

Performance analyses of a combined ORC-TFC cycle with various low-GWP refrigerants by using Excel

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Abstract

This paper presents a combined organic cycle for recovering the energy of low-temperature heat sources that utilises the Trilateral Flash Cycle (TFC) in the high-temperature circuit and the Organic Rankine Cycle (ORC) in the low-temperature circuit with a cascade heat-exchanger in between. Although the combined cycle generally improves the performance of each organic cycle working separately, its actual performance depends on temperature of the heat source and that of the cascade condenser between the two circuits of the cycle as well as the pair of refrigerants used in the two circuits. In this study, the performance of the cycle is initially gauged against that of another combined ORC-TFC cycle with a 300°C heat source using R245fa in the low-temperature circuit and water in the high-temperature circuit. It is then analysed with a 120°C heat-source by using R152a as a single fluid and R1234yf/R152a as a two-fluid pair and the results are compared to those of the simple ORC using R152a. Finally, the cycle's performance is analysed with the 120°C heat-source using six working fluids that have zero ozone-layer depletion potential (ODP) and zero or low global-warming potential (GWP), which are R290, R600a, R717, R152a, R1234yf, and R245fa. Comparison of the power produced, thermal efficiency and exergetic efficiency of the cycle at various temperatures in the cascade condenser shows that the best performance with a single fluid is achieved by R245fa, while R1234yf/R152a and R1234yf/R245fa achieved the best performance as pairs. The model developed for analysing the cycle's performance uses Microsoft Excel with special VBA functions for determining the fluid properties. The VBA functions are first verified by comparing their estimations for the simple ORC and TFC with published data.

Keywords: Organic Rankine cycle, trilateral flash cycle, low-temperature energy utilisation, Microsoft Excel, VBA

1. Introduction

The mounting concerns about the climate changes due to the use of fossil-fuels for electricity generation have inspired intensive research for the development of new technologies that utilise low-grade energy sources such as solar and geothermal energy and waste heat from industries. One of the promising technologies in this respect is the ORC [1,2]. Unlike the conventional Rankine cycle that requires steam at very high temperatures before expanding in the turbine, the ORC can be adapted to heat sources at much lower temperatures by using an organic refrigerant as the working fluid instead of water. However, because of its lower temperature the efficiency of the ORC is also lower and, like the conventional cycle, various improvements to the simple cycle have been considered such as reheating, regeneration, and recuperation [3]. Although such modifications proved to be effective, they do not address an inherent drawback of the ORC which is the mismatch between the temperature of a finite heating source and that of the working fluid during the heat-addition process in the evaporator. While the temperature of the heating source keeps dropping, that of the working fluid remains constant [4]. The TFC has been proposed as a solution to this problem. In this cycle, the working fluid is not allowed to vaporise by the heat recovery process but taken as saturated liquid to expand in a two-phase expander [4-8]. The problem with the TFC is that the technology of expanders is relatively immature and, therefore, they are generally found less efficient compared to the conventional ORC. The two-phase expander is generally seen as the main factor for degrading the cycle's efficiency [3-6].

Many previous studies analysed the performance of the ORC or the TFC for utilising low-grade energy or compared them with other cycles by using theoretical models. The following brief review of the literature gives examples of these studies with a focus on four related aspects which are: (i) the type and temperature of the heat source, (ii) the working fluids used, (iii) the software used for developing the analytical models, and (iv) the main findings of the study. Wolf et al. [1] investigated a solar powered ORC using pure iso-pentane and two iso-pentane/CO₂ zeotropic mixtures. Modelling of the system was done by using the Engineering Equation Solver (EES) software and the thermodynamic properties were determined by using REFPROP. Their exergy and exergo-economic analyses show that the investigated unit is 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 is able to compete with existing renewable power generating systems in terms of specific cost of electricity. Bidgoli and Yanagihara [2] 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 compared the performance of four working fluids including R123, n-butane, n-pentane, hexane, and n-heptane. Their results show that a net power of up to 40 MW could be generated with R123.

Yari et al. [5] considered a low-grade heat source with a temperature of 120°C and conducted parametric studies of the ORC and TFC cycles with several working fluids by using the EES software. Their results show that increasing the inlet temperature of the TFC expander increases the net output power and decreases the product cost, but this was not the case for the ORC system. Lai et al. [6] compared the ORC and the

TFC feeding with the same low-temperature (80-100°C) heat source. In the ORC a small fraction of the supplied waste heat was continuously provided to the expander with the aim of approaching a quasi-isothermal process instead of the adiabatic one. They performed parametric thermodynamic analyses of the two cycles, with R1234ze(E), R1234yf, R1233zd(E), and R134a as working fluids by using Aspen Plus software. Their techno-economic study of the ORC with the four working fluids show that R1233zd(E) is the most proper working fluid for the cycle.

Ketfi et al [8] conducted thermodynamic-based comparison of ORC, TFC and organic flash cycle (OFC) systems for waste heat recovery from a marine diesel engine at 80°C. They selected working fluids with favourable thermo-physical properties, namely n-butane (R600), i-butane (R600a), n-pentane (R601), i-pentane (R601a), and toluene and developed their program by using EES. Their results indicate that the ORC system achieves a higher thermal efficiency compared to the TFC and OFC systems. However, the TFC system exhibits the advantage of a lower specific volume of the working fluid at the end of the heat addition process (expander inlet) since it remains in a liquid state. This characteristic allows for the use of smaller-sized expanders, making the TFC system particularly appealing for marine applications. Furthermore, it was observed that all the selected working fluids outperform R245fa in terms of power generation within the ORC system. In the case of the TFC and OFC systems, only R601, R601a, and toluene surpass the performance of R245fa.

For better utilisation of the available energy sources, the previous researchers also considered two-stage cascaded ORC systems [9,10], ORC systems combined with vapour-compression refrigeration [11], and combined and integrated systems [12-14]. Yu et al [15] and Li et al [16] considered recovering the waste energy from the exhaust gas of a marine engine. To obtain a better matching between the heat source and the working fluid, they proposed a combined TLC-ORC system in which the hot gas initially heats the working fluid of the TFC in its evaporator (evaporator 1). The working fluid of the ORC is initially heated by the condensing fluid of the TFC in a cascade condenser (condenser 1) before being heated further in the evaporator of the ORC (evaporator 2) until the saturated vapour state by the hot source gas leaving evaporator 1. By using Cyclohexane, Toluene, Benzene and water for the TFC circuit and R245fa for the ORC circuit Yu et al [15] studied the effects of the evaporating temperature for the high temperature (HT) and low-temperature (LT) cycles and compared their proposed TLC-ORC system with the conventional dual-loop ORC system. Their results obtained by using the REFPROP software show that the maximum net power output, thermal efficiency, and exergy efficiency of the conventional dual-loop ORC are 8.8 kW, 18.7%, and 50.0%, respectively, obtained by using cyclohexane as HT working fluid at $T_{HT,evap} = 470\text{K}$ and $T_{LT,evap} = 343\text{K}$. While for the T_{LC-ORC} , the corresponding values are 11.8 kW, 25.0%, and 65.6%, obtained by using toluene as a HT working fluid at $T_{HT,evap} = 470\text{K}$ and $T_{LT,evap} = 343\text{K}$, which are 34.1%, 33.7%, and 31.2% higher than that of a conventional dual-loop ORC.

The brief literature review given above and the more comprehensive review given in [17] show that most previous studies dealt with the ORC or the TFC separately, but the cycle that combines the two organic cycles has received little attention. Apart from minimising the mismatch between the working fluid and the heat source, the

combined ORC-TFC cycle enables different organic fluids to be used that suit the high and low temperature ranges better than a single fluid. In this regard, the present paper contributes to knowledge by analysing the performance of a new combined ORC-TFC cycle [18] at two levels of the heat-source temperature; 300°C and 120°C, using a number of natural fluids and synthetic refrigerants with zero ODP and low GWP. From another perspective, the literature review shows that most researchers used commercial software for their analyses. However, such software may not be available to independent researchers and engineering students particularly in developing countries [19, 20]. In this respect, the present study shows how Microsoft Excel can be used as the modelling platform by determining the fluids thermodynamic properties with special VBA functions. The paper initially confirms the accuracy of the VBA functions for the thermodynamic analyses with the data provided by Yari et al. [5], Lai et al. [6] and Yu et al [15].

2. Analytical models for the ORC and TFC

Figure 1 shows schematic diagrams of the simple ORC and TFC systems, while Figure 2 shows their corresponding T - s diagrams.

