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Analysis of different control strategies for improved performance at off design operation in a CO_2 heat pump water heater

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ABSTRACT

A commercial heat pump water heater using CO_2 as refrigerant is considered in this work. A theoretical model was first validated via experimental data, and then tested to evaluate various operations. A fundamental and well-known parameter for optimizing the system performance is controlling the high stage pressure. In this work, a logic control was derived to maximize performance even under off-design conditions. Poor stratification, or improper sizing of the water system may lead to compressor outlet temperature too high, which should be limited to avoid problems with lubricants. Usually, the maximum compressor discharge temperature is 140 °C and, to control this temperature, this work proposes and compares three different control logic, acting on the back pressure valve or on the internal heat exchanger (IHX) with the aim of gaining efficiency and reducing the operating cost.

Keywords: Refrigeration, Carbon Dioxide, Heat Pump, COP, Energy Efficiency

1. INTRODUCTION

Heat pumps for hot water production using carbon dioxide as refrigerant are gaining market share in the last years. Both the use of heat pumps in place of gas boilers, and of a natural refrigerant in place of hydrofluorocarbons, are central actions for the reduction of greenhouse gas emissions. A heat pump working with CO_2 requires transcritical operation, which is a drawback for the efficiency of the refrigerating cycle, partly mitigated by the effective heat transfer of CO_2 in the supercritical region. The open literature reports some papers of authors dealing with CO_2 heat pumps and their control. Wang et al (2013) made tests on a prototype to identify correlations for the optimal gas cooler pressure as a function of the ambient temperature and the water outlet temperature. They claimed that their correlation was not of general purpose, however the empirical method for its fitting is reproducible for other systems. Also Qi et al (2013) gave a correlation for the optimal discharge pressure, claiming it is dependent almost upon the temperature of CO_2 at the gas cooler exit, rather than outdoor temperature.

However, efficiency remains the main goal for the designers of such heat pumps, and among the solutions to improve their COP the Internal Heat Exchanger (IHX) is one of the simplest but very effective. Kim et al (2005) investigated the effect of the IHX size on a water-to-water CO_2 heat pump, finding that a reduction in the optimal discharge pressure as well as refrigerant flow rate and consequently compressor power are related to the size of the IHX.

Also Cao et al (2020) experimentally investigated the effect of a given IHX on the optimal gas cooler pressure from both an energy and exergy point of view. They concluded that the optimal discharge pressure can be reduced thanks to the operation of the IHX, especially at low ambient temperature, or high water inlet temperature or high water outlet temperature. The same authors set up a model to investigate the effect of the IHX at various values of its efficiency, to properly consider the effect of pinch point in the heat exchanger (Ye et al, 2020). They stated that at low water return temperature the pinch point limits the heat exchange capability thus impairing the COP, while they concluded that the increase of the efficiency if the IHX always acts in favour of a reduction of the gas cooler pressure. However, while the IHX is known for improving the efficiency with some refrigerants, it is also known for the probable increase in the discharge temperature. This is particularly the case when return water temperature at the inlet of the gas cooler increases. Otón-Martinez et al (2022) carried out an experimental work to validate their model of a transient heating-up process, observing this behaviour and concluding that higher efficiency of the IHX lead to higher efficiency of the system, reducing gas cooler pressure. In this paper, a CO_2 water heater heat pump with IHX is analysed through a numerical model and then validated via experimental data; three different control rules for its operation are proposed and examined with the aim of increasing performance while controlling the discharge temperature at off-design conditions.

2. THE SYSTEM

A simplified scheme of the Heat Pump is proposed in figure 1, It's a typical transcritical one stage CO_2 cycle with a compressor, gas-cooler, an internal heat exchanger (IHX), high pressure valve (HPV) that allows the pressure control in the gas-cooler and downstream of the evaporator is the liquid receiver. The water circuit is controlled by a pump (WP) with the aim of achieving a specific water temperature at the outlet of the gas cooler. The gas-cooler and the IHX are brazed plate heat exchangers with area of 8.1 m^2 and 0.5 m^2 and a nominal heating capacity of 129 kW kW and 12.9 kW respectively, the volumetric displacement of the semi-hermetic compressor is 27.29 m^3/s at 60 Hz and a finned tube heat exchanger is used as evaporator.

