Analysis of the impact of different operating conditions on the performance of a reversible heat pump with domestic hot water production

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Highlights

- A validated mathematical model for a liquid-to-water heat pump is presented
- The model considers actual dimensions and heat transfer features of all components
- A LVHX is beneficial when DHW is produced
- An optimum subcooling degree that maximizes the system performance is predicted
- Different control strategies may be considered with or without DHW production
Analysis of the impact of different operating conditions on the performance of a reversible heat pump with domestic hot water production

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ABSTRACT

This paper presents the mathematical modelling of a liquid-to-water heat pump with scroll compressor, brazed plate heat exchangers, additionally built-in liquid-vapor heat exchanger (LVHX) and a desuperheater for domestic hot water (DHW) production. The refrigerant is the zeotropic mixture R407C and the liquid used in the outdoor loop is a propylene-glycol water mixture. Developed mathematical model is validated on experimental data and used as a tool for the heat pump analysis. Simulation results are obtained for the effect of the degree of superheat at the evaporator outlet, the subcooling degree at the condenser outlet, the effect of using or not the LVHX and the effect of using or not the desuperheater for DHW for typical operating conditions of liquid-to-water heat pumps (EN-14511-2, 2011) in the cooling and
heating modes (low and medium temperature applications). Results show that the effect of the
degree of superheat or the decision on the suitability of using or not a LVHX may be different
for heat pumps that include or not a desuperheater for DHW. In particular, if DHW is a
priority, the use of a LVHX is recommended because it leads to higher COP (or EER) values
as well as to higher DHW heating powers.

**Keywords:**
Heat pump, mathematical model, desuperheater, liquid-vapor heat exchanger, experimental
results

**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>A</td>
<td>Area (heat exchanger)</td>
<td>(m²)</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of performance (heating)</td>
<td>-</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat capacity</td>
<td>(J·kg⁻¹·K⁻¹)</td>
</tr>
<tr>
<td>d</td>
<td>Diameter</td>
<td>(m)</td>
</tr>
<tr>
<td>e</td>
<td>Thickness</td>
<td>(m)</td>
</tr>
<tr>
<td>EER</td>
<td>Energy efficiency ratio (cooling)</td>
<td>-</td>
</tr>
<tr>
<td>$F_t$</td>
<td>Correction factor</td>
<td>-</td>
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<td>Symbol</td>
<td>Description</td>
<td>Unit</td>
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<td>--------</td>
<td>-------------------------------------------------</td>
<td>---------------</td>
</tr>
<tr>
<td>$g$</td>
<td>Gravity acceleration</td>
<td>(m·s$^{-2}$)</td>
</tr>
<tr>
<td>$H$</td>
<td>Height</td>
<td>(m)</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>Mass flow rate</td>
<td>(kg·s$^{-1}$)</td>
</tr>
<tr>
<td>$\dot{m}_{\text{flux}}$</td>
<td>Mass flux</td>
<td>(kg·s$^{-1}$·m$^{-2}$)</td>
</tr>
<tr>
<td>$M$</td>
<td>Molecular weight</td>
<td>(kg·kmol$^{-1}$)</td>
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<tr>
<td>$N$</td>
<td>Compressor speed</td>
<td>(s$^{-1}$)</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure</td>
<td>(bar)</td>
</tr>
<tr>
<td>$p_{\text{red}}$</td>
<td>Reduced pressure</td>
<td>(-)</td>
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<tr>
<td>$PR$</td>
<td>Pressure ratio</td>
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<tr>
<td>$\dot{q}$</td>
<td>Heat flux</td>
<td>(W·m$^{-2}$)</td>
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<tr>
<td>$\dot{Q}$</td>
<td>Heat transfer rate</td>
<td>(W)</td>
</tr>
<tr>
<td>$R_p$</td>
<td>Plate roughness</td>
<td>(μm)</td>
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<tr>
<td>$T$</td>
<td>Temperature</td>
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<td>$U$</td>
<td>Overall heat transfer coefficient</td>
<td>(W·m$^{-2}$·K$^{-1}$)</td>
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<tr>
<td>$\nu$</td>
<td>Specific volume</td>
<td>(m$^3$·kg$^{-1}$)</td>
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<td>$\dot{V}$</td>
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<td>(m$^3$·s$^{-1}$)</td>
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<tr>
<td>$X$</td>
<td>Vapor quality</td>
<td>-</td>
</tr>
<tr>
<td>$\dot{W}$</td>
<td>Power</td>
<td>(W)</td>
</tr>
</tbody>
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**Greek symbols**
$\alpha$  Convective heat transfer coefficient  \( (\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}) \)

$\Delta p$  Pressure drop  \( \text{(bar)} \)

$\Delta T$  Temperature difference  \( \text{(K)} \)

$\Delta T_{ln}$  Logarithmic mean temperature difference  \( \text{(K)} \)

$\Delta X$  Vapor quality difference  \(-\)

$\lambda$  Thermal conductivity  \( (\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}) \)

$\rho$  Density  \( (\text{kg} \cdot \text{m}^{-3}) \)

$\eta$  Efficiency  \(-\)

**Subscripts**

boil  Boiling phase (evaporation of saturated liquid)