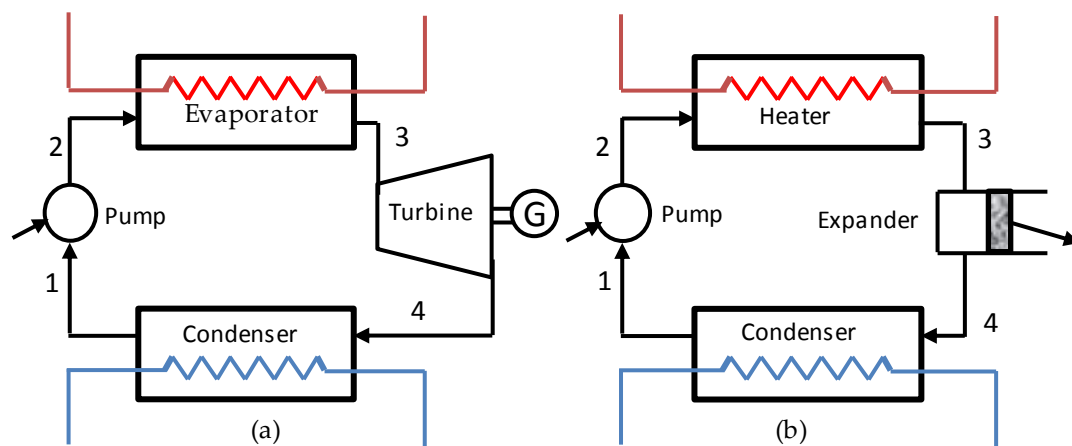


Figure 1: Schematic diagrams of: (a) the ORC system and (b) the TFC system

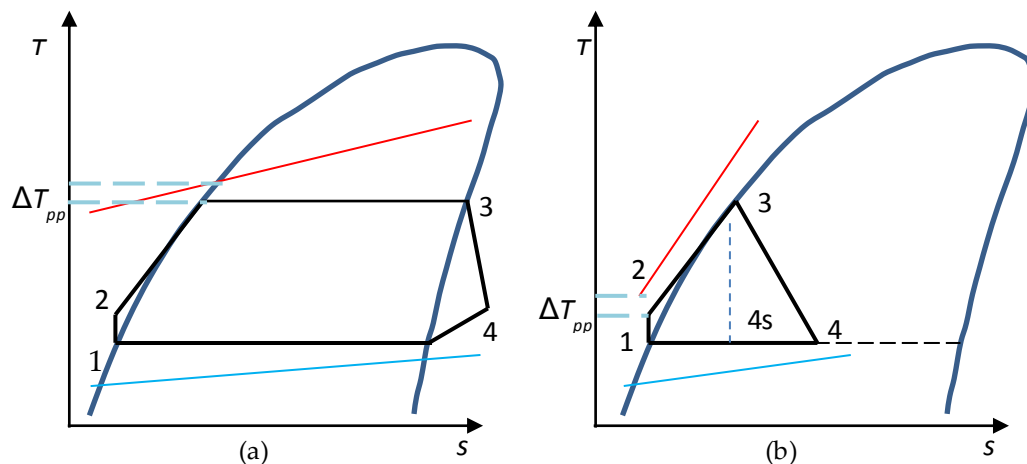


Figure 2: T - s diagrams of: (a) the ORC cycle and (b) the TFC cycle

As Figure 1.a shows, the ORC system has the same components as those of the conventional Rankine cycle which are the evaporator (boiler), the turbine, the condenser, and the pump. To avoid the problem of blade erosion at the last turbine stages, dry or isentropic fluids are usually selected for which the saturated liquid-vapour curves have negative or infinite slopes, respectively [3,14]. The TFC system is different from the ORC system in that it heats the working fluid without vaporising it so that the hot pressurised fluid expands in the two-phase region. As Figure 2.b shows, this cycle leads to a uniform temperature glide between the heat source and the working fluid; which reduces the losses during the heat-transfer process and improves the system's performance. Typically, TFC systems can provide 50% more work than ORC systems for the same energy input. However, they need sophisticated expanders to handle the liquid-phase presence during the expansion process [3]. Apart from increasing the investment cost, two-phase expanders are generally less efficient than conventional vapour turbines.

2.1. The analytical model for the TFC

The analytical models applied for the present thermodynamic analyses assume steady-state operation and neglect pressure losses and heat-transfer losses in the various components. However, the losses due to irreversibility in the pumps, the turbine, and the two-phase expander are taken into consideration via the relevant isentropic efficiencies [21]. Referring to Figure 2.b, the TFC pump specific work, w_p , is given by:

$$w_p = v_1(p_2 - p_1)/\eta_p \quad (1)$$

Where, p_1 and p_2 are the pressures of the TFC fluid in the condenser and heater, respectively, and η_p is the isentropic efficiency of the pump. Equation (2) is then used to determine the enthalpy of the fluid at state 2 after the pump:

$$h_2 = h_1 + w_p \quad (2)$$

The temperature T_2 is determined from h_2 and the temperature of the heat source exiting the heater of the TFC, T_{hso} , is determined from:

$$T_{hso} = T_2 + \Delta T_{pp} \quad (3)$$

Where, ΔT_{pp} , is the specified pinch-point temperature difference. Given the inlet temperature of the heat source, T_{hsi} , and its mass flow rate, $\dot{m}_{hs}$, the rate of heat transfer from the heating fluid to the working fluid of the TFC, $\dot{Q}_{hs,TFC}$, and the mass flow rate of the working fluid, $\dot{m}_{TFC}$, are given by:

$$\dot{Q}_{hs,TFC} = \dot{m}_{hs} c_p (T_{hsi} - T_{hso}) \quad (4)$$

$$\dot{m}_{TFC} = \dot{Q}_{hs,TFC} / (h_3 - h_2) \quad (5)$$

Where, c_p is the specific heat of the heating medium. The specific work in the TFC expander, w_{exp} , is evaluated from the enthalpy change as follows:

$$w_{exp} = (h_3 - h_{4s}) \times \eta_{exp} \quad (6)$$

Where, η_{exp} is the isentropic efficiency of the expander and h_{4s} is the enthalpy of the fluid after an isentropic expansion. The thermal efficiency of the TFC is given by:

$$\eta_{TFC} = \dot{m}_{TFC} (w_{exp} - w_p) / \dot{Q}_{hs,TFC} \quad (7)$$

2.2. The analytical model for the ORC

Referring to Figure 1b, the pump specific work is given by:

$$w_p = v_1 (p_2 - p_1) / \eta_p \quad (8)$$

Where, η_p is the isentropic efficiency of the ORC pump. The enthalpy h_4 is determined by taking into consideration the irreversibility of the turbine as follows:

$$h_4 = h_3 + (h_3 - h_{4s}) \times \eta_t \quad (9)$$

Where h_{4s} is the enthalpy after an isentropic expansion and η_t is the isentropic efficiency of the turbine. The turbine's specific work, w_t , is then calculated from:

$$w_t = h_3 - h_4 \quad (10)$$

The thermal efficiency of the ORC is given by:

$$\eta_{ORC} = \dot{m}_{ORC} (w_t - w_p) / \dot{Q}_{hs,ORC} \quad (11)$$

Where, $\dot{Q}_{hs,ORC}$ is the heat recovered in the ORC, which is given by:

$$\dot{Q}_{hs,ORC} = \dot{m}_{hs} c_p (T_{hsi} - T_{hso}) \quad (12)$$

3. Development and validation of the Excel models

The Excel-based models developed for analysing the organic cycles in this study determine the thermodynamic properties of the fluids by using the special VBA user-defined functions described in [22] that determine the properties of saturated liquids and saturated vapours by interpolating the data given by ASHRAE [23]. To determine the enthalpy and entropy of superheated refrigerants the functions use ideal-gas equations in which the specific heat is determined at an adjusted pressure calculated by multiplying the actual pressure with a "compressibility factor". El-Awad et al. [24] assessed the functions' accuracy by comparing their results with those obtained by Atalay and Conan [25] who analysed a simple VCR system using REFPROP for two ozone-friendly refrigerants, R410A and R1234yf, and two natural refrigerants, R290 and, R744. El-Awad et al. [24] also analysed a cascade VCR system with R507A in the high-temperature circuit and R23 in the low-temperature circuit and compared their results with those obtained by Parekh and Tailor [26] who used EES. This section describes the two models developed for the ORC and TFC and validates their results with relevant published data.

3.1. The ORC model

Yari et al. [5] analysed the simple ORC with a heat source at 120°C using the data shown on Table 1. Their results for seven working fluids, which are: R134a, R1234yf, R152a, propane (R290), n-butane (R600), iso-butane (R600a), and ammonia (R717),

were used to verify the VBA property functions for this cycle. Figure 4 shows the Excel model developed for the ORC by using their data.

