Comparison of ORC and TFC for power production from low temperature waste heat sources

Daniel Rohde[1], Stian Trædal[2], Brede Hagen[1], Trond Andresen[1], Trygve M. Eikevik[2]

[1] SINTEF Energy Research, Postboks 4761 Sluppen, 7465 Trondheim, Norway
[2] The Norwegian University of Science and Technology (NTNU),
Kolbjørn Hejes vei 1B, 7034 Trondheim, Norway

Abstract

In this paper, the Organic Rankine Cycle (ORC) and the Trilateral Flash Cycle (TFC) are compared for power production from low temperature heat sources. Both cycles are simulated with air as heat source at 10 kg/s and temperatures of 100, 150 and 200 °C and heat rejection to water at 20 °C. All cases are simulated for seven different working fluids: R134a, R245fa, R123, R1234ze(E), butane, isopentane and propane. The cycles are optimized for maximum net power production with a variable efficiency for the TFC's two-phase expander.

The results show 10% higher power outputs for the best TFC compared to the best ORC for the 100 °C case. However, the required heat exchanger area and expander outlet volume flow rate are significantly higher for the TFC, indicating a larger and more expensive system. At 150 °C and 200 °C, the power outputs are very similar for both cycles and the difference in system size decreases.

Natural working fluids with low environmental impact show similar performance compared to standard refrigerants. Isopentane showed the best performance when used in a TFC system but does not outperform the ORC.

Keywords: ORC; TFC; Power production; Low temperature; Waste heat

1. Introduction

The world's energy demand is rising and the emission of greenhouse gases such as CO_2 is increasing as an effect. It is therefore necessary to search for alternate energy sources and improve existing methods for power production. In this study, we look at the possibility of improving power production from low temperature waste heat sources by using the Trilateral Flash Cycle (TFC) instead of the Organic Rankine Cycle (ORC).

The ORC is a well-established technology for power production from low to medium temperature heat sources. Its applications range from waste heat recovery to solar and geothermal power plants (Quoilin, Broek et al. 2013). A known drawback of the ORC from a thermodynamic point of view is the suboptimal temperature matching in the heat recovery heat exchanger (HRHE), see Figure 1. This is a result of the constant evaporation temperature of the working fluid and leads to exergy losses (Galanis, Cayer et al. 2009).

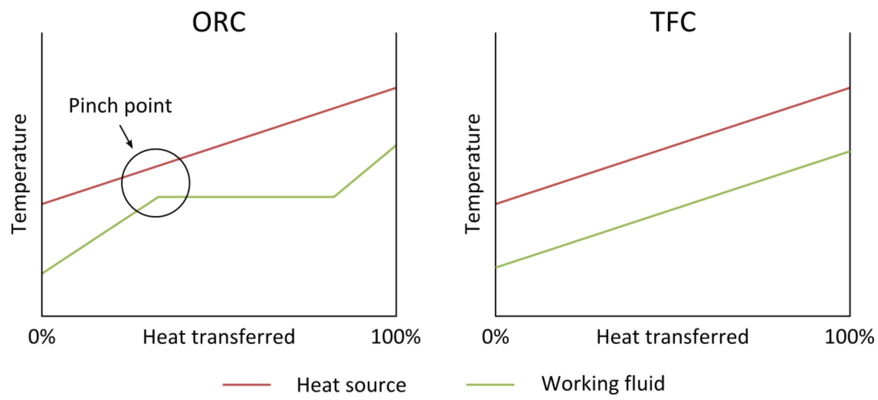

Figure 1: Variation in stream temperature during heat addition process for ORC (left) and TFC (right)

To improve the temperature matching, the use of zeotropic working fluid mixtures or supercritical pressures has been suggested to absorb the heat source at a gliding and thus better temperature profile. However, the best temperature match can be obtained when both fluids are in liquid phase so that the temperature difference in the heat exchanger is almost constant (DiPippo 2007; Ho, Mao et al. 2012). This is the case for the TFC, which can be seen in Figure 2.

