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Parametric Analysis of Heat Source and Sink and Design of Heat Exchangers for Trilateral Flash Cycle (TFC)

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Abstract – The utilization of a low-grade heat source (<100 ºC) can improve energy conservation and reduce the harmful emissions contributing effectively to the environment protection. Consequently, the advancement of technologies related to heat-to-power conversion applications needs to be considered, since large amounts of energy from industrial applications are rejected to the ambient. Trilateral Flash Cycle (TFC) is a thermodynamic cycle that employs the same components as an organic Rankine cycle (ORC) unit, while the main difference with the latter is that the working fluid does not evaporate at the heating phase but expands from a saturated liquid state. The present work aims to achieve primarily the investigation of a TFC unit towards the commercialization if such cycles, focusing on the heat exchangers and their detailed design, before the manufacturing and the experimental study of the unit. The simulation models of the TFC unit have been developed employing the commercial software Aspen Plus for process modelling and Aspen Exchanger Design and Rating (EDR). The Aspen Plus is used to simulate the overall system while heat exchangers are modelled by Aspen EDR.

Keywords**:** Trilateral flash cycle, waste heat recovery, heat-to-power engines, condenser

1. Introduction

Nowadays, there is a growing debate over the efficiency of the energy systems, since the global climate change has become an important issue, while the rising fuel prices make the industries to look for innovative solutions that contribute to operating costs reduction. A comprehensive, high-efficiency and clean utilization of energy is critical for a sustainable development [1]. Waste heat is usually a 24-hour discharged heat from industrial processes, and approximately 63% of the rejected heat is considered low-temperature (<100 ºC). Efficient reuse of waste heat improves the energy efficiency of the systems, drives into reduced harmful emissions and improves the cost effectiveness. Among all, the waste heat-to-power technologies, the organic Rankine cycle (ORC) is considered a mature technology, featuring high reliability and low cost. It is arguably the most efficient energy conversion solution for low-temperature waste heat power generation at present, while it is widely proposed to recover industrial waste heat, geothermal energy, biomass heat and solar thermal energy. The Trilateral Flash Cycle (TFC) system is an alternative solution to low-temperature sites for energy saving, which is relatively unexplored and may lead to economic benefits compared to conventional technologies, since it is aimed at promoting the utilization of waste heat in future power generation. In order to maximize the benefits of the system and system optimization in terms of efficiency, several studies have been conducted exploring components design or favorable working fluids for a designated heat source, which is one of the most important phases in building TFC and ORC systems [2]. In short, in the TFC concept, a liquid-liquid heater replaces the evaporator of the ORC engine, since heat gain is achieved without phase change of the organic working fluid, and the expansion process therefore starts from the saturated liquid state rather than a vapour phase (Fig. 1). The working fluid is pressurized adiabatically, heated at constant pressure to its saturation point, expanded adiabatically as a two-phase mixture and eventually condensed at constant pressure [3]. Most of research works conducted in the field focus on parametric investigations, regarding the net electrical power output (kW), overall energy and exergy efficiency (%), in order to better understand TFC performance not only by comparing different working fluids [4], but also by comparing TFC applications with other thermodynamic cycles [2]. The current research work represents the continuation of the study of Antonopoulou et al. [5], which aimed to elaborate on TFC thermodynamic analysis, highlight its efficiency and compare the overall cycle performance by using different working fluids.