Table 1: Values of the parameters used for the simple ORC model [5]

Parameter	Value
T_{hs} ($^{\circ}\text{C}$)	120
$\dot{m}_{hs}$ (kg/s)	100
T_{cond} ($^{\circ}\text{C}$)	40
ΔT_{pp} (K)	10
η_p (%)	85
η_t (%)	85

Pevap : ✕ ✓ f_x =RefPsat_T(Fluid,Tevap)												
	A	B	C	D	E	F	G	H	I	J	K	L
1	Fluid	R152a										
2	Heating source (hot water)		Pevap	2342.4	kPa	s4s	2.0198			Q_evap	190.28	kJ/kg
3	Ths_in	120	oC			h4s	515.2144			Q_sensible	79.83881	kJ/kg
4	mflow	100	kg/s	P_cond	909.27	kPa	h4	519.4468		Q_total	270.12	kJ/kg
5	p	943.10					x4	0.954475		Q_out	16600.63	kW
6	cp	4.24		h3	543.43					Qhs_tot	18074.17	kW
7				s3	2.0198		T1	40		Work_t	1604.763	kW
8	Tevap	80	oC				h1	271.35		Work_P	131.227	kW
9	T_cond	40	oC	hsat.liq	353.15		v1	0.001163	m3/kg	Work_net	1473.536	kW
10	ΔT_hs2	10.00	K				h2	273.3112		Ths_out	77.41	oC
11				Ths_cc	90.00					therm eff	8.153	%
12	ηt_isen	0.85					s_hs	1.5279		exg eff	25.798	%
13	ηp_isen	0.85		mflowref	66.91	kg/s	s_0	0.3672				
14							Eerg_hs	5711.73				
15	T_0	298.15	K									
16	P_0	101.325	kPa									
17												

Figure 4: The Excel model for the simple ORC with the data given by [5]

The ORC model consists of four blocks of cells. The first block on the left side of the sheet stores the specified data, while the second and third blocks in the middle perform the cycle's intermediate calculations. As the formula bar shows, the sheet uses cell-naming. The fourth block on the right side of the sheet determines the overall cycle parameters that include the total amount of recovered heat (Q_{total}), the net power produced by the system ($Work_{net}$), the exit temperature of the heating source (Ths_{out}), and the cycle's energetic and exergetic efficiencies ($therm_{eff}$, exg_{eff}). The sheet uses R152a as the working fluid, but the name of the fluid is stored as a variable so that it can be used for other fluids. The results of Yari et al. [5] showed that the ORC has a certain temperature in its evaporator that maximises its output power, which is 80°C . Therefore, this temperature is used in the present model as shown on Figure 4. By using the model, the system's power output, thermal efficiency, and exergetic efficiency were calculated at various values of the turbine's inlet temperature, T_3 , which is the temperature of the working fluid in the evaporator. The results obtained by the present model for the seven fluids are compared on Figures 5 and 6 with those obtained by Yari et al [5] who developed their model by using the EES software. The two figures show close agreement between the results of the present model and those obtained by Yari et al [5].

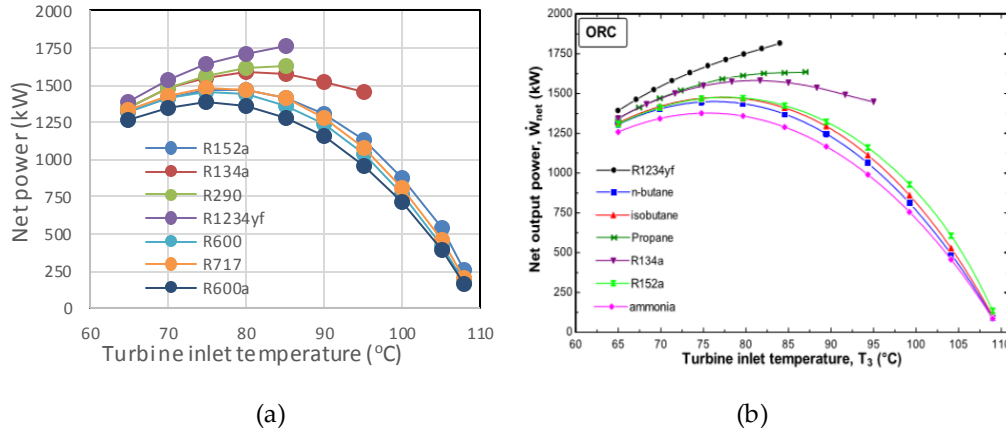


Figure 5: Comparison of (a) the model's estimations for the net power of the ORC at various values of T_3 with (b) the reference [5]

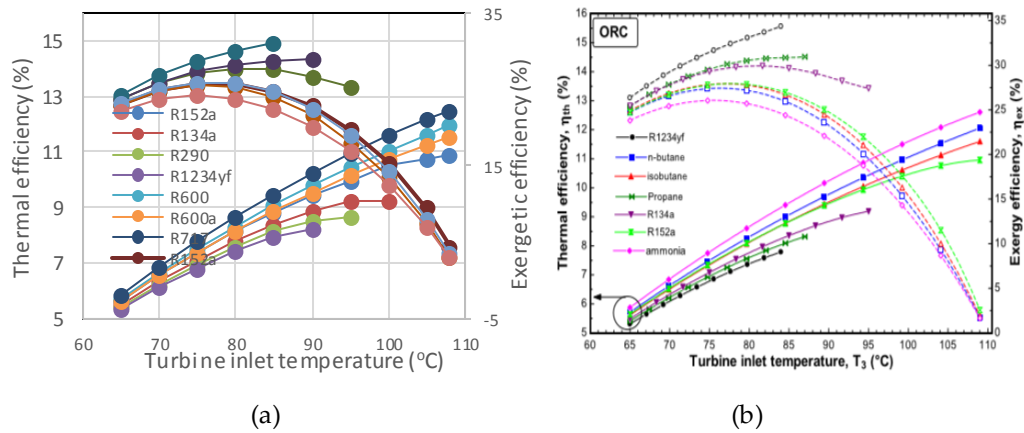


Figure 6: Comparison of (a) the model's estimations for the thermal efficiency and exergetic efficiency of the ORC at various values of T_3 with (b) the reference [5]

3.2. The TFC model

Figure 7 shows the model developed for the TFC using the data provided by Lai et al. [6] who analysed the cycle with a heat source at 80°C. In their analysis, the mass flow rate of the heat source is 4.16 kg/s, the heat sink temperature is maintained at 30°C, the fluid's inlet temperature to the condenser is 37°C, and the isentropic efficiency of the pump is 0.7.

$\eta_{\text{nozzle}} = 0.865 + 0.00175 * (1 / \text{Ref}_v_T_x(\text{Fluid_TFC}, \text{Tcond_TFC}, 1))$															
1	Fluid_ORC		TFC cycle												
2	Fluid_TFC	R134a	Ths_out	42.00		s2	1.18013		x4	0.248143		Work_TFC	26.34512	kW	
3	Heating source (hot water)		h3	295.765		h3	295.765		ηrotor	0.655647		Workp_TFC	18.270	kW	
4	Ths_in	80	Pevap_TFC	1890.2		s3	1.30885					Worknet_TFC	8.075621	kW	
5	mflow	4.160	Pcond_TFC	937.5		ss4	1.30885		ED_exp	2.2291	kW				
6	p	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	17.30465	kW		η_TFC	1.217	%
8	TFC		mf_TFC	15.56109		ηnozzle	0.945441						e_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					
11			T2	37.79358		s4	1.30933		Exerg_hs	81.64					
12	ΔT_hs2	5.00													
13	ηp	0.7													
14															
15	T_0	298.15													
16	P_0	101.325													
17															

Figure 7: The Excel-aided model for the TFC using R152a with the data of Lai et al. [6]

Lai et al. [6] fixed the operating temperature at the inlet of the pump (T_2) at 35°C. The model shown on Figure 7 uses their data, but T_2 is evaluated by the model and not fixed. For their analyses, Lai et al. [6] selected four working fluids which are R134a, R236fa, R245fa, and R1233zd and evaluated the fluid properties by using the REFPROP software. Their results are shown on Figure 8. The comparison was made with three fluids, which are R134a, R245fa, and R1233zd. Figure 9 that shows the results obtained by the present model also shows a close agreement with the results of the reference model. While R245fa and R1233zd have high and comparable values of the power and thermal efficiency, R134a has considerably lower values.

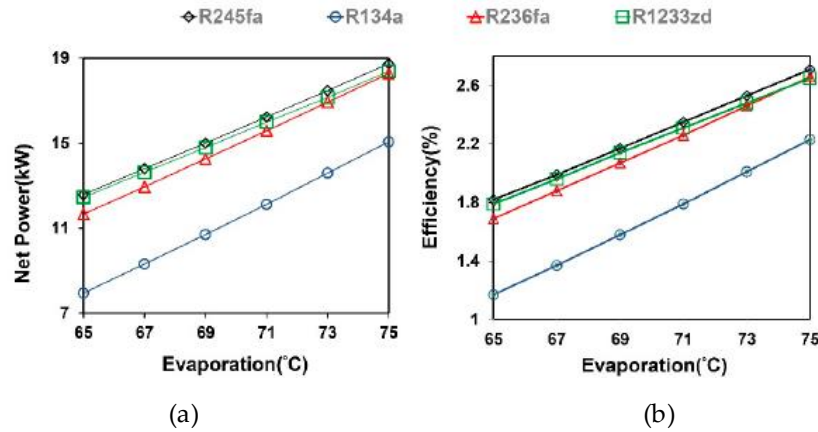


Figure 8: Variations of the net power and thermal efficiency of the TFC with the evaporation temperature as given by Lai et al. [6]

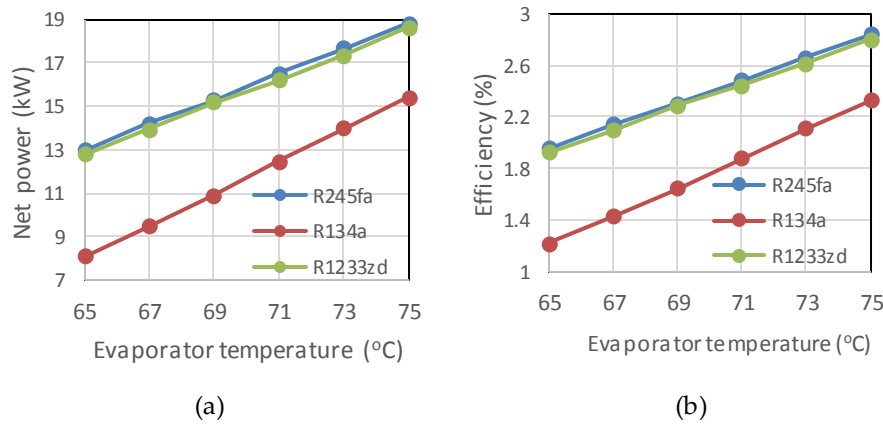


Figure 9: Variations of the net power and thermal efficiency of the TFC with the evaporation temperature as obtained by the present model

4. The analytical model for the combined ORC-TFC cycle

Figure 10.a shows a schematic diagram of the combined-cycle system in which the heat source is first used to heat the working fluid of the TFC circuit before heating the working fluid of the ORC circuit which then enters the cascade condenser. 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 the cascade condenser before being pumped into the TFC heater. After expanding in the turbine, the working fluid of the ORC circuit is condensed by the cooling medium. Figure 10.b shows the T - s diagram of the combined cycle.