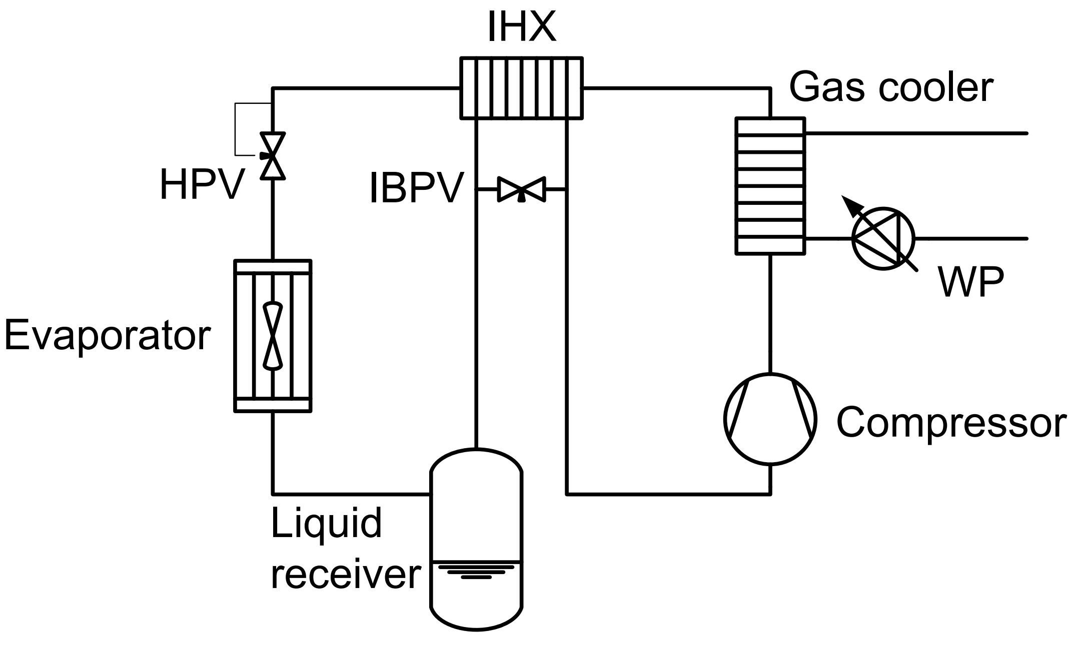

Figure 1 Simplified heat pump water heater scheme

Finally, there is an internal by-pass valve (IBPV) that allows a by-pass of the internal heat exchanger. The IBPV is normally closed; bypassing the heat exchanger allows lower compressor suction temperatures, resulting in lower discharge temperatures. In this paper, analyses are conducted maintaining a limit temperature at the compressor outlet of 140 °C, to avoid degradation of lubricating oil.

3. NUMERICAL MODEL AND VALIDATION

The numerical model for the heat pump behavior has been developed in MATLAB, estimating the thermodynamical properties of the fluids with the software REFPROP (Lemmon E. W. et al., 2018), and then validated via experimental data. The mass flow rate and the power consumption of the compressor are estimated from equation 1, where the coefficients, provided by the manufacturer, are listed in table 1.

$$X = C1 + C2 \cdot To + C3 \cdot Pgc + C4 \cdot To^2 + C5 \cdot To \cdot Pgc + C6 \cdot Pgc^2 + C7 \cdot To^3 + C8 \cdot Pgc \cdot To^2 + C9 \cdot To \cdot Pgc^2 + C10 \cdot Pgc^3 \quad \text{Eq. (1)}$$

Table 1 Coefficients for compressor numerical model.

