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Highlights:

- The energy performance and volumetric heating capacity of different vapour compression configurations are compared.
- HCFO-1233zd(E), HFO-1336mzz(Z), Butane and n-Pentane are considered as alternative working fluids for HFC-245fa.
- The alternatives suggested increase the energy performance in all conditions and configurations.
- The proper configuration selection is highly dependent on the temperature lift between the evaporation and condensing temperatures.
- n-Pentane achieves the highest COP of 3.85 for heating production up to 150 ºC.

Theoretical evaluation of different high-temperature heat pump configurations for low-grade waste heat recovery

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Abstract

The introduction of high-temperature heat pumps for waste heat recovery with low GWP refrigerants can reduce the greenhouse gas emissions in the industrial sector. This article evaluates the energy performance and the volumetric heating capacity of five vapour compression system configurations using n-Pentane, Butane, HCFO-1233zd(E) and HFO-1336mzz(Z) as HFC-245fa low GWP alternative fluids for heating production at temperatures of 110, 130 and 150 °C and different temperature lifts. The selected architectures and the equations are presented, and the most appropriate method to calculate the intermediate pressure is selected. The results of the simulation show that single-stage cycle with an internal heat exchanger (IHX) becomes the most efficient configuration at lower temperature lifts whereas two-stage cycle with IHX at higher lifts. While n-Pentane provides the highest energy performance values, Butane (only up to 130 °C) and HCFO-1233zd(E) highlight in the heating volumetric capacities. HFO-1336mzz(Z) provides intermediate values in both parameters. Consequently, the working fluid selection is highly dependent on the specifications and the energetic and installation costs.

Keywords: waste heat recovery; low GWP refrigerants; industrial heat pump; energy efficiency; greenhouse gases; climate change.

Nomenclature

$\dot{Q}$	thermal capacity (kW)
$\dot{V}$	volumetric flow rate ($m^3 \cdot s^{-1}$)
$\dot{W}$	electric power consumption (kW)
$\dot{m}$	refrigerant mass flow rate ($kg \cdot s^{-1}$)
COP	coefficient of performance
h	enthalpy ($kJ \cdot kg^{-1}$)
P	pressure (MPa)
q_v	volumetric heating capacity ($kJ \cdot m^{-3}$)
SC	sub-cooling degree (K)
SH	superheat degree (K)
T	temperature (ºC)
Greek symbols	
η	Efficiency (-)
ε	effectiveness (-)
ρ	density ($kg \cdot m^{-3}$)
Subscripts	

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c	compressor
em	electromechanical
HS	high stage
I	intermediate
in	inlet
is	isentropic
k	condenser
LS	low stage
o	evaporator
out	outlet
sec	secondary circuit
suc	suction
temp	temperature
vol	volumetric
Abbreviations	
CFC	Chlorofluorocarbon
GHG	Greenhouse gas
GWP	Global warming potential
HCFO	Hydrochlorofluoroolefin
HFC	Hydrofluorocarbon
HFO	Hydrofluoroolefin
HTHP	High-temperature heat pump
IHX	Internal heat exchanger
ODP	Ozone depletion potential
POE	Polyolester

1. Introduction

The mitigation of the climate change is one of the greatest challenges that humanity faces in this century. To address it, greenhouse gas emissions (GHG), which is the main cause of the global warming, need to be controlled. Therefore, the efforts to improve the energy efficiency has been intensified, especially in the industrial sector as being one of the three main energy consuming sectors (Brückner et al., 2015). Intergovernmental Panel on Climate Change considers the use of heat recovery technologies as a tool to improve the energy efficiency (Miró et al., 2015), decreasing the energy consumption of the industrial processes and therefore, the GHGs emissions.

Many industrial processes reject a significant amount of waste heat below 100 ºC and hence, a substantial quantity of waste heat energy remain unused due to economic and technical barriers (Forman et al., 2016). However, a higher range of temperature is required by some industrial processes, especially between 100 and 130 ºC (Chua et al., 2010; Seck et al., 2015). Thus, the attempts to introduce high-temperature heat pumps (HTHPs) have attracted significant attention for waste heat recovery in industrial processes (Langan and O'Toole, 2017). In addition, the integration of HTHPs as a heat source for the industrial processes leads to a reduction of the fossil fuels dependence, promoting a decarbonised and sustainable industrial sector.

At present, HFC-245fa, with a global warming potential (GWP) of 858, is widely used as a working fluid for HTHPs and Organic Rankine Cycles (Mounier et al., 2017). Nevertheless, the (European Commission, 2017) has adopted a proposal to ratify the Kigali amendment to the Montreal Protocol to gradually limit the HFCs production and consumption to mitigate the global warming. The use of renewable energy sources for heating becomes a possible solution for this challenge (Esen and Yuksel, 2013). Thus, the use of HTHPs for waste heat recovery using working fluids with low GWP becomes a solution to climate change mitigation by improving the industrial processes energy efficiency, using environmental friendly working fluids.

A few studies realised a fundamental performance analysis of different low GWP working fluids for HTHPs (Fukuda et al., 2014; Kondou and Koyama, 2015). Moreover, vapour compression systems can use different configurations alternatives to basic cycle to improve the energy performance (Molés et al., 2014; Mota-Babiloni et al., 2014). HTHPs can also use these variations to increase the temperature of the heat source to a higher and more useful temperature. These heat sources can come from the ground (Esen et al., 2006) or from the combination of ground source with solar energy (Esen et al., 2017; Esen, 2000). The use of ground as a heat source requires a detailed study of the temperature distribution (Balbay and Esen, 2013) and especial attention in cold climates (Balbay and Esen, 2010). However, waste heat can used as a proper heat source for HTHP. Hence, Cao et al. (2014) studied different heat pump systems configurations for using waste heat recovery with an average temperature of 45 ºC as a heat source and heating production temperature up to 95 ºC. Similar system configurations than Cao et al. (2014) are analysed by Antonijević et al., (2012) for groundwater HTHPs. Nevertheless, studies that analyse HTHP configurations with low GWP working fluids at different heating production temperatures are not found in the literature.

