Parametric analysis of heat source and sink and design of heat exchangers for Trilateral Flash Cycle (TFC)

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ΓΕΩΠΟΝΙΚΟ ΠΑΝΕΠΙΣΤΗΜΙΟ ΑΘΗΝΩΝ AGRICULTURAL UNIVERSITY OF ATHENS
CPERI Chemical Process & Energy Resources Institute
CERTH CENTRE FOR RESEARCH & TECHNOLOGY HELLAS

Christina Antonopoulou, Apostolos Gkountas, Konstantinos Atsonios, Panteleimon Bakalis, Anastasios Skiadopoulos, Panagiotis Grammelis, Dimitrios Manolakos

INTRODUCTION

- 63% of the rejected heat is considered low-temperature (<100 °C).
- The Trilateral Flash Cycle (TFC) is a relatively unexplored heat-to-power solution for low-temperature heat rejection.
- In the TFC concept, a liquid-liquid heater replaces the evaporator of the ORC, so the expansion starts from the two-phase region.
- The current work studies the TFC thermodynamically, investigating the efficiency and overall performance for different working fluids.
- Parametric investigations of the TFC cycle were performed to determine the optimal operation aspects in terms of net power output, thermal efficiency, exergy, and total recovery efficiency.
- The design process of the heat exchangers of the TFC is examined, to find out the optimum design to manufacture it for a real biogas power plant.

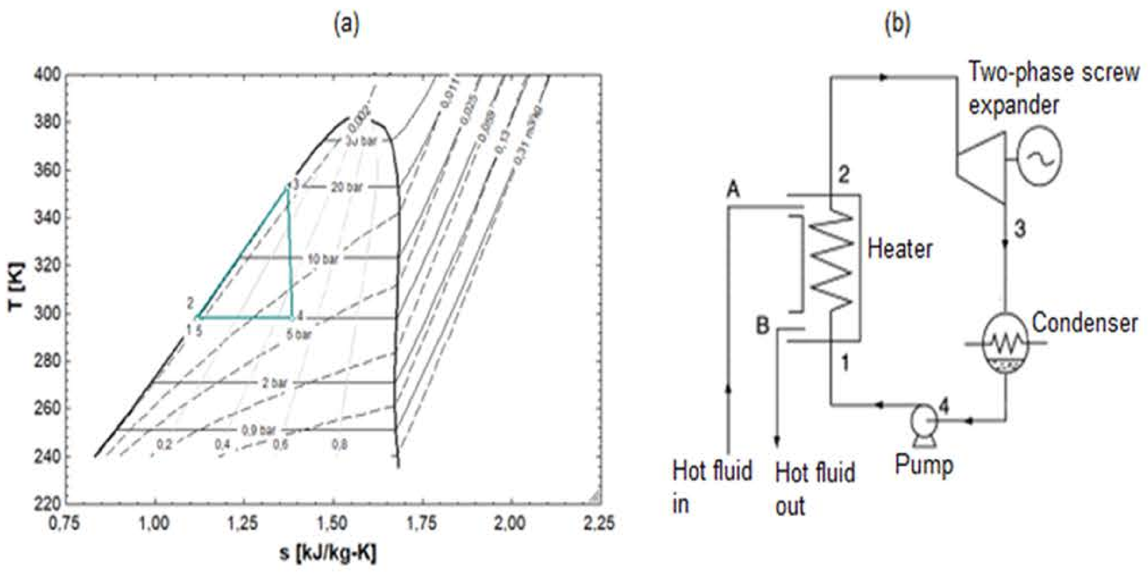

Fig.1: Thermodynamic cycle (a), and configuration (b) of Trilateral Flash Cycle.

MODELLING AND PARAMETRIC ANALYSIS

Working fluid investigation based on heat source temperature

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.

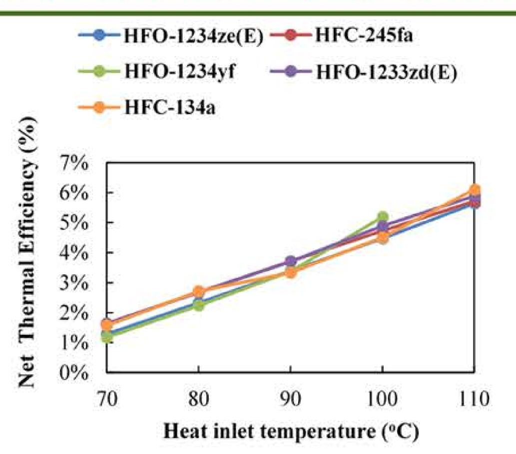

Fig.2: Impact of the heat source temperature on the thermal efficiency.

