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Research Paper

Compressor model for CO_2-doped blends. Correction factor of EN 12900 for mixtures with R152a, R290, R1233zd(E)

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ABSTRACT

It has been experimentally verified that doping CO_2 with small amounts of other fluids enhances the COP of refrigeration systems thanks to the modification of the thermophysical properties, as it increases the critical temperature and pressure, and moves the operation to subcritical at high ambient temperatures. Up to now, theoretical studies are mainly based on the compressor information provided by the manufacturer according to the standard EN 12900 which represent the behaviour of the compressor using polynomial relations. However, manufacturers only offer information for pure-CO_2 compressor operation and there are no tools to predict their performance when working with CO_2-doped fluids. This work fills this gap since it develops a correlation for predicting the performance of CO_2 compressors when used with mixtures. A CO_2 semi-hermetic compressor was tested with mixtures (R152a at 5 and 10 %; R290 at 5 %; and R1233zd(E) at 5 % by mass) and the data was used to define a correction factor of the standard EN 12900 (for pure CO_2) to predict compressor power consumption and mass flow rate when working with mixtures. In addition, the model also allows determination of compressor's discharge temperature, volumetric and overall efficiencies. The model, developed ensuring good balance between complexity and accuracy, allows determination of these variables with a precision of 5 % using as input variables the reduced discharge and suction pressures and the reduced suction temperature. This model can be used to extend the investigation of CO_2-doped fluids in more complex refrigeration architectures.

1. Introduction

CO_2 is widely used for refrigeration applications of medium and large capacities and, during the last decades, numerous cycle improvements have been implemented to enhance its performance, such as parallel compression [1,2], subcooling systems [3–5], ejectors [6] and hybrid systems [7] among others. During the last years, there has been a rising interest to introduce refrigerant blending as a cycle enhancement mechanism [8]. Addition of other refrigerants in low quantities, maintaining low flammability [9], causes changes in the cycle operating conditions that provides COP increments when appropriate mixtures and compositions are selected. Literature about CO_2-based mixtures is not wide, however, several studies have been conducted about this matter.

From a theoretical point of view, Yoji et al. [10] studied the mixture CO_2/RE170 for a heat pump water heater working in both subcritical and transcritical operation and predicted COP enhancements up to 7.5 %. Xiaowei et al. [11] studied the pair CO_2/R290 [80/20 %] in a heat pump system where COP enhancements were predicted. In 2019, Kumar and Kumar [12] simulated the mixture CO_2/R290 [85/15 %] in a refrigeration application for a cycle with IHX and predicted similar COP values compared to CO_2, however, higher critical temperature and lower critical pressure were noted. Later Xie et al. [13] proposed CO_2/R152a and CO_2/R161 as fluids for refrigeration applications and predicted COP increments up to 30.1 % and 32.5 % respectively. Vaccaro et al. [8] performed a theoretical analysis with six fluids (R600a, R600, R290, R1234ze(E), R1234ze(Z) and R1233zd(E)) concluding that at −15 °C of evaporation and 40 °C at the exit of the gas-cooler blends can offer up to 12.8 % COP increments. Sánchez et al. [14] proposed R1270 and R32 as doping agents and predicted COP increments up to 8.7 % and 21.4 % respectively. Recently, Vaccaro et al. [15] proposed DME as additive to CO_2 with CO_2 increments up to 21.0 %.

Using an experimental approach, Kim et al. [16] tested CO_2/R290 [75/25 %] as a working fluid for air conditioning applications and measured a maximum COP increment of 12.8 %. Tobaly et al. [17] measured 19.7 % of COP increment using a scroll compressor with the fluid CO_2/R290 [90/10 %]. Yu et al. [18] worked with the same mixture

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Nomenclature

BP	back-pressure valve
CF	correction factor
COP	coefficient of performance
CR	compression ratio (−)
DME	dimethyl ether
GWP	global warming potential
h	specific enthalpy, $kJ \cdot kg^{-1}$
HC	refers to hydrocarbons
HCFO	refers to hydroclorofluoroolefins
HFC	refers to hydrofluorocarbons
$\dot{m}$	refrigerant mass flow rate, $kg \cdot s^{-1}$
NBP	normal boiling point, °C
p	pressure, bar
P_{Cons}	power consumption, W
POE	refers to synthetic Polyol Ester
s	specific volume, $m^3 \cdot kg^{-1}$
t	temperature, °C
X	mass fraction of a component (−)

Subscripts

crit	refers to the critical point of a fluid
dis	compressor discharge
fit	calculated parameter with the model
g	refers to compressor overall efficiency
gc	gas-cooler
me	mechanical–electrical
mixt	refers to mixtures
o	refers to evaporation or evaporator
suc	refers to compressor suction
vol	volumetric

Greek symbols

η	efficiency

in automobile air conditioning systems, obtaining 22 % of COP increment. Sánchez et al. [19] evaluated three blends (with R290, R1270 and R32) for positive temperature applications; they measured energy consumption reductions up to 17.2 %.

Semi-hermetic type are the most used compressors for medium to large refrigeration systems. Numerous studies have modelled these compressors with different strategies since modelling is a valuable tool for predicting compressor outputs such as mass flow rate, power consumption or discharge temperature; all of them key parameters in component selection, design, control, and problems detection. Three main approaches are followed in the literature. First, detailed models, that consider compressor geometry, oil properties or heat transfer dynamics among others which are used in compressor design by manufacturers [20]. Second, a semi-empirical approach commonly used for determining heat losses to environment or heat transfer coefficients [21]. Lastly, the empirical models which employ measurable data for fitting polynomial equations [22]. Empirical models are the most extended and used for engineering design and evaluation of refrigeration architectures; they are usually represented by the AHRI correlations [23]. AHRI correlations correspond to fitted polynomials obtained from experimental data in calorimeter systems, with 10 coefficients for compressors operating at nominal speed and with 20 for variable-speed compressors [24]. Ossorio et al. [25] recently discussed the validity of the methods and assessed other approaches for better accuracy and for simplification of their characterization. However, none of the existing studies have considered how the behaviour of a CO_2 compressor is modified when using CO_2-doped fluids. It needs to be mentioned that no manufacturer has produced yet compressors for the operation with CO_2-doped blends, and that experimentation done with CO_2-doped blends is conducted using CO_2 compressors. As this line of research is promising, as discussed previously, scientific community requires accurate models or relations to further extend the investigation with CO_2-doped fluids.

Attending to the growing interest in CO_2 mixtures, this work analyses experimentally the impact of four CO_2-based mixtures on a semi-hermetic fixed-speed compressor working in a wide range of operating conditions. Standard compressor modelling procedure (AHRI/EN 12900) is conducted with the obtained experimental data. Next, after founding poor predictions by the standard, a correction factor for the compressor working with CO_2-base mixtures is developed with a subset of the whole experimental data set. New variables which have the mixture information embedded on them (reduced pressures and temperatures) are included in the model and a statistical analysis is conducted for avoiding overfitting, reaching agreement between model complexity and accuracy. Lastly, the proposed model is validated using it against the unused data subset. Developed model constitutes a new and useful tool on the study of cycles operating CO_2-based mixtures.

