ANALYSIS OF THE TRILATERAL FLASH CYCLE AND THE PARTIALLY EVAPORATING CYCLE FOR POWER PRODUCTION FROM LOW TEMPERATURE HEAT SOURCES

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Abstract The Trilateral Flash Cycle (TFC) and the Partially Evaporating Cycle (PEC) have been analyzed and compared to the Organic Rankine Cycle (ORC). Three cases have been investigated. Case I uses air at 100 °C, Case II air at 150 °C and Case III air at 200 °C as the heat source. Water at 20 °C is used as the heat sink for all cases. The cycles are optimized for maximum net power production with eight different working fluids, R123, R134a, R245fa, R1234ze, butane, pentane, isopentane and propane. Detailed heat exchanger models to calculate the pressure drops and heat transfer coefficients are included in the model.

The results show that the TFC has the lowest power production for all cases, and the total system size is estimated to be larger for the TFC compared to the other cycles. The PEC doesn't show any advantage over the ORC for the cases analyzed here.

Keywords Power generation, low temperature heat sources, Organic Rankine Cycle, Trilateral Flash Cycle, Partially Evaporating Cycle, working fluids

1. INTRODUCTION

Several studies have found the Trilateral Flash Cycle (TFC) very promising for power production from low temperature heat sources, like discussed in our previous paper […]. However, none of these studies includes pressure losses in the system and calculation of heat transfer coefficients for each working fluid. In addition, the required heat exchanger areas are usually not considered and most of the studies assumes constant and equal expander efficiency for the TFC and the ORC. The PEC is found to be promising for low temperature heat sources when the working fluid pump efficiency is low [1].

In our previous paper on the TFC […], the cycle was compared to the ORC for three cases defined by different heat source inlet temperatures. Air with a mass flow of 10 kg/s and temperatures of 100, 150 and 200 °C were used as the heat source for Case I, II and III respectively. Water at 20 °C where used as the heat sink for all cases. The cycles were simulated with seven different working fluids, R123, R134a, R245fa, R1234ze, butane, isopentane and propane. The pressure losses were neglected, and the heat transfer coefficients where assumed to be the same for all working fluids. A variable two-phase expander efficiency was used. The TFC was then found to have a higher net power production than the ORC for the 100 °C case, and the difference was found to be insignificant for the 150 °C and 200 °C cases. The system size was estimated to be larger for the TFC, especially at lower heat source temperatures.

In this study, a detailed heat exchanger model is included to calculate the pressure losses and heat transfer coefficients. The cases investigated are the same as in our previous study.

This paper begins with a description of the TFC and PEC. A model description then follows before the results and conclusions are presented.

2. DESCRIPTION OF THE TFC AND PEC

The main components of the ORC, TFC and PEC are pump, heat recovery heat exchanger (HRHE), expander and condenser. The TFCs main difference from the ORC is that the heating process ends at the boiling point of the working fluid, i.e. there is no evaporation and superheating. Power is produced in a two-phase expander after the heating process. A temperature-entropy diagram for an ORC (left) and a TFC (right) with propane as working fluid is shown in Figure 1 below.

