).

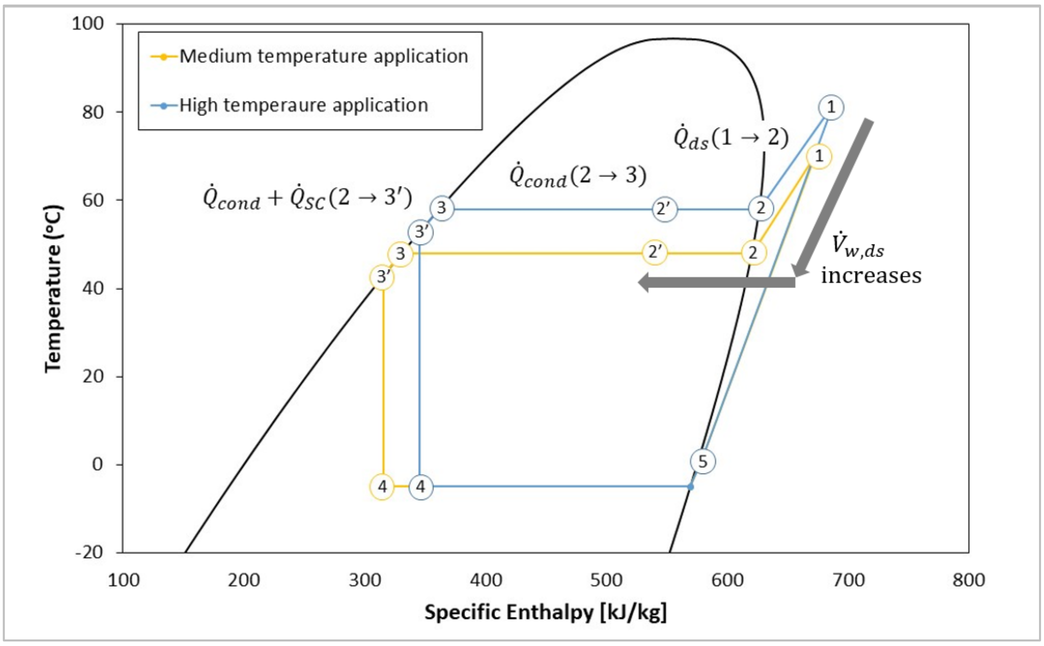


Figure 4. Behaviors of desuperheater and condenser on the high pressure side of CASE 2 in T-h diagram

In order to map the system performance depending on the water flow rate of the desuperheater within the conditions that no partial condensation of propane occurs in it, simulation using IMST-ART is conducted to predict the variation of the condensation temperature, COP_1 and β. The simulation is based on the experimental testing conditions in which the water inlet temperature to the desuperheater is 15 ^{o}C and inlet/outlet of the condenser is controlled to be 40/45 ^{o}C. Fig. 5 shows this variation – the condensation temperature subtly increases as the volumetric flow rate of water increases and subsequently decreases in a more evident manner. In contrast, the trend of COP variation behaves in the opposite way as a result.

Such variations are attributed to the temperature profiles and proportion variation in the areas of two-phase heat transfer in the condenser. As more heating capacity shifts from the condenser to the desuperheater with increasing water flow rate in it, the water flows through the condenser with imposed inlet and outlet temperature, in this case, 40 °C and 45 °C, decreases in terms of the volumetric flow rate. It leads to a reduction in the heat transfer coefficient on the water side. However, the two-phase heat transfer of condensation becomes more proportionally dominant in the condenser area due to that the refrigerant is desuperheated in the other heat exchanger.

In Fig. 5, it is observed that the positive contributions to the overall heat transfer coefficient from the refrigerant side overpass the negative from the water side when the volumetric flow rate of water in the desuperheater is larger than 0.09 m^3/h. The condensation temperature approaches more to the water temperature and thus COP_1 slightly increases by approximately 1%. In addition, β increases from 9.0% to 20.74%. In general, this part of simulation shows that the Scenario B unit is feasible to provide varied capacity for DHW use and consequently varied capacity for SH with adjustment in the water flow rate to both heat exchangers; the COP_1 and condensation temperature vary almost undetectably referring to their scales in this figure.