Therefore, the aim of this article is to compare theoretically the energy performance and the volumetric heating capacity of five vapour compression system configurations using n-Pentane, Butane, HCFO-1233zd(E) and HFO-1336mzz(Z) as low GWP alternative fluids to HFC-245fa for heating production temperatures of 110 and 130 ºC. Moreover, this article evaluates the proper low GWP working fluid and system configuration for heating production temperature up to 150 ºC, which is a situation not reachable by the traditional refrigerant HFC-245fa.

2. Refrigerants under consideration

Despite hydrofluorocarbons (HFCs) have been commonly used as refrigerants for HTHPs, the European Commission is gradually limiting the production and use of these chemicals. Hence, hydrofluoroolefins (HFO), hydrochlorofluoroolefin (HCFO) and natural refrigerants become potential alternatives. In addition, all the potential replacements must have comparable or higher critical temperature than HFC-245fa, as is the parameter that defines the possibility to achieve high heating production temperatures in a subcritical cycle.

HCFO-1233zd(E) and HFO-1336mzz(Z) are synthetic fluids widely used for organic Rankine cycle applications (Molés et al., 2016; Navarro-Esbrí et al., 2017) and, moreover, those refrigerants are being considered in HTHP applications (Juhasz, 2017). Both fluids are considered non-flammable and non-toxic. Butane (R-600) is an commonly known hydrocarbon limited by its critical temperature to HTHP applications with heating production temperatures below 130 ºC (Moisi and Rieberer, 2017; Pan et al., 2011), whereas n-Pentane (R-601), is only considered by Yamazaki and Kubo (1985) and Jakobs et al. (2010). However, the higher critical temperature of n-Pentane compared with the other refrigerants makes it an interesting candidate. Both hydrocarbons are flammable but non-toxic refrigerants.

Table 1 provides a general overview of several relevant characteristics of these HTHP refrigerants. As it can be seen, the molar mass of the hydrocarbons is much lower than that of synthetic fluids (HFC, HFO and HCFO). From this initial comparison, the molecular weight difference can be highlighted, because of the influence of the density that can affect the HTHPs components design.

Table 1: Characteristics of refrigerant selected (Klein, 2006)

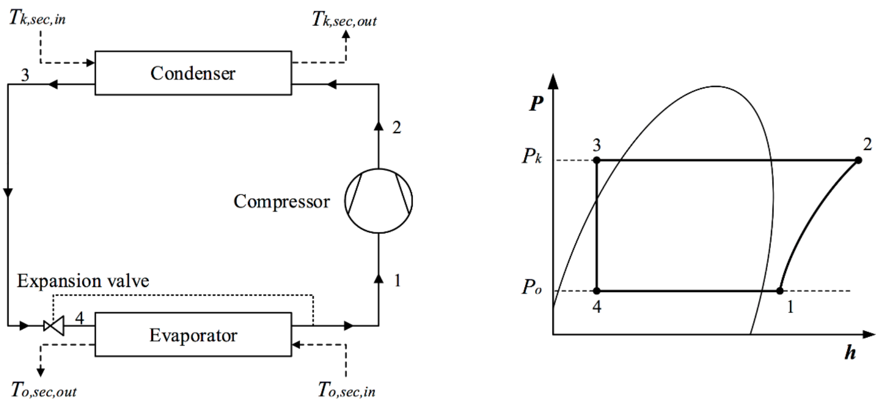
	ASHRAE Standards 34	Molar mass [g·mol	P_{crit} [MPa]	Normal boiling point	T_{crit} [ºC]	ODP (ASHRAE, 2017)	100-yr GWP (ASHRAE,

	(ASHRAE, 2017)	-1]		[ºC]			2017)
HFC-245fa	B1	134.05	3.65	15.18	154.01	0	858
HCFO-1233zd(E)	A1	130.50	3.77	18.32	165.60	0	1
HFO-1336mzz(Z)	A1	164.06	2.90	33.47	171.27	0	2
Butane	A3	58.12	3.80	-0.52	151.98	0	20
n-Pentane	A3	72.15	3.37	35.87	196.54	0	20

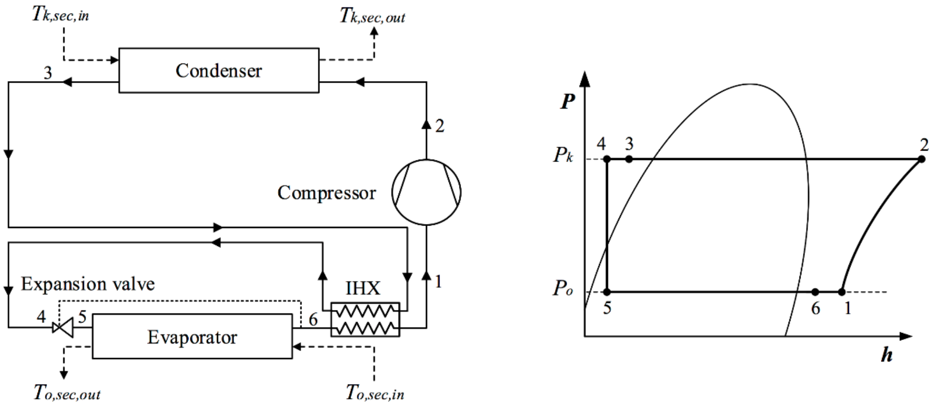
3. System configurations

In this paper, five vapor-compression architectures are considered to provide an adapted solution to provide useful heat from different grades of industrial waste heat: single-stage cycle (SS), single-stage cycle with internal heat exchanger (S-S IHX), two-stage cycle (T-S), two-stage cycle with IHX (T-S IHX) and two-stage cycle with intermediate-IHX (T-S I-IHX).

