ESTUDIO NÚMERICO Y EXPERIMENTAL DE UNA BOMBA DE CALOR AGUA-AGUA USANDO CO_2 COMO REFRIGERANTE

Presentado por

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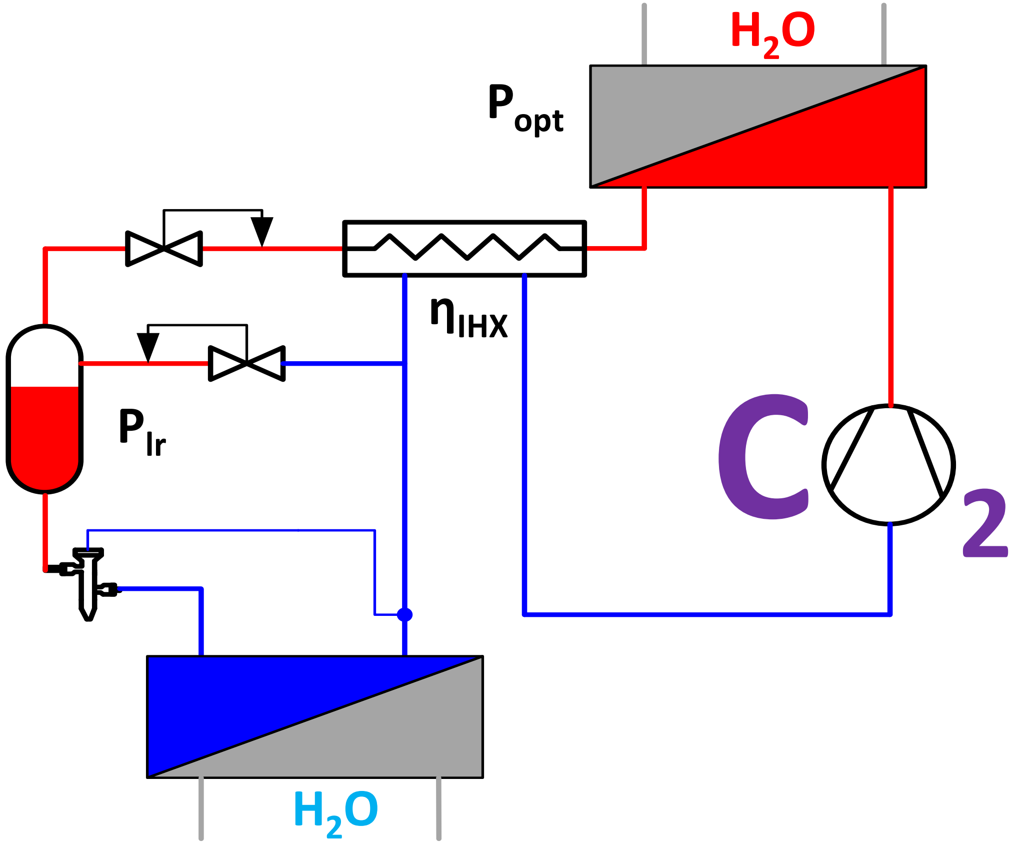
0. Content

INTRODUCTION

1. Introduction

- CO_2 heat pump for hot water generation, EN-14511-2 and UNE-EN-16147
- The optimal pressure (P_{opt})
- Influence of liquid receiver pressure (P_{lr}) and the internal heat exchanger (IHX)
- Comparison of different configurations



STATE OF THE ART

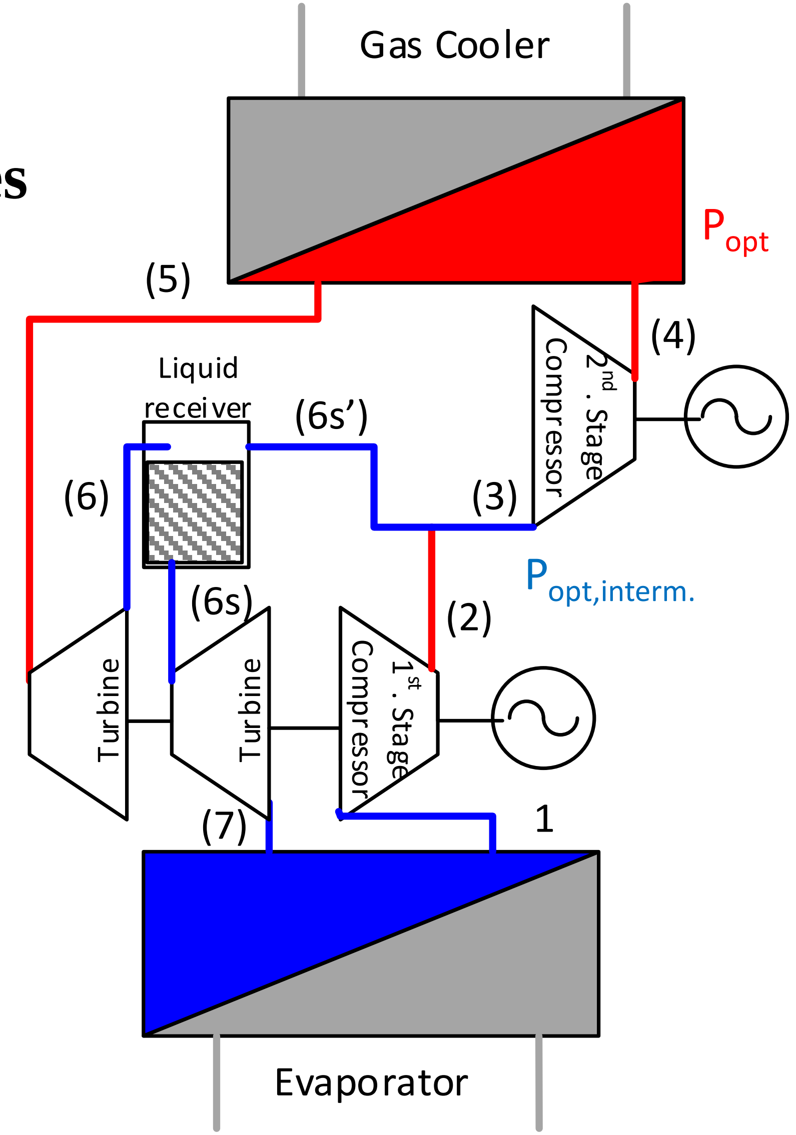
2.1. CO_2 and other refrigerants

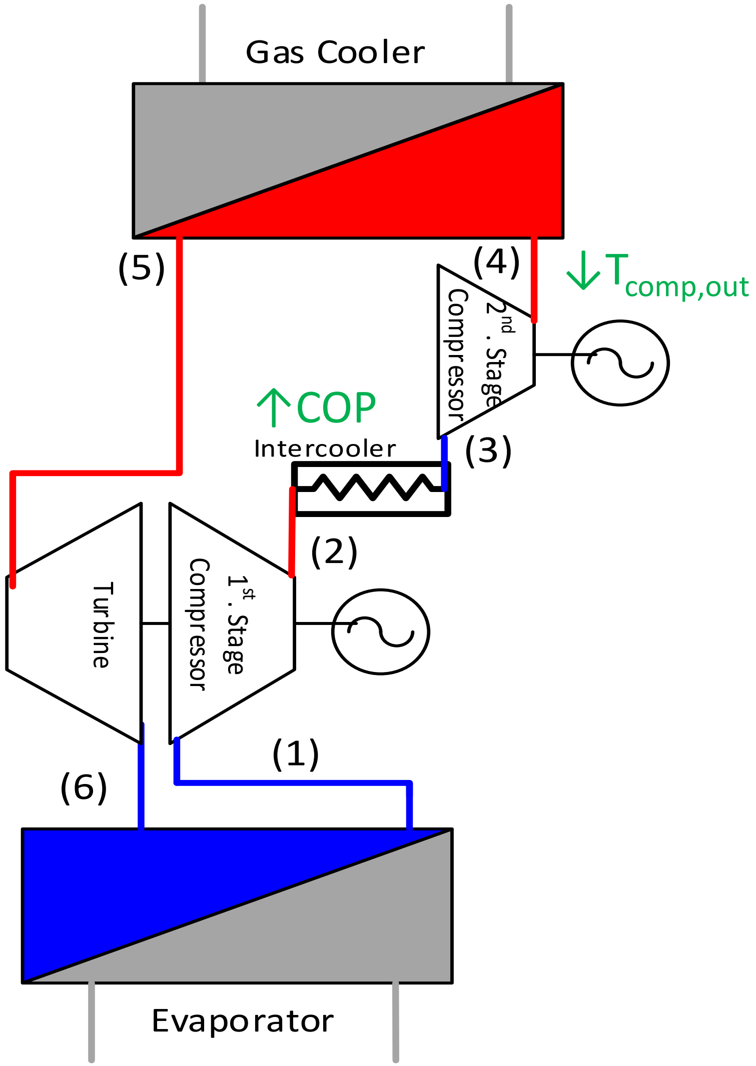
Refrigerant group	Refrigerant examples	Estimated period	ODP	$GWP_{100\ -yrs}$	Atmospheric lifetime (years)	Flammability
CFCs	R11, R12, R13, R115	1930-1990	0.6-1	4750-14400	45 to 1700	Nonflammable
HCFCs	R22, R141b, R124	1930-1990	0.02-0.11	400-1800	1 to 20	Nonflammable
HFCs	R407C, R32, **R41**, R23, *R134a*, *R410A*, R404A	1990-2010	0	100-14900	1 to 300	Nonflammable or middle flammable
HFOs	R1234yf, R1234ze, R1234yz, R513A	2010-present	0	0-573	-	Middle flammable
Natural refrigerants	HC (**R290**, R600, R600a), R1270).		0	0-4	Few days	HCs: Highly flammable
Natural refrigerants	R717		0	0	Few days	Flammable
Natural refrigerants	R744	1830-1930	0	1 (or zero)	Few days	Nonflammable

For nearly all of refrigerants, the critical pressure ranges between 30 bar and 60 bar, except water vapor (221 bar), ammonia (113 bar), and CO_2 (73.8 bar). (Lorentzen, 1994)

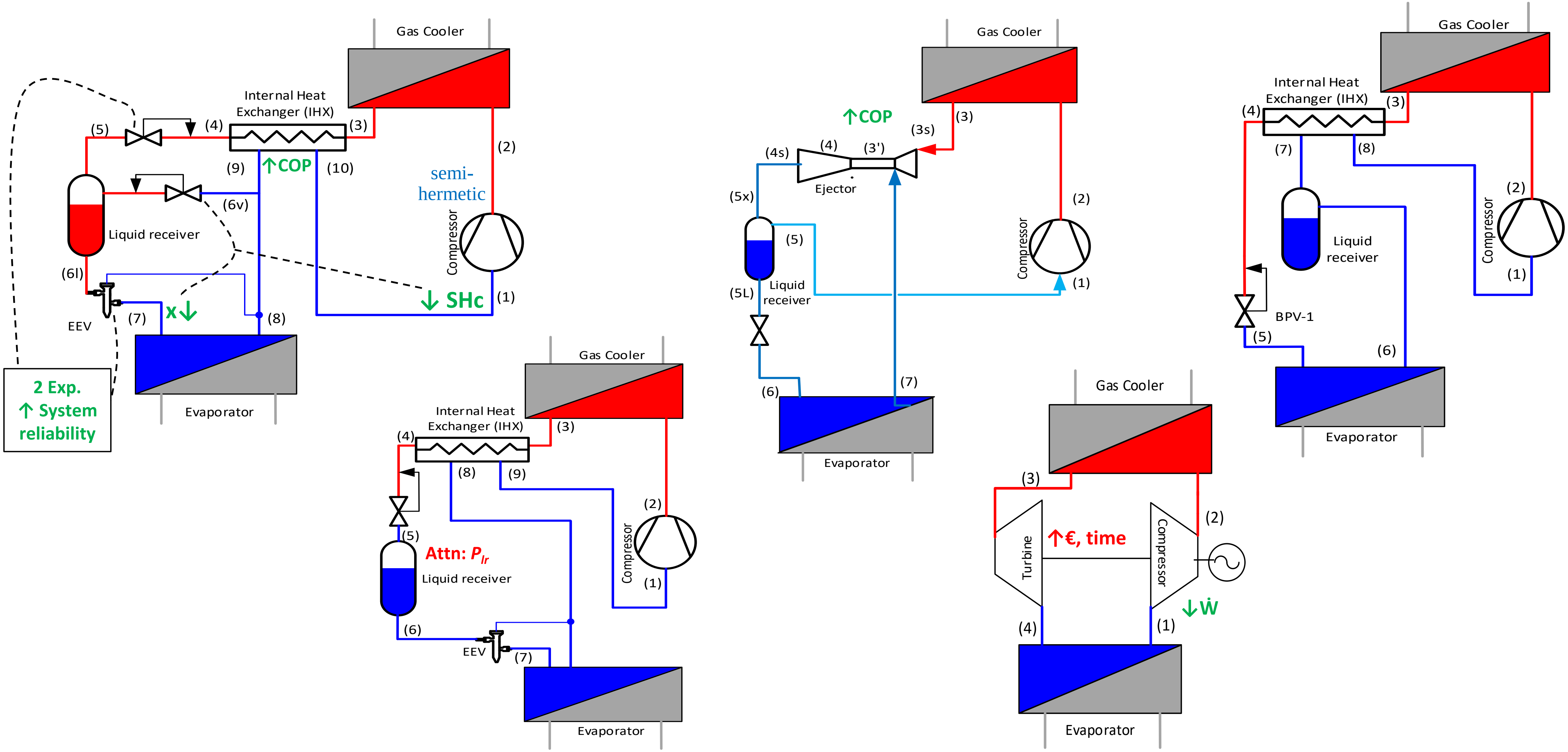
2.2. Different configurations used in CO_2 applications

Multiple-stage cycles



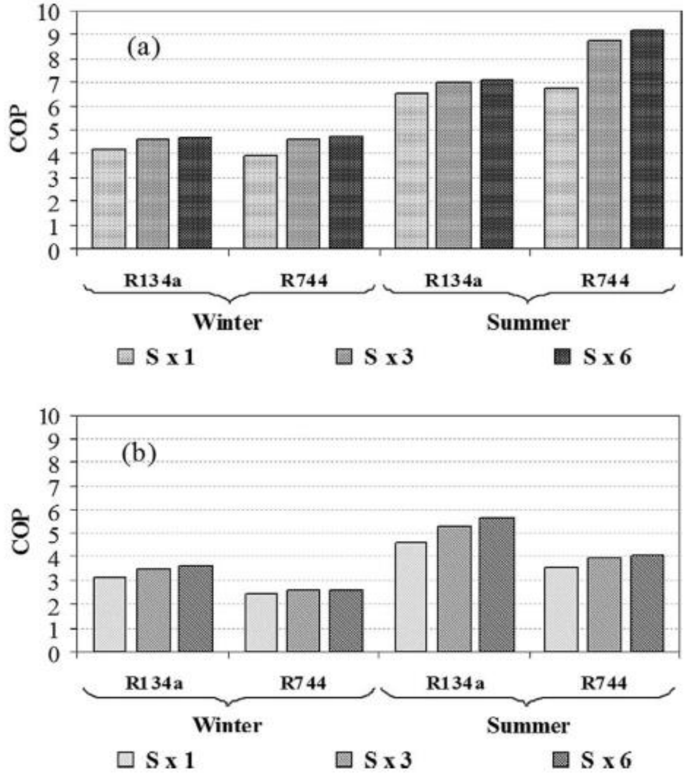


Single-stage cycles

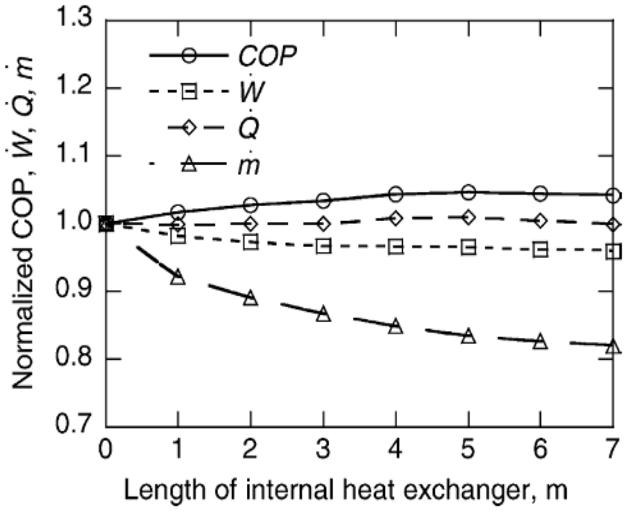


2. State of the art

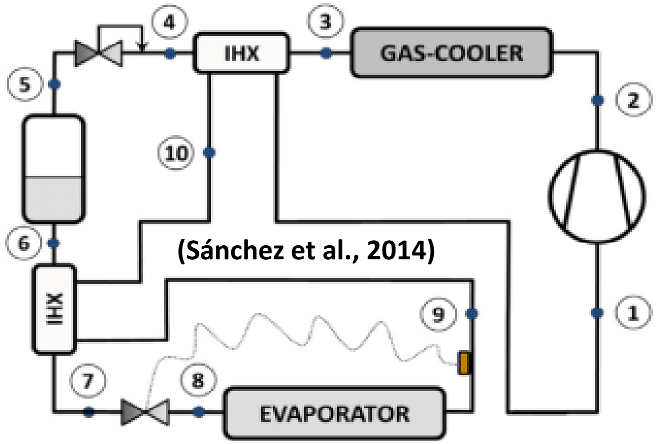
2.3. Heat exchangers optimization

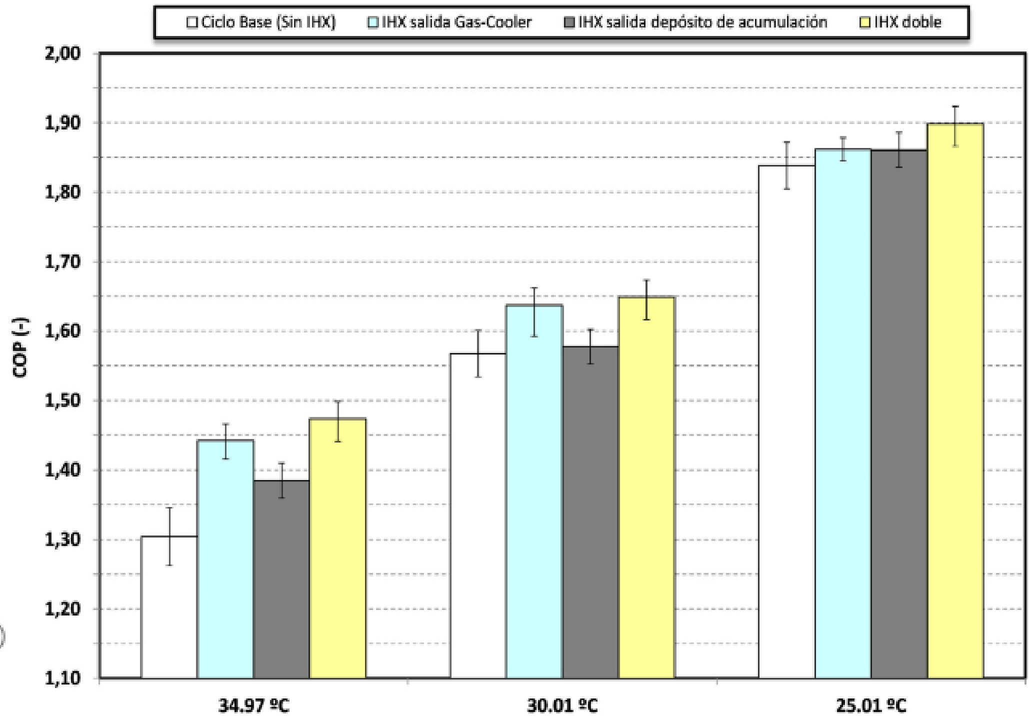


Gas cooler surface. COP values for perfect stratification (a) and for perfect mixing (b) hypothesis. (Cecchinato et al., 2005)



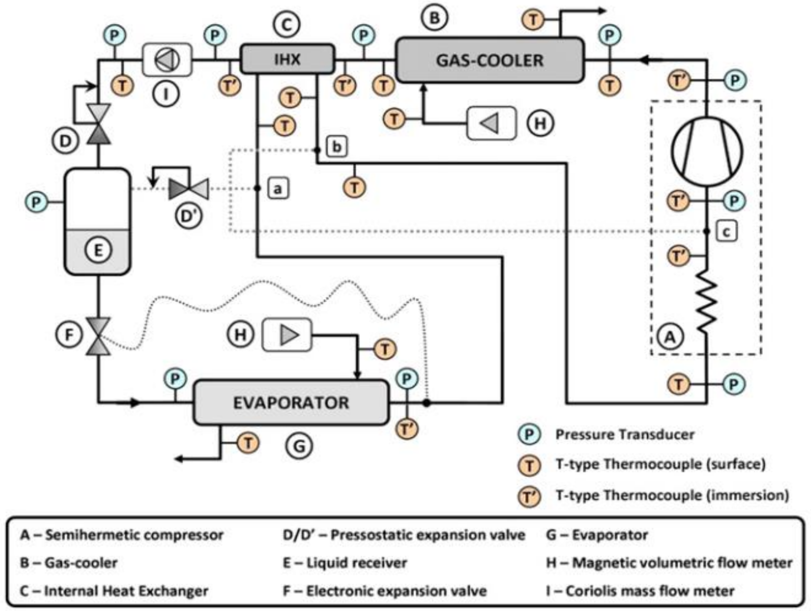
System performances with respect to the length of internal heat exchanger. (Kim et al., 2005)





