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Semi-empirical and environmental assessment of the low GWP refrigerant HCFO-1224yd(Z) to replace HFC-245fa in high temperature heat pumps[☆]

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ABSTRACT

This paper investigates a promising low-GWP refrigerant, HCFO-1224yd(Z), as an environmentally friendly replacement for HFC-245fa in high temperature heat pumps (HTHPs). First, the thermophysical properties of both fluids were analysed, and a single-stage cycle with Internal Heat Exchanger (IHX) was modelled. Then, a semi-empirical drop-in test replacement was assessed. The low-temperature reservoir was fixed at 80 °C, whereas the high-temperature reservoir was varied from 110 to 140 °C to cover a wide range of industrial applications. The theoretical and semi-empirical results illustrate that HCFO-1224yd(Z) heating capacity becomes around 8.9% lower than HFC-245fa. However, the higher suction density of HCFO-1224yd(Z) compared to the reference fluid can compensate for this effect, reducing the compressor power consumption. Hence, HCFO-1224yd(Z) presents a COP increase up to 4.5% compared to the reference fluid. At high-temperature reservoir of 140 °C, HCFO-1224yd(Z) shows a COP of 2.33, whereas HFC-245fa only reaches 2.23. The carbon footprint assessment illustrates a significant equivalent $CO_{2,eq}$ emissions reduction down to 90%, depending on the country (carbon emission factor). Therefore, HCFO-1224yd(Z) can be used as an alternative to HFC-245fa in HTHP systems due to its beneficial operating, energetic and environmental characteristics.

Évaluation semi-empirique et environnementale du frigorigène à faible PRP HCFO-1224yd(Z) en remplacement du HFC-245fa dans les pompes à chaleur à haute température

Mots clés: Hydrochlorofluorooléfine (HCFO); Fluides actifs; Pompes à chaleur à haute température; Récupération de chaleur perdue; Chauffage industriel; Décarbonation

1. Introduction

Heat pumps are systems based on thermodynamic principles that can revalorise heat using relatively small electric energy. Recent heat pump prototypes have efficiently extended the operating ranges of this technology becoming possible to produce heating at temperatures from 90 to 140 °C (Arpagaus and Bertsch, 2019; Mateu-Royo et al., 2019b). This technological improvement modified the classification of the heat pumps, becoming known as high temperature heat pumps (HTHPs) (Arpagaus et al., 2018). HTHPs represent a strategical tool to decarbonise different EU

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Nomenclature

ε	effectiveness
ρ	density (kg m^{-3})
h	enthalpy (kJ kg^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})
η	efficiency
$\dot{Q}_k$	heating capacity (kW)
$\dot{W}_c$	compressor power consumption (kW)
VHC	volumetric heating capacity (kJ m^{-3})
Subscripts	
in	inlet
is	isentropic
o	evaporating
out	outlet
ref	refrigerant
sink	high-temperature reservoir
source	low-temperature reservoir
suc	suction
vol	volumetric
Abbreviatures	
COP	coefficient of performance
GWP	global warming potential
HFO	hydrofluoroolefin
HTHP	high temperature heat pump
HFC	hydrofluorocarbon
HCFO	hydrochlorofluoroolefin
IHX	internal heat exchanger
ODP	ozone depletion potential

industries with heat consumption between 100 and 150 °C, covering around 21.5 TWh per year (Kosmadakis, 2019). Apart from the HTHPs potential because of the gap in these temperatures, they also present higher energy performance, flexibility in design, and they can be combined with other technologies in trigeneration systems (Urbanucci et al., 2019), or district heating networks (Mateu-Royo et al., 2020c), among other representative examples.

Previous experimental studies illustrate that HTHP prototypes are based on different technologies and working fluids in basic cycle configurations. Bamigbetan et al. (2019) used HC-600 in a basic cycle with an internal heat exchanger (IHX) and receiver to deliver heat at 115 °C. They use a modified piston compressor due to the operation at a high temperature, lubricated with synthetic oil and tested at frequencies from 30 to 50 Hz. The total and isentropic efficiencies of the compressor were 74% and 83%, respectively. Chamoun et al. (2014) proposed twin-screw as the compressor technology and water as the refrigerant. The system is also composed of a plate condenser, flash tank, electrical expansion valve, and falling film evaporator. The rotational speed of the compressor was 5000 rpm, whereas the compression ratio achieved was 4.5. Bobelin et al. (2012) obtained a supply heat up to 125 °C with a developmental mixture as the refrigerant, using an HTHP composed of a scroll compressor, brazed-plate evaporator, condenser, sub-cooler, and electronic expansion valve. The COP obtained was 4.48, for evaporating and condensing temperatures of 50 and 100 °C, respectively. Meroni et al. (2018) validated a centrifugal compressor model using air, HFC-134a, and carbon dioxide, developing a coupled system with a heat pump to supply steam at 150 °C.

