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Increased power production through enhancements to the Organic Flash Cycle (OFC)

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ABSTRACT

The previously proposed Organic Flash Cycle (OFC) was shown to potentially improve power generation from finite thermal energy reservoirs. The OFC's primary advantage was improved temperature matching to the energy reservoir, thereby reducing exergy destruction and losses during energy transfer to the cycle working fluid. This advantage was negated though by irreversibilities introduced by flash evaporation. The present study proposes several design enhancements to the OFC. A theoretical analysis is conducted using the BACKONE, Span–Wagner, and REFPROP equations of state and ten aromatic hydrocarbon and siloxane working fluids previously suggested are reexamined. A comparison and discussion of the different improvements are also presented. Results showed that by splitting the expansion process into two steps and combining the liquid stream from flash evaporation in a secondary expansion stage, utilization efficiency gains of 10% above the optimized basic Organic Rankine Cycle (ORC) are achievable. The greatest efficiency gains were observed when combining this modification and replacing the flash evaporation throttling valve with a two-phase expander. Results showed that proposed enhancements had the greatest effect using aromatic hydrocarbon working fluids, increasing the utilization efficiency by 5%–20% over the optimized ORC; whereas for siloxanes, the improvement was limited to 2%–4%.

1. Introduction

The inclusion of electricity production from energy sources beyond conventional fossil fuels will become inevitably necessary in order to meet increasing global energy demands. Efficient and cost-effective utilization of renewable energy sources such as solar thermal and geothermal energy will lessen the reliance on fossil fuels and reduce the emission of pollutants and potential climate changing agents. In addition, greater care must be taken to improve efficiency of all processes and reduce the amount of energy that is wasted or unused. In a number of manufacturing industries, process heat containing significant amounts of energy is vented and lost to the environment. This high quality waste energy though has the potential to be efficiently converted to electricity which would reduce thermal pollution and overall plant operating costs as the electricity generated from the waste heat could be used to partly power the plant itself or be sold back to the grid. This is a topic of applied research in the industries including ceramics [1], metallurgical [2], paper and pulp [3], food and beverage, oil refining industries [4], and cement. Particularly in the cement industry, a number of exergy and optimization studies have been conducted on recovering and utilizing high temperature thermal energy from the kilns [5,6]. The Organic Rankine Cycle (ORC) has been shown to be a viable method in recovering this waste heat energy in cement plants [7]. Steinbess provides an overview of past, present, and future methods in utilizing waste heat energy in cement plants [8]. Aside from cement and other manufacturing industries, energy from the exit stream of gas turbines in high temperature Brayton cycles could also be used to generate electricity; in fact, utilizing this energy is the premise of combined cycle plants [9,10]. These sources of thermal energy are often termed “finite thermal energy reservoirs” because the source temperature decreases dramatically as heat is transferred to the power cycle. In improving the efficiency with which these renewable energy sources are utilized, there is a greater potential in the investment and widespread adoption of these technologies due to less risk in their high initial capital costs.

In addition to the aforementioned waste heat energy sources, one-axis concentrating solar thermal (collectors such as linear Fresnel lens and parabolic troughs) [11] and high grade geothermal are also in the temperature range of 200–400 °C [12]. For applications in this temperature range, Rankine vapor cycles are the most suitable power cycle [13]. However, the traditional steam

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working fluid is economically unsuitable for low temperature energy sources below 370 °C [14,15] due to a number of complications that include low vapor pressures in the condenser [13], high turbine volumetric flow ratios [16], and low thermal efficiencies caused by low boiler pressures that are required for superheating the steam and avoiding moisture from forming during the final expansion stages in the turbine [13]. High volume flow ratios requires larger, more complicated, and more expensive turbines; low vapor pressures in the condenser requires deaerators to remove air that leaks into the system [13]. Liquid droplets and moisture forming in the turbine during expansion erodes and damages the turbine blades [13,17]; aside from expensive, reinforced blades being necessary, it is also known that turbine isentropic efficiency decreases linearly with the fluid's average moisture content by the so-called "Baumann rule" [17]. Organic working fluids on the other hand can overcome many of these obstacles and have been utilized in the ORC. Organic fluids have turbine volume ratios that can be an order of magnitude smaller than steam [13], they have higher vapor pressures compared to steam [13], and because the majority of them are "dry" or "isentropic" fluids they do not require superheating prior to expansion in order to avoid droplet formation in the turbine [16,18]. "Dry" and "isentropic" fluids exhibit a positively or infinitely sloped saturated vapor curve respectively on a Temperature–Entropy (T–S) diagram. Unlike "wet" fluids like water that have a negatively sloped saturated vapor curve, isentropic expansion from a saturated vapor state for "dry" and "isentropic" fluids will always result in a saturated vapor or superheated vapor; therefore, there is no concern for liquid droplets forming and damaging turbine blades [19,20].