COP for different configurations of IHX in a refrigeration plant. ($T_{ev,water+glycol,in=5°C}$) (Sánchez et al., 2014)

2.4. The liquid receiver bypass



Test type	Configuration	Evaporator				Gas-cooler		Liquid receiver	Maximum variation						
		$T_{Glic,in}$ (°C)	$q_{Glic,in}$ ($m^3\ h^{-1}$)	P_{ev} (MPa)	ΔP_{ev} (%)	$T_{W,in}$ (°C)	$q_{W,in}$ ($m^3\ h^{-1}$)	$P_{Liq\ Rec}$ (MPa)	TSH (°C)	$\Delta \dot{m}_{GC}$ (%)	$\Delta \dot{m}_{Ev}$ (%)	$\Delta \dot{Q}_{ev}$ (%)	ΔP_C (%)	ΔCOP (%)	ΔT_{Dis} (°C)
1	Inj. before IHX	5.03	1.01	2.59	4.42	24.99	1.01	3.03	28.35	9.0	−24.1	5.10	1.57	3.48	−5.57
1	Inj. after IHX	4.90	1.02	2.59	3.69	25.01	1.02	3.04	25.77	10.9	−24.3	5.12	1.63	3.43	−8.95
1	Inj. Compressor	4.97	1.02	2.59	4.26	25.08	1.02	3.02	25.59	11.45	−23.76	5.34	2.79	2.49	−8.95
2	Inj. before IHX	5.07	1.01	2.73	4.09	34.90	1.03	4.10	31.31	10.5	−28.1	4.10	2.60	1.47	−6.61
2	Inj. after IHX	5.01	0.99	2.76	4.77	35.15	1.02	3.22	26.43	16.5	−35.18	6.78	2.15	4.53	−12.50
2	Inj. Compressor	4.94	1.02	2.74	4.54	34.85	1.02	3.25	23.71	16.2	−31.4	9.81	2.62	7.01	−14.68
3	Inj. before IHX	15.11	1.02	3.21	2.44	25.12	1.02	4.01	21.59	8.23	−25.2	3.87	0.56	3.30	−4.19
3	Inj. after IHX	15.06	1.02	3.21	2.65	24.93	1.02	3.97	19.87	8.44	−26.8	0.49	−0.35	0.84	−6.98
3	Inj. Compressor	15.00	1.01	3.22	2.76	24.92	1.02	3.81	17.02	10.5	−26.5	2.72	0.92	1.78	−8.95
4	Inj. before IHX	14.99	1.00	3.38	2.68	34.59	1.00	4.07	23.28	9.6	−31	7.67	1.73	5.84	−6.56
4	Inj. after IHX	14.99	1.01	3.44	4.21	35.05	1.00	4.10	19.57	16.4	−29.3	5.17	0.59	4.55	−11.54
4	Inj. compressor	15.07	1.01	3.46	5.10	35.04	1.01	3.85	16.10	17.4	−33.3	6.12	2.07	3.97	−14.60

Results obtained for different bypass injection points, and the improvement (%) when compared to closed bypass cycle in a refrigeration plant. (Cabello et al., 2012)

2.5. The gas cooler optimal pressure

2. State of the art

$T_c = T_{cond,out}$=

$T_{gc,ro}$ = Refrigerant temperature out of the gas cooler.

$T_e = T_{evap}$ = Refrigerant evaporation temperature.

Author	Expression	Conditions, when…	Mode
Single-stage cycles			
(Kauf, 1999)	$P_{opt}\ (bar) = 2.6 \cdot T_{amb} = 2.6 \cdot T_c - 7.54.$	35 °C $< T_{amb} <$ 50 °C, $and\ 91\ bar < P_{opt} <$ 130 bar.	Cooling
(Liao et al., 2000)	$P_{opt}\ (bar) = (2.778 - 0.0157 \cdot T_e) \cdot T_c + 0.381 \cdot T_e - 9.34.$	$-10°C < T_c < 20°C, 30°C < T_c < 60°C, and\ 71\ bar < P_{opt} < 120\ bar.$	Cooling
(Sarkar et al., 2004)	$P_{opt}\ (bar) = 4.9 + 2.256 \cdot T_c - 0.17 \cdot T_e + 0.002 \cdot {T_c}^2.$	$-10°C < T_e < 10, and\ 35°C < T_c < 50°C.$	Both
(Chen and Gu, 2005)	$P_{opt}\ (bar) = 2.304 \cdot T_{amb} + 19.29.$	There is not IHX and evaporator quality is x=1.0.	Cooling
	$2.68 \cdot T_{amb} + 0.975 = 2.68 \cdot T_c - 6.797.$	Using 2.9 °C as the temperature approach at the gas cooler outlet. $-10°C < T_e < 10°C; 35°C < T_c < 50°C, and\ 80\ bar < P_{opt} < 135\ bar.$	Cooling
(Sarkar et al., 2006)	$P_{opt}\ (bar) = 85.45 + 0.774 \cdot T_{GC,wi}.$	20 °C $< T_{wi} <$ 40 °C.	Both
(Sawalha, 2008)	$P_{opt}\ (bar) = 2.7 \cdot (T_{amb} + T_c) - 6.1.$	$25°C < T_{amb} < 45°C, and\ 75\ bar < P_{opt} < 135\ bar.$	Cooling
(Kim et al., 2009)	$P_{opt}\ (bar) = 1.938 \cdot T_c + 9.872.$	$25°C < T_{amb} < 45°C, and\ 75\ bar < P_{opt} < 135\ bar.$	Cooling
(Aprea and Maiorino, 2009)	$P_{opt}\ (bar) = P_{opt,Liao} - 0.003 \cdot T_c + 0.174.$	$P_{opt,Liao}\ (bar) = (2.778 - 0.0157 \cdot T_e) \cdot T_c + 0.381 \cdot T_e - 9.34.$	Cooling
(Zhang et al., 2010)	$P_{opt}\ (bar) = P_{opt,Liao} + 0.00473 \cdot T_c - 0.1801.$	$P_{opt,Liao}\ (bar) = (2.778 - 0.0157 \cdot T_e) \cdot T_c + 0.381 \cdot T_e - 9.34.$	Heating
(Ge and Tassou, 2011a)	$P_{opt}\ (bar) = 1.352 \cdot T_{amb} + 44.34.$	$0°C \le T_{amb} \le 20°C.$	Cooling
	$72.05\ bar.$	$20°C \le T_{amb} \le 22°C\ (subcritical\ cycle).$	
	$75\ bar.$	$20°C \le T_{amb} \le 22°C.$	
	$75\ bar.$	$22°C \le T_{amb} \le 27°C.$	
	$P_{opt}\ (bar) = 2.3426 \cdot T_{amb} + 11.541.$	$T_{amb} \ge 27°C.$	
(Ge and Tassou, 2011b)	$P_{opt}\ (bar) = 0.119 T_{amb}^{0.887}(-28.437 R_{ba}^2 - 8.09 R_{ba} + 1.178 \varepsilon_{SHX}^2 - 0.844 \varepsilon_{SHX} - 4.772 R_{ba} \varepsilon_{SHX} + 33.82).$	R_{ba} is the coefficient ratio of high-temperature compressor. ε_{SHX} the effectiveness of suction line heat exchanger.	Cooling
(Wang et al., 2013)	$P_{opt}\ (bar) = 23.08391 + 1.22379 \cdot T_{GC,wo} - 0.004707 \cdot {T_{wo}}^2 + 0.16207 \cdot T_{amb}.$	$-15°C \le T_{amb} \le 5°C.$ T_{wo}, is water outlet temperature, in the range: $55°C < T_{wo} < 80°C.$	Heating
	$P_{opt}\ (bar) = 10.98 + 1.06442 \cdot T_{wo} + 1.01404 \cdot T_{amb} - 0.01216\, {T_{amb}}^2.$		
(Qi et al., 2013)	$P_{opt}\ (bar) = 132.2 - 8.4 \cdot T_c + 0.3 \cdot T_c^2 - 27.7 \cdot 10^{-4} \cdot T_c^3.$	$-15°C < T_{amb} < 30°C, and\ 25°C < Tc < 45°C.$	Heating

2. State of the art

$T_c = T_{cond,out} =$

$T_{gc,ro}$ = Refrigerant temperature out of the gas cooler.

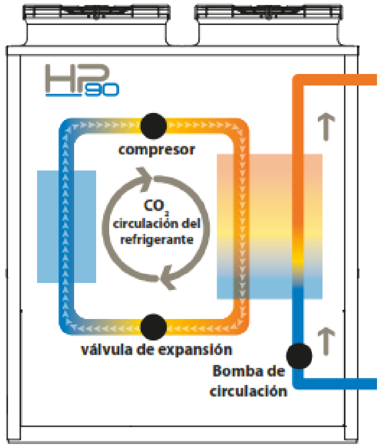
$T_e = T_{evap}$ = Refrigerant evaporation temperature.

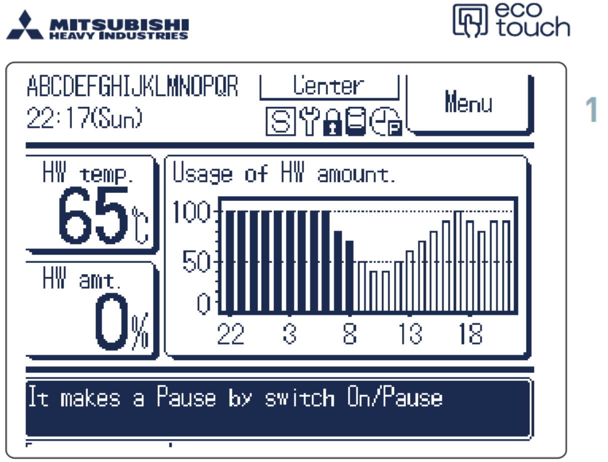
Author	Expression	Conditions, when…	Mode
(Yang et al., 2015)	$P_{opt}(bar) = 2.918T_c + 0.471T_e - 0.018T_eT_c - 13.955.$ $P_{opt}\ (bar) = 2.759T_c - 9.912.$	$-10°C < T_c < 20°C, 30°C < T_c < 60°C, and\ 71 < P_{opt} < 120\ bar.$	Cooling
(Shao et al., 2018)	$P_{opt}(MPa) = min\{0.240 \cdot T_c, 0.132 \cdot T_{c,max} + 3.89\}.$	$40°C \leq T_{c,max} \leq 60°C, and\ 30°C \leq T_c \leq T_{c,max}.$	Both
(Song and Cao, 2018)	$P_{opt}\ (bar) = 34.5 + 1.135 \cdot T_{w,f} + 1.1 \cdot (T_{w,s} - T_{w,f}) + 0.7 \cdot T_{air}.$	$Ambient\ temperature: -20°C < T_{amb} < -7°C$ $Water\ return\ temperature, (wf): 40°C < w,f < 50°C,$ $and\ water\ suply\ temperature\ (w,s): 50°C < w,s < 70°C.$	Heating
Using ejector			
(Sarkar, 2008)	$P_{opt}\ (bar) = 22.7 + 0.21T_e + 1.06T_c - 0.0094T_eT_c + 0.0213T_c^2.$	$-45°C < T_e < 5°C, 30°c < T_c < 60°C.$	Both
(Elbel and Hrnjak, 2008)	$P_{opt}\ (bar) = 1.6T_c + 30.$	$35°C < T_{amb} < 50°C, 88\ bar < P_{opt} < 120\ bar.$	Cooling
(Xu et al., 2012)	$P_{opt}\ (bar) = 1.18T_c + 56.$		Heating
Two-stage cycle			
(Agrawal et al., 2007)	$P_{opt}\ (bar) = 25.11 - 0.087 \cdot T_e + (0.973 + 0.019 \cdot T_c) \cdot T_c.$ (Flash gas bypass).	$-50°C < T_e < -30°C,\ and\ 30°C < T_c < 50°C.$	Heating
	$P_{opt}(bar) = 16.94 - 0.08T_e + (1.201 + 0.0201T_c)T_c.$ (Flash intercooling).		
	$P_{opt}\ (bar) = -18.13 - 0.202T_e + (2.741 + 0.006T_c)T_c.$ (Compression intercooling).		
(Yari, 2009)	$P_{opt}\ (bar) = -10.78 - 0.323T_e - 0.00134T_e^2 + 3.001T_c - 0.00775T_c^2 + 0.0068T_eT_c.$	Two-stage, using ejector.	Cooling
Computation algorithms			
(Cecchinato et al., 2012, 2010; Hu et al., 2015; Liu et al., 2018, 2017; Minetto, 2011; Peñarrocha et al., 2014; Rampazzo et al., 2019; Song et al., 2019; Yin et al., 2019; Zhang and Zhang, 2011)	Several computational strategies have been studied (the order is not necessarily related to the authors citing). -Computational intelligence algorithms and particle swarm technique. -Correlation-free on-line optimal control method. -A logic control able to develop a real time calculation of COP. -On-line artificial neural network system identification technique (ANN). -Model-based optimization strategy that employed a genetic algorithm. -Group method of data handling-type (GMDH) and a PSO-BP-type (particle swarm optimization and back-propagation) neural network. -Perturb and Observe approach (P&O), instead of maximizing the COP. -A perturbation based extremum seeking control scheme (ESC).		Heating, Cooling Heating Heating Heating Cooling Cooling Cooling, Cooling Cooling

2.6. Technological situation of the heat pumps currently in the market, including CO_2 heat pumps

- Most common conventional refrigerants: R134a and R410A.
- For $T_{ev,wi}$: 0°C to 25°C, CO_2 offers good COPs around 4.0, 3.6, and 3.0 for $T_{gc,wi/o}$: 10/60°C, 20/60°C, and 30/60°C, respectively

MAX COP®





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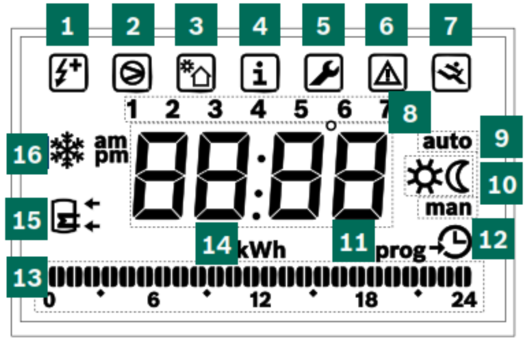


OBJECTIVES

3. Objectives

3.1. Main objective

- To experimental and numerically study different configurations of a CO_2 water-to-water heat pump for hot water generation for both space heating and DHW applications in order to contribute to the design and optimization of CO_2 water-to-water heat pumps and the environmental preservation.

3.2. Specific objectives

- To study the influence that the liquid receiver pressure and the IHX efficiency have on the COP when working at a constant pressure (out of the optimal).

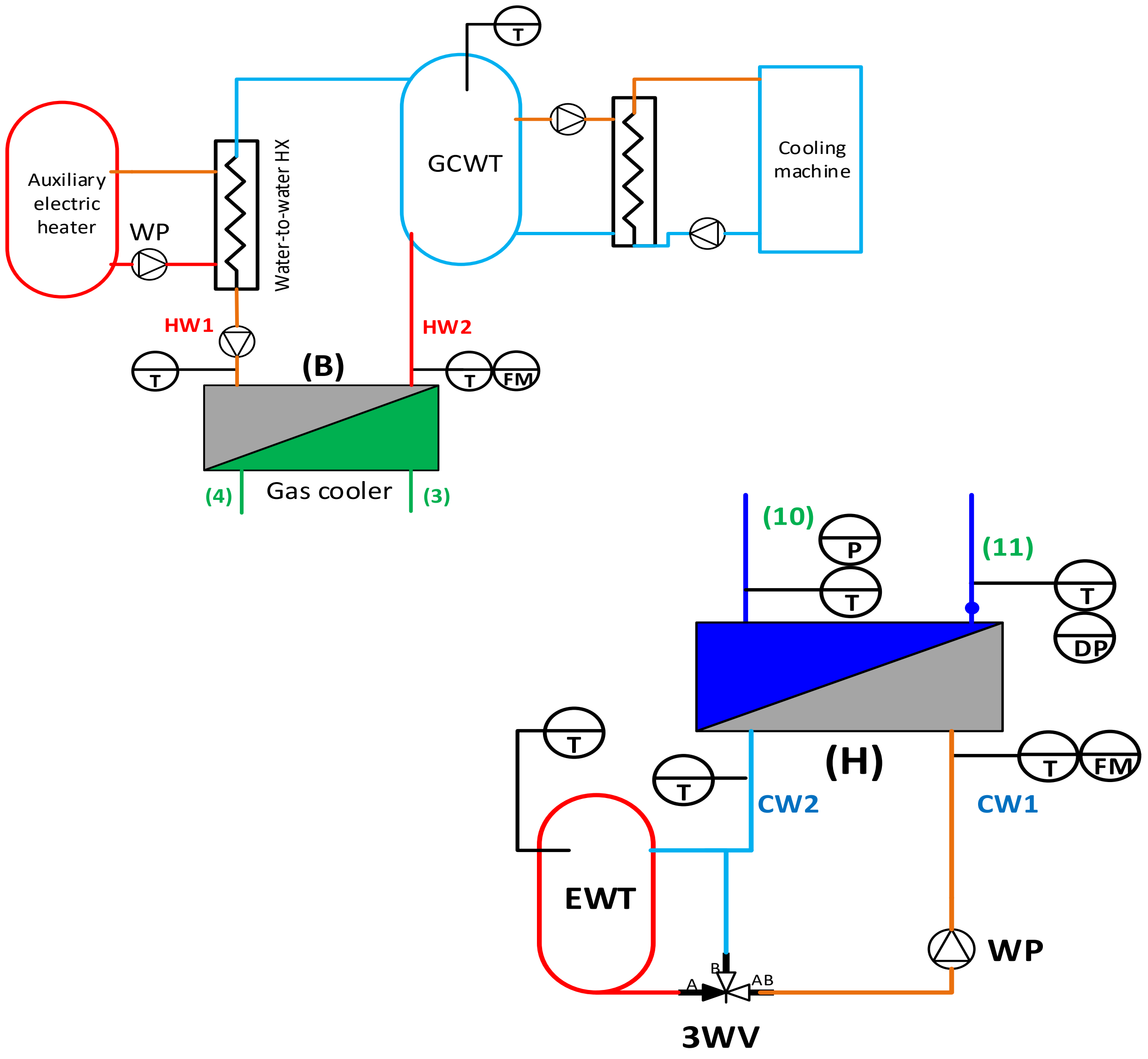
- To analyze the optimal pressure, numerical and experimentally, and the influence that different variables have on this optimal pressure of operation.

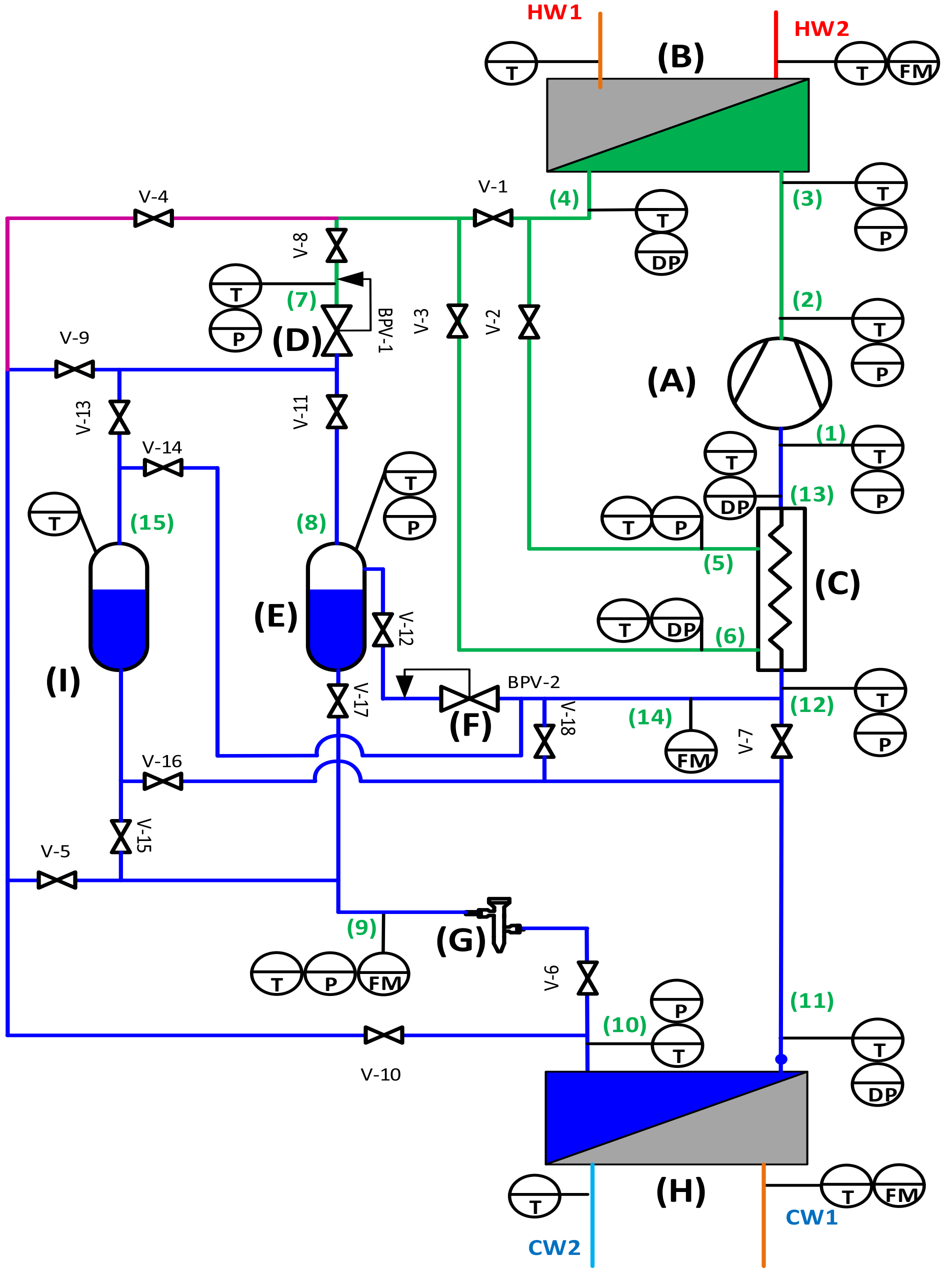
- To evaluate the CO_2 water-to-water heat pump when working on space heating application, comparing the different configurations and analyzing the influence of different variables, including a numerical study about varying the heat transfer area of the different heat exchangers.