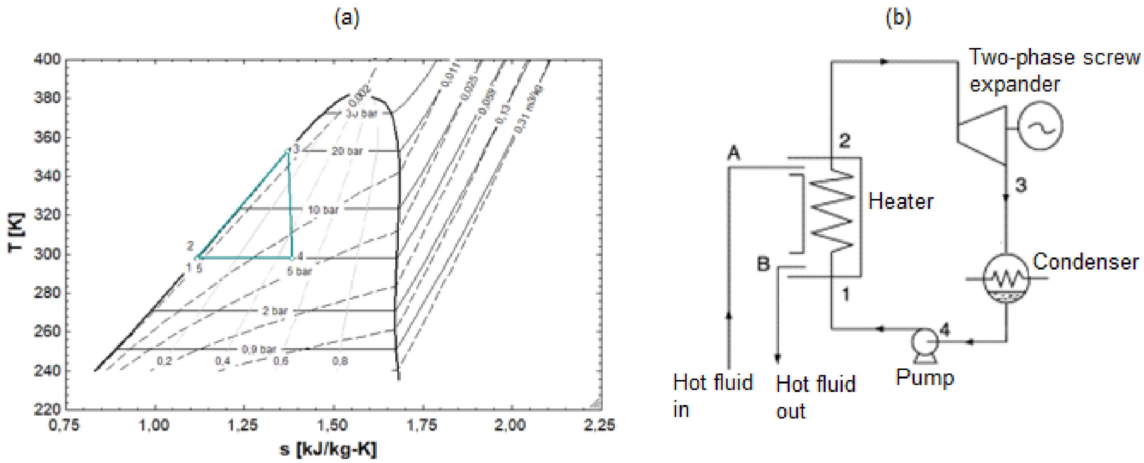

Fig.1: Thermodynamic cycle (a), and configuration (b) of Trilateral Flash Cycle.

Parametric investigations of the TFC cycle were performed to determine the optimal operation aspects in terms of net power output, gross and net thermal efficiency, exergy, and total recovery efficiency. This paper aims to elucidate the accurate design of the heat exchangers of the trilateral application, as the research team tends to manufacture, install and test a TFC system on a real biogas power plant. Among the objectives of this study is to provide comparative results of different working fluids selection in TLC system for utilizing low heat sources, studying both 4th generation refrigerants with low GWP and conventional HFCs with proved performance but eliminated eco-friendliness. Table 1 presents the selected refrigerants along with their key thermophysical properties.

Table 1: Characteristics and properties of the examined working fluids.

Working fluid	Generation	Formula	Molecular Mass (kg/kmol)	T_{crit} (°C)	GWP (100 years)
HFO-1234yf	4	$C_3F_4H_2$	114.04	94.7	4
HFO-1234ze(E)	4	$C_3F_4H_2$	114.04	109.4	6
HFO-1233zd(E)	4	$C_3H_2ClF_3$	130.50	165.5	1
HFC-245fa	3	$C_3F_5H_3$	134.05	154.0	1020
HFC-134a	3	$C_2F_4H_2$	102.03	101.1	1320

2. Modelling and Parametric Analysis

To investigate the impact of heat source and sink temperatures, a simulation model has been developed using the commercial tool Aspen Plus, which is specialized in process modelling and employs blocks for each cycle's component, namely the pump, the two-phase expander, the heater and the condenser. For the heater and the condenser, a HeatX block is selected, while a pump and an expander from the pressure changers blocks are also used. The simulation tool employs the Nist REFPROP database [6] to estimate the thermophysical properties of the working fluids, whereas IAPWS-95 and Peng-Robinson for the water and air properties, respectively [7, 8]. The operating conditions and constant parameters used in the analysis are presented in Table 2.

Table 2: Operating conditions and basic constant parameters summary.

	Heat source parametric analysis	Heat sink parametric analysis
Working fluid mass flow rate $\dot{m}_f$ (kg/s)	3.5	
Hot carrier-liquid water mass flow rate $\dot{m}_h$ (kg/s)	8	
Cold carrier-vapor water mass flow rate $\dot{m}_{cwater}$ (kg/s)	8	
Cold carrier-vapor air mass flow rate $\dot{m}_{cair}$ (kg/s)	-	28
Cold carrier- liquid water temperature $T_{in,c}$ (ºC)	-	90
Hot carrier- liquid water temperature $T_{in,h}$ (ºC)	28	-
Hot carrier pressure, $P_{in,h}$ (bar)	2	1
Cold carrier pressure $P_{in,c}$ (bar)	1	
Working fluid temperature change in the condenser, $\Delta T_{f,cond}$(ºC)	5	
Working fluid temperature in heater outlet, $T_{fout,heater}$ (ºC)	-	80
Temperature difference between heat source and refrigerant at heater outlet, ΔT (ºC)	10	-
Heat transfer coefficient, U (W/m^2 ºC)	850	
Pump isentropic efficiency, n_{pump}	0.8	
Expnder isentropic efficiency, n_{exp}	0.75	

2.1. Working fluid investigation based on heat source temperature

The heat source inlet temperature is studied in the range 70 - 110 ºC, corresponding to varying saturation pressure for each refrigerant, which in turns leads to different pumping work. Table 3 presents the temperature and pressure values at the saturated liquid state for each working fluid that are obtained by the REFPROP method in Aspen Plus [6]. As expected, the required pressure rise gets higher as the heat source temperature, corresponding to the heating temperature of the refrigerant, increases from 60 ºC to 100 °C, taken under consideration the temperature difference between heat source and refrigerant that is stated in Table 2.