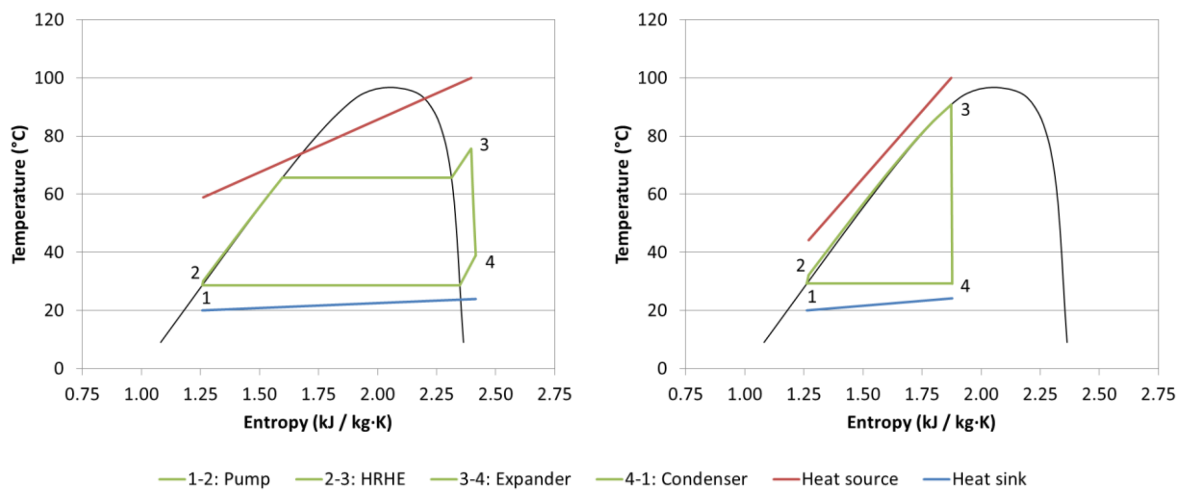

Figure 2: T,s-diagrams of ORC (left) and TFC (right) with propane as working fluid

The main components of both ORC and TFC are pump, HRHE, expander and condenser. However, the expansion process is entirely different: While an ORC has its expansion in the superheated vapor region, a TFC expands into the two-phase region. Turbines and expanders commonly used in ORCs do not tolerate liquids or even liquid droplets, so this two-phase expansion requires specially designed expanders which are described below.

1.1. Expander technology

The technically most challenging component of the TFC is the two-phase expander, as the expansion into the two-phase area includes flash evaporation of the working fluid (giving the TFC its name). This leads to much higher volume ratios between inlet and outlet than for dry vapor expanders and also makes the flow characteristics harder to predict and optimize. An efficient expansion process is vital for the TFC though if it is to outperform an ORC with a high efficiency dry vapor turbine.

There are a few expander types that tolerate two-phase flow. For small-scale (1-10 kW) applications, scroll or rotary vane expanders can be used (Bao and Zhao 2013). For slightly larger systems, a reciprocating piston might be suitable. However, since the piston itself does not tolerate two-phase flow, it can only be used as expander in a TFC when a cyclone for phase separation is implemented upstream (Steffen, Löffler et al. 2013). For medium-scale (50-250 KW) systems as investigated in this paper, screw expanders and the Variable Phase Turbine (VPT) are the most relevant expander types and are described below.

A screw expander is a positive displacement expander that consists of a pair of meshing helical rotors in a casing. The volume trapped between the rotors and the casing changes as the rotors rotate. Whether the volume increases or decreases depends only on the direction of rotation which is why the same machine can be used as compressor and expander. The energy transfer between the fluid and the rotor depends mainly on the pressure on the rotors and only to a small extent on the dynamic effects of fluid motion. This is why the presence of liquid in the expander has little effect on its mode of operation or efficiency (Smith, Stosic et al. 1994; Smith, Stosic et al. 2005).

The VPT consists of a set of individual fixed nozzles and an axial impulse rotor. In the nozzles, the working fluid's enthalpy is partly converted to kinetic energy in a near isentropic expansion. The liquid phase is broken up into small droplets by the expanding gas, and momentum is transferred from the gas to the droplets by pressure and shear forces. The small size of the droplets leads to a close coupling of the gas and the liquid and efficient acceleration of both phases. The inlet to the nozzle can be liquid, two-phase, supercritical or vapor. The kinetic energy of the two-phase jets is converted to shaft power in an axial impulse turbine that allows direct driving of the generator, so no gearbox and lube oil systems are needed (Welch and Boyle 2009).

1.2. Literature study

Currently, only a few studies on the TFC can be found. The TFC was originally developed for geothermal applications and was investigated as far back as 1989 (Hu, Wang et al. 1989). The first publications describing the TFC in detail are from Smith and indicate great potential for geothermal applications (Smith 1993; Smith, Stosic et al. 1995). Smith and Stosic focused on the implementation of screw expanders and pointed out cost-advantages of TFC systems with screw expanders compared to commonly used ORC systems with dry vapor turbines. They state that low temperature heat sources can be utilized substantially cheaper in the range of approximately 20-500 kW because screw

expanders neither require a gearbox nor a lubrication system (Smith, Stosic et al. 2007). However, no report of a commercial application could be found.
Recently, Öhman and Lundqvist tested a screw expander with R134a as working fluid and power outputs around 30 kW (Öhman and Lundqvist 2013). The expander was tested with superheated, saturated and mixed inlet state, and efficiencies around 0.75 were measured with a peak of 0.92. However, the authors stress that it is notoriously difficult to get absolute levels of efficiencies derived from measurements in two-phase conditions.

