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Experimental Study of a R407C Liquid-to-Water Heat Pump with Domestic Hot Water Production

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ABSTRACT

Currently, a number of companies manufacture liquid-to-water heat pumps for geothermal applications, based on the scroll compressor technology and using refrigerant R407C. The performance of this kind of machines depends greatly on the operating conditions, basically on the temperature levels of the liquid and water fluids. This paper presents an experimental study of a liquid-to-water heat pump with the possibility of domestic hot water (DHW) production. The refrigerant used is R407C. The main components of the heat pump are a scroll type compressor, a desuperheater for hot water production, two brazed plate heat exchangers (condenser and evaporator) and a thermostatic expansion valve with external equalizer. A liquid-vapor heat exchanger (LVHEx) has been included to check the effect of this heat exchanger on the heat pump performance. Experimental results for typical operating conditions of liquid-to-water heat pumps (EN 14511-2, 2013) in the heating mode of operation (low and medium temperature applications) are presented and discussed. Additional tests were performed to check the performance of the heat pump with DHW production at different water temperature values. From the experimental measurements, parameters such as the COP, heating powers or compressor electric power consumption were obtained and analyzed. The final goal of the paper is to characterize and to explain from a thermodynamic perspective the performance of the heat pump under different operating conditions.

1. INTRODUCTION

The use of heat pumps has been increasing during the last decades. According to Calm (1987) in the decade of the late 70's and early 80's electric heat pumps were available in one of every four new constructions in the USA; becoming a third at the end of the 80's. This type of machines allows the air conditioning of a space (in both the winter and summer seasons) and the simultaneous or unique generation of DHW. Hepbasli and Kalinci (2009) collected information from 1976 to 2007 about heat pump water heating systems; they analyzed various types of heat pumps and their respective efficiencies, comparing them with other traditional technologies for developing the same functions. They highlight that residential heat pump (HP) water heating systems can supply much more heat just with the same amount of electric input used in units equipped with conventional heaters, which consume fossil fuels or electricity.

Focusing on more recent data, the HP market has been increasing in recent years in developed countries as reflected in Lee (2009). He points that installations of ground source heat pumps (GSHP) systems are rapidly increasing in both private and public sectors. The GSHP systems are extremely energy efficient, clean and environmentally friendly (Lee, 2009). According to Bayer *et al.* (2012), GSHPs appear an attractive technology in order to reduce greenhouse gas emissions, especially if the primary energy for the electricity consumed by the heat pump is renewable, nuclear and/or hydropower. Urchueguía *et al.* (2008) showed that in some climates, GSHP represent a technically viable technology for heating, cooling and domestic hot water systems in buildings. They also indicate that GSHP provide energy efficiency savings when compared to an air source heat pump.

Many authors have studied the effect of different variables on the performance of GSHP. The water temperature in the condenser and the production of DHW affect the system performance of a geothermal heat pump (GHP), as explained by Fernández-Seara *et al.* (2012). Sebarchievici and Sarbu (2015) showed a COP improvement of a GHP with domestic hot water production through the addition of an intermediate tank (buffer tank) and an adjustable water pump. Hengel *et al.* (2016) developed and validated a simulation model, for the analysis of air-source heat pump with desuperheater for DHW production. By simulations they obtained energy saving potentials of heat pumps with desuperheaters compared to the reference systems without desuperheater.

Related with the effects of a liquid-vapor heat exchanger Domanski *et al.* (1994) presented a theoretical evaluation of the performance effects resulting from using this type of heat exchanger. They showed that the benefit of using the LVHEx depends on a combination of operating conditions and refrigerant properties. Kim (2002) carried out experiments in order to study the effects of using the LVHEx with different refrigerants and compared the results with data obtained with R22. They obtained that the effects of the LVHEx depends of the refrigerant; in particular, a small improvement in the efficiency for R407C was obtained. Bakirci and Colak (2012) studied the effect of the LVHEx in a GHP with R134a. They obtained a small decrease in the performance of the system when the LVHEx was used. Janković *et al.* (2018) presented a numerical analysis of a liquid-to-water heat pump with DHW production. They showed that increasing the degree of superheat at the evaporator outlet leads to lower COP values and heating powers (for low and medium temperature applications) either with or without DHW production. Also, they showed that there is an optimum subcooling degree value that maximized the system performance. With regards to the effect of the LVHEx, they showed that for equal subcooling and superheating degree values, using the LVHEx has a slight negative effect on the system performance when DHW production is not active; however, it turns to a slight positive effect when DHW production is considered.