C  Condenser

Comp  Compressor

DHW  Domestic hot water

E  Evaporator

elec  Electric

in  Inlet

is  Isentropic

L  Liquid

ln  Logarithmic

LVHX  Liquid vapor heat exchanger
1. INTRODUCTION

Energy consumption (mostly electricity) for driving the vapor compression devices, new regulations about refrigerants and their replacements, the new system design and optimization of main components are current research topics of refrigeration systems in HVAC applications. Some of those topics are included in the ASHRAE Research Strategic Plan 2010 – 2018 [1]. To introduce major breakthroughs in any of these research lines large number of system designs need to be evaluated, either through simulations or building prototypes and performing measurements. The second option is obviously expensive and time consuming. Hence, the development cost of vapor compression systems can be reduced by using proper simulation models and solvers, as pointed out by Beshr et.al. [2] and Negrão and Hermes [3].
Mathematical modelling of vapor compression systems requires the combination of thermodynamic and heat transfer calculations as well as experimental confirmation to prepare the model as a useful tool for such analyses. The models can be used by engineers to predict the interactive effect of the components size on the system design and its operational characteristics. The capability to predict these effects can enhance the ability of engineers to make wise decisions in the design process and thereby shortening the design cycle, as pointed out by Qiao et al. [4]. The level of the model depends on the kind of analyses intended to be performed. Balance between the model complexity and calculation speed is important for the model choice. The research in this field is comprehensive. Some models relevant for the present research and developed by other authors will be mentioned here. Cheung and Braun [5] developed gray-box component models for eight different cooling systems for the purpose of research of faults impact on performance for systems that have been previously tested under normal and faulted conditions. Overall, the models provide accurate predictions over a wide range of conditions. Negrao and Hermes [3] presented the simulation-based design methodology for household refrigerators and freezers focused on energy savings and cost reduction. The system simulation model was validated against experimental data with discrepancies not exceeding ±10% error bounds. The optimization methodology was used to size the condenser and evaporator areas and the insulation thickness, as also to provide the minimum system cost for a target energy consumption and to study the impact of the compressor efficiency and stroke volume on the overall system cost and performance. A new model for predicting the energy performance of heat pumps and chillers, aimed to achieve good accuracy results by the use of catalog data was presented and published by Scarpa et al. [6]. The model was validated with water-to-water and air-to-water heat pumps and chillers. The validation showed that the model is able to predict the performance within 10% in accuracy, the same accuracy range found in previous literature for more complex models. The
program is extremely fast in calculations and is able to predict the performance of heat pumps in heating and cooling modes even out of nominal boundary conditions (mass flow rates and secondary fluid inlet temperatures). Hengel et al. [7] developed and validated a simulation model, for the analysis of air-source heat pump with desuperheater for domestic how water production (DHW) in order to compare the annual performance for different working conditions to reference system conditions. By simulations they presented energy saving potentials of heat pumps with desuperheaters compared to the reference systems without desuperheater. The effect of using or not a liquid-vapor heat exchanger (LVHX) was not considered in their analysis. Ghoubali et al. [8] performed a simulation study of a heat pump with different refrigerants coupled to buildings. The overall heat transfer coefficients of the different heat exchangers were obtained from linear regression analyses of experimental results, rather than from general correlations. They considered the use of a desuperheater for DHW but the effect of using a LVHX was not taken into account.

The previous literature review reveals a lack of studies about the effect of using LVHX in heat pumps with DHW production under different operating conditions. Moreover, the effects of operating parameters such as the degree of superheat at the evaporator outlet or the degree of subcooling at the condenser outlet are not generally considered.

The goal of this paper is to present the mathematical modelling of a liquid-to-water heat pump which methodology can be applied on similar heat pumps and chillers (water-to-water or liquid-to-water) in order to predict the system performance, components selection or optimization and system design. The model is validated against experimental data from a commercial liquid-to-water heat pump. Later, the model is used to predict the effect of different parameters on the system performance as well as the impact of using a LVHX under different operating conditions.

2. DESCRIPTION OF THE HEAT PUMP AND ITS MAIN COMPONENTS
The considered heat pump is a liquid-to-water heat pump for heating and cooling and also for 
DHW production (Figure 1). The heat pump is reversible and includes a scroll compressor, 
brazed plate heat exchangers (indoor and outdoor heat exchangers in the function of 
condenser/evaporator and desuperheater), a coaxial LVHX, a 4-way valve and a thermostatic 
expansion valve. The refrigerant is the zeotropic mixture R407C. The liquid used in the 
outdoor loop is a 15% of propylene-glycol and water mixture (mass basis).

The compressor is a hermetic scroll compressor with a displacement of 9.44 m$^3$/h for a 
rotational speed of 2900 rpm. The desuperheater has 30 effective plates. Indoor and outdoor 
heat exchangers are identical brazed plate heat exchangers with 20 effective plates.

In the heating (cooling) mode of operation of the heat pump, the outdoor heat exchanger 
behaves as an evaporator (condenser), whereas the indoor heat exchanger behaves as a 
condenser (evaporator).

The LVHX is used with the goal of analyzing the effect on the system performance of using 
or not this heat exchanger. According to Figure 1, when the LVHX is used (not used) the 
valves A1, A2, B1 and B2 are opened (closed), and the valves A3 and B3 are closed (opened).

The 4-way valve is used for the reversion between the winter and the summer cycles.

3. THERMODYNAMIC AND HEAT TRANSFER MODEL OF THE HEAT PUMP

The mathematical model takes into account specific data and dimensions of components with 
goal to predict the actual operating conditions of the heat pump, which follows the model 
description published by researches Janković et al. [9]. The following general assumptions are 
made:

- The heat losses in refrigerant pipes are negligible (pipes are well insulated);
- Heat transfer between the heat exchangers and surroundings is negligible;
Pressure drops in the refrigerant pipes between main components are negligible.