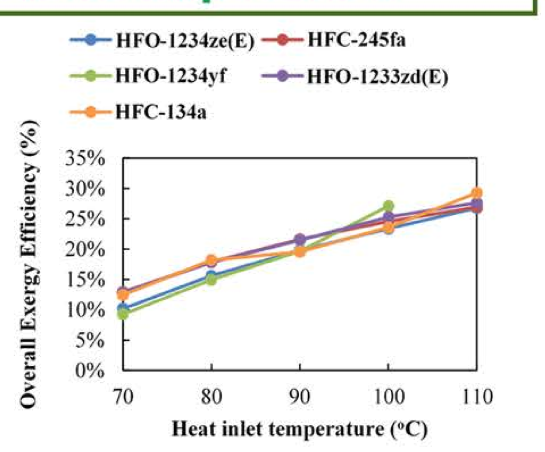

Fig.3: Impact of the heat source temperature on overall exergy efficiency.

Working fluid investigation based on heat sink temperature

An air cooler would probably be a beneficial condenser type choice

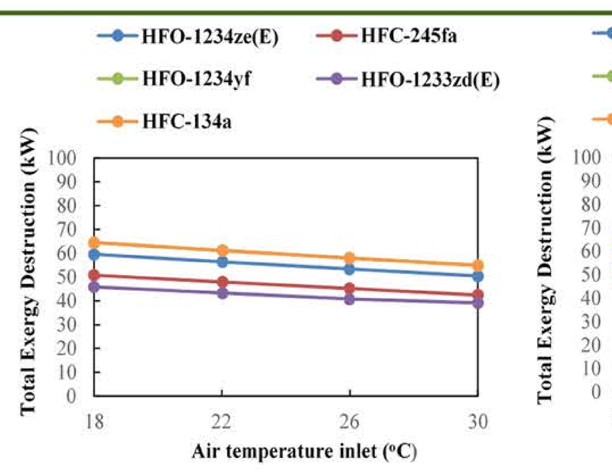

Fig.4: Impact of the heat source temperature on the total exergy destruction (kW) for cooling air.

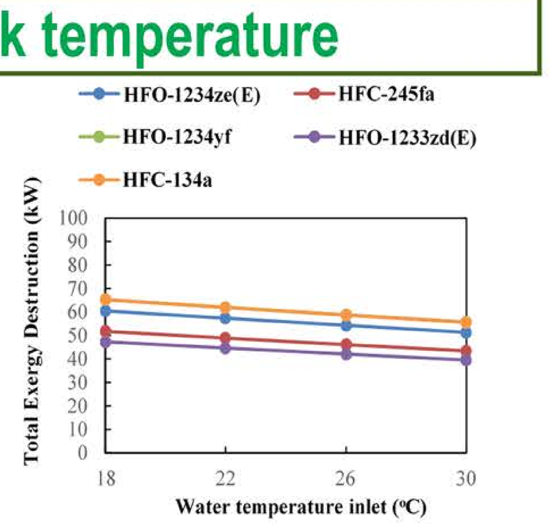

Fig.5: Impact of the heat source temperature on the total exergy destruction (kW) for cooling water.

As the air inlet temperature increases, the heat exchanger surface (m2) gets higher, while the overall heat transfer coefficient increases too for each working fluid.

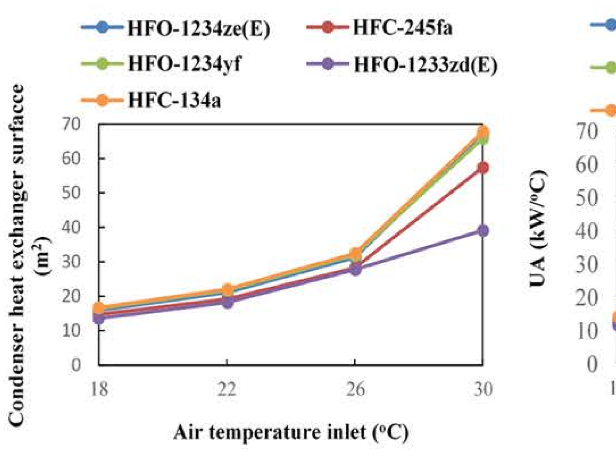

Fig.6: Impact of the cooling air inlet temperature on the condenser surface (m2).