- To study the CO_2 water-to-water heat pump in DHW application, comparing the different configurations, including when working at optimal or constant pressure.

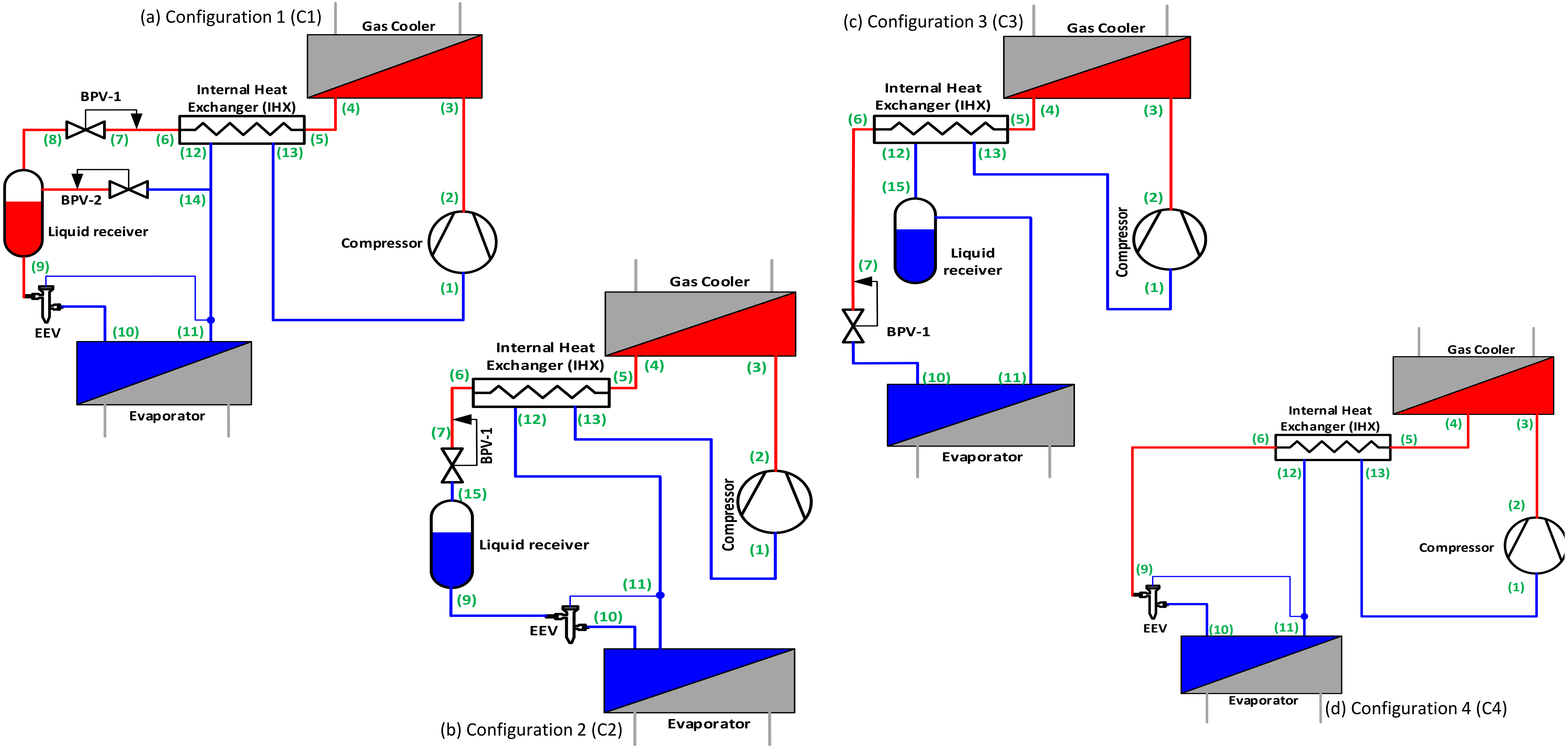
FACILITY DESCRIPTION

4.1. General sketch of the CO2 facility





4.2. Configurations to be studied



4. Facility description

4.3. Main elements of the refrigerant loop

Element	Manufacturer	Model	Specifications
Evaporator	Swep PHE	B8Tx26P	Heat transfer area (A_{ht}) 0.552 m^2
Compressor	Dorin	CD300H	Displacement volume 1.46 m^3/h
Gas cooler	Swep PHE	B16x34P	A_{ht} = 1.31 m^2
Liquid to vapor heat exchanger	Swep PHE	B17x4P	A_{ht} = 0.082 m^2
Back pressure expansion valve	Carel	E2V11	
Thermostatic expansion valve	Carel	E2V24	

4.4. Main elements of the water loop

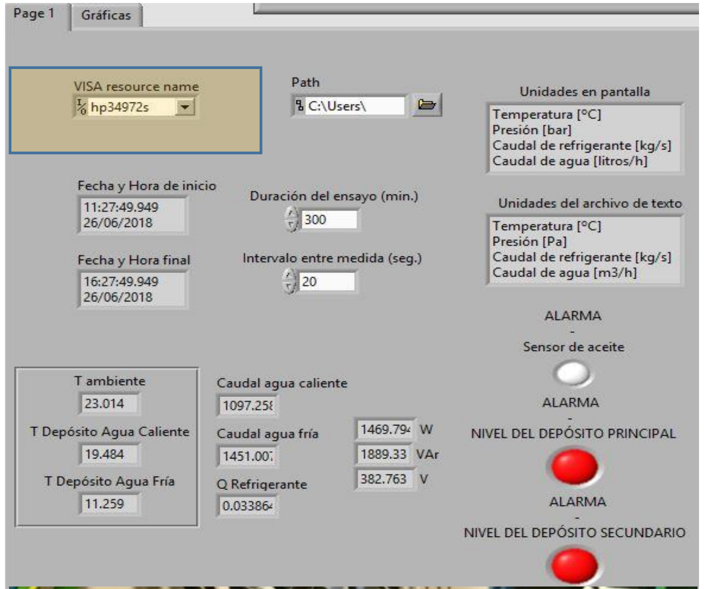
Element	Manufacturer	Model, specifications
Three-ways valve	Sauter	AVM105SF132
Water circulation pumps	Wilo	Stratos 30/1-12
Water-to-water heat exchanger	Alfa Laval PHE	T2-BFG, A_{ht} =0.14 m^2
PID for the temperature control	WATLOW	EZ Zone

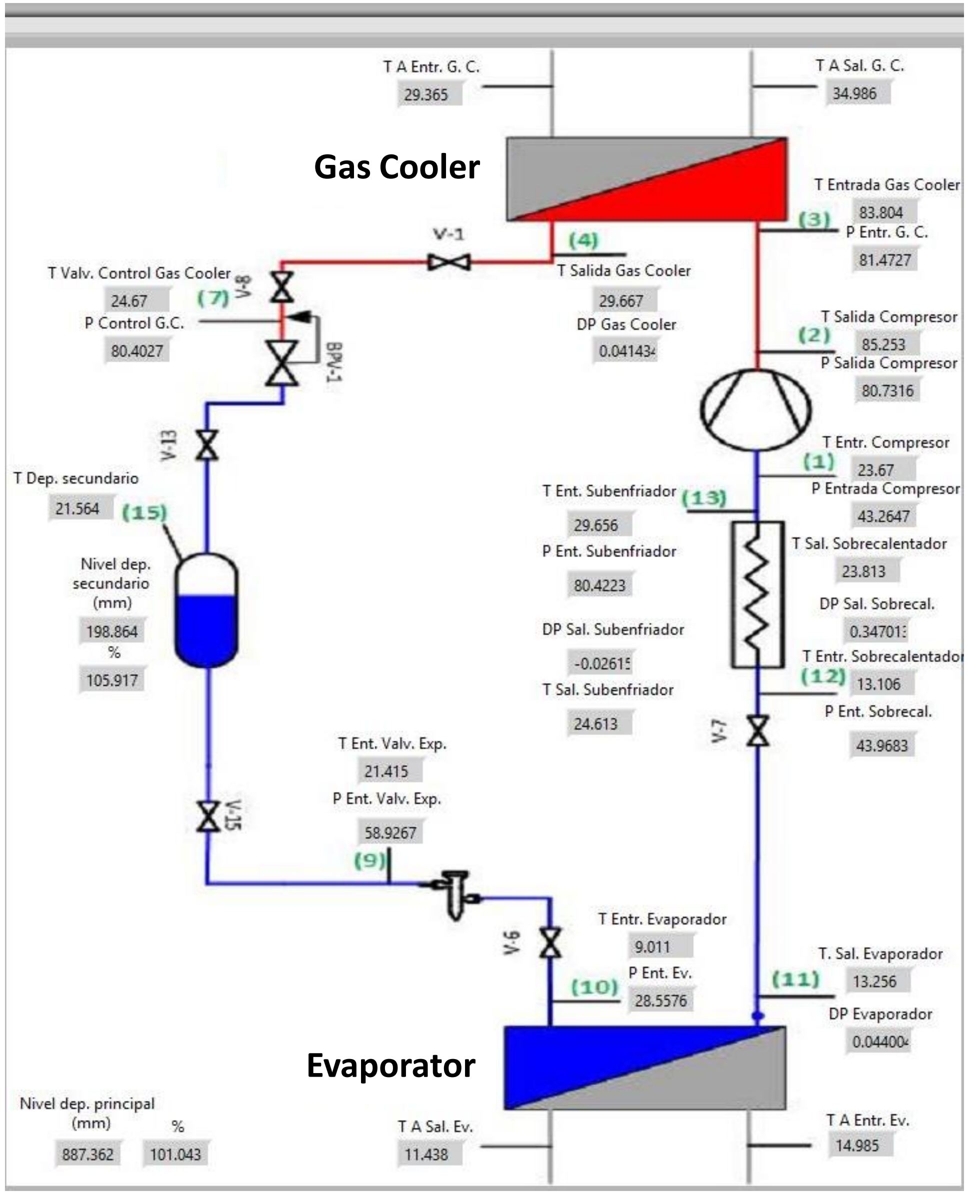
4.5. Measuring devices and their accuracy

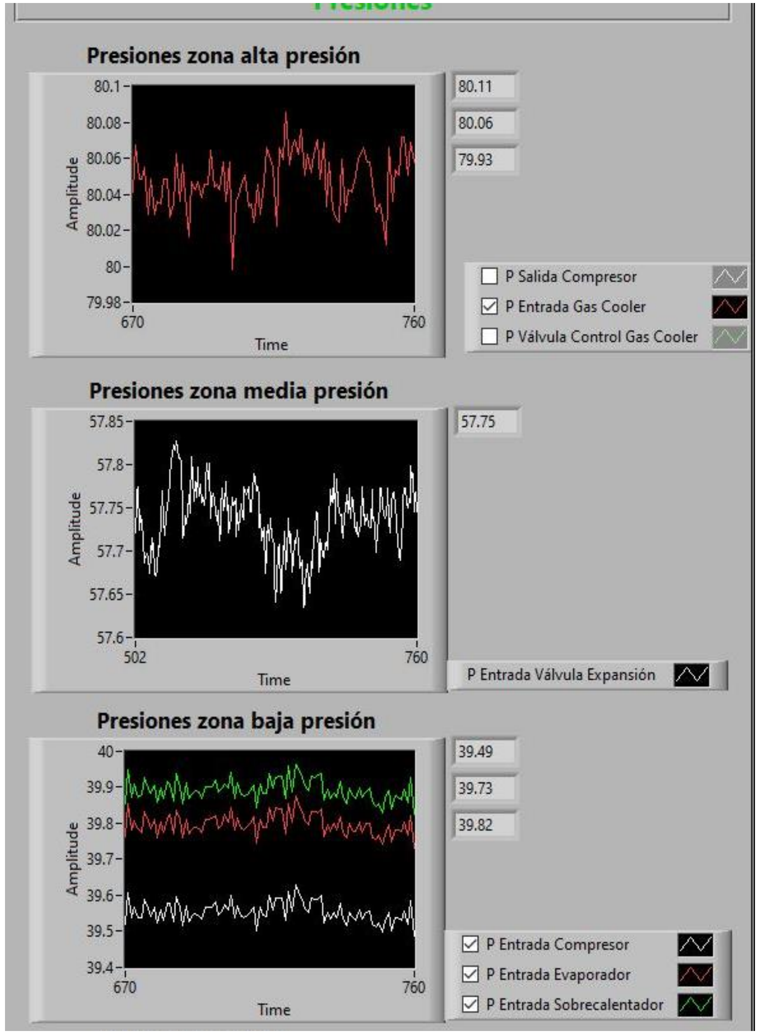
Measured variable	Device	Point	Measuring range	Accuracy
Temperature	Pt100 RTD class A 1/10 DIN	1-13, 15, CW1&2, HW1&2	223 – 523 K	$\pm$ 0.1 [K]
Absolute pressure	Yokogawa EJX510A ECS	1, 10, 11	0 – 5 MPa	$\pm 2.1\cdot10^{-3}$ [MPa]
	Yokogawa EJX510A JDS	2, 3, 5, 7, 9	0 – 12 MPa	$\pm 5\cdot10^{-3}$ [MPa]
Differential pressure	Yokogawa EJX110A JHS	4, 6, 11, 13, HW2, CW2	0 – 0.1 MPa	$\pm 0.26\cdot10^{-3}$ [MPa]
Mass flow rate	Yokogawa RCCS32	9	0 – 0.1 $kg\cdot s^{-1}$	$\pm 0.0027\cdot\dot{m} \pm 5.28\cdot10^{-6}$ [$kg\cdot s^{-1}$]
	Yokogawa RCCT34	14	0 – 0.1 $kg\cdot s^{-1}$	$\pm 0.0056\cdot\dot{m} \pm 4.17\cdot10^{-5}$ [$kg\cdot s^{-1}$]
Water flow rate	SIEMENS FM MAGFLO MAG5100	CW1	0 – $0.1111\cdot10^{-2}$ $m^3\cdot s^{-1}$	$\pm 0.002\cdot\dot{V} \pm 4.91\cdot10^{-7}$ [$m^3\cdot s^{-1}$]
	SIEMENS FM MAGFLO MAG5100	HW1	0 – $0.1111\cdot10^{-2}$ $m^3\cdot s^{-1}$	$\pm 0.002\cdot\dot{V} \pm 4.91\cdot10^{-7}$ [$m^3\cdot s^{-1}$]
Electric power	SINEAX M563	1	0 – 2500 W	$\pm 0.01\cdot\dot{W}$ [W]

TEST METHODOLOGY AND RESULTS

5.1. Variables recording and data treatment





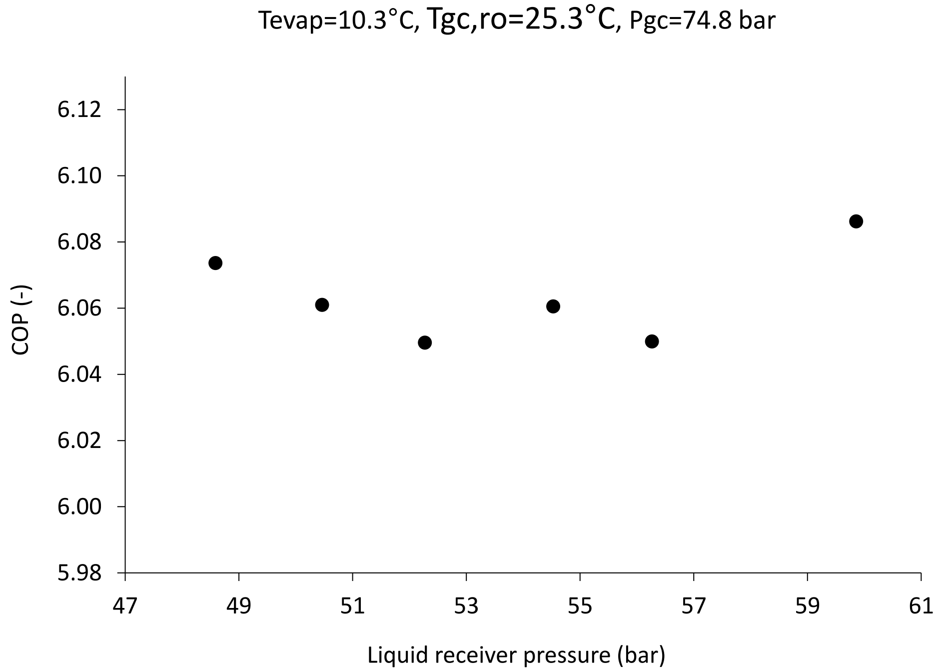


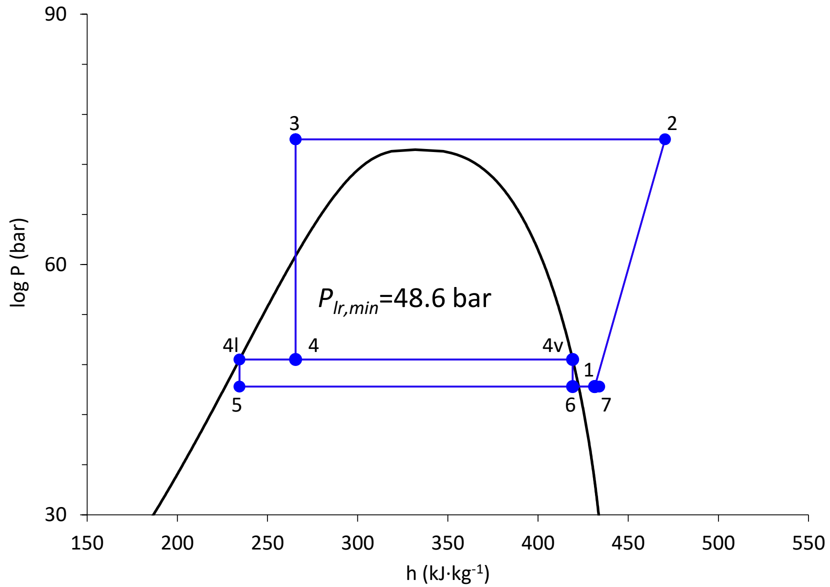
Stable test: ±0.1 K; 30 minutes

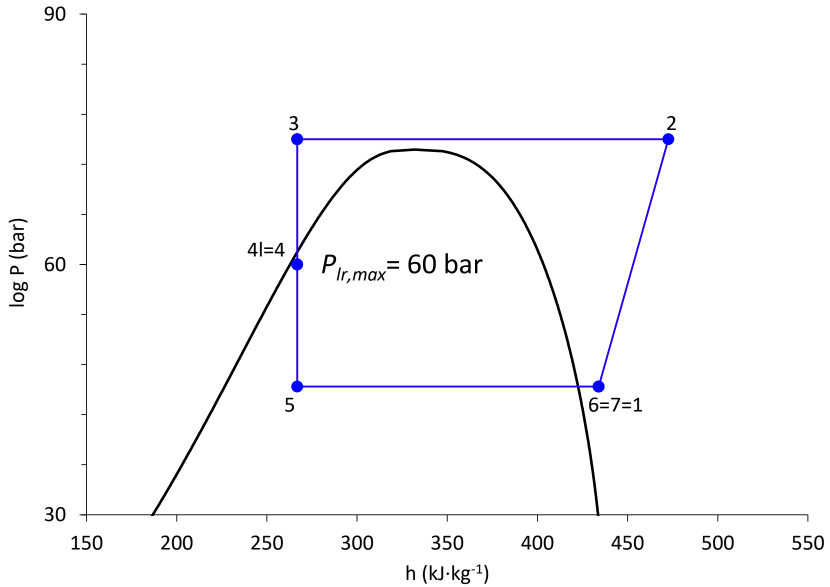
5. Tests methodology and results

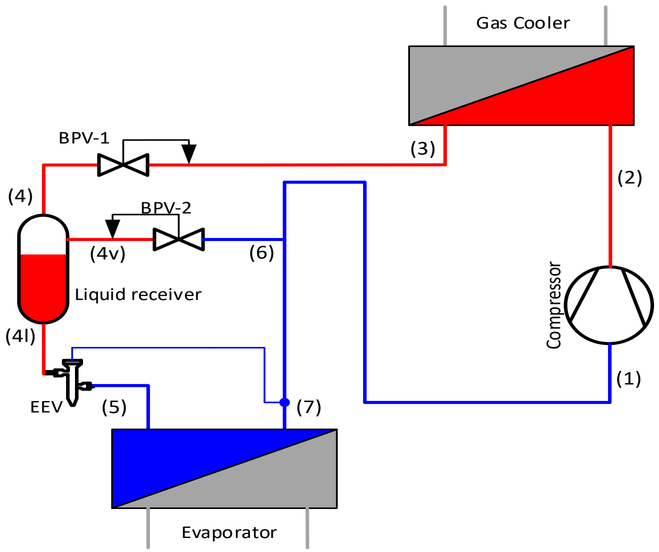
5.2. Influence of the liquid receiver pressure. Experimental results.