Table 3: Temperature and pressure values of the working fluid at the saturated liquid state.

Working fluid	$T_{out,heater}$	60 ºC	70 ºC	80 ºC	90 ºC	100 ºC
HFO-1234ze(E)	$P_{in,heater}$	12.8 bar	16.15 bar	20.1 bar	24.76 bar	30.27 bar
HFC-245fa		4.63 bar	6.1 bar	7.9 bar	10.06 bar	12.65 bar
HFO-1234yf		16.42 bar	20.45 bar	25.2 bar	30.82 bar	-
HFC-134a		16.85 bar	21.18 bar	26.35 bar	32.6 bar	39.72 bar
HFO-1233zd(E)		3.95 bar	5.12 bar	6.58 bar	8.35 bar	10.45 bar

Based on the above mentioned parametric analysis, the impact of the heat source inlet temperature on the net thermal efficiency and the overall exergy efficiency was investigated. It was revealed that regardless of the working fluid, the increase of the heat source temperature improves the thermal efficiency of the cycle, achieving an efficiency about 3.5% for 90 °C (Fig. 2), while the exergy efficiency is estimated slightly higher than 20% when HFC-1233zd(E) is employed at 90 °C.

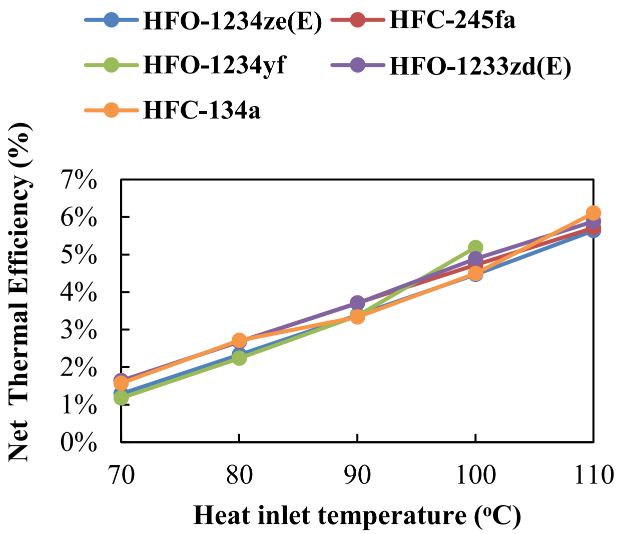

Fig.2: Impact of the heat source temperature on net thermal efficiency.

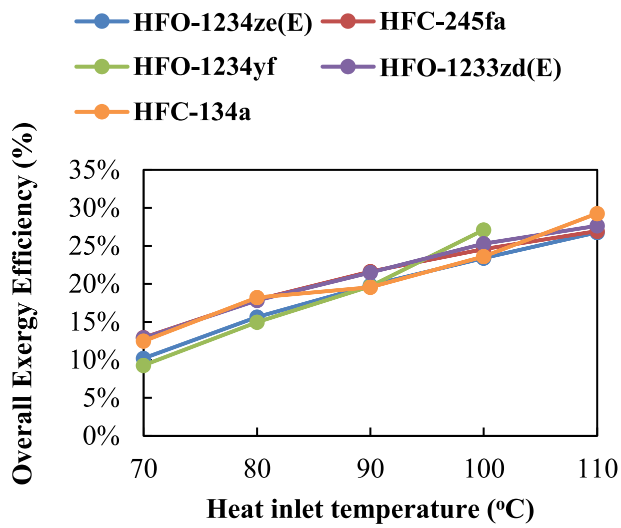

Fig. 3: Impact of the heat source temperature on overall exergy efficiency.