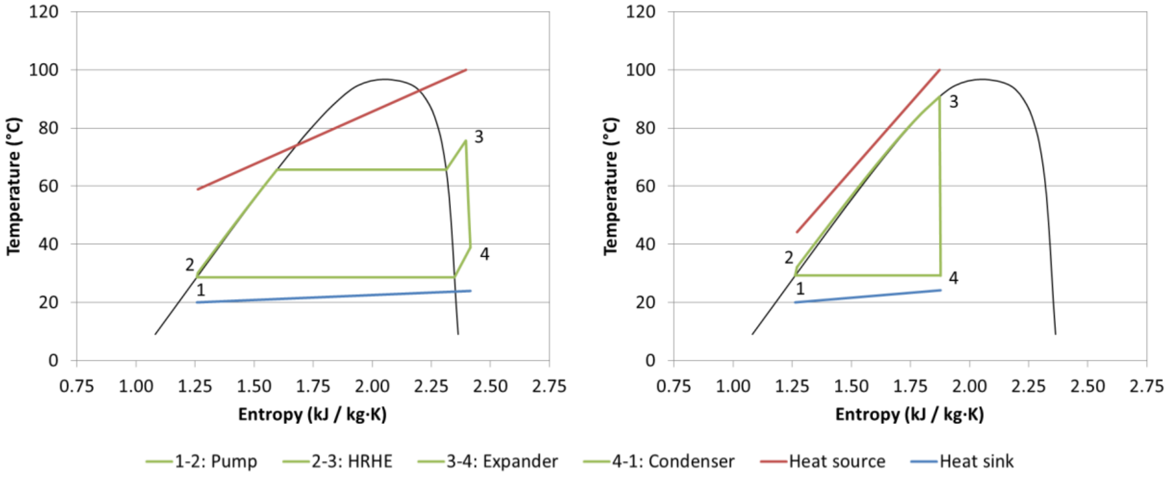

Figure 1: Temperature-entropy diagrams of ORC (left) and TFC (right) with propane as working fluid

The fact that there is no evaporation and superheating in the TFC leads to the possibility of a better temperature match between the working fluid and the heat source in the heat recovery heat exchanger. This is illustrated in the temperature vs. heat transferred diagrams in Figure 2.

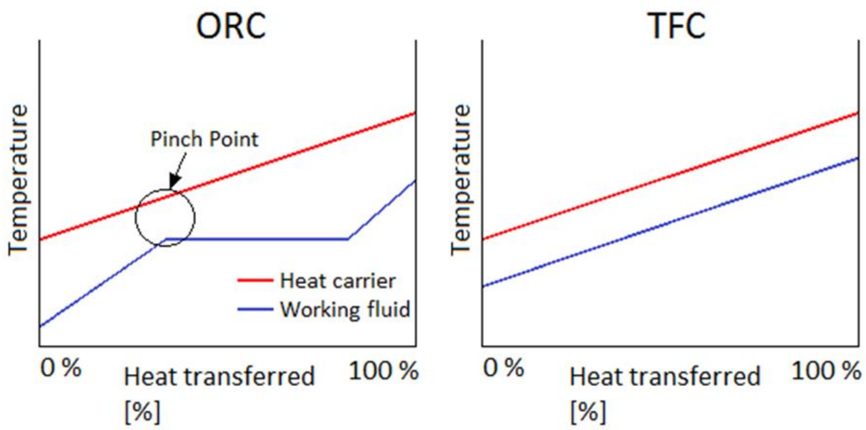

Figure 2: Variation in stream temperature during heat addition for ORC (left) and TFC (right).

As seen in Figure 2, there is a suboptimal temperature matching in the heat recovery heat exchanger for the ORC. This is a result of the constant evaporation temperature of the working fluid and leads to exergy losses. In the TFC both fluids are liquid during the heating process, such that the temperature difference in the heat exchanger is nearly constant.

The cost pr. kWh for TFC systems have been estimated to be lower than for ORC systems due to the elimination of the evaporator, separator drum, gear box, lube oil system and the fact that simpler heat exchangers can be used [2].

In the PEC the working fluid is allowed to be partially evaporated in the heating process. This is done in an attempt to combine the advantages of the TFC and ORC.

The two-phase expansion is the technically most challenging part of the TFC and PEC. For medium scale (50 – 250 kW) applications, like investigated in this study, screw expanders and the Variable Phase Turbine (VPT) are the most relevant expander types. These are described in [3] and [4], respectively.

3. MODEL DESCRIPTION

To analyze the TFC and PEC and compare them to the ORC, the three cycles have been simulated using Microsoft Excel with Visual Basic for Applications (VBA). REFPROP 9 [5] is used for the thermodynamic properties of the working fluids and the Excel solver is used for the optimization.