2. Materials and methods

2.1. Experimental facility description

The evaluation of CO_2-doped blends was made with a water-to-water single-stage transcritical refrigeration plant with a double-stage expansion system. Three different architectures, base cycle, base cycle with internal heat exchanger (IHX) and cycle with integrated mechanical subcooling (IMS) were all tested using CO_2-doped mixtures. An E2V18 electronic valve controls the gas-cooler pressure (BP) and an E2V14 valve working as thermostatic the evaporating process. The reference compressor, Fig. 1, is a Dorin model CD380H CO_2 semi hermetic compressor [26]. It has 2 cylinders (bore and stroke lengths of 28 mm) and a displacement of 3.0 $m^3 \cdot h^{-1}$ at nominal speed (1450 rpm). It employs synthetic Polyol Ester (POE) as lubricant. Evaporator and condenser/gas-cooler are both brazed plate exchangers with 4.792 m^2 and 1.224 m^2 of heat exchange area respectively. The internal heat exchanger is a counterflow type and the subcooler used in the IMS configuration is a brazed plate exchanger of 0.850 m^2.

The main compressor (Dorin CD380H) is fully instrumented (Fig. 1). In the suction line, it has an immersion thermocouple and a pressure gauge; while in the discharge line, a surface thermocouple and a high-pressure gauge are placed. The mass-flow rate is measured by a Coriolis flow meter located upstream of the back pressure and the compressor power consumption is measured with a digital wattmeter. All the measurement devices precision as well as their calibration range are detailed in Table 1.

2.2. Experimental campaign. Test procedure

The experimental data used to develop the compressor model was obtained from the experimentation of three different layouts of the plant working with 5 different fluids [27,28]:

- CO_2
- CO_2/R152a [90/10 %]
- CO_2/R152a [95/5 %]
- CO_2/R-290 [95/5 %]
- CO_2/R-233zd(E) [95/5 %]

To generalize the model three different types of additives were

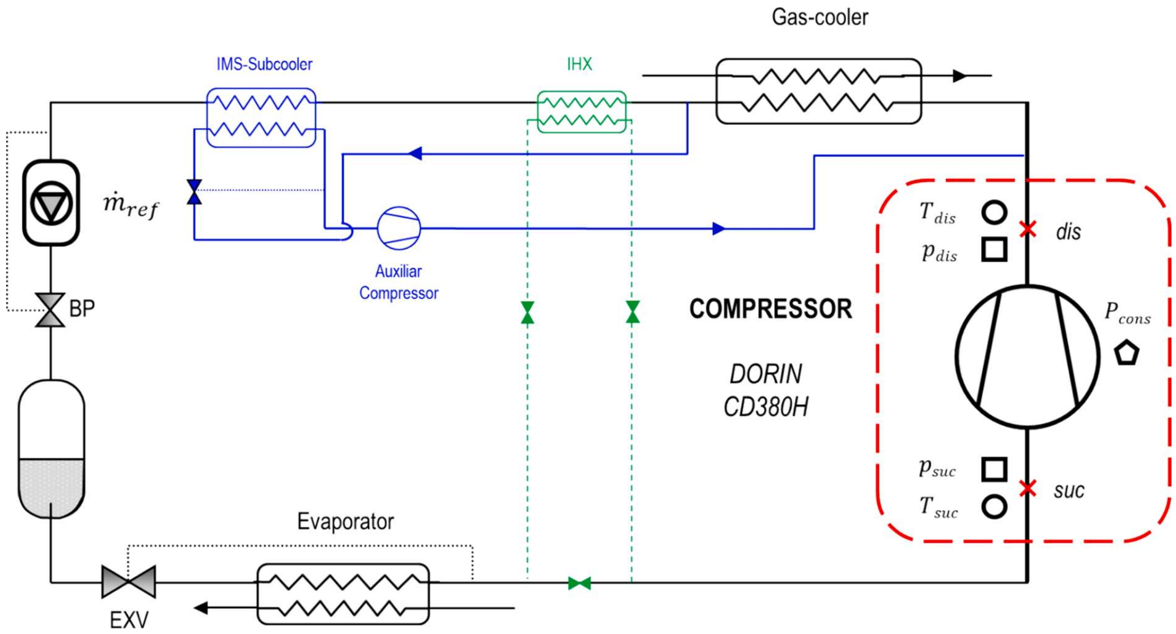

Fig. 1. Schematic diagram of the experimental set up. IHX cycle (green), IMS cycle (blue) and compressor instrumentation. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

Table 1
Calibration range and measurement error of instrumentation.

Measured variable	Device	Range	Calibrated error
Temperature	T-type thermocouple	−40.0 °C to 145.0 °C	±0.5 K
High pressure	Pressure gauge	0.0 bar to 160.0 bar	±0.96 bar
Low pressure	Pressure gauge	0.0 bar to 60.0 bar	±0.36 bar
Refrigerant mass flow rate	Coriolis	0.0 $kg·s^{-1}$ to 0.5 $kg·s^{-1}$	±0.1 % of reading
Power consumption	Digital wattmeter	0.0 kW to 5.0 kW	±0.5 % of reading

employed: CO_2/HFC, CO_2/HCFO and CO_2/HC. Molecules of additives have different polarity so, interactions with CO_2 are expected to be different. The thermophysical properties of the fluids are specified in Table 2 which have been calculated using REFPROP v.10 [29].

As shown in Table 2, doping carbon dioxide causes changes in thermophysical properties of the resulting fluid. Increased critical temperature and appearance of glide are common traits for all studied mixtures. Regarding compressor-related characteristics, suction specific volume increases with mixtures compared to pure CO_2. Furthermore, the use of mixtures causes increment on the specific isentropic compression work reaching a maximum increment of 76 % for CO_2/R1233zd(E) [95/5 %].

For each fluid, the plant was prepared the same way. Firstly, the plant was emptied, and a vacuum process of minimum two hours was conducted. The second step was the charging procedure which consisted for all cases of introducing 12 kg of total working fluid in the plant; this charge is an adequate quantity for ensuring liquid presence in the vessel which was checked visually through an eyehole. The first refrigerant to be charged is the one with higher NBP, in our case the additive (R152a, R1233zd(E) or R290). Then, CO_2 is added to complete the 12 kg in total. The entire charging process was controlled using a certified mass balance with measurement accuracy of 5.0 g.

External conditions are fixed in both condenser/gas-cooler and evaporator. The gas-cooler outlet temperature is controlled by an independent cycle that reject the heat in the gas-cooler and maintains a desired water inlet temperature to the gas-cooler. Regarding the evaporator, the evaporating level is controlled with a water/propylene glycol solution (60/40 % by volume) as secondary fluid where its inlet temperature to the evaporator is maintained constant by using electrical resistors whose control is carried out by a commercial PID controller.

Table 2
Thermophysical properties of tested mixtures (*evaluated with REFPROP v.10*).