Total heat transfer coefficients for all heat exchangers are calculated according to the following equation:

\[
U_{\text{heat exchanger}} = \frac{1}{\frac{1}{\alpha_{\text{refrigerant}}} + \frac{e_{\text{wall}}}{\lambda_{\text{wall}}} + \frac{1}{\alpha_{\text{second flow medium}}}}.
\]  

(1)

In the following sections the model equations are given for the heating mode of operation; i.e. the outdoor and indoor heat exchangers behave as the evaporator and condenser, respectively. For the sake of brevity, the equations for the cooling mode of operation are not given, but they can be easily obtained by introducing minor changes in the model equations. Essentially, these modifications involve interchanging the condenser and evaporator refrigerant-side equations (for heat transfer and pressure drop) for the indoor and outdoor heat exchangers, respectively. The mathematical model has been programmed using Engineering Equation Solver (EES) [10]. Heat transfer correlations for calculating the heat transfer coefficients in the evaporator, condenser, desuperheater and LVHX are summarized in table 1.

3.1 Hermetic scroll compressor

From the manufacturer’s data two efficiencies were defined as a function of the pressure ratio \((PR)\) to predict the refrigerant mass flow rate and electric power for the compression process.

The volumetric efficiency was obtained by a simple regression analysis, with a coefficient of determination \(R^2\) of 96.5%:

\[
\eta_{\text{vol}} = 1.04783 - 0.028886 \cdot PR
\]  

(2)

For the overall efficiency, a higher order polynomial fit as a function of pressure ratio was obtained, with a coefficient of determination \(R^2\) of 91.6 %:
\[ \eta_{ov} = 0.257 + 0.366 \cdot PR - 0.09517 \cdot PR^2 + 0.00679 \cdot PR^3 \]  
\[ (3) \]

Pressures at the compressor’s outlet and at inlet define the pressure ratio:
\[ PR = \frac{p_{[2]}}{p_{[1]}} \]  
\[ (4) \]

The refrigerant state at the compressor outlet was obtained using the next two equations:
\[ \eta_{s} = \frac{(h_{[2]s} - h_{[1]})}{h_{[2]} - h_{[1]}} \]  
\[ (5) \]
\[ \eta_{ls} = 0.8302 - 0.0332 \cdot PR - 6.551 \cdot 10^{-5} \cdot T_{dew@p_{[1]}} \]  
\[ (6) \]

where the presented coefficients given in equation (6) were obtained from linear regression of experimental results as a function of the pressure ratio and the dew-point temperature at the compressor suction pressure. The refrigerant mass flow rate \( \dot{m}_{\text{ref}} \) is calculated from the compressor displacement \( \dot{V}_{\text{comp}} \), the volumetric efficiency \( \eta_{\text{vol}} \) and the refrigerant density at the compressor inlet \( \rho_{[1]} \):
\[ \dot{m}_{\text{ref}} = \dot{V}_{\text{comp}} \cdot \eta_{\text{vol}} \cdot \rho_{[1]} \]  
\[ (7) \]

The electric power of compressor \( \dot{W}_{\text{comp,elec}} \) is obtained from the overall efficiency and the isentropic power:
\[ \eta_{ov} = \frac{\dot{W}_{ls}}{\dot{W}_{\text{comp,elec}}} = \frac{\dot{m}_{\text{ref}} \cdot (h_{[2]s} - h_{[1]})}{\dot{W}_{\text{comp,elec}}} \]  
\[ (8) \]

The idea is to use only compressor manufacturer data about overall efficiency in order to calculate the input electric to drive the compressor. The manufacturer overall efficiency data is based on the compressor isentropic power, so the total engaged electric power is obtained by dividing isentropic power of the compressor with overall efficiency from equation (8).
3.2 Thermostatic expansion valve

A thermostatic expansion valve with external equalizer is used. The valve is well insulated; therefore, an isenthalpic process \( h_\text{e} = h_\text{r} \) is considered. It is assumed that the thermostatic expansion valve is adjusted in order to control the degree of superheat at the evaporator’s outlet.

3.3 Evaporator (outdoor heat exchanger)

The total heat transfer rate for the evaporation process is evaluated by applying energy balances for the refrigerant flow and for the propylene-glycol water mixture:

\[
\dot{Q}_E = \dot{m}_\text{ref} \cdot (h_\text{e} - h_\text{r}),
\]  
\[
\dot{Q}_E = \dot{m}_\text{PGW} \cdot c_\text{PGW} \cdot (T_{\text{PGW,E,in}} - T_{\text{PGW,E,out}}),
\]  
\[
\dot{Q}_E = \dot{Q}_E_{\text{boil}} + \dot{Q}_E_{\text{sup}}.
\]

The evaporation heat transfer process is observed in two parts: heat transfer in the boiling region and heat transfer in the superheated region. Researches Dutto et al. [11] and Primal et al. [12] suggested a model with the logarithmic mean temperature difference for both parts. The same method is applied in Longo and Gasparella [13] for the vaporization process by applying the following equations:

\[
\Delta T_{E,\text{boil}} \frac{\dot{Q}_E}{\dot{Q}_E_{\text{boil}}} + \Delta T_{E,\text{sup}} \frac{\dot{Q}_E_{\text{sup}}}{\dot{Q}_E_{\text{sup}}}.
\]  
\[
A_E = A_{E,\text{boil}} + A_{E,\text{sup}},
\]

where the boiling heat transfer rate is obtained as:
\[ \dot{Q}_{E,\text{boil}} = A_{E,\text{boil}} \cdot U_{E,\text{boil}} \cdot \Delta T_{E,\text{in,boil}}, \quad (14) \]

