THERMODYNAMIC ANALYSIS ON TRILATERAL FLASH CYCLE (TFC) FOR LOW GRADE HEAT TO POWER GENERATION USING DIFFERENT WORKING FLUIDS

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ABSTRACT

Climate change and the global energy crisis strongly emphasize the need not only for low carbon energy production but also for the development of advanced technologies related to heat to power conversion applications. Recent technological advancements indicate that low-grade heat source (<100 °C), could potentially become a more sustainable power source. Trilateral Flash Cycle (TFC) is a thermodynamic cycle, the main difference of which compared to the widely used ORC is that the working fluid does not evaporate at the heating phase, but expands from a saturated liquid state. In particular, the working fluid is pressurized, heated at constant pressure, expanded as a two-phase mixture and eventually condensed at constant pressure. Beyond that, TFC employs the same components as an ORC application. This study aims to elaborate TFC thermodynamic analysis, highlight its efficiency and compare the overall cycle performance by using different working fluids. Working fluid selection is primarily focus on the comparison between promising fourth generation refrigerants such as HFO-1234yf and HFO-1234ze(E), with low GWP and zero ODP and more common ones such as HFCs. For each of the refrigerants, parametric investigations are performed, in order to determine the optimal operation aspects in terms of net power output, gross and net thermal efficiency, exergy and total recovery efficiency. The analysis is carried out with Aspen Plus while REFPROP calculation method was selected. The analysis highlighted the importance that the temperature difference across the heater and the expander isentropic efficiency have for the overall cycle performance. Thermodynamic results revealed that HFO-1234yf had the best power production and thermal efficiency performance, while HFC-245fa had a strong exergy potential for a heat source temperature of 90 °C.

1 INTRODUCTION

In a time of overwhelming population growth and decreasing fossil fuel reserves, the globalized world is mounting pressure upon both energy consumers and providers to explore the idea of sourcing from more sustainable energy sources and technologies (Iqbal *et al* 2020). Low-grade heat is the most common form of thermal energy which industrial processes make available as a byproduct (Marchionni *et al* 2019) and can be found from many sources such as geothermal resources, solar thermal and waste heat (Oreijah *et al* 2013). Moreover, low and medium grade heat sources demonstrate a huge energy recovery potential into mechanical and electrical forms (Cipollone *et al* 2017). Based on energy statistics in Eurostat Database, the total European industrial low-grade waste heat potential is 469 TWh, 12.7 % of the global amount (Bianchi *et al* 2017). The Paris Agreement points out that except of the energy saving measures that must be implemented to lower energy consumptions, the recovery of thermal waste streams is considered as a promising approach to fulfil the relevant to the climate crisis environmental targets.

Waste heat recovery technologies could achieve a conversion of the waste heat to electrical energy among others by using thermodynamic cycles (Zhang *et al* 2013). One of the heat to power generation cycles, the Organic Rankine Cycle (ORC) has been broadly utilized in the existing power plants for conversion of heat to power from low and moderate temperature heat sources (Ahmadi *et al* 2019). The Trilateral Flash Cycle (TFC) also known as the Trilateral Cycle (TLC) holds a good promise as an alternative to ORC (Bianchi *et al* 2018) as might achieve higher net output power (Yari *et al* 2015) and thermal and exergy efficiency (Marchionni *et al* 2019, Nini *et al* 2019) compared to the latter.

Choosing a suitable working fluid for thermodynamic cycles is crucial since it influences on the system efficiency, operating conditions, economic viability and environmental impact (Ahmadi *et al 2019)*. Yari *et al* (2015) studied the variation of the thermal efficiency and the net output power of TLC regarding expander inlet temperature, assuming its isentropic efficiency 0.75 and a water stream of 120 °C as the heat source. Among others refrigerants Yari *et al* (2015) compared HFO-1234yf and HFC-134a, with the former performing better. Cipollone *et al* (2017) conducted a comparison analysis between pure fluids and mixtures on TFC application performance. At this study for which a trans-critical CO_2 whose temperature decreases from 100 °C to 40 °C was considered as a hot source, HFO-1234yf and HFO-1234ze(E) gross overall cycle efficiency was compared and the first was higher. Nevertheless, HFC-134a exergy performance managed to exceed both of the previous ones. At Nini *et al* (2019) study, heat source inlet temperature varied from 60 °C to 160 °C. By analyzing the optimal heat carrier inlet temperature HFO-1234yf, HFC-134a and HFC-245fa compared and the best exergy performance was shown by the latter. Bianchi *et al* (2017) tested HFO-1234yf and HFO-1234ze(E) at a TLC application whereas the hot source is a water stream at 1 kg/s and 90 °C. The results indicate that HFO-1234ze(E) had better performance. It is evident that the availability of new working fluids having a reduced environmental impact, which have begun to be considered in the literature, offers at the same time a new necessity and opportunity for research on their potentialities, as it seems that there is still no clear picture of who the best candidates are and under what operating conditions.