One major challenge for ORCs is temperature matching to the thermal energy source stream while heat is transferred to the ORC working fluid stream. Temperature matching to the source stream is important in minimizing the irreversibilities caused by heat transfer across a finite temperature difference [21,22]. When using the ORC to produce electricity from a finite thermal reservoir though, temperature mismatching often is inevitable because the source stream is single-phase and possesses a near linear temperature profile along the heat exchanger. For an ORC employing a pure working fluid though, the fluid is heated first as a liquid, then vaporized, and if necessary, further superheated as a vapor. The resulting temperature profile along the heat exchanger for the ORC will first be near linear, then constant during phase change, and then near linear again. This temperature mismatching between streams is ultimately the largest source of the cycle irreversibility and lost exergy (potential work) [23]; this has led to much research into innovative methods to produce closer temperature matches between the energy source and working fluid. A comprehensive review of many of these proposed methods can be found in ref. [18,24]; they include the transcritical and supercritical ORCs [25,26] and CO_2 Rankine cycles [27], zeotropic Rankine cycles [19,20], the Kalina cycle [28,29], and the Trilateral Flash Cycle [30]. Two of the more often suggested methods are adding heat while the fluid is above the critical pressure or using a multi-component mixture working fluid instead of a pure one component fluid. In the former case, isothermal phase change is avoided because heat addition occurs above the critical pressure and the fluid avoids the vapor dome completely. In the latter case, the working fluid mixture changes temperature during phase change because the different components of the mixture vaporize at different temperatures for a common pressure.

In a previous study [24], the authors proposed that heat addition occur with the cycle working fluid remaining in the liquid state. In other words, heat is transferred to the ORC until the working fluid reaches a saturated liquid state; this would ensure near perfect temperature matching to the source stream. The fluid would then be flash evaporated to produce a two-phase mixture; the saturated vapor would be separated and then expanded to produce power. This cycle was termed the Organic Flash Cycle (OFC) and the cycle schematic and T–S diagram are shown in Fig. 1 for a "dry" working fluid. The premise of the OFC was that the advantages of increased exergetic efficiency in the heat addition process due to better temperature matching would outweigh the disadvantages of the irreversibilities generated by uncontrolled expansion during the flash evaporation process. A finite thermal source initially at 300 °C was considered in the study to be a broad representation of the aforementioned energy sources. Ten different aromatic hydrocarbons and siloxanes were analyzed as potential working fluids using a combination of the BACKONE, Span–Wagner, and REFPROP equations of state. The OFC was compared to the optimized ORC and results showed that the two cycles possessed comparable performances and gave similar power production per unit finite thermal energy reservoir flow rate [24]. It was determined that although the OFC possessed higher heat addition exergetic efficiency, that did not

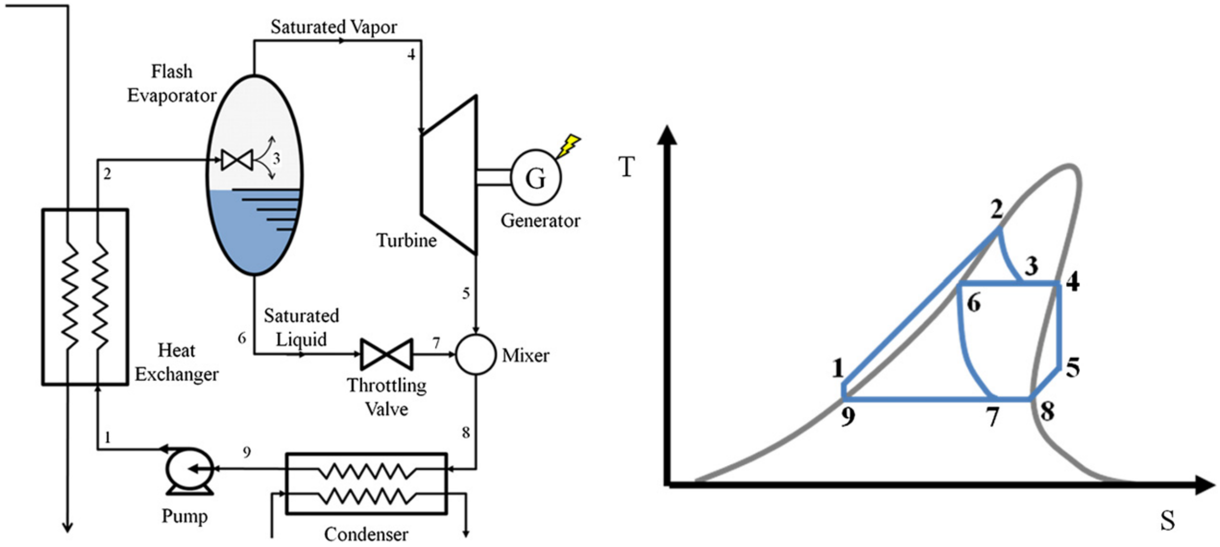

Fig. 1. System schematic and T–S diagram for the basic OFC [24].

overcome the exergy destruction introduced by the throttling valve in the flash evaporation process and the net effect was comparable power output for the basic OFC and optimized ORC. In this present study, a number of potential improvements to the basic OFC are proposed and analyzed for improving power production.