The main objective of this article is to obtain a better understanding of the performance of a heat pump with the refrigerant R407C under different operating conditions. Experimental results for typical operating conditions of liquid-to-water heat pumps (EN 14511-2, 2013) in the heating mode of operation (low and medium temperature applications) are presented and discussed. Additional tests were performed to check the performance of the heat pump with DHW production at different water temperature values. Also, a liquid-vapor heat exchanger has been included to study the effect of this heat exchanger on the heat pump performance.

2. DESCRIPTION OF THE EXPERIMENTAL SET-UP

The main component of the experimental set-up is a liquid to water heat pump, which can work under the heating or cooling modes of operation. Production of DHW is also possible in the desuperheater. The refrigerant used is R407C.

Figure 1 shows a schematic representation of the heat pump. The heat pump has a scroll type compressor (220-230V/50Hz) with a displacement of 9.44 $m^3\ h^{-1}$ for a rotational speed of 2900 rpm. The desuperheater (used for DHW production) is a brazed plate heat exchanger with 30 effective plates. Condenser and evaporator are also brazed plate heat exchangers with 20 effective plates each; this allows interchanging the functionally of the mentioned heat exchangers depending on the working mode of operation. In the heating (cooling) mode of operation, the outdoor heat exchanger functions as the evaporator (condenser) and the indoor heat exchanger as the condenser (evaporator). The expansion valve is a thermostatic expansion valve with adjustable superheat degree and external equalizer. There is also a tube-in-tube liquid-vapor heat exchanger; the superheated vapor from the evaporator flows through the external tube and the liquid refrigerant from the condenser flows through the internal tube. Experiments can be done using or not this heat exchanger. If valves A1, A2, B1 and B2 are opened (closed) and valves A3 and B3 are closed (opened) then the LVHEx is used (not used).

A 15% (mass basis) propylene-glycol water mixture (PGW) is pumped through the outdoor heat exchanger from a 750 L PGW tank. Similarly, water is pumped from a 280 L tank to the indoor heat exchanger. Variable speed pumps were used in order to control the PGW and water flow rates through the outdoor and indoor heat exchangers, respectively. The supply temperature to each heat exchanger was controlled by adjusting the temperature levels of the corresponding tanks using an additional water circuit and an air-to-water heat pump (not shown in Figure 1). For the DHW tests, water is pumped from a 300 L tank to the desuperheater at the desired temperature.

Table 1: Types of sensors and accuracies

Variable	Instrumentation	Accuracy
Water and PGW volumetric flow rate	Electromagnetic flow meter	±0.2% reading
DHW volumetric flow rate	Paddle-wheel flow sensor	±(0.1 L min^{-1}+2.5% reading)
Water and PGW temperatures	PT 100 Class B 1/10	±0.1 K
DHW temperature variation	Thermocouple T type	±0.3 K
Refrigerant temperatures	PT 100 Class A	±0.3 K
Refrigerant pressures (gauge)	Pressure transmitter	±0.2 bar
Atmospheric pressure	Digital barometer	±0.003 bar
Compressor's electrical power	Power transducer	±0.5% reading

The experimental facility has been equipped with a supervisory control and data acquisition (SCADA) system based on a PC and a programmable controller. Figure 1 shows the location of the different sensors used in the experimental setup and Table 1 summarize their main characteristics.