3.1 Cycle simulations

Before the simulations, the working fluid and boundary conditions are specified. The mass flow, inlet temperature and pressure of the heat source are given, and the inlet temperature and pressure of the heat sink. The following parameters are then optimized during the simulations: working fluid mass flow, pump outlet pressure, vapor quality at expander inlet (only PEC), condensation temperature, heat sink outlet temperature and superheat at the expander inlet (only ORC). Heat source and sink specifications as well as component efficiencies are constant inputs which have to be chosen by the user. The only exception to this are the expander efficiencies for the TFC and PEC, which are calculated automatically. The heat transfer coefficients and pressure drops are calculated using the correlations specified in section 3.2.

The cycles are calculated based on enthalpy balances with a number of constraints to avoid unfeasible solutions: The minimum pinch point temperatures in the heat exchangers and the minimum superheat throughout the ORC expander are specified. Both are calculated by dividing the component into several small sections and ensuring the limits in each section. Also, the pump outlet pressure is limited to 95% of the working fluid's critical pressure and the condensation pressure is kept above 1 bar to avoid a sub-atmospheric system. The total heat exchanger area is constrained during the case studies to compare the performances of the cycles at different system sizes. The cooling of the heat source is not limited although some practical limitations can occur in reality.

The optimization target for the Excel solver is the net power output, which is set to be maximized. Using the net power output instead of cycle efficiency is recommended for waste heat sources, as explained in [6]. The net power output is defined as

$$P_{net} = P_{expander} \cdot \eta_{generator} - \frac{P_{pumps}}{\eta_{motor}} - \frac{P_{fan}}{\eta_{motor}}$$

with P_{pumps} being the sum of working fluid pump power and heat sink pump power, and P_{fan} being the heat source fan power, both in kW. To perform the optimization, the Excel solver is used with the GRG Nonlinear solving method with forward derivatives. A high number of starting points are used to ensure that the global maximum is found with a high probability.

To calculate the heat exchanger area, the well-known logarithmic mean temperature difference (LMTD) method is used. Although the assumptions made in this method are not fully fulfilled in all of our cases, it gives a good approximation. The area for each section of the heat exchangers are calculated individually with the following formula:

$$A = \frac{\dot{Q}}{U \cdot \Delta T_{lm}}$$

Here $\dot{Q}$ is the heat flow in the heat exchanger section in kW, U is the heat transfer coefficient in W/m^{2}K, and ΔT_{lm} is the LMTD in K.

All component efficiencies are assumed constant except for the expander efficiency in the TFC and PEC. The two-phase expander is modeled as a VPT. The nozzle and rotor efficiencies are calculated separately. The nozzle efficiency is calculated as

$$\eta_{\mathrm{nozzle}} = 0.865 + 0.00175 \cdot \mathrm{d}_{4,\mathrm{vapor}}$$

where $d_{4,vapor}$ is the vapor density in kg/m^3 of the working fluid at condensing pressure. The rotor efficiency is calculated as

$$\eta_{\mathrm{rotor}} = 0.575 + 0.325 \cdot \mathrm{q}_4$$

where q_4 is the vapor quality of the working fluid at the nozzle exit. These relationships are based on values given in [7] and [8] and personal communication with Lance Hays, author of [7]. They allow a more realistic comparison between different working fluids, as the main influencing properties (vapor density and vapor quality) are taken into account. However, these relationships are simplified and not validated and should therefore not be taken as accurate.

3.2 Heat exchanger models

Heat exchanger models have been made for both the heater and condenser. The heater is modelled as a plate-finned tube heat exchanger, and the condenser as a plate heat exchanger. For the simulations, some geometry specifications and the mass fluxes needs to be specified by the user, and the inlet and outlet conditions for all the sections are taken from the cycle calculations. Multiple iterations are performed such that the parameters are calculated under the correct conditions. The overall heat transfer coefficients are calculated by the methods of [9].

The heater is divided into a preheating section, an evaporating section (for ORC and PEC) and a superheating section (for ORC). The heat transfer coefficient and pressure drop is calculated separately for each section, using the following correlations:

Single phase (working fluid): The heat transfer coefficients and friction factors for the single phase sections (preheating and superheating) are calculated from the correlation by Gnielinski [10]. **Two-phase:** In the evaporating section for the ORC and PEC the heat transfer coefficient is found from the Liu and Winterton correlation [11]. The pressure loss is calculated from the correlation method of Friedel [12]. The pressure losses in the header and tube entrance is estimated using the method from [9] and the pressure losses in the bends are calculated using the method from [13]. **Air side:** The heat transfer coefficient and friction factor is calculated from the Gray and Webb correlations [14].