	T_{crit} (°C)	p_{crit} (bar)	p (bar) at (−10 °C)*	Latent heat ($kJ·kg^{-1}$) at (−10 °C)	Glide (K) at (−10 °C)	p (bar) at (25 °C)*	Latent heat ($kJ·kg^{-1}$) at (25 °C)	Glide (K) at (25 °C)*	V_{suc} ($m^3·kg^{-1}$) (−10 °C)	$\dot{W}_{cp,s}$($kJ·kg^{-1}$) (−10 °C to 25 °C)
CO_2	30.97	73.77	26.49	258.61	0	64.34	119.64	0	0.01405	36.33
CO_2/R152a [95/5 %]	36.58	76.93	24.65	271.21	7.03	59.52	154.15	3.37	0.01946	47.48
CO_2/R152a [90/10 %]	41.58	78.83	22.84	280.08	13.06	54.99	178.77	7.33	0.02611	52.21
CO_2/R1233zd(E) [95/5 %]	37.23	79.21	25.32	273.36	12.41	60.66	159.10	5.18	0.02615	63.98
CO_2/R290 [95/5 %]	31.16	70.96	25.52	256.99	1.05	61.62	118.77	0.59	0.01496	36.91

* Properties evaluated with a vapor quality of 0.5.

Once the plant reached steady state conditions the test was conducted, registering the data from sensors during 10 min with a sampling rate of 5 s. During the tests, compressor was kept at nominal frequency (50 Hz). Table A.1 resumes the range of tested conditions for the compressor including pressures, temperatures, efficiencies, or mass flow rates, among others.

3. Compressor's experimental performance

Fluid thermophysical properties change between mixtures, consequently, changes in the compressor behaviour are expected. This section presents the obtained results for the compressor during the experimental campaign. The main studied parameters are the efficiencies, discharge temperature, compression ratio, mass flow rate and power consumption. All these parameters are important for understanding the effect mixtures have on the compressor. All thermophysical properties were calculated using REFPROP v.10 [29].

3.1. Compressor efficiencies

The overall efficiency of the compressor (η_g) is calculated as the product of the isentropic enthalpy difference in the compressor and the mass flow rate divided by the power consumption of the compressor as explained in Eq. (1).

$$\eta_g = \frac{\dot{m}_{ref}\cdot(h_{dis,is} - h_{suc})}{P_{cons}} \tag{1}$$

$$h_{dis,is} = f(T_{dis}; s_{suc}; mixture) \tag{2}$$

Fig. 2 depicts the overall efficiency of the main compressor in relation to the compression ratio (CR). When using mixtures overall efficiency has lower dependence on the compression ratio; however, biggest compression ratios are obtained with mixtures for the same external conditions.

The volumetric efficiency (η_{vol}) represents the efficiency of a compressor cylinder to compress gas. This parameter is calculated as stated in Eq. (3), the mass flow rate multiplied by specific volume at suction conditions divided by the compressor geometrical displacement at nominal conditions.

$$\eta_{vol} = \frac{\dot{m}_{ref}\cdot v_{suc}}{\dot{V}_g} \tag{3}$$

Fig. 3 depicts the volumetric efficiency of the main compressor in relation to the CR. As mentioned before, mixtures increase compression ratio of the compressor and depending on which is the mixture efficiencies are enhanced or reduced compared to pure CO_2.

3.2. Compressor outputs

Discharge temperature is a key operation parameter of compressors, so it is also analysed as an important output. Table A.1 collects the range of discharge temperatures for each refrigerant. The maximum registered discharge temperature for pure CO_2 is 134.5 °C. Due to additives introduction, this temperature is higher for all tested mixtures except for CO_2/R-290 [95/5 %]. With CO_2/R152a and CO_2/R1233zd(E) discharge temperatures surpassed 140 °C. Small proportions of additive have significant impact in compressor discharge temperature, furthermore, depending on which is the doping agent the temperature can increase, decrease, or remain similar.

Along with discharge temperature increment, mixtures also lead to an increased compression ratio. As commented in previous figures, introduction of additives increase CR. Biggest compression ratio is measured for CO_2/R152a [90/10 %] with a value of 4.38 while lowest CR is measured for CO_2/R-290 [95/5 %] with a value of 2.46 which is like the one obtained with CO_2.

Mass flow rate is an important parameter in the cycle design and the introduction of a second mixture component has significant implications on it. CO_2/R152a and CO_2/R1233zd(E) mixtures cause $\dot{m}_{ref}$ decrements along the entire tested range, while CO_2/R-290 [95/5 %] maintain similar or even higher values. This general mass flow rate decrease leads to a lower compressor power consumption that occurs in all the tested mixtures with the biggest reduction taking place working with CO_2/R152a [90/10 %].

4. Compressor mathematical model

Compressor outputs as well as both overall and volumetric efficiency suffer changes due to the use of the mixtures, hence, models able to predict this modified behaviour are required. Standard EN 12900 [30] stablishes the rating conditions and tolerances to present the compressor's performance data by the manufacturers. This norm dates from 2013 and nowadays an update project is ongoing. This section discusses the deviation of this standard when the compressor uses CO_2 doped blends, and then proposes a modification of the polynomials included in

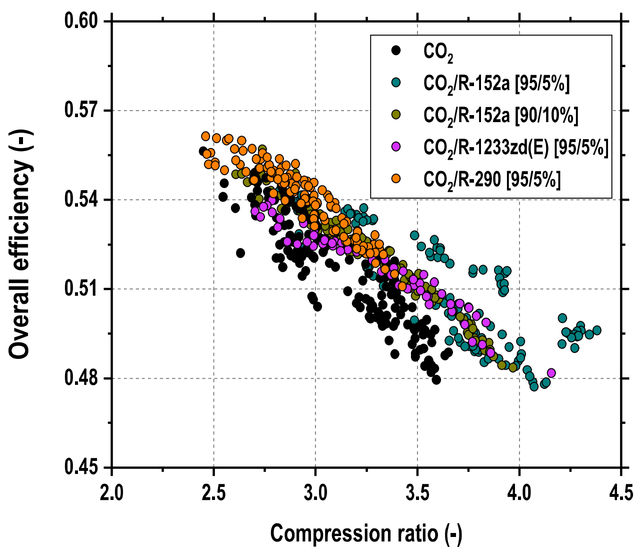

Fig. 2. Overall efficiency of the compressor.

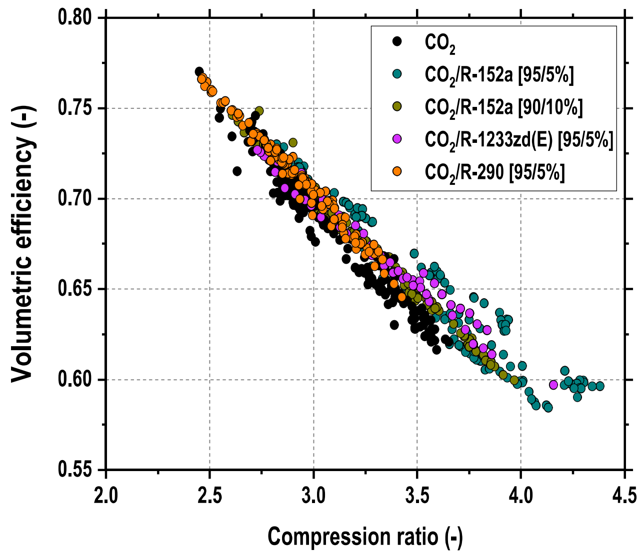

Fig. 3. Volumetric efficiency of the compressor.