HTHPs, mainly based on vapour compression technology, typically use hydrofluorocarbons (HFCs) as working fluids (refrigerants), which have high values of global warming potential (GWP) (Arpagaus et al., 2018). However, in 2014, the EU Regulation No. 517/2014 (European Commission and The European Parliament and the Council of the European Union, 2014) gradually limited the acquisition of HFCs, establishing market quotas to their consumption (put into the market). Given this situation, the commonly called fourth generation of refrigerants appeared to replace HFCs (Calm, 2008). HFC-245fa is considered the reference fluid for high temperature applications up to 140 °C. Therefore, several low GWP alternatives are available to replace HFC-245fa to promote the sustainable development of HTHPs systems (Mateu-Royo et al., 2019a).

A prominent candidate is HCFO-1224yd(Z), an unsaturated organic compound approved in 2017 by the American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE) (ANSI/ASHRAE, 2016) and proposed for HFC-245fa replacement in applications like centrifugal chiller, HTHPs, or ORCs, among others. The main advantages are its GWP value below the unity (AGC Chemicals, 2017), and it is considered a low-pressure refrigerant, as happens for HFC-245fa, and is classified as A1 refrigerant (low toxicity and non-flammable). Furthermore, the magnitude for some properties is comparable, resulting in similar thermodynamic performance (Eyerer et al., 2019; Navarro-Esbrí et al., 2020). Consequently, HCFO-1224yd(Z) could help spread HTHPs (Frate et al., 2019).

The interest of the heat pump industry in HCFO-1224yd(Z) is demonstrated through the rising yearly number of publications, once that its properties are being accurately determined (Fedele et al., 2020). HCFO-1224yd(Z) has been considered in HTHPs based on different configurations, such as cycles including ejectors (Bai et al., 2019; Mateu-Royo et al., 2020a) or two-stage cascades (Mota-Babiloni et al., 2018). However, up to this day, only Arpagaus and Bertsch (2019) experimentally investigated HCFO-1224yd(Z), illustrating the high potential of this refrigerant for future HTHP applications and retrofit systems.

Semi-empirical models enable accurate prediction of system operation and performance at varying conditions (Dechesne et al., 2019). It is a powerful tool for prototype design at an early stage of the development of any technology. At this time, there is a lack of HCFO-1224yd(Z) results in HTHPs and the environmental, energetic and safety benefits that could provide the introduction of this refrigerant in these systems. This paper presents a semi-empirical evaluation of the alternative low-GWP refrigerant HCFO-1224yd(Z) as a replacement for HFC-245fa in high temperature heat pumps through a thermodynamic assessment and semi-empirical comparison. A single-stage cycle with IHX is used as the test bench configuration, operating with a fixed low-temperature reservoir inlet of 80 °C. In contrast, the high-temperature reservoir outlet varies from 110 to 140 °C. The main parameters studied are heating capacity, compressor power consumption, volumetric heating capacity, and coefficient of performance (COP). Finally, expected theoretical results and the semi-empirical results are compared and discussed to comprehensively evaluate HCFO-1224yd(Z) potential as an alternative to HFC-245fa.

2. Overview of studied refrigerants

HCFO-1224yd(Z) is a non-flammable substance proposed to replace HFC-245fa with minor system modifications. Both refrigerants are pure fluids with slight differences in thermophysical properties. Thus, Table 1 shows the main thermodynamic and transport properties of both working fluids.

This alternative low-GWP refrigerant is classified as an A1 safety class: lower toxicity refrigerant with no flame propagation. Its GWP is more than a thousand times lower than HFC-245fa, reducing the direct greenhouse emissions when this alternative refrigerant is leaked. Whereas HFC-245fa has zero ODP, HCFO-1224yd(Z) contains chlorine in its molecule, showing an ODP of 0.00012. Nevertheless, the consequence of leakages on the atmo-

Table 1
Main thermodynamic and transport properties of HFC-245fa and HCFO-1224yd(Z).

Parameters	HFC-245fa	HCFO-1224yd(Z)
Molecular weight (g mol^{-1})	134.05	148.49
Critical temperature (°C)	153.86	155.54
Critical pressure (MPa)	3.65	3.34
Normal boiling point (NBP) (°C)	15.05	14.62
Latent heat of condensation[a] (kJ kg^{-1})	97.25	86.10
Suction density[b] (kg m^{-3})	25.45	27.11
ODP (CFC-11=1) (ASHRAE, 2017)	0	0.00012
$GWP_{100\text{-years}}$ (ASHRAE, 2017)	858	<1
ASHRAE safety class (ASHRAE, 2017)	B1	A1

[a] At a condensing temperature of 130 °C.
[b] At an evaporation temperature of 60 °C.

Table 2
Operating conditions in the experimental tests.

Controlled parameters	Values
High-temperature reservoir outlet	110–140 °C
Low-temperature reservoir inlet	80 °C
Superheating degree (SH) with IHX	35 K
Compressor drive frequency	50 Hz

spheric ozone atmosphere could be considered barely negligible (Patten and Wuebbles, 2010). The critical temperature of HCFO-1224yd(Z) is 1.7 °C higher than HFC-245fa. Therefore, the alternative refrigerant can reach even slightly higher heating production temperatures than the reference working fluid.