The objective of this study is to provide results of four different working fluids selection in TLC system for utilizing low heat sources, which are shown in Table 1 (Fukuda *et al* 2014, Le *et al* 2014, Minor & Spatz 2008, Yamada *et al* 2012), and are either fourth generation refrigerants with low GWP, either more common ones in order to be compared. This comparative analysis approach will contribute to the potentially expanded introduction of the use of novel thermodynamic cycle applications more advantageous from an energy and environmental aspect. Examined refrigerants energy and exergy performance are initially analyzed based on the temperature difference throughout the heater. Parametric study on the impact of the expander isentropic efficiency is also performed.

Table 1: Characteristics and properties of the examined working fluids

Working fluid	**Formula**	**Flammability & Toxicity**	**Molecular Mass (kg/kmol)**	**T_{crit} (ºC)**	**ODP**	**GWP (100 years)**
HFO-1234yf	$C_3F_4H_2$	No	114.04	94.7	~ 0	4
HFO-1234ze(E)	$C_3F_4H_2$	No	114.04	109.4	~ 0	6
HFC-245fa	$C_3F_5H_3$	No	134.05	154.01	~ 0	1020
HFC-134a	$C_2F_4H_2$	No	102.03	187.2	~ 0	1320

2 TRILATERAL CYCLE FUNDAMENTALS

2.1 Description of the TLC system

Smith (1993) made the first publications describing in detail the Trilateral Cycle which employs the same components as an ORC application – feed pump, heater,expander, condenser- (Ajimotokan 2017, Smith 1993). The actual difference of TLC and ORC is that the working fluid at the entrance of the TLC two-phase isentropic expander is in saturated liquid state (Nini *et al* 2019) instead of the superheated vapor in the ORC (Kanno & Shikazono 2014), and consiquently the fluid expansion through the expander occurs entirely within the two-phase region (Yamada *et al* 2012).

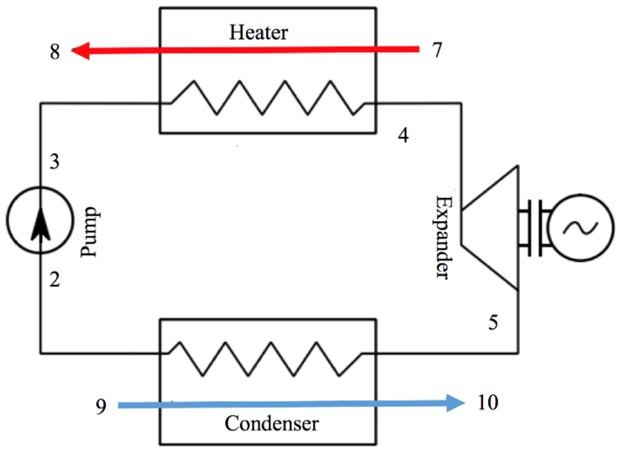

Figure 1a: TLC system components

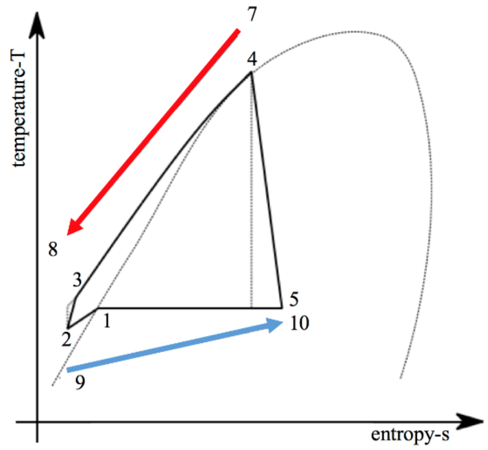

Figure 1b: TLC system temperature-entropy chart

With reference to the schematic cycle configuration and the T-s diagram displayed in Figures 1a and 1b (Bianchi *et al* 2017), the working fluid is pressurizes adiabatically, heated at constant pressure to its saturation point, expanded adiabatically as a two-phase stream and eventually condensed at constant pressure (McGinty *et al* 2017).