Brown and Mines performed a fluid study for the TFC with 20 working fluids and analyzed two different inlet temperatures of the heat source (geothermal brine), which were set to 93°C and 160°C. The performance criteria for the working fluid were net power output, expansion ratio and critical temperature. No working fluid was in the top 10 in all categories. They then compared the cycle performance of an ORC with isobutane to a TFC with n-pentane as working fluids and find that the TFC can give higher power output when the expander efficiencies are equal. However, it requires a larger heat exchanger area. They mention that the two-phase expansion would very likely be less efficient and show that it needs to be at least 0.76 to match the performance of the ORC (Brown and Mines 1998).

Energent Corporation has successfully used their Variable Phase Turbine (VPT) in refrigeration systems for 10 years with power outputs up to 20 kW. Due to good test results from a TFC testrig, a 1 MW TFC power plant with R134a as working fluid has been built at Coso Geothermal in California which is designed for power production from a 113 °C brine stream (Welch, Boyle et al. 2010). The system has been operated up to 795 kW (Boyle and Hays 2013) and several other installations from 1.5 kW to 1.5 MW are reported in (Welch and Boyle 2009). A 500 kW geothermal plant is now planned with an advanced VPT, which is hermetically sealed with turbine and generator submerged in the working fluid R134a (Welch, Boyle et al. 2011).

Zamfirescu and Dincer analyzed a TFC with a mixture of ammonia and water as working fluid for a geothermal heat source with 150 °C. They found that it outperforms the ORC, but mention possible difficulties with material selection due to the corrosiveness of the ammonia–water solution (Zamfirescu and Dincer 2008).

Fischer (Fischer 2011) analyzed heat source temperatures from 150 °C to 350 °C with water as working fluid for the TFC. He found that the exergy efficiency (net power output divided by incoming exergy flow of the heat source) was 14 – 29 % higher for the TFC than the ORC. However, water had been reported to be unsuitable by Smith (Smith 1993) and Fischer comes to the same conclusion, mainly due to very high volume flows. In a follow-up study, Lai and Fischer analyzed more working fluids (all pure organic substances) with exergy efficiency as optimization target (Lai and Fischer 2012). Cyclopentane is stated as most promising for low temperatures, although other working fluids like pentane and butane gave similar efficiency and volume flow. The volume flows at the expander outlet were found to be higher for the TFCs compared to the ORCs, indicating larger systems for the TFCs. Working fluid mixtures are mentioned as promising but were not investigated. An expander efficiency of 0.85 was used for both ORC and TFC in these studies.

More sophisticated cycles using two phase expanders have also been analyzed. Smith suggested a cycle with two expansion stages of which one would be two-phase and one dry vapor, which requires a phase-separation after the first expansion (Smith, Stosic et al. 2004). Ho investigated similar and even more advanced configurations with several expansion stages and reports efficiency improvements for low temperature applications compared to ORC (Ho, Mao et al. 2012). Lecompte studied the use of a partially evaporating cycle to combine the advantages of TFC and ORC and reports improvements over a pure TFC, especially when assuming low pump efficiencies (due to the reduced mass flow in a partially evaporating cycle) (Lecompte, van den Broek et al. 2013). While these are only theoretical considerations, they indicate that advanced cycle configurations can have a better performance than a standard TFC.

2. Calculations

To compare the TFC to the ORC, both cycles have been simulated in Microsoft Excel. Visual Basic for Applications (VBA) was used to automatize the calculations, REFPROP 9 by the National Institute of Standards and Technology (NIST) was used for the thermodynamic properties of the working fluids (Lemmon, Huber et al. 2013) and the Excel solver was used as optimization engine.

2.1. Cycle simulation

Table 1 gives an overview over all parameters that were used during the calculations. It can be seen that the following variables were optimized during the simulations: Working fluids mass flow, pump outlet pressure, condensation temperature, heat sink outlet temperature, superheat at expander inlet (only ORC) and recuperator capacity (only ORC). Heat source and sink specifications as well as pressure drops, overall heat transfer coefficients and component efficiencies are constant inputs which have to be chosen by the user. The only exception to this is the expander efficiency for the TFC which is calculated automatically.