The first system configuration presented is a single-stage cycle, Fig. 1a. This basic configuration is composed of the following elements: compressor, condenser, expansion valve and evaporator.
This cycle is used as a reference for the system configuration analysis. Then, adding a heat exchanger between the suction and the liquid line to this basic configuration the single-stage cycle with IHX configuration is achieved Fig. 1b. The IHX reduces the liquid temperature before the expansion valve (sub-cooling) by means of heating the suction vapour (superheat). In contrast to refrigeration systems (Molés et al., 2014), the interest of IHX for HTHP lies more in heating this suction gas to increase the gas discharge temperature instead of increasing the sub-cooling degree of the system. Higher discharge temperature increases the performance of heat pump applications, considering the components temperature limits.



(a)

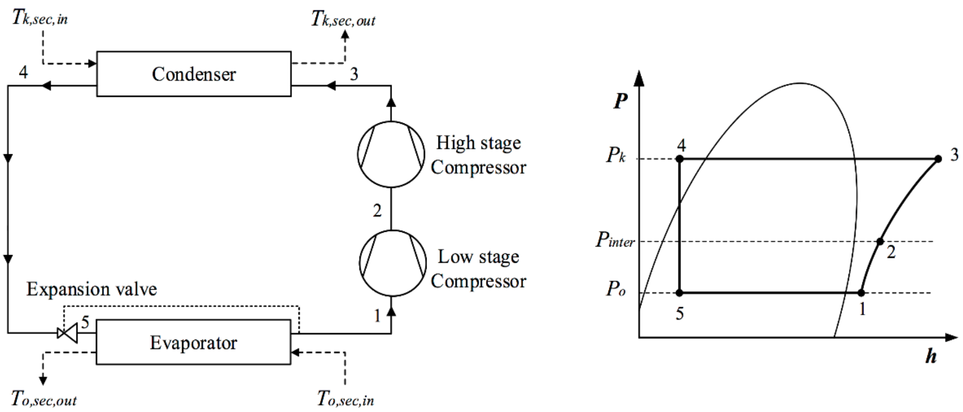


(b)

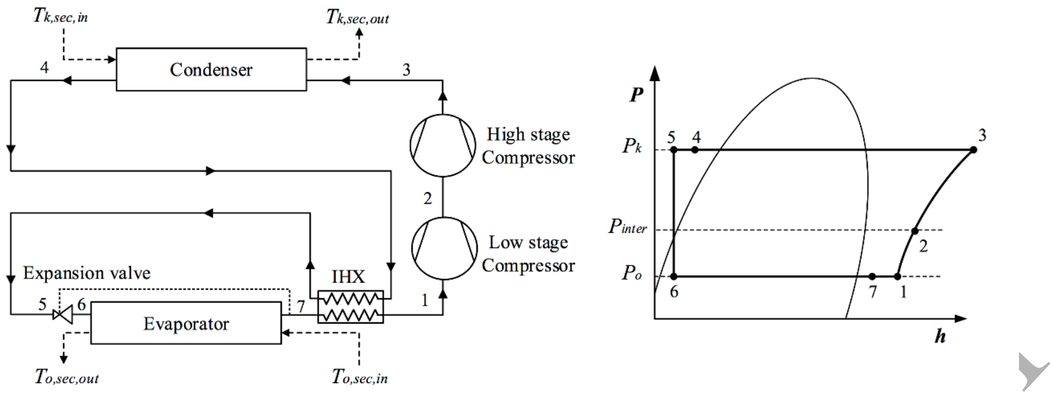
Fig. 1- Schematic and P-h diagram of single-stages configurations: a) single-stage cycle and b) single-stage cycle with IHX.

Two-stage cycles can be obtained by the division of the compression stage in two stages. The introduction of two-stage cycles, Fig. 2a, reports an increase in the cycle performance through a reduction of the pressure ratio of compression stages. In refrigeration applications, this system configuration has an inter-cooler or liquid injection between the compressors with the purpose of decrease the gas discharge temperature (Nemati et al., 2017). Nevertheless, in heating applications, it is interesting to increase the discharge temperature to enhance the useful heat transfer to the secondary fluid ($T_{k,sec,out}$). Therefore, two modifications of the initial two-stage configuration are proposed for the analysis of the HTHPs performance. IHX can be added between the evaporator and the low stage (LS) compressor and hence two-stage cycle with IHX configuration is obtained, Fig. 2b. This configuration, similar to single-stage cycle with IHX, increases the superheat before the LS compressor at the expense of cooling the liquid before the expansion valve. It produces an increase of the gas discharge temperature of the LS compressor and consequently, an increase of the gas discharge pressure of the high stage (HS) compressor.

Finally, the IHX could be located between the LS compressor and the HS compressor, giving rise to the last configuration two-stage cycle with intermediate-IHX. In contrast to single-stage cycle with IHX, this configuration increases the vapour suction temperature of the HS compressor instead of the LS compressor, as shown in Fig. 2c. Hence, the gas discharge temperature of LS compressor remains similar to the two-stage cycle configuration, but the gas discharge temperature of the HS compressor increases. It can result interesting for the analysis because HS gas discharge temperature is a parameter relevant to the useful heat production. Nevertheless, a comprehensive analysis is required to find the proper system configuration and the appropriate working fluid.



