Thermodynamic Evaluation of a Combined ORC-TFC Cycle for Low-Temperature Power Generation by using Microsoft Excel

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Abstract-This paper presents a new cycle for utilising the energy of low-temperature heat sources that combines the Organic Rankine Cycle (ORC) with the Trilateral Flash Cycle (TFC) via a cascade condenser. The performance of the combined cycle that allows different working fluids to be used in the two circuits is compared with those of the simple ORC and TFC cycles for a heat-source of 120^{o}C by using R152a as the working fluid in the three cycles. Compared to the simple ORC, the results show that the combined cycle gives more power and a higher thermal efficiency. Compared to the simple TFC, the results show that the cycle gives less power, but a higher thermal efficiency. The cycle's performance with R1234yf in the ORC circuit and R152a in the TFC circuit is also compared with that with R152a in both circuits at the same condenser temperature. The results show that the R1234yf/R152a pair gives more power than R152a alone, but less efficiency. The model developed for the cycle uses Microsoft Excel as the modelling platform and determines the fluid properties with VBA functions. The functions are validated with published data for the TFC model with three suitable organic working fluids which are R134a, R245fa, and R1233zd.

***Keywords*:** Low-temperature heat sources, ORC, TFC, combined cycle, Excel

1. Introduction

The mounting concerns about climate changes caused by the large-scale use of fossil-fuels for electricity generation have inspired intensive research for the development of new technologies for utilising low and moderate temperature energy sources such as solar and geothermal energy and waste heat from industries. One of the promising technologies in this respect is the organic Rankine cycle (ORC) [1,2]. Unlike the conventional Rankine cycle that requires very high temperatures to raise the steam to the required pressure, the ORC can adapt to the low-temperature heat sources by using an organic working fluid instead of steam. Like the conventional cycle, though, the efficiency of the simple ORC deteriorates if the temperature of the heat source is reduced or that of the condenser is increased. Various modifications to simple ORC have been considered to improve its efficiency, such as reheating, regeneration, and recuperation [3]. However, such modifications do not address a main drawback of the cycle which is the mismatch between the temperature of the heating source and that of the working fluid during the heat-addition process in the evaporator since the temperature of the working fluid remains constant while that of the heating source drops [4]. A newly proposed cycle that can solve this problem is the trilateral flash cycle (TFC) in which the working fluid is not allowed to vaporise during the heat recovery process, but directly taken as saturated liquid to expand in a two-phase expander [4,5]. Since the technology of TFC expanders is relatively immature compared to the conventional turbines, they are costlier and less efficient [3,5].

The literature on the ORC and TFC analyses is large and numerous previous researchers developed theoretical models for comparing the performance of the ORC and TFC from thermodynamics and thermoeconomic viewpoints or compared them with other cycles such as the Kalina cycle. For example, Bidgoli and Yanagihara [1] analysed an ORC system for recovering the waste heat from the intercoolers of the compression units of a large processing plant. By using Aspen HYSYS, they analysed the performance of various working fluids including R123, n-butane, n-pentane, hexane, and n-heptane. Their results showed that a net power of up to 40 MW could be generated with R123. Wolf et al. [2] investigated a solar powered ORC that uses pure iso-pentane and two iso-pentane/CO_2 zeotropic mixtures. Modelling of the system was done by using EES and all thermodynamic properties were determined by using REFPROP. By conducting both exergy and exergo-economic analyses of the investigated unit, they showed that the unit was capable of co-producing approximately 30 kW electricity and 160 kW district heating with an exergetic efficiency exceeding 60%. Therefore, they concluded that the unit was able to compete with existing renewable power generating systems in terms of specific cost of electricity.