The condenser is divided into a desuperheating section and a condensing section. The heat transfer coefficient and pressure drop is calculated separately for each section, using the following correlations:

Single phase: The Martin correlations [15] are used to calculate the heat transfer coefficient and pressure drop for the desuperheating on the working fluid side and for the heating on the heat sink side. It is a semi-theoretical correlation for single phase flow in chevron type heat exchangers. **Two-phase:** The Han, Lee et al. correlations [16] are used to find the heat transfer coefficient and friction factor for the condensation of the working fluid. It uses corrugation pitch and chevron angle as parameters. The pressure losses for the inlet and outlet ports as well as the acceleration pressure drop are calculated using the method of [16].

4. CASE STUDIES

Most of the specifications and boundary conditions for the ORC and TFC are the same, as shown in Table 1. To define a case, all user input values have to be chosen which is explained below.

4.1 Case definitions

Three different heat sources were chosen as case studies. Air was selected as heat source fluid with a mass flow of 10 kg/s and the temperatures were set to 100, 150 and 200 °C for Case I, Case II and Case III, respectively. Water at 20 °C was chosen as heat sink.

To show how the power output changes with system size, the maximum total heat exchanger area was constrained during the calculations. All cases were calculated with seven different limitations: 1000, 1500, 2000, 2500, 3000, 3500 and 4000 m^2. The maximum total heat exchanger area includes all calculated sections: heater, evaporator, superheater, desuperheater and condenser.

All pump efficiencies were set to 0.70, the ORC expander efficiency was set to 0.80, all motor and generator efficiencies were set to 0.90 and the fan efficiency was set to 0.92. All user input values are shown in Table 1.

Table 1: Input data for the ORC, TFC and PEC simulations

	Parameter	**Value**
Cycle	Working Fluid Total area restriction	R123, R134a, R245fa, R1234ze, butane, pentane, isopentane and propane 1000, 1500, 2000, 2500, 3000, 3500 and 4000 m^2
Heat source	Fluid Mass flow Inlet temperature Inlet pressure Fan efficiency Pressure drop	Air 10 kg/s 100, 150 and 200 °C 0.3 MPa 0.92 Calculated in heat exchanger simulation
Heat sink	Fluid Inlet temperature Inlet pressure Pump efficiency Motor efficiency Pressure drop	Water 20 °C 0.3 MPa 0.70 0.95 Calculated in heat exchanger simulation
Pump	Isentropic efficiency Motor efficiency	0.70 0.95
HRHE	HTC heater HTC evaporator (ORC and PEC) HTC superheater (ORC) Minimum pinch temperature Pressure drop	Calculated in heat exchanger simulation Calculated in heat exchanger simulation Calculated in heat exchanger simulation 1 °C Calculated in heat exchanger simulation
Expander	Isentropic efficiency (ORC) Isentropic efficiency, nozzle (TFC and PEC) Isentropic efficiency, rotor (TFC and PEC)	0.80 $0.865 + 0.00175 * vapor\ density$ $0.575 + 0.325 * vapor\ fraction$
Condenser	HTC condenser HTC desuperheater Pressure drop	Calculated in heat exchanger simulation Calculated in heat exchanger simulation Calculated in heat exchanger simulation

4.2 Working fluid selection

Many working fluid studies can be found in the literature, mostly presenting comparisons of thermodynamic performance. However, other aspects like safety (toxicity, flammability, etc.) and environmental impact (global warming potential (GWP) and ozone depletion potential (ODP)) should also be considered.