Table 3
Coefficients for compressor CD380H provided by manufacturer.

	m [kg/s]	P [W]
C0	0.084356	−3929.08
C1	0.002386	−156.067
C2	−0.00039	162.7711
C3	2.26E-05	−1.98566
C4	−3.2E-06	2.398034
C5	1.03E-06	−1.10325
C6	0	−0.00823
C7	0	0.009257
C8	0	−0.0063
C9	0	0.002907

the standard to predict the behaviour of the compressor with the blends. The modification is stablished using data from CO_2/R152a and CO_2/R290 mixtures and then is validated using the experimental behaviour with CO_2/R1233zd(E).

4.1. Standard EN 12900

Compressor manufacturers provide performance data in the compressor's technical data which include values of mass flow rate and power consumption. These values are calculated using a polynomial expression given by the EN 12900 [30] which is an European regulation that specifies the nominal conditions, tolerances, and form of data presentation to be followed by the compressors manufacturers. EN 12900 propose the polynomial expressions (Eqs. (4) and (5)) as fitting expressions for compressor's power consumption and circulating mass flow rate. Specifically, the manufacturer Dorin used these standard expressions but replacing the condensing temperature by the compressor discharge pressure. For the considered compressor (CD380H), coefficients provided by the manufacturer are detailed in Table 3.

$$\dot{m}_{ref} = C_0 + C_1 \cdot T_o + C_2 \cdot p_{dis} + C_3 \cdot T_o^2 + C_4 \cdot T_o \cdot p_{dis} + C_5 \cdot p_{dis}^2 + C_6 \cdot T_o^3 + C_7 \cdot p_{dis} \cdot T_o^2 + C_8 \cdot T_o \cdot p_{dis}^2 + C_9 \cdot p_{dis}^3 \quad (5)$$

$$\dot{W}_{cp} = C_0 + C_1 \cdot T_o + C_2 \cdot p_{dis} + C_3 \cdot T_o^2 + C_4 \cdot T_o \cdot p_{dis} + C_5 \cdot p_{dis}^2 + C_6 \cdot T_o^3 + C_7 \cdot p_{dis} \cdot T_o^2 + C_8 \cdot T_o \cdot p_{dis}^2 + C_9 \cdot p_{dis}^3 \quad (4)$$

Figs. 4 and 5 present the comparison of mass flow rate and power consumption using the EN 12900 regressions (Table 3) and the data obtained through experimental tests for all tested fluids. For compressor power consumption with mixtures the predictions are sufficient in most case. From all the data set, 69 % of tests are well predicted with deviations below 5 %. The maximum deviation for isolated points reaches 27.1 %.

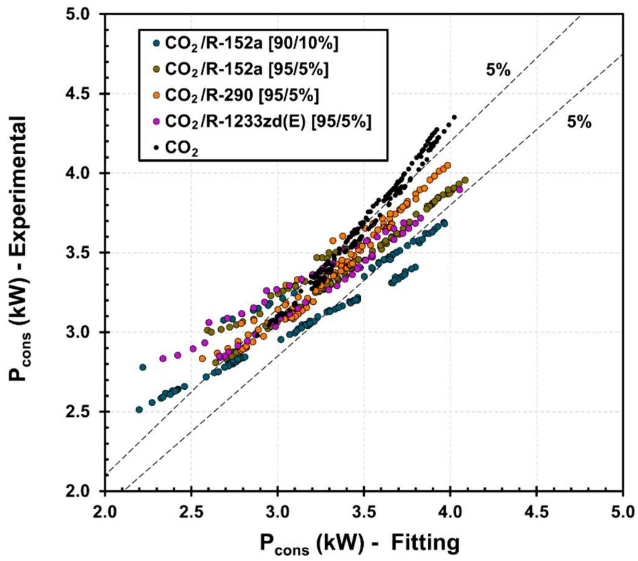

Fig. 4. Compressor output calculation, EN 12900. Power consumption.

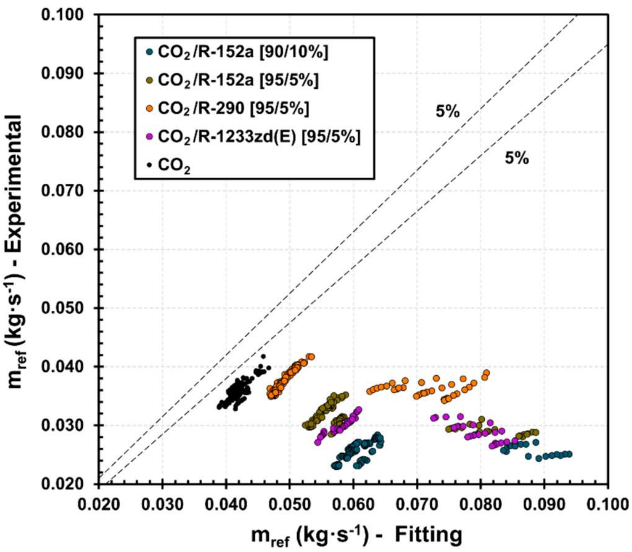

Fig. 5. Compressor output calculation, EN 12900. Mass flow rate.

Regarding mass flow rate predictions using EN 12900 correlation the results are shown in Fig. 5 where no trend nor tendency is observed. The correlation cannot accurately calculate mass flows in the main compressor, points from all the tested mixtures have deviations over 100 %. Hence, it is concluded that this expression cannot be used for calculating compressor mass flow rate with mixtures.

Regarding pure CO_2, EN 12900 can predict compressor power consumption with acceptable accuracy. Compressor mass flow rate is not so well predicted and bigger deviations are noted, nevertheless, the results follow a trend and a tendency close to reality can be inferred.

As observed, EN 12900 standard for CO_2 doped fluids cannot predict the mass flow rate and the power consumption present moderate deviations. Therefore, to further extend the investigation of CO_2-doped fluids a more precise model is required.

4.2. Correction factor of EN 12900 standard for CO_2 mixtures

The objective of this section is to present a model develop to predict the same outputs as EN 12900 with sufficient precision when using CO_2-doped fluid. This model is based on the expressions given by the manufacturers multiplied by a correction factor that considers the mixture effect on the compressor. This expression is specified by Eqs. (6) and (7) where CF_{mixt} is the correction factor applied for the mixtures while $\dot{m}_{ref,CO2}$ or $P_{cons,CO2}$ are the compressor parameters calculated with the manufacturer expressions and coefficients for pure CO_2 (Eqs. (4) and (5)).

$$\dot{m}_{ref} = \dot{m}_{ref,CO2} \cdot CF_{mixt,mref} \quad (6)$$

Table 4
Linear multipliers for Eqs. (8) and (9).