HCFO-1224yd(Z) presents a decrease of the latent heat of condensation (about 11.5%) compared to HFC-245fa, as shown in Table 2. This reduction illustrates that a heating capacity decrease is expected in a system with a constant compression volume. Nevertheless, a considerable mass flow rate increase can compensate for this phenomenon. However, the mass flow rate is highly influenced by suction density. In this case, HCFO-1224yd(Z) shows a slight increase of 6.5% of the suction density compared to HFC-245fa. Thus, a heating capacity decrease between 6 and 8% is expected with HTHP operating with HCFO-1224yd(Z) instead of HFC-245fa.

The heating capacity decrease can be compensated with a considerable reduction of the compressor power consumption, operating with HCFO-1224yd(Z). The suction density increase of the alternative refrigerant will produce a slight increment of the mass flow rate that can be compensated with lower compressor specific work. Additionally, based on the investigation of Mateu-Royo et al. (2019a), HCFO-1224yd(Z) and HFC-245fa are pure fluids with slightly similar heat transfer parameters based on the analysis of figures of merit.

Comparing both refrigerants in P-h and T-s diagrams illustrates that the slope of the isentropic line of HFC-245fa is higher than HCFO-1224yd(Z), as shown in Fig. 1. Hence, lower compressor specific work is required using HCFO-1224yd(Z). Therefore, the mass flow rate increase of HCFO-1224yd(Z) can be compensated, resulting in lower compressor power consumption. However, a comprehensive theoretical and semi-empirical analysis must provide a detailed performance comparison between both refrigerants.

3. Methodology

This section presents the methodology followed in this study where a preliminary theoretical study is carried out to illustrate the potential performance difference between the novel refrigerant HCFO-1224yd(Z) and the reference fluid HFC-245fa. Then, the experimental setup and acquisition procedure to obtain the experimental data of HFC-245fa is presented. Thus, experimental data is used to calculate the expected HCFO-1224yd(Z) performance results through an approach based on the thermophysical differences between both refrigerants, obtaining a semi-empirical evaluation. Finally, environmental analysis is realised using the TEWI metric.

A preliminary theoretical study is carried out to estimate the main parameters for both refrigerants analysed. The high-temperature reservoirs between 110 and 145 °C simulate an industrial process's thermal level, whereas the low-temperature reservoir is fixed at 80 °C. In this theoretical study, the following assumptions are established:

- Constant electro-mechanic efficiency of 0.75 is considered.
- Variable isentropic and volumetric efficiency are considered based on experimental data (Mateu-Royo et al., 2019b)
- Sub-cooling degree of 2 K is considered at the condenser outlet.
- Superheating degree of 5 K is considered at the evaporator outlet.
- Pitch point is considered 3 K at the condenser and between 2.5 to 11 K in the evaporator.
- ΔT between inlet and outlet of the secondary circuits is considered 15 K in both, condenser and evaporator.
- The IHX effectiveness varies between 0.1 and 0.5, depending on the outlet high-temperature reservoir.
- Isenthalpic process is considered at the expansion valve.
- Either pressure losses or heat losses in any components are neglected.

The refrigerant mass flow rate ($\dot{m}_{ref}$) is calculated as follows:

$$\dot{m}_{ref} = \rho_{suc} V_c \eta_{vol} \left(\frac{N}{60} \right) \tag{1}$$

The heating capacity ($\dot{Q}_k$) is defined as the product of the refrigerant mass flow rate and the refrigerating effect (enthalpy difference between evaporator outlet and inlet):

$$\dot{Q}_k = \dot{m}_{ref} \left(h_{k,\ in} - h_{k,out} \right) \tag{2}$$

The suction enthalpy is calculated with Eq. (3), using an experimental value of IHX effectiveness for each operating condition and the condenser and evaporator's outlet states. For an IHX with a fixed area, the effectiveness values change in the different conditions. Therefore, the values used in this study are based on the experimental data obtained in the previous test for HFC-245fa. In the theoretical analysis, the IHX effectiveness is assumed the same for both refrigerants. However, the semi-empirical analysis uses a thermodynamic approach of this value based on the thermophysical properties.

On the other hand, it is used an enthalpy difference instead of a temperature ratio because the specific heat value significantly increases where the point becomes close to the two-phase region. Thus, the temperature difference ratio becomes not a proper method to calculate the IHX effectiveness, whereas the enthalpy difference ratio provides more accurate results. This phenomenon is observed in several experimental tests, realising the energy balance and improving the theoretical calculations using enthalpy difference for the refrigerant side where some point can be close to the two-phase region. Then, the suction temperature is calculated with the evaporating pressure and the calculated suction enthalpy.

$$\varepsilon_{\mathrm{IHX}} = \frac{h_{suc} - h_{o,out}}{h_{k,out} - h_{o,out}} \tag{3}$$

The electric power consumption of the compressor ($\dot{W}_C$) is expressed in Eq. (4) as the product of the mass flow rate and the isentropic enthalpy difference at the compressor divided by the isentropic and electromechanical efficiencies.