(a)



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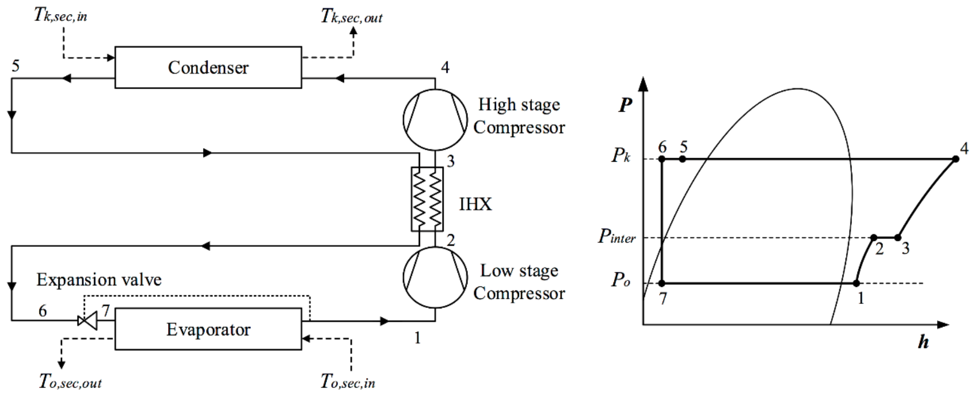


(c)

Fig. 2- Schematic and P-h diagram of two-stages configurations: a) two-stage cycle, b) two-stage cycle with IHX and c) two-stage cycle with intermediate-IHX

4. Operating conditions and model description

The heat absorbed by the evaporator ($\dot{Q}_o$) is an input of the model and it is considered constant for all the operating conditions. Considering the high-temperature applications of this performance analysis and the low-grade waste heat sources, two operating parameters are combined to simulate the typical working conditions: outlet condenser temperature of the secondary fluid or heating production temperature ($T_{k,sec,out}$) and inlet evaporator temperature of the secondary fluid or waste heat temperature ($T_{o,sec,in}$).

Firstly, heating production temperatures of 110 and 130 ºC are proposed to select an alternative working fluid and proper configuration to HFC-245fa in HTHP applications. Additionally, heating production temperature of 150 ºC is considered due to the industrial development interest of higher temperatures in heat pump applications. The sub-cooling degree is assumed to be 10 K and the temperature difference in the condenser of the secondary fluid between the inlet ($T_{k,sec,in}$) and the outlet ($T_{k,sec,out}$) is considered to be 15 K.

Secondly, inlet evaporator secondary fluid temperatures of 50, 70 and 90 ºC are considered in the simulations due to the interest of low-grade waste heat recovery. The superheat degree is considered to be 15 K and the temperature difference in the evaporator of the secondary fluid between the inlet ($T_{o,sec,in}$) and the outlet ($T_{o,sec,out}$) is assumed to be 10 K. For both condenser and evaporator, the pitch point is assumed to be 5 K.

The refrigerant thermodynamic properties have been evaluated using software Engineering Equation Solver (EES) (Klein, 2006). Moreover, isenthalpic process is considered at the expansion valve and heat transfer to the surroundings and pressure drops are neglected.

Isentropic and volumetric efficiency of the compressors are calculated using the Pierre's correlations for "good" reciprocating compressors (Granryd et al., 1999). In this way, volumetric efficiency, η_{vol}, is obtained from Eq. (1):

$$\eta_{vol} = k_1 \cdot \left(1 + k_s \cdot \frac{t_{2k} - 18}{100}\right) \cdot \exp(k_2 \cdot \frac{p_1}{p_2}) \tag{1}$$

where t_{2k} is the inlet temperature to the compressor and p_1/p_2 is the pressure ratio. The remaining symbols k_1, k_s and k_2 are constants and the values of these are 1.04, 0.15, and -0.07, respectively.
Following the Pierre's correlations, isentropic efficiency, η_{is}, is calculated with the Eq. (2) and the volumetric efficiency calculated with the Eq. (1).

$$(\frac{\eta_{vol}}{\eta_{is}}) = \left(1 + k_e \cdot \frac{t_{2k} - 18}{100}\right) \cdot \exp(a \cdot \frac{T_1}{T_2} + b) \tag{2}$$

where T_1/T_2 is the ratio of the absolute temperatures (in kelvin) of the condensation and evaporation corresponding to the discharge and the suction compressor pressures. The constants values considered are those given by Pierre for CFC-12, and therefore, k_e, a and b are -0.1, -2.40 and 2.88, respectively. This constant values are taken from the literature (Granryd et al., 1999). The electromechanical efficiency, η_{em}, is assumed to be 0.95.

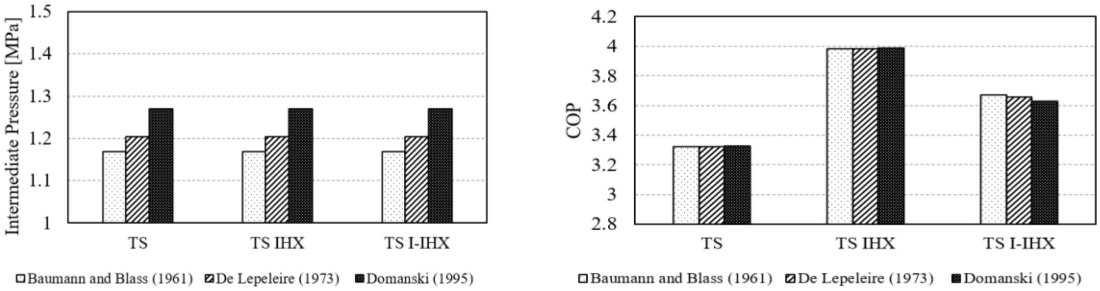
In two-stage system configurations, it is necessary to select the appropriate intermediate pressure to achieve competitive system efficiencies. Thus, the equations proposed by Baumann and Blass (1961) for theoretical cycles and assuming vapour as a perfect gas, Eq. (3), De Lepeleire (1973), Eq. (4), and Domanski (1995), Eq. (5), has been analysed for HTHP applications (Purohit et al., 2016).