Table 1: Calculation parameter types

	Parameter	**Type**
Heat source	Fluid	User input
	Mass flow	User input
	Inlet temperature	User input
	Inlet pressure	User input
	Outlet temperature	Calculated
Heat sink	Fluid	User input
	Mass flow	Calculated
	Inlet temperature	User input
	Inlet pressure	User input
	Outlet temperature	Optimized
	Pump efficiency	User input
	Motor efficiency	User input
	Pump work	Calculated
Pump	Mass flow working fluid	Optimized
	Isentropic efficiency	User input
	Motor efficiency	User input
	Inlet pressure	Calculated
	Outlet pressure	Optimized
	Work	Calculated

HRHE	Pressure drops	User input
	U heater	User input
	U evaporator (ORC)	User input
	U superheater (ORC)	User input
	Amount of superheat (ORC)	Optimized
	Pinch point	Calculated
	Area	Calculated
Expander	Isentropic efficiency (ORC)	User input
	Nozzle efficiency (TFC)	Calculated
	Rotor efficiency (TFC)	Calculated
	Generator efficiency	User input
	Minimum vapor fraction (ORC)	User input
	Work	Calculated
Condenser	Condensation temperature	Optimized
	Working fluid pressure drop	User input
	Heat sink pressure drop	User input
	U desuperheater	User input
	U condenser	User input
	Pinch point	Calculated
	Area	Calculated
Cycle	Working fluid	User input
	Heat loss to ambient	Neglected
	Total heat exchanger area	Calculated
	Efficiency	Calculated
	Net power output	Maximized

The cycle is 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 were set to 1 °C. Both were calculated by dividing the component into several small sections and ensuring the limits in each section. This is necessary because the phase envelope of the working fluid is not taken into account if only inlet and outlet states are considered. Also, the pump outlet pressure was limited to 95% of the working fluid's critical pressure and the condensation pressure was kept above 1 bar to avoid a sub-atmospheric system. The total heat exchanger area was 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, e.g. in aluminum production plants (Børgund 2009).

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 (Quoilin, Declaye et al. 2011). It is defined as

$$P_{net} = P_{expander} \cdot \eta_{generator} - \frac{P_{pumps}}{\eta_{motor}} \qquad (1)$$

with P_{pumps} being the sum of working fluid pump power and heat sink pump power. To perform the optimization, the Excel solver is used with the GRG Nonlinear solving method with forward derivatives. This kind of optimization problem can have many local maximums and the starting values of the optimization variables determine which local maximum is found. As there is no way for the solver to recognize the global maximum or to know how many local maximums exist, it cannot find the global maximum with absolute certainty. Thus, a high number of starting points was chosen to ensure that the global maximum is found with high probability.

To calculate the heat exchanger area, the well-known log mean temperature difference (LMTD) is used. Although the assumptions made in this method are not fully fulfilled in all of our cases, it gives a good approximation. The heat transfer coefficients for the heat exchangers are defined per section, see Table 1. The heat recovery heat exchanger (HRHE) is divided into a heating, boiling, and superheating part. The condenser is divided into a desuperheating and a condensing part. The area for each section is calculated individually with the following formula:

$$A = \frac{\dot{Q}}{U \cdot \Delta T_{lm}} \tag{2}$$

Here $\dot{Q}$ is the heat flow in the heat exchanger section in, U is the heat transfer coefficient and ΔT_{lm} is the LMTD.

All component efficiencies are assumed constant except for the expander efficiency in the TFC. The two-phase expander is modeled as a VPT which is explained in Section 1.1. The nozzle and rotor efficiencies are calculated separately. The nozzle efficiency is calculated as

$$\eta_{nozzle} = 0.865 + 0.00175 \cdot d_{4,vapor} \tag{3}$$

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

$$\eta_{rotor} = 0.575 + 0.325 \cdot q_4 \tag{4}$$

where q_4 is the vapor quality (dimensionless) of the working fluid at nozzle exit. These relationships are based on values given in (Hays 2010) and (Welch and Boyle 2009) and personal communication with Lance Hays, author of (Hays 2010). They allow a more realistic comparison between different working fluids, as the main influencing properties (vapor density and vapor fraction) are taken into account. However, these relationships are simplified and not validated and should therefore not be taken as accurate.

2.2. 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.

Case definition

Three fictive waste 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 and kept constant for all cases.

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 five different limitations: 500, 1000, 1500, 2000 and 2500 m^2. The maximum total heat exchanger area includes all calculated sections: Heater, evaporator, superheater, desuperheater and condenser.

The heat transfer coefficients are assumed constant for each part in the HRHE and condenser. They were set according to values presented in (Ho 2012) where an order of magnitude approximation was used to define convective heat transfer coefficients for different stream combinations which were comparable to other published values.

All pressure drops were neglected, except for the heat sink side of the condenser. Here, the pressure drop is needed to calculate the heat sink pump power which influences the optimization target (net power output) and 100 kPa was chosen as value. As systems with lower absolute pressure usually suffer more from pressure drops than systems with higher absolute pressure, neglecting all other pressure drops introduces an error in favor of the low pressure systems.