Specific refrigerant conditions, without IHX						
Test order (#)	T_{evap} (°C)	*SH* (K)	$T_{gc,ro}$ (°C)	P_{gc} (bar)	P_{lr} (bar)	*COP* (-)
1	10.3	5 K	25.3	75	**48.6 (min)**	**6.0736**
2					50.5	6.0610
3					52.3	6.0496
4					54.5	6.0605
5					56.3	6.0499
6					59.9 (max)	**6.0862**





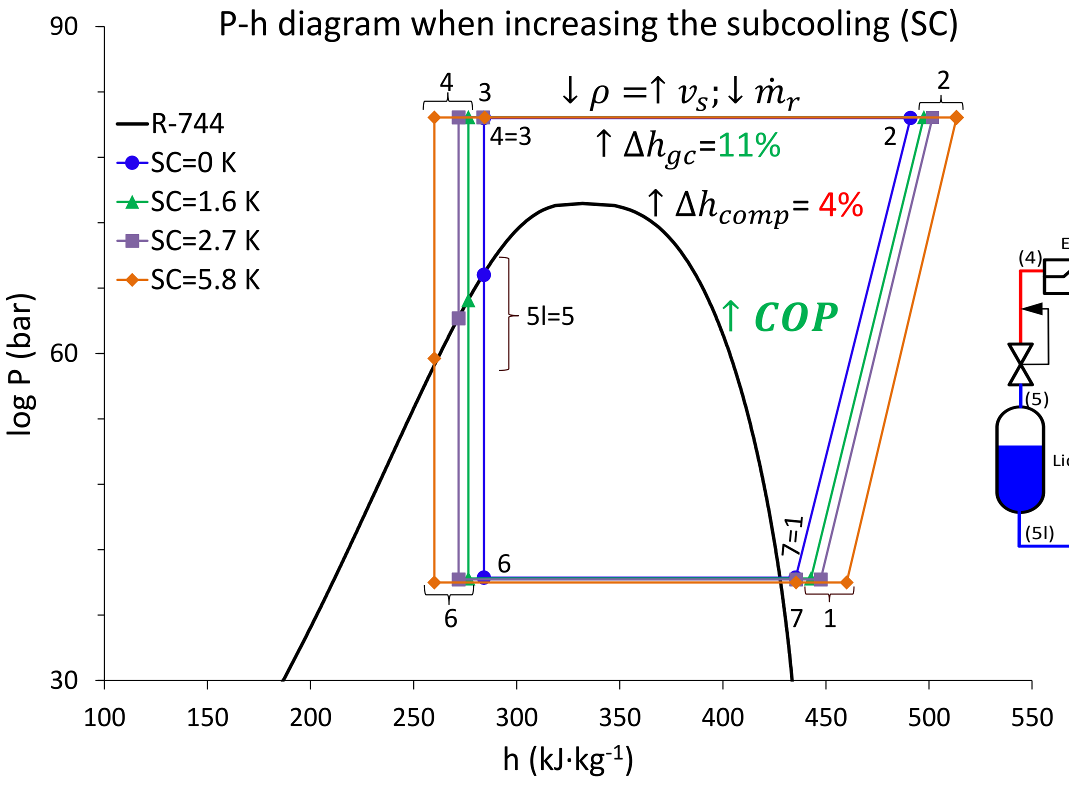




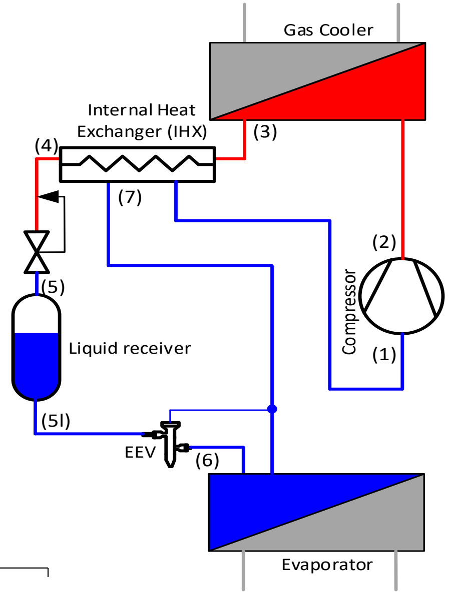
5.3. Influence of the internal heat exchanger (IHX). Experimental results.

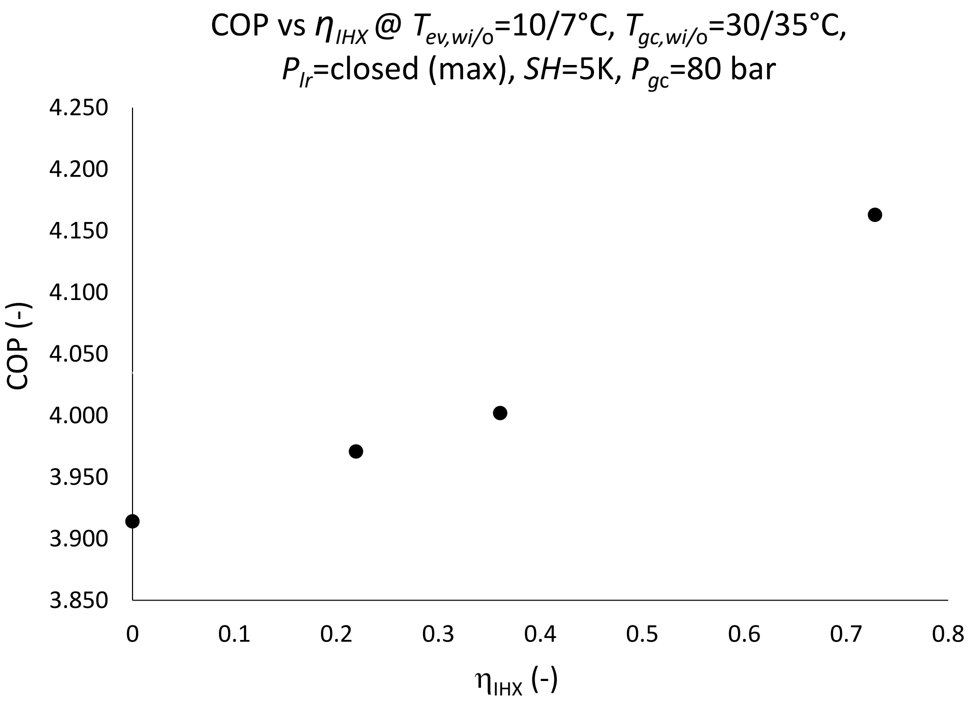
Test order (#)	*SC* (K)	IHX valve position	$T_{ev,wi/o}$ (°C)	*SH* (K)	$T_{gc,wi/o}$ (°C)	P_{gc} (bar)	$*P_{lr}$ (bar)	η_{IHX}	*COP* (-)
1	0 K	Totally closed	10/7	5K	30/35	80	67.2	0 %	3.9142
2	1.6 K	Partially					64.9	22 %	3.9709
3	2.7 K	Opened					63.2	36 %	4.0021
4	5.8 K	Totally opened					59.6	73 %	4.1630

$*P_{lr}$ is calculated based on the measured temperature.



Summary of the four cycles when increasing η_{IHX} through the *SC*, (all-in-one).





5.4. The optimal pressure study

5.4.1 Numerical model

Simplified model based only on the refrigerant cycle.

T_{evap} (°C)	*SH* (K)	η_{IHX} (-)	P_{lr} (bar)	$T_{gc,ro}$ (°C)	P_{gc} (bar)
5-25	3-7	0-0.9	P_{min} & P_{max}	10-60	74-140

Compressor modeling: AHRI 540-2015

$$\dot{m}_r = C_1 + C_2 \cdot T_0 + C_3 \cdot p_c + C_4 \cdot T_0^2 + C_5 \cdot T_0 \cdot p_c + C_6 \cdot p_c^2 + C_7 \cdot T_0^3 + C_8 \cdot p_c \cdot T_0^2 + C_9 \cdot T_0 \cdot p_c^2 + C_{10} \cdot p_c^3$$

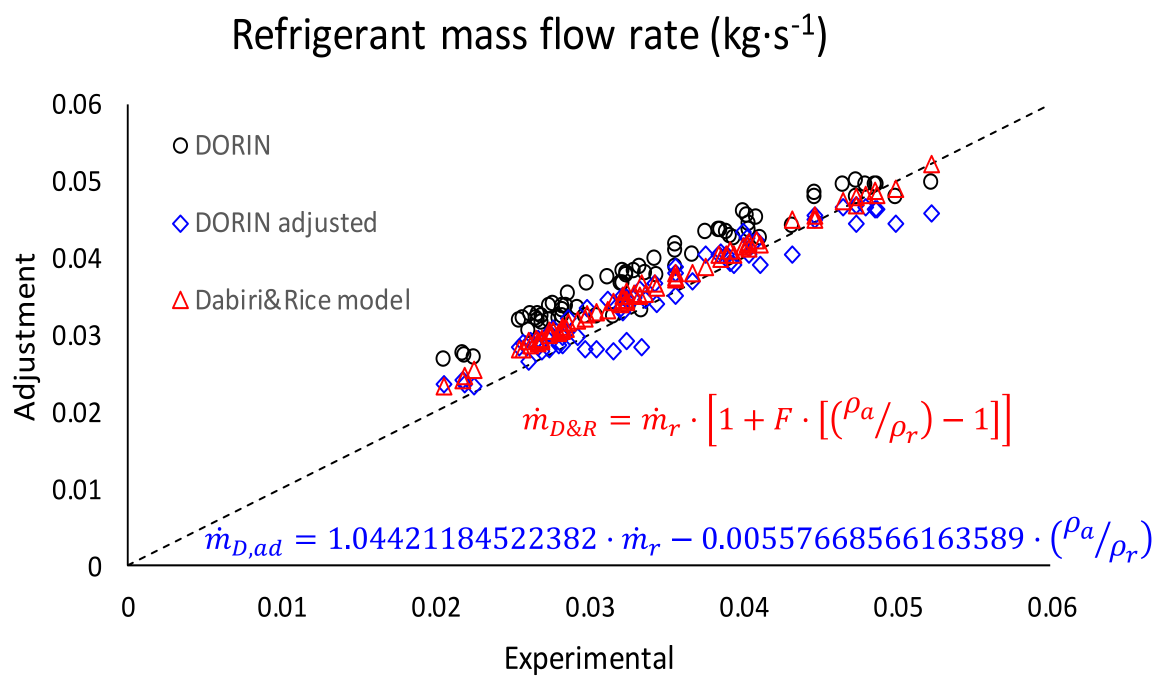
$$\dot{W}_r = C_1 + C_2 \cdot T_0 + C_3 \cdot p_c + C_4 \cdot T_0^2 + C_5 \cdot T_0 \cdot p_c + C_6 \cdot p_c^2 + C_7 \cdot T_0^3 + C_8 \cdot p_c \cdot T_0^2 + C_9 \cdot T_0 \cdot p_c^2 + C_{10} \cdot p_c^3$$

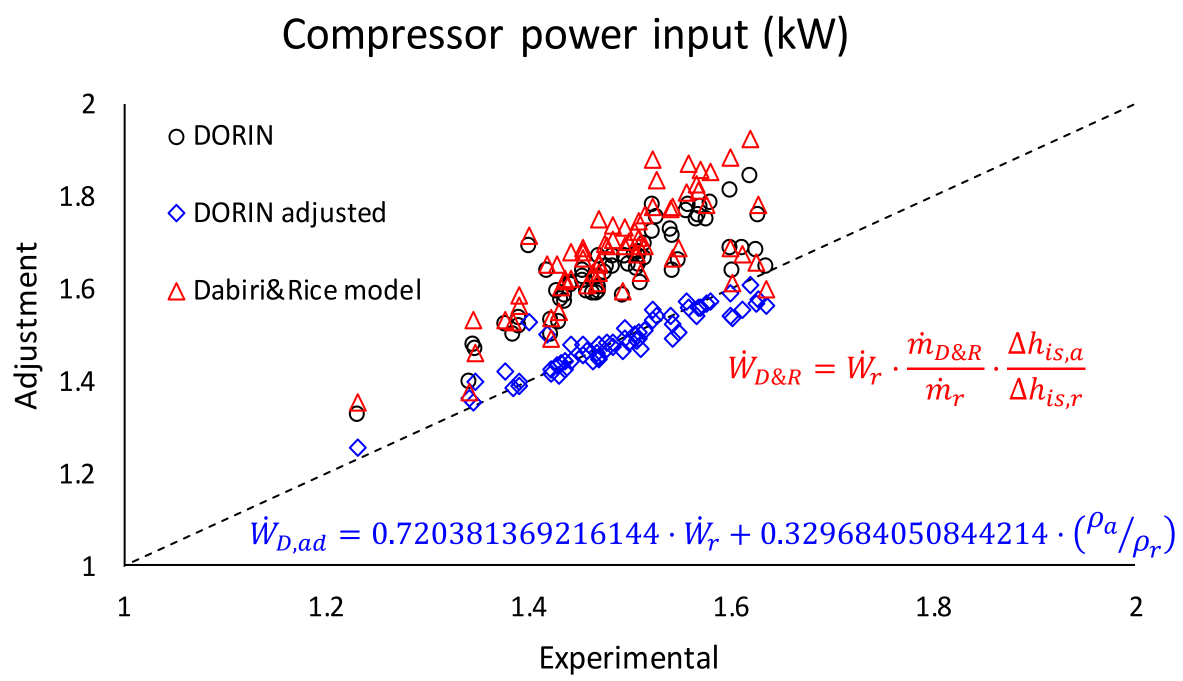
	C_1	C_2	C_3	C_4	C_5	C_6	C_7	C_8	C_9	C_{10}
$\dot{m}$	0.040044	0.001141	-0.000197	$1.08 \cdot 10^{-5}$	$-1.7 \cdot 10^{-6}$	$5.5 \cdot 10^{-7}$	0	0	0	0
$\dot{W}$	-1904.9	-77.21	78.17	-1.033	1.195	-0.5297	-0.0048	0.0052	-0.0032	0.0014

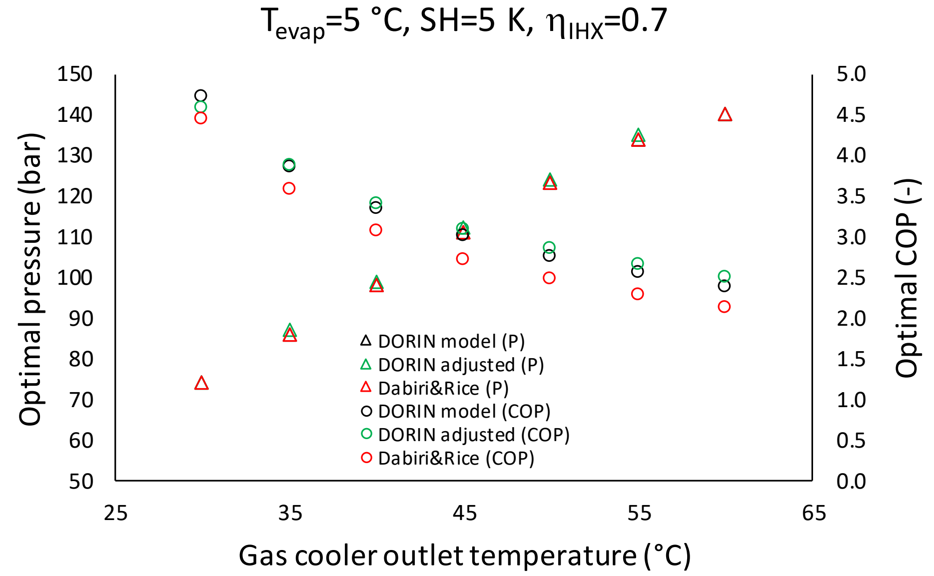
$$\dot{m}_{D\&R} = \dot{m}_r \cdot \left[1 + F \cdot \left[\left({}^{\rho_a}/_{\rho_r}\right) - 1\right]\right]$$

$$\dot{W}_{D\&R} = \dot{W}_r \cdot \frac{\dot{m}_{D\&R}}{\dot{m}_r} \cdot \frac{\Delta h_{is,a}}{\Delta h_{is,r}}$$

5. Tests methodology and results







5. Tests methodology and results

5.4.2. Experimental results

$T_{gc,ro}$ (°C)= 30.5, 32.5, and 36. And T_{evap} (°C)= 5 and 10 °C.

T_{evap} (°C)	SH (°C)	$T_{gc,ro}$ (°C)	η_{IHX} (-)	P_{gc} (bar)	P_{lr} (bar)	COP (-)
5.02	5.06	30.49	0.79	74.04	60.58	4.3983
5.03	5.03	**30.49**	0.76	**76.07**	**59.28**	**4.4011**
5.06	5.06	30.51	0.73	80.05	57.85	4.2783
5.01	5.03	30.49	0.70	84.99	56.25	4.0779
5.05	5.03	30.49	0.68	89.97	54.22	3.8871
5.01	5.06	**30.51**	0.76	**76.08**	**45.56**	**4.3847**
5.02	5.01	32.56	0.95	76.15	67.42	3.3275
5.07	5.04	32.54	0.81	78.07	61.29	4.0485
5.02	5.06	**32.53**	0.77	**80.09**	**59.30**	**4.0580**
5.04	5.11	32.54	0.76	81.00	59.78	4.0101
5.09	5.00	32.55	0.73	84.96	58.03	3.9303
5.01	5.03	32.52	0.70	90.04	56.94	3.7669
5.05	5.02	**32.54**	0.77	**80.05**	**46.21**	**4.0435**
5.05	5.04	36.00	0.92	80.04	69.53	2.8378
4.98	5.06	36.06	0.88	83.07	64.64	3.3450
5.01	5.12	36.03	0.81	85.05	62.20	3.5090
5.04	4.94	**35.99**	0.75	**88.96**	**59.74**	**3.5479**
5.03	5.07	36.00	0.74	91.03	58.53	3.5263
5.03	4.99	**36.00**	0.75	**88.96**	**46.29**	**3.5221**

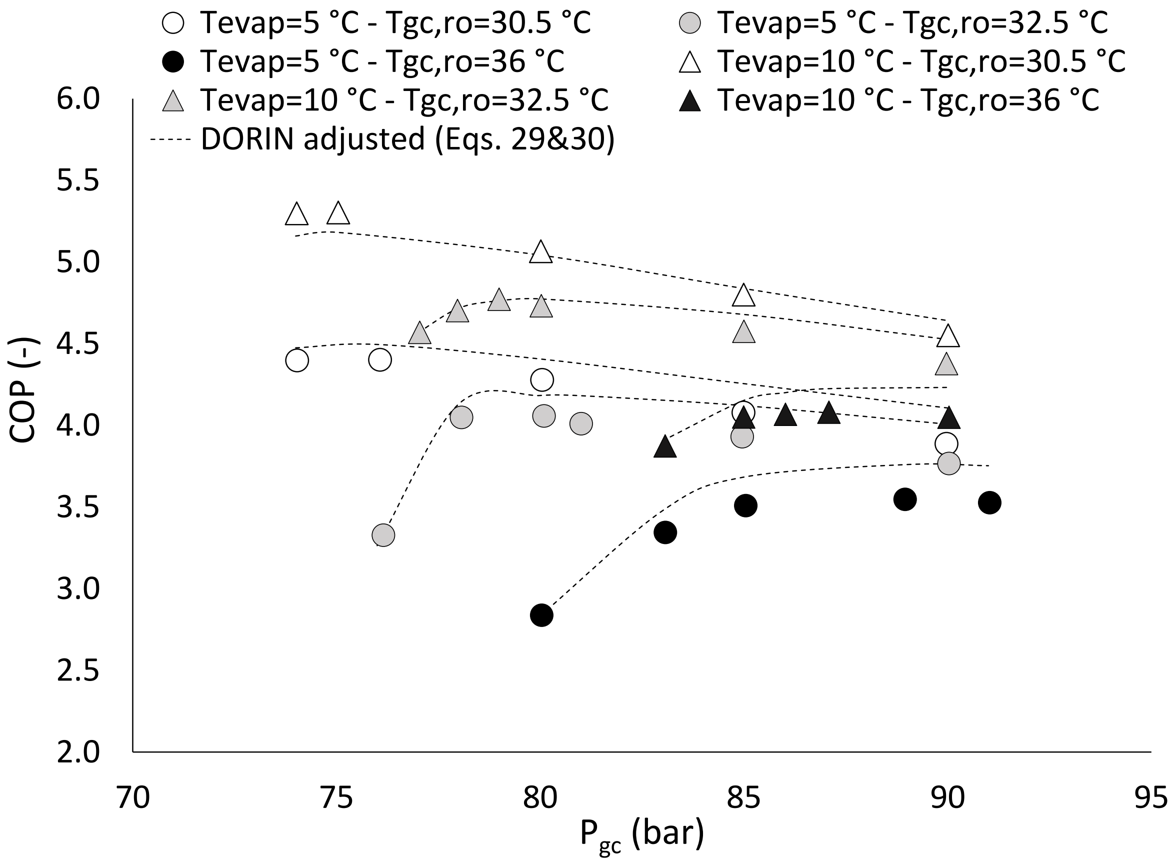
T_{evap} (°C)	SH (°C)	$T_{gc,ro}$ (°C)	η_{IHX} (-)	P_{gc} (bar)	P_{lr} (bar)	COP (-)
10.02	5.03	30.52	0.79	74.03	53.77	5.2987
10.02	4.96	30.50	0.76	75.05	53.96	5.3039
10.01	5.03	**30.49**	0.72	**80.02**	51.15	**5.0667**
10.04	4.97	30.51	0.69	84.99	48.90	4.8008
10.05	6.80	30.47	0.67	90.01	46.82	4.5518
10.00	5.02	32.55	0.87	77.04	57.29	4.5695
9.98	5.02	32.56	0.82	77.97	55.91	4.7028
10.03	4.98	**32.55**	0.79	**78.99**	54.97	**4.7717**
9.97	5.00	32.53	0.77	80.03	53.93	4.7323
10.02	4.99	32.55	0.72	85.00	51.75	4.5759
9.99	5.00	32.55	0.69	89.97	49.58	4.3785
10.00	4.95	36.01	0.88	83.06	58.73	3.8738
10.03	4.97	36.00	0.80	84.99	56.18	4.0517
10.03	4.97	36.01	0.79	86.02	55.29	4.0675
10.05	4.97	**36.00**	0.77	**87.08**	54.33	**4.0811**
10.01	5.01	36.03	0.74	90.03	52.99	4.0503