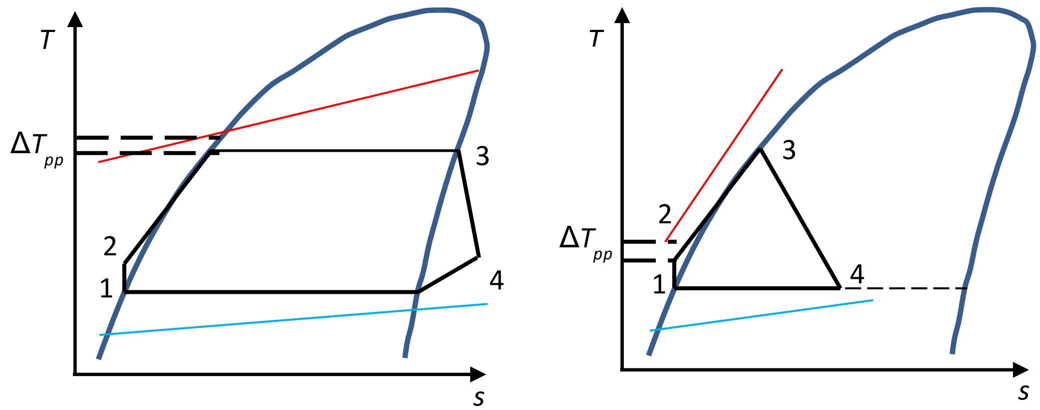
Some studies investigated the combination of ORC and vapour-compression or vapour-absorption refrigeration cycles. For example, Sun et al. [6] investigated the ORC combined with the absorption refrigeration cycle (ARC) and the ejector refrigeration cycle (ERC) to recover the waste heat from the low-temperature flue gas for generating both power and cooling for external uses. Their results with R113 as the ORC working fluid showed that the net power output, the refrigerating capacity and the resultant exergy efficiency of the ORC-ARC combination are all higher than those of the ORC-ERC combination for evaporation temperatures of the basic ORC exceeding 153°C. Toujani et al. [7] combined the ORC with the vapour compression refrigeration (VCR) cycle for cogeneration applications. Three configurations were examined in terms of the net power, refrigeration capacity and overall efficiency. They used n-hexane as working fluids for the ORC and R600 for the VCR cycle. Their results obtained by using the EES software show that, for a hot spring of 1000 kW, the cycle can provide simultaneously, a maximum net work of 17 kW and a maximum net cooling capacity of 160 kW and an overall coefficient of the order of 0.3.

The above literature review and the recent and more comprehensive review conducted by Jiménez-García et al. [8] indicate that a cycle that combines the TFC in the high-temperature circuit (HTC) and the ORC in the low-temperature circuit (LTC) has not been considered before. While minimising the mismatch between the working fluid and the heat source for the ORC circuit, this cycle enables different organic fluids to be used in the HTC and LTC that suit the high and low temperature ranges better than a single fluid and gives cogeneration systems more flexibility than a single ORC or TFC. The present paper contributes to knowledge by presenting a thermodynamic evaluation of this new cycle. From another perspective, the above literature review shows that most researchers used commercial software for their studies. However, the use of general-purpose software can encourage independent researchers and engineering students to contribute to the development of innovative systems for utilising low-temperature heat sources. In this respect, the present paper uses Microsoft Excel as the modelling platform with special VBA functions to determine the thermodynamic properties of the working fluids. The functions are verified by comparing the estimations of the TFC model with published data for three organic working fluids, which are R134a, R1233zd, and R245fa.

2. The ORC, TFC, and the combined cycle

Figure 1 shows the *T-s* diagrams of the simple ORC and TFC. The ORC system has the same components as those of the conventional steam-turbine power plant which are the evaporator (boiler), the turbine, the condenser, and the pump. The difference between the ORC and TFC systems is that the TFC system heats the working fluid without going into the vaporisation process so that the hot pressurised fluid expands in a two-phase expander. Organic fluids are used in the ORC and TFC instead of water because they have higher boiling pressures and, therefore, suit the low-temperature applications better. As Figure 1.b shows, the TFC leads to a uniform temperature glide between the heat source and the working fluid which minimises the losses during the heat-transfer process and improves the performance of the system. According to [3], TFC systems can provide 50% more work than ORC systems for the same energy input, but they need sophisticated expanders to adequately handle the liquid-phase presence during the expansion process.