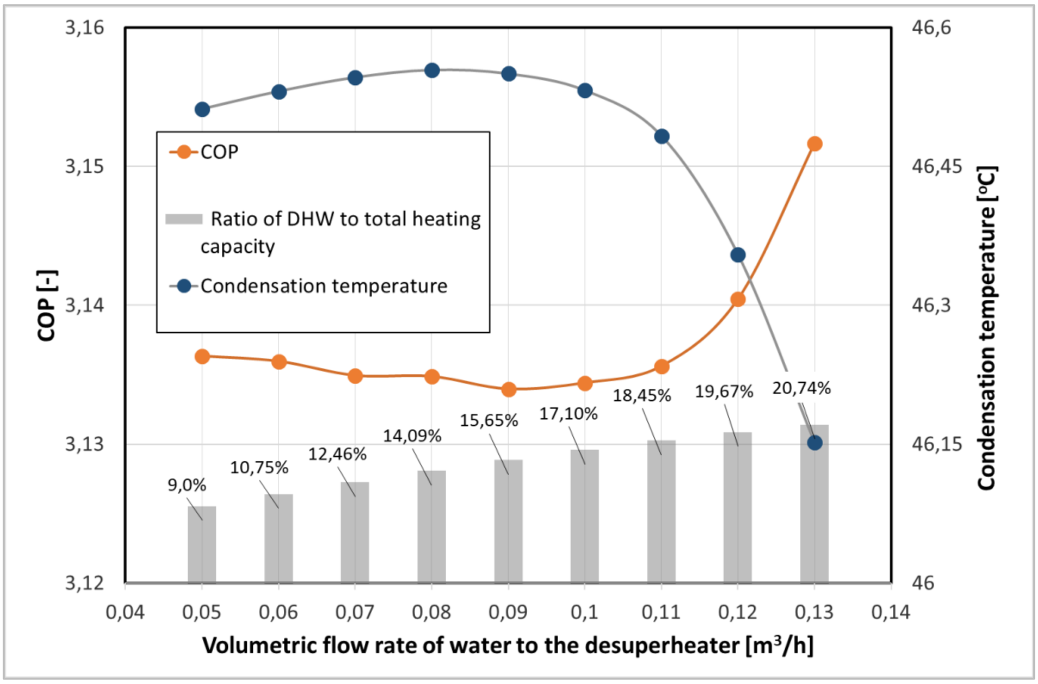
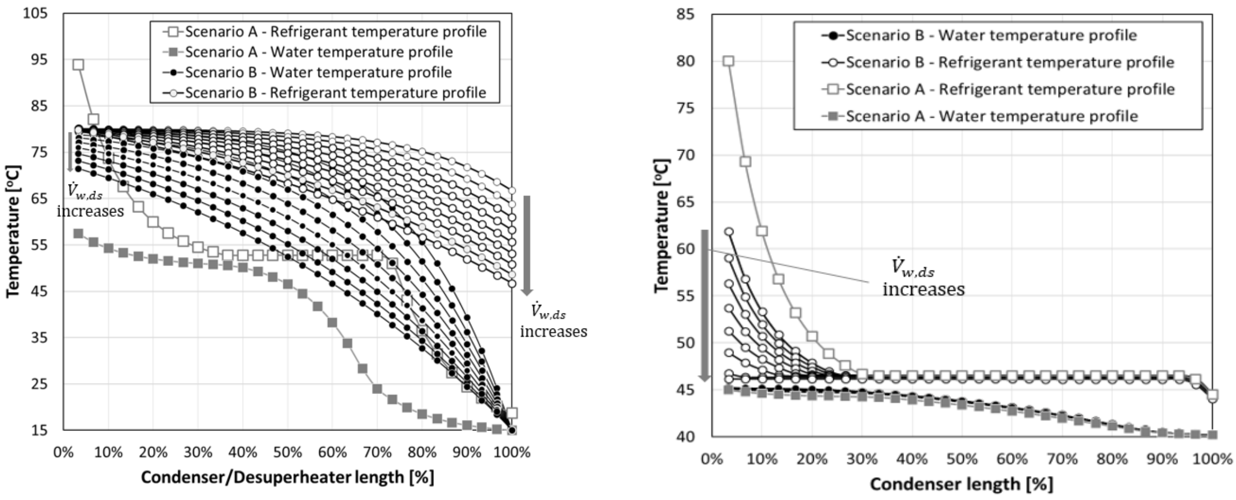


Figure 5. Effects of desuperheater water flow rate on the cycle performance

The effects of the increasing desuperheating degree in the desuperheater of Scenario B can be reflected in the temperature profiles of refrigerant and water in both heat exchangers. The temperature evolution of two fluids is calculated by IMST-ART software with known mass flow rates and inlet temperature. Fig. 6 (a) shows the comparisons between Scenario A and Scenario B of the temperature evolution along the heat exchanger which only provides DHW for users, while (b) shows that of the condenser providing SH in both cases.

It can be observed in Fig. 6 (a) that the more heating load allocated to the desuperheater by increasing the water flow rate going through that, the more two-phase area of refrigerant condensation will take up in the condenser. This behavior in Scenario B narrows down the temperature approach between the refrigerant and water at the entrance regions of refrigerant in the condenser comparing to Scenario A. As a consequence, the condensation temperature is slightly decreased and COP_1 is improved as demonstrated in Fig. 5.