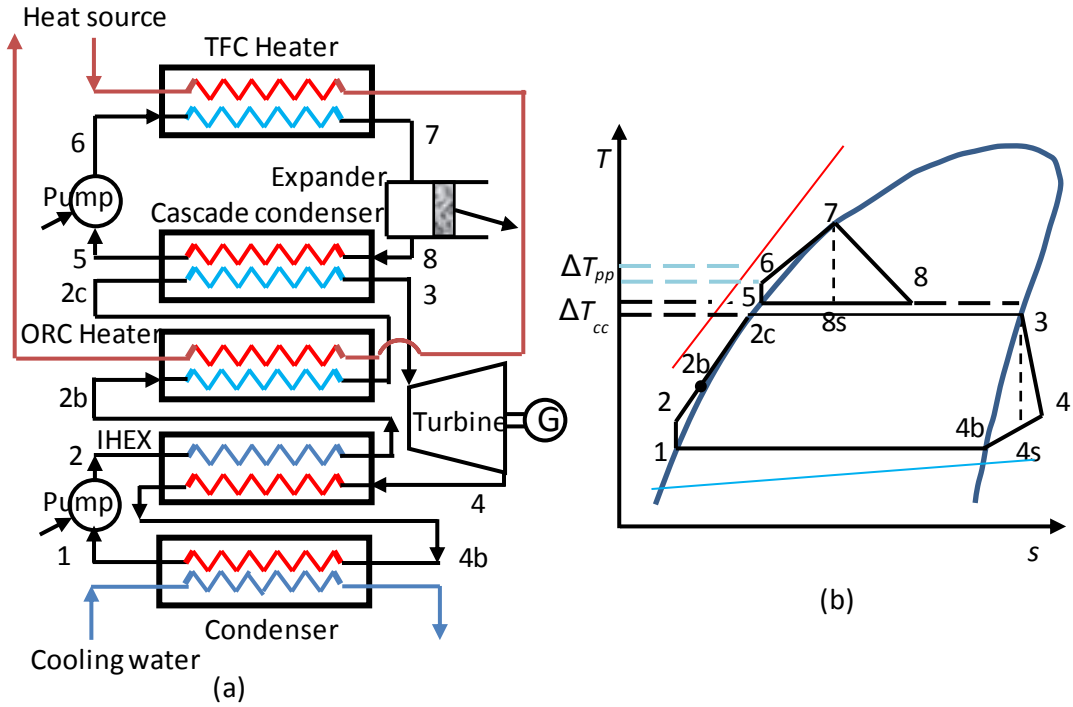


Figure 10: Schematic and T-s diagrams for the combined ORC-TFC cycle

In the cycle described by Yu et al. [15], the ORC fluid is initially heated by that of the TFC in the cascade condenser before receiving more heat from the heat source at the high-temperature end. Unlike the present cycle, the temperature of the ORC fluid in their cascade condenser also varies while that of the TFC fluid remains constant. Therefore, the problem with the temperature difference is only partially removed in their cycle. Moreover, the present ORC is a recuperative one in which the superheated fluid exiting the turbine is used to heat the cold fluid exiting the pump in an internal heat-exchanger (IHEx). After being cooled in the ORC condenser to the saturated-liquid state, the working fluid is 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 to become saturated vapour.

Unlike the simple TFC in which the pinch point occurs where the heating fluid exits the system that of the combined cycle occurs in the middle of the heating process. Although this forces the heating source to exit the system at a higher temperature than that of the simple TFC and limits the amount of heat recovered and the work produced by the ORC circuit, the combined cycle can still produce more work than the simple ORC alone because of the additional work produced by the TFC circuit. Compared to a simple TFC, the combined cycle may not be able to produce the same 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 that suit the two temperature-ranges better.

Referring to Figure 10, the mass flow rates of the two working fluids are related by:

$$\dot{m}_{ORC} = \dot{m}_{TFC} (h_8 - h_5) / (h_3 - h_{2c}) \quad (13)$$

The energy received from the heat source in the ORC circuit is given by:

$$\dot{Q}_{hs,ORC} = \dot{m}_{ORC} (h_{2c} - h_{2b}) \quad (14)$$

Where, h_{2b} is the specific energy of the ORC fluid after the IHEX. Energy balance over the IHEX gives:

$$h_{2b} = h_2 + (h_4 - h_{4b}) \quad (15)$$

Where, h_{4b} is the enthalpy of the saturated vapour at the condenser pressure. The total heat recovered from the heat source, the total net power produced, the overall thermal efficiency, and the second-law (exergetic) efficiency of the combined cycle, respectively, are given by:

$$\dot{Q}_{hs,tot} = \dot{Q}_{hs,TFC} + \dot{Q}_{hs,ORC} \quad (16)$$

$$\dot{W}_{tot} = \dot{m}_{ORC} (w_t - w_{p,ORC}) + \dot{m}_{TFC} (w_{exp} - w_{p,TFC}) \quad (17)$$

$$\eta_{tot} = \dot{W}_{tot} / \dot{Q}_{hs,tot} \quad (18)$$

$$\varepsilon_{tot} = \dot{W}_{tot} / \dot{E}_{hs,in} \quad (19)$$

Where, $\dot{E}_{hs,in}$ in Equation (19) is the rate of exergy flow of the heat source entering the system. Following the definition adopted by Yari et al. [5]:

$$\dot{E}_{hs,in} = \dot{m}_{hs} [(h_{hs,in} - h_0) - T_0 (s_{hs,in} - s_0)] \quad (20)$$

The temperature of the heat source exiting the system is given by:

$$T_{hs,out} = T_{hs,in} - \dot{Q}_{hs,tot} / \dot{m}_{hs} c_p \quad (21)$$

5. Gauging the performance of the combined ORC-TFC cycle with $T_{source} = 300^\circ\text{C}$

The performance of the combined cycle is first gauged by comparison with that of the cycle described by Yu et al [15] who considered an exhaust gas at 573K (300°C). The model's input data given by Yu et al [15] are shown on Table 2.

Table 2: Input data for the combined-cycle analysis [15]

Parameters	Value and Unit
The inlet temperature of the exhaust	573 K (300°C)
The outlet temperature of the exhaust	383 K (110°C)
Mass flow rate of exhaust	804 kg/h (0.223 kg/s)
Condensing temperature of HT working fluid	358 K (85°C)
Condensing temperature of LT working fluid	308 K (35°C)
Evaporator pinch point temperature difference	30 K
Condenser pinch point temperature difference	10 K
Expander isentropic efficiency	0.85
Pump isentropic efficiency	0.65

Because of the relatively high temperature of the gas, Yu et al [15] used cyclohexane, toluene, benzene, octane, and water for the high-temperature TFC circuit and R245fa for the low-temperature ORC circuit. They studied the effects of the TFC evaporator temperature (T_{Evap_TFC}) on the performance of the combined cycle. Since the present group of VBA functions do not cover cyclohexane, toluene, benzene, or octane, only water will be considered as the working fluid for the TFC circuit in the following analysis. In order to avoid the condensation of water vapour in the exhaust gas, Yu et al [15] imposed a lower limit on the exhaust-gas temperature which is 110°C. However, this condition is difficult to apply with the present cycle since the energy required for the sensible heating of the LT fluid comes from the gas not the HT fluid. It should be mentioned that this lower limit won't be required when considering other low-grade energy sources with similar temperatures such as geothermal energy and industrial waste. Needless to say that excluding energy sources with temperatures below 110°C makes the cycle unable to utilise the largest part of the low-grade energy sources which are available at 120°C and below. For these reasons, no lower limit was imposed on the exhaust-gas temperature by the present model.

Figure 11 shows the Excel sheet developed for the combined-cycle using the data of Yu et al [15]. The figure shows four blocks of cells the first of which on the left side of the sheet 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. R718 is used in the TFC circuit and R245fa in the ORC circuit, but the names of the working fluids in the two circuits are treated as variables so that the model can be used for different fluid pairs. The temperature of the TFC fluid in the evaporator, T_{evap_TFC} , is specified as 257°C, and its temperature in the cascade condenser, T_{cc_TFC} , as 80°C. With a temperature difference in the cascade-condenser of 10°C, the temperature of the ORC fluid in the cc, T_{cc_ORC} , is 70°C. The temperature of the ORC condenser, T_{cond_ORC} , is specified as 35°C. With a mass flow rate of the gas of 0.223 kg/s, the figure shows that the net power produced by the system, thermal efficiency and exergetic efficiency are 10.36 kW, 20.22%, and 58.33%, respectively.