2.2 Thermodynamic model: Energy and Exergy analysis

The simulation models of this study have been developed with Aspen Plus version 11.1 commercial tool. In Aspen Plus have already been performed many application tests for thermodynamic cycles (Ambarita & Sihombing 2020, Invernizzi *et al* 2016, Meinel *et al* 2014, Salehi *et al* 2020). Five blocks have been used as except of the four necessary components of the tested cycle, a cooler after the condenser has been also used. For the heater and the condenser, a HeatX block is selected, while for the cooler a Heater block. A pump and a turbine from the pressure changers blocks are also taking place. REFPROP (Lemmon *et al* 2018) is the selected and more suitable property method for calculating the thermophysical properties of the working fluids, whereas IAPWS-95 and PENG-ROB for the water and air properties calculation respectively, since water was selected as the heating and air as the cooling medium, respectively (Peng & Robinson 1976, Wagnera & Pruß 2002). The selected methods have been used in previous simulation works (Atsonios *et al* 2017, Atsonios *et al* 2013, Atsonios *et al* 2021). The following thermodynamic equations are used to analyse thermodynamic performance of TLC system by the different working fluids at the following section. The exergy transfer due to the heat and work can be expressed as:

$$\dot{E}_Q = \sum(1-\frac{T_0}{T})\dot{Q} \tag{1}$$

where the dead state temperature (T_o) and pressure are 25 °C and 1 bar respectively.
The exergy efficiency of the expander is defined:

$$\eta_{exp} = \frac{\dot{W}_{exp}}{\dot{E}_4-\dot{E}_5} \tag{2}$$

Equations 3-7 express the net work done by the system and the energy, the exergy, the heat recovery efficiency and the total heat recovery efficiency respectively.

$$\dot{W}_{net} = \dot{W}_{exp} - \dot{W}_p \tag{3}$$

$$\eta_{th} = \frac{\dot{W}_{exp}}{\dot{E}_3-\dot{E}_4} \tag{4}$$

$$\eta_{exg} = \frac{\dot{W}_{net}}{\dot{E}_7-\dot{E}_8} \tag{5}$$

$$\varphi = \frac{T_{h,in}-T_{h,out}}{T_{h,in}-T_o} \tag{6}$$

$$\eta_{tot} = n_{th}\varphi \tag{7}$$

3 RESULTS AND DISCUSSION

3.1 Working fluid screening based on temperature difference at the heater

In this thermal analysis, a constant temperature difference between the heat source inlet temperature and the working fluid output temperature is assumed, in order to obtain the following output quantities: expander power output (Fig. 2a), gross thermal efficiency (Fig. 2b), net thermal efficiency (Fig. 2c), UA (Fig. 2d), heater heat transfer area (Fig. 2e), pressure ratio (Fig. 2f), overall exergy efficiency (Fig. 2g) and total heat recovery efficiency (Fig. 2h). The operating conditions and constant parameters summary of the analysis are presented in Table 2. For instance, a considered 5 °C temperature difference across the heater indicates that the maximum cycle pressure is the saturation state pressure of each of the tested working fluids at 85 °C.

Table 2: Operating conditions and constant parameters summary

Working fluid mass flow rate, $\dot{m}$ (kg/s)	3.5
Hot carrier-liquid water mass flow rate, $\dot{m}$ (kg/s)	8
Hot carrier-liquid water temperature, $T_{h,in}$ (°C)	90
Hot carrier-liquid water pressure, $P_{ih,in}$ (bar)	5
Cold carrier-vapor air mass flow rate, $\dot{m}$ (kg/s)	22
Cold carrier-vapor air temperature, $T_{c,in}$ (°C)	28
Cold carrier-vapor air mass pressure, $P_{c,in}$ (bar)	1
Heat transfer coefficient ,U (W/m^2 °C)	850
$T_{out\ exp}$ (°C)	40
$\Delta T_{f,cond}$ (°C)	5
Pump isentropic efficiency, η_p	0.8
Expander isentropic efficiency, η_{exp}	0.75

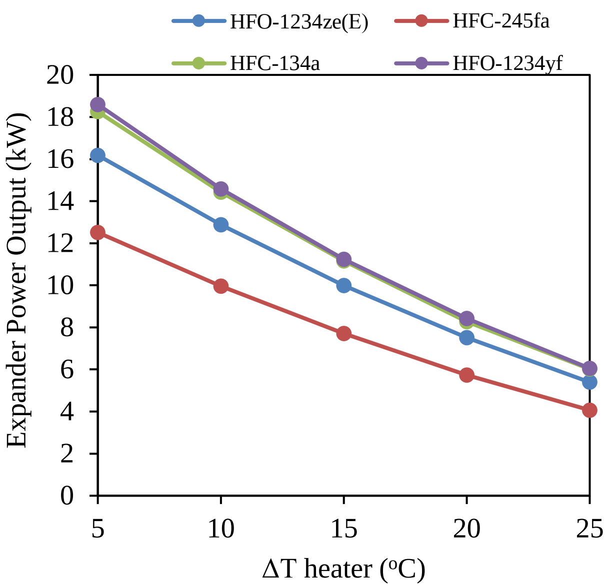

Figure 2a: Impact of temperature difference throughout the heater- Expander power output (kW)