Figure 2.a shows a schematic diagram of the proposed combined-cycle system in which the heat source heats the working fluid of the TFC circuit first and then heats the working fluid in the ORC circuit. After expanding in the two-phase expander to produce power, the working fluid of the TFC circuit is condensed by the cooler working fluid of the ORC circuit in a cascade condenser. The liquefied fluid is then pumped by the TFC pump back into the heater where it is heated to the saturation temperature. The ORC system shown on Figure 2.a is a recuperative system in which the superheated fluid exiting the turbine is used to heat the cold fluid exiting the pump in an internal heat-exchanger (IHEX). Accordingly, the initially superheated fluid enters the condenser of the ORC as saturated vapour. In the condenser, the working fluid is cooled by the cold-sink fluid to the saturated-liquid state before being pumped into the cold side of the IHEX. After the IHEX, the fluid is heated by the heat source to the state of saturated liquid before entering the cascade condenser where it is heated by the condensing fluid of the TFC circuit until it becomes saturated vapour.



(a) (b)

Fig. 1: *T-s* diagrams of: (a) the simple ORC and (b) the simple TFC

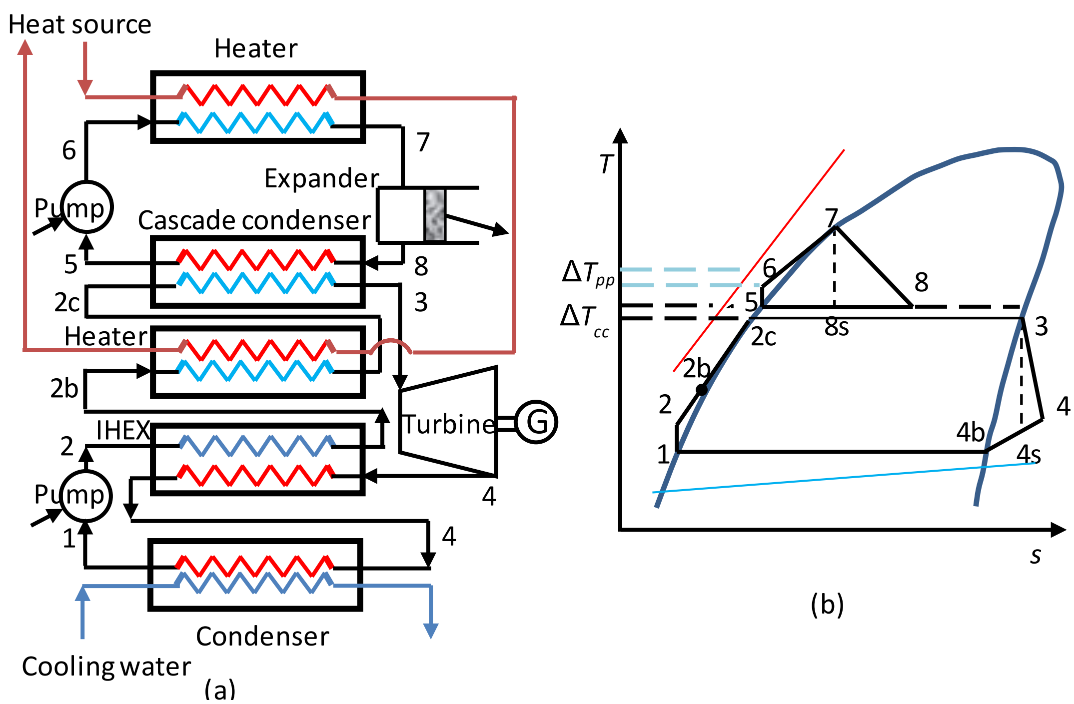
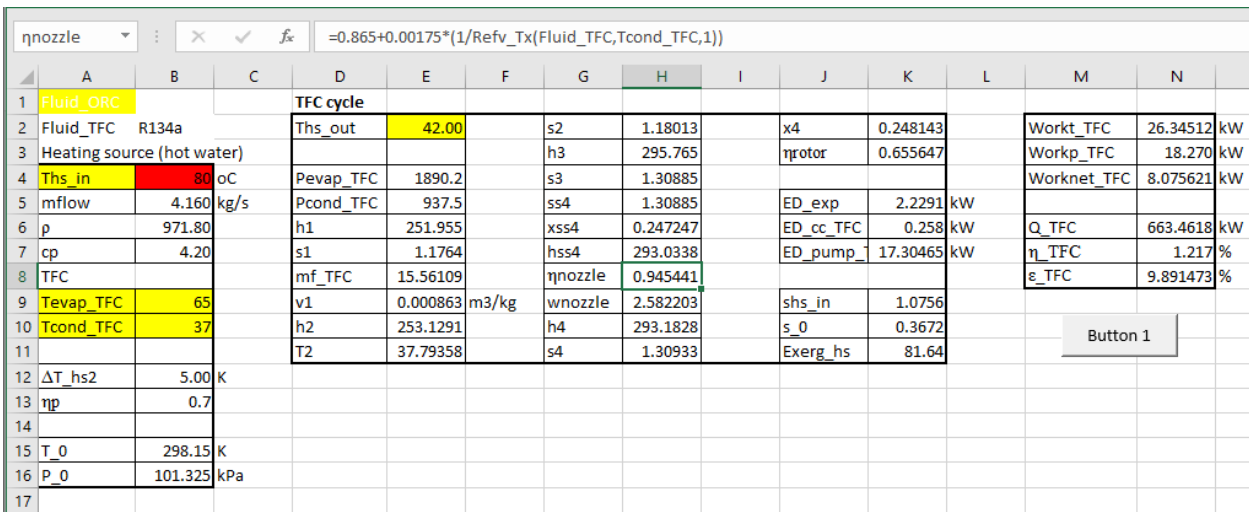