Tcc_ORC		fx =Tcc_TFC-ΔTcc													
A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	
1	Working fluids		TFC cycle									TFC			
2	Fluid_TFC	R718	T_hsi	110.00		s6	1.91248		h8	987.65666		Workt_TFC	7.223849	kW	
3	Fluid_ORC	R245fa				h7	1120.122		s8	2.9236736		Workp_TFC	0.3819	kW	
4	Heating source (exhaust gs)		Pevap_TFC	4470.66		s7	2.85748					Worknet_TFC	6.841977	kW	
5	T_hs	300	Pcc_TFC	47.41								Q_TFC	42.43333	kW	
6	mflow	0.223	h5	335.01		ss8	2.85748					η_TFC	16.124	%	
7	p	1.20	s5	1.0756		xss8	0.272646								
8	cp	1.00	mf_TFC	0.0545339		hss8	964.2804		Δh_hs	236.38		ORC			
9	TFC		v5	0.001029	m3/kg				Δs0_hs	0.53		Q_sensible	39.7424	kJ/kg	
10	Tevap_TFC	257	h6	342.0125					Exerg_hs	17.77		Q_ORC	8.817941	kW	
11	Tcc_TFC	80	T6	157								Workt_ORC	3.626418	kW	
12	ΔTcc	10	ORC cycle									Workp_ORC	0.104	kW	
13	ΔT_hs2	30	Tcc_ORC	70	oC	s4s	1.7736		h2	246.27184		Worknet_ORC	3.522837	kW	
14	ORC		Pcc_ORC	609.6	kPa	h4s	435.7715		T2	35.345764		η_ORC	7.933	%	
15	Tcond_ORC	35	Pcond_ORC	211.815	kPa	h4	438.6558		s2	1.1588214		Overall parameters			
16	ΔT_sc	0.00				T4	43.64492					Q_total	51.25127	kW	
17	ηt_isen	0.85	h3	455		s4	1.782746		h2b	254.8476		T_hsout	70.52	oC	
18	ηp_isen	0.65	s3	1.7736					T2b	41.674347	kJ/kg	Q_out	40.88646	kW	
19	ηx_isen	0.85	h2c	294.59		T1	35	oC	s2b	1.1861997	kg/s				
20	T_0	298.15	s2c	1.3062		h1	245.805	kJ/kg				W_total	10.36481	kW	
21	P_0	101.325	v1	0.000763	m3/kg				Qevap_ORC	160.41	kW	η_overall	20.224	%	
22			h4b	430.08		s1	1.1573		mf_ORC	0.22	kW	e_overall	58.33144	%	
23			s4b	1.7553											
24															

Figure 11. The Excel sheet of the combined cycle for the heat-source at 300°C

By fixing the temperature of the ORC evaporator at 70°C and varying the temperature of the TFC evaporator from 167°C (440K) to 257°C (530K), the model was used to calculate the net power, thermal efficiency, and exergetic efficiency of the cycle as determined by Equations (17), (18) and (19). Figures 12 to 14 compare the estimations of the present combined cycle with those obtained by Yu et al [15] for their combined cycle and the conventional dual-loop cycle with the same input data.

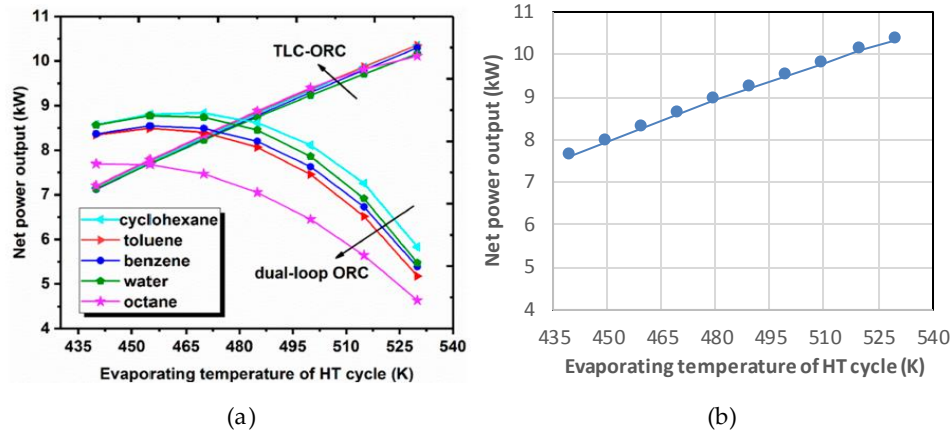


Figure 12. Comparison of the net power; (a) Ref. [15], (b) present cycle

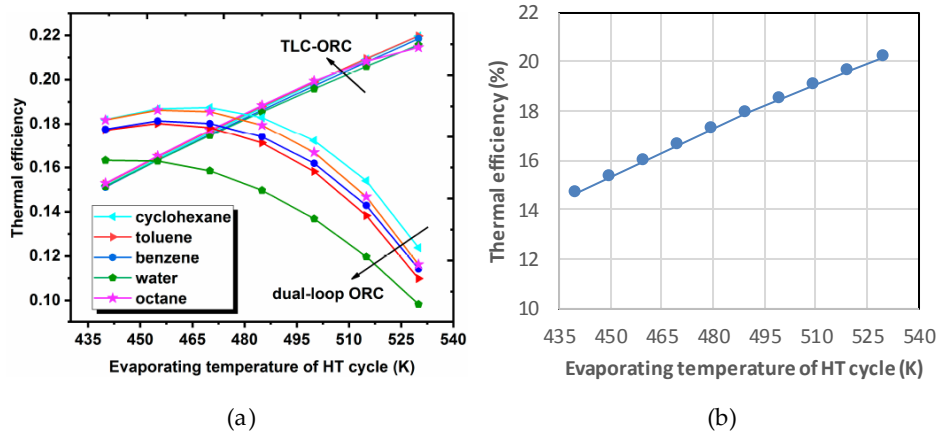


Figure 13. Comparison of the thermal efficiency; (a) Ref. [15], (b) present cycle

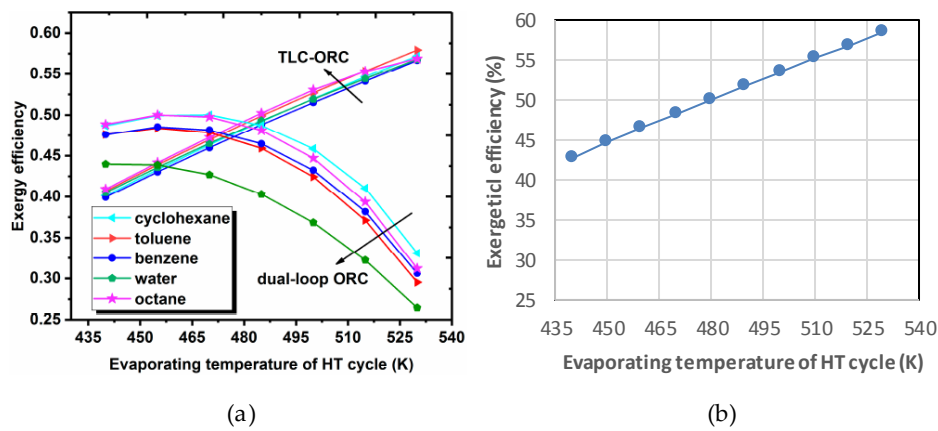


Figure 14. Comparison of the exergetic efficiency; (a) Ref. [15], (b) present cycle

The figures show that the net power, thermal efficiency, and exergetic efficiency of both combined cycles increase as T_{Evap_TFC} increases, while those of the conventional

dual loop deteriorate if T_{Evap_TFC} is increased beyond 465K. The figures also show that the estimations of the present cycle show slightly higher net power and exergetic efficiency and slightly lower thermal efficiency. Table 3 shows the exit temperature of the exhaust gas ($T_{hs,out}$) calculated by the present model at different values of the HT evaporator temperatures (T_{Evap_TFC}) together with the amount of heat recovered by the system ($\dot{Q}_{hs,Tot}$) and the corresponding amount of heat recovered by the combined cycle of Ref. [15]. By allowing lower exit temperatures of the discharged gas, the present cycle could recover more energy and, therefore, produce more net power and give higher exergetic efficiencies than the cycle described by Yu et al [15]. However, reducing the average temperature of the heat-transfer process resulted in the lower thermal efficiencies.