2. Enhancements to the basic OFC

From the previous analysis [24], it was observed that a major source of irreversibility and exergy destruction in the basic OFC resulted from the flash evaporation process (state 2 → 3 in Fig. 1) and the liquid throttling process (state 6 → 7). These two processes respectively, caused about 13% and 6% of the total initial theoretically available work in the finite thermal energy source stream to be destroyed for aromatic hydrocarbon working fluids. In the following sections, several modifications to the basic OFC are proposed to mitigate the exergy destroyed by these two processes.

2.1. The double flash OFC

The motivation of the double flash OFC is similar to that of the well known double flash steam cycle in geothermal energy. By splitting the flash evaporation process into two steps instead of only one, more of the fluid is vaporized and consequently, more of the fluid can be expanded for power production [17]. Fig. 2 shows the system schematic for the double flash OFC and its corresponding T–S diagram. Note that the expansion process now occurs in two stages, one at a high pressure after the first flash evaporation step (state 4 → 5 in Fig. 2) and a secondary expansion stage occurs at a lower, intermediate pressure after the second flash evaporation step (state 8 → 9). Geothermal studies have shown that by introducing a secondary flash step, the double flash steam cycle can generate 15%–20% more power than the single flash steam cycle for the same geofluid [17].

2.2. The modified OFC

The schematic and T–S diagram for the modified OFC is shown in Fig. 3. From the preceding analysis [24], results showed that the "drying" nature of the organic working fluids caused a substantial degree of superheat at the turbine exit, particularly for siloxanes. Siloxanes are molecularly complex, which have been shown to result in less positively sloped saturated vapor curves on a T–S diagram and correspondingly more superheat after expansion from a saturated vapor state [31]. The modified OFC was designed with this observation in mind; the working fluid is first flashed to a two-phase mixture (state 2 → 3 in Fig. 3). Similar to the basic OFC, the two-phase mixture is separated (state 3 → 4 and 3 → 6), the saturated vapor is expanded through the high pressure turbine to an intermediate pressure (state 4 → 5), and the saturated liquid is throttled to the same intermediate pressure (state 6 → 7). The improvement to the basic OFC occurs next where the superheated vapor (state 5) and saturated mixture (state 7) are combined in the mixer to produce a saturated vapor at the intermediate pressure (state 8), which is then expanded to the condenser pressure producing power.

There are three distinct advantages to this design; the first being more of the fluid goes through the expansion process to produce work. In the basic OFC, saturated liquid separated after the flash evaporation step is throttled to the condensing pressure and never used to produce work and the energy it possessed was essentially lost. In the modified OFC, the saturated liquid does produce work after it recombines with the high pressure turbine exhaust and is then expanded in the low pressure turbine (state 8 → 9). The second advantage that stems from this design is that the fluid is less superheated at the low pressure turbine exit; this can be seen more clearly in the T–S diagram of Fig. 3. Expansion to the condenser pressure from a saturated vapor at a lower pressure (e.g. state 8) will produce a state less superheated than expansion from a saturated vapor at a higher pressure (e.g. state 4). Effectively, the excess superheat due to expansion of a "dry" fluid has been used to vaporize more fluid and generate more work. Also, from Carnot considerations, the thermal efficiency of the cycle will increase because heat is now being rejected at a lower temperature since the fluid is at a lower temperature prior to the condenser (state 9). These two advantages should allow for decreased exergy destruction in the condenser and throttling valve compared to the basic OFC. The third advantage of this design is that the fluid is flashed to a lower quality which results in the fluid being at a higher temperature and pressure prior to the high pressure turbine. This also results in reduced exergy destruction in the flash evaporation process since the separated liquid will still be used to produce power in the low pressure turbine.