	$CF_{mixt,mref}$		$CF_{mixt,Pc}$	
	Regression coefficient – a_n value	p-value	Regression coefficient – a_n value	p-value
a_0	−0.00288	6.57E−01	1.0851	0
a_1	−0.08785	0	−0.36231	0
a_2	2.32881	0	0.85379	0
a_3	−0.71747	0	0.19381	0

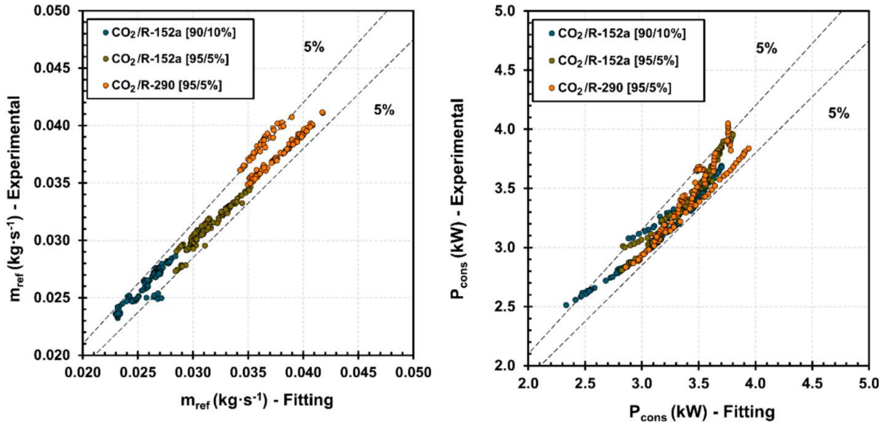

Fig. 6. $\dot{m}_{ref}$And p_{cons} adjusted with CF_{mixt} (Eqs. (8) and (9)).

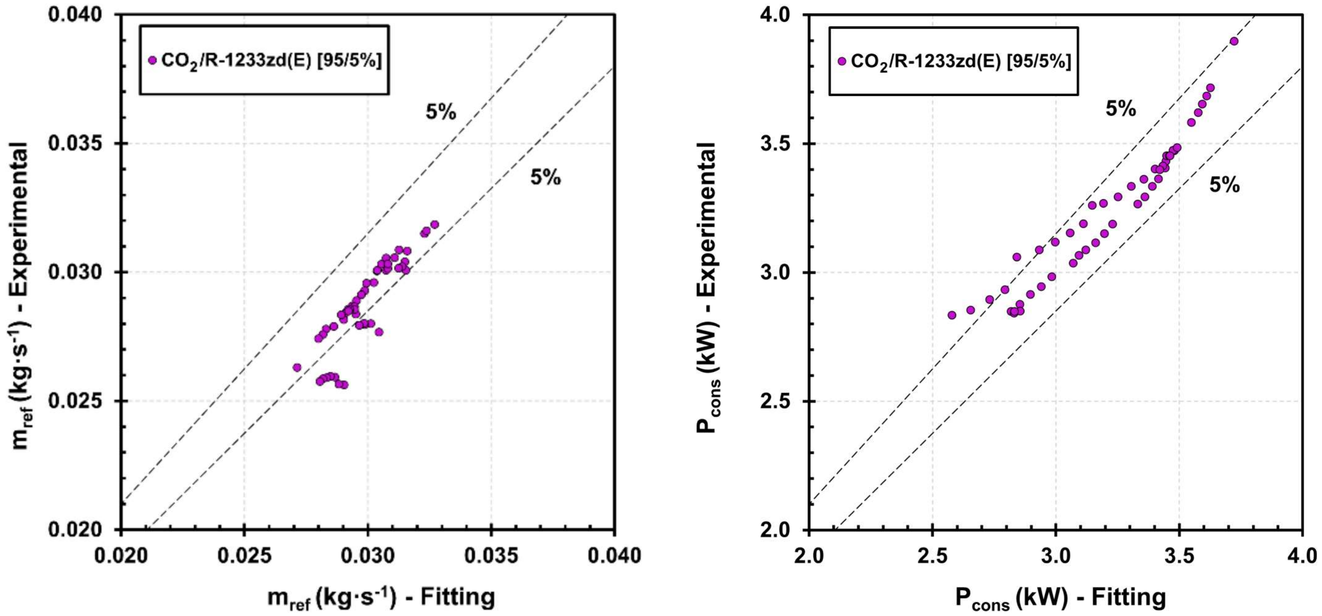

Fig. 7. Validation with CO_2/R1233zd(E) – Compressor's mass flow rate (left) and power consumption (right).

$$P_{cons} = P_{cons,CO2} \cdot CF_{mixt,Pc} \tag{7}$$

Correction factor (CF_{mixt}) should contain enough information about the refrigerant mixture, so the molar mass of the mixture, the mass fraction of CO_2, and the critical point properties of the mixture were considered. Authors investigated the most significant parameters on the fitting procedures, with the objective of obtaining the most simplified expression and avoiding overfitting problems [31,32]. Through a *t*-test analysis we concluded that the molar mass of the mixture and the mass fraction of CO_2 were not relevant, since all the information characterising the mixture was already included in the critical point definition. Therefore, authors proposed the definition of the correction factor defined by Eqs. (8) and (9), where the information of the mixture is included as reduced pressures and temperatures using the critical point properties (P_{crit}, T_{crit}) calculated with REFPROP v.10.

$$CF_{mixt,mref} = a_0 + a_1 \cdot \frac{p_{dis}}{p_{crit}} + a_2 \cdot \frac{p_{suc}}{p_{crit}} + a_3 \cdot \frac{T_{suc}}{T_{crit}} \tag{8}$$

$$CF_{mixt,Pc} = a_0 + a_1 \cdot \frac{p_{dis}}{p_{crit}} + a_2 \cdot \frac{p_{suc}}{p_{crit}} + a_3 \cdot \frac{T_{suc}}{T_{crit}} \tag{9}$$

Coefficients a_0 to a_3 were fitted using experimental data with the mixtures CO_2/R152a [90/10 %], CO_2/R152a [95/5 %] and CO_2/R290 [95/5 %] through a least squares regression, the adjusted coefficients and the p-values of the fitting procedure being detailed in Table 4. Fig. 6 depicts experimental $\dot{m}_{ref}$ and P_{cons} against calculated values with Eqs. (8) and (9). Significant improvements are accomplished with the introduction of the correction factor. Power consumption calculated values reach a maximum deviation of 6.1 % while the biggest deviation for $\dot{m}_{ref}$ is 6.7 %. Mass flow rate deviation is below 5 % for 90.9 % of the whole data set while none of the test was inside this deviation range using EN 12900 expression.

Accordingly, the proposed correction factor can predict with high accuracy, at least for the fluids considered for the fitting procedure, the output parameters of a compressor when using CO_2-doped mixtures.