$$\dot{W}_C = \frac{\dot{m} \Delta h_{is,c}}{\eta_{is}\, \eta_{em}} \tag{4}$$

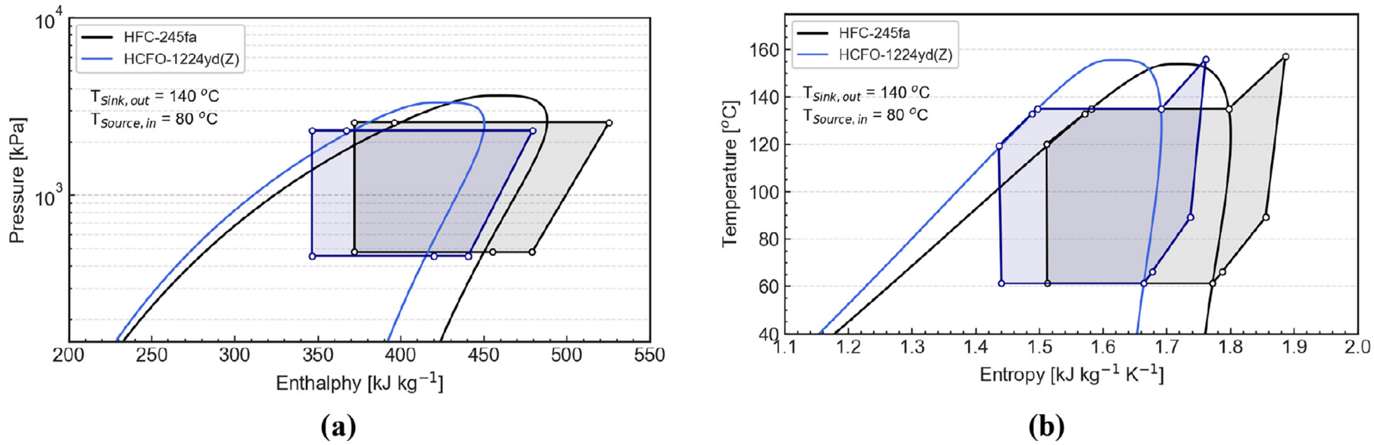

Fig. 1. Thermodynamic diagrams of a single-stage cycle with IHX, using HFC-245fa and HCFO-1224yd(Z) as working fluids: (a) Pressure-enthalpy, and (b) temperature-entropy.

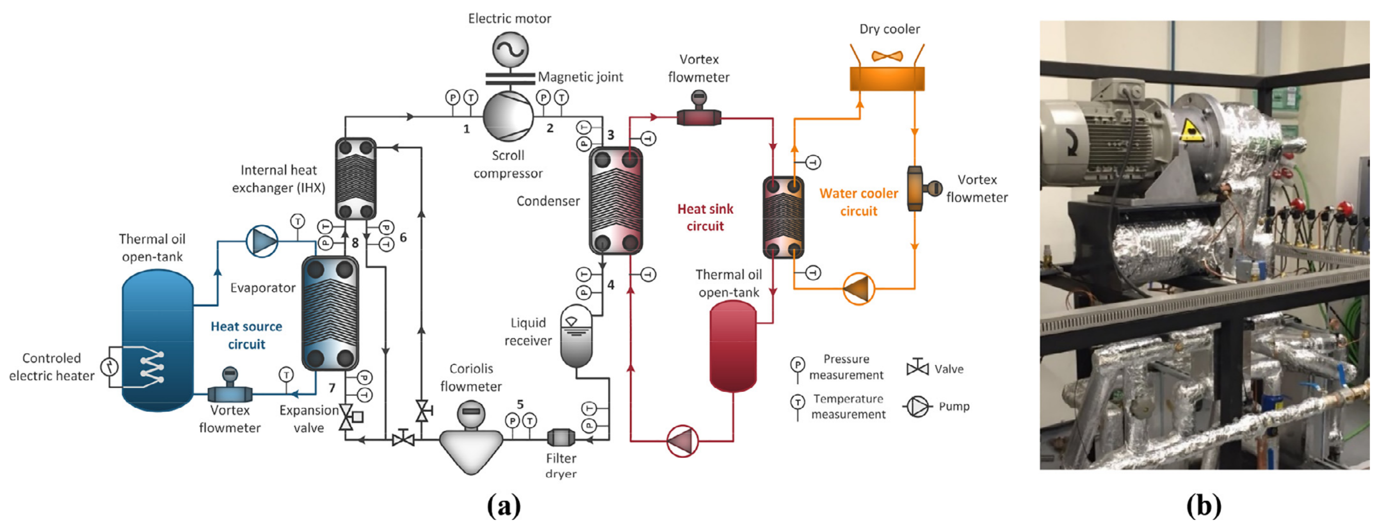

Fig. 2. HTHP experimental setup: (a) Schematic diagram, and (b) photo of the prototype.