All pump efficiencies were set to 0.7, the ORC expander efficiency was set to 0.8 and all motor and generator efficiencies were set to 0.9. All user input values are shown in Table 2, except for the working fluids which are explained separately and shown in Table 3.

Table 2: User input values for case studies

	Parameter	**Value**
Heat source	Fluid	Air
	Mass flow	10 kg/s
	Inlet temperature	100, 150, 200 °C
	Inlet pressure	0.3 MPa
Heat sink	Fluid	Water
	Inlet temperature	20 °C
	Inlet pressure	0.3 MPa
	Pump efficiency	0.70
	Motor efficiency	0.90
Pump	Isentropic efficiency	0.70
	Motor efficiency	0.90
HRHE	Pressure drops	Neglected
	U heater	90 W/(m^2·K)
	U evaporator (ORC)	100 W/(m^2·K)
	U superheater (ORC)	50 W/(m^2·K)
Expander	Isentropic efficiency (ORC)	0.80
	Generator efficiency	0.90
	Minimum vapor fraction (ORC)	100%
Condenser	Working fluid pressure drop	Neglected
	Heat sink pressure drop	100 kPa
	U desuperheater	90 W/(m^2·K)
	U condenser	900 W/(m^2·K)

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 (Quoilin, Broek et al. 2013) and are therefore chosen as reference. R123 was chosen as it is suitable for low temperature applications (Li, Wang et al. 2013). R1234ze(E) was chosen as it is a promising refrigerant for the future (Zyhowski and Brown 2011). To compare the performance of these refrigerants to natural working fluids, three hydrocarbons with different critical temperatures were chosen. One of the biggest disadvantages of hydrocarbons is their high flammability which can be reduced by mixing them with CO_2 (Garg, Kumar et al. 2013). However, in this paper only pure fluids are investigated.

All selected fluids are listed in Table 3 with their critical temperatures, GWP and safety class (according to ASHRAE Standard 34). The ODP is not listed as it is negligible for all selected fluids.

Table 3: Properties of selected working fluids (Calm and Hourahan 2011; Fukuda, Kondou et al. 2014)

Fluid	**T_{crit} (°C)**	**GWP**	**Safety class**
R134a	101.1	1300	A1
R245fa	154.0	1030	B1
R123	183.7	77	B1
R1234ze(E)	109.4	6	A2L
Butane	152.0	20	A3
Isopentane	187.2	20	A3
Propane	96.7	20	A3

3. Results and discussion

The results for all cases and working fluids are presented below. The values for net power production for each working fluid are shown in Figure 3, Figure 4 and Figure 5 for Case I, II and III, respectively. The net power is shown in a separate column for each working fluid and cycle. The columns are divided into colored sections, representing the different constraint values for total heat exchanger area. Naturally, more power requires more heat exchanger area. However, when the pinch point limitation (1 °C) is reached in both heat exchangers, no area can be added anymore and increasing the maximum allowed area thus has no effect on the results anymore.

Adding a recuperator generally increases the power cycle efficiency and is thus favorable from a thermodynamic point of view. However, it adds cost and complexity to the system, so it should only be included if it is economically reasonable. Some cases in our calculations showed an available temperature difference for recuperation of 60 °C. However, when a recuperator was added, this difference decreased as the area for the recuperator was no longer available for the other heat exchangers due to the constrained total heat exchanger area. Thus, the net power output only increased slightly for some cases but not significantly enough to be included in the results.

3.1. Results for Case I (heat source inlet temperature: 100 °C)

The calculated net power output for Case I is very similar for all fluids: Around 30 kW can be produced when the heat source inlet temperature is 100 °C.