	$C1$	$C2$	$C3$	$C4$	$C5$	$C6$	$C7$	$C8$	$C9$	$C10$
$\dot{m}$	0.74684	0.01984	$-3.11 \cdot 10^{-3}$	$1.781 \cdot 10^{-3}$	$-1.69 \cdot 10^{-5}$	$9.98 \cdot 10^{-5}$	0	0	0	0
W	-33554.10	-1185.445	1364.262	-11.41558	17.41383	-9.318666	-0.010705	0.032412	-0.048737	0.02816

Mass flow rate and compressor power consumption are corrected for the actual superheating compared to the 10 K value at which the coefficients listed in table (1) are calculated, with Eqs. (2) and (3), respectively

$$\frac{\dot{m}_{new}}{\dot{m}_{data}} = 1 + F\left(\frac{\rho_{suc,new}}{\rho_{suc,data}} - 1\right) \quad \text{Eq. (2)}$$

$$\frac{W_{new}}{W_{data}} = \frac{\dot{m}_{new}}{\dot{m}_{data}} \frac{\Delta \mathrm{h}_{is,new}}{\Delta \mathrm{h}_{is,data}} \quad \text{Eq. (3)}$$

where the correction factor F is assumed to be 0.75, ρ is density, new represents the corrected superheat state, $data$ is the state from the experimental data, and $\Delta \mathrm{h}_{is}$ denotes enthalpy difference in an isentropic process (Illán-Gómez et al., 2021). Regarding the calculation of the gas-cooler and the internal heat exchanger, both streams are discretized in *n* part where the temperature difference is assumed constant. The procedure presented by Otón-Martínez et al. (2022) is used. The inlet conditions (mass flow, pressure, enthalpy) of both fluids are known, and the hot fluid outlet temperature is estimated like in Eq. (4) at the first iteration.

$$T_{hot,out} = \frac{T_{high} + T_{low}}{2} \quad \text{Eq. (4)}$$

where

$$T_{high} = T_{hot,in} \quad \text{Eq. (5)}$$

$$T_{low} = T_{cold,in} \quad \text{Eq. (6)}$$

Then, outlet high enthalpy is calculated as well as the overall heat exchanged; applying the energy conservation equation to each control volume, the thermodynamic variables of all nodes of the cold stream are calculated. The experimental correlation by Bogaert and Böles (1995) is used for the calculation of the local heat transfer coefficient at the average thermophysical properties in each cell and thus the exchange area of each control volume. Finally, an overall exchange area is calculated (A_{calc}) and compared with the experimental area (A_o) to re-calculate $T_{hot,out}$ as shown in Eq. (7) and Eq. (8).

If $A_o < A_{calc}$ Then $T_{high} = T_{hot,out}$ Eq. (7)

If $A_o > A_{calc}$ Then $T_{low} = T_{hot,out}$ Eq. (8)

The convergence criterion is $T_{high} = T_{hot,out} = T_{low}$ (Otón-Martínez et al., 2022).

The HPV is modelled as isenthalpic expansion, the evaporation temperature is considered as an input in this model and the liquid receiver outlet is considered as saturated vapour. Comparisons between the theoretical model and experimental data are presented in Figures 2 to 5.