\[ \Delta T_{E,\text{in,boil}} = \frac{(T_{\text{PGW,E,m}} - T_{\text{Ex}(x=1,p=p[8])}) - (T_{\text{PGW,E,out}} - T[7])}{\ln \left( \frac{T_{\text{PGW,E,m}} - T_{\text{Ex}(x=1,p=p[8])}}{T_{\text{PGW,E,out}} - T[7]} \right)}, \quad (15) \]

and for the superheating zone:

\[ \dot{Q}_{E,\text{sup}} = A_{E,\text{sup}} \cdot U_{E,\text{sup}} \cdot \Delta T_{E,\text{in,sup}}, \quad (16) \]

\[ \Delta T_{E,\text{in,sup}} = \frac{(T_{\text{PGW,E,in}} - T[8]) - (T_{\text{PGW,E,m}} - T_{\text{Ex}(x=1,p=p[8])})}{\ln \left( \frac{T_{\text{PGW,E,in}} - T[8]}{T_{\text{PGW,E,m}} - T_{\text{Ex}(x=1,p=p[8])}} \right)}, \quad (17) \]

The total heat transfer coefficients for the boiling region \( U_{E,\text{boil}} \) and for the superheating region \( U_{E,\text{sup}} \) are calculated according to equation (1). The temperature of the propylene-glycol water mixture \( T_{\text{PGW,E,m}} \) at the end of the boiling region (start of the superheated region) is calculated from an energy balance:

\[ T_{\text{PGW,E,m}} = T_{\text{PGW,E,in}} - \frac{m_{\text{ref}} \cdot c_{p,\text{ref}} \left( T[8] - T_{\text{Ex}(x=1,p=p[8])} \right)}{\dot{m}_{\text{E,PGW}} \cdot c_{p,\text{E,PGW}}}. \quad (18) \]

The heat transfer coefficient on the refrigerant side for the boiling process \( \alpha_{E,\text{ref,boil}} \) is calculated according to the Copper correlations [14]. The heat transfer coefficient for the propylene-glycol water mixture \( \alpha_{E,\text{PGW}} \) and for the superheating phase (refrigerant single phase – vapor) in the evaporator are calculated using correlations from Thonon [15].

3.4 Condenser (indoor heat exchanger)
In general, the condensation process in this type of equipment can be divided into three zones: vapor desuperheating, condensation and liquid subcooling. However, in our case, the refrigerant side surface temperature of the condenser was always lower than the refrigerant dew point temperature. As a result, only two zones (condensation and liquid subcooling) are considered in the analysis.

The total heat transfer in the condenser is given by the three following equations:

\[
\dot{Q}_C = \dot{m}_{\text{ref}} \cdot (h_{[4]} - h_{[5]}), \quad (19)
\]

\[
\dot{Q}_C = \dot{m}_{C,W} \cdot c_{p,C,W} \cdot (T_{C,W,\text{out}} - T_{C,W,\text{in}}), \quad (20)
\]

\[
\dot{Q}_C = \dot{Q}_{C,\text{cond}} + \dot{Q}_{C,\text{sub}}. \quad (21)
\]

The surface of the condenser is also divided into two zones:

\[
A_C = A_{C,\text{cond}} + A_{C,\text{sub}}. \quad (22)
\]

For the condensation process it is considered that the subcooling region starts somewhere inside (denoted as ‘*’) of the condenser (presumably close to the outlet port), then:

\[
\dot{Q}_{C,\text{cond}} = \dot{m}_{\text{ref}} \cdot (h_{[4]} - h_{[*\text{group}[5]})), \quad (23)
\]

\[
\dot{Q}_{C,\text{cond}} = \dot{m}_{C,W} \cdot c_{p,C,W} \cdot (T_{C,W,\text{out}} - T_{*}^*), \quad (24)
\]

\[
\dot{Q}_{C,\text{cond}} = A_{C,\text{cond}} \cdot U_{C,\text{cond}} \cdot \Delta T_{C,\text{cond},\text{ln}} \cdot F_t. \quad (25)
\]

The logarithmic mean temperature difference for the condensation process is calculated by the following equation:
\[ \Delta T_{\text{cond,ln}} = \frac{\left( T_{C(x=1,p_p[4])} - T_{C,W,\text{out}} \right) - \left( T_{C(x=0,p_p[5])} - T^*_{C,W} \right)}{\ln \left( \frac{T_{C(x=1,p_p[4])} - T_{C,W,\text{out}}}{T_{C(x=0,p_p[5])} - T^*_{C,W}} \right)}. \] (26)

Similarly, for the subcooling part of the condenser, the next equations are obtained:

\[ \dot{Q}_{\text{sub}} = \dot{m}_{\text{ref}} \left( h_{C,W}^* - h_s \right), \] (27)

\[ \dot{Q}_{\text{sub}} = m_{C,W} c_{p,C,W} \left( T^*_{C,W} - T_{C,W,\text{in}} \right), \] (28)

\[ \dot{Q}_{\text{sub}} = A_{\text{sub}} U_{C,\text{sub}} \Delta T_{\text{sub,ln}} F_t, \] (29)

\[ \Delta T_{\text{sub,ln}} = \frac{\left( T_{C(x=0,p_p[5])} - T^*_{C,W} \right) - \left( T_{[s]} - T_{C,W,\text{in}} \right)}{\ln \left( \frac{T_{C(x=0,p_p[5])} - T^*_{C,W}}{T_{[s]} - T_{C,W,\text{in}}} \right)}. \] (30)

In this paper correction factor \((F_t)\) for the condensation of R407C has been included as suggested by researchers Shah and Focke [16]. The correction factor is calculated as a function of the number of transfer units, according to Mancin et al. [17] and [18]. The heat transfer coefficient for the water side \((\alpha_{C,W})\) is calculated from Thono [15] and the computation approach suggested by Mancin et al. [17] is used for the refrigerant side.