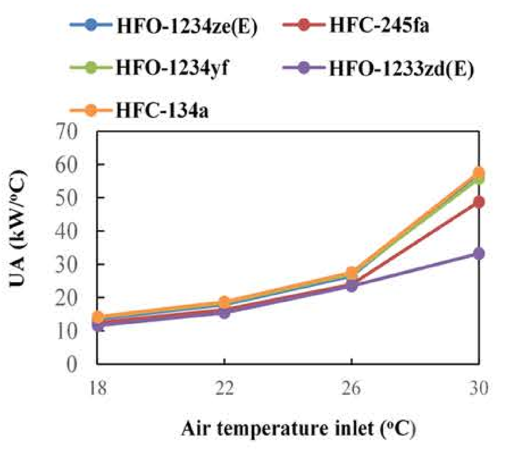

Fig.6: Impact of the cooling air inlet temperature on the overall heat transfer ((kW/ °C).

The simulation model has been developed in Aspen Plus software

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HEAT EXCHANGERS DETAILED DESIGN

Aspen EDR software used for the HEX design & geometry configuration. The design fully meets the following criteria: ASME Design Code, Code Sec VIII Div 1, Normal Service class, R-refinery service TEMA class, ASME Material standard

Heater – Shell & Tube Basic Geometry and Specification

The heater is selected as a shell and tube heat exchanger with vertical baffles since the heater has a two-phase fluid at the outlet and it is proposed for such applications

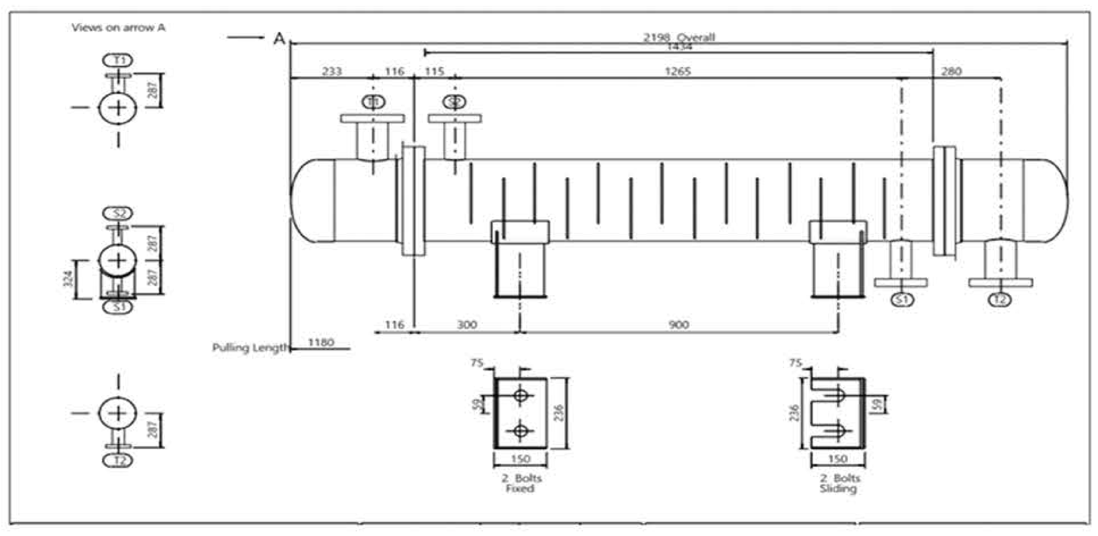

Fig.8: Geometry of the shell and tube heat exchanger designed as heater

HFO-1234ze(E) passes through the shell, while the hot water flows through the tubes side. The horizontal shell and tube heat exchanger type was estimated as a BEM type.

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		

Table 1: Proposed Shell & Tube detailed design data

Condenser – Air Cooler – Basic Geometry and Specification

The air-cooled condenser is selected as a plate-fin heat exchanger. The heat exchanged was calculated 284.6 kW, while the bundle face area of 7.5 m2, which seems to be relatively high due to the high value of 159 kJ/kW of latent heat, based on the thermodynamic cycle conditions.

Air Cooler Basic Geometry			
Bays per unit	1		
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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		

Table 2: Proposed air cooler's basic geometrical data

CONCLUSIONS

- Regardless of the working fluid, increasing the heat source temperature improves the net thermal and the overall exergy efficiency of the TFC cycle
- 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.
- 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 will be used as a valuable basis for the TFC unit manufacture and the respective cost analysis and techno-economic assessment.