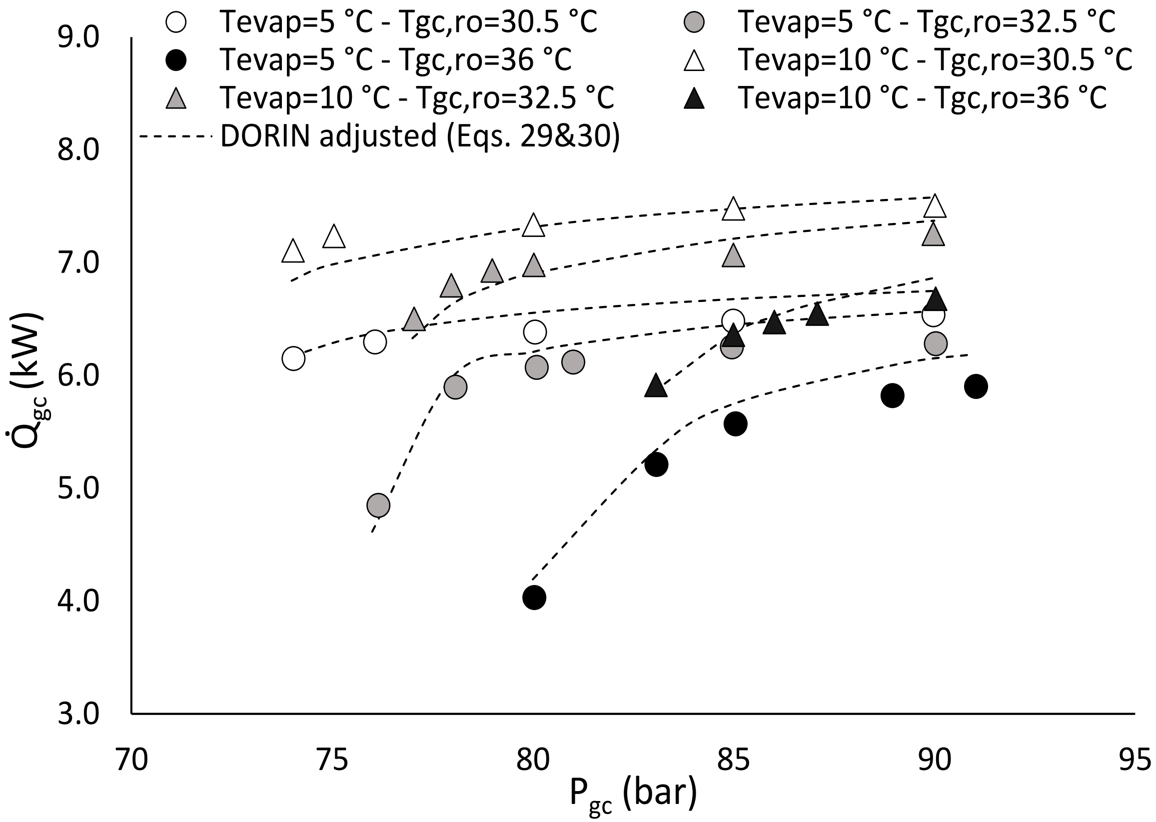
Bold: Indicate the optimal pressure with the maximum and minimum liquid receiver pressure.

5. Tests methodology and results

Comparison between predicted and experimentally measured values of COP and Q_{gc} (SH=5 K, η_{IHX}≈0.7). DORIN Adjusted ($\dot{W}_{D,ad}$ and $\dot{m}_{D,ad}$)

Model is able to reproduce the experimental results



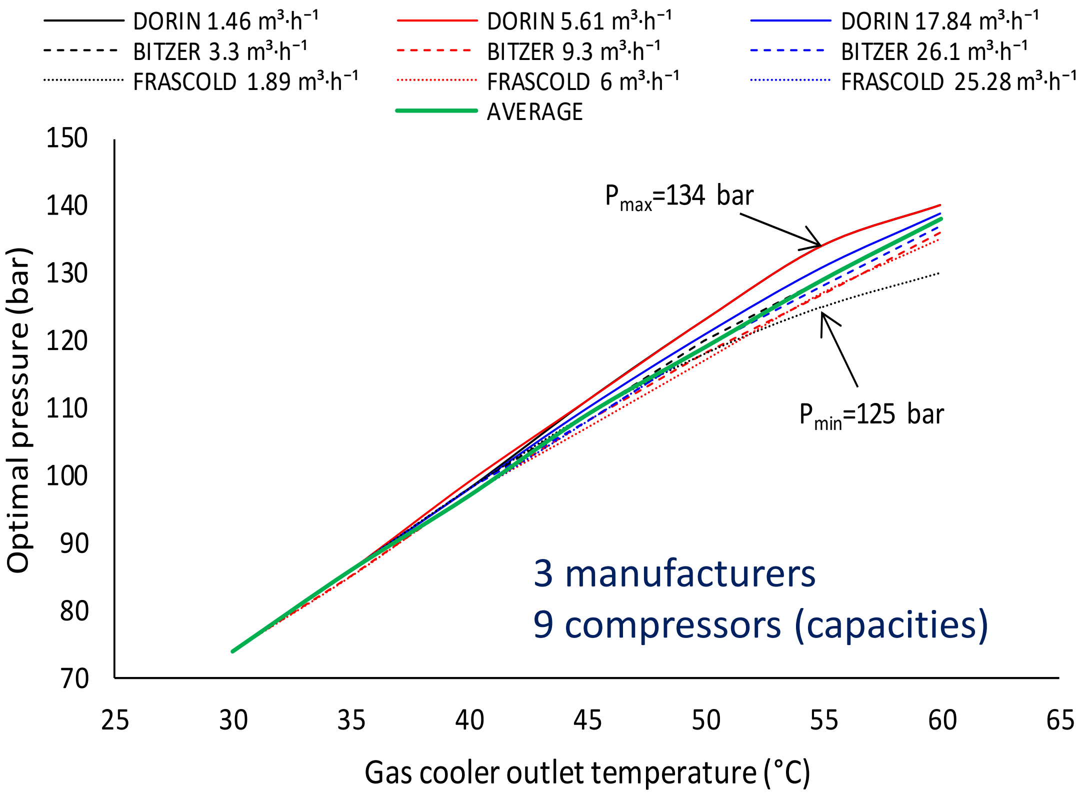


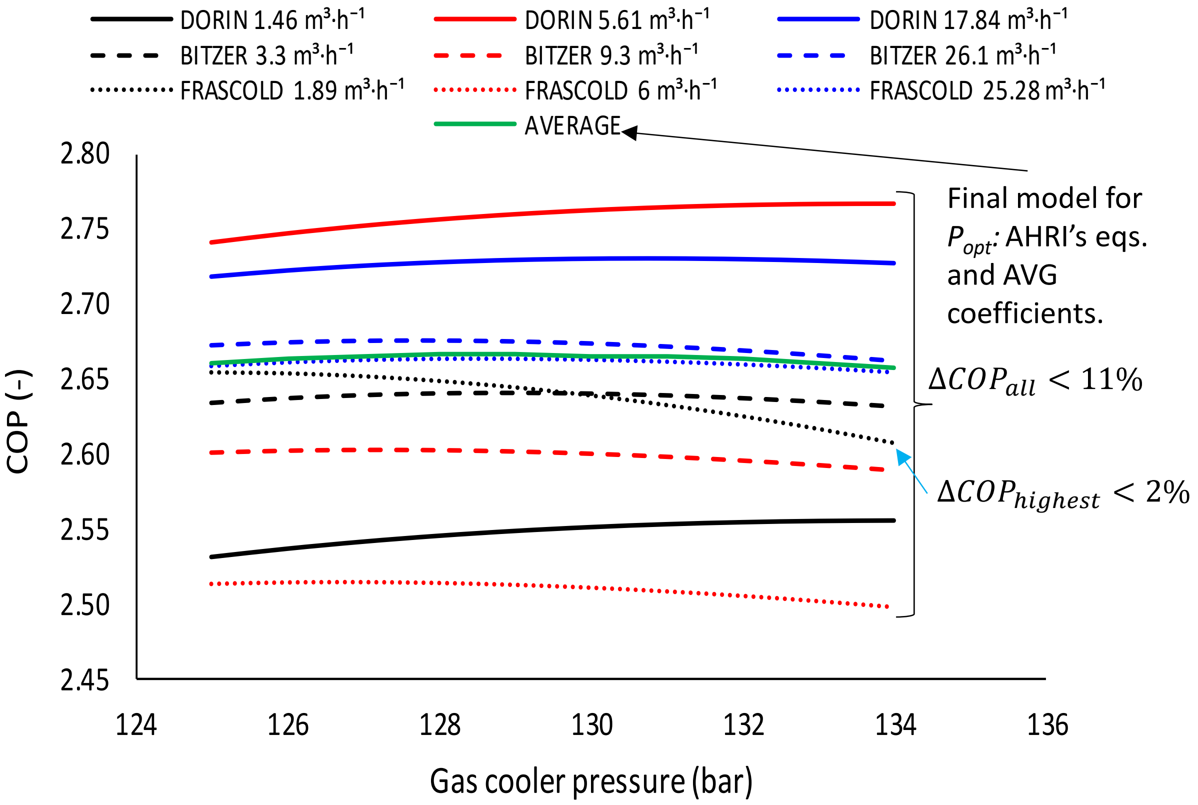
$$\dot{m}_{D,ad} = 1.04421184522382 \cdot \dot{m}_r - 0.00557668566163589 \cdot \left({\rho_a}/{\rho_r}\right)$$

$$\dot{W}_{D,ad} = 0.720381369216144 \cdot \dot{W}_r + 0.329684050844214 \cdot \left({\rho_a}/{\rho_r}\right)$$

5.4.3. Influence of the compressor characteristics

Influence of compressor characteristics on system's behavior (T_{evap}=5 °C, SH=5 K, η_{IHX}=0.7).





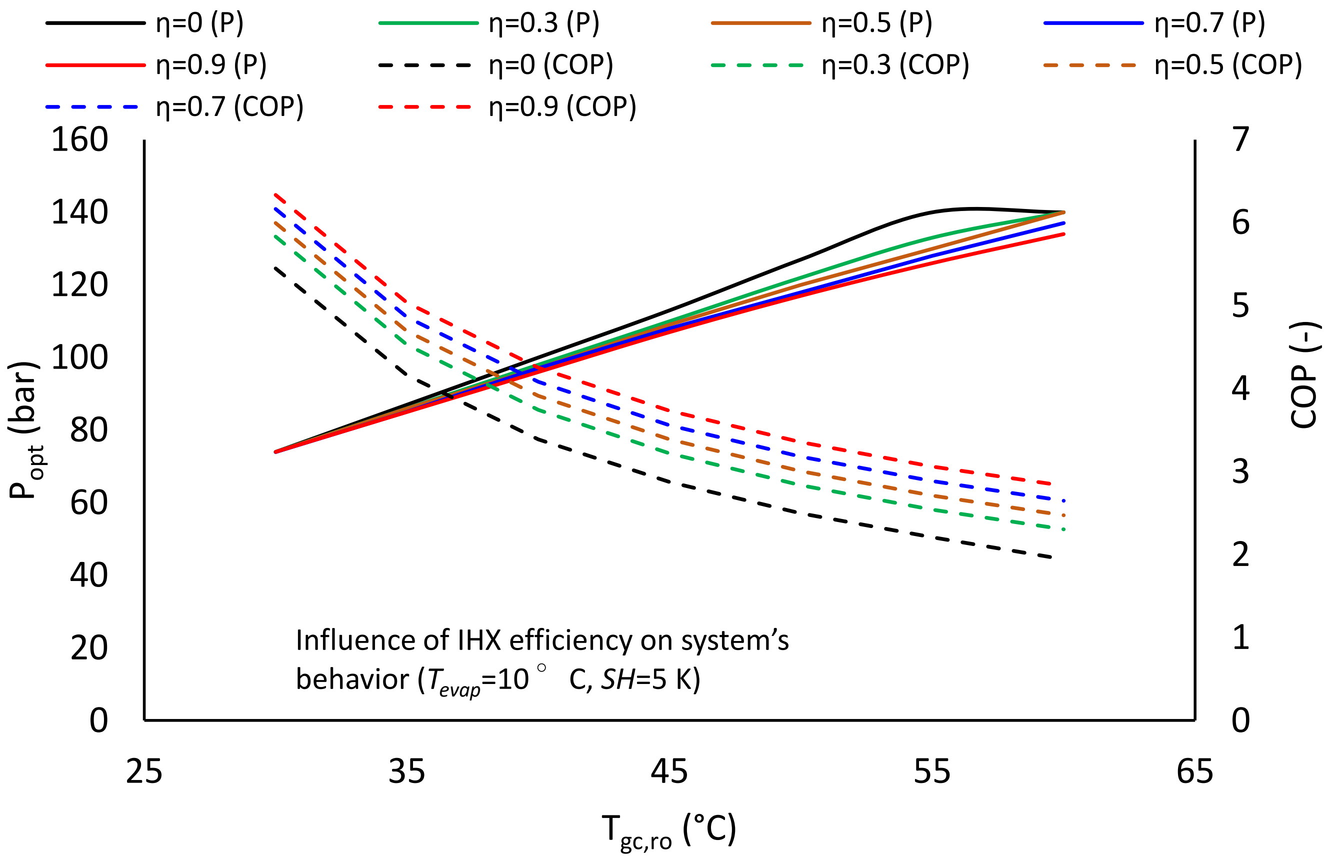
Compressor modeling: AHRI 540-2015

$$\dot{m}_r = C_1 + C_2 \cdot T_0 + C_3 \cdot p_c + C_4 \cdot T_0^2 + C_5 \cdot T_0 \cdot p_c + C_6 \cdot p_c^2 + C_7 \cdot T_0^3 + C_8 \cdot p_c \cdot T_0^2 + C_9 \cdot T_0 \cdot p_c^2 + C_{10} \cdot p_c^3$$

$$\dot{W}_r = C_1 + C_2 \cdot T_0 + C_3 \cdot p_c + C_4 \cdot T_0^2 + C_5 \cdot T_0 \cdot p_c + C_6 \cdot p_c^2 + C_7 \cdot T_0^3 + C_8 \cdot p_c \cdot T_0^2 + C_9 \cdot T_0 \cdot p_c^2 + C_{10} \cdot p_c^3$$

5. Tests methodology and results

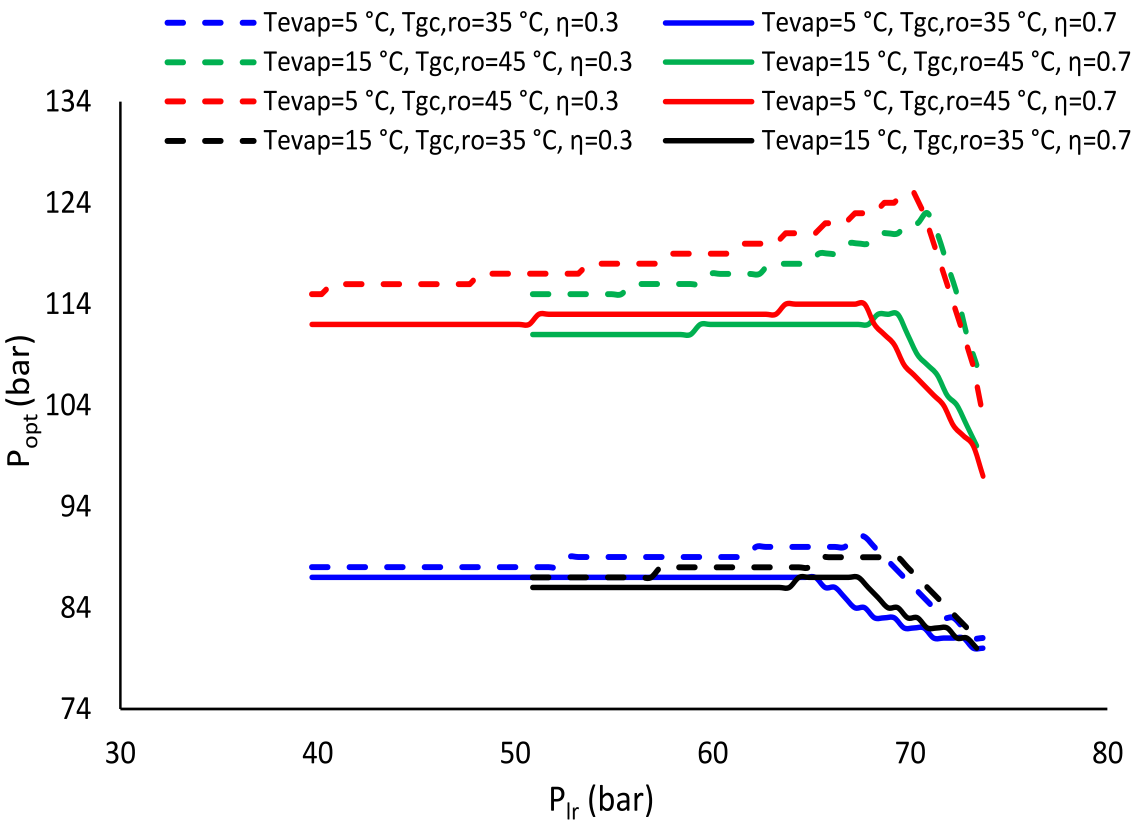
5.4.4. Influence that the IHX efficiency has on the optimal pressure

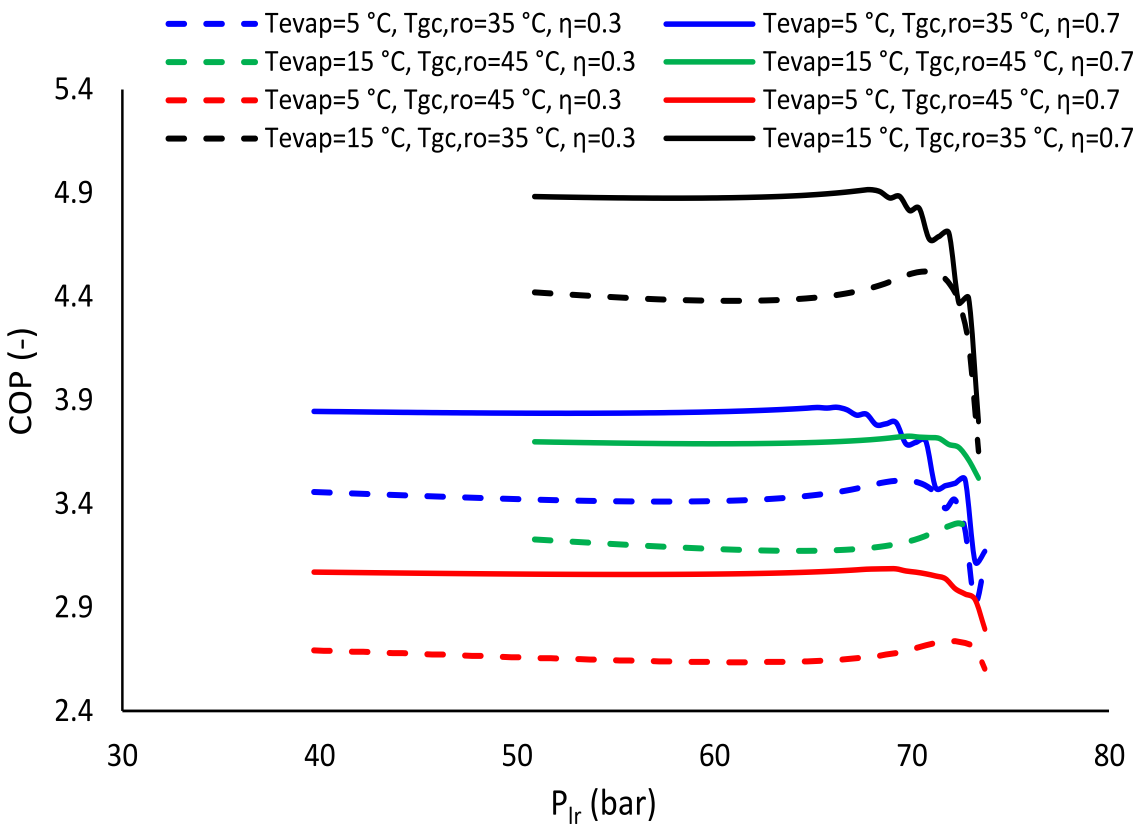


T_{evap} (°C)	SH (K)	$T_{gc,ro}$ (°C)	η_{IHX} (-)	$T_{comp,out}$ (°C)	P_{opt} (bar)	COP (-)	Δ COP (%)
5	5	50	0.7	145	124	2.80	-
Reduce $T_{comp,out}$, bypassing the IHX or by decreasing the high pressure?			0.0	130	131	2.25	-20%
			0.7	130	108 (no P_{opt})	2.64	-5.7%