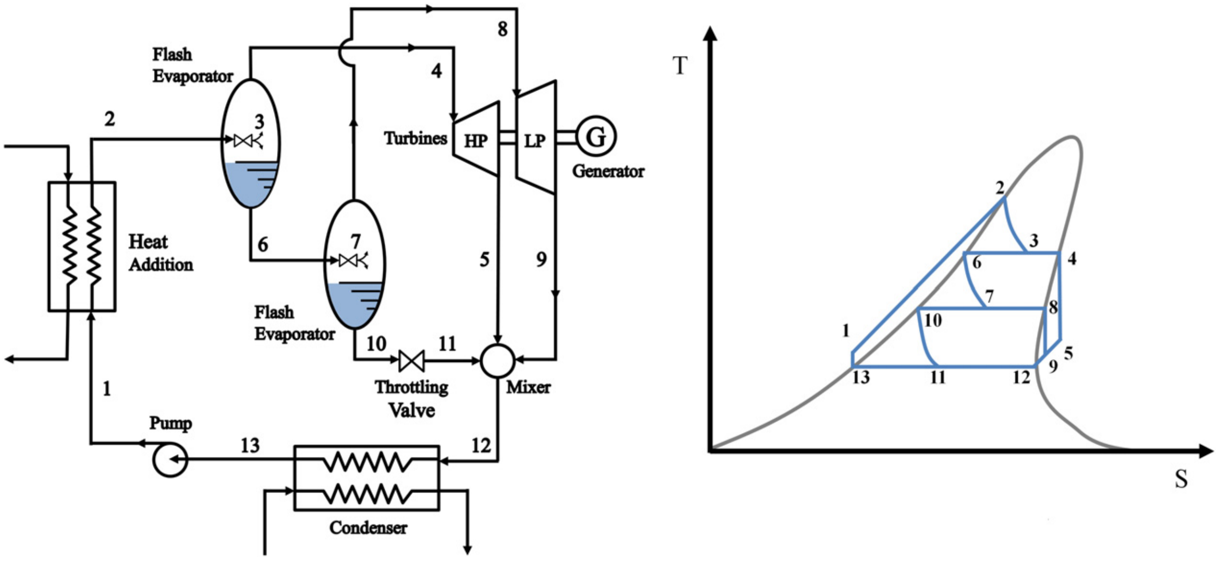

Fig. 2. System schematic and T–S diagram for the double flash OFC.

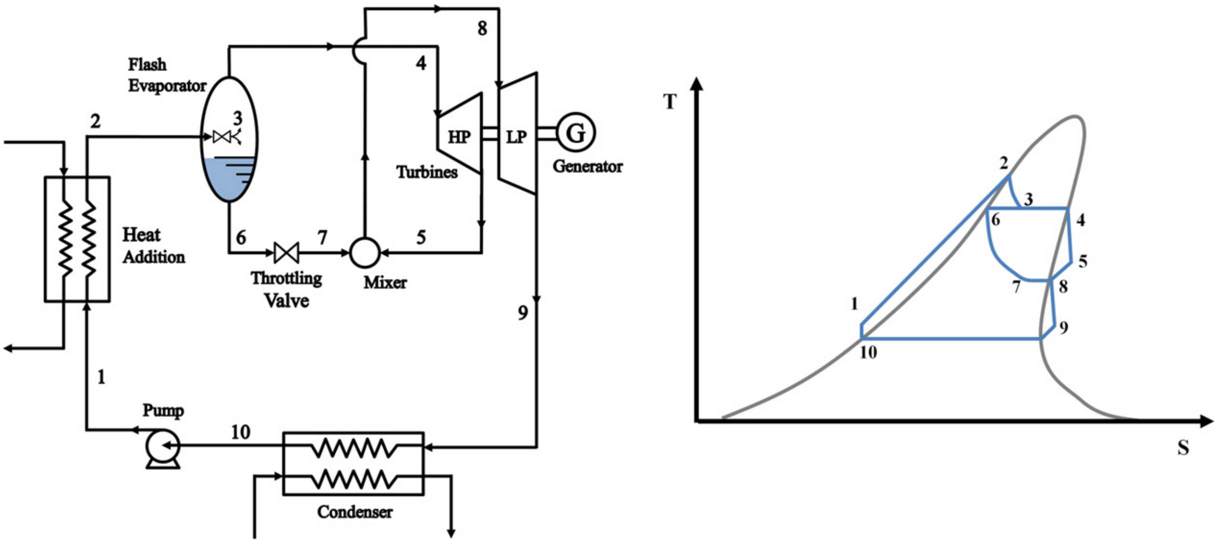

Fig. 3. System schematic and T–S diagram for the modified OFC.

2.3. The 2-phase OFC

Traditionally, the task of designing a reliable and efficient two-phase turbine has been very challenging because it requires the turbine to be able to handle a fluid with both liquid and vapor behaviors. Tailoring the turbine specifically to one phase or the other is thus not appropriate in this case which has made coming up with a suitable design difficult [30]. Presently, radial inflow turbine manufacturers have reported isentropic efficiencies of about 70% can be achieved reliably in the two-phase regime [32,33]. Significant advances have also been achieved recently for screw-type and scroll-type expanders [12,34]. Smith et al. have designed, built, and tested a twin-screw machine for two-phase expansion that achieved isentropic efficiencies of about 70%; the device is claimed to be both simple and cheap to manufacture [35]. The attractiveness of two-phase expanders is that they could potentially replace throttling valves, thereby producing useful work where a source of irreversibility had previously existed. In particular, the refrigeration [36] and LNG [37] industries are highly interested in this technology. In the proposed 2-phase OFC, the throttling valve (state $2 \rightarrow 3$ in Fig. 1) in the basic OFC is replaced with a two-phase expander. The system schematic and T–S diagram of the 2-phase OFC are shown in Fig. 4. The resulting cycle closely resembles the so-called “Smith Cycle” which used a n-pentane working fluid [33]. Results from Smith et al. showed increases in power output up to 30% over the conventional ORC for the same geothermal source [33]. Since two-phase expanders are still very much in the research and development phase, in the present analysis, the authors have assumed a range of isentropic efficiencies from 5% to 70% for a hypothetical two-phase expander.