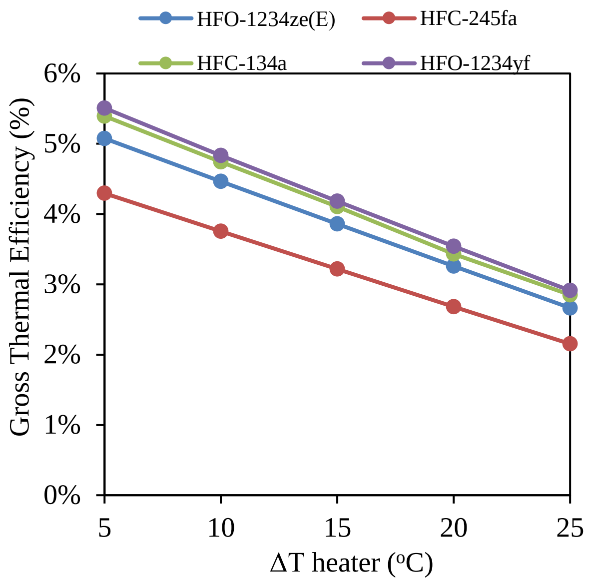

Figure 2b: Impact of temperature difference throughout the heater- Gross thermal efficiency (%)

The results of Figure 2 show that regardless of the working fluid, the decrease in power output (Fig. 2a) and efficiencies (Fig. 2b, Fig. 2c, Fig. 2g, and Fig. 2h) is evident as the maximum cycle temperature and consequently the cycle pressure ratio decrease. Moreover, when the temperature difference across the heater increases, the UA value (Fig. 2d) and the heater heat transfer area (Fig. 2e) present very significant reduction. HFO-1234yf in general presents the best performance for the TFC simulation in terms of power output, thermal efficiency analysis and total heat recovery analysis, while HFC-134a follows with a very slight difference. Obtained results came in accordance with the best performance that HFO-1234yf presented at Yari *et al* (2015) analysis, based on the conducted literature review. With reference to a temperature difference at the heater at 5 °C, HFO-1234yf could produce 18.6 kW at the expander with a gross thermal efficiency of 5.51 %, a net thermal efficiency of 3.35 % and a total heat recovery efficiency of 0.53 %. In terms of net thermal efficiency (Fig. 3c) and overall exergy efficiency (Fig. 2g), the best candidate working fluid seems to be the HFC-245fa, as also Nini *et al* (2019) had presented. Its saturation pressure is much lower comparing to the rest candidates, a fact that leads to low pumping power needs and explains the pressure ratio analysis (Fig. 2f).

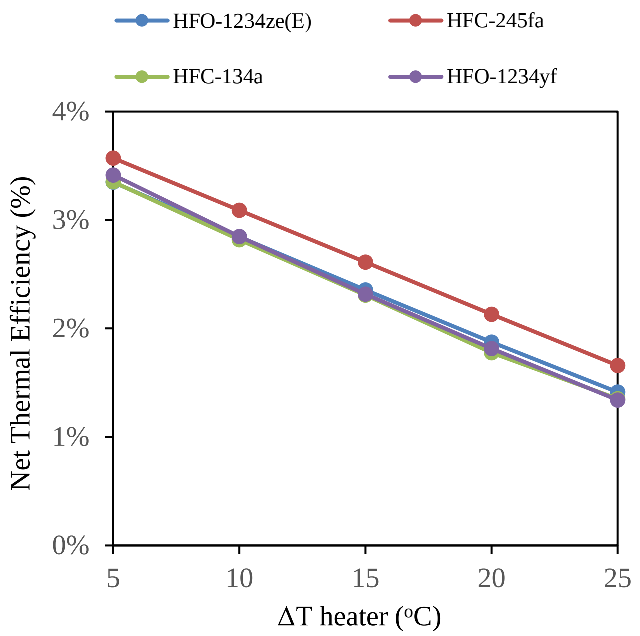

Figure 2c: Impact of temperature difference throughout the heater- Net thermal efficiency (%)