The volumetric heating capacity (VHC) is calculated using Eq. (5) to compare the influence of the volumetric flow rate correctly at the compressor suction and the heating capacity.

$$VHC = \frac{\dot{Q}_k}{\dot{m}_{ref}} \eta_{vol} \rho_{suc} \tag{5}$$

The theoretical COP only depends on thermodynamic states at the inlet and outlet of the condenser and the compressor, defined as:

$$COP = \frac{\dot{Q}_k}{\dot{W}_C} \tag{6}$$

The experimental setup is a high temperature heat pump prototype with a heating capacity of 17.5 kW, as presented in Fig. 2. The vapour compression circuit consists of a magnetic coupling scroll compressor with a variable-speed electric motor of 7.5 kW nominal power. All the heat exchangers (condenser, evaporator and IHX) are brazed plate type. A liquid receiver between the condenser and sub-cooler is used to ensure the absence of bubbles before the expansion process. Due to the lack of electronic expansion valves for high-temperature applications, a hand expansion valve is installed to control the correct superheating degree at the evaporator's outlet. Polyester oil with a viscosity of 40 $mm^2\ s^{-1}$ (at 60 °C) is used to ensure the required compressor lubrication at high operating temperatures for both refrigerants. However, the oil behaviour between both refrigerants is not analysed in this study.

The secondary circuits are used to set the targeted evaporation and condensation temperatures. The high-temperature reservoir circuit uses thermal oil as the secondary fluid and is cooled by a closed-type cooling system that controls the thermal oil temperature. Moreover, the volumetric flow rate can be adjusted using a variable-speed pump. This cooling water system is composed of a Proportional Integrative Derivative (PID) controlled dry cooler to ensure the high-temperature reservoir circuit stability.

The waste heat simulation system (low-temperature reservoir circuit) also regulates the secondary thermal oil temperature through a set of immersed PID controlled electrical resistances. The test operating conditions for the experimental tests of the HTHP system are described in Table 2.

The refrigerant's thermodynamic states are calculated using pressure and temperature measurements in the required locations of the system. These values are measured with calibrated thermocouples and pressure transducers. The refrigerant mass flow rate is measured by a Coriolis mass flow meter located at the liquid line, while the power compressor consumption is measured using a digital wattmeter. The secondary circuits include temperature and volumetric flow rate sensors. Table 3 shows the characteristics of all sensors and their uncertainty. Finally, the thermodynamic prop-

Table 3
Summary of measurement sensors and uncertainties.

Measured parameters	Sensor	Uncertainty
Temperatures	J-type thermocouples	± 0.3 K
Pressures	Piezoelectric pressure transducers	±0.04% of reading
Refrigerant mass flow rate	Coriolis mass flow meter	±0.17% of reading
Low-temperature reservoir volumetric flow rate	Vortex flow meter	±0.5% of reading
High-temperature reservoir volumetric flow rate	Vortex flow meter	±0.028 m^3 h^{-1}
Compressor power consumption	Digital wattmeter	±1.55% of reading