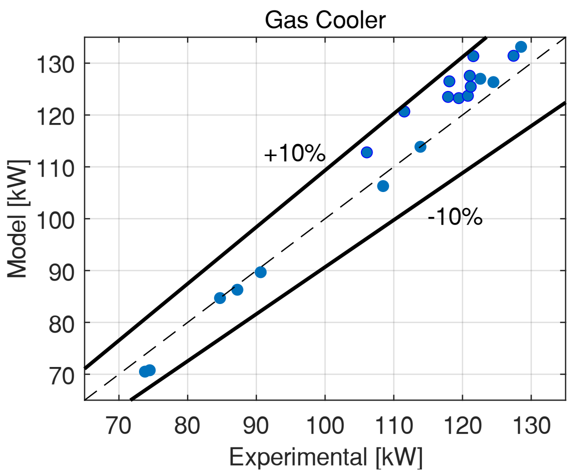

Figure 2 Gas-Cooler heating capacity validation

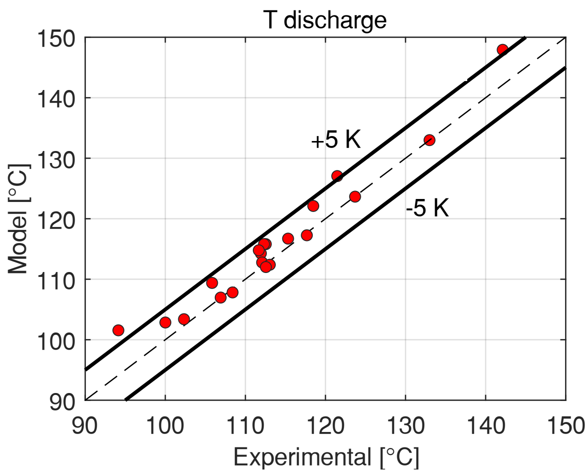

Figure 3 Compressor discharge temperature validation

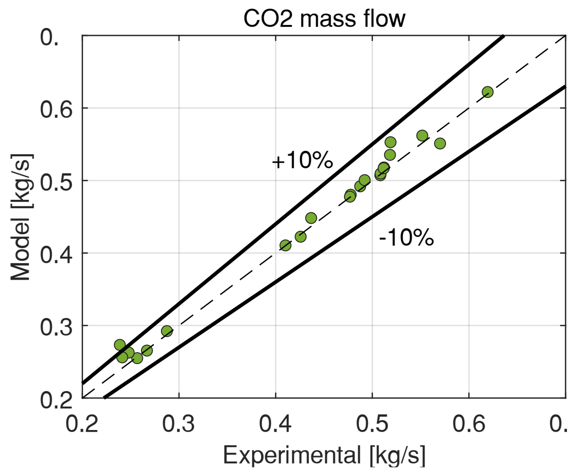

Figure 4 CO2 mass flow validation

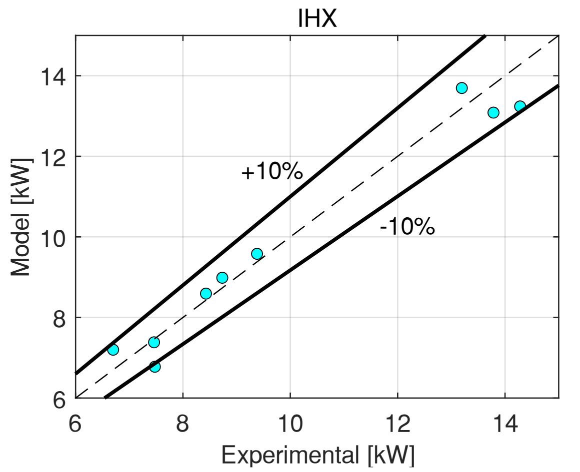

Figure 5 Internal Heat Exchanger validation

4. OFF-DESIGN PERFORMANCE

As mentioned before, the aim is to predict the performance of the heat pump at off-design conditions; in this work, three control logics are proposed and analysed in terms of COP. The conditions for which the control logics must be applied are shown in figure 6; the reference parameter is the compressor discharge temperature (maximum 140 °C) and the water outlet temperature is fixed at 80°C.