The standard fluids used in low temperature ORC plants today are R134a and R245fa [17] and are therefore chosen as reference. R123 was chosen as it is suitable for low temperature applications [18]. R1234ze(E) was chosen as it is a promising refrigerant for the future [19]. To compare the performance of these refrigerants to natural working fluids, four hydrocarbons with different critical temperatures were chosen. One of the biggest disadvantages of hydrocarbons is their high flammability.

5. RESULTS AND DISCUSSION

Selected result details for the two best working fluids for each cycle with maximum area are given in Table 2, Table 3 and Table 4 for Case I, II and III respectively.

Table 2: Case I – selected result details for the two best ORCs, TFCs and PECs

Case I, 100 °C		**ORC**		**TFC**		**PEC**	
Working fluid		R134a	R1234ze	R134a	Propane	R134a	R1234ze
Net power	kW	32.0	31.9	28.7	27.4	33.1	33.1
Heat input	kW	497	492	646	618	407	488
Heater area	m^2	800	789	2246	805	807	1091
Condenser area	m^2	200	211	255	195	193	213
Working fluid mass flow	kg/s	2.5	2.6	5.3	2.6	2.6	2.8
Volume flow expander outlet	m^3/s	0.09	0.11	0.11	0.09	0.08	0.12

The ORC and PEC outperforms the TFC for Case I. The best PEC, with R134a, has a power production that is 15 % higher than that of the best TFC, with R134a, and 3 % higher than that of the best ORC, with R134a. The vapor quality at the expander inlet is 1.00 for the PEC, which makes it an ORC without superheating. Both the heat input and power produced in the expander are highest for the TFC, but the required working fluid pump power and heat source fan power are also much higher for the TFC resulting in the lowest net power production. The reason for the high pump power is that the working fluid mass flow is much higher than for the two other cycles, and that it needs to be pumped to a higher pressure. The fan power is higher because the higher heater area requirement leads to higher pressure losses in the heater (the front area of the heater is the same for all cycles).

The best TFC requires a 178 % larger heater, a 32 % larger condenser and a 110 % higher working fluid mass flow than the best PEC. The volume flow at the expander outlet is about 30 % higher for the best TFC compared to the PEC and ORC. The reason the difference in volume flow isn't higher is that the expander outlet temperature is much higher for the PEC and ORC. When comparing the average values for mass flow of working fluid, volume flow rate at the expander outlet and total heat exchanger area requirement over all the different working fluids, the TFC is estimated to have a significantly larger total system size than the ORC and PEC for Case I.

The total expander efficiency ranges from 0.59 to 0.71 for the TFC and 0.59 to 0.82 for the PEC. The expander efficiency is slightly better for the best PECs compared to the ORC because the variable two-phase efficiency is at its optimum for these cases.

Table 3: Case II – selected result details for the two best ORCs, TFCs and PECs

Case II, 150 °C		**ORC**		**TFC**		**PEC**	
Working fluid		R1234ze	R134a	Butane	R245fa	R1234ze	Butane
Net power	kW	114.1	110.1	107.3	90.0	111.1	107.1
Heat input	kW	1180	1150	1183	1114	1141	1094
Heater	m^2	3266	2551	2086	3812	3525	2578
Condenser	m^2	519	449	414	188	479	420
Working fluid mass flow	kg/s	5.8	5.3	3.4	5.9	6.3	3.3
Volume flow expander outlet	m^3/s	0.25	0.19	0.47	0.48	0.25	0.47

The ORC has the highest power production for Case II. The best ORC, with R1234ze, has a power production that is 3 % higher than that of the best PEC, with R245fa and 6 % higher than that of the best TFC, with butane. The best PEC has a vapor quality of 1.00 at the expander inlet, which makes it an ORC. The best TFC requires 36 % less heater area and 20 % less condenser area compared to the best ORC, due to higher heat transfer coefficients. The working fluid mass flow is 42 % lower, and the volume flow at the expander outlet is 87 % higher for the best TFC compared to the ORC. When comparing the average values over all the different working fluids, the TFC still has a significantly larger estimated system size than the ORC, however not as much as for Case I. The estimated system size for the PEC is between the ORC and the TFC.

The total expander efficiency ranges from 0.67 to 0.74 for the TFC and 0.70 to 0.81 for the PEC.