Table 3: The heat recovered by the present cycle compared to that of Ref. [15]

T_{Evap_TFC} [K]	$T_{hs,out}$ [°C]	$\dot{Q}_{hs,Tot}$ [kW]	$\dot{Q}_{hs,Tot}$ [15] [kW]	Diff. (%)
440	67.142	52.005	45.87	13.375
470	68.351	51.735	45.83	12.885
500	69.469	51.485	45.78	12.462
530	70.517	51.251	45.69	12.171

6. Performance of the combined cycle with a single fluid and with two-fluid pairs

The ability to use two different fluids in the ORC and TFC circuits gives the combined cycle an important advantage over the simple ORC and TFC. This section highlights this advantage by comparing the performance of the cycle with a single fluid and with two different fluids in the two circuits. To compare the performance of the combined cycle with that of the simple ORC, the case considered for the analysis is similar to that of the simple ORC shown on Table 1 with an inlet temperature of 120°C. The two fluids selected for this exercise are R152 and R1234yf for their superior performance. Figure 15 shows the Excel model of the combined cycle with the necessary modifications to the data part and three cells K8, K9, and K10 that calculate the heat-source exergy according to Equation (20). R152a is the working fluid in both the ORC and TFC circuits. The temperature of the TFC evaporator, T_{Evap_TFC} , is specified as 110°C and the temperature of the ORC condenser, T_{cond_ORC} , as 40°C. The temperature of the TFC fluid in the cascade condenser, T_{cc_TFC} , is specified as 83°C and the cascade-condenser temperature difference as 3°C. Therefore, the temperature of the ORC fluid in the cascade condenser, T_{cc_ORC} , is 80°C. With a mass flow rate of the hot stream of 100 kg/s, the cycle's net power, thermal efficiency and exergetic efficiency are 1522.9 kW, 9.026%, and 43.85%, respectively.

The model was used to analyse the cycle's performance at various temperatures in the cascade condenser with R152a as the single fluid and with R1234yf in the ORC circuit and R152a in the TFC circuit. Figures 16 to 19 show the heat added, power produced, thermal efficiency, and exit temperature of the heat source calculated for the R1234yf/R152a pair compared to the corresponding values obtained when R152a is used in both circuits.

Tcc_ORC		=Tcc_TFC-ΔTcc													
	Name Box	B	C	D	E	F	G	H	I	J	K	L	M	N	O
1	Working fluids			TFC cycle								TFC			
2	Fluid_TFC	R152a		T_hsi	93.00		s6	1.508015		h8	434.86189		Workt_TFC	656.4599	kW
3	Fluid_ORC	R152a					h7	439.22		s8	1.7098828		Workp_TFC	433.9989	kW
4	Heating source (exhaust gs)			Pevap_TFC	4243.2		s7	1.7058					Worknet_TFC	222.4611	kW
5	T_hs	120	oC	Pcc_TFC	2503.08								Q_TFC	11458.8	kW
6	mflow	100.0	kg/s	h5	360.266		ss8	1.7058					η_TFC	1.941	%
7	p	943.10	kg/m3	s5	1.50023		xss8	0.40023							
8	cp	4.24	kJ/kg.K	mf_TFC	150.62947		hss8	433.4092		Δh_hs	1.5279		ORC		
9	TFC			v5	0.0014074	m3/kg				Δs0_hs	0.3672		Q_sensible	91.67203	kJ/kg
10	Tevap_TFC	110	oC	h6	363.1472					Exerg_hs	3472.92		Q_ORC	5413.38	kW
11	Tcc_TFC	83	oC	T6	84.214554								Workt_ORC	1416.248	kW
12	ΔTcc	3	K	ORC cycle											
13	ΔT_hs2	10	K	Tcc_ORC	80	oC	s4s	2.0198		h2	273.31119		Workp_ORC	115.811	kW
14	ORC			Pcc_ORC	2342.4	kPa	h4s	515.2144		T2	41.037827		Worknet_ORC	1300.436	kW
15	Tcond_ORC	40	oC	Pcond_ORC	909.27	kPa	h4	519.4468		s2	1.2472232		η_ORC	7.811	%
16	ΔT_sc	0.00	K				T4	40					Overall parameters		
17	ηt_isen	0.85		h3	543.43		s4	2.033315		h2b	261.47797		Q_total	16872.18	kW
18	ηp_isen	0.85		s3	2.0198					T2b	34.724121	kJ/kg	T_hsout	80.24	oC
19	ηx_isen	0.75		h2c	353.15		T1	40	oC	s2b	1.2098085	kg/s	Q_out	15349.28	kW
20	T_0	298.15	K	s2c	1.481		h1	271.35	kJ/kg				W_total	1522.897	kW
21	P_0	101.325	kPa	v1	0.001163	m3/kg	v1	0.001163	m3/kg	Qevap_ORC	190.28	kW	η_overall	9.026	%
22				h4b	531.28		s1	1.2411		mf_ORC	59.05	kW	ε_overall	43.85063	%
23				s4b	2.0711										
24															