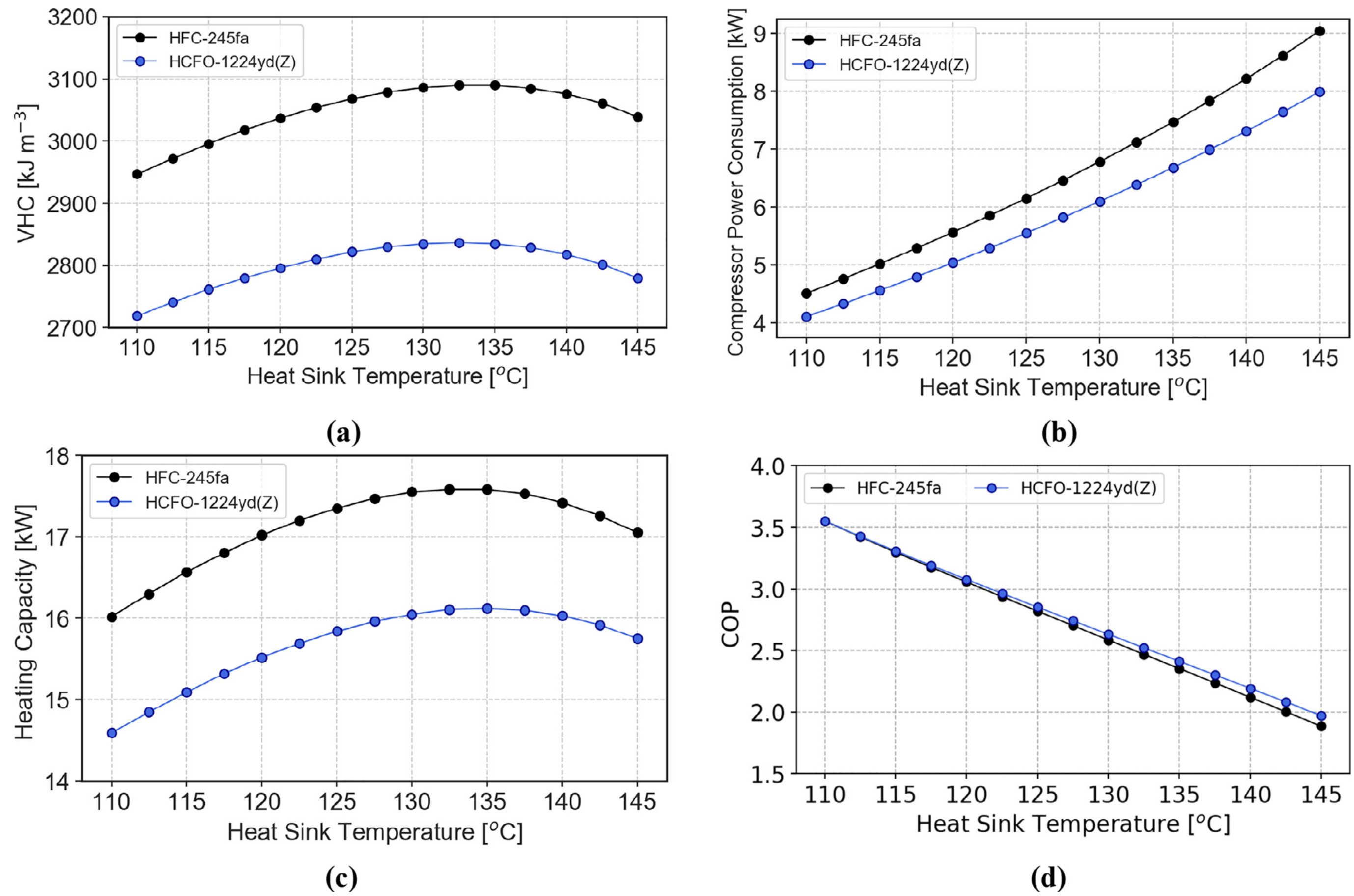

Fig. 3. Theoretical evaluation of single-stage cycle with IHX using HFC-245fa and HCFO-1224yd(Z) with a constant inlet high-temperature reservoir of 80 °C: (a) Volumetric heating capacity, (b) compressor power consumption, (c) heating capacity, and (d) COP.

erties are calculated using the REFPROP software (Lemmon et al., 2018).

The process for selecting a steady-state test consists of measuring for a time period of 40 min, with a sample period of 1 s. The condensing and evaporating pressure are within an interval of ± 3.5 kPa. During the test, all the temperatures are within a deviation of ± 0.25 K and the refrigerant mass flow rate is within ± 0.0007 kg s^{-1}. Then, once the steady-state operation is achieved (it involves 2400 direct measurement), the data is processed with an algorithm to find the steady-state period of 10 min (600 direct measurements) with the highest precision and accuracy.

Finally, the carbon footprint using HFC-245fa and HCFO-1224yd(Z) is compared using the TEWI metric, which has proved to be accurate enough to have an idea of the effects of refrigerant replacement in different applications (Mota-Babiloni et al., 2020). The TEWI metric is calculated as indicated in Eq. (7), being the two main components of the equation, the direct and the indirect $CO_{2,eq}$ emissions.

$$TEWI = GWP\ L\ n + GWP\ m_r\ (1-\alpha) + n\ E_a\ \beta \tag{7}$$

Some necessary assumptions can be considered to carry out every TEWI analysis. The lifespan of the system (n) is considered 15 years, and the recycling factor of the refrigerant (α) is assumed 0.7. It is considered a refrigerant charge (m_r) of 20 kg, based on the experimental field analysis. Finally, different values of the annual leakage rate and the indirect emission factor (β) are considered in order to provide a practical environmental evaluation of these parameters. High-temperature and low-temperature reservoirs considered are 140 °C and 80 °C, respectively.

4. Results and discussion

4.1. Theoretical analysis

Fig. 3 presents the theoretical comparison of the main performance parameters (volumetric heating capacity, compressor power consumption, heating capacity and COP) for the reference fluid HFC-245fa and its alternative low-GWP refrigerant HCFO-1224yd(Z). The volumetric heating capacity of HCFO-1224yd(Z) becomes lower than the reference working fluid HFC-245fa, pre-