2.4. The modified 2-phase OFC

Combining the enhancements introduced in section 2.2 and 2.3, the modified 2-phase OFC replaces the throttling valve in the flash evaporation process with a two-phase expander. It also uses two separate vapor expansion stages to de-superheat the exhaust from the high pressure turbine and generate more vapor to produce work. The system schematic and corresponding T–S diagram for the modified 2-phase OFC is shown in Fig. 5. This system is likely to produce the most power; however, it is noted that the system is the most complex of the enhancements suggested. There are a total of three expanders in the system: the two-phase expander and the high pressure and low pressure vapor turbines. The increase in power output will need to be weighed against the cost of additional equipment when determining the merits of the modified 2-phase OFC and the other enhancements to the OFC that have been proposed here.

3. Methods of analysis

3.1. Equations of state and thermodynamic/exergetic analysis

In the authors' previous work [24], a combination of modern equations of state was used to calculate working fluid thermodynamic properties. These equations of state included the semi-empirical BACKONE equations [38–40], the empirical, multi-parameter Span–Wagner equations [41–44], and the equations of state compiled in REFPROP 8.0 [45]. A brief overview is provided here; greater details on the applications of these equations of state can be found in the authors' previous works [24,46]. Helmholtz-explicit equations of state like the BACKONE and Span–Wagner equations can often be cast into the general form shown in eq. (1) [41],

$$\frac{a(T,\rho)}{RT} = \alpha(T,\rho) = \alpha^0 + \alpha^R \tag{1}$$

where a is the Helmholtz free energy, R is the universal gas constant, T is temperature, ρ is density, and α, α^0, and α^R are the total, ideal gas component, and residual component of the reduced Helmholtz energy. The ideal gas portion can be determined relatively straightforward if the ideal gas heat capacity is known. The residual portion though is more complex and can be cast into the general form shown in eq. (2),

$$\alpha^R(\tau,\rho) = \sum_{i=1}^{I_{pol}} n_i \tau^{t_i} \delta^{d_i} + \sum_{i=I_{pol}+1}^{I_{pol}+I_{exp}} n_i \tau^{t_i} \delta^{d_i} \exp(-g_i \delta^{p_i}) \tag{2}$$

where n_i, t_i, d_i, g_i, and p_i are constants dependent on the specific Helmholtz-explicit equation of state selected, δ and τ are respectively the reduced density and inverse reduced temperature with respect to the critical properties, and I_{pol} and I_{exp} are the number of polynomial and exponential terms, respectively [41]. The Span–Wagner equations use optimized empirical data to determine the coefficients in eq. (2), whereas the BACKONE equations use a combination of theoretical/physical basis and empirical data.