2.2. Working fluid investigation based on heat sink temperature

Apart from the hot source temperature, the temperature of the cooling medium was also investigated, as it may be varied from the kind of application and significantly affects the size of the heat exchanger. This study initially focused on the nature of the sink to define the condenser type that would ultimately be designed and manufactured (water-cooler or air-cooled engine). The total exergy destruction (kW) was investigated for varying water and air inlet temperature from 18 °C to 30 °C, illustrating that in both cases the exergy destruction at the condenser slightly decreases as the sink carrier temperature increases. In addition, the analysis pointed out that in the case of air the exergy destruction is slightly lower than the case of water-cooled engine. Therefore, an air cooler would probably be a beneficial condenser type choice. In addition, for the whole working fluid sorting procedure, the results demonstrate that HFOs could achieve competitive overall cycle performance, compared to more common refrigerants such as HFCs that do not comply with F-Gas regulation.

Fig. 6 and Fig. 7 present the condenser surface (m^2) and the overall heat transfer coefficient, (kW/°C), respectively, as the air inlet temperature increases. As it is expected, as the air inlet temperature increases, the heat exchanger surface (m^2) gets higher, while the overall heat transfer coefficient increases too for each working fluid.

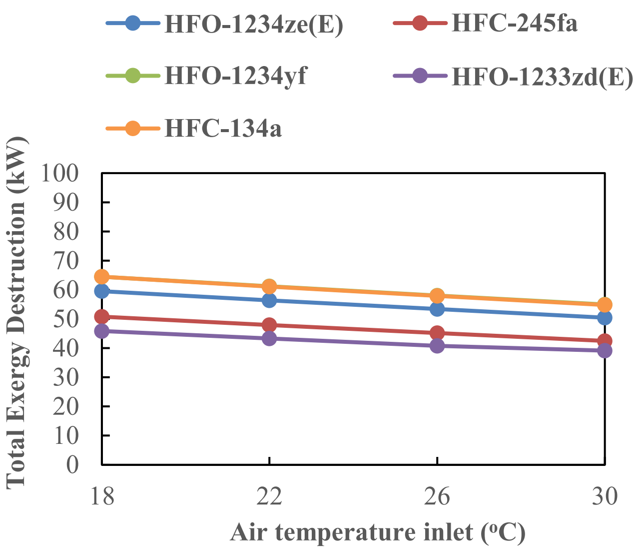

Fig. 4: Impact of the heat sink temperature on the total exergy destruction (kW) for cooling air.

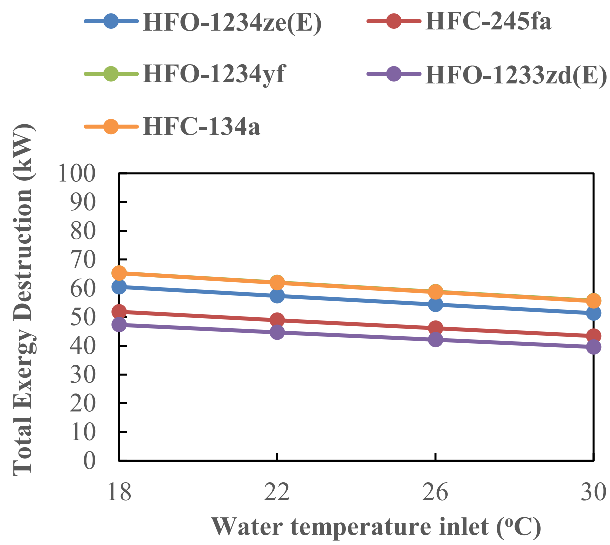

Fig.5: Impact of the heat sink temperature on the total exergy destruction (kW) for cooling water.