(a) BPHE providing DHW

(b) condenser providing SH

Figure 6. Temperature profile in two heat exchangers of Scenario A and Scenario B

On the other hand, the temperature profiles vary more evidently in the desuperheater due to the reduction in both the refrigerant and water outlet temperatures with increasing water flow rate. Fig. 6 (b) illustrates that the temperature profiles of two fluids match better with incremental desuperheating degree; the temperature approach decreases by approximately 20 ^{o}C at the refrigerant outlet. Comparing to Scenario A, in which the condenser provides full load of DHW heating from 15 ^{o}C to 60 ^{o}C at optimum subcooling of 35 ^{o}C, Scenario B works with a lower condensation temperature that is determined by the water inlet and outlet temperature of the condenser instead of the desuperheater itself. The temperature approach at the water outlet is much smaller in Scenario B than Scenario A due to the lower condensation temperature and discharge temperature from the compressor.

4. CONCLUSIONS

In this study, analysis of using large-capacity propane heat pumps to cover hot water demands with varying DHW demand ratio in residential buildings is presented. Two heat pump prototypes which have been experimentally characterized are served as the basis in the present studies. More detailed evaluation on two proposed scenarios fulfilling SH and DHW individually by two separate units or integrally by one unit is conducted on the basis of experimentally validated models. The comparisons of these two scenarios, namely Scenario A and Scenario B respectively, are summarized in the following points:

- The IMST-ART models' results show that the total heating capacity of Scenario A is around 53 kW to 58 kW while of Scenario B is around 40 kW to 43 kW.
- Two scenarios have almost equal efficiency in terms of the $COP_{1.}$ The relative difference between $COP_{1,A}$ and $COP_{1,B}$ is within 6% on all compared operational conditions.
- $COP_{1,A}$ is slightly lower than $COP_{1,B}$ on W40/45 conditions but generally higher on W 47/55 conditions (see Fig. 3); it decreases with increasing water inlet temperature to the BPHE producing DHW.

Due to the needs of varying DHW heating capacity in Scenario B in different working conditions of the heat pump, an optimization process is conducted by simulating increasing water flow rate to the desuperheater in step. The $COP_{1,B}$ varies almost undetectably around 3.15 with the ratio of DHW to the total heating capacity increasing from 9% to about 20%. The temperature profiles in both the condenser and the desuperheater show that by increasing water flow rate to the desuperheater for DHW production from 0.05 to 0.14 m^3/h, the temperature approach between the refrigerant and the water is narrowed down. The temperature profiles match better in Scenario B than Scenario A in BPHEs.

In a nutshell, both scenarios are capable of satisfying the SH and DHW heating demands for residential buildings with similar efficiency when they are properly sized. Scenario B can only produce DHW simultaneously with providing SH. However, the operation of the desuperheater can be optimized in order to produce various share of DHW heating capacity to the total.

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REFERENCES

Corberán JM, Gonzálvez J, Montes P, Blasco R. ‘ART’ A Computer Code to Assist the Design of Refrigeration and A/C Equipment. In: International Refrigeration and Air Conditioning Conference (Purdue University). Paper 570; 2002. Vol 40, 37-59.

Corberan JM, Fernandez de Cordoba P, Gonzalvez J, Alias F, 2001. Semiexplicit method for wall temperature linked equations (sewtle): a general finite-volume technique for the calculation of complex heat exchangers. Numerical Heat Transfer Part B: Fundamentals.

Dokka, T.H., Hermstad, K., 2006: Energieffektive boliger for framtiden – en håndbok for planlegging av passivhus og lavenergihus (Energy Efficient Houses for the Future – A Handbook for Planning of Passive Houses and Low-Energy Houses). Norwegian final report from ECBCS Annex 38, Sustainable Solar Housing. SINTEF Building and Infrastructure.

EHPA, 2016. European Heat Pump Market and Statistics Report 2016, Brussels, Belgium, 41 p.
European Commission 7th framework program. Next Generation of Heat Pumps working with Natural fluids. http://cordis.europa.eu/project/rcn/105998_en.html

IMST-ART: Simulation tool to assist the selection, design and optimization of refrigerator (v3.70), [accessed 07-07-2016].

Janson,U., 2010. Passive houses in Sweden, From Design to Evaluation of Four Demonstration Projects, Doctoral Thesis, Lund University, Lund, Sweden, 27 p.

Palm, B., 2008. Hydrocarbons as refrigerants in small heat pump and refrigeration systems – A review. International Journal of Refrigeration 31, 552-563.

Piscopiello, S., et al., 2016. Performance Evaluation of a Large Capacity Water-Water Heat Pump using Propane as Refrigerant, Paper ID 1129, Gustav Lorentzen Natural Working Fluids Conference, Edinburgh, Scotland.

Pitarch, M., 2017. High capacity heat pump development for sanitary hot water production. Doctoral thesis, Universitat Politècnica de València, Valencia, Spain, 58 p.

Pitarch, M., et al., 2017. Experimental study of a subcritical heat pump booster for sanitary hot water production using a subcooler in order to enhance the efficiency of the system with a natural refrigerant (R290). International Journal of Refrigeration, *73*, 226-234.

Sköldberg, H., et al., 2014. The heating market in Sweden-an overall view, Stockholm, Sweden, 18 p.