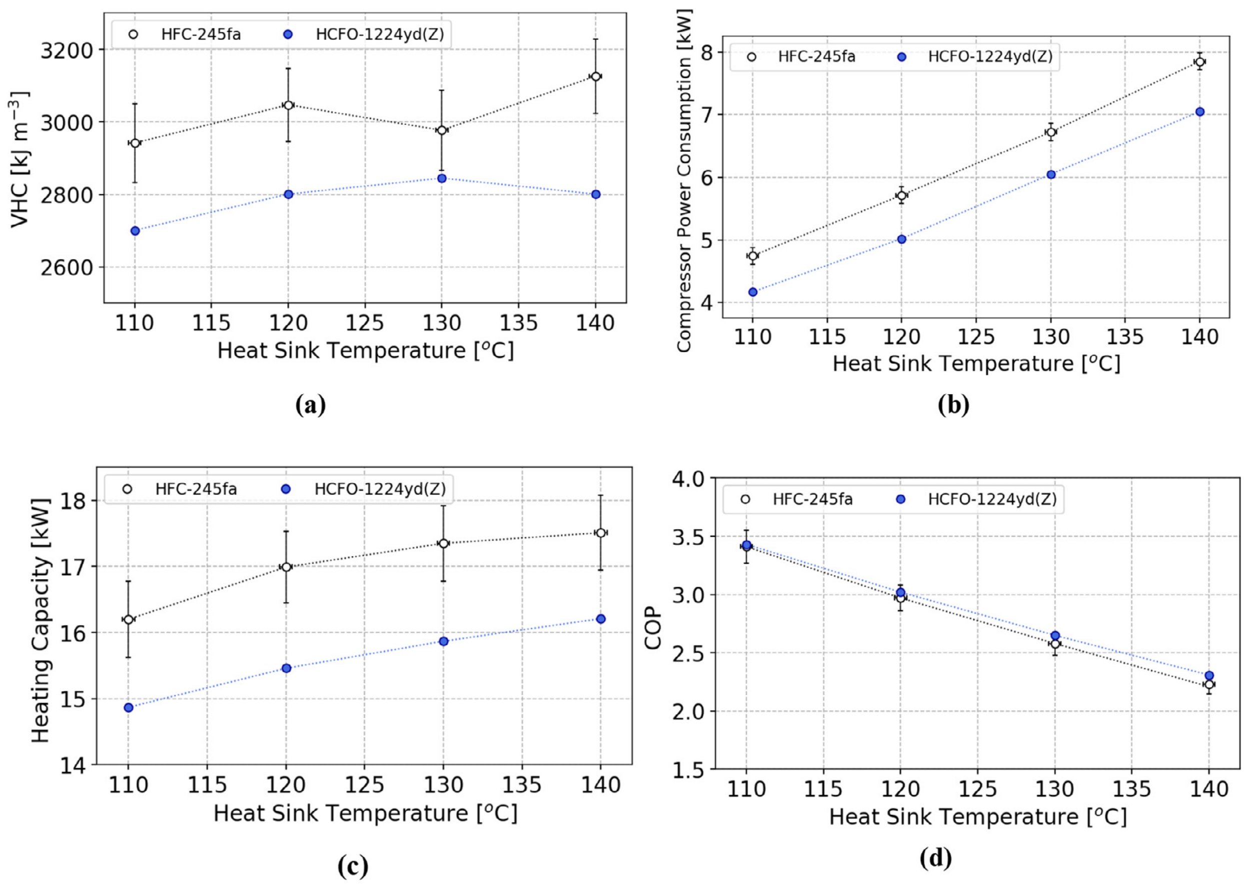

Fig. 4. Semi-empirical results of HFC-245fa and HCFO-1224yd(Z) with a constant inlet low-temperature reservoir temperature of 80 °C: (a) Heating capacity, (b) compressor power consumption, (c) volumetric heating capacity, and (d) COP.

cisely down to 8.52% less volumetric heating capacity, as shown in Fig. 3a. This phenomenon results from the heating capacity reduction that presents HCFO-1224yd(Z) compared with HFC-245fa. The lower latent heat of condensation that exhibits HCFO-1224yd(Z), compared with HFC-245fa, is not compensated by a mass flow rate increase, resulting in a lower heating capacity shown in Fig. 3c.

Nevertheless, HCFO-1224yd(Z) presents a decrease incompressor power consumption, down to 11.52% compared with HFC-245fa, illustrated in Fig. 3b. As the high-temperature reservoir increase, the difference in compressor power consumption increases. According to the previous refrigerants analysis, this reduction can be resulting in a lower specific work that shows HCFO-1224yd(Z) compared to HFC-245fa. Hence, the compressor power consumption drop can compensate for the heating capacity reduction, providing similar or even higher COP up to 4.75% when the HTHP operates with HCFO-1224yd(Z) instead of HFC-245fa.

4.2. Semi-empirical comparison

This section describes the semi-empirical results obtained with HCFO-1224yd(Z) based on experimental data from HFC-245fa, showing the main energy performance parameters, previously analysed in the theoretical evaluation: volumetric heating capacity, compressor power consumption, heating capacity, and COP (Fig. 4).

The semi-empirical calculations of the volumetric heating capacity present a similar trend to the expected theoretical evaluation, showing lower values for HCFO-1224yd(Z) than the reference HFC-245fa. About the compressor power consumption, Fig. 4b illustrates a considerable reduction, between 7% and 11% of HCFO-1224yd(Z) in comparison of HFC-245fa, showing similar trends to the expected theoretical evaluation. Although the mass flow rate value of HCFO-1224yd(Z) results higher than HFC-245fa, the isentropic lines slope difference produces a variation in each refrigerant's specific compressor work that becomes higher as the high-temperature reservoir increase.