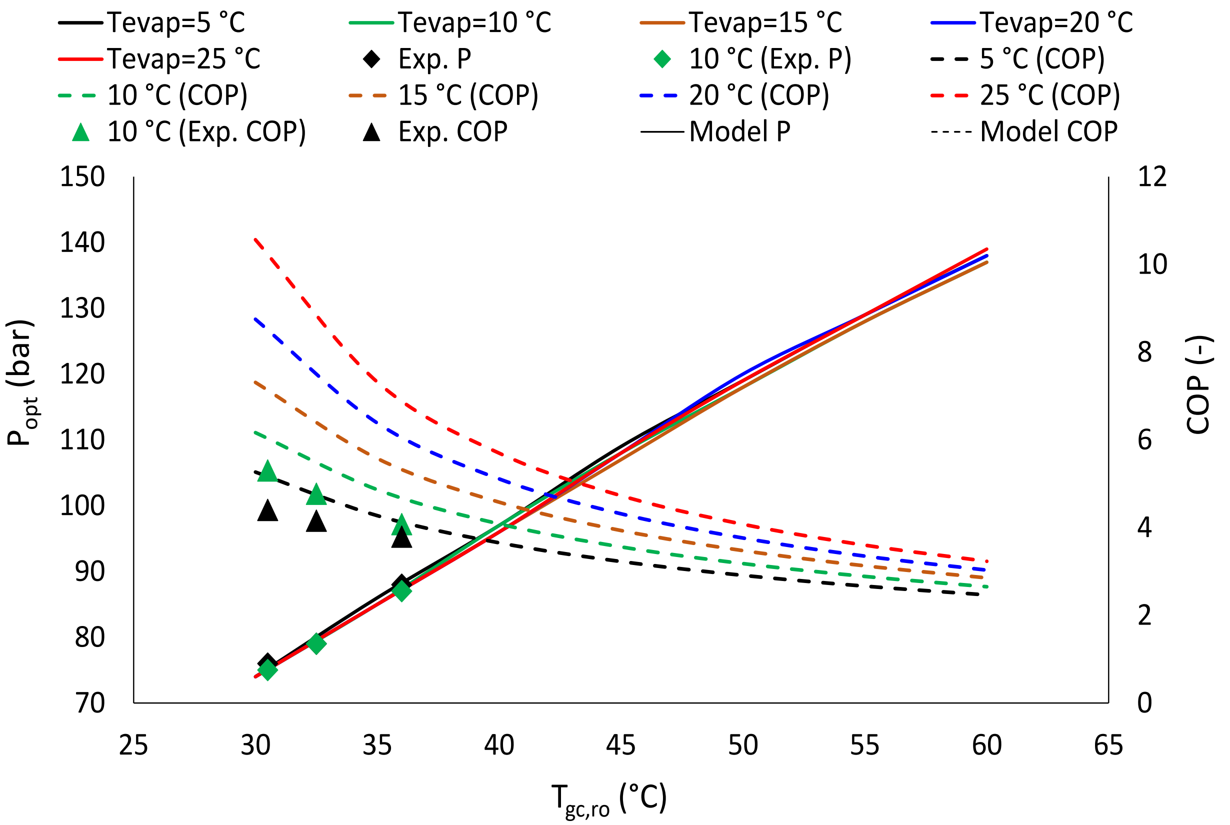
↑η_{IHX}
↑$T_{comp,out}$

5.4.5. Influence that the liquid receiver pressure has on the optimal pressure

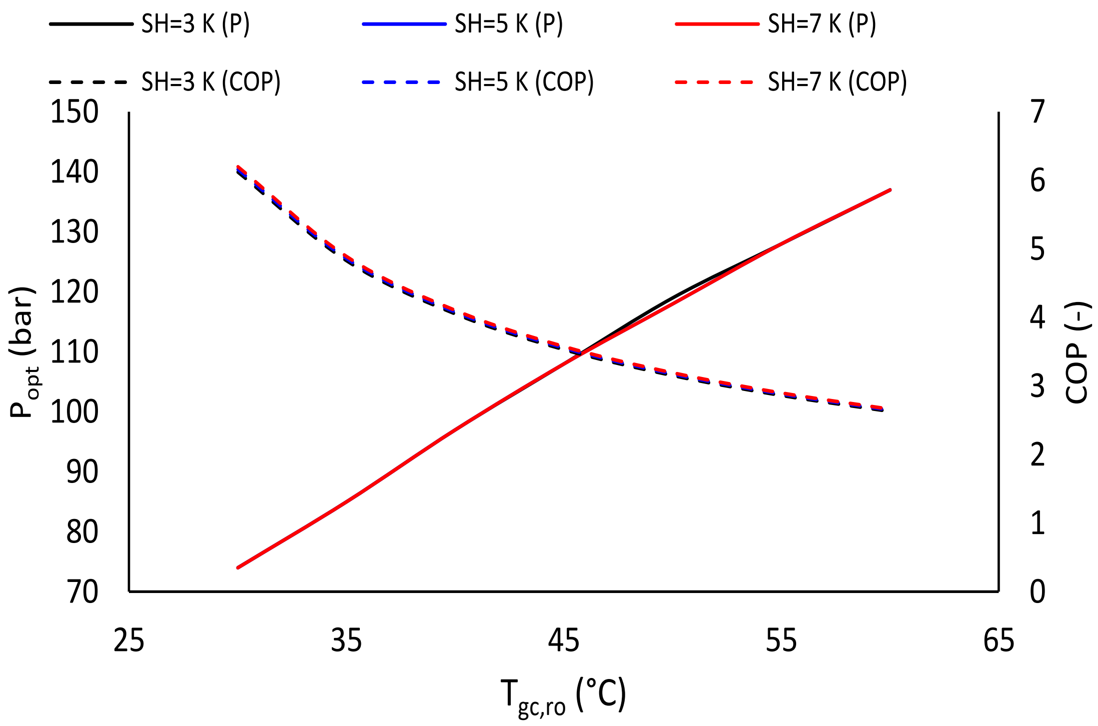




5.4.6. Influence that the evaporation temperature and the superheating have on the optimal pressure



Influence of evaporation temperature on system's behavior (*SH*=5 K, η_{IHX}=0.7).

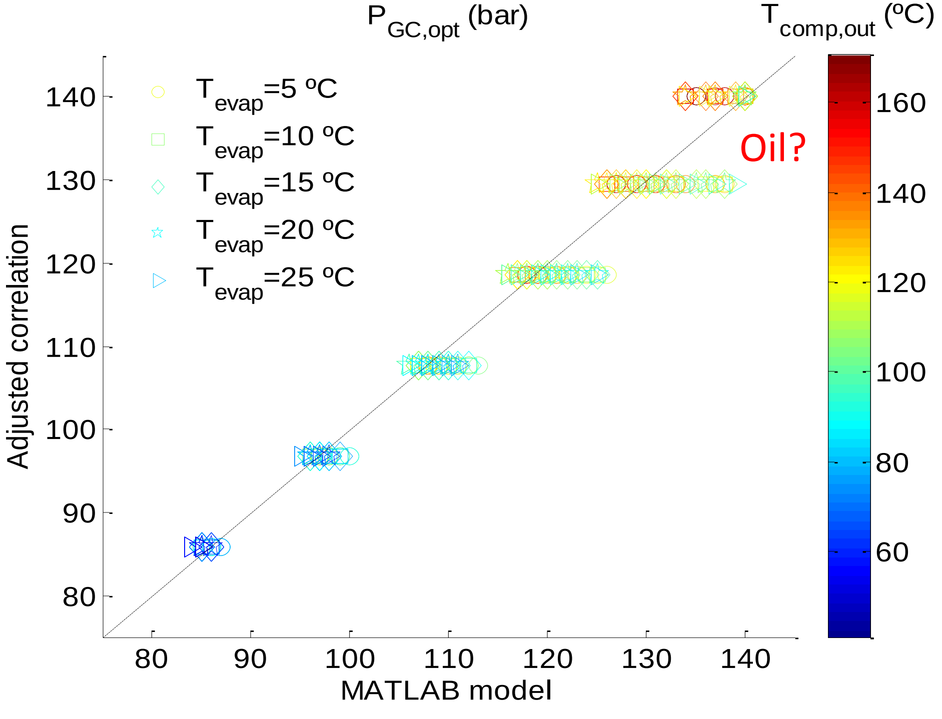


Influence of superheating on system's behavior (T_{evap}=10 ° C, η_{IHX}=0.7).

5. Tests methodology and results

5.4.7. Correlation for the optimal heat rejection pressure control

$$P_{gc,opt} = \min(140;\ 9.3267 + 2.1857 \cdot T_{gc,ro})$$



(T_{evap}=5-25° C, SH=3-7 K, η_{IHX}=0-0.9, $T_{gc,ro}$=30-60° C).

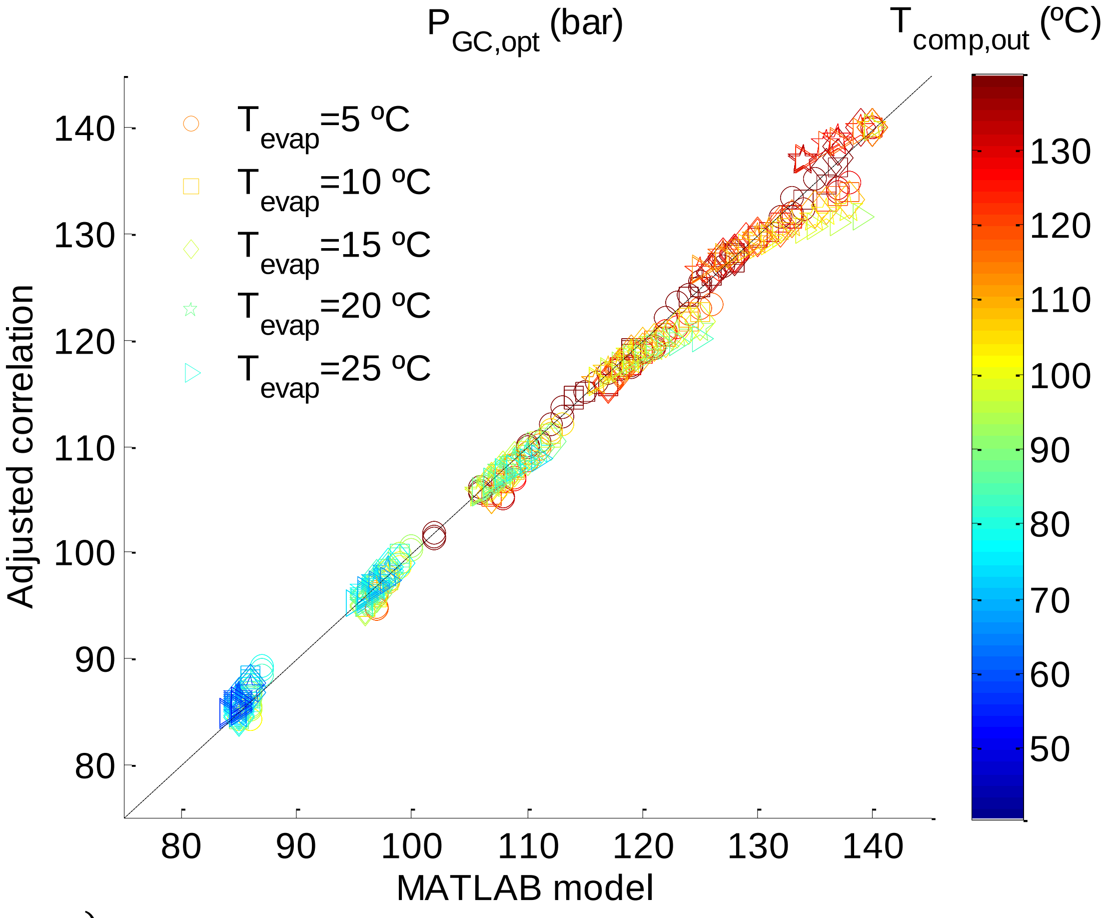
$$T_{comp,out} = 13.403 + 2.0657 \cdot T_{gc,ro} - 2.3525 \cdot T_{evap} + 0.86806 \cdot T_{comp,in}$$

If $T_{comp,out} < 140$ °C:

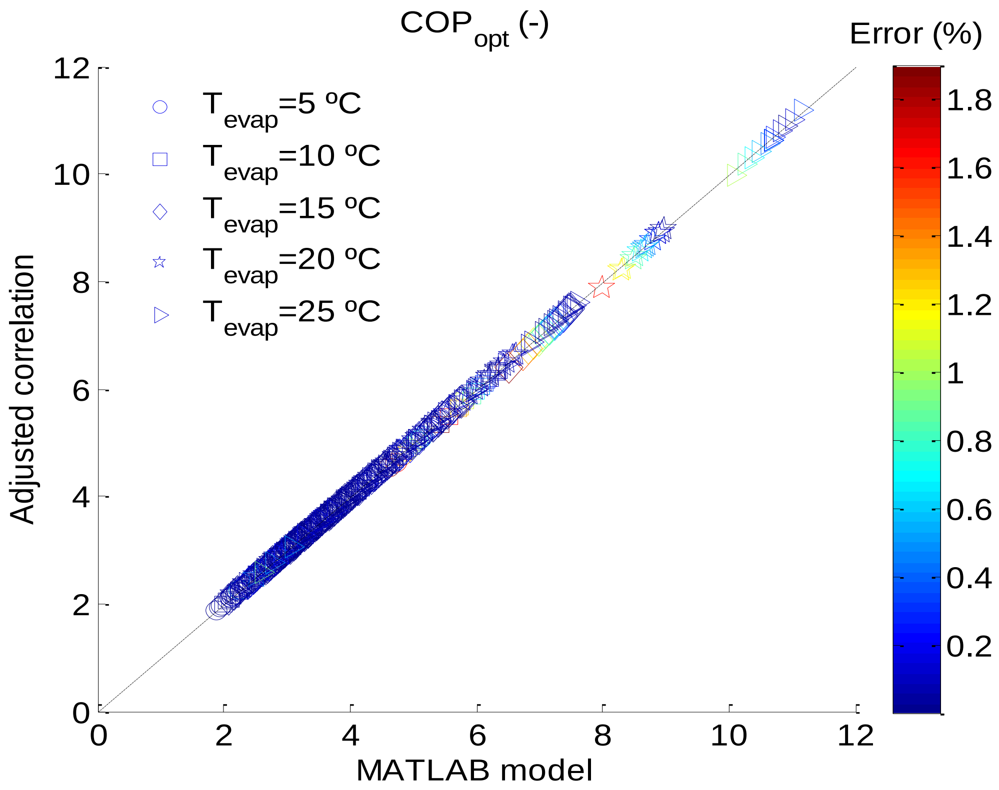
$$P_{gc,opt} = \min(140;\ 11.047 + 2.2756 \cdot T_{gc,ro} + 0.047279 \cdot T_{evap} - 0.20814 \cdot T_{comp,in})$$

If $T_{comp,out} \geq 140$ °C:

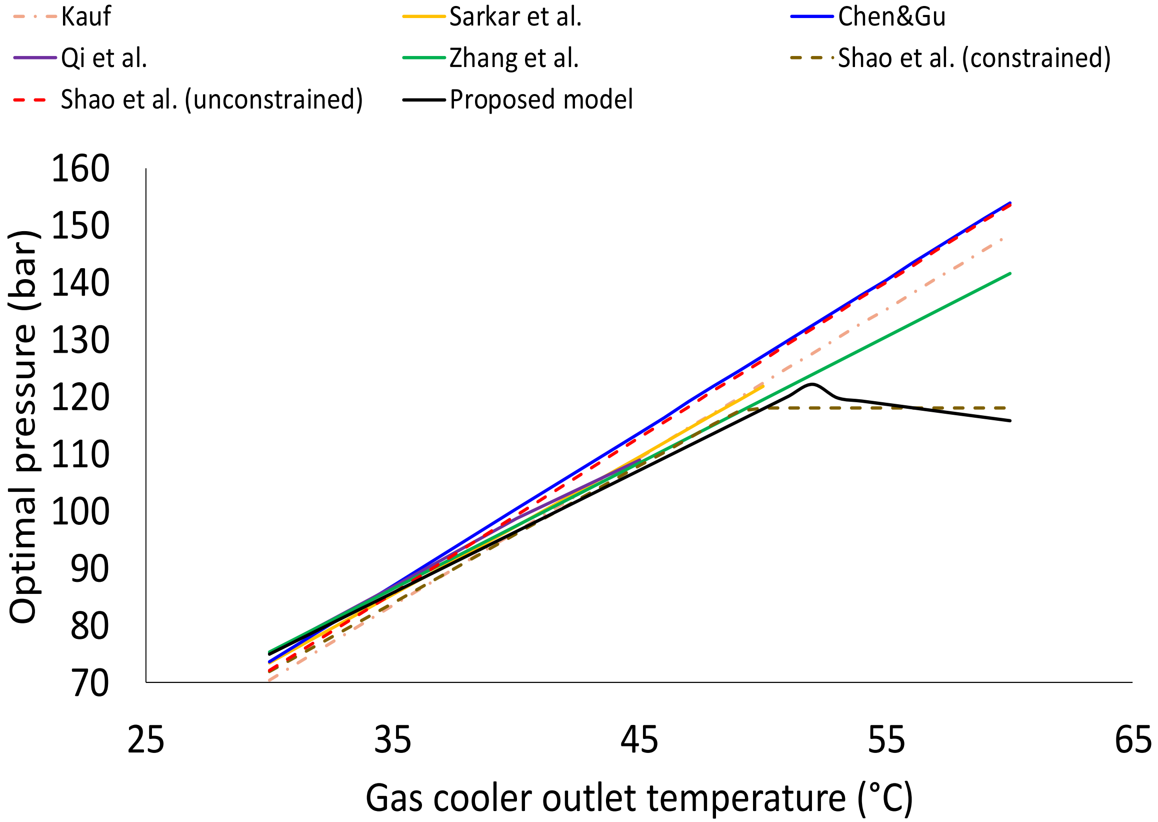
$$P_{gc,opt} = \min(140;\ 140.74 + 0.031555 \cdot T_{gc,ro} + 2.7227 \cdot T_{evap} - 1.0086 \cdot T_{comp,in})$$



(T_{evap}=5-25° C, SH=3-7 K, η_{IHX}=0-0.9, $T_{gc,ro}$=30-60° C).



Deviation between the COP values calculated with the MATLAB® model under optimal pressure conditions, and those with the same MATLAB® model, but with the optimal pressure values obtained with proposed equations.