Fig. 2: Schematic and *T-s* diagrams of the combined cycle

Figure 2.b shows the *T-s* diagram of the combined cycle. Unlike the TFC in which the pinch point occurs at the exit point of the heating source, that of the combined cycle occurs in the middle of the heating process which forces the heating source to exit the system at a higher temperature than that of the TFC. Although this limits the amount of energy recovered and the work produced by the ORC circuit, the cycle can still produce more power than a simple ORC because of the additional work produced by the TFC circuit. Compared to the simple TFC, the combined cycle may not be able to produce more power, but it can have a higher thermal efficiency by replacing the two-phase expander with the single-phase turbine and by allowing two different working fluids to be used in the TFC and ORC circuits. Two fluids suit the low-temperature and high-temperature heat-transfer parts better than a single fluid and lead to a better overall performance of the system.

3. Verification of the VBA functions for fluid properties

The Excel-based models developed for the present analyses determine the thermal fluid properties by using a special Excel add-in developed with VBA [9]. The functions related to the present analyses, which are those of the refrigerants group, simply store and interpolate the data given by ASHRAE [10] for saturated liquids and saturated vapours. The specific volume of superheated refrigerants is determined by the Redlich-Kwang equation and the functions that determine their enthalpy and entropy use ideal-gas equations in which the specific heat is determined at an adjusted pressure by multiplying the actual pressure by a "compressibility factor" for which an average value of 0.5 is adopted [11]. Although this factor extends the usefulness of these functions to supercritical conditions, the accuracy of the functions needs to be verified. In what follows, the functions are verified by comparing the results of the model developed for the simple TFC with the data provided by Lai et al. [12] who conducted a thermodynamics analysis of the cycle by using the REFPROP software. Lai et al. [12] fixed the temperature and the mass flow rate of heat source at 80^{o}C and 4.16 kg/s, respectively, the heat sink temperature to 30^{o}C, the condenser inlet temperature to 37^{o}C, and used an isentropic efficiency of 0.7 for the pump. Figure 3 shows the Excel-based model developed for the TFC using their data.