To generalize the proposed correction factor, the model detailed by Eqs. (8) and (9) was used to predict the behaviour of the compressor when operating with CO_2/R1233zd(E) [95/5 %] mixture. R1233zd(E) is an HFO with different polarity to R152a or R-290, and even the mixing rules used by REFPROP v.10 are estimated due to the lack of experimental measurements of this pair. Fig. 7 shows the validation of

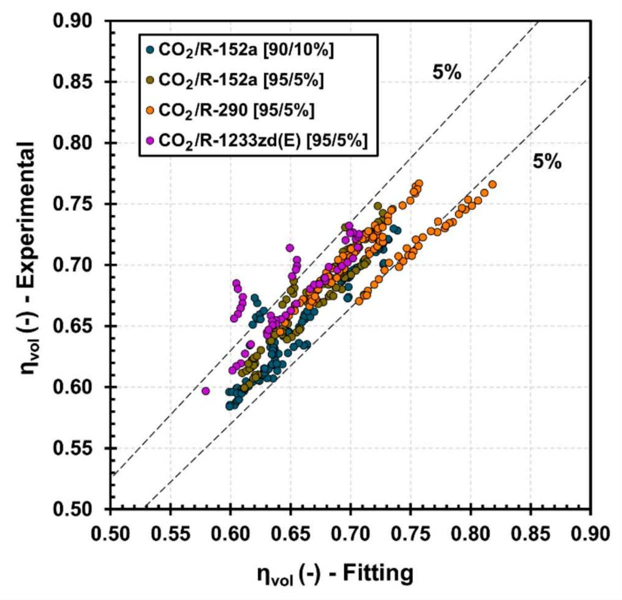

Fig. 8. Estimated vs. experimental compressor's volumetric efficiency.

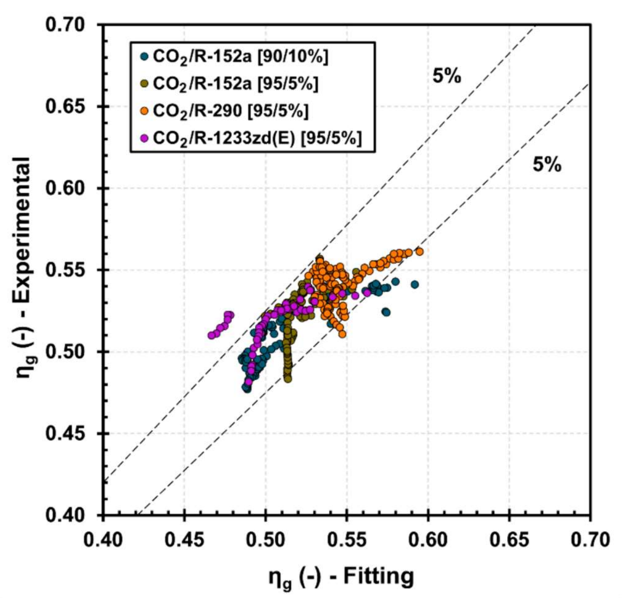

Fig. 9. Estimated vs. experimental compressor's overall efficiency.

compressor outputs for CO_2/R1233zd(E) [95/5 %] mixture. Regressions predict with accuracy both the mass flow rate and power consumption since all errors are maintained below 10 %. The maximum deviations are 11.7 % and 9.0 % for mass flow rate and power consumption, respectively.

Therefore, it can be stated that the proposed correction factor is able to predict the output parameters of a compressor operating with CO_2-doped fluids using only the data provided by the manufacturer and the information of the blend (T_{crit}, P_{crit}) with enough accuracy to investigate refrigeration cycles.

4.3. Compressor efficiencies and discharge temperature

Although standard EN 12900 does not include a specific relationship

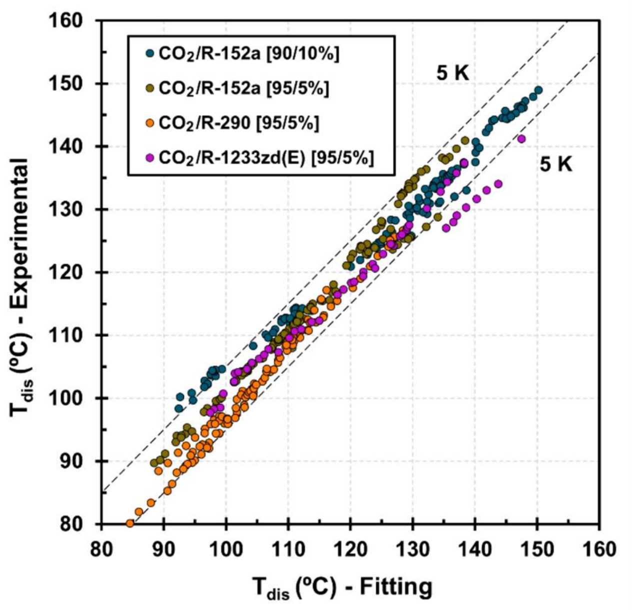

Fig. 10. Estimated vs. experimental compressor's discharge temperature.

to calculate the compressor's discharge temperature and their efficiencies (volumetric and overall), these values are of relevant importance to investigate thermodynamic cycles, especially with refrigerant mixtures for heat pump purposes. Therefore, this section uses the previous adjusted model (Eqs. (6)–(9)), to determine the compressor's efficiencies and use them to predict the compressor's discharge temperature. The validation is made for all considered refrigerant mixtures.

4.3.1. Compressor's efficiencies

Volumetric efficiency is calculated as the product of mass flow rate and specific suction volume divided by the compressor displacement, as stated in Eq. (10). Mass flow rate used to estimate the volumetric efficiency is that previously calculated with Eq. (8), where the subindex '*fit*' indicates the parameter is calculated using the developed model.

$$\eta_{vol,fit} = \frac{\dot{m}_{ref,fit} \cdot \nu_{suc}}{\dot{V}_g} \tag{10}$$

Compressor overall efficiency is computed as the product of mass flow rate and the specific isentropic compression work divided by the estimated compressor's power consumption, as stablished by Eq. (9). Eq. (11) defines the compressor overall efficiency as the isentropic work of the compressor multiplied by the mass flow rate divided by the compressor power consumption previously obtained. Mass flow rate and compressor power consumption are calculated with Eqs. (8) and (9) respectively.

$$\eta_{g,fit} = \frac{\dot{m}_{ref,fit} \cdot (h_{dis,s} - h_{suc})}{P_{c,fit}} \tag{11}$$

Fig. 8 presents the relationship between the measured and estimated compressor's volumetric efficiency and Fig. 9 of the compressor's overall efficiency. Each graph depicts the measured value of each variable versus the same variable value calculated using Eqs. (10) and (11).

For overall efficiency calculations, the maximum error has an absolute value of 9.6 % while for the volumetric efficiency, the error is maintained under 12.3 %. A total 9.6 % of calculated efficiencies have deviations below 5 % while 88.0 % of calculated η_{vol} are below the same limit. These low error values, confirm the quality of the regressions and the proposed model and enlarge their use for computing compressor efficiencies which are commonly used parameters in compressor studies.