Figure 15: The Excel model for the combined cycle using R152a in both circuits

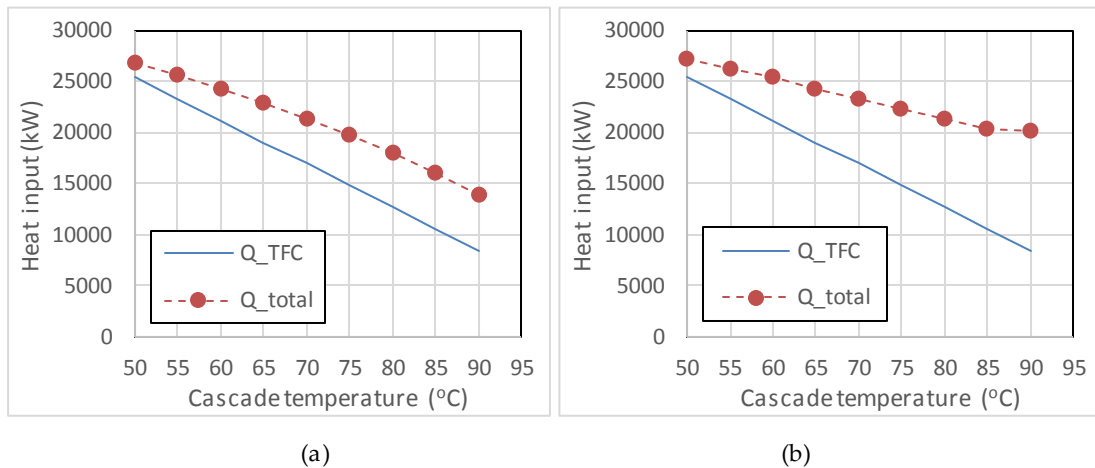


Figure 16: The cycle's heat input (a) R152a/R152a (b) R1234yf/R152a

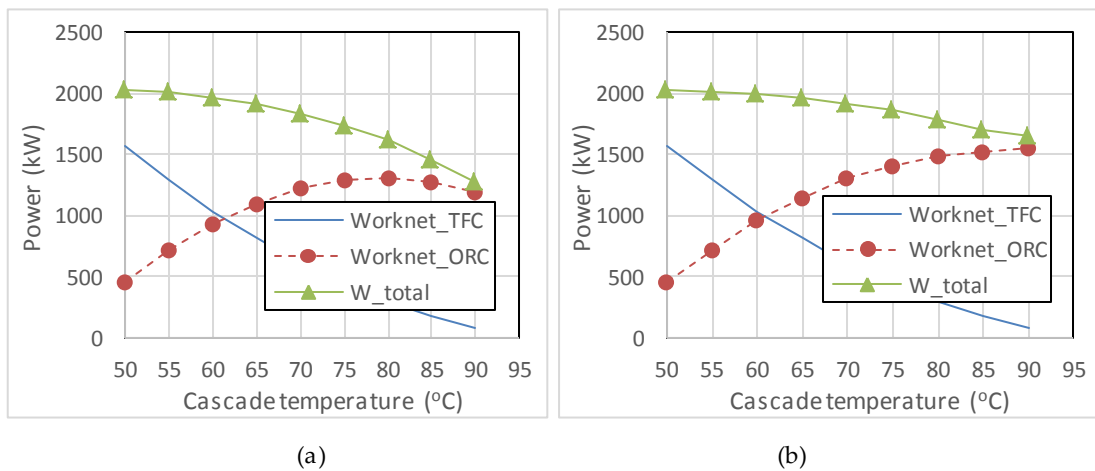


Figure 17: The cycle's power for (a) R152a/R152a (b) R1234yf/R152a

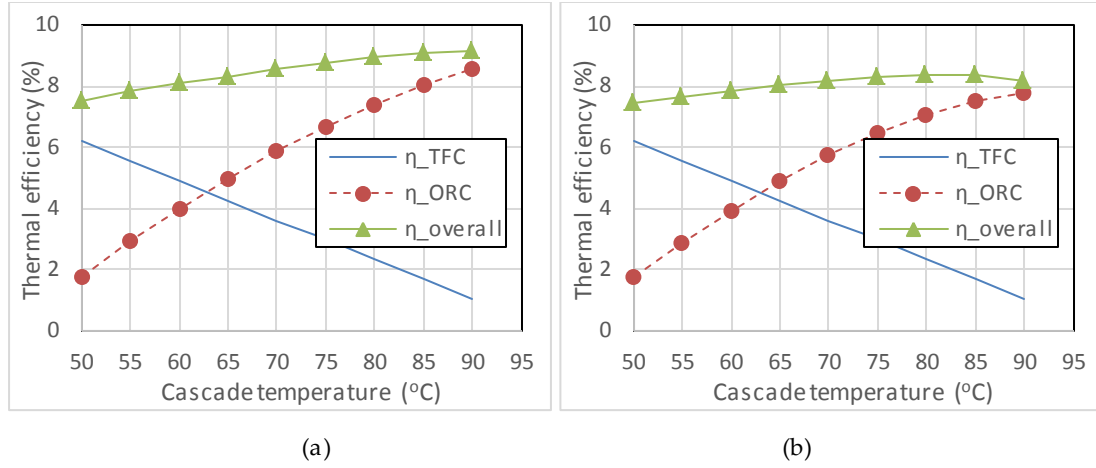


Figure 18: The cycle's thermal efficiency for (a) R152a/R152a (b) R1234yf/R152a

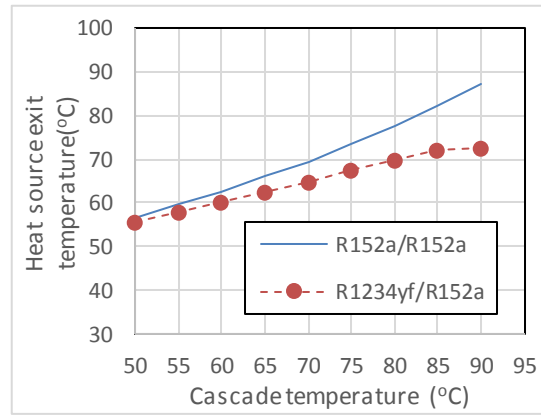


Figure 19: The exit temperature of the heat source for the two fluid pairs

Figure 16 that shows the variation of the heat recovered in the TFC (Q_{TFC}) and the total energy recovered by the combined cycle (Q_{total}) shows that both of them decrease with the increase in the cascade temperature, but the reduction in Q_{total} is less for the pair R1234yf/R152a than that for R152a alone. Noting that the variation of the energy recovered in the TFC is the same for R1234yf/R152a and R152a/R152a, this figure indicates that the ORC recovers more energy with the pair R1234yf/R152a than with R152a alone. Figure 17 shows the variations of the power produced by the ORC and TFC circuits separately and total power produced. For R152a/R152a, Figure 17.a shows that the work produced by the ORC passes through a maximum value at about 80°C, while Figure 17.b shows that the work produced by the ORC for the R1234yf/R152a keeps growing steadily. Therefore, the two-fluids pair recovers more energy and produces more total work than R152a alone at the high values of the condenser temperature. However, Figure 18 shows that the thermal efficiency of the ORC and the overall thermal efficiency for the R1234yf/R152a pair are lower than those obtained by R152a alone. Figure 19 shows that the hot water stream exits at a lower temperature with the R1234yf/R152a pair than with R152a alone. Therefore, the heat transfer with the R1234yf/R152a pair occurs at a lower average temperature, which is the reason for its lower thermal efficiency.

Figure 20 shows the variation of the power and thermal efficiency of the cycle with the cascade temperature obtained for the pair R1234yf/R152a and for R152a alone compared to those of the simple ORC using R152a with the same inlet temperature

and flow rate of the energy-source. The figure shows that both the power and thermal efficiency of the combined cycle for both R1234yf/R152a and R152a/R152a are higher than those of the simple ORC at values of the cascade temperature below 80°C, i.e., when the contribution of the TFC circuit is dominant.

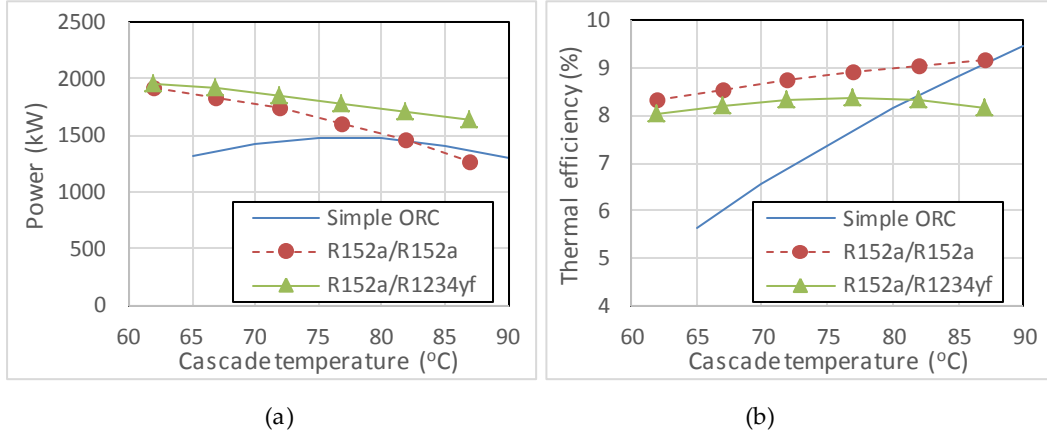


Figure 20: Comparison of (a) the power and (b) thermal efficiency of the combined cycle with R1234yf/R152a and R152a/R152a with those of the simple ORC

7. Analysis of the combined cycle at $T_{source} = 120^{\circ}\text{C}$ with six low-GWP fluids

Most of the recoverable low-grade energy sources are those below 120°C with those below 100°C making up 63% of the total global low-grade energy [27]. Therefore, the selection of suitable working fluids for the combined ORC-TFC cycle in this temperature range is crucial for its acceptability. This section analyses the performance of the cycle for a heat-source at 120°C with six fluids that have zero ODP and low GWP which are R290, R600a, R717, R152a, R1234yf, and R245fa. Table 4 shows their boiling and critical temperatures, GWP, and safety groups [23].

Table 4: Characteristics and safety groups of the six selected fluids [23]

Fluid	Normal boiling point ($^{\circ}\text{C}$)	Critical temperature ($^{\circ}\text{C}$)	GWP	Safety group
R245fa	15.3°C	154 °C	1030	B1
R600a	-12	134.66	-20	A3
R717	-33.33	132.25	0	B2L
R152a	-24	113.26	138	A2
R290	-42.1°C	96.7°C	0.02	A3
R1234yf	-29.4	94.7	<1	A2L

Table 4 arranges the six refrigerants according to their critical temperatures with R245fa having the highest temperature and R1234yf having the lowest one. For an evaporation temperature of 110°C , the first four of these refrigerants can be used in both the TFC and ORC circuits. Since the critical temperatures of R290 and R1234yf are lower than 100°C , they can be used in the ORC circuit only with one of the other four refrigerants in the TFC. In this section the performance of the combined cycle is

first analysed by using a single fluid in both circuits and then by using pairs of two refrigerants in the two circuits with either R1234yf or R290 in the ORC.

7.1. Performance with single fluids

Figure 21 and Figure 22 compare the performance of the first four refrigerants in Table 4 used in both the ORC and TFC circuits. Figure 21.a shows that the power produced by the cycle drops as the cascade temperature is increased for all four refrigerants, but the highest values are those obtained by R245fa and R152a and the lowest values are those obtained by R717. Figure 21.b and Figure 22 show that both the thermal efficiency and exergetic efficiency increase with the cascade temperature, but the highest values are those of R245fa followed by R600a, while R717 and R152a obtained the lowest values. These results indicate that the most suitable single fluid from energetic point of view is R245fa followed by R600a, while R717 resulted in the lowest values of all three performance indicators.

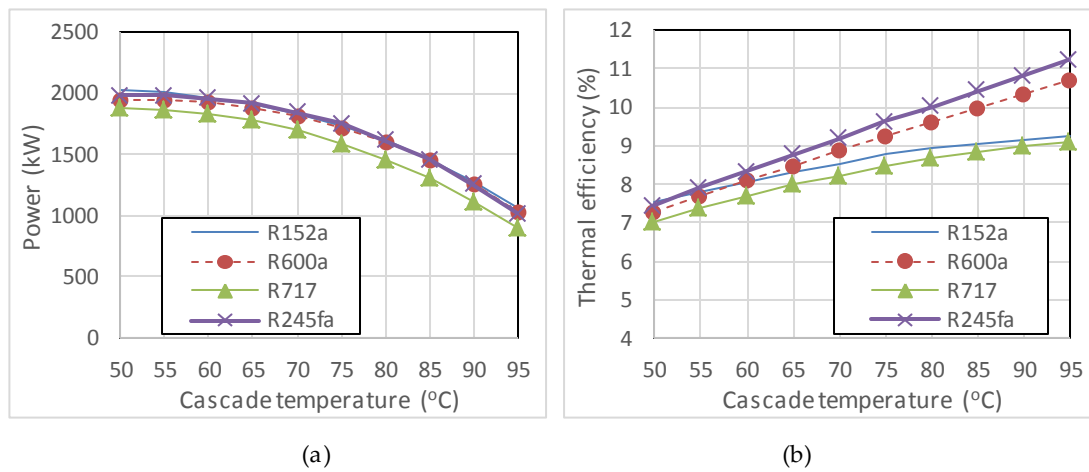


Figure 21: The power and thermal efficiency of the combined cycle using single fluids

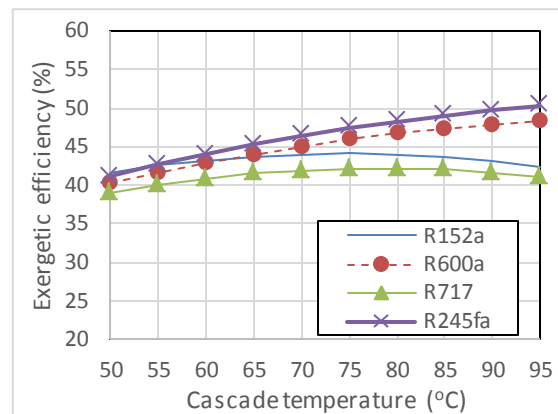


Figure 22: The exergetic efficiency of the combined cycle using single fluids

7.2. Performance with two-fluid pairs

For this analysis, the model was used to evaluate the performance of the combined cycle with six pairs of refrigerants having either R1234yf or R290 in the ORC. The performance of the R1234yf/R152a pair for the same heat source has already been discussed in Section 6. The two other pairs that have R1234yf in the ORC are R1234yf/R245fa and R1234yf/R600a. The three pairs that have R290 in the ORC are

R290/R152a, R290/R245fa, and R290/R600a. Figures 23 to 25 compare the power produced, thermal efficiency, and exergetic efficiency of the combined cycle with these six fluid pairs to those obtained by R152a alone.