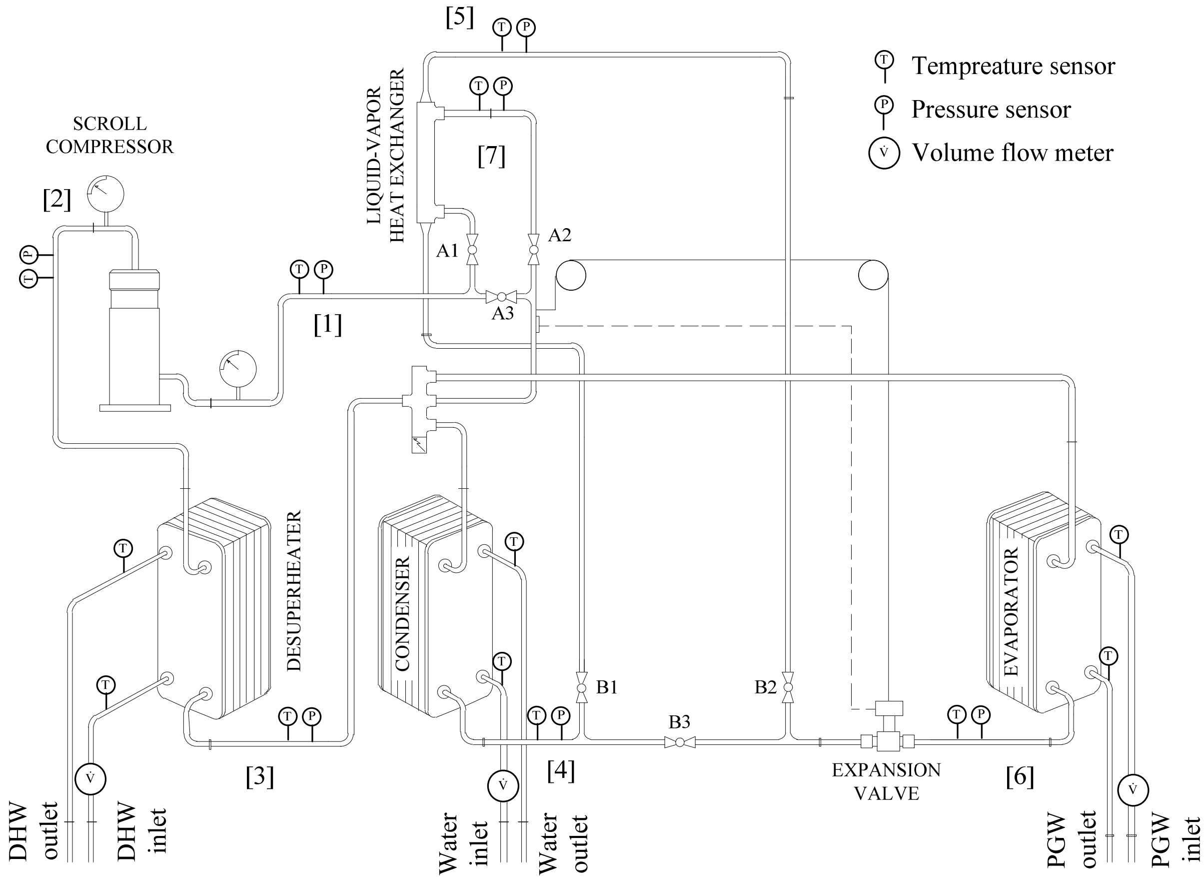

Figure 1: Schematic representation of the heat pump (winter-heating mode) *(Numbers in square brackets are related to temperature and pressure sensors)*

3. TEST CONDITIONS AND EXPERIMENTAL PROCEDURE

Experiments were performed for typical operating conditions of liquid-to-water heat pumps (EN 14511-2, 2013) in the heating mode of operation for low and medium temperature applications. Additional tests were carried out to check the performance of the heat pump with DHW production at different water temperature values. Two different water temperature levels (43.5°C and 54.5°C) at the inlet of the desuperheater were considered with a nearly constant (5.2 L min^{-1}) water flow rate. All experiments were performed twice, with and without the LVHEx, in order to evaluate the effect of this heat exchanger on the heat pump performance. Combining these different operating conditions gives a total of 12 experiments, which are summarized in Table 2.

Table 2: Operating conditions for the experimental tests

LVHEx		**Water temperatures (°C) (indoor HEx)**		**DHW temperatures (°C) (desuperheater)**		
ON	OFF	30-35	40-45	DHW not used	43.5	54.5
Additional fixed operating conditions: -DHW flow rate 5.4 L min^{-1} (for experiments with DHW production) -PWG temperatures (outdoor heat exchanger): 7/10 (°C)						

The degree of superheat at the evaporator outlet was adjusted initially by manual rotation of the side stem of the thermostatic expansion valve. During the experiments described in Table 2, the degree of superheat at the evaporator outlet was nearly constant and equal to 3.5 ±0.2 K.

All sensors were scanned within a time interval of 3 s. The SCADA system allows an automatic control of the PGW and water temperatures at the inlet of the outdoor and indoor heat exchangers, respectively. The corresponding temperatures at the outlet of these heat exchangers were also controlled automatically by means of PID controllers that regulate the PGW and water flow rates. The DHW temperature at the inlet of the desuperheater and the DHW flow rate were adjusted manually for each experiment.

During each experiment, steady state conditions were confirmed by visualizing the measured variables in the SCADA system. In general, steady state operation was reached within a period of 1-2 hours. Thereafter, data recording during the next 40-60 minutes was performed for all the measured variables.