$$P_I = \sqrt{P_k P_o} \tag{3}$$

$$P_I = \sqrt{P_k P_o} + 0.35\ [bar] \tag{4}$$

$$\theta = \frac{T_I - T_o}{T_k - T_o} \approx 0.5 \tag{5}$$

Fig. 3a shows that the resulting intermediate pressures are different depending on the configuration analysed and the equation used. On the other hand, the system performance remains almost constant in TS and TS IHX configurations whereas for the TS I-IHX configuration the equation proposed by Baumann and Blass (1961) achieves higher efficiencies than the others, as shown in Fig. 3b. Consequently, Eq. (3) is selected as an equation to calculate the intermediate pressure in all two-stage system configurations.



(a) (b)

Fig. 3- Intermediate pressure and COP comparison analysis for different two-stage system configurations.

The refrigerant mass flow rate for all system configurations is calculated using Eq. (6).

$$\dot{m} = \frac{\dot{Q}_o}{(h_{o,out} - h_{o,in})} \tag{6}$$

The required volumetric flow rate at the compressor suction, $\dot{V}_{suc}$, is obtained from the mass flow rate, the suction density and the volumetric efficiency, Eq. (7).

$$\dot{V}_{suc} = \frac{\dot{m}}{\eta_{vol}\,\rho_{suc}} \tag{7}$$

The IHX effectiveness (ε_{IHX}) is limited to the maximum discharge temperature T_{disc} at 180 ºC. Degradation of HFO fluids and POE lubricants can be observed above 200 ºC (Kontomaris, 2012; Navarro-Esbrí et al., 2015). The discharge temperature is dependent on the enthalpy after the compressor, $h_{c,out}$, and the condenser pressure, P_K. The vapour suction temperature T_{suc} is calculated using the Eq. (8).

$$\varepsilon_{IHX} = \frac{T_{suc} - T_{o,out}}{T_{k,out} - T_{o,out}} \tag{8}$$

The electric power consumption for the single-stage compressor, $\dot{W}_C$, is expressed in Eq. (9) as the product of the mass flow rate and the isentropic enthalpy increase at the compressor divided by the isentropic and electromechanical efficiencies.

$$\dot{W}_C = \frac{\dot{m}\,\Delta h_{is,c}}{\eta_{is}\,\eta_{em}} \tag{9}$$

For the two-stage compressors, the electric power consumption is calculated using the Eq. (10).

$$\dot{W}_C = \frac{\dot{m}}{\eta_{em}}\left(\frac{\Delta \dot{h}_{is,c,HS}}{\eta_{is,HS}} + \frac{\Delta h_{is,c,LS}}{\eta_{is,LS}}\right) \tag{10}$$

The heating capacity, $\dot{Q}_k$, is calculated using the product of the mass flow rate and the enthalpy difference between the inlet and outlet in the condenser, Eq. (11).

$$\dot{Q}_k = \dot{m}\,(h_{k,in} - h_{k,out}) \tag{11}$$

To properly compare the influence of the volumetric flow rate at the compressor suction and the heating capacity, volumetric heating capacity (q_v) is calculate using Eq. (12).

$$q_v = \frac{\dot{Q}_k}{\dot{V}_{suc}} \tag{12}$$

Finally, the Coefficient of Performance, COP, is calculated from the heating capacity and the compressor electric power consumption, using Eq. (13).

$$COP = \frac{\dot{Q}_k}{\dot{W}_C} \tag{13}$$

5. Results and discussion

The main results of the simulations are obtained considering five configurations and five working fluids at different heating production temperature ($T_{k,sec,out}$) and different waste heat temperatures ($T_{o,sec,in}$). As exposed in Section 2, refrigerants HFO-1336mzz(Z), HCFO-1233zd(E), Butane and n-Pentane have been selected as low GWP alternatives for HFC-245fa in high-temperature applications with $T_{k,sec,out}$ below 130 ºC. Then, for applications with $T_{k,sec,out}$ of 150 ºC, HFC-245fa and Butane have been discarded by the limitation of their critical temperature, and HFO-1336mzz(Z), HCFO-1233zd(E) and n-Pentane have been analysed.

The parameters chosen in this study to evaluate the appropriate configuration and working fluid is the COP and the volumetric heating capacity. COP is commonly used to evaluate the energy performance of the system and the volumetric heating capacity is used to compare the relationship between the heating capacity and compressor and installation size.

5.1 Comparison of low GWP alternatives to HFC-245fa in moderate-high temperature applications

For moderate and high-temperature applications, $T_{k,sec,out}$ of 110 and 130 ºC respectively, the results for alternative low GWP working fluids and configurations are shown as a relative difference (%COP and $\%q_v$) between each pair refrigerant-configuration taken and HFC-245fa single-stage configuration, as indicated Eq. (15) and Eq. (16).

$$\%COP = \left(\frac{COP_{conf,alt.fluid} - COP_{SS,HFC-245fa}}{COP_{SS,HFC-245fa}}\right) \tag{15}$$

$$\%q_v = \left(\frac{q_{v\,conf,alt.fluid} - q_{v\,SS,HFC-245fa}}{q_{v\,SS,HFC-245fa}}\right) \tag{16}$$

Fig. 4 presents the results for %COP at three inlet secondary fluid temperatures on the heat source, two outlet secondary fluid temperatures on the heat sink (110 and 130 ºC), five different configurations and five refrigerants. The difference between the evaporation and condensing temperature is known as temperature lift (Eq. 17), which is commonly used to compare the heat pump system performance. Hence, all simulation results are grouped into four different temperature lifts in order to select and analyse the proper configuration depending on each parameter.

$$Temperature\ lift = T_{condensation} - T_{evaporation} \tag{17}$$

On the one hand, for lift temperatures of 40 and 60 K, single-stage cycle IHX has higher performance (COP) than the other configurations. At lower temperature lifts, two-stage cycles have smaller pressure ratios than single-stage cycles, causing a decrease of the combined isentropic compressor efficiency. It increases the electric power consumption, and therefore, decrements the COP until the pressure ratio reaches higher values.