4.3.2. Compressor's discharge temperature

The last considered compressor output of interest the compressor discharge temperature, which is calculated via the discharge enthalpy, using Eq. (12). Discharge temperature is a function of discharge pressure, discharge enthalpy and working fluid (13).

$$h_{dis} = \frac{P_{C,fluid}}{\dot{m}_{ref,fit}} + h_{suc} \tag{12}$$

$$T_{dis} = f(p_{dis}; h_{dis}; mixture) \tag{13}$$

In Eq. (12) the term noted as $P_{C,\mathrm{fluid}}$ is the power consumption invested in the compression of the fluid. Total power consumption of the compressor also takes into consideration heat transfer to the environment, other losses and irreversibilities. $P_{C,\mathrm{fluid}}$ is defined as the total power consumption multiplied by a term that takes into consideration electrical and mechanical losses. In the literature, this factor takes different values depending on the study. Sánchez [33] considered a constant value of 0.8 for a CO_2 semi-hermetic compressor. Attending to literature, the electro-mechanical efficiency is fixed at a constant value of 0.78 for all the working fluids since it mainly depends on the compressor (Eq. (15)).

$$P_{C,fluid} = P_{C,fit} \cdot \eta_{me} \tag{14}$$

$$\eta_{me} = 0.78 \tag{15}$$

Fig. 10 shows the validation of the compressor discharge temperature using the developed model in relation to the measured data in the real compressor. The estimated value calculated with Eqs. (12)–(15) has low deviations compared to experimental data. In general, 94.1 % of the calculated temperatures have deviations below 5 K reaching a maximum deviation of 7.7 K for the adjusting data set. For non-fitting data set the maximum deviation reaches 9.7 K. These values are low; hence, the approximation is considered sufficient for their application in modelling; nevertheless, this value which is accurate according to our data and provides good predictions may be different for other compressors.

Accordingly, the proposed model predicts compressor behaviour using the manufacturer expressions adjusted with CF_{mixt} coefficients reaching sufficient accuracy and being consistent along all the tested conditions/configuration for all the studied mixtures.

5. Conclusions

This work presents a simplified numerical model of a CO_2 semi-hermetic compressor that allows estimation of mass flow rate, power consumption and discharge temperature when the compressor is used with CO_2-doped mixtures. The model has been developed based on the dada provided by the manufacturer which complies the standard EN 12900. These expressions have been corrected using a correction factor that characterizes the main thermophysical properties of the mixture.

Development of the model is based on experimental results of a CO_2 semi-hermetic compressor that was operated with pure CO_2 and four CO_2-based mixtures: CO_2/R152a (5 % and 10 % of R152a), CO_2/R-290 (5 % of additive) and CO_2/R1233zd(E) (5 % of additive). The experimental campaign was performed for a constant evaporating level and heat rejection levels ranging from 20 °C to 40 °C.

The proposed correction factor only depends on the use of reduced pressures and temperatures and allows estimating the performance of the compressor from the manufacturer data multiplied by a correction factor. Selected inputs are the suction and discharge pressures, the suction temperature and the critical temperature and pressures for the considered mixture. Selected correction factor corresponds to a polynomial with only four coefficients. Determination of the correlation coefficients was made using experimental data of CO_2/R152a (5 % and 10 % of R152a) and CO_2/R290 (5 % of additive) and the model was validated using the experimental data of the mixture CO_2/R1233zd(E) (5 % of additive). In relation to the experimental data, proposed regressions predict data from fitting mixtures with high accuracy. From all mass flow results, 90.9 % of them have deviations below 5 % while the power consumption results show a maximum deviation of 6.1 % and 93.2 % of the points with errors below 5 %. Regarding the utilization of the model against unseen data, 76 % of points have deviations below 5 % (maximum 11.7 %) for the mass flow while 90 % of points present deviation below 5 % (maximum 9 %) for power consumption. Furthermore, discharge temperature has been discussed: with CO_2/R1233zd(E) 85.7 % points have deviation below 5 K and a maximum difference of 9.7 K has been noted. The model can predict the behaviour of the compressor when working with the adjusting mixtures and it was validated with the additive R1233zd(E) which is a fluid with significant differences in relation to the others tested. Hence, mixtures of CO_2 with other additives, including HFC/HFO/HC, are expected to be well-predicted by the model.

Therefore, this study proposes a correction factor of EN 12900 coefficients which can be used to predict the behaviour of CO_2 compressors when operating with CO_2 refrigerant mixtures with low proportion of additive, an identified gap in the literature. This correction factor only depends on three variables and corrects completely the poor results obtained with EN 12900. This work represents a contribution on the study CO_2 mixtures as its results can be used to generate models with high accuracy for thermodynamic cycles evaluation.

CRediT authorship contribution statement

M. Martínez-Ángeles: Writing – original draft, Validation, Software, Methodology, Investigation, Formal analysis, Conceptualization. **D. Calleja-Anta:** Writing – review & editing. **L. Nebot-Andrés:** Writing – original draft, Supervision, Data curation. **R. Llopis:** Writing – review & editing, Supervision, Funding acquisition.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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Appendix A

Table A1
Variable ranges for tested mixtures.

	P_{suc} (bar)		P_{dis} (bar)		RC (−)		T_{suc} (°C)		T_{dis} (°C)		$\dot{m}_{ref}$ (kg·s^{-1})		P_{Cons} (kW)		η_g(−)		η_{vol}(−)		N° of tests
	min	max	min	max	min	max	min	max	min	max	min	max	min	max	min	max	min	max	
CO_2	23.92	28.05	64.08	99.89	2.45	3.65	−3.99	12.08	78.58	134.51	0.0328	0.0418	2.98	4.35	0.479	0.556	0.616	0.773	172
CO_2/R152a [90/10 %]	19.25	22.22	54.99	91.48	2.82	4.38	3.98	18.77	98.40	148.96	0.0230	0.0284	2.51	3.69	0.477	0.543	0.584	0.730	119
CO_2/R152a [95/5 %]	22.50	24.47	61.08	95.39	2.61	3.97	0.98	17.08	89.76	140.98	0.0281	0.0352	2.81	3.96	0.484	0.557	0.600	0.748	118
CO_2/R290 [95/5 %]	24.09	28.76	62.23	94.37	2.46	3.43	−2.59	14.10	78.40	126.70	0.0343	0.0418	2.83	4.05	0.511	0.561	0.646	0.767	113
CO_2/R1233zd(E) [95/5 %]	21.91	22.87	61.11	94.91	2.70	4.16	3.13	16.05	97.75	142.48	0.0265	0.0327	2.83	3.90	0.482	0.540	0.597	0.732	50

References

[1] M. Karampour, S. Sawalha, State-of-the-art integrated CO2 refrigeration system for supermarkets: a comparative analysis, Int. J. Refrig 86 (2018) 239–257.

[2] L. Nebot-Andrés, D. Sánchez, D. Calleja-Anta, R. Cabello, R. Llopis, Experimental determination of the optimum intermediate and gas-cooler pressures of a commercial transcritical CO2 refrigeration plant with parallel compression, Appl. Therm. Eng. 189 (2021).