HCFO-1224yd(Z) heating capacity suffers a reduction compared to HFC-245fa, specifically, between 7.2% and 8.9%, depending on the high-temperature reservoir. Operating at a high-temperature reservoir of 140 °C, HCFO-1224yd(Z) provides a heating capacity of 16.4 kW, whereas HFC-245fa presents 17.5 kW. Finally, the semi-empirical COP shows slightly similar behaviour than the expected theoretically, obtaining an increment up to 4.5% operating with HCFO-1224yd(Z) instead of HFC-245fa. At the maximum high-temperature reservoir of 140 °C, HFC-245fa ends with a COP of 2.23, whereas HCFO-1224yd(Z) presents a value of 2.33.

4.3. Carbon footprint comparison

As an application that has attracted attention recently, HTHP systems do not have previous guidelines published for carbon footprint analysis, and therefore, it is challenging to select concrete input values. Therefore, following that recommended by Mota-Babiloni et al. (2020), several carbon emission factors and leakage rates are simulated. Moreover, Fig. 5 contains relative results (TEWI reduction) between HCFO-1224yd(Z) and HFC-1245fa to minimise the effect of other input parameters. Emission factors of selected countries and the European Union have been placed as indicators of the magnitude of these values.

The use of the low-GWP refrigerant HCFO-1224yd(Z) instead of HFC-245fa provides a significant equivalent $CO_{2,eq}$ emissions re-

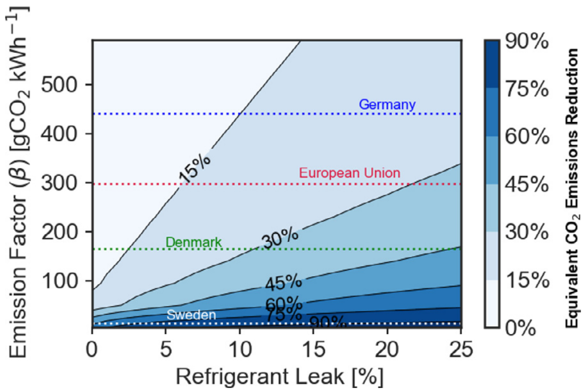

Fig. 5. TEWI reduction using HCFO-1224yd(Z) instead of HFC-245fa.

duction positive for climate change mitigation, as shown in Fig. 5. Considering an annual refrigerant leakage of 5% for HTHP in the machinery room, an equivalent $CO_{2,eq}$ emission reduction up to 15% is achieved, using the European Union emission factor. This reduction becomes higher for countries with lower emission factor. For instance, it can reduce 45% if the highest refrigerant leak is considered (Denmark). Therefore, low GWP refrigerants as HCFO-1224yd(Z) will improve these countries' decarbonisation. Hence, HCFO-1224yd(Z) becomes a potential sustainable refrigerant to replace HFC-245fa in HTHPs systems due to its low-GWP value and higher energy efficiency in high temperature applications.

5. Conclusions

This paper presents a theoretical and semi-empirical analysis of HCFO-1224yd(Z) as a drop-in replacement for HFC-245fa in a high temperature heat pump (HTHP) based on a single-stage cycle with IHX. The HFC-245fa tests have been performed varying the high-temperature reservoir from 110 to 140 °C with a constant low-temperature reservoir of 80 °C. Volumetric heating capacity, compressor power consumption, heating capacity and COP are used as performance indicators in the refrigerant evaluation. The main conclusions of this paper can be summarised as follows:

- HCFO-1224yd(Z) volumetric heating capacity is approximately 8.5% lower than HFC-245fa used as a drop-in in replacement in an HFC-245fa experimental setup. This difference is strongly related to heating capacity reduction, operating with the same volumetric flow rate.
- The heating capacity using HCFO-1224yd(Z) becomes about 8.9% lower than that obtained with HFC-245fa, which can be compensated with the 11.3% compressor power consumption decrease that presents HCFO-1224yd(Z) compared to HFC-245fa.
- The COP values obtained with HCFO-1224yd(Z) are comparable or even 4.5% higher than those obtained with HFC-245fa. At the maximum high-temperature reservoir of 140 °C, HCFO-1224yd(Z) provides a COP of 2.33, whereas HFC-245fa shows a value of 2.23.
- The environmental analysis using the TEWI metrics illustrates that HCFO-1224yd(Z) instead of HFC-245fa provide a significant equivalent CO_2 emissions reduction. Considering an annual refrigerant leakage of 5%, equivalent $CO_{2,eq}$ emissions reduction between 10 and 90% can be achieved, depending on the energy mix of each country.

Finally, from the semi-empirical results, it can be concluded that the energy performance parameters of HCFO-1224yd(Z) in a drop-in replacement are close to those obtained with HFC-245fa in high temperature heat pumps. Hence, HCFO-1224yd(Z) becomes a potential low-GWP alternative to replace the reference working fluid HFC-245fa.

The comparison between the theoretical and semi-empirical results confirms the latter analysis as a useful tool to evaluate the potential of recently developed refrigerants in existing systems. Semi-empirical assessment is recommended to complement the theoretical analysis and confirm a refrigerant potential in vapour compression systems before starting experimental campaigns at extreme conditions as those existing in HTHPs. Then, semi-empirical data can be used to validate experimental results by comparing both in a more accurate way than taking theoretical values as the reference.

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.

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