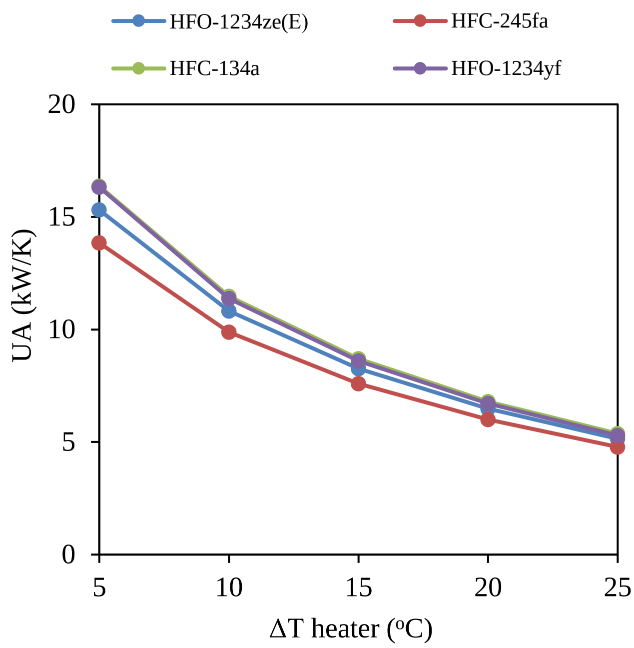

Figure 2d: Impact of temperature difference throughout the heater- UA (kW/K)

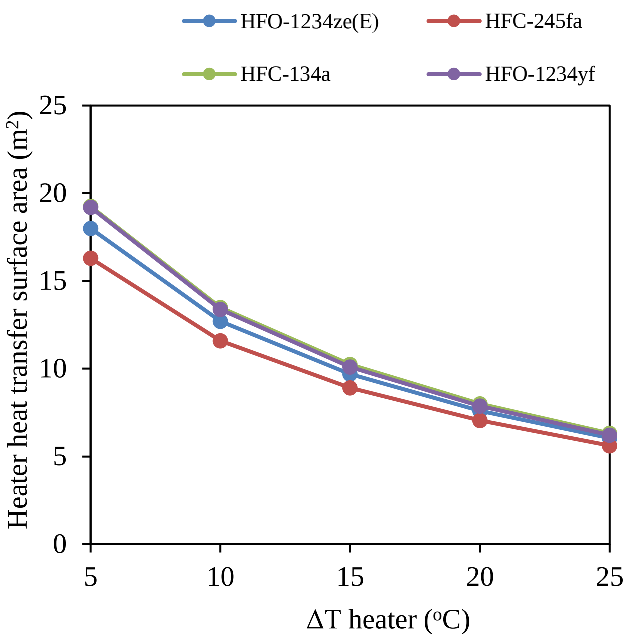

Figure 2e: Impact of temperature difference throughout the heater- Heater heat transfer area (m^2)

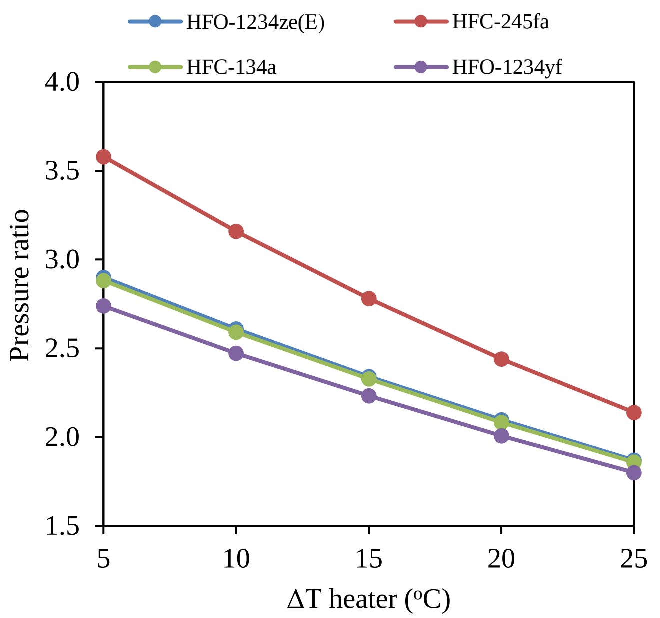

Figure 2f: Impact of temperature difference throughout the heater- Pressure ratio