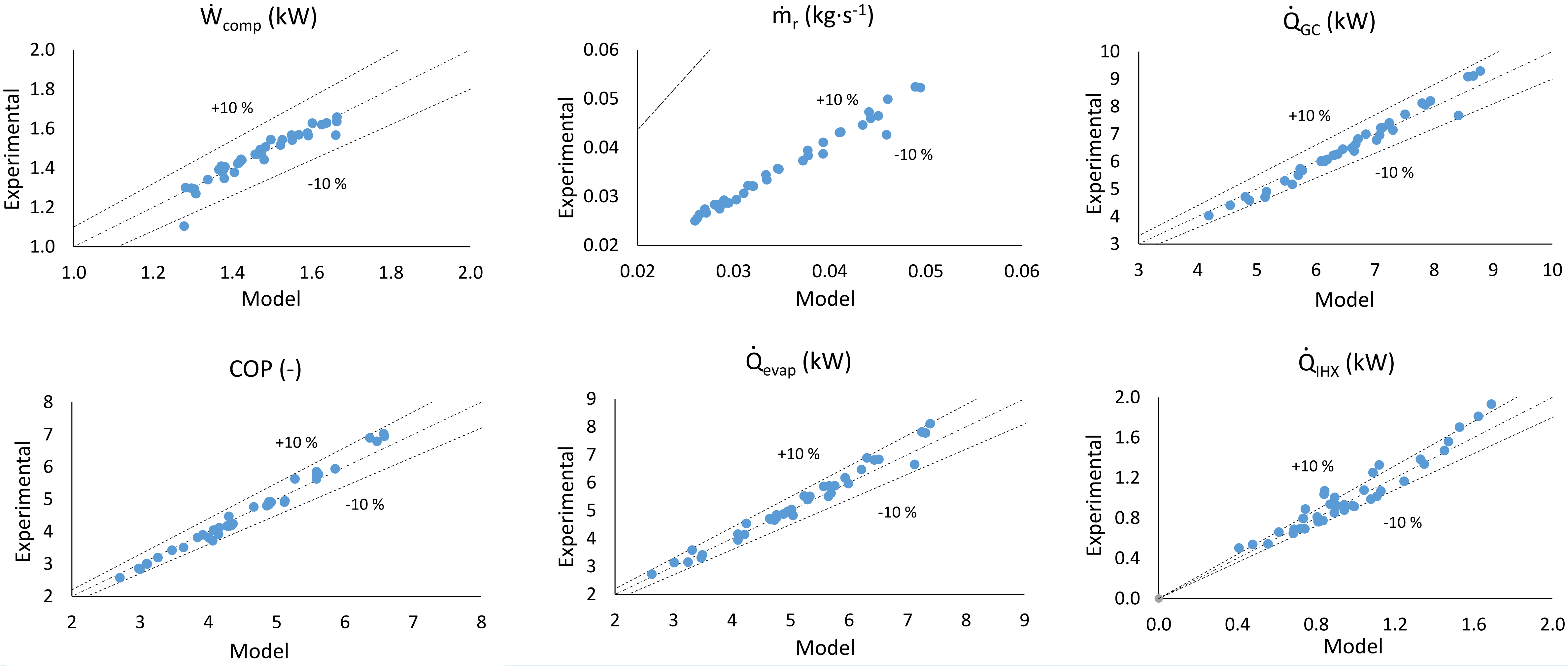


Comparison between the proposed model and previous models (T_{evap}=5 ° C, *SH*=5 K, η_{IHX}=0.6.)

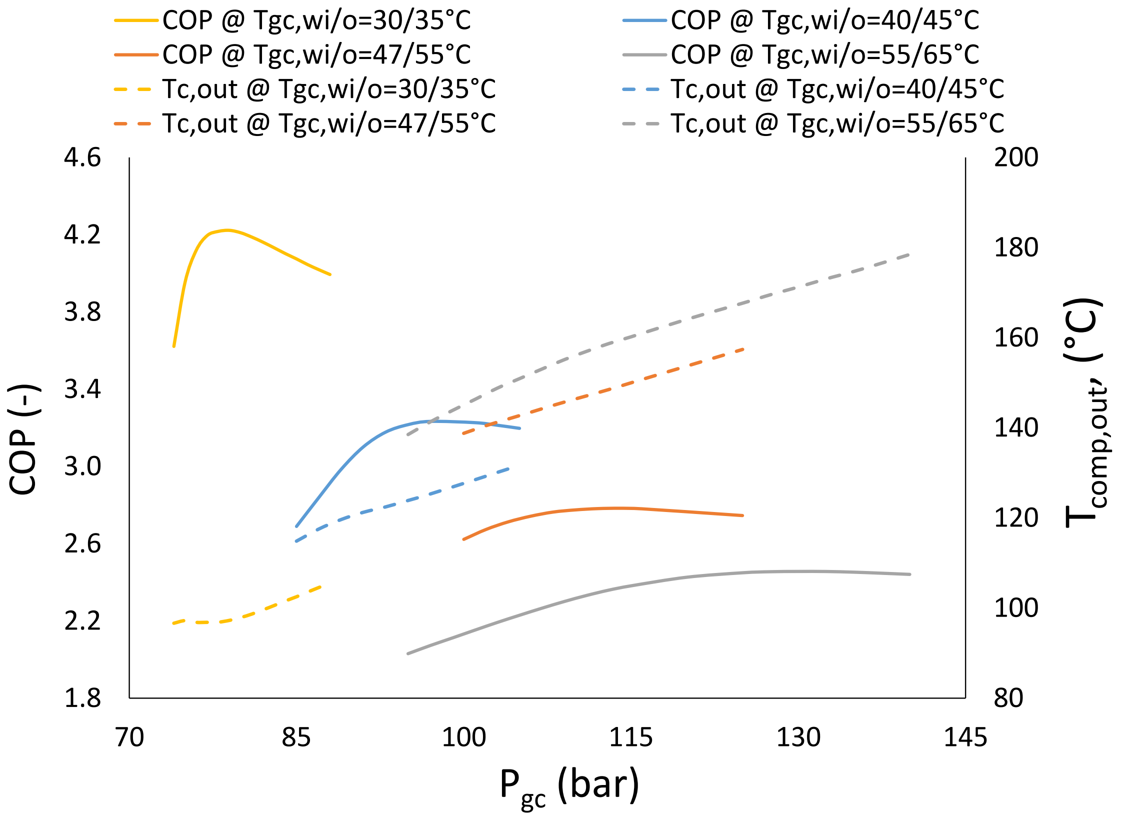
5.5. Space heating application

A detailed model for the hot water generation (low, medium, high and very high temperature), and the different heat exchangers.

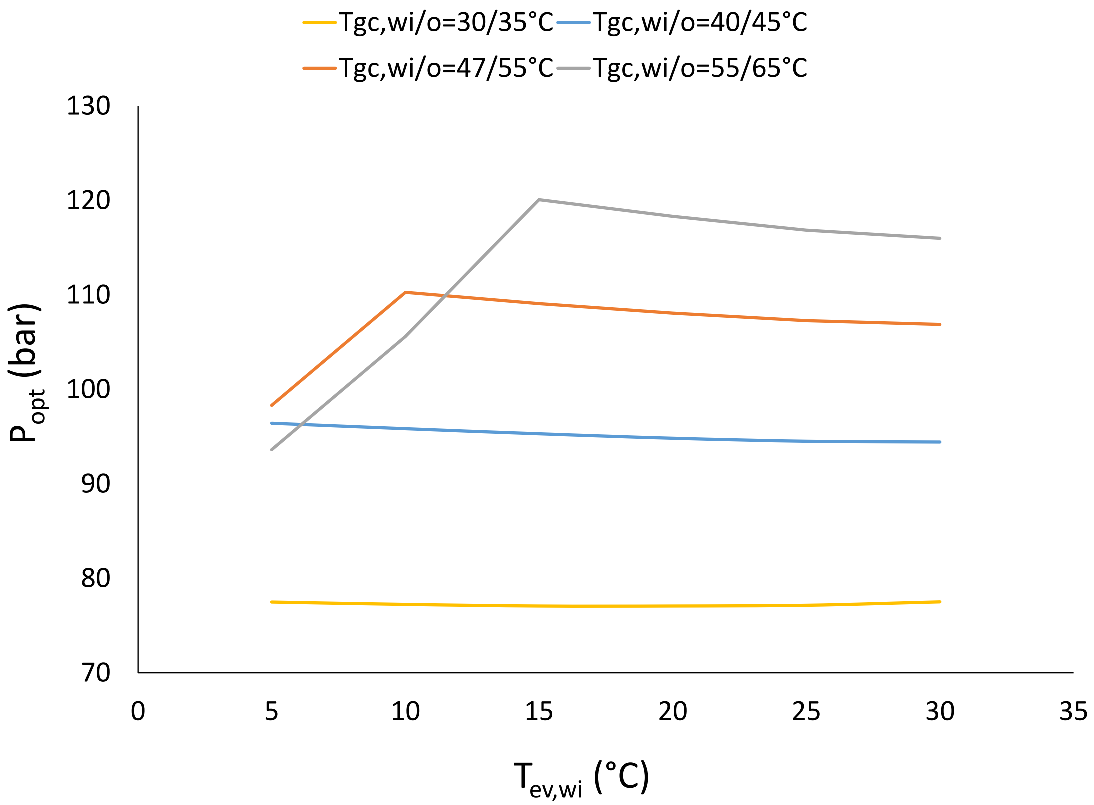
5.5.1. Model validation



5.5.2. The optimal pressure in space heating



COP and compressor outlet temperature according to the high pressure for different $T_{gc,wi/o}$ and $T_{ev,wi/o}$=10/7° C.

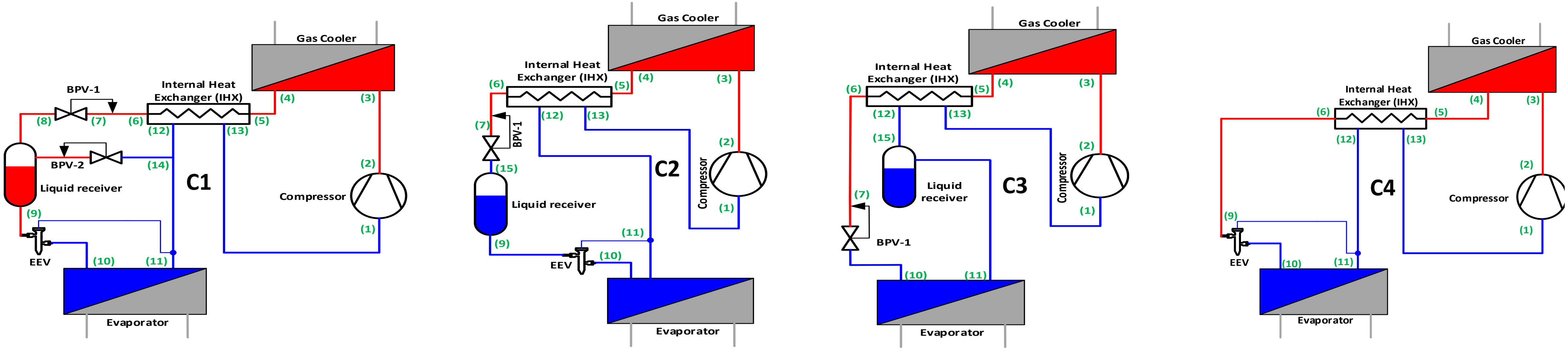


Optimal pressure according to the $T_{ev,wi}$ and limiting the $T_{comp,out}$ for different $T_{gc,wi/o}$ and $T_{ev,wi/o}$.

5.5.3. Experimental comparison of the different configurations

Tests order (#)	Evaporator		Gas cooler		$\dot{m}_{w,gc}$ ($kg \cdot s^{-1}$)			$Q_{gc,water}$ (kW)			$COP_{h,water}$ (-)		
	$T_{ev,wi}$ °C)	$T_{ev,wo}$ (°C)	$T_{gc,wi}$ (°C)	$T_{gc,wo}$ (°C)	C1&C2	C3	C4	C1&C2	C3	C4	C1&C2	C3	C4
1	10	7	30	35	0.291	0.289	0.212	6.008	5.992	4.398	4.259	4.276	2.629
2	15	X	X	35	0.291	0.289	0.211	6.665	6.678	3.806	4.951	4.994	2.314
3	20	Y	Y	35	0.291	0.289		7.393	7.296		5.855	5.943	
4	25	Z	Z	35	0.291	0.289		8.750	8.895		7.339	7.468	
5	10	7	40	45	0.237	0.235	0.196	4.886	4.862	4.109	2.958	2.942	2.404
6	15	X	X	45	0.237	0.234	0.197	5.655	5.567	3.053	3.465	3.438	1.757
7	20	Y	Y	45	0.237	0.235		6.412	6.205		4.031	3.947	
8	25	Z	Z	45	0.238	0.234		7.151	6.957		4.709	4.665	

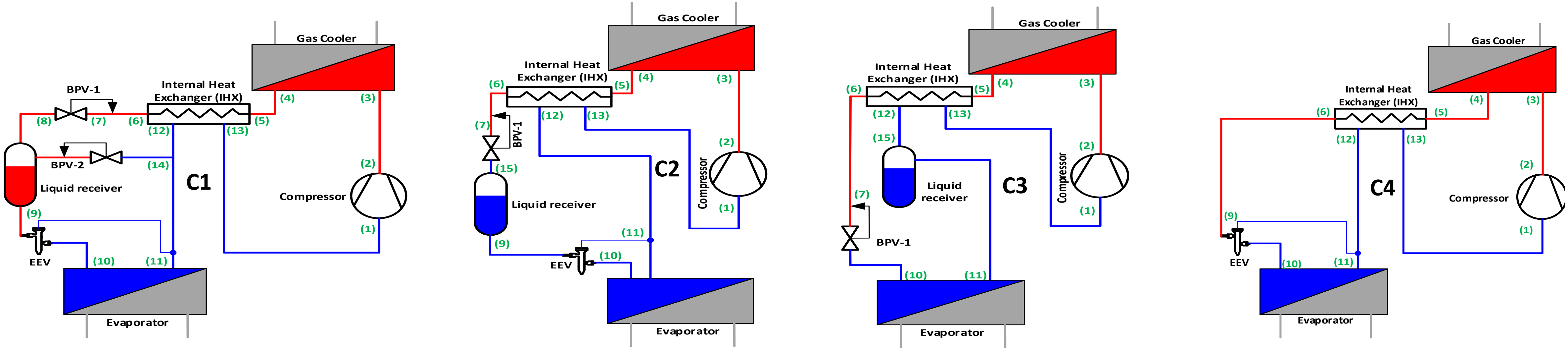
Space heating, C1&C2, C3 and C4, experimental results according to UNE-EN 14511 standard



5.5.3. Experimental comparison of the different configurations

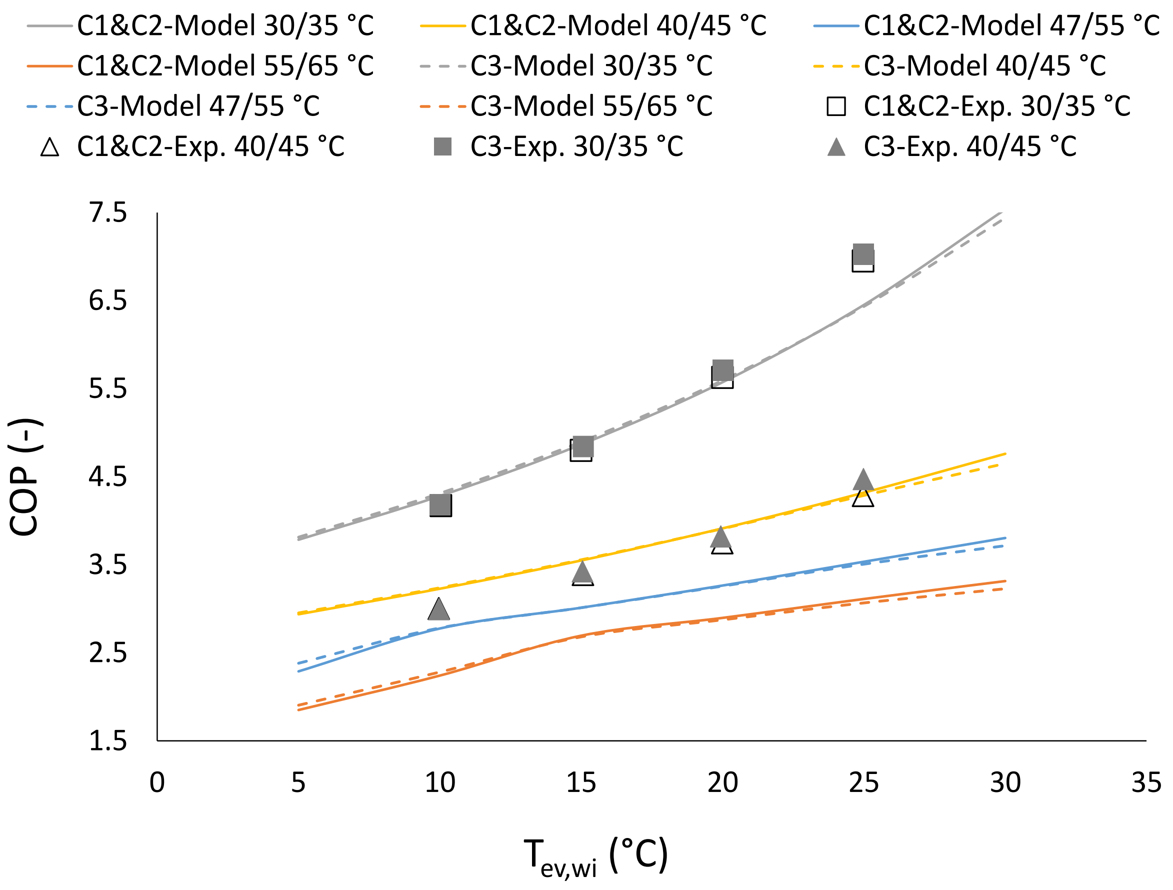
Tests order (#)	Evaporator		Gas cooler		COP_c		
	$T_{ev,wi}$ (°C)	$T_{ev,wo}$ (°C)	$T_{gc,wi}$ (°C)	$T_{gc,wo}$ (°C)	C1	C2	C3
1	12	7	30	35	3.2891	3.3180	3.2486
2	12	7	25	30	3.9280	3.9901	3.9947

Space heating, C1&C2, C3 and C4, experimental results according to UNE-EN 14511 standard

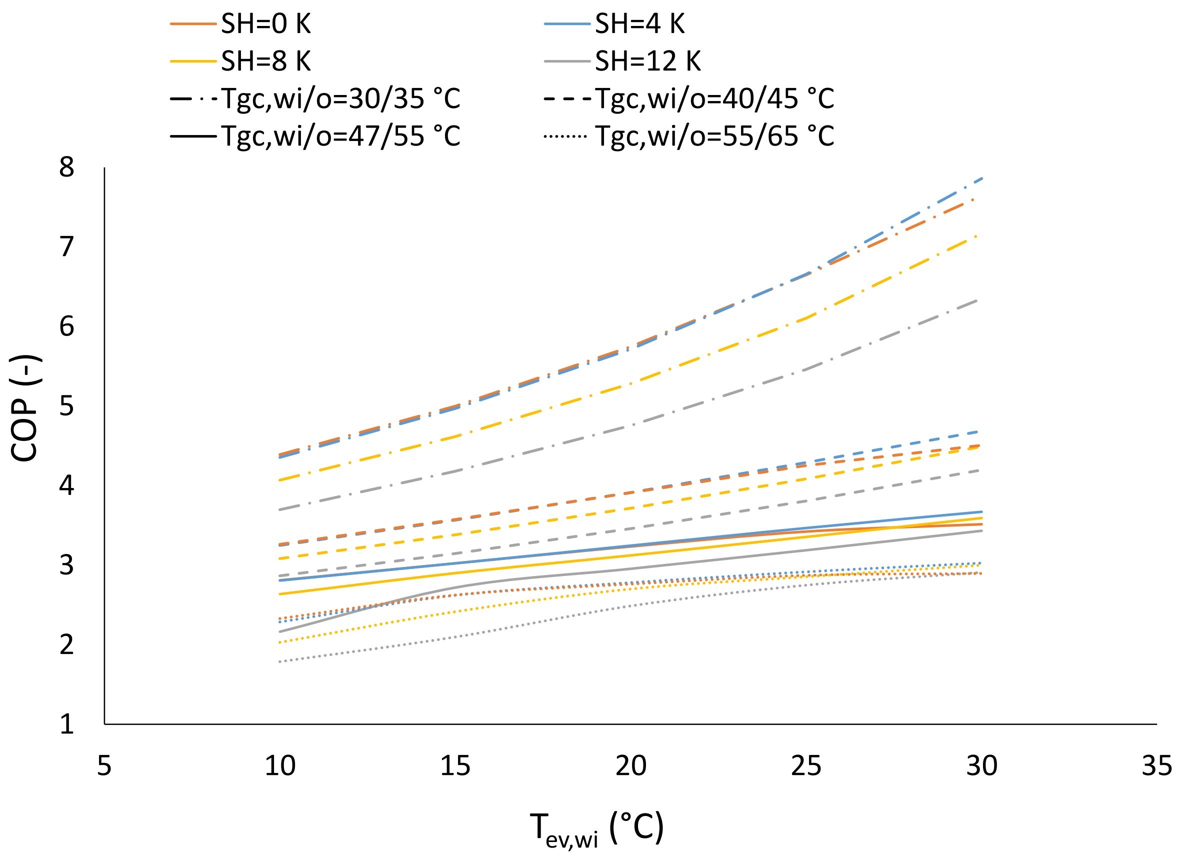


5.5.4. Influence of the evaporator water temperature and the superheating (SH)

Numerical and experimental $COP_{refrigerant}$ for different $T_{gc,wi/o}$ and $T_{ev,wi/o}$.