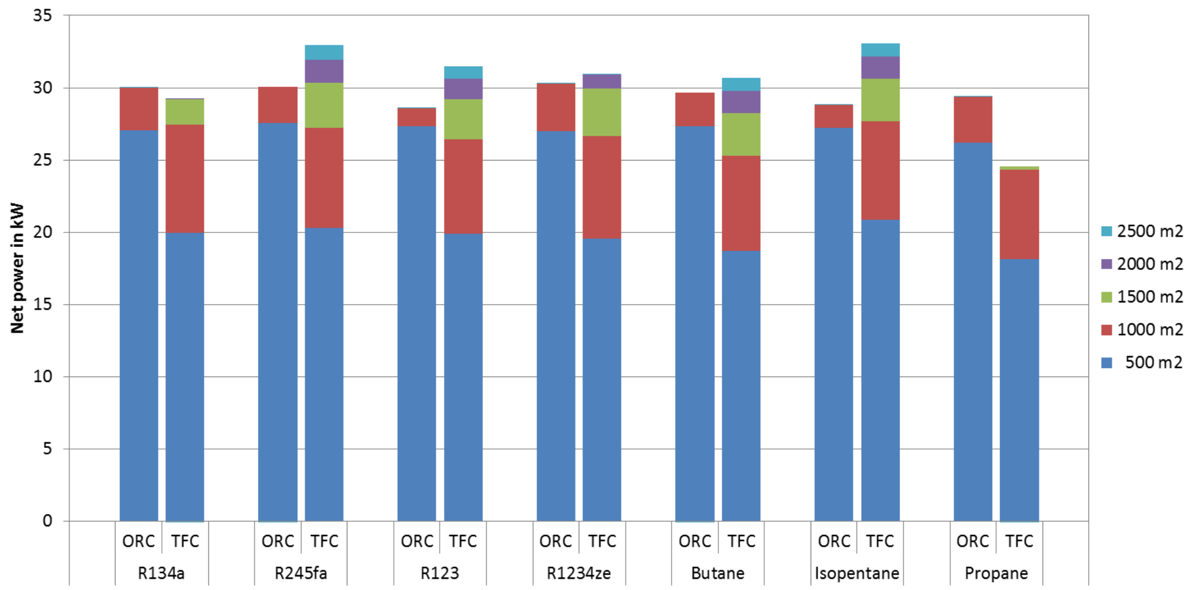

Figure 3: Net power for ORC and TFC for Case I (heat source inlet temperature: 100 °C)

The total expander efficiencies for the TFCs range from 0.617 to 0.705. Detailed results for the different systems with maximum heat exchanger area are shown in Table 4.

Table 4: Case I – selected result details (ORCs and TFCs with highest net power highlighted)

Fluid	**Cycle**	**Net power output**	**Total heat exchanger area**	**Heat source outlet**	**Pump outlet pressure**	**Mass flow working fluid**	**Expander volume flow (out)**
		kW	m2	°C	bar	kg/s	m3/s
R134a	ORC	30.0	822	53.1	1.93	2.3	0.08
	TFC	29.2	1494	33.7	3.51	5.9	0.11
R245fa	ORC	30.0	747	54.6	0.53	2.1	0.26
	TFC	32.9	2500	30.9	1.20	6.9	0.39
R123	ORC	28.6	652	57.2	0.33	2.2	0.37
	TFC	31.5	2500	32.1	0.74	9.2	0.58
R1234ze(E)	ORC	30.3	826	52.1	1.48	2.6	0.10
	TFC	30.9	1980	31.5	2.75	6.4	0.14
Butane	ORC	29.7	745	54.7	0.71	1.1	0.19
	TFC	30.7	2500	31.4	1.46	3.6	0.28
Isopentane	ORC	28.8	686	56.2	0.31	1.1	0.39
	TFC	33.1	2500	32.1	0.68	4.0	0.61
Propane	ORC	29.4	837	53.2	2.39	1.2	0.06
	TFC	24.5	1093	38.3	3.99	2.7	0.08

3.2. Results for Case II (heat source inlet temperature: 150 °C)

Unlike for Case I, big differences can be seen for Case II: The net power ranges from 50 to 110 kW for a heat source inlet temperature of 150 °C.