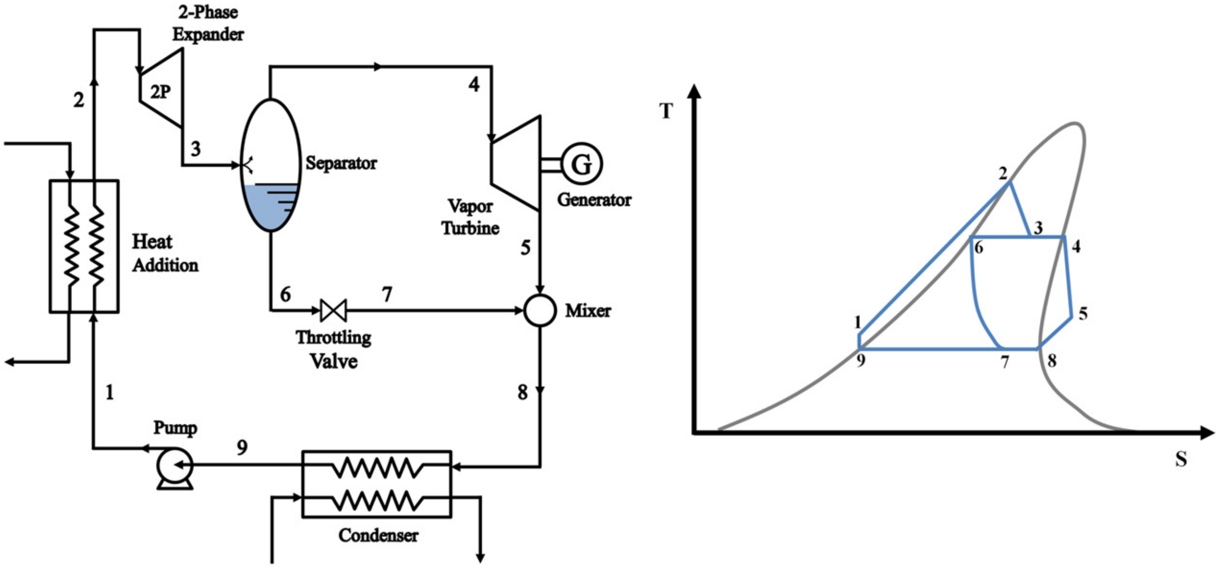

Fig. 4. System schematic and T–S diagram for the 2-phase OFC.

The appeal of the Helmholtz-explicit equations of state is that thermodynamic properties can be determined just from a combination of temperature, density, Helmholtz energy, and partial derivatives of Helmholtz energy with respect to temperature and/or density. The disadvantage however, is that complex iterative schemes are necessary when starting from known properties other than temperature and density. The thermodynamic properties derivations and common iterative schemes can be found in ref. [41]. NIST REFPROP 8.0 uses an equation of state explicit in Helmholtz energy, a modified Benedict-Webb-Rubin equation of state, or an Extended Corresponding States (ECS) model depending on the fluid of interest [45]. REFPROP, BACKONE, and Span–Wagner have all been utilized in a number of ORC studies and these equations of state have been shown to provide the necessary accuracy for technical analysis of ORC systems.

In power cycle analysis, exergy χ defined in eq. (3), is an important variable to quantify a fluid's usefulness and potential for producing work.

$$\chi = h - T_d s \tag{3}$$

In eq. (3), the subscript d designates the property at the dead state which has been taken to be the conditions of the surrounding environment which are assumed to be at 30 °C and atmospheric pressure. It has been assumed that no chemical reactions occur in the cycle and thus changes in chemical exergy are small and negligible. Note that in eq. (3), exergy is referring to only the thermo-mechanical exergy as the chemical exergy has been neglected. It is assumed the energy remaining in the source stream is lost to the environment after it has transferred heat to the power cycle. In this case, the goal is to maximize the power output from the power cycle per unit flow rate of the finite thermal reservoir. In eq. (4), a nondimensional power parameter termed the utilization efficiency is given,

$$\varepsilon = \frac{W_{\text{net}}}{\dot{m}_s(\chi_{s,\text{in}} - \chi_{s,d})} \tag{4}$$

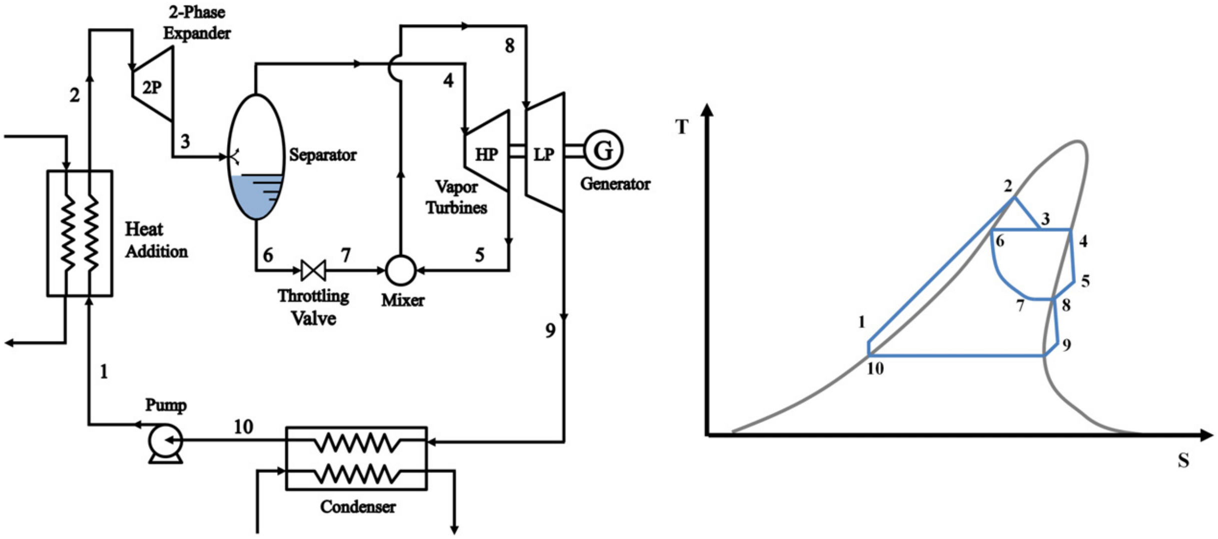

Fig. 5. System schematic and T–S diagram for the modified 2-phase OFC.

where $\dot{m}_s$ represents the source flow rate, $\chi_{s,in}$ is the source's inlet exergy, and $\chi_{s,d}$ is the source's exergy at the dead state. The utilization efficiency is essentially a ratio of the actual power produced to the maximum theoretical power initially available in the thermal energy reservoir. Note that this efficiency definition is preferred over the 1st law efficiency in waste heat applications as it penalizes the cycle if it is poor at absorbing energy from the waste heat stream whereas the 1st law rewards a cycle for using little "fuel." In waste heat applications however, the waste energy is to be lost to the environment if not absorbed by the cycle; therefore, consideration must also be given to how well the cycle absorbs energy from the waste heat stream.