4. DATA REDUCTION AND ANALYSIS

From the measured data in the experimental setup, the thermodynamic conditions of the refrigerant, the water and the PGW at the various states of the system are determined at steady state conditions. The thermodynamic conditions at the inlet of the evaporator are determined from the conditions of the refrigerant at the inlet of the thermostatic expansion valve (condenser outlet or LVHEx outlet when this heat exchanger is used) and assuming an isenthalpic process through the valve. All refrigerant and water properties were obtained using the Engineering Equation Solver (Klein, 1992-2016). The PGW properties were obtained using technical data from the propylene-glycol supplier. A detailed analysis of the experimental uncertainties was made and some typical uncertainty bands are included in the results. The uncertainties were determined following the guidelines given in ISO (2008).

The following assumptions were considered in the thermal analysis of the heat pump:

1. Heat losses/gains in the refrigerant and external fluids (water and PGW) pipes are negligible (pipes are insulated);
2. Heat transfer between the heat exchangers and the surroundings is negligible;
3. Pressure drops in the refrigerant pipes between main components are negligible;
4. The refrigerant's enthalpy at the evaporator inlet is equal to refrigerant's enthalpy at the expansion valve inlet;
5. Heat transfer in the 4-way valve is negligible.

Turning the attention to the water flows across the condenser (indoor heat exchanger) and the desuperheater, the heat transfer rate for each heat exchanger is calculated by applying simple energy balances:

$$\dot{Q}_C = \dot{V}_{C,W} \cdot \rho_{C,W,i} \cdot C_{p,C,W} \cdot \left(T_{C,W,o} - T_{C,W,i}\right) \quad (1)$$

$$\dot{Q}_{DS} = \dot{V}_{DS,W} \cdot \rho_{DS,W,i} \cdot C_{p,DS,W} \cdot \left(T_{DS,W,o} - T_{DS,W,i}\right) \quad (2)$$

where the specific heats are evaluated at the average fluid temperature across each heat exchanger, and the densities at the corresponding inlet temperature values.

Similarly, an energy balance in the evaporator gives:

$$\dot{Q}_E = \dot{V}_{E,PGW} \cdot \rho_{E,PGW,i} \cdot C_{p,E,PGW} \cdot (T_{E,i} - T_{E,o}) \tag{3}$$

where the specific heat of PGW is evaluated at the PWG average temperature and the density is evaluated at the PGW inlet temperature value.
The coefficient of performance (COP) of the heat pump is calculated from the heating power in the condenser, the heat power used for DHW production (if any) in the desuperheater and the compressor's electric power consumption:

$$COP = \frac{\dot{Q}_C + \dot{Q}_{DS}}{\dot{W}_{elec}} \tag{4}$$

Taking into account the previous assumptions, an energy balance in the refrigerant side for both the condenser (indoor heat exchanger) and the desuperheater can be applied in order to determine the refrigerant mass flow rate through the condenser (and desuperheater):

$$\dot{Q}_C + \dot{Q}_{DS} = \dot{m}_{C,ref} \cdot (h_2 - h_4) \tag{5}$$

Similarly,

$$\dot{Q}_E = \dot{m}_{E,ref} \cdot (h_7 - h_6) \tag{6}$$

Ideally, both calculated mass flow rate values should be the same. Deviation between the refrigerant mass flow rate through the condenser and evaporator provides a general idea about the confidence on the experimental data and thermal analysis assumptions. The actual refrigerant mass flow rate ($\dot{m}_{ref}$) is calculated as the average of the values obtained in the condenser and evaporator, that is:

$$\dot{m}_{ref} = \frac{\dot{m}_{E,ref} + \dot{m}_{C,ref}}{2} \tag{7}$$

Additionally, from the compressor manufacturer's data, the volumetric efficiency can be obtained as a function of the pressure ratio (*PR*). The following equation was obtained by a simple regression analysis, with a coefficient of determination R^2 of 96.5%:

$$\eta_{vol} = 1.04783 - 0.028886 \cdot PR \tag{8}$$

where

$$PR = \frac{P_2}{P_1} \tag{9}$$

Then, the refrigerant mass flow rate can also be determined as:

$$\dot{m}_{ref,comp} = \dot{V}_{comp} \cdot \eta_{vol} \cdot \rho_1 \tag{10}$$

Figure 2a) compares the refrigerant mass flow rate values calculated from an energy balance in a control volume that includes the condenser and desuperheater with those values obtained from an energy balance in the evaporator. Figure 2b) compares the refrigerant mass flow rate obtained from equation (7) with the values estimated with equation (10) using the compressor manufacturer's data. Results in Figure 2 show that the discrepancies of the calculated mass flow rate values are within 5% and, in general, within the uncertainty of the estimated parameters.