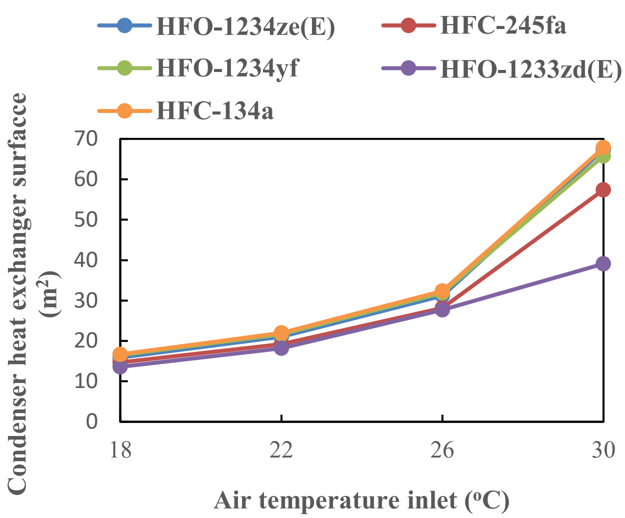

Fig.6: Impact of the cooling air inlet temperature on the condenser surface (m^2).

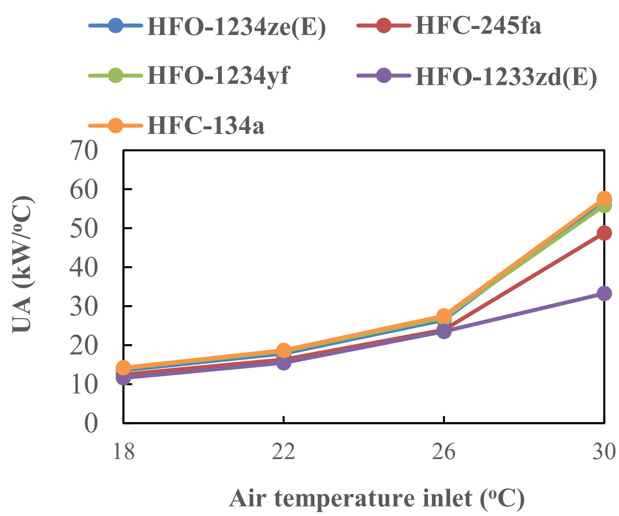

Fig 7: Impact of the cooling air inlet temperature on the overall heat transfer (kW/°C).

3. Heat Exchangers Detailed Design

After determining basic design parameters, a heat exchanger design procedure is proposed for both heat exchangers to calculate the heat transfer area and determine heat exchanger geometrical characteristic that fulfils all design specifications provided by the thermodynamic cycle at design condition. The heat exchangers geometry configuration and the design results are taken by Aspen EDR (Exchanger Design & Rating) software that is a reliable tool for exchanger design, already used by previous studies on thermodynamic cycles [1, 9, 10]. The design fully meets the following criteria:

- Design Code: ASME Code Sec VIII Div 1

- Service class: Normal
- TEMA class: R-refinery service
- Material standard: ASME

The heater is selected as a shell and tube heat exchanger with vertical baffles, while the air-cooled condenser is selected as a plate-fin heat exchanger, where the organic refrigerant fluids through the tubes. Following, the specifications of the design for each heat exchanger type are presented.

3.1. Heater specifications

The heat exchanger was modeled as a shell and tube heat exchanger, where the HFO-1234ze(E) passes through the shell, while the hot water flows through the tubes side. The type of heat exchanger was selected as shell and tube instead of plate heat exchanger, since the heater has a two-phase fluid at the outlet and it is proposed for such applications. The inlet and outlet conditions, as well as the refrigerant mass flow, are determined by the thermodynamic modelling of the cycle. The horizontal shell and tube heat exchanger type was estimated as a BEM type, namely bonnet bolted or integral with tube sheet, having one pass shell and bonnet type heads being the most common for use in applications where the head does not have to be removed frequently. Figure 8 presents the sketch of the heater with the basic geometric characteristics.

Table 4: Specifications of the shell & tube heater.