Common simplifying assumptions such as neglecting frictional losses in the piping and heat exchangers have been employed in this study [24,47]. It is also assumed that working fluid's kinetic and potential energies changes are small and negligible and that the cycle components are well insulated so they do not lose significant energy to the surroundings. Throttling valves are modeled as isenthalpic devices; mixing chambers and separators are assumed to be isobaric and adiabatic with no work interactions. Vapor turbines and feed pumps are assumed to have 85% isentropic efficiency. A thermodynamic model for the OFC and the proposed enhancements was constructed using MATLAB; the equations of states were also programmed into MATLAB to determine thermodynamic properties of the 10 working fluids considered in this study. These fluids are shown in Table 1 with the respective equation of state that was used also listed.

3.2. Optimization of the OFC with proposed enhancements

For the basic OFC, optimization entailed determining the conditions at state 3 of Fig. 1 that would yield the greatest power output. The "equal-temperature-split-rule" was shown to be a good initial estimate for state 3 [17] for use in an iterative algorithm for the optimized basic OFC [24]. For the double flash OFC, an extra degree of freedom makes the optimization more complex; however, the "equal-temperature-split-rule" again provides a good initial estimate to start the simultaneous iterative functions to determine the optimal states for state 3 and state 7 of Fig. 2. For the modified OFC, the fluid is set to a saturated vapor state before entering the low pressure turbine so that the fluid is minimally superheated prior to the condenser. The optimization procedure for the modified OFC is again more complex than the basic OFC because of the added degree of freedom in selecting the intermediate pressure between the high pressure turbine inlet and the condensing pressure. For the 2-phase OFC, the number of degrees of freedom is similar to that of the basic OFC. The modified 2-phase OFC also possesses an additional degree of freedom than the basic OFC; similar to the modified OFC, simultaneous iterative functions for state 3 and the intermediate pressure at state 5,7, and 8 of Fig. 5 are required to determine the optimal conditions for maximum power output. The proposed cycles were optimized with iterative functions available in the MATLAB optimization toolbox.

Table 1
List of working fluids analyzed for OFC, ORC, zeotropic, and transcritical vapor cycles.

Fluid name	T_{crit} [K]	Equation of state and reference
Toluene	591.75	REFPROP [47]
Ethylbenzene	617.20	BACKONE [40–42]
Butylbenzene	660.05	BACKONE [40–42]
o-Xylene	630.33	BACKONE [40–42]
m-Xylene	617.05	BACKONE [40–42]
p-Xylene	616.23	BACKONE [40–42]
Tetradecamethylhexasiloxane (MD4M)	653.2	Span-Wagner [43–46]
Decamethylcyclopentasiloxane (D5)	619.2	Span-Wagner [43–46]
Dodecamethylpentasiloxane (MD3M)	628.36	Span-Wagner [43–46]
Dodecamethylcyclohexasiloxane (D6)	645.78	Span-Wagner [43–46]

4. Results and discussion

The proposed enhancements to the OFC were analyzed for the 10 aromatic hydrocarbon and siloxane working fluids that had been suggested in the authors' previous study [24]. A finite thermal energy source in the form of a 1 kg/s stream of water available initially at 300 °C is again assumed as it is representative of the previously discussed energy sources such as industrial waste heat and single-axis solar thermal technology. A heat sink is assumed to be available that allows for a minimum temperature of 40 °C for the saturated liquid at the exit of the condenser. A minimum 10 °C temperature difference is assumed necessary to properly transfer energy in the heat exchangers; the mass flow rate of the power cycle is adjusted to the maximum value that produces maximum power while still maintaining a 10 °C pinch.

Fig. 6 shows the resulting utilization efficiencies for the optimized basic ORC, OFC, double flash OFC, modified OFC, and 2-phase OFC and modified 2-phase OFC both with a 70% isentropic 2-phase expander. Fig. 7 shows the resulting percent increases or decreases in power output for the OFCs compared to the optimized ORC, which had been analyzed in detail in ref. [24]. From Fig. 6, it is observed that the modified 2-phase OFC produced the greatest power from the same finite thermal sources of all the cycles analyzed. This was true for all 10 working fluids examined; in addition, it can be seen that the proposed enhancements to the OFC had a much bigger effect for aromatic hydrocarbon working fluids than for siloxanes.