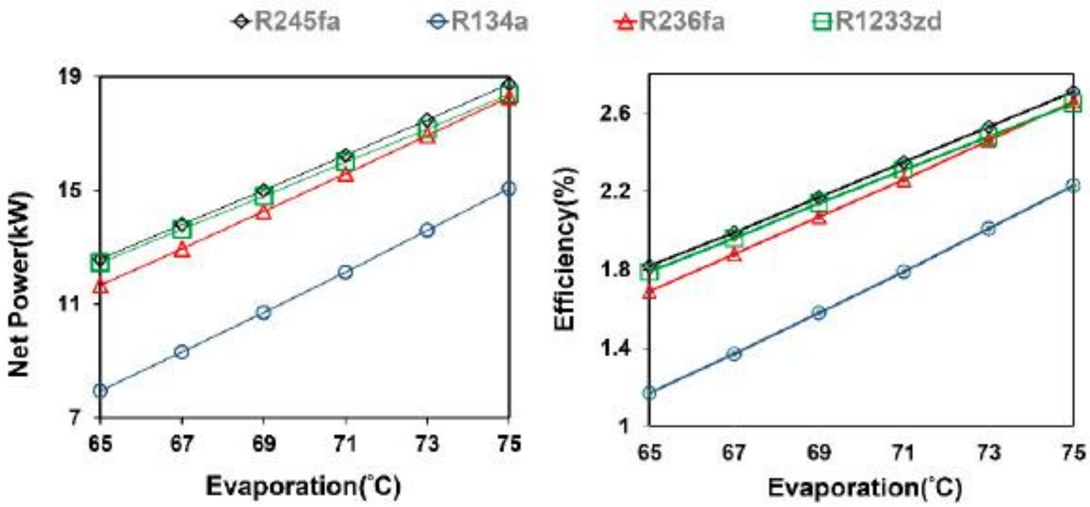


ηnozzle =0.865+0.00175*(1/Refv_Tx(Fluid_TFC,Tcond_TFC,1))

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	
1	Fluid_ORC			TFC cycle											
2	Fluid_TFC	R134a		Ths_out	42.00		s2	1.18013		x4	0.248143		Workt_TFC	26.34512	kW
3	Heating source (hot water)						h3	295.765		ηrotor	0.655647		Workp_TFC	18.270	kW
4	Ths_in	80	oC	Pevap_TFC	1890.2		s3	1.30885					Worknet_TFC	8.075621	kW
5	mflow	4.160	kg/s	Pcond_TFC	937.5		ss4	1.30885		ED_exp	2.2291	kW			
6	ρ	971.80		h1	251.955		xss4	0.247247		ED_cc_TFC	0.258	kW	Q_TFC	663.4618	kW
7	cp	4.20		s1	1.1764		hss4	293.0338		ED_pump_1	17.30465	kW	η_TFC	1.217	%
8	TFC			mf_TFC	15.56109		ηnozzle	0.945441					ε_TFC	9.891473	%
9	Tevap_TFC	65		v1	0.000863	m3/kg	wnozzle	2.582203		shs_in	1.0756				
10	Tcond_TFC	37		h2	253.1291		h4	293.1828		s_0	0.3672		Button 1		
11				T2	37.79358		s4	1.30933		Exerg_hs	81.64				
12	ΔT_hs2	5.00	K												
13	ηp	0.7													
14															
15	T_0	298.15	K												
16	P_0	101.325	kPa												
17															

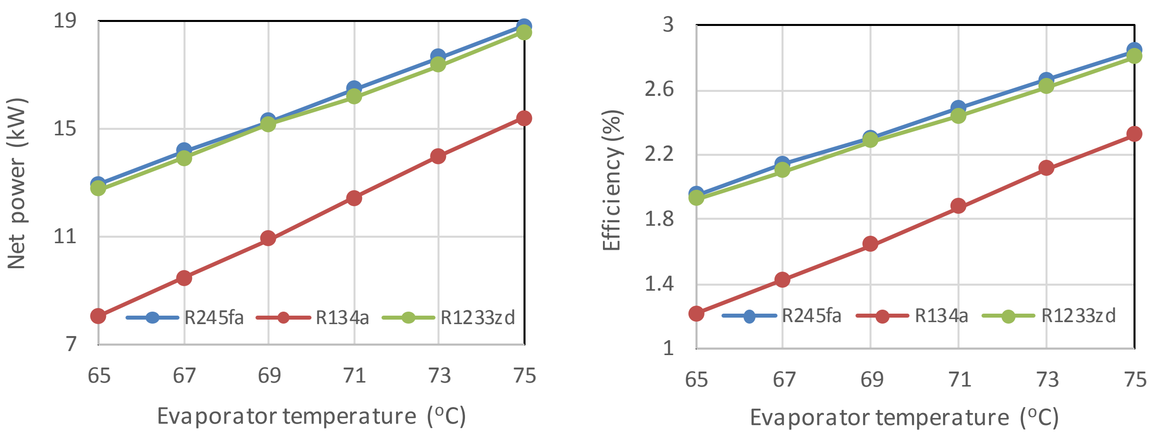
Fig. 3: The Excel-aided model for the TFC using R134a with the data of Lai et al. [12]