On the other hand, for lift temperatures of 80 and 100 K, two-stage cycle IHX has higher performance than the other configurations, as shown in Fig. 4. Two-stage cycle IHX achieves higher discharge temperature, heating capacity and isentropic compressor efficiency than other configurations. Therefore, higher the temperature lift, higher the performance improvement difference between two-stage cycle IHX and the other configurations.

Besides, two-stage cycle IHX and two-stage cycle intermediate IHX have similar isentropic HS compressor efficiency, but the position of the IHX produces that two-stage cycle IHX has higher isentropic LS compressor efficiency than the other configuration. Thus, two-stage cycle IHX has lower overall electric power consumption and higher heating capacity, achieving the maximum COP.

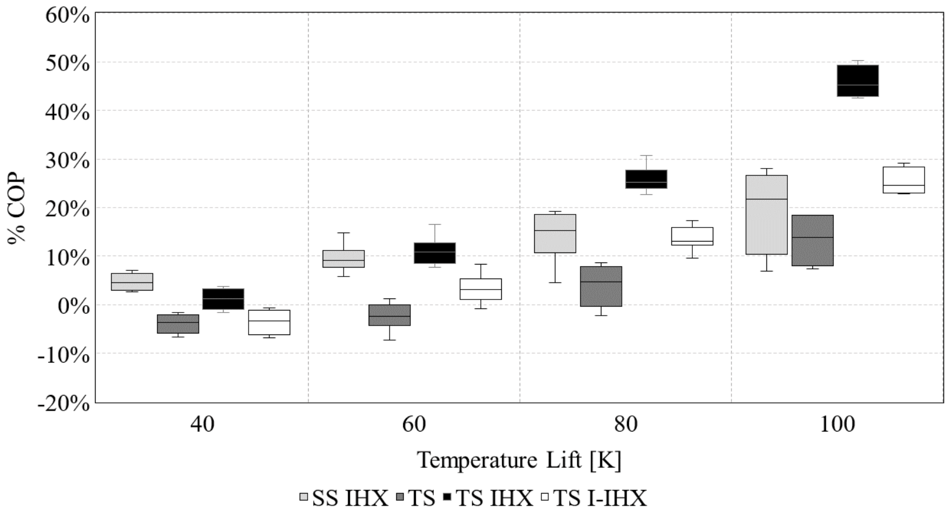
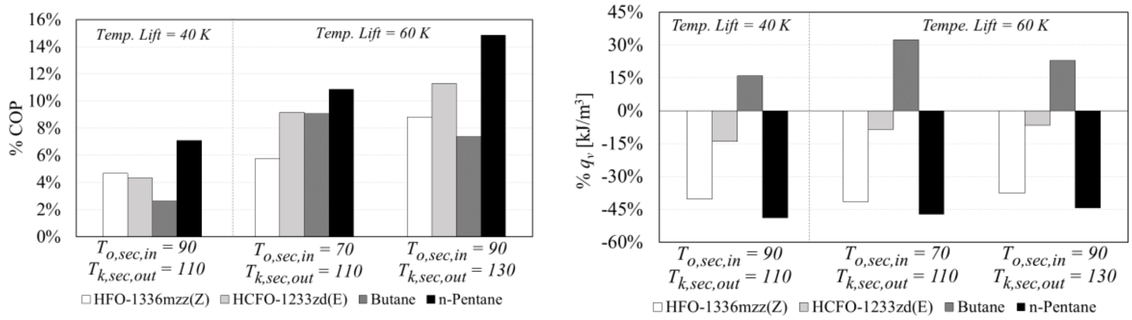


Fig. 4-Simulations results of %COP compared with single-stage cycle using HFC-245fa for different temperature lifts.

Attending to the COP results, single-stage with IHX is selected for analysis of the alternative working fluid in applications with temperature lifts of 40 and 60 K, whereas two-stage cycle IHX is selected for applications with temperature lifts of 80 and 100 K. Similar results are presented by (Antonijević et al., 2012; Cao et al., 2014). These studies concluded that two-stage configuration with modifications is the proper system solution for high temperature lifts. Nevertheless, none study has been found for proper architecture of high-temperature heat pumps operating at low temperature lift.

Fig. 5a presents the performance comparison of the different working fluids for a single-stage cycle with IHX configuration with temperature lifts of 40 and 60 K. All the alternative refrigerants perform better than HFC-245fa, given the positive differences. For the temperature lift of 40 K, higher %COP values are obtained by n-Pentane, with a performance increment of 7% compared to HFC-245fa single-stage cycle. Then, HFO-1336mzz(Z) and HCFO-1233zd(E) show an improvement of 5% and 4%, respectively. As the temperature lift increases, the %COP increases in the same way and for the temperature lift of 60 K, n-Pentane obtain a COP increase of 13%-15% whereas HCFO-1233zd(E) achieves an improvement of 9%-11%, being the HFO-1336mzz(Z) lower than those. The lower critical temperature of Butane causes that at higher evaporation temperatures it has worse improvements than the other refrigerants.