Fluid allocation	**Shell Side**		**Tube Side**	
Fluid name	***HFO-1234ze(E)***		***H_2O***	
Fluid quantity (kg/s)	2.6		7.7	
	IN	**OUT**	**IN**	**OUT**
Vapor (kg/s)	0		0	
Liquid (kg/s)	2.6		7.7	
Temperature (ºC)	25.6	80	90	83.3
Pressure (bar)	20.07	20.05	5	4.84
Density (kg/m^3)	1168.9	929.8	965.5	969.8
Viscosity (mPa s)	0.195	0.093	0.314	0.34
Special heat (kJ/kg K)	1.37	1.8	4.2	4.2
Thermal conductivity (W/m K)	0.075	0.057	0.673	0.663
Latent heat (kJ/kg)	110.2	110.4		
Pressure drop (bar)	0.1	0.04	0.4	0.17
Velocity mean/max (m/s)	0.23 / 0.29		1.5 / 1.29	
Reynolds Number	8141	17000	31083	31015.5
Prandtl Number	3.56	2.96	1.96	2.13
Heat exchanged (kW)	215.9			
Transfer rate (W/m^2 K)	759.4			
Clean U (W/m^2 K)	1207.2			

As stated in Table 4, the heat exchanged is 215.9 kW and the transfer rate 759.4 W/ m^2 K. Table 5 presents the results of the Aspen Plus simulations, covering the required conditions of the designed thermodynamic cycle. The total number of tubes will be 276, the effective tube length is 1500 mm and the effective surface area per shell 12.4 m^2.

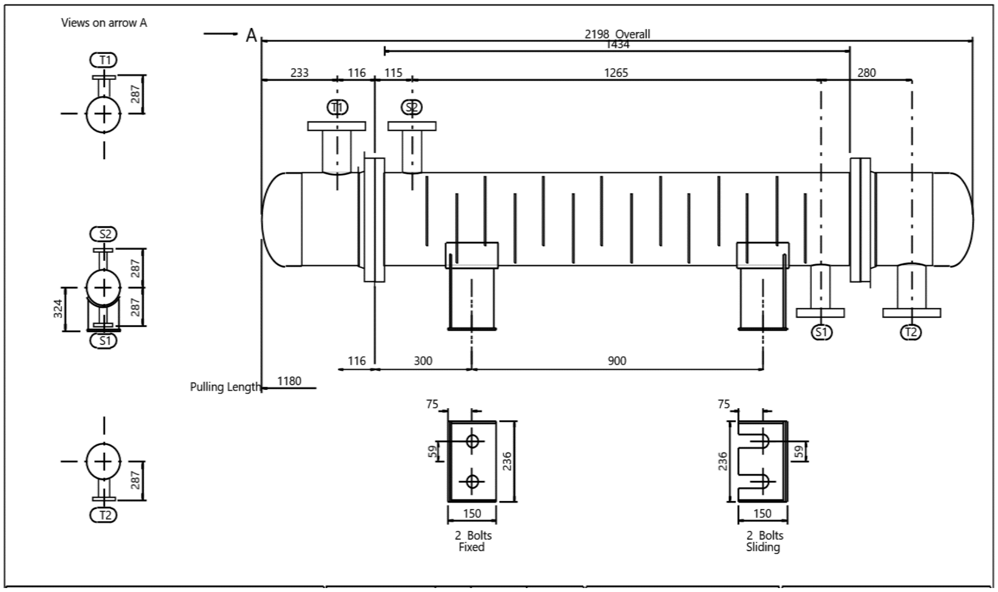

Fig. 8: Geometry of the shell and tube heat exchanger designed as heater.

Table 5: Proposed Shell & Tube detailed design data.

Shell & Tube - Basic Geometry			
Tubes - Geometry		**Baffles -Geometry**	
Type	Plain	Type	Single segmental
Total number of tubes	276	Number	14
Tube length actual (mm)	1500	Spacing (center – center) (mm)	90
Tube length effective (mm)	1435	Spacing at inlet (mm)	132.48
Tube passes	1	Spacing at outlet (mm)	132.47
Outside diameter (mm)	10	End lengthof the front head (mm)	165
Inside diameter (mm)	7	End length of the rear head (mm)	165
Wall thickness (mm)	1.5	Actual Baffle cut (%diameter)	24
Tube pitch (mm)	13	Cut orientation	Horizontal
Tube pattern (mm)	30	Cut thickness (mm)	3.18
Material	Carbon Steel		
Thermal conductivity (W/m K)	50.8		
Bundle - Geometry			
Shell ID to center 1st tube row:			
From top (mm)		67.44	
From bottom (mm)		67.44	
From right (mm)		6.73	
From left (mm)		6.73	
Gross surface area per shell (m^2)		13	
Effective surface area per shell (m^2)		12.4	