3.5 Liquid – vapor heat exchanger

Energy balances for the liquid and vapor sides of this heat exchanger are described by the following equations:

\[ \dot{Q}_{\text{LVHX}} = \dot{m}_{\text{ref}} \left( h_{[s]} - h_{[s]} \right). \] (31)

\[ h_{[s]} - h_{[s]} = h_{[s]} - h_{[s]} \] (32)
For the liquid phase, the heat transfer coefficient is calculated from the widely known correlations for friction factor and Nusselt number according to Petukhov [19] and Gnielinski [20], respectively. The heat transfer coefficient for the vapor flow in the concentric annular duct was calculated according to the Petukhov and Roizen [21] and Petukhov [19], using a modified form of the correlation for turbulent flow from Gnielinski [20].

### 3.6 Desuperheater

In this paper, the mathematical model of the desuperheater was simplified based on the domestic hot water temperature values used for the heat pump analysis covered in section 4. As a result, it is considered that during hot water production the refrigerant will not condense. The heat exchanger is modelled as a single-phase flow heat exchanger in both sides using the widely used correlations of Thonon [15].

### 3.7 Pressure drops in the refrigerant circuit

The pressure drops on the refrigerant side in the evaporator, condenser and desuperheater are calculated based on accordance to Collier and Thome [22].

The total pressure drop in the evaporator ($\Delta p_{E,t}$) is calculated from the frictional ($\Delta p_{E,f}$), momentum ($\Delta p_{E,a}$), gravity ($\Delta p_{E,g}$) and the manifolds and ports ($\Delta p_{E,c}$) pressure drops:

$$\Delta p_{E,t} = \Delta p_{E,f} + \Delta p_{E,a} + \Delta p_{E,g} + \Delta p_{E,c}.$$  (33)

The momentum and gravity pressure drops are estimated by the homogeneous model for the two-phase flow as follows:

$$\Delta p_{E,a} = \dot{m}_{flux,E}^2 \left( v_{v,E} - v_{l,E} \right) \cdot \Delta X_E,$$  (34)

$$\Delta p_{E,g} = g \cdot \rho_{m,E} \cdot H_E,$$  (35)
where \( \Delta X_E \) is the vapor quality change between inlet and outlet of the evaporator. The average two-phase density between inlet and outlet is calculated by the following equation:

\[
\rho_{m,E} = \left[ \frac{X_{m,E}}{\rho_{V,E}} + \frac{1 - X_{m,E}}{\rho_{L,E}} \right]^{-1},
\]

where \( X_{m,E} \) is the average vapor quality between inlet and outlet of the evaporator.

The pressure drops in the inlet and outlet manifolds and ports are empirically estimated, in accordance with Collier and Thome [22], as follows:

\[
\Delta p_{E,c} = \frac{1.5 \cdot \dot{m}_{\text{flx},E}^2}{2 \cdot \rho_{m,E}},
\]

The frictional pressure drop calculation was based on the experimental data from Longo and Gasparella [13] and Longo and Gasparella [23]:

\[
\Delta p_{E,f} = 1553 \cdot \frac{\dot{m}_{\text{flx},E}^2}{2 \cdot \rho_{m,E}},
\]

In the condenser, a similar procedure was followed. For calculating the frictional pressure drops the correlation from Longo [24] was used. The same model is set for the desuperheater where, in this case, the momentum pressure rise is neglected because it is considered that refrigerant will not condense (no phase change).

Pressure drops in the LVHX are also included in the mathematical model, the friction factor for the liquid and vapor sides (concentric annular duct) are calculated (for turbulent flow) from Petukhov [19] and Gnielinski [20].

The pressure drop in the 4-way valve is considered to be equal to 14000 Pa, based on manufacturer data.
3.8 System performance of the heat pump

The coefficient of performance (COP) and energy efficiency ratio (EER) are calculated from the heating $Q_C$ and cooling $Q_E$ capacities (winter/summer mode), the heat power used for domestic hot water (DHW) (if any) and the total electric power consumption of the heat pump:

$$COP = \left( \frac{Q_C + Q_{DHW}}{W_{TOTAL,elec}} \right),$$  \hspace{1cm} (39)

$$EER = \left( \frac{Q_E + Q_{DHW}}{W_{TOTAL,elec}} \right),$$  \hspace{1cm} (40)

The total power consumption includes the compressor and three pumps power: for the propylene-glycol water mixture in the outdoor heat exchanger, for the water pump in the indoor heat exchanger, and also for the DHW in the desuperheater. For all three pumps, the electrical power is obtained from the manufacturer data as a function of the volumetric flow rate using the following general equations with coefficients $a$, $b$ and $c$:

$$\dot{W}_{elec,pump} = -a \cdot V^2 + b \cdot V + c,$$  \hspace{1cm} (41)

$$\dot{W}_{TOTAL,elec} = \dot{W}_{comp,elec} + \dot{W}_{elec,pump,PGW} + \dot{W}_{elec,pump,W} + \dot{W}_{elec,pump,DHW},$$  \hspace{1cm} (42)

4. VALIDATION OF THE MODEL

The model was validated against steady state tests performed on an experimental station with the heat pump described in section 2. The experimental station was equipped with proper instrumentation. Figure 1 shows the location of the different sensors used in the experimental setup and table 2 summarize their main characteristics.
For the validation of the mathematical model a set of 19 steady state tests were considered. For these tests the LVHX and desuperheater were not used and the heat pump was run for the heating mode of operation (winter cycle) as described in section 2.