Attending to the q_v, Fig. 5b shows a reduction of 45% and 35% when n-Pentane and HFO-1336mzz(Z) are used instead of HFC-245fa, whereas the reduction of the volumetric heating capacity using HCFO-1233zd(E) is between 5% and 15%. Nevertheless, Butane is the unique refrigerant analysed that increases the volumetric heating capacity between 15-30%, therefore, lower compressor displacement and installations size is required than using HFC-245fa.

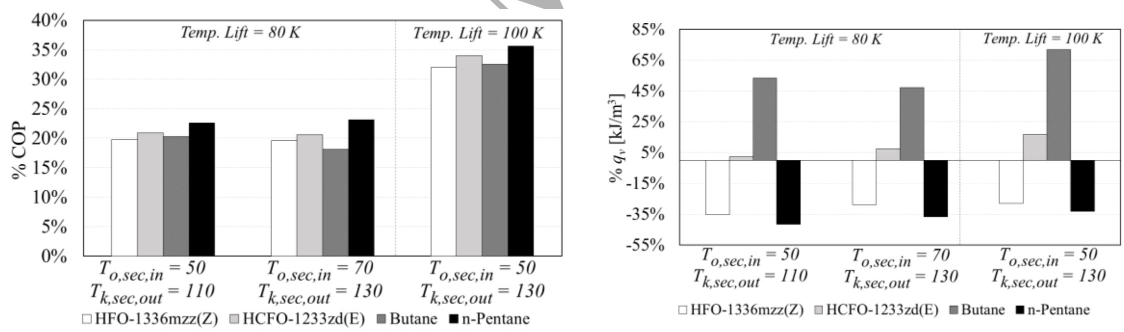


(a) (b)

Fig. 5- Simulation relative results of COP and volumetric heating capacity for temperature lifts of 40 and 60 K.

Fig. 6a illustrates the COP variation for temperature lift of 80 and 100 K in two-stage cycle with IHX. A higher temperature lift will increase the COP difference compared with the reference system. The %COP for all the analysed refrigerants is at a similar level but again, the highest COP is obtained by n-Pentane, being 23-35% than that of HFC-245fa single-stage cycle.

However, Fig. 6b shows a great variation of the volumetric heating capacity between the different working fluids simulated. While n-Pentane and HFO-1336mzz(Z) has a reduction of the volumetric heating capacity, Butane and HCFO-1233zd(E) presents an increment. Concretely, 50-70% for Butane and 2-15% for HCFO-1233zd(E). As a result, with similar variations of COP and providing the same heating capacity, HCFO-1233zd(E) requires a slightly similar compressor and installation size whereas Butane requires a lower compressor and installation size comparing with HFC-245fa.



(a) (b)

Fig. 6- Simulation relative results of COP and volumetric heating capacity for temperature lifts of 80 and 100 K.

It can be concluded that HCFO-1233zd(E) presents promising results in all the simulations carried out, achieving remarkable increments of COP with a slight variation of the volumetric heating capacity compared to the reference HFC-245fa (positive for higher temperature lifts and negative for lower). Although the highest increment of COP is achieved by n-Pentane, this working fluid obtained the most significant reduction of the volumetric heating capacity in all the cases and would require a significantly higher investment cost than HFC-245fa (in addition to the already needed because of the security measures). Butane has the largest increment of volumetric heating capacity, reducing the installation cost and proving higher heating capacity than HFC-245fa. Moreover, Butane has an increment of COP in all the simulations realised, nonetheless, when the heat source temperature is bringing closer to its critical temperature, this increment of COP is reduced, as it can be seen in Fig. 5a. Hence, Butane resulted in a proper solution if the heat source temperature does not come close to its critical temperature. Finally, HFO-1336mzz(Z) has a general increment of COP, most of the cases lower than the other fluids, and in addition, it has a highly decrement of the volumetric heating capacity. Thus, HFO-1336mzz(Z) is not a proper solution when the production temperature is below 130 ºC.

5.2 Analysis for high-temperature application solutions

This section presents the results for the heating production temperature of 150 ºC and heat source temperatures of 50, 70 and 90 ºC. HFC-245fa and Butane have a critical temperature below that value, therefore, these refrigerants are not considered for these conditions. Therefore, the results cannot be compared with HFC-245fa and are shown as HFO-1336mzz(Z), HCFO-1233zd(E) and n-Pentane absolute values of COP and q_v, Fig. 7 and Fig. 8.

With a heat source temperature of 90 ºC, Fig. 7 shows that n-Pentane and HFO-1336mzz(Z) obtain a COP of 3.85 and 3.59, respectively. Nevertheless, Fig. 8 illustrates the volumetric heating capacity of these simulations, where HFO-1336mzz(Z) has slightly higher volumetric heating capacity than n-Pentane. Note that n-Pentane requires security requirements because of its flammability and slight higher investment cost compared with the other two candidates due to its low volumetric heating capacity.

For heat source temperature of 70 ºC, Fig. 7 shows that, like in the previous application, n-Pentane and HFO-1336mzz(Z) are the potential candidates with a COP of 2.98 and 2.74, respectively. Nevertheless, HCFO-1233zd(E) achieves a COP of 2.50 with higher volumetric heating capacity than the other two candidates, as shown in Fig. 8. For heat source temperature of 50 ºC, similar behaviour than that for heat source temperature of 70 ºC is observed. Similar results are obtained by (Juhasz, 2017), who compares the energy performance of HFO-1336mzz(Z) with other low GWP, obtaining better COP for HFO-1336mzz(Z) than for HCFO-1233zd(E).