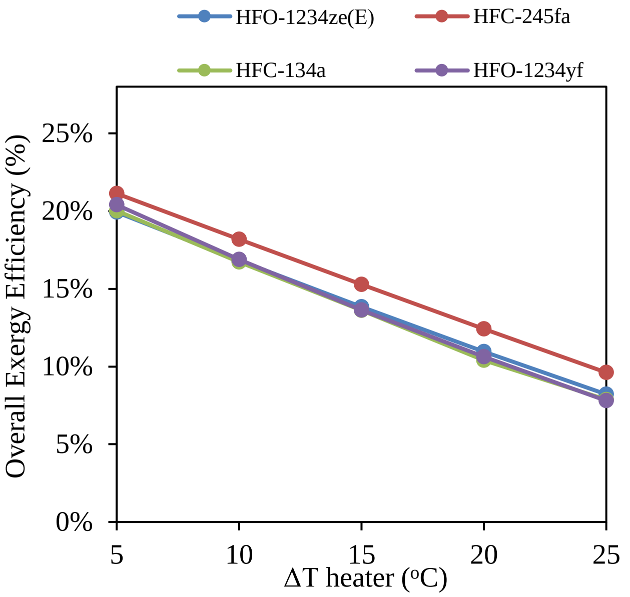

Figure 2g: Impact of temperature difference throughout the heater- Overall exergy efficiency (%)

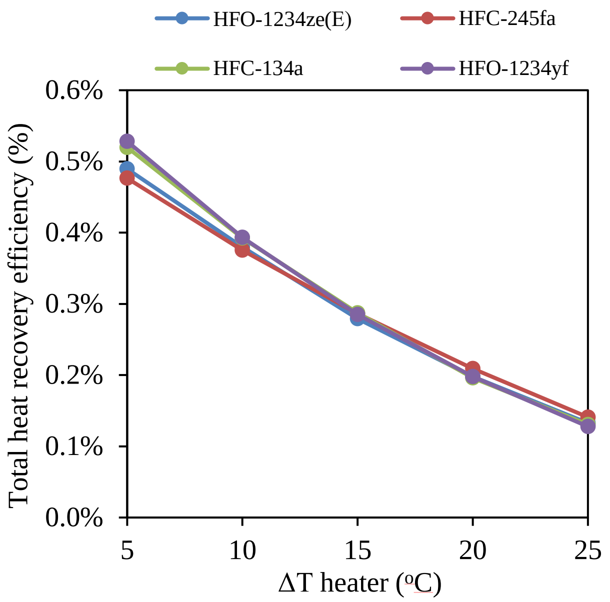

Figure 2h: Impact of temperature difference throughout the heater- Total heat recovery efficiency (%)

As noted, the different thermophysical properties of the selected working fluids whose results are presented above are relevant to the pressure-temperature-enthalpy relations of each working fluid. Figure 3 is an explanatory graph for the different pairs of pressure and temperature values -for the saturation liquid state-, obtained by the REFPROP method in Aspen that validates the aforementioned results as concerns the expander work output. Considering that only that vapor phase contributes to the power production, the smaller the curvature of the curve, the greater the expansion work.