The propylene-glycol water mixture (PGW) is pumped to the outdoor heat exchanger from a 750 L PGW tank. Similarly, water is pumped from a 280 L tank to the indoor heat exchanger. The PGW and water flow rates through the outdoor and indoor heat exchangers were controlled manually. 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).

Experiments were carried out with PGW supply temperatures (outdoor heat exchanger) from 8 to 12 °C and water supply temperatures (indoor heat exchanger) from 18ºC to 36.5ºC. The PGW and water flow rates were nearly constant during all experiments, with values equal to $(6.85 \pm 0.05) \times 10^{-4} \text{ m}^3/\text{s}$ and $(5.9 \pm 0.1) \times 10^{-4} \text{ m}^3/\text{s}$, respectively. The degree of superheat at the evaporator outlet was adjusted initially by manual rotation of the side stem of the thermostatic expansion valve. During all the experiments, the degree of superheat at the evaporator outlet was nearly constant and equal to $3.0 \pm 0.5$ K.

During each experiment, steady state conditions were reached within a period of 2-3 hours while data recording during the next 60 minutes was done for measured temperatures and pressures.

The inputs used in order to validate the model were: inlet temperature and volumetric flow rate of PGW at the outdoor heat exchanger (evaporator), inlet temperature and volumetric flow rate of water at the indoor heat exchanger (condenser), degree of superheat controlled by thermostatic expansion valve and subcooling degree measured at condenser outlet.
Figures 2 and 3 present the deviations between experimental and simulation results (pressures and temperatures). Deviation of the temperatures between experimental and numerical results are inside 2.5 K bands. Pressure deviations shown in Figure 3 are mostly within 0.3 bar bands. Figure 3 also shows that experimental and simulation results predict non-negligible pressure drops in the high and low pressure refrigerant circuit of the heat pump (for example, average pressure differences of 0.4 bar were measured between compressor outlet and condenser outlet). Since pressure drops affect adversely the system performance, it was necessary to include them in the mathematical model (as described in section 3.7).

Figure 4 shows that the deviation for total electric power is in the range from 5 to 10%. It should be clarified that the electric power measured by the power transducer includes the compressor, pumps, electronic control board and additional electric components; the later values are not included in the simulation calculations, so the simulated results are expected to be slightly lower than the experimental ones.

It can be seen, that the accordance between experimental and simulation results are reasonable. As a result the model can be used as an effective tool for further heat pump analyses.

5. SIMULATION RESULTS AND DISCUSSION

The effect that different operating conditions or components design have on the heat pump performance can be evaluated by using the developed simulation mathematical model.

In this section the effect of the degree of superheat at the evaporator outlet, the subcooling degree at the condenser outlet, the effect of using or not the LVHX and the effect of using or not the desuperheater for DHW production will be analyzed for typical operating conditions of liquid-to-water heat pumps. The conditions used for these analyses are selected rating
conditions (EN-14511-2, 2011) for liquid-to-water heat pumps in the cooling and heating modes (low and medium temperature applications). Unless otherwise stated, the following parameters were considered: degree of superheat of 3 K; subcooling degree of 2 K; and, for those cases with hot water production, water temperature values at the inlet and outlet of the desuperheater of 45 °C and 60 °C, respectively. The propylene-glycol water mixture concentration for all cases was 15% (mass basis).

5.1 Heating mode of operation (low temperature)

The analysis is initially performed for the low temperature heating mode of operation, considering inlet/outlet temperatures of 0/-3ºC for the outdoor heat exchanger (evaporator) and 30/35ºC for the indoor heat exchanger (condenser).

Figure 5 shows the effect of the subcooling degree and degree of superheat on the COP when the desuperheater is used (DHW-ON) and not used (DHW-OFF) for DHW production. Results are presented when the LVHX is used (LVHX-ON) or not used (LVHX-OFF). From these results it can be seen that using the desuperheater leads to higher COP values of around 2 % for the different analyzed cases. On the other hand, using the LVHX (LVHX-ON) has a slight negative effect on the system coefficient of performance when the desuperheater is not used (DHW-OFF) but a slight positive effect when it is used (DHW-ON). Increasing the degree of superheat has a negative effect on the COP, so its value should be adjusted to a minimum that guarantees that no liquid reaches the compressor. By increasing the subcooling degree, it is seen that, initially, the COP increases up to a maximum value and then the COP decreases with larger subcooling degree values. There is an optimum subcooling degree value that maximizes the system COP. For the cases analyzed, the optimum subcooling degree is around 3 K. For lower subcooling degree values than the optimum one, the COP variations are negligible; however, for subcooling degrees above 6 K the COP can decrease by more than 2 % with respect to the optimum value.
It is known that in systems without a liquid receiver, condenser subcooling can be obtained by using a fraction of the condenser heat transfer area for cooling the refrigerant below the saturation temperature. This can be somehow varied during the refrigerant charging procedure as shown by Pottker and Hrnjak [25].

Figure 6 presents on a $p-h$ diagram the heat pump cycle for subcooling degree values of 1.5, 3.5 and 7 K when neither the LVHX nor the desuperheater are used. As shown in figure 6, for the three cases considered, the evaporator pressure and the compressor suction conditions are nearly identical. Also, figure 6 shows that increasing the subcooling degree leads to higher condensation temperatures and pressures and, consequently, higher pressure ratios. As a result, we should expect higher specific compressor works and lower COP values when the subcooling degree increases. On the other hand, increasing the subcooling degree also leads to lower temperature (enthalpy) values at the condenser outlet which would lead to higher heating effects per unit mass and COP values. For the results shown in figure 6, the COP reaches a maximum for a subcooling degree of 3 K, as a result of the trade-off between increasing the heating effect and compression work.