The sheet consists of three blocks of cells. The first block on the left side of the sheet stores the specified data, while the second block calculates the various temperatures, pressures, and fluid properties. The third block on the right side of the sheet determines the overall parameters that include the total amount of recovered heat (Q_TFC), the net power produced by the system (Worknet_TFC), and the overall energetic and exergetic efficiencies, η_TFC and ε_TFC, respectively. The model uses R134a as the working fluid, but the name of the fluid is stored as a variable so that the sheet can be used for other fluids. Lai et al. [12] determined the specific work of the expander by following the method described by Trædal [13]. The formula bar in Figure 3 shows the function used to determine the efficiency of the nozzle according to this method. They also fixed the operating temperature at the inlet of the pump (T_2) at 35^{o}C. In the present analysis, T_2 is not fixed but evaluated by the model. For their analysis, Lai et al. [12] selected four working fluids that are commonly used in low-temperature power generation systems, which are R134a, R236fa, R245fa, and R1233zd. Since R236fa is not supported by present add-in, comparison is made with their results for three fluids which are R134a, R245fa, and R1233zd. Figure 4 and Figure 5 compare the results obtained by Lai et al. [12] and by the present model. According to both models, R245fa and R1233zd have comparable power and thermal efficiency, but R134a has considerably lower values. In general, the figures show a close agreement between the estimations of the two models.



(a) (b)

Fig. 4: Variations of the net power and thermal efficiency of the TFC with the evaporation temperature as given by Lai et al. [12]



(a) (b)

Fig. 5: Variations of the net power and thermal efficiency of the TFC with the evaporation temperature as obtained by the present model

4. Comparison of the combined cycle with the ORC and TFC

Two other models for the simple ORC and for the combined cycle have been developed in order to compare the performance of the combined cycle to those of the ORC and TFC. Figure 6 shows the model developed for the combined cycle in which the first block of cells on the left side stores the specified data, while the second and third blocks in the middle perform the calculations for the TFC and ORC circuits, respectively. The fourth block on the right side of the sheet determines the overall cycle parameters. The names of the working fluids in the TFC and ORC circuits are stored as variables so that the model can be used for different fluid pairs, but R152a is used in both circuits for the sake of comparison. In this section, the three cycles will be compared on the basis of the data given by Yari et al. [5] for a heat source at 120°C with the data shown on Table 1. The temperature of the TFC fluid entering the expander, T_6, is taken as 110°C and the temperature difference in the cascade-condenser is taken as 3°C. The temperature of the fluid in the TFC circuit in the cascade condenser is specified as 83°C so that the temperature of the fluid in the ORC circuit is 80°C, which is the optimum evaporator temperature for the simple ORC as shown by Yari et al. [5]. Figure 6 shows the results of the combined cycle.