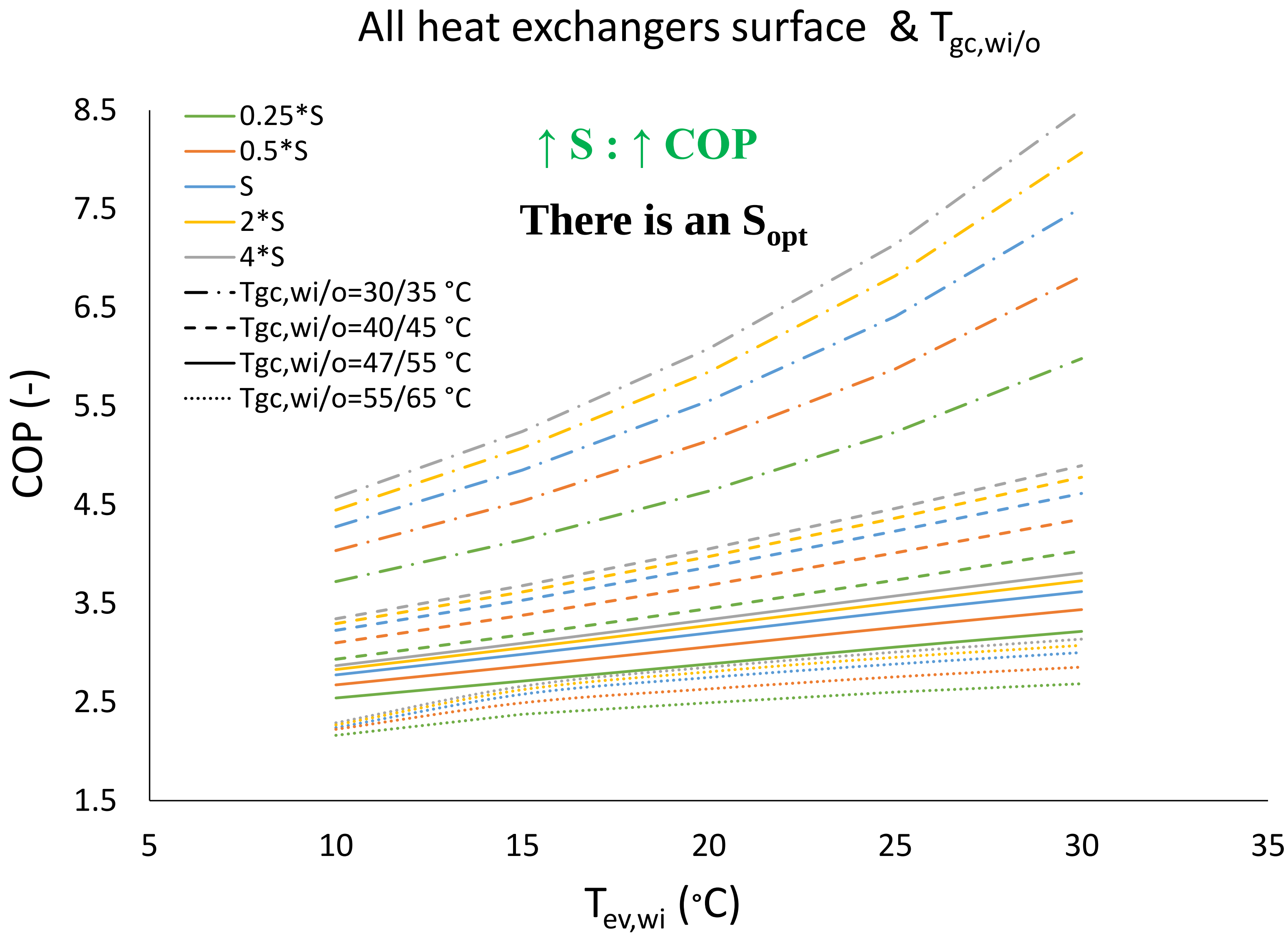


Influence of the *SH* on the $COP_{refrigerant}$ for different $T_{gc,wi/o}$ and $T_{ev,wi/o}$; $\Delta T_{ev,wi/o}$=3 K.



5. Tests methodology and results

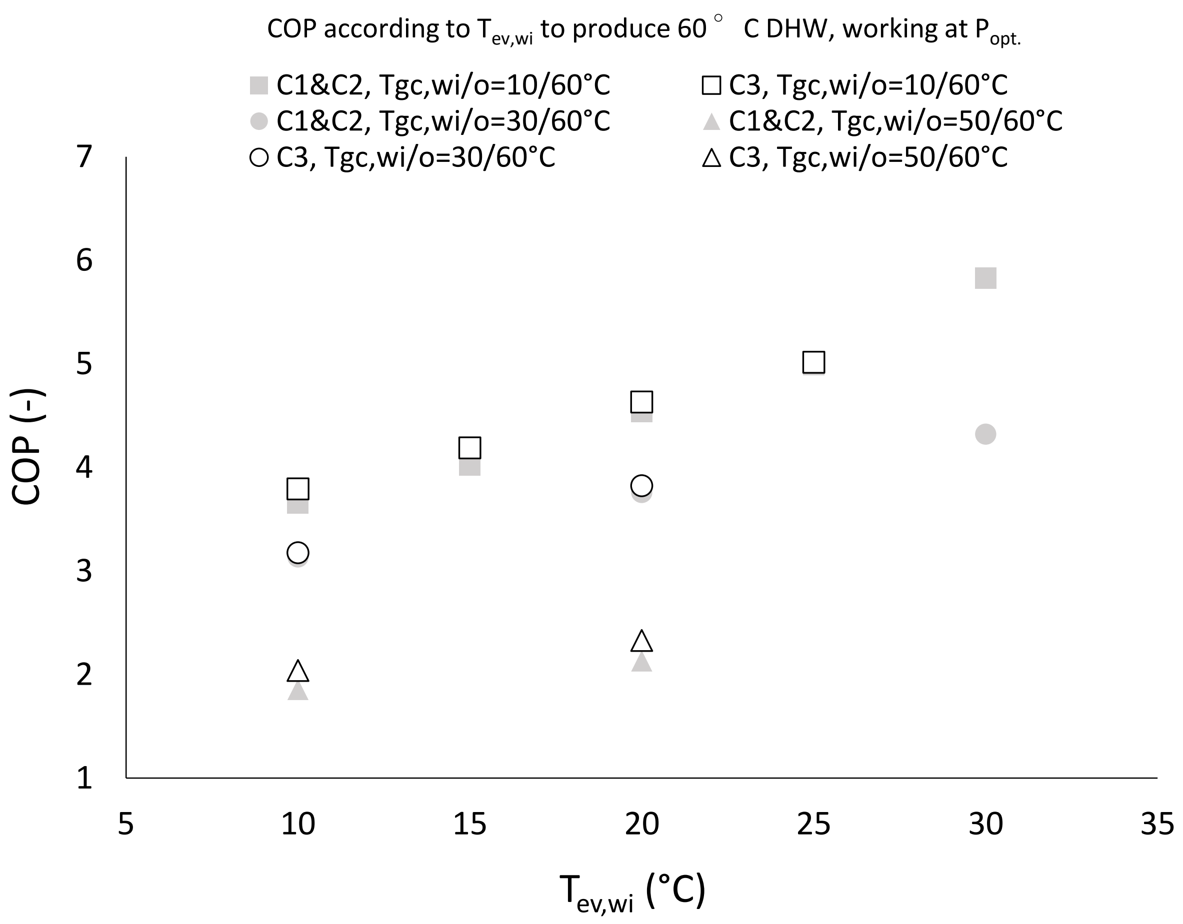
5.5.5. Varying all heat exchangers surface

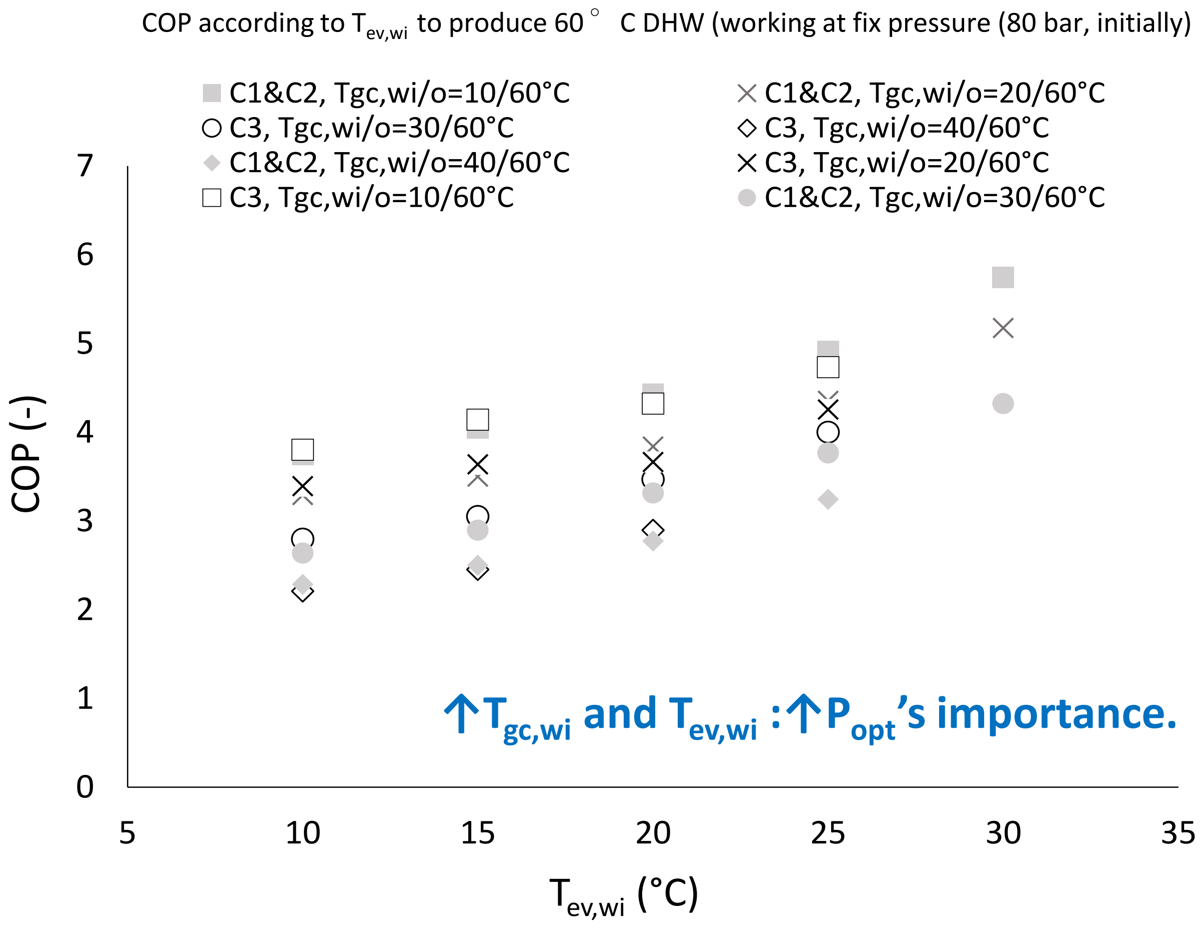


5.6. Domestic hot water (DHW) generation

5.6.1. Comparison of different configurations w/ and w/o P_{opt} to produce 60°C DHW

$T_{gc,wi}$ (°C) : 10-50 °C. **$T_{ev,wi}$** (°C) **:**10 to 30 = **$\Delta T_{ev,wi}$** (reference test)= 5 K.





CONCLUSIONS

6.1. The gas cooler optimal pressure

- The most influential parameter is the $T_{gc,ro}$. The influence of the rests of the parameters is negligible
- It is possible to obtain an expression for the optimal pressure based only on the $T_{gc,ro}$.

$$P_{gc,opt} = \min(140;\ 9.3267 + 2.1857 \cdot T_{gc,ro})$$

However, such expression does not limit the **compressor outlet temperature**, which could lead to oil degradation and compressor damages. Then, new correlations considering this parameters have been developed, and they are applicable for a wide range of reciprocating semi-hermetic compressors models since they were developed using coeficients of compressors that are already in the market.

$$T_{comp,out} = 13.403 + 2.0657 \cdot T_{gc,ro} - 2.3525 \cdot T_{evap} + 0.86806 \cdot T_{comp,in}$$

If $T_{comp,out} < 140$ °C: $P_{gc,opt} = \min(140;\ 11.047 + 2.2756 \cdot T_{gc,ro} + 0.047279 \cdot T_{evap} - 0.20814 \cdot T_{comp,in})$.

If $T_{comp,out} \geq 140$ °C: $P_{gc,opt} = \min(140;\ 140.74 + 0.031555 \cdot T_{gc,ro} + 2.7227 \cdot T_{evap} - 1.0086 \cdot T_{comp,in})$.

- Since the expressions proposed only depend on cycle conditions, they can be applied to any type of heat pump (water-to-water, air-to-water or even air-to-air) and can be easily programmed in a PLC.

6.2. Comparison of the different configurations in space heating and domestic hot water generation

- The importance of the IHX in CO_2 transcritical cycles has been confirmed.

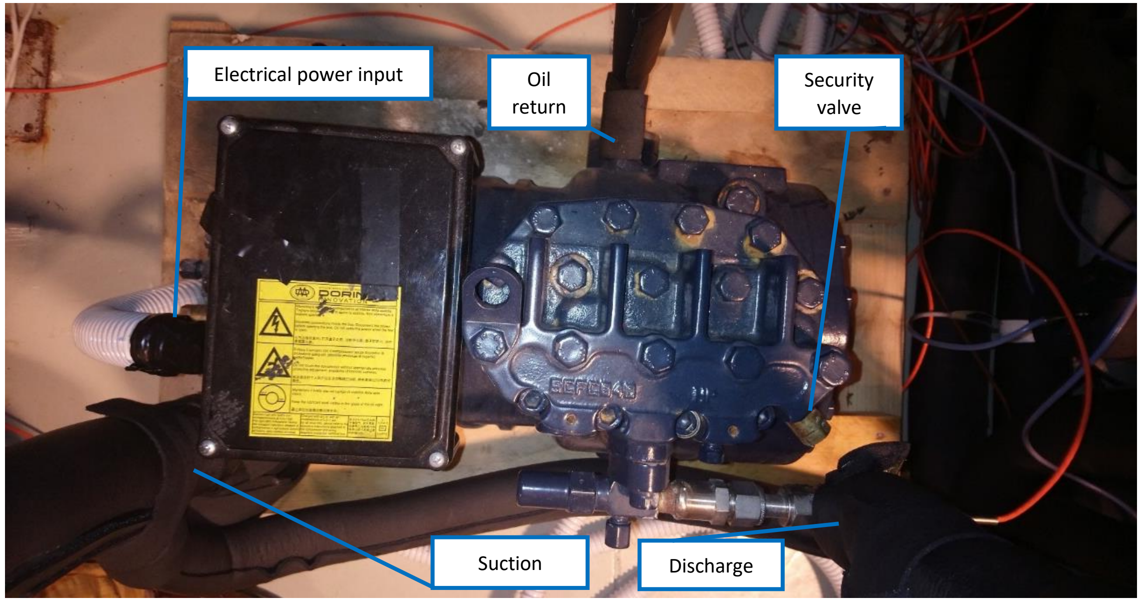
- The influence of the liquid receiver pressure is negligible.

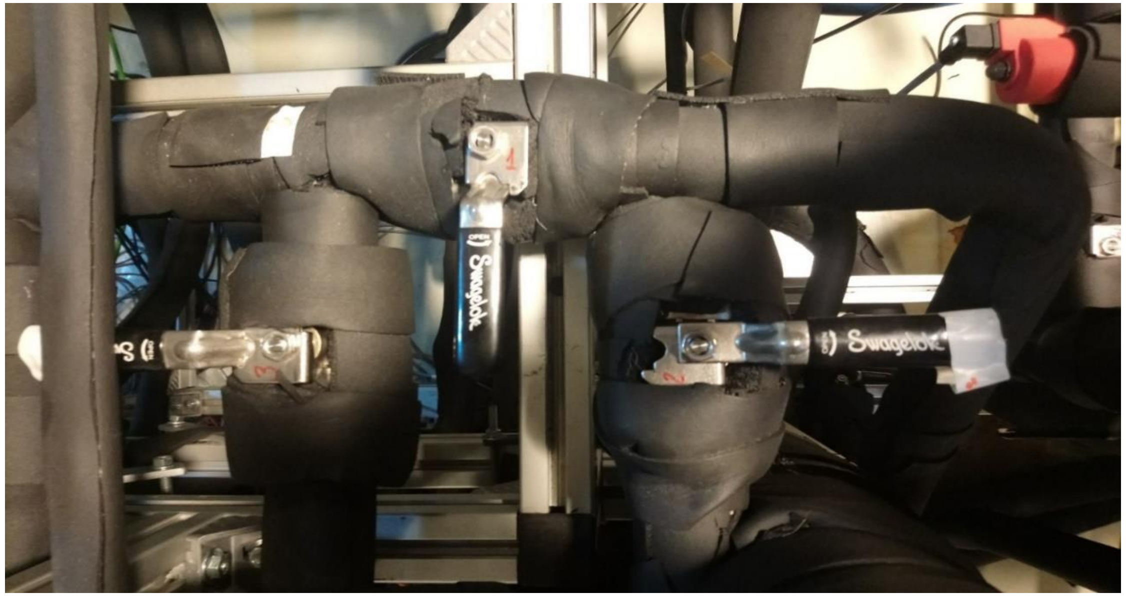
- Difference is negligible when using *flooded* or *dry evaporator (with liquid receiver)*.

- For low $T_{gc,wi/o}$ (30/35°C) the system could be designed for working around the optimal pressure, which is independent of the $T_{ev,w}$ (within the studied range). For the rests (medium, high and very high), an optimal pressure control is needed. For DWH, the more the $T_{ev,wi}$ or the $T_{gc,wi}$ increase (being the outlet 60°C), the more important is to control the optimal pressure.

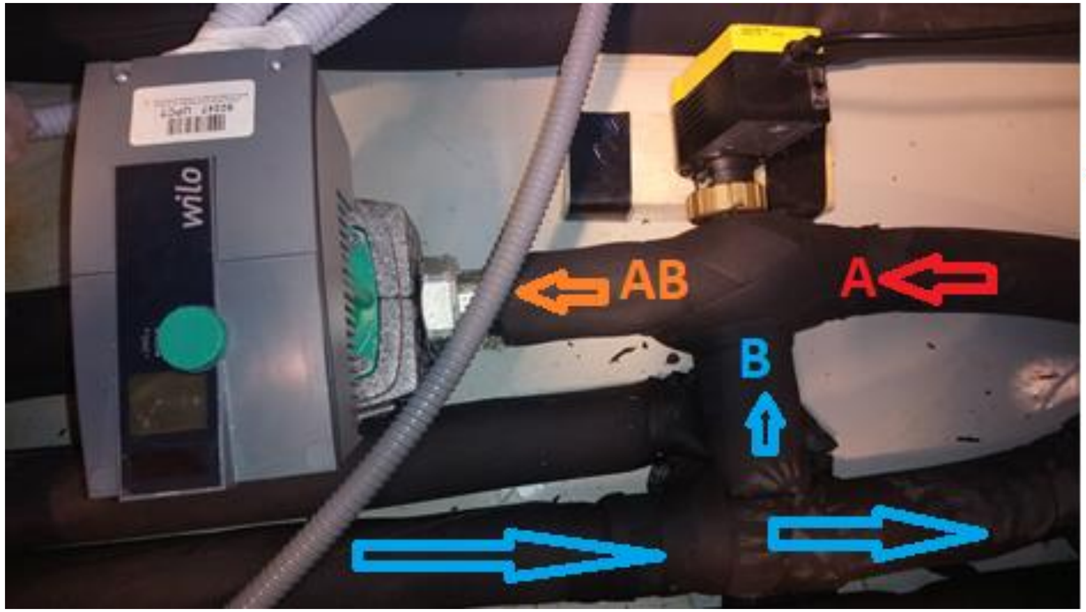
- Varying the surface of the different heat exchangers improves the system, but there exists an optimal surface for the different conditions of operation, which should be considered when designing heat exchangers.

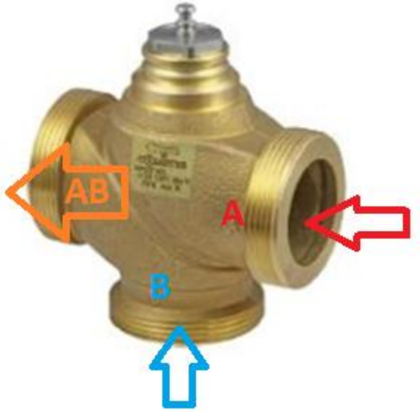
4. Facility description

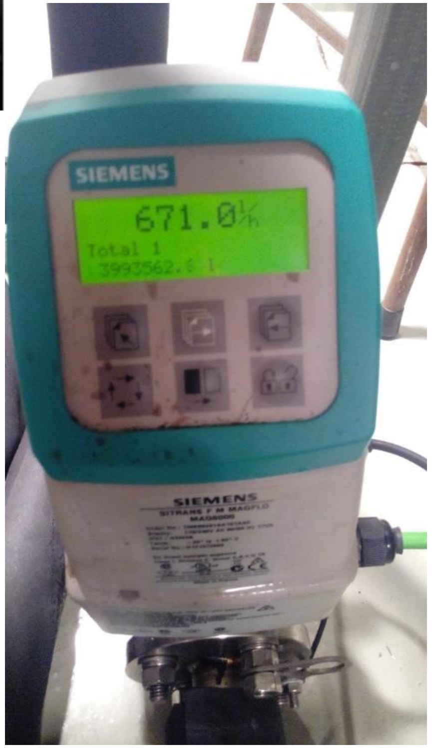




4. Facility description







Thank you for your attention

THANK YOU

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EDUCACIÓN

ESTUDIO NÚMERICO Y EXPERIMENTAL DE UNA BOMBA DE CALOR AGUA-AGUA USANDO CO_2 COMO REFRIGERANTE

Presentado por

Víctor Francisco Sena Cuevas

Universidad Politécnica de Cartagena

14 de Agosto, 2024