Figure 7 shows the corresponding heat transfer rates in the condenser and desuperheater (when DHW-ON) for the same cases analyzed in figure 5. Focusing on the DHW-OFF case, it can be seen that increasing the degree of superheat has a negative effect on the heating power of the heat pump, whereas increasing the subcooling degree has a positive effect. Moreover, using the LVHX has a negligible effect on the heating power, as both curves (LVHX-ON and LVHX-OFF) collapse into a single one. This can be clearly seen in the $p-h$ diagrams shown in figure 8a), where it can be seen that using the LVHX has two opposite effects on the heating power. One hand, using the LVHX leads to higher discharge temperature which will lead to higher heating powers; but on the other hand, if the LVHX is used, the compressor suction temperature and specific volume increase, which leads to lower mass flow rates and,
consequently, lower heating powers. Results in figure 7 show that both effects nearly cancel each other, and the heating power is almost unaffected by the LVHX.

A similar analysis has been performed when the desuperheater is used for hot water production (DHW-ON case) and the results of this analysis are also included in figure 7. As shown in the p-h diagram of figure 8b) using the LVHX leads to higher compressor suction and discharge temperatures which finally leads to a greater amount of heat that can be used for hot water production, as confirmed in figure 7. As a consequence, the heating power in the condenser decreases in a similar value, being the total power (condenser and desuperheater) nearly the same. The results also show that, for the DHW-ON case, increasing the subcooling degree leads to low variations for the heat transfer values in the condenser but higher heat transfer values for hot water production in the desuperheater. With regards to the effect of the degree of superheat, figure 7 shows that increasing the degree of superheat leads to higher powers for hot water production which may be interesting in those situations where hot water production is a priority.

5.2 Heating mode of operation (medium temperature)

A similar analysis is performed for the medium temperature heating mode of operation, considering inlet/outlet temperatures of 0/-3 °C for the outdoor heat exchanger (evaporator) and 40/45 °C for the indoor heat exchanger (condenser). The results are collected in figures 9 and 10.

Figure 9 shows similar trend of COP’s to the ones obtained for the low temperature applications; however, the COP values for the medium temperature applications are around 23% and 25% (0.67 and 0.70) lower for the DHW-ON and DHW-OFF cases, respectively. For the DHW-OFF case, the effect on the COP of the LVHX is insignificant, but for the DHW-ON case, using the LVHX has a slight positive effect on the COP of the heat pump.
Figure 10 also shows that the trends for the heat transfer values are similar to the ones obtained for the low temperature heating mode of operation; though the heating power values in the condenser are lower (16% and 5.1% in average for the DHW-ON and DHW-OFF cases, respectively), while the heating power that can be used for DHW is higher (around 0.8 kW or 90%) due to the higher vapor temperatures at the compressor outlet. Once again, using the LVHX has a positive effect on the heating power that can be used for DHW, whereas it has a slight negative effect on the heating power obtained in the condenser. The effects of the subcooling and superheating degree follow similar trends to those obtained for the low temperature mode of operation.

5.3 Cooling mode of operation

Finally, an analysis is performed for the cooling mode of operation, considering inlet/outlet temperatures of 30/35 °C for the outdoor heat exchanger (condenser) and 12/7 °C for the indoor heat exchanger (evaporator).

Figure 11 shows the effect of the subcooling degree and degree of superheat on the cooling energy efficiency ratio (EER) when the desuperheater is used (DHW-ON) and not used (DHW-OFF) for DHW production. Results are shown when the LVHX is used (LVHX-ON) or not used (LVHX-OFF). It can be seen that using the desuperheater improves clearly the system EER by factors between 5 to 10%. On the other hand, using the LVHX has a negligible effect for the DHW-OFF case, but leads to EER values which increase around 1 % for the DHW-ON case. The optimum subcooling degree value that maximizes the system EER, is around 4 K; though, for the range of analyzed values the EER does not drop more than 1% from its maximum value. Finally, increasing the degree of superheat has a negative effect on the EER for the DHW-OFF case, but a positive effect when the desuperheater is used for DHW production. Then, the degree of superheat could be varied based on the mode of operation (DHW-OFF or DHW-ON) in order to optimize the system performance.
Figure 12 shows the corresponding heat fluxes in the evaporator and desuperheater for the same cases analyzed in figure 11. For the DHW-OFF case, it can be seen that increasing the degree of superheat has a negative effect on the cooling power of the heat pump, whereas increasing the subcooling degree has a positive effect. Using the LVHX has a negligible effect on the cooling power, as both curves (LVHX-ON and LVHX-OFF) collapse into a single one. A similar analysis has been performed when the desuperheater is used for DHW production. Now, it can be seen that using the LVHX leads to the greater amount of heat that can be used for DHW production. Results also show that increasing the degree of superheat leads to higher heat transfer values for DHW production in the desuperheater.