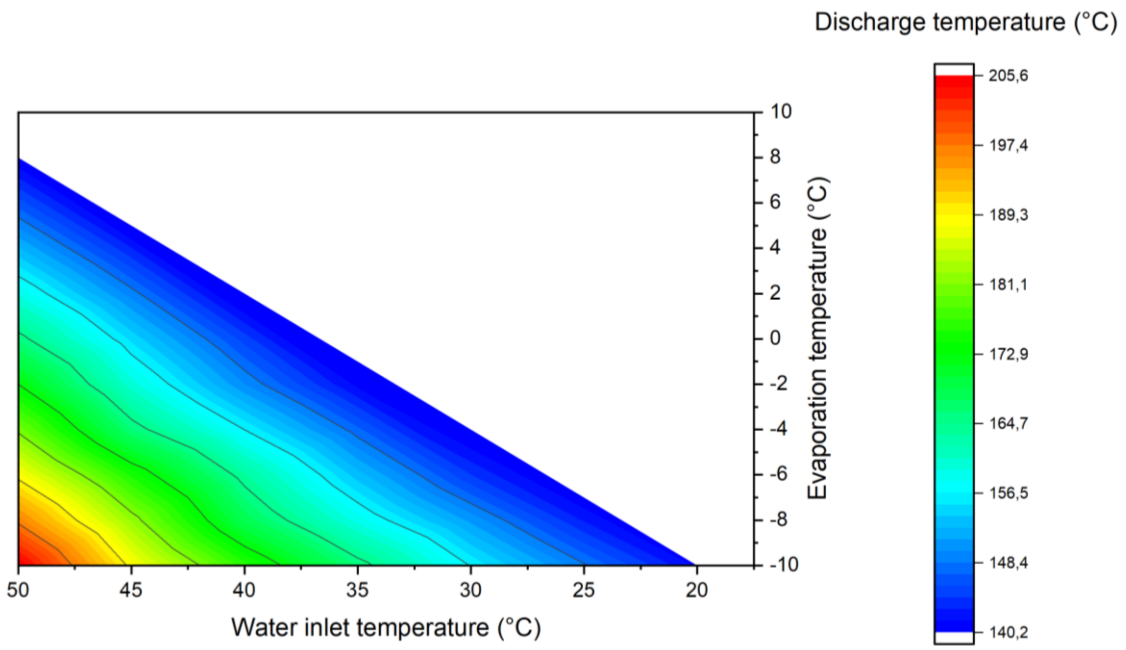

Figure 6 Off-Design conditions

Figure 6 shows the discharge temperature trend as water temperature inlet and evaporation temperature change without any control limitations, so in this study, all conditions included in the coloured map will be considered; this set of conditions will later be referred to as "off-design conditions".

4.1.1 Control rules

Three different control logic rules are proposed:

- **Control rule 1:** limiting the discharge temperature by completely bypassing the IHX **(CR1)**
- **Control rule 2:** limiting the discharge temperature by partially bypassing the IHX **(CR2)**
- **Control rule 3:** limiting the discharge temperature by acting on the HPV **(CR3)**

In off-design conditions, two different behaviours are evident when control rules are applied; in order to analyse them, a comparison of the three different control rules, in term of COP at different gas-cooler pressure is shown in figures 7 and 8.