[3] L. Nebot-Andrés, D. Calleja-Anta, C. Fossi, D. Sánchez, R. Cabello, R. Llopis, Experimental assessment of different extraction points for the integrated mechanical subcooling system of a CO2 transcritical plant, Int. J. Refrig. 136 (2022) 8–16.

[4] L. Nebot-Andrés, J. Catalán-Gil, D. Sánchez, D. Calleja-Anta, R. Cabello, R. Llopis, Experimental determination of the optimum working conditions of a transcritical CO2 refrigeration plant with integrated mechanical subcooling, Int. J. Refrig. 113 (2020) 266–275.

[5] A. Paez, B. Ballot-Miguet, B. Michel, P. Tobaly, R. Revellin, Experimental investigation of a new CO2 refrigeration system arrangement for supermarket applications, Int. J. Refrig. (2024).

[6] A. Hafner, S. Försterling, K. Banasiak, Multi-ejector concept for R-744 supermarket refrigeration, Int. J. Refrig. 43 (2014) 1–13.

[7] L. Nebot-Andrés, M.G. Del Duca, C. Aprea, A. Žerovnik, J. Tušek, R. Llopis, A. Maiorino, Improving efficiency of transcritical CO2 cycles through a magnetic refrigeration subcooling system, Energ. Conver. Manage. 265 (2022).

[8] G. Vaccaro, A. Milazzo, L. Talluri, Thermodynamic assessment of trans-critical refrigeration systems utilizing CO2-based mixtures, Int. J. Refrig. 147 (2023) 61–70.

[9] D. Calleja-Anta, L. Nebot-Andres, R. Cabello, D. Sánchez, R. Llopis, A3 and A2 refrigerants: border determination and hunt for A2 low-GWP blends, Int. J. Refrig. 134 (2022) 86–94.

[10] O. Yoji, M. Akio, T. Koutaro, K. Shigeru, Analysis of Heat Pump Cycle Using CO2/DME Mixture Refrigerant, in: International Refrigeration and Air Conditioning Conference, Purdue University, 2008.

[11] F. Xiaowei, Z. Xianping, W. Fengkun, Simulation study on a heat pump system using R744/R290 as refrigerant, J. Civil Eng. Architect. 7 (2013).

[12] K. Kumar, P. Kumar, Analysis of Propane + CO2 mixture as a working fluid in a vapor compression refrigeration system, Refrig. Sci. Technol. 2019 (2019) 319–326.

[13] J. Xie, J. Wang, Y. Lyu, D. Wang, X. Peng, H. Liu, S. Xiang, Numerical investigation on thermodynamic performance of CO2-based mixed refrigerants applied in transcritical system, J. Therm. Anal. Calorim. 147 (2021) 6883–6892.

[14] D. Sánchez, F. Vidan-Falomir, R. Larrondo-Sancho, R. Llopis, R. Cabello, Alternative Co2-based blends for transcritical refrigeration systems, Int. J. Refrig. (2023).

[15] G. Vaccaro, A. Milazzo, L. Talluri, A proposal for a non-flammable, fluorine-free, CO2-based mixture as a low TEWI refrigerant, Int. J. Refrig. 158 (2024) 157–163.

[16] J.H. Kim, J.M. Cho, M.S. Kim, Cooling performance of several CO2/propane mixtures and glide matching with secondary heat transfer fluid, Int. J. Refrig. 31 (2008) 800–806.

[17] P. Tobaly, M.F. Terrier, P. Bouteiller, CO2+ propane mixture as working fluid for refrigeration in hot climates. Experimental results of energy efficiency tests, Refrig. Sci. Technol. 2018 (2018) 790–797.

[18] B. Yu, D. Wang, C. Liu, F. Jiang, J. Shi, J. Chen, Performance improvements evaluation of an automobile air conditioning system using CO2-propane mixture as a refrigerant, Int. J. Refrig. 88 (2018) 172–181.

[19] D. Sánchez, F. Vidan-Falomir, L. Nebot-Andrés, R. Llopis, R. Cabello, Alternative blends of CO2 for transcritical refrigeration systems. Experimental approach and energy analysis, Energy Convers. Managem. 279 (2023) 116690.

[20] Y. Chen, N.P. Halm, E.A. Groll, J.E. Braun, Mathematical modeling of scroll compressors—part I: compression process modeling, Int. J. Refrig. 25 (2002) 731–750.

[21] C. Cuevas, J. Lebrun, V. Lemort, E. Winandy, Characterization of a scroll compressor under extended operating conditions, Appl. Therm. Eng. 30 (2010) 605–615.

[22] C. Aprea, C. Renno, An experimental analysis of a thermodynamic model of a vapour compression refrigeration plant on varying the compressor speed, Int. J. Energy Res. 28 (2004) 537–549.

[23] V. Aute, C. Martin, R.J.C.P. Radermacher, MD: Air-Conditioning, Heating, R. Institute, A study of methods to represent compressor performance data over an operating envelope based on a finite set of test data (no. 8013), (2015).

[24] H. Wan, T. Cao, Y. Hwang, S.-D. Chang, Y.-J. Yoon, Machine-learning-based compressor models: a case study for variable refrigerant flow systems, Int. J. Refrig. 123 (2021) 23–33.

[25] R. Ossorio, E. Navarro-Peris, Testing of Variable-Speed Scroll Compressors and their inverters for the evaluation of compact energy consumption models, Appl. Therm. Eng. 230 (2023).

[26] DORIN, CO2 semi-hermetic compressors, cd series. CO2 transcritical Application, in: I. Officine Mario Dorin S.p.A. (Ed.), 2018.

[27] E. Sicco, M. Martínez-Ángeles, G. Toffoletti, L. Nebot-Andrés, D. Sánchez, R. Cabello, G. Cortella, R. Llopis, Experimental evaluation of CO2/R-152a mixtures in a refrigeration plant with and without IHX, Int. J. Refrig. 159 (2024) 371–384.

[28] M. Martínez-Ángeles, L. Nebot-Andrés, D. Calleja-Anta, R. Llopis, Experimental assessment of CO2/R-152a mixtures in a refrigeration plant with integrated mechanical subcooling, Int. J. Refrig. 158 (2024) 288–302.

[29] E.W. Lemmon, I.H. Bell, M.L. Huber, M.O. McLinden, NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 10.0, National Institute of Standards and Technology, 2018.

[30] AENOR, UNE-EN 12900. Refrigerant compressors. Rating conditions, tolerances and presentation of manufacturer's performance data, 2014.

[31] M. Hu, F. Xiao, H. Cheung, Identification of simplified energy performance models of variable-speed air conditioners using likelihood ratio test method, Sci. Technol. Built Environ. 26 (2019) 75–88.

[32] P. Bacher, H. Madsen, Identifying suitable models for the heat dynamics of buildings, Energ. Build. 43 (2011) 1511–1522.

[33] D. Sánchez, E. Torrella, R. Cabello, R. Llopis, Influence of the superheat associated to a semihermetic compressor of a transcritical CO2 refrigeration plant, Appl. Therm. Eng. 30 (2010) 302–309.