6. CONCLUSION

This paper presents numerical results of a reversible liquid-to-water heat pump with domestic hot water production. The modelling procedure considered the actual dimensions and heat transfer features of all main components of the heat pump. Numerical results were validated against experimental data with discrepancies not exceeding ±10% error bounds. By using the mathematical model, results for the effect of different variables were investigated for typical operating conditions of liquid-to-water heat pumps (EN-14511-2, 2011) in the cooling and heating modes (low and medium temperature applications) of operation including or not DHW production. The following conclusions are obtained:

- Increasing the degree of superheat at the evaporator leads to lower COP values and heating powers for all the heating modes of operation analyzed (low and medium temperature applications) either with or without DHW production. Similar results are obtained for the EER and the cooling power under the cooling mode of operation; however, the EER increases with the degree of superheat when DHW production is active.
In general, there is an optimum subcooling degree value that maximizes the system performance. The optimum value depends on the system operating conditions, but for the analyzed cases it was always lower than 5 K. Anyway, for the range of cases analyzed in the paper, the effect of the subcooling degree on the system performance is in general lower than 2%

Using a LVHX has a slight negative effect on the system performance when DHW production is not active. However, if DHW is a priority, the use of a LVHX is recommended because it leads to higher COP (or EER) values as well as to higher DHW heating powers.

After all, it is shown that different control strategies (degree of superheat or LVHX) may be implemented if DHW production is a priority or not.

REFERENCES


### Table 1: Heat transfer coefficient correlations

<table>
<thead>
<tr>
<th>Authors</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooper [14]</td>
<td>$\alpha = 55 \frac{P_{\text{red}}^{0.12 - 0.2 \log_{10} R_p}}{(- \log_{10} P_{\text{red}})^{-0.55}} q^{0.67} M_{\text{ref}}$</td>
</tr>
<tr>
<td>Thonon [15]</td>
<td>$Nu = 0.2946 , Re^{0.7} , Pr^{1/3}$</td>
</tr>
<tr>
<td>Mancin et al. [17]</td>
<td>$\alpha_{\text{Nu}} = 0.943 \left( \frac{\rho_L}{\rho_v} \right)^{0.3685} \left( \frac{\mu_L}{\mu_v} \right)^{0.2363} \left( 1 - \frac{\mu_v}{\mu_L} \right)^{2.144} Pr^{-0.1} \left( 1 + 1.128 X^{0.817} \right)$</td>
</tr>
<tr>
<td></td>
<td>$\alpha_A = \alpha_{\text{LO}} \left( 1 + 1.128 X^{0.817} \right)$</td>
</tr>
<tr>
<td></td>
<td>$\alpha_{\text{LO}}$ from Thonon [15] using liquid properties with the total flow rate</td>
</tr>
<tr>
<td>Petukhov [19]</td>
<td>$f = \frac{1}{(0.79 \ln(Re) - 1.64)^2}$</td>
</tr>
<tr>
<td>Gnielinski [20]</td>
<td>$Nu = \frac{f/8 , Re , Pr}{1.07 + 12.7 \sqrt{f/8 , (Pr^{2/3} - 1)}}$</td>
</tr>
<tr>
<td>Petukhov and Roizen [21]</td>
<td>$Nu = Nu_{\text{ct}} \left( \frac{d_{\text{outer}}}{d_{\text{inner}}} \right)^{-0.16}$</td>
</tr>
<tr>
<td></td>
<td>$Nu_{\text{ct}}$ from Gnielinski [20]</td>
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</table>

### Table 2: Types of sensors used and accuracies

<table>
<thead>
<tr>
<th>Variable</th>
<th>Instrumentation</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant temperatures</td>
<td>Pt100 Class A</td>
<td>± 0.5 K</td>
</tr>
<tr>
<td>PGW and water temperatures</td>
<td>Pt100 Class B 1/10</td>
<td>± 0.1 K</td>
</tr>
<tr>
<td>PGW and water flowrates</td>
<td>Electromagnetic flow meter</td>
<td>± 0.2% reading</td>
</tr>
<tr>
<td>Pressure</td>
<td>Pressure transmitter</td>
<td>± 0.2 bar</td>
</tr>
<tr>
<td>Heat pump power consumption</td>
<td>Power transducer</td>
<td>± 0.45% reading</td>
</tr>
</tbody>
</table>
FIGURES

Figure 1 Schematic representation of the heat pump with LVHX (cooling and heating mode)
(Numbers in square brackets are related to refrigerant states in the model description)
Figure 2 Deviation between simulated and experimental values of temperatures (R407C)

Figure 3 Deviation between simulated and experimental values of pressures (R407C)
Figure 4 Deviation between simulated and experimental values of total input power for the heat pump (electric power of compressor and three pumps) (R407C)
Figure 5 Influence on the system performance (COP) of the subcooling and superheating degrees (R407C). The low temperature heating mode of operation is considered.
Figure 6 p-h diagram of the heat pump refrigeration process (R407C) for different subcooling values.
Figure 7 Influence on the heating and DHW heating powers of the subcooling and superheating degrees (R407C). The low temperature heating mode of operation is considered. (Arrows beside curves indicate that the right side ordinate should be used)
Figure 8 p-h diagrams for the refrigerant cycles (R407C) with and without LVHX for the cases with and without DHW production.
Figure 9 Influence on the system performance (COP) of the subcooling and superheating degrees (R407C). The medium temperature heating mode of operation is considered.
Figure 10 Influence on the heating and DHW heating powers of the subcooling and superheating degrees (R407C). The medium temperature heating mode of operation is considered. (Arrows beside curves indicate that the right side ordinate should be used)
Figure 11 Influence on the system performance (EER) of the subcooling and superheating degrees (R407C). The cooling mode of operation is considered.
Figure 12: Influence on the cooling power and the DHW heating power of the subcooling and superheating degrees (R407C). The cooling mode of operation is considered. (Arrows beside curves indicate that the right side ordinate should be used)