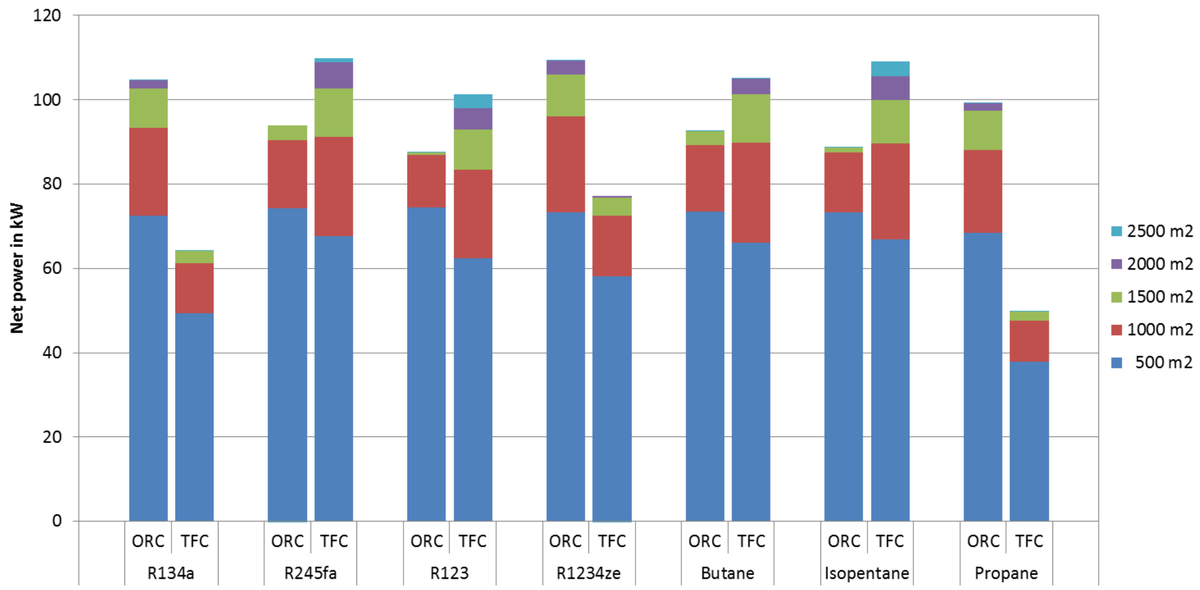

Figure 4: Net power for ORC and TFC for Case II (heat source inlet temperature: 150 °C)

The total expander efficiencies for the TFCs range from 0.693 to 0.740. Detailed results for the different systems with maximum heat exchanger area are shown in Table 5.

Table 5: Case II – selected result details (ORCs and TFCs with highest net power highlighted)

Fluid	Cycle	Net power output	Total heat exchanger area	Heat source outlet	Pump outlet pressure	Mass flow working fluid	Expander volume flow (out)
		kW	m2	°C	bar	kg/s	m3/s
R134a	ORC	104.6	1731	37.1	3.86	5.4	0.18
	TFC	64.2	1520	29.1	3.86	9.6	0.19
R245fa	ORC	93.9	1346	58.0	1.13	3.9	0.50
	TFC	109.8	2273	33.2	2.93	6.6	0.62
R123	ORC	87.4	1069	65.1	0.66	4.1	0.69
	TFC	101.2	2500	36.1	1.93	8.8	0.92
R1234ze(E)	ORC	109.3	1814	36.2	3.45	5.8	0.23
	TFC	77.2	1680	28.8	3.45	9.0	0.25
Butane	ORC	92.6	1328	58.8	1.38	2.0	0.36
	TFC	105.0	1957	35.4	3.17	3.4	0.45
Isopentane	ORC	88.7	1168	61.9	0.62	2.0	0.74
	TFC	109.0	2500	36.2	1.73	3.7	0.96
Propane	ORC	99.2	1699	38.3	4.04	2.7	0.15
	TFC	49.7	1449	29.8	4.04	5.3	0.16

3.3. Results for Case III (heat source inlet temperature: 200 °C)

As expected, the differences in performance are even bigger for Case III: Power outputs from 70 to 225 kW are calculated for a heat source inlet temperature of 200 °C.

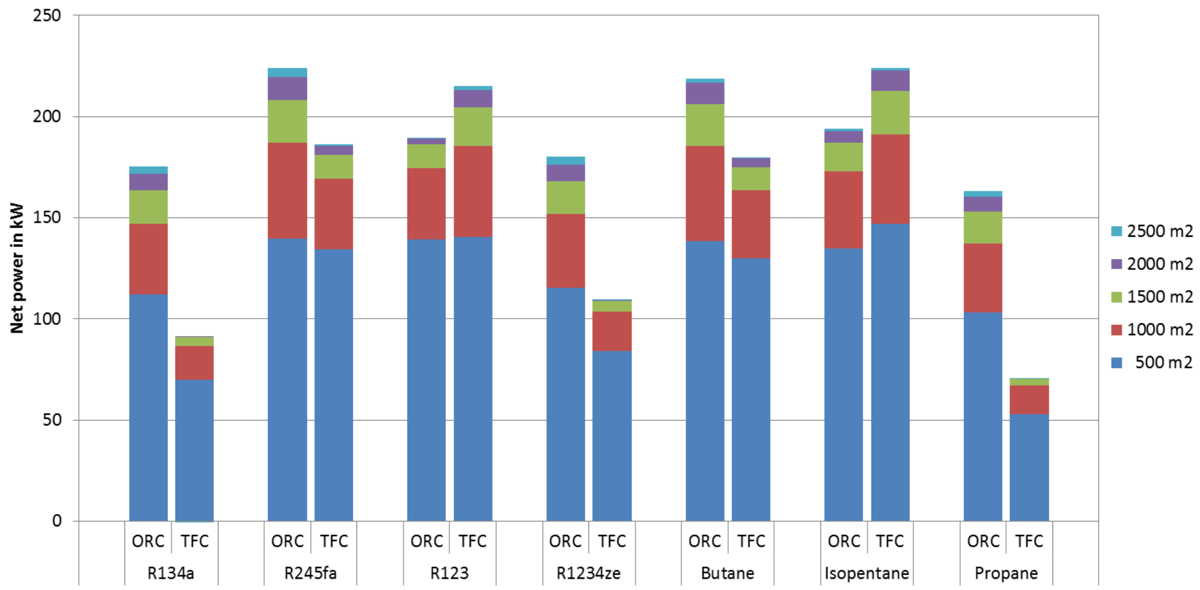

Figure 5: Net power for ORC and TFC for Case III (heat source inlet temperature: 200 °C)

The total expander efficiencies for the TFCs range from 0.703 to 0.783. Detailed results for the different systems with maximum heat exchanger area are shown in Table 6.