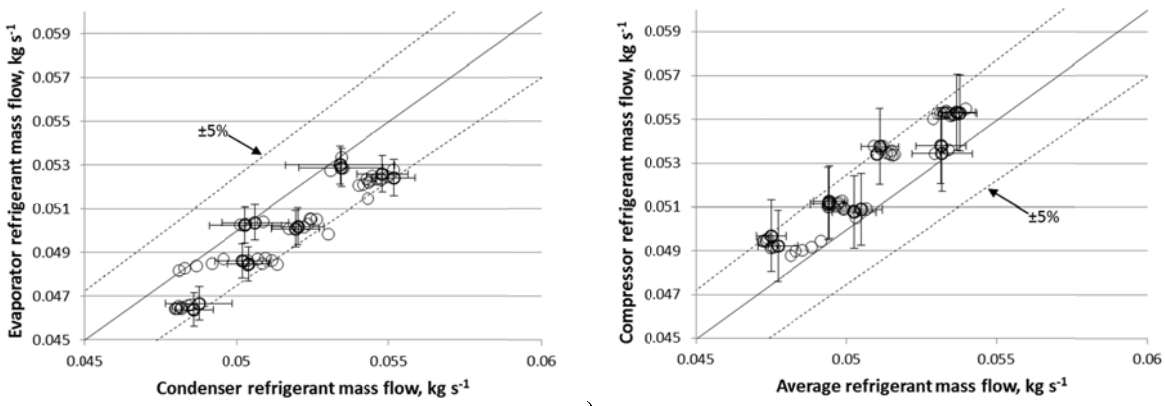

Figure 2: Deviation between calculated refrigerant mass flow rate values. a) From energy balances at condenser and evaporator. b) From energy balances and compressor manufacturer data

5. RESULTS AND DISCUSSION

A set of 12 experiments were performed for the operating conditions collected in Table 2. Figure 3 shows the heating power in the condenser, the compressor's electric power and the COP for different water inlet/outlet temperature values in the condenser, when the desuperheater is not used for DHW production. Results are presented when the LVHEx is used (LVHEx-ON) or not used (LVHEx-OFF). As expected, when increasing the water temperature (from 30/35°C to 40/45°C) the compressor's required power increases and the heating power and COP decrease. Results in Figure 3 show that using the LVHEx leads to lower heating powers in the condenser and lower compression powers, being the COP nearly the same when using or not the LVHEx. These results contrast sharply with the results of Janković *et al.* (2018), who predicted a negligible effect of the LVHEx on the heating power in the condenser. However, their results were based on equal subcooling degree values at the condenser outlet; though, as shown later if Figure 5, this is not the case for the present experimental tests.

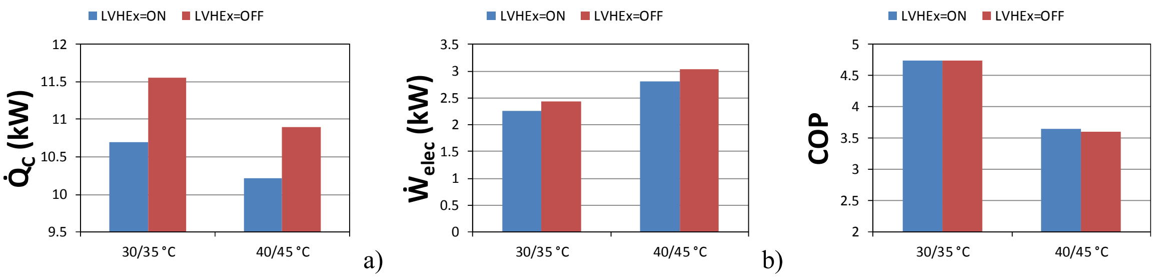

Figure 3: a) Heating power, b) compressor's electric power and c) COP, for low and medium temperature applications under the heating mode of operation using or not the LVHEx. DHW production is not considered

Figure 4 shows the suction and discharge pressures as well as the pressure ratio for the same cases considered in Figure 3. The discharge pressure and compression ratio follow the expected trends with the water temperature levels in the condenser; i.e. both are higher for medium temperature applications (40/45°C) than for low temperature applications (30/35°C). Results in Figure 3 showed that the heating power is higher when the LVHEx is not used; then, we should expect higher condensation (and discharge) pressures and higher pressure ratio values when the LVHEx is not used than when it is used, as confirmed in Figure 4. Finally, a clear effect of the water temperature levels (low or medium) on the suction pressure cannot be concluded; though, the variations are small when compared with those obtained for the discharge pressure.