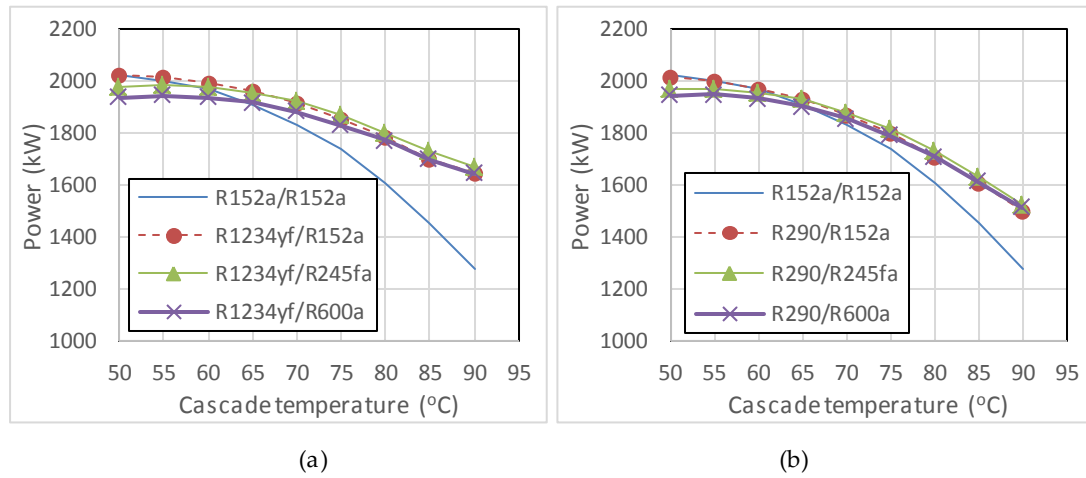


Figure 23: The power of the combined cycle using fluid pairs

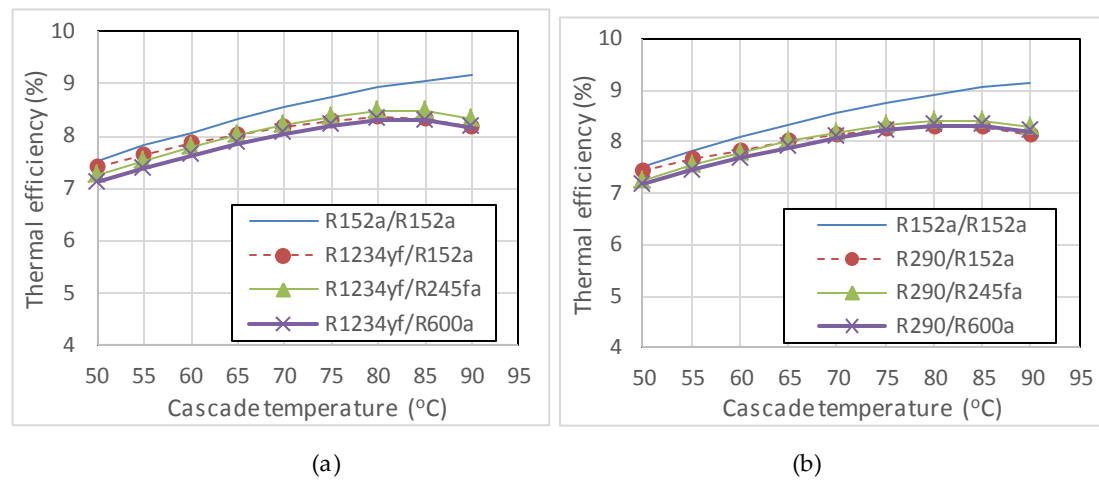


Figure 24: The thermal efficiency of the combined cycle using fluid pairs

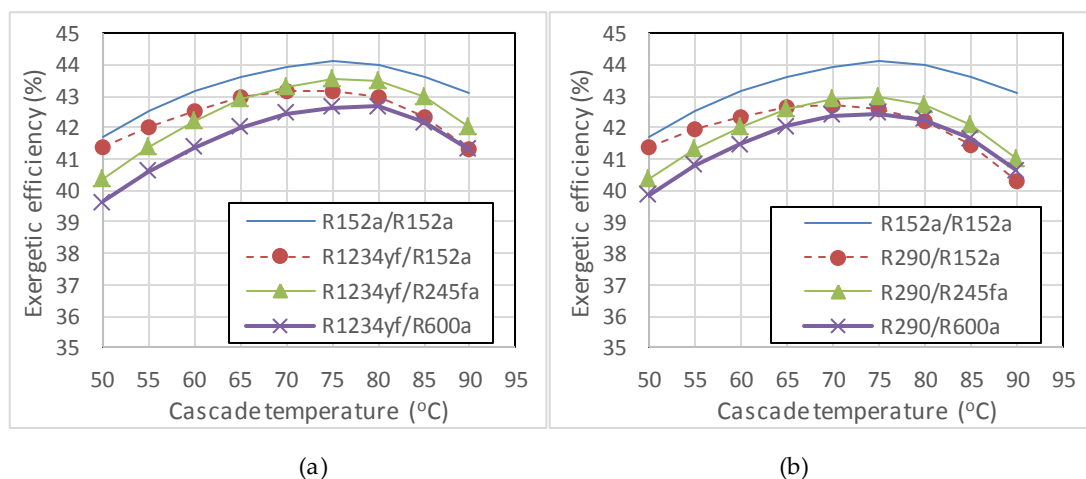


Figure 25: The exergetic efficiency of the combined cycle using fluid pairs

Figure 23 shows that all the six two-fluid pairs produce more power than R152a alone at the high condenser temperatures. However, comparing Figure 23.a with Figure 23.b shows that the three pairs with R1234yf in the ORC circuit also produce

more power than the three pairs involving R290 at the high cascade temperatures. Figure 24 and Figure 25 show that all the two-fluid pairs with either R1234yf or R290 in the ORC have lower thermal and exergetic efficiencies than R152a alone. The figures also show that the two pairs involving R600a produce lower values of the thermal and exergetic efficiencies than those involving R152a or R245fa. Comparison of Figure 25.a with Figure 25.b shows that the exergetic efficiencies with R290 are slightly lower than the corresponding values with R1234yf. Figure 25 also shows that the two pairs involving R152a in the TFC achieve better thermal and exergetic efficiencies at the low condenser temperatures than the pairs involving R245fa, but the trend is reversed at the high condenser temperatures.

8. Conclusions

This paper presents a combined ORC-TFC cycle for power generation from low-temperature heat sources and analyses its performance by using an Excel-based model that utilises VBA functions to determine the refrigerants properties. Initially, the accuracy of the VBA function is confirmed by comparing their estimations with published data for the simple ORC and TFC using various refrigerants as working fluids. The combined cycle is first analysed for a 300°C heat-source from the exhaust of a marine engine and its estimations for the power, thermal efficiency, and exergetic efficiency are gauged with those of another combined ORC-TFC cycle using R245fa in the ORC circuit and water in the TFC circuit. The cycle's performance is then analysed by using a single fluid, which is R152a, in both circuits and by using two different fluids, which are R1234yf in the ORC and R152a in the TFC. The results show that the hot water stream exits at a lower temperature with the R1234yf/R152a pair than with R152a alone. Therefore, the two-fluids pair recovers more energy and produces more work than R152a alone, but its thermal efficiency is lower than that reached by R152a alone since the heat transfer with the R1234yf/R152a pair occurs at a lower average temperature.

The performance of the combined cycle is also analysed for a heat-source at 120°C with six fluids of low GWP, which are R290, R600a, R717, R152a, R1234yf, and R245fa, first by using a single fluid in both circuits and then by using pairs of two refrigerants in the two circuits with either R1234yf or R290 in the ORC. The analyses with single fluids indicate that the most suitable fluid from thermodynamic point of view is R245fa followed by R600a, while R717 is not preferred. The analyses with two-fluid pairs show that the two pairs involving R152a in the TFC achieve better thermal and exergetic efficiencies at the low condenser temperatures than the pairs involving R245fa, but the trend is reversed at the high condenser temperatures. It should be mentioned that selecting the most suitable pair of working fluids for the cycle should take into consideration the economic and safety factors and determining the appropriate cascade temperature for the selected fluids requires a trade-off between the power of the cycle on one side and its thermal and exergetic efficiencies on the other side.

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