Table 6: Case III – selected result details (ORCs and TFCs with highest net power highlighted)

Fluid	**Cycle**	**Net power output**	**Total heat exchanger area**	**Heat source outlet**	**Pump outlet pressure**	**Mass flow working fluid**	**Expander volume flow (out)**
		kW	m2	°C	bar	kg/s	m3/s
R134a	ORC	175.4	2472	30.5	3.86	6.4	0.26
	TFC	91.0	1520	29.3	3.86	13.7	0.27
R245fa	ORC	224.2	2373	34.8	3.47	6.5	0.86
	TFC	186.4	2216	28.2	3.47	8.7	0.90
R123	ORC	189.1	1829	60.7	1.40	6.3	1.09
	TFC	215.1	2212	34.7	3.48	9.0	1.27
R1234ze(E)	ORC	180.2	2500	30.7	3.45	6.6	0.33
	TFC	109.3	1589	29.0	3.45	12.8	0.35
Butane	ORC	218.9	2186	36.3	3.61	3.4	0.62
	TFC	179.8	2157	28.7	3.61	4.6	0.66
Isopentane	ORC	193.9	2242	52.3	1.30	3.0	1.17
	TFC	224.1	2276	35.4	3.07	3.8	1.31
Propane	ORC	163.4	2417	30.7	4.04	3.3	0.22
	TFC	70.6	1493	30.0	4.04	7.5	0.23

3.4. Discussion

The results presented in the previous sections show that the calculated expander efficiency for the TFC was always less than the assumed 0.8 for the ORC. This was expected as explained in Section 1.1. The efficiency increases with increasing heat source temperature and reaches a maximum of 0.783. The expander outlet volume flows are generally higher for the TFC, indicating a larger system.

The heat source utilization is much higher for the TFC for Case I, which leads to lower heat source outlet temperatures. However, this advantage comes at the cost of a higher total heat exchanger area which again leads to a larger and more expensive system. This difference between ORC and TFC decreases as the heat source temperature increases and the heat source outlet temperatures and total heat exchanger areas are similar for both cycles for Case III.

The two effects explained above lead to a 10% higher maximum net power output for the TFC for Case I. For Case II and Case III, the net power outputs are very similar. However, the total system size is believed to be higher for the TFC for all cases.

The investigated refrigerants are well-suited for power production and are commonly used in ORC systems. It can be seen that the critical temperature is an important parameter for the working fluid choice in a TFC. A high difference between heat source inlet temperature and critical temperature of the working fluid leads to poor temperature matching and thus low performance, see R134a, R1234ze(E) and propane for Case II and Case III.
The natural working fluids showed similar performance to the refrigerants, especially the TFC systems with isopentane. This might be an environmentally friendly alternative to the standard systems used today. However, the differences were rather small and replacing R134a with R1234ze(E) in existing systems also seemed reasonable.

The presented results deviate from the results of many other studies like (Zamfirescu and Dincer 2008), (Fischer 2011), (Lai and Fischer 2012), or (Chan, Ling-Chin et al. 2013) by being less promising for the TFC. This is mainly due to the variable expander efficiency that was used for the two-phase expander in the TFC. All of the above mentioned studies assumed a constant (and rather optimistic) efficiency. An even more detailed and, most importantly, verified calculation method for the expander efficiency of both ORC and TFC is required for a more realistic comparison.

Future work will involve the implementation of correlations for the heat transfer coefficients and pressure drops in both heat exchangers. The use of working fluid mixtures and more complex cycle configurations like the Smith cycle (Smith, Stosic et al. 2005), a partially evaporating cycle (Lecompte, van den Broek et al. 2013) or a dual stage cycle (Choi and Kim 2013) might be promising and could also be investigated.

4. Conclusions

The following can be concluded from the performed calculations:

- For a heat source inlet temperature of 100 °C, up to 10% more power can be produced with a TFC. At 150 and 200 °C, the difference between the best ORC and TFC is insignificant.
- The system size is estimated to be larger for the TFC. Both the heat exchanger area and the expander outlet volume flow are generally higher for the TFC systems.
- This study thus shows less promising results for the TFC than other studies, mainly due to the lower expander efficiency.
- Natural working fluids with low environmental impact can match standard refrigerants in terms of performance, especially isopentane used in a TFC. However, the flammability of the investigated hydrocarbons is a disadvantage that needs to be kept in mind.

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