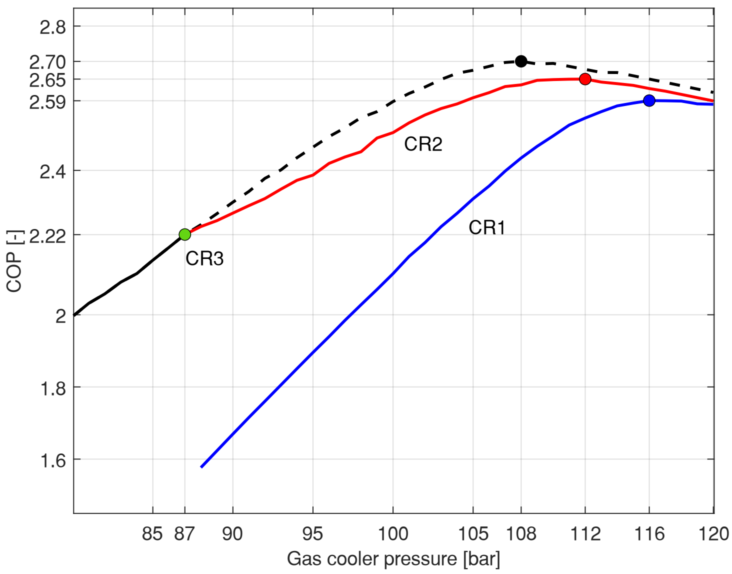

Figure 7 Optimization analysis for water inlet temperature at 40°C and evaporation temperature at -5°C

Figure 7 shows the COP by varying the gas-cooler pressure when the water inlet temperature is 40°C and the evaporation temperature is -5°C; up to 87 bar, the discharge temperature is lower than 140°C, then following the standard heat pump rule maximizing the COP (dotted line) the pressure increase until 108 bar with a corresponding COP of 2.70 (T_{dis} = 159.1 °C); in order to limit the discharge temperature, the standard control cannot be applied, but other logics must be followed. With CR1 (blue line) the IHX is bypassed, the maximum COP of 2.59 (T_{dis}=119.7°C) is reached at a gas-cooler pressure of 116 bar ; with CR2 (red line) an higher COP of 2.65 is reached at a lower gas-cooler pressure of 112 bar, and then CR3 (Green point) where the pressure is limited at 87 bar, the COP is limited to 2.22. Thus, the best control logic is the CR2, i.e. partial by pass of the IHX, followed by CR1 and CR3. It is important to mention that while CR2 and CR3 are control logics that keep the maximum discharge temperature constant at 140°C, for CR1 (IHX completely bypassed) T_{dis} is normally lower because, without an IHX, the discharge temperature does not increase as much as the optimum pressure increases.

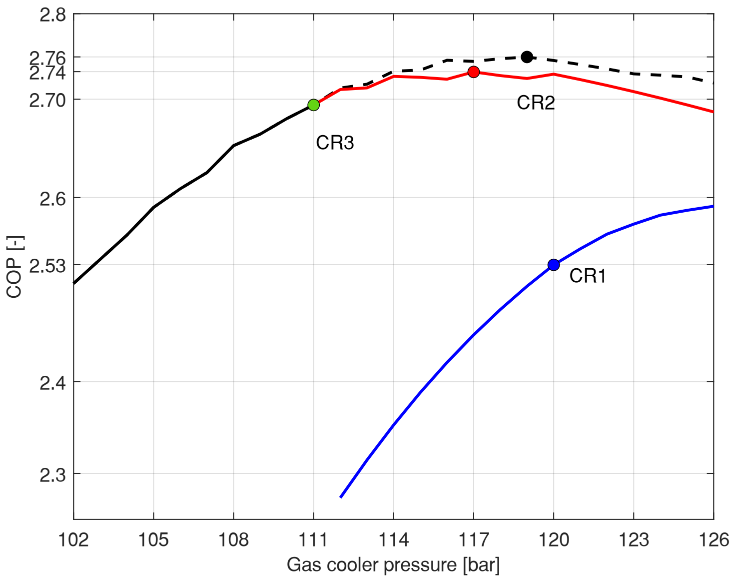

Figure 8 Optimization analysis for water inlet temperature at 47°C and evaporation temperature at 5°C

In figure 8, a different behaviour is shown: while, as above CR2, is the rule with the highest COP (2.74 at 117 bar), in this off-design condition (water inlet temperature at 47°C and evaporation temperature at 5°C) acting on the pressure at the gas cooler (CR3) instead of bypassing even partially the IHX, increases the performance more. Specifically, the COP is 2.70 at 111 bar when CR3 is used and COP 2.53 at 120 bar when CR1 is used. This pressure value has been considered as the maximum gas cooler pressure allowed for the heat pump. Therefore, for this specific case analysed, it can be confirmed that partially bypassing the IHX (CR2) is a control rule that can successfully improve the performance instead of using the complete bypass (CR1) or simply limiting the gas cooler pressure (CR3).