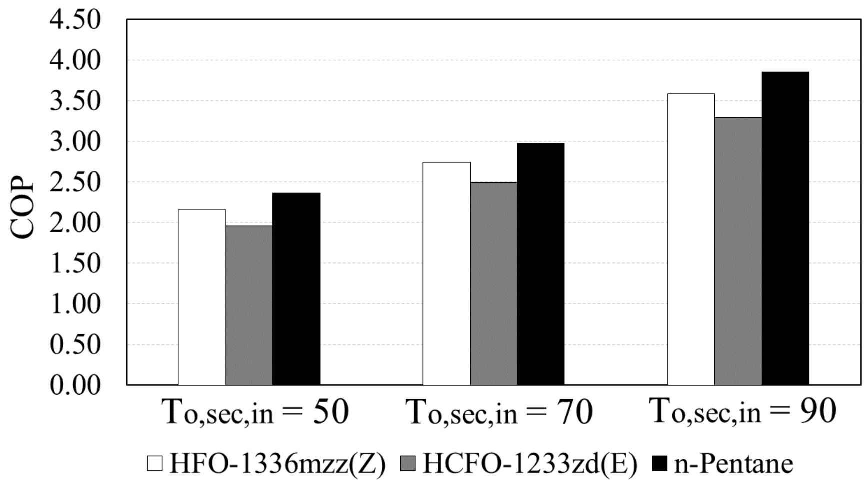


Fig. 7- Two-stage cycle with IHX simulations results of COP for heating production temperature of 150 ºC.

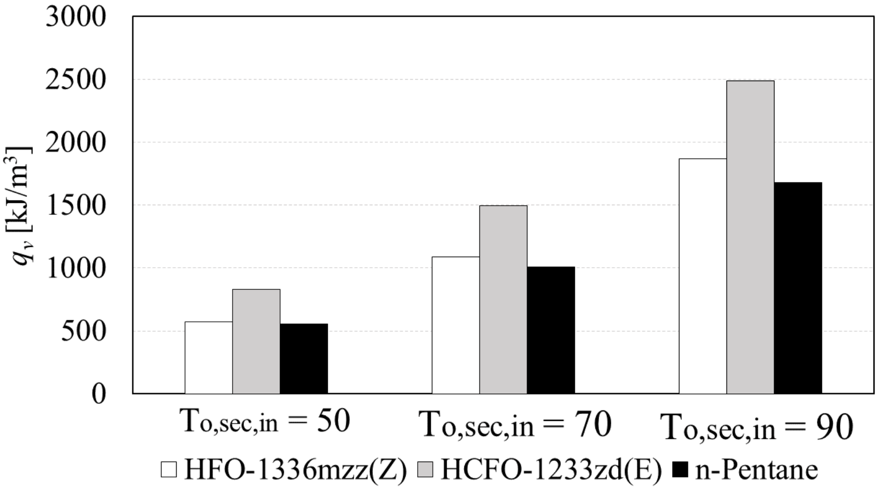


Fig. 8 - Two-stage cycle with IHX simulations results of volumetric heating capacity for heating production.

In general, higher COPs are achieved by n-Pentane in all the options simulated, followed by HFO-1336mzz(Z). Nevertheless, both refrigerants obtain lower volumetric heating capacity than HCFO-1233zd(E). This means that higher displacement compressor and installation size are needed for n-Pentane and HFO-1336mzz(Z) than for HCFO-1233zd(E).

6. Conclusions

Due to the concern for the effects of greenhouse gasses emissions on the global environment there is large interest in Europe and elsewhere for the use of high-temperature heat pumps for waste heat recovery with low GWP refrigerants. This paper has investigated the energy performance of five different system configurations for high-temperature heat pumps using low GWP alternatives to HFC-245fa.

n-Pentane, Butane, HFO-1336mzz(Z) and HCFO-1233zd(E) are the proposed replacements because of the high critical temperatures and promising thermodynamic and transport properties. The proposed configurations are single-stage and two-stage cycles with and without internal heat exchanger in different positions. The optimal intermediate pressure has been calculated for each situation.

Heating production and waste heat source temperatures are highly sensitive to the configuration and working fluid selection. Therefore, temperature lift is a key parameter to analyse the proper configuration. For temperature lift of 40 K, single-stage with IHX presents higher performance than the other configuration, but as temperature lift increases, two-stage with IHX becomes the proper system configuration.

For moderate-high temperature applications with low-temperature lift, single-stage cycle with IHX using n-Pentane achieves a COP improvement up to 15% compared with a single-stage cycle with HFC-245fa. However, n-Pentane has a reduction of 45% of the volumetric heating capacity whereas HCFO-1233zd(E) keeps this reduction up to 5% with a COP improvement up to 11%. For higher temperature lifts, the configuration with highest energy performance is a two-stage cycle with IHX. The COP variation is slightly different for each refrigerant; therefore, volumetric heating capacity becomes the significative parameter. Butane and HCFO-1233zd(E) obtain an increment of the volumetric heating capacity of 70% and 15%, respectively, whereas n-Pentane and HFO-1336mzz(Z) decrease this parameter in 40% and 35%.

For high-temperature applications with production heating temperature up to 150 ºC, two-stage cycle with IHX using n-Pentane achieves the highest COP, followed by HFO-1336mzz(Z) (up to 3.85 and 3.59, respectively). Otherwise, n-Pentane and HFO-1336mzz(Z) obtain up to 50%

lower volumetric heating capacity than HCFO-1233zd(E), which has a COP of 3.42. Hence, depends on the energy or the installation cost, most proper working fluid would be selected.

In conclusion, it seems likely that single-stage and two-stage cycles with IHX will be used as system configurations in high-temperature heat pumps. While the system configuration candidates are clear, working fluid selection is highly dependent on the energy and installation costs aside from the regulation and safety measures of flammable fluid in the installations.

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