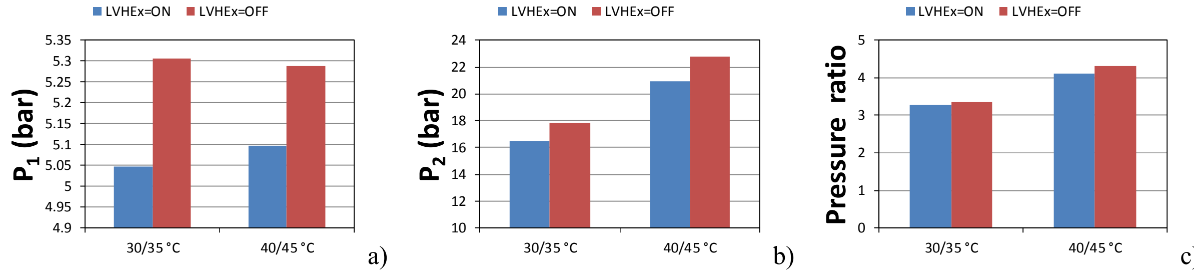

Figure 4: a) Suction pressure, b) discharge pressure and c) pressure ratio, for low and medium temperature applications under the heating mode of operation using or not the LVHEx. DHW production is not considered

Figure 5 shows the suction and discharge temperatures as well as the degree of subcooling at the condenser outlet for the same cases considered in Figure 3. Obviously, the discharge temperature increases with the temperature level application in the indoor water loop (low or medium). Using the LVHEx produces an additional reheat of the vapor coming from the evaporator, so higher suction temperatures are expected when the LVHEx is used. For higher suction temperature values, higher discharge temperature values should also be expected, as confirmed from the results in Figure 5b). The experimental results for the degree of subcooling at the condenser outlet are represented in Figure 5c). It can be seen that the degree of subcooling is somehow higher (around 1.5 K) for the medium temperature application than for the low temperature one. A significant change in the degree of subcooling (between 6.0 K and 6.7 K) is observed when considering the experimental results when the LVHEx is used or not used. This last behavior might be explained from a structural point of view. Since a liquid refrigerant tank is not used in the heat pump, the refrigerant charge is distributed through the heat pump components and lines. The addition of the LVHEx component adds an extra volume to the heat pump. This volume must be filled with refrigerant. A fraction of the LVHEx is filled with liquid refrigerant (with a density much higher than the vapor phase). Also, when using the LVHEx, the liquid line (from the condenser to the expansion valve and through the LVHEx) becomes larger than when the LVHEx is not used (see Figure 1). As a result, when the LVHEx is used a higher mass of refrigerant can be accommodated in the heat pump pipes and LVHEx, which prevents the condenser from containing too much refrigerant. In contrast, when the LVHEx is not used, a larger amount of refrigerant accumulates in the condenser and, as a result, a larger fraction of the condenser heat transfer area is used for cooling the refrigerant below the saturation temperature (i.e. increasing the degree of subcooling) as confirmed in the experimental results in Figure 5c). Since a larger fraction of the heat transfer area is filled with liquid, a lower heat transfer area is available for the phase change (condensation) process. As a result, a higher refrigerant condensation pressure (and temperature) is required, as confirmed in the experimental results shown in Figure 4b).

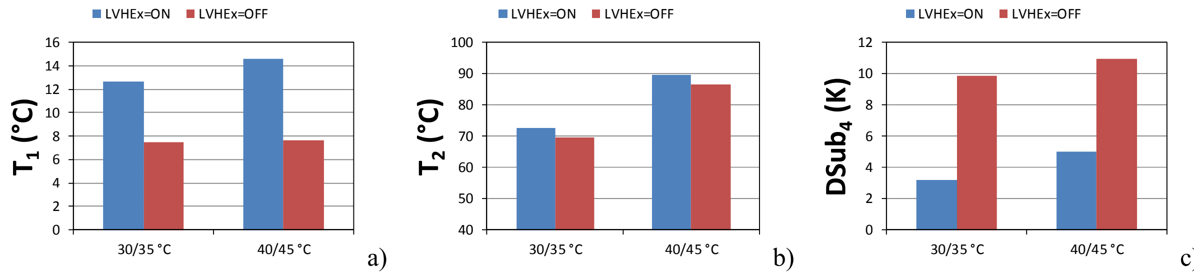

Figure 5: a) Suction temperature, b) discharge temperature and c) degree of subcooling at the condenser outlet, for low and medium temperature applications under the heating mode of operation using or not the LVHEx. DHW production is not considered