5. RESULTS

Following the optimisation rules described above, an overall comparison in off-design conditions between the CR1 (IHX by-pass) and CR3 (acting on pressure) and between the CR2 (partially IHX by-pass) and CR3 are shown in Figures 9 and 10 respectively.

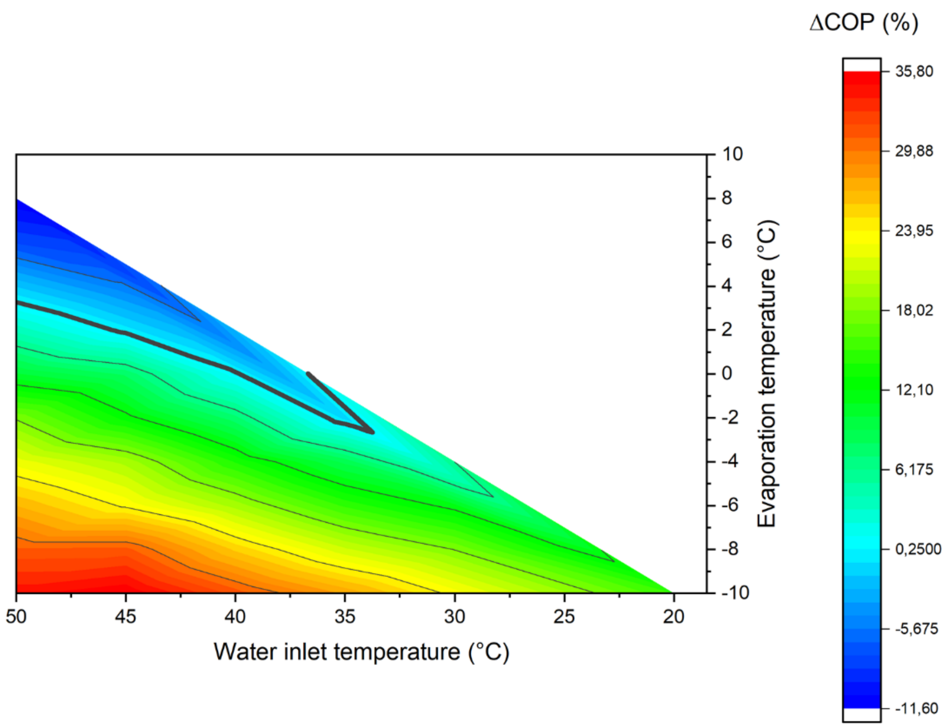

Figure 9 Percentage difference in COP between CR1 (full IHX bypass) and CR3 (gas cooler pressure control)

ΔCOP(%) is defined as the percentage difference between the use of CR1 and CR3; the black line corresponds to ΔCOP(%)=0, so it separates the set of conditions where CR3 gives better performance (ΔCOP(%)<0) in the region above the black line, from the opposite case where CR1 provides better benefits (ΔCOP(%)>0) in the region below the black line. It may be noted that the CR3 gives better performance only for evaporation temperature greater than about -2 °C; one of the causes is the increase in irreversibility as the evaporation pressure decreases when a high superheating is maintained, making it more efficient to reduce superheating as the evaporation temperature decreases. On the other side, the comparison between CR2 and CR3 is represented in Figure 10, as percentage difference ΔCOP(%).