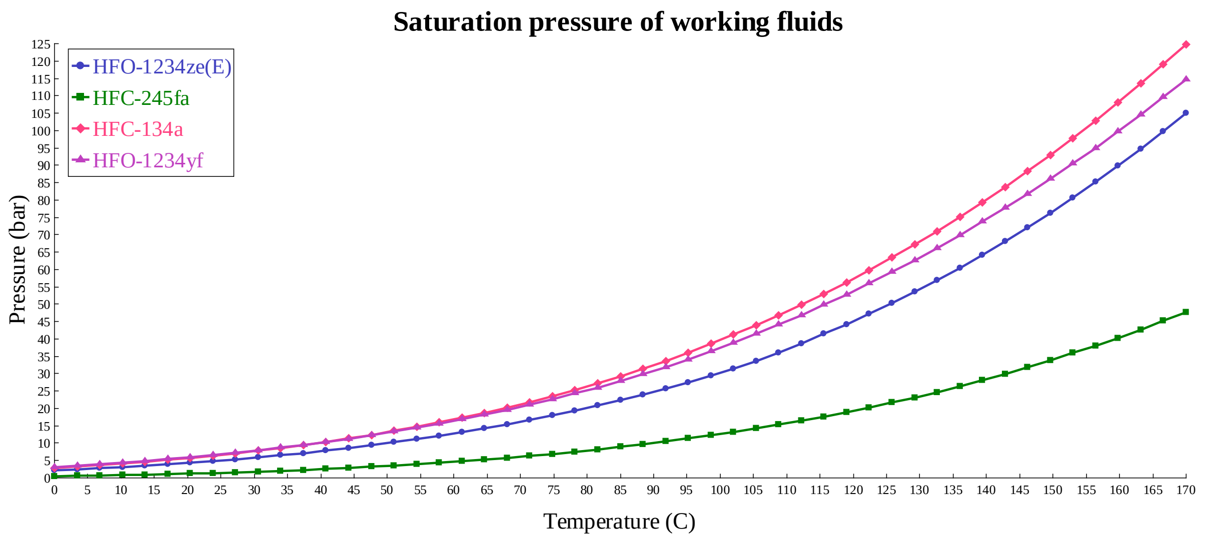

Figure 3: Saturation liquid state for the selected working fluids-ASPEN graph

3.2 Working fluid screening based on expander isentropic efficiency

Parametric analysis on the potential impact of the expander isentropic efficiency is reported in Figure 4. For the current analysis, in addition to the data in Table 2 data, a 10 °C pinch point temperature difference is assumed. Pump isentropic efficiency has a low impact on the cycle performance, but as already mentioned, the two-phase expander is the most important and challenging component of the whole TFC unit.

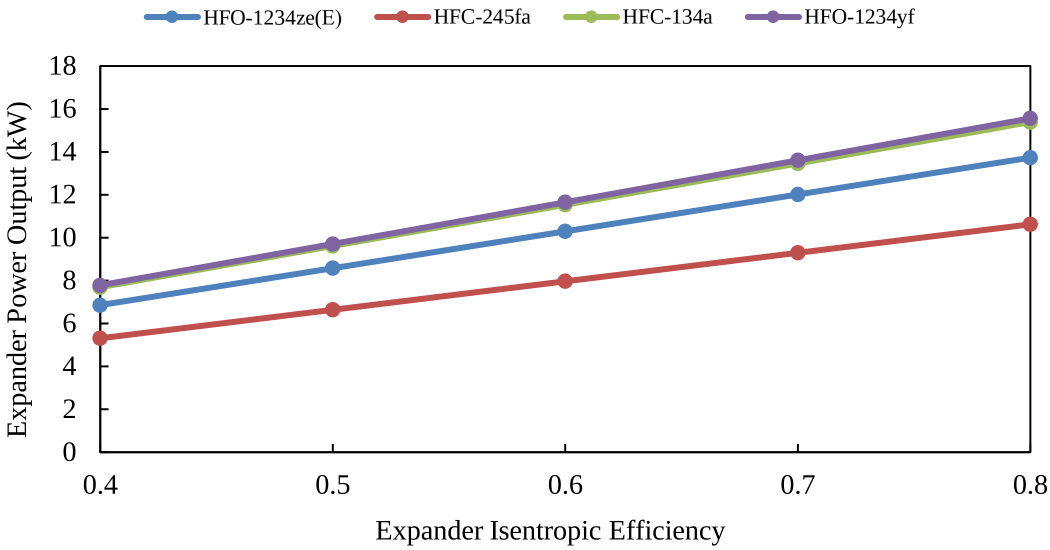

Figure 4a: Effect of expander isentropic efficiency on Expander Power Output (kW)

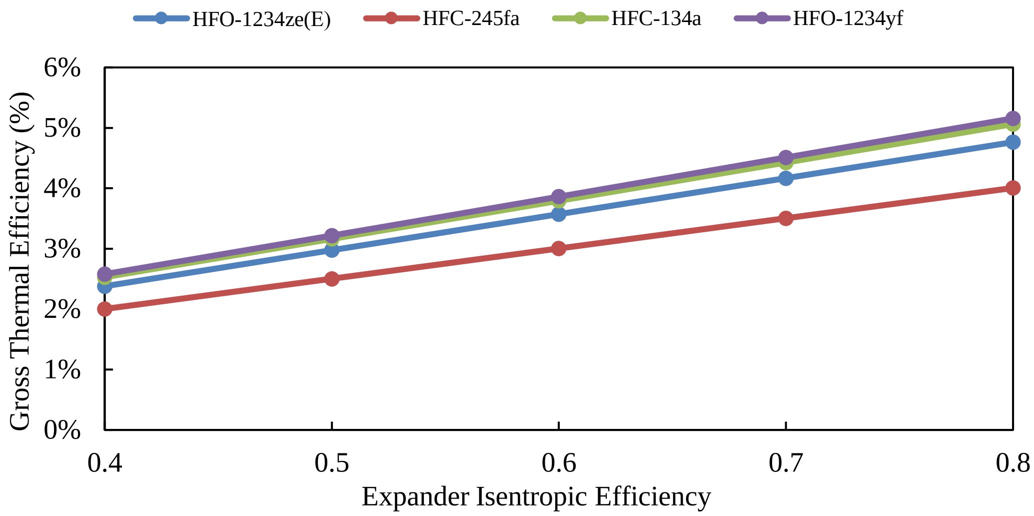

Figure 4b: Effect of expander isentropic efficiency- Gross Thermal Efficiency (%)