η_overall =W_total/Q_total*100

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O
1	Fluid_TFC	R152a		**TFC cycle**									TFC		
2	Fluid_ORC	R152a		T_hsi	93.00		s6	1.508015					Workt_TFC	656.4599	kW
3	Heating source (hot water)						h7	439.22					Workp_TFC	433.9989	kW
4	T_hs	120	oC	Pevap_TFC	4243.2		s7	1.7058					Worknet_TFC	222.4611	kW
5	mflow	100	kg/s	Pcond_TFC	2503.08		ss8	1.7058		ED_exp	183.3613	kW	Q_TFC	11458.8	kW
6	ρ	943.10		h5	360.266		xss8	0.40023		ED_cc_TFC	13.682	kW	η_TFC	1.941	%
7	cp	4.24		s5	1.50023		hss8	433.4092		ED_pump_TFC	349.6387	kW			
8	TFC			mf_TFC	150.62947								ORC		
9	Tevap_TFC	110		v5	0.0014074	m3/kg				s_hs	1.5279		Q_sensible	91.67203	
10	Tcond_TFC	83		h6	363.1472		h8	434.8619		s_0	0.3672		Q_ORC	5413.38	
11	ΔTcc	3	oC	T6	84.214554		s8	1.709883		Eerg_hs	5711.73		Workt_ORC	1416.248	kW
12	ΔT_hs2	10	K	**ORC cycle**									Workp_ORC	115.811	kW
13				Tevap_ORC	80	oC	s4s	2.0198		T2	41.03783		Worknet_ORC	1300.436	kW
14	ORC			Pevap_ORC	2342.4	kPa	h4s	515.2144		s2	1.247223		η_ORC	7.811	%
15	Tcond_ORC	40	oC	Pcond_ORC	909.27	kPa	h4	519.4468		h2b	261.478				
16	ΔT_sc	0.00	K				T4	40		T2b	34.72412	kJ/kg	Overall parameters		
17	ηt_isen	0.85		h3	543.43		s4	2.033315		s2b	1.209809	kg/s	W_total	1522.897	kW
18	ηp_isen	0.85		s3	2.0198					Qevap_ORC	190.28	kW	Q_total	16872.18	kW
19	ηx_isen	0.75		hsat.liq	353.15		T1	40	oC	mf_ORC	59.05	kW	T_hsout	80.24	oC
20	T_0	298.15	K	ssat.liq	1.481		h1	271.35	kJ/kg	ED_evap	-0.138596	kW	Q_out	15349.28	
21	P_0	101.325	kPa				v1	0.001163	m3/kg	ED_turb	237.9403	kW	ED_total	893.162	
22				hsat.vap	531.28		s1	1.2411		ED_cc_ORC	0.871		η_overall	9.026	%
23				ssat.vap	2.0711		h2	273.3112		ED_pump	107.8061		exg eff	26.66263	%
24															

Fig. 6: Excel-aided model for the combined using R152a with the data shown on Table 1

Table 1: Values of the parameters used for validating the ORC model (Yari et al. [5])

Parameter	Value
T_{hs} [°C]	120
$\dot{m}_{hs}$ [kg/s]	100
ΔT_{pp} [K]	10
η_p (%)	85
η_t [%]	85

Table 2 gathers the results obtained by the three models. The table figures show that the lowest exit temperature for the heat source is that of the TFC, which is 50°C. Both the ORC and the combined cycle produced higher exit temperatures, but the highest exit temperature is that of the combined cycle, which is 80.24°C. Therefore, both the amounts of energy recovered and energy rejected to the cooling water in the condenser by the combined cycle are the lowest among the three cycles. A lower heat input means that there is more energy left in the heat source that can be recovered for cogeneration and a lower rate of heat rejection means that the capacity of the cooling system is reduced. The table figures also show that the combined cycle achieved the highest thermal efficiency of 9.026% compared to 7.466% for the simple TFC and 8.153% for the simple ORC. The power produced by the combined cycle is less than that of the simple TFC by 31.3%, but more than that of the simple ORC by 3.3%.

Table 2: Comparison of the thermodynamic performance of the ORC, TFC, and combined cycle

	TFC	ORC	Combined cycle
Exit temperature of heat source, T_{hsout} (°C)	50	77.41	80.24
Amount of recovered heat, Q_{total}, (kW)	29708	18074.17	16872.18
Amount of rejected heat, Q_{out}, (kW)	27489.92	16600.63	15349.28
Power produced, W_{total}, (kW)	2218.085	1473.536	1522.897
Overall thermal efficiency (%)	7.466	8.153	9.026
Overall exergetic efficiency (%)	38.83386	25.798	26.66263

5. Comparison of the combined cycle with the ORC at various cascade temperatures

The comparison of the three cycles presented in the previous section did not take into consideration an important advantage of the combined cycle, which is the allowance to use different working fluids in the TFC and ORC circuits. This section analyses the performance of the combined cycle with R152a and R1234yf as a pair with the same data of Table 1. The critical temperature of R1234fy is lower than the temperature of the heat source and Yari et al. [5] showed that it can achieve a higher thermal efficiency in the lower-temperature range than R152a. Therefore, it is used in the ORC circuit while R152a is used in the TFC circuit. Figure 7 compares the power and thermal efficiency of the combined cycle using this fluid pair with those of R152a in both circuits and the simple ORC at various temperatures in the cascade condenser.