The effect of DHW production at different temperature levels has also been experimentally evaluated. Figure 6 shows the heating power in the condenser, desuperheater and the sum of the heating power in both heat exchangers for the low and medium temperature heating modes of operation when DHW production is not considered and also for DHW production with water inlet temperatures of 54.5°C and 43.5°C. In these experiments the LVHEx was not considered. Results show that the condenser and total heating powers are higher for the 30/35°C mode of operation than for the 40/45°C one. However, for the 40/45°C application higher discharge temperatures are obtained than for 30/35°C one; this explains that for the former much higher values of $\dot{Q}_{DS}$ (almost a two-fold factor) are obtained than for the later. On the other hand, it can be seen that when decreasing the DHW inlet water temperature, the DHW heating power increases but the condenser heating power decreases, being the total heating power nearly constant.

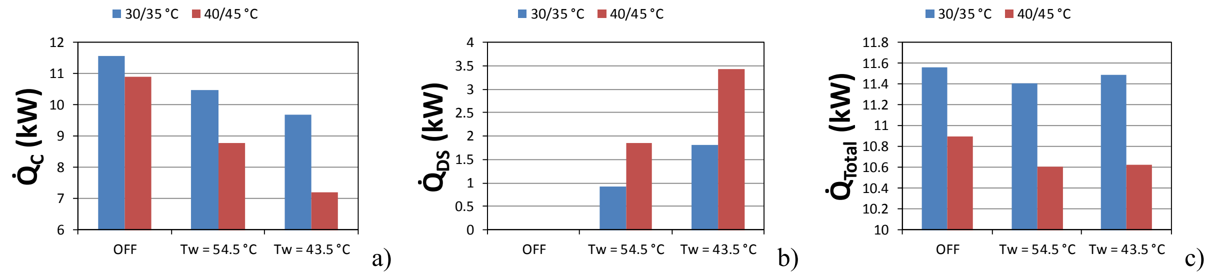

Figure 6: a) Heating power in the condenser, b) in the desuperheater (for DHW) and c) total heating power, for low and medium temperature applications under the heating mode of operation and for different DHW inlet temperatures. The use of the LVHEx is not considered

Figure 7 shows the compressor's electric power and the COP for the different cases considered in Figure 6. For the 30/35°C case, the compressor's electric power is nearly unaffected by the DHW production; however, for the 40/45°C case the compressor's electric power is lower when DHW production is considered and decreases with lower DHW water temperature values. Finally, the heat pump COP is slightly higher with DHW production than without DHW production. Results in Figure 7b) suggests that the COP increases when the DHW temperature at the desuperheater inlet decreases.

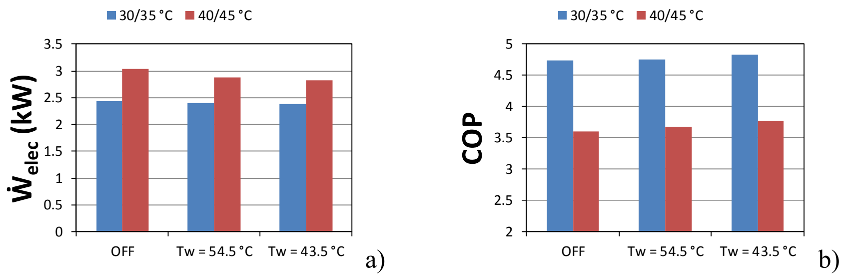

Figure 7: a) Compressor's electric power and b) COP, for low and medium temperature applications under the heating mode of operation and for different DHW inlet temperatures. The use of the LVHEx is not considered