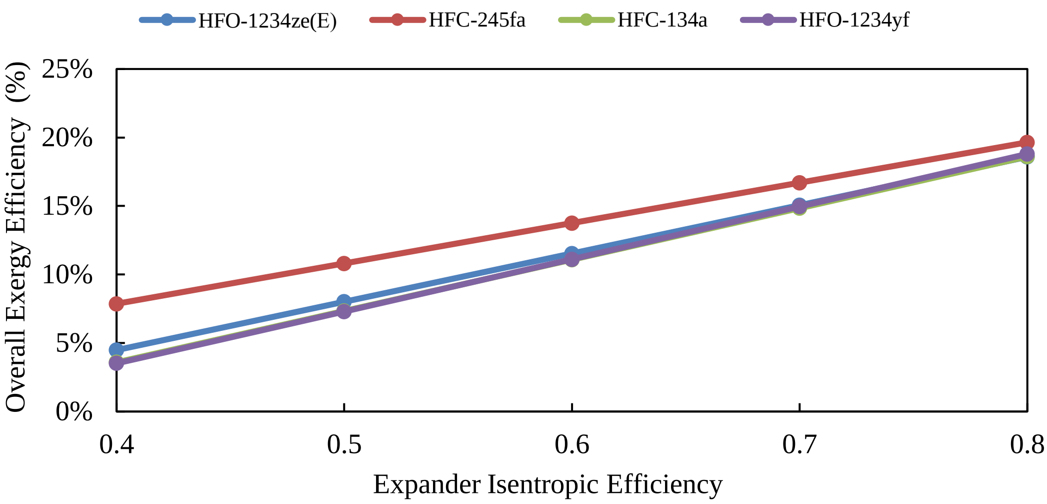

Figure 4c: Effect of expander isentropic efficiency- Overall Exergy Efficiency (%)

Following the same pattern with figures in section 3.1, the isentropic efficiency analysis shows that HFO-1234yf reached the highest power output values, as for 0.8 expander efficiency produced 15.6 kW_e. The expander isentropic efficiency plays an important role in the cycle power output and performance, implying that should be taken special attention in the manufacturing of such a novel compound. HFC-134a and HFO-1234ze(E) follow while the lowest values achieved by the HFC-245fa which, as also in section 3.1, presents the higher overall exergy efficiency. The latter is explained by the much lower pressure ratio that HFC-245fa has during the expansion cycle, compared to the other refrigerants. Different pressure values for the liquid saturation state lead to different amounts of heat duty and in this case, higher rates of exergy efficiency.

4 CONCLUSIONS

Following the working fluid sorting procedure, the results were particularly encouraging as they demonstrate that primarily HFO-1234yf but also even HFO-1234ze(E) could achieve competitive overall cycle performance, compared to more common refrigerants such as HFCs whose GPW is much higher and therefore their use do not keep pace up with climate crisis adaptation trends. Even if simulations with HFC-245fa presented the highest overall exergy and net efficiency, HFO-1234yf achieved the best power production and thermal efficiency performance. The analysis also confirms that for the case of the TLC system, as the expander inlet temperature approaches the temperature of the heat source, which means the smaller the temperature difference across the heater is, the power output and the thermal and exergy efficiency increase. Moreover, the two-phase isentropic expander is the key element of the cycle performance as its impact on it is undoubtedly significant.

NOMENCLATURE

ODP	ozone depletion potential	
GWP	global warming potential	
TLC	trilateral cycle	
TFC	trilateral flash cycle	
ORC	Organic Rankine Cycle	
HFO	hydrofluoro-olefin	
HFC	hydrofluorocarbon	
T	temperature	(ºC)
P	pressure	(bar)
$\dot{m}$	mass flow rate	(kg/s)
U	heat transfer coefficient	(W/ m^2 ºC)
$\dot{E}$	exergy rate	(W)
$\dot{Q}$	heat transfer	(W)
$\dot{W}$	work transfer	(W)
η	efficiency	(%)
φ	heat recovery efficiency	(%)

Subscript

crit	critical
in	inlet
out	outlet
f	working fluid
h	hot water
c	cold air
exp	expander
p	pump
th	thermal
exg	exergy

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NIST Reference Fluid Thermodynamic and Transport Properties Database (REFPROP)

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