3.2. Condenser- Air cooler heat exchanger detailed design

The condenser was simulated also with the commercial Aspen Plus and EDR. It was designed for forced air configuration, employing copper tubes. The design was based on manufacturing restrictions given by the manufacturer. 6 presents the operating conditions of the refrigerant and air for the condenser, while Table 7 presents the basic of the air cooler. The heat exchanged was calculated 284.6 kW, while the bundle face area of 7.5 m^2, which is listed in 7, seems to be relatively high due to the high value of 159 kJ/kW of latent heat, based on the thermodynamic cycle

Table 6: Performance of the Air Cooler unit.

Fluid allocation	**Tube Side**		**X-Side**	
Fluid name	***HFO-1234ze(E)***		***Air (50% humidity)***	
Total flow (kg/s)	2.6		39.6	
	IN	**OUT**	**IN**	**OUT**
Vapor(kg/s)	1.74	0	39.6	39.6
Liquid (kg/s)	0.86	2.6	0	
Condensed (kg/s)	1.74		0	
Temperature (°C)	34.98	32.84	20	27.16
Pressure (bar)	6.67	6.61	0.1	0.1
Velocity (m/s)	0.9	0.04	8.08	8.28
	Liquid/ Vapor			
Density (kg/m^3)	1129.4 / 35.25	1136.7	1.19	1.16
Viscosity (mPa s)	0.168 /0.01	0.172	0.0182	0.0186
Specific heat (kJ/kg K)	1.421 /1.023	1.413	1.003	
Thermal Conductivity (W/m K)	0.0709 / 0.0145	0.0716	0.0256	0.0261
Reynolds No	1366.9 / 36269.4	4025.8	8361.7	8207.8
Prandtl No	3.37 / 0.91	3.41	0.71	
Latent heat (kJ/kg)	159			
Heat exchanged (kW)	284.6			
Clean U (W/m^2 K)	686.6			

Table 7: Proposed air cooler's basic geometrical data.

Air Cooler Basic Geometry			
Bays per unit		1	
Bundles per bay		1	
Bay width (m)		3.37	
Bundle width (m)		3.15	
Unit length (m)		2.95	
Unit height (m)		2.45	
Bundle - Geometry		**Fins - Geometry**	
Tubes per bundle	320	Type	Tube-in-plate
Tubes rows per bundle	4	Material	Aluminum 1060
Tubes per row per bundle	80	Tip diameter (mm)	140
Tube passes per bundle	1	Fin height (mm)	13.17
Total tube length (m)	2.42	Mean fin thickness (mm)	0.22
Effective tube length (m)	2.3442	Fin frequency (#m)	400

Tubesheet thickness (mm)	31.75	Conductivity (W/m K)	232.61
Tube support width (mm)	25	Density kg/m^3	2768
Bundle face area (m^2)	7.5	**Circular tubes - Geometry**	
Tube row arrangement	Staggered-even rows to right	OD (mm)	15.875
Tube transverse pitch (mm)	40	ID (mm)	14.875
Tube row longitudinal pitch (mm)	35	Wall thickness (mm)	0.5
Layout angle (degrees)	30		

4. Conclusion

The analysis performed showed that regardless of the working fluid, increasing the heat source temperature improves the net thermal and the overall exergy efficiency of the TFC cycle. Investigating the nature of the cooling medium nature showed that the total exergy destruction (kW) for varying water and air inlet temperature slightly decreases as the sink carrier temperature increases. As in the case of air, the exergy destruction value is slightly lower than in the case of a water-cooled engine, an air cooler is considered a beneficial condenser type choice. In addition, the temperature of the air cooling medium was also investigated and shown to significantly affects the size of the heat exchanger. The results of the working fluid screening process indicate that HFOs could achieve competitive overall cycle performance, compared to more common refrigerants such as HFCs that do not comply with F-Gas regulation. The heat exchanger design proposed for both heat exchangers to calculate the heat transfer area and determine heat exchanger geometrical characteristic that fulfils all design specifications provided by the thermodynamic cycle at design conditions will be used as a valuable basis for the TFC unit manufacture and the respective cost analysis and techno-economic assessment.

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