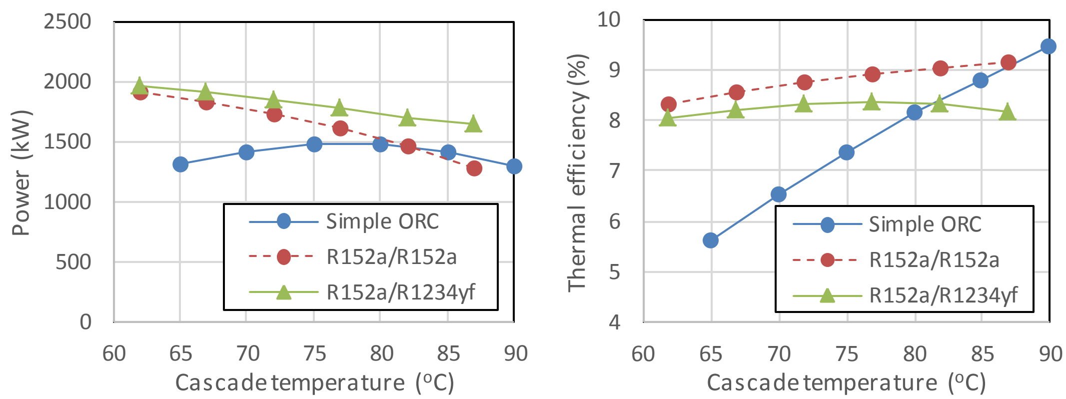


Fig. 7: Comparison of (a) the power and (b) thermal efficiency of the combined cycle using R152a/R1234yf and R152a/R152a with those of the simple ORC

Figure 7.a shows that the power of the combined cycle with both fluid pairs is higher than that of the simple ORC at low values of the cascade temperature, i.e., when the contribution of the TFC circuit is dominant. Although the cycle's power decreases by increasing the cascade temperature, the figure shows that the power of the combined cycle with the R152a/R1234yf pair remains higher than that of the R152a/R152a pair and the simple ORC over the whole range of cascade temperatures. However, Figure 7.b shows that the thermal efficiency of the combined cycle with the R152a/R152a pair remains higher than that of the R152a/R1234yf pair. While the thermal efficiency of the simple ORC increases significantly as the cascade temperature increases, Figure 7.b shows that the efficiency of the combined cycle increases slowly but remains higher than that of the simple ORC for all the temperatures below 80°C, which the optimum evaporator temperature for the simple ORC. Below 80°C, both the power and thermal efficiency of the combined cycle with the two fluid pairs are higher than those of the simple ORC.

6. Conclusion

This paper presents a new cycle for the utilisation of low-temperature heat sources for power generation that combines the organic Rankine cycle (ORC) and the trilateral flash cycle (TFC). The performance of the combined cycle is compared to those of the two basic cycles for a heat source available at 120°C. Two performance parameters are used for comparison which are the power produced by the cycle and the thermal efficiency of the cycle. The results obtained with R152a as the working fluid show that the combined cycle gives more power than the simple ORC and higher thermal efficiency. Compared to the simple TFC, the combined cycle produces less power but achieves a higher thermal efficiency. The results also show that the exit temperature of the heating source for the combined cycle is high enough to be utilised for cogeneration. The models developed for the analyses of the three cycles use Excel as s modelling platform with special VBA functions to determine the thermodynamic properties of the organic working fluid. The Excel-based platform enables the analytical model to be developed in a compact form. By extending the analytical model to include the economic and environmental factors, it can be used for multi-objective optimisation analyses of the cycle [14, 15].

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