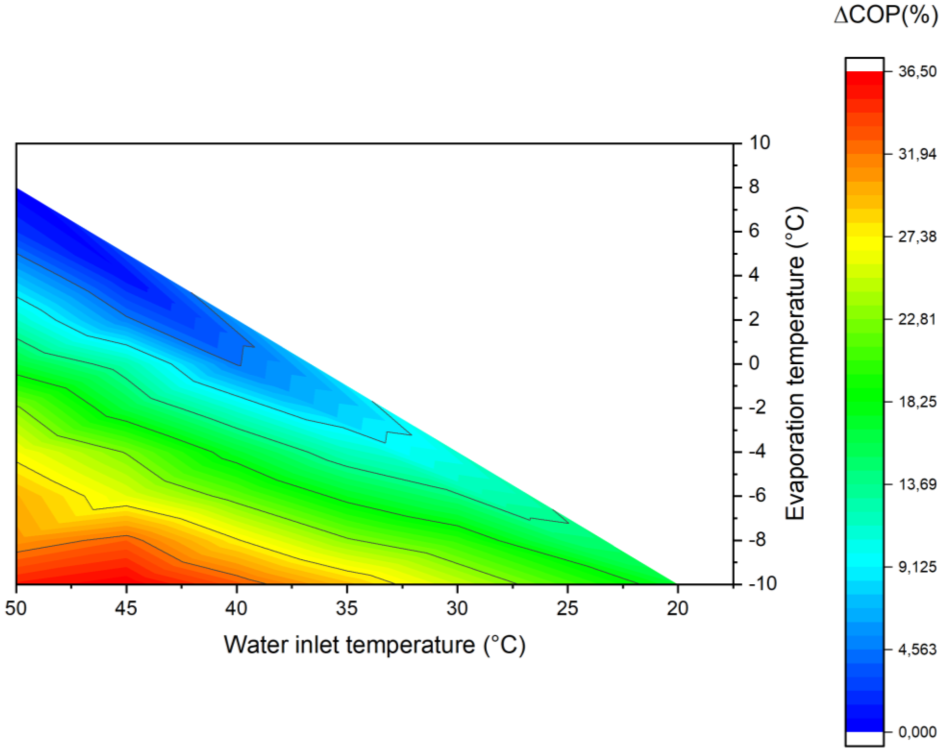

Figure 10 Percentage difference in COP between CR2 (partial IHX bypass) and CR3 (gas cooler pressure control)

Gradual bypass of the IHX (CR2) is the control rule that gives the best performance in all off-design conditions. Nevertheless, an analysis of the difference in terms of COP can still be useful: in fact, it may be profitable to adopt the CR3 anyway for the reduction of investment costs (no by-pass required) and simplification of the control rules. Actually, the performance drop occurs only when the heat pump works in off-design conditions, and a thorough design of the water storage and its stratification can limit the occurrence of such heavy conditions.

CONCLUSIONS

A numerical model for a heat pump water heater has been developed, and then validated via experimental data. Based on the numerical model, and with an outlet water temperature target fixed at 80°C, a set of conditions has been identified as "off-design conditions”; three different control rules are proposed in order to avoid discharge temperature at the compressor greater than 140 °C, compared to each other in terms of efficiency. In the present study, the control logic that gives the best performance during off-design conditions is the CR2 (i.e. gradually adjusting the bypass) increasing performance compared to the others up to 36.5%. Nevertheless, other control logics can be considered if they involve a lower cost of heat pump components and simplicity. Future developments of the work will be the experimental validation of the results, and a more in-depth cost-benefit analysis considering the overall period of work in off-design conditions.

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NOMENCLATURE

COP	Coefficient of performance	A	Area (m^2)
CR1	Control rule 1	h	Specific enthalpy ($kJ \times kg^{-1}$)
CR2	Control rule 2	$\dot{m}$	Mass flow rate ($kg \times s^{-1}$)
CR3	Control rule 3	T_{dis}	Compressor discharge temperature (°C)
HPV	High pressure valve	W	Compressor work per unit time (kW)
IBPV	Internal bay-pass valve	WP	Water Pump
IHX	Internal heat exchanger	ρ	Density (kg x m^{-3})

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