Finally, Figure 8 shows experimental results for the degree of subcooling at the condenser outlet, discharge pressure and also for the dew point temperature at the desuperheater outlet pressure, for the same operating conditions considered in Figure 6. It can be seen that the dew point temperature is around 45°C for the low-temperature (30/35°C) experiments but increases up to 52-55°C for the medium-temperature (40/45°C) ones. Then, for the low-temperature (30/35°C) experiments no refrigerant condensation (or a negligible amount) takes place in the desuperheater for DHW inlet temperatures of 43.5 or 54.5°C. As a result, the degree of subcooling remains nearly constant for these experiments. However, for the medium-temperature (40/45°C) experiments, partial condensation occurs in the desuperheater, especially for the DHW temperature of 43.5°C; which explains the variations observed in the degree of subcooling shown in Figure 8a) for the 40/45°C experiments. If a smaller amount of refrigerant accumulates in the condenser, the degree of subcooling decreases and so does the system high pressure due to an increase of the heat transfer area available for the phase change (condensation) process (see Figure 8c).

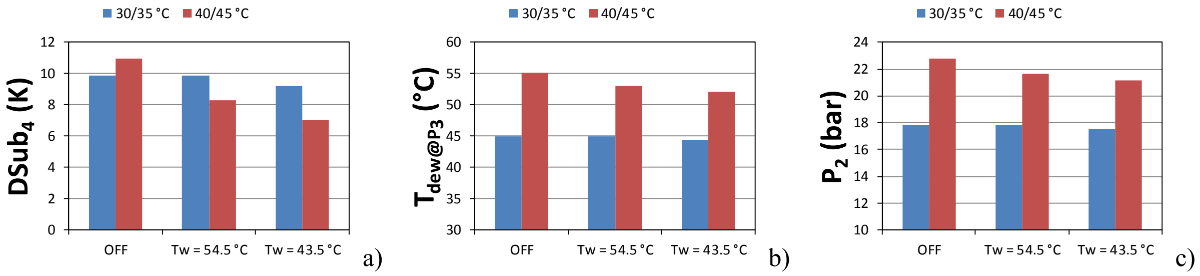

Figure 8: a) Degree of subcooling at the condenser outlet, b) dew point temperature at the desuperheater outlet pressure and c) discharge pressure, for low and medium temperature applications under the heating mode of operation and for different DHW inlet temperatures. The use of the LVHEx is not considered

6. CONCLUSIONS

This paper presents an experimental study of a R407C liquid-to-water heat pump with the possibility of domestic hot water production. Experimental results for typical operating conditions of liquid-to-water heat pumps (EN 14511-2, 2013) in the heating mode of operation (low and medium temperature applications) were presented and discussed. Experiments were also performed in order to evaluate the effect of using or not a liquid-vapor heat exchanger. The following conclusions are obtained:

- The use of a LVHEx without including a refrigerant tank produced a considerably drop of the degree of subcooling at the condenser outlet. For the experimental conditions tested, using a LVHEx lead to lower heating powers and lower compressor's electric powers than when the LVHEx was not used, but the COP remained nearly the same.
- When DHW production was active, results suggest that partial condensation inside the desuperheater occurred, especially for the low DHW inlet temperature and high condensation temperature experiments. As a result, a lower amount of liquid refrigerant built up in the condenser, leading to lower subcooling degrees at the condenser outlet and lower condensation pressures.
- Decreasing the DHW inlet water temperature led to higher DHW heating powers but lower condenser heating powers, being the total heating power nearly constant.
- The total heating power (condenser and desuperheater) and compressor's electric power were slightly higher when DHW was not considered than when DHW was active; however, the COP improved slightly when DHW production was considered.

NOMENCLATURE

C_p	Specific heat capacity	$(J \cdot kg^{-1} \cdot K^{-1})$
COP	Coefficient of performance	(–)
DSub	Degree of subcooling	(K)
h	Specific enthalpy	$(J \cdot kg^{-1})$
$\dot{m}$	Mass flow rate	$(kg \cdot s^{-1})$
P	Pressure	(bar)
PR	Pressure ratio	(–)
$\dot{Q}$	Heat Power	(W)
T	Temperature	(°C, K)
$\dot{V}$	Volume flow rate	$(m^3 \cdot s^{-1})$
$\dot{V}_{comp}$	Compressor's displacement	$(m^3 \cdot s^{-1})$
$\dot{W}_{elec}$	Electric power	(W)

Greek symbol

η	Efficiency	(–)
ρ	Density	$(kg\ m^{-3})$

Subscript

C	Condenser

comp	Compressor
dew	Dew point
DS	Desuperheater
E	Evaporator
i	Inlet
o	Outlet
PGW	Propylene-glycol water mixture
ref	Refrigerant
Total	Condenser and desuperheater
vol	Volumetric
W	Water

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