Table 4: Case III – selected result details for the two best ORCs, TFCs and PECs

Case III, 200 °C		**ORC**		**TFC**		**PEC**	
Working fluid		Butane	R245fa	Isopentane	Butane	Butane	R245fa
Net power	kW	220.4	200.6	185.6	183.9	217.2	194.8
Heat input	kW	1696	1662	1622	1772	1680	1629
Heater	m^2	2011	3758	1930	2015	2428	3758
Condenser	m^2	491	242	70	486	573	243
Working fluid mass flow	kg/s	3.4	6.3	3.6	4.6	3.7	6.8
Volume flow expander outlet	m^3/s	0.61	0.67	0.84	0.66	0.63	0.66

The ORC has the highest power production for Case III. The ORC with butane has a power production that is 19 % higher than that of the best TFC, with isopentane. The best PEC, with butane, has a vapor quality of 1.00 at the expander inlet and is an ORC. The required heater area are about the same for the best ORC and TFC, but the ORC requires 599 % more condenser area. The mass flow of working fluid is similar for the ORC and TFC and the volume flow at the expander outlet is 38 % higher for the TFC. When comparing the average values over all the working fluids, the TFC requires a significantly larger mass flow of working fluid, about the same volume flow at the expander outlet, and a slightly lower total heat exchanger area. This is also the case for the PEC, and both cycles are estimated to have a larger system size compared to the ORC.

The total expander efficiency ranges from 0.70 to 0.78 for the TFC and 0.77 to 0.81 for the PEC.

The calculated expander efficiency for the TFC is lower than the assumed 0.8 for the ORC expander for all cases. The efficiency increases with increasing heat source temperature and has a maximum of 0.78 for isopentane in the 200 °C case.

The pressure losses in the evaporator are significantly higher for the natural working fluids butane, pentane and isopentane compared to the standard refrigerants. Despite this, these working fluids are among the best for the 150 °C and 200 °C cases.

The results from this study deviates from the results found in our previous study […]. Now that the pressure losses and heat transfer coefficients are included the TFC can no longer outperform the ORC. The reason is that the increase in pressure losses and associated increase in pump power outweighs the increase in heat input and higher expander power output obtained by increasing the heater area. The heat exchanger coefficient for the TFC heater was also overestimated in the previous study. The new heat exchanger models have shown that the heat transfer coefficients are lower than earlier predicted, and that there are big differences between the different working fluids used. This study deviates even more from other published studies on the TFC by being less promising. This is mainly due to the variable two-phase expander efficiency used here, and that none of the other studies considers pressure losses in the system or calculation of heat transfer coefficients for each working fluid.

To improve this model the massflux of working fluid should be optimized in the cycle simulations to get the optimal tradeoff between the heat transfer coefficient and pressure loss. To do this the calculations needs to be done in another software. The use of working fluid mixtures and more complex cycle configurations could also be investigated.

6. CONCLUSIONS

- The TFC has the lowest power production in all cases, and the total system size is estimated to be larger for the TFC compared to the other cycles, especially for the lower heat source temperature cases.
- The top three PECs are close to pure ORCs or TFCs in all cases, and does not show an advantage over the ORC for the cases studied here.

- The TFC system is estimated to have a lower cost than the ORC due to the elimination of the evaporator, separator drum, gear box, lube oil system and the fact that simpler heat exchangers can be used. The TFC can therefore be suitable in systems with a heat source in the 100 °C to 150 °C range when system size is not a critical factor.
- The calculated expander efficiency for the TFC is lower than the assumed 0.80 for the ORC expander for all cases. The efficiency increases with increasing heat source temperature.
- The results are much less promising for the TFC than the results from our previous study […] and other published studies on the TFC now that the pressure losses, heat transfer coefficients for each working fluid and better two-phase expander model are included.
- The natural working fluids with low environmental impact can match standard refrigerants in terms of performance despite having significantly higher pressure losses in the evaporator. However, the flammability of the investigated hydrocarbons is a disadvantage that needs to be kept in mind.

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