Table 2 and Table 3 give the conditions at several important states for the optimized basic OFC and double flash OFC respectively. Note the bottom caption of Table 2 that the basic OFC and OFC with enhancements have the same optimal T_2 and same mass flow rate as the proposed enhancements only occur after heat addition; therefore, they do not affect the near perfect temperature matching to the finite thermal reservoir. The main advantage of double flash OFC is the generation of additional vapor for power

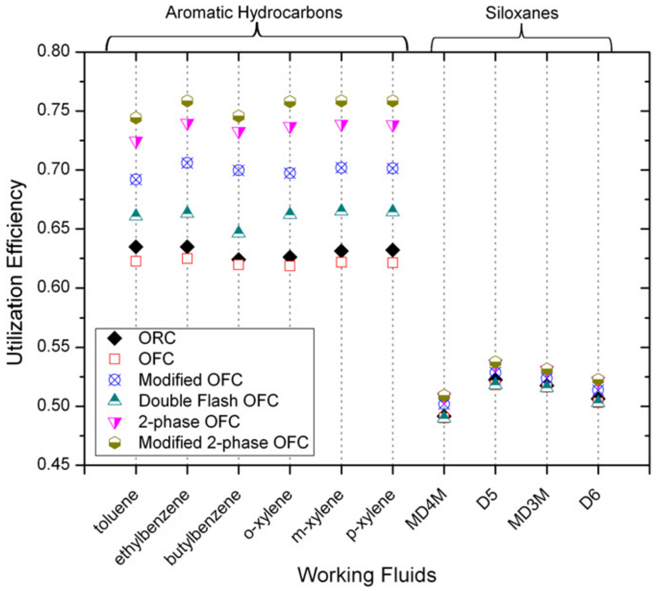

Fig. 6. Utilization efficiencies for the optimized ORC and the OFC with proposed enhancements.

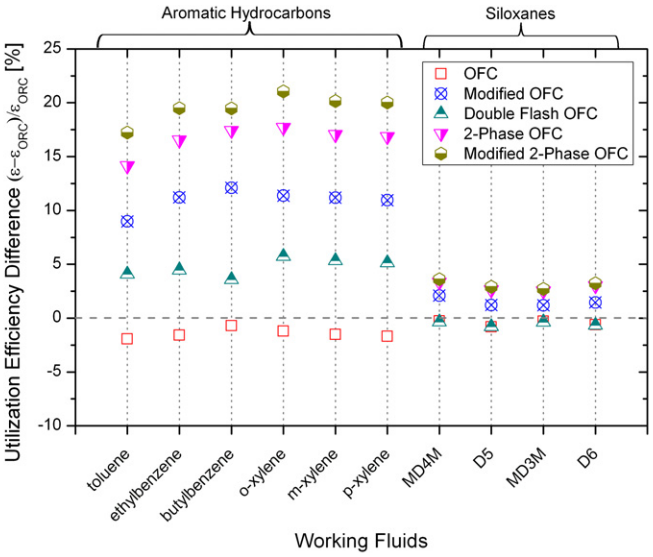

Fig. 7. OFC percent difference in power output compared to the optimized ORC.

production. Comparing Table 2 and Table 3, it can be seen that for aromatic hydrocarbon working fluids, the vapor produced is at a higher temperature but lower quality than for the basic OFC. Although the total amount of fluid expanded are similar for the single flash OFC and double flash OFC, more power is produced because the first flash step results in a higher exergetic state. Results showed the exergy destruction from the flash evaporation processes decreased by approximately 20–30% for the double flash OFC compared to the single flash OFC. In theory, additional flash steps could be added on, but the improvements in efficiency diminish with each additional step and any more than a secondary flash would not be cost-justified [17].

From Table 3, it is noted that for siloxanes, maximum power output was achieved for conditions where the working fluid was completely flash evaporated into a saturated vapor in the first step. In other words, for maximum power output for siloxanes, all the flow should be vaporized at step 3 and sent through the high pressure turbine. The optimal double flash OFC for siloxanes actually defaults to the optimized single flash OFC. This was a significant difference from aromatic hydrocarbons which showed greater power output with two flash steps. For siloxanes, their vapor dome on a T–S diagram is actually very narrow due in part to their high molecular complexity and very “drying” behavior; therefore, the net change in specific entropy between saturated liquid and saturate vapor states is much smaller for siloxanes than for aromatic hydrocarbons. This property causes substantially less exergy to be destroyed during isenthalpic throttling from a saturated liquid to a saturated vapor for siloxanes. At lower temperatures, the vapor dome broadens and the exergetic efficiency of the isenthalpic throttling begins to decrease. In a double flash OFC for siloxanes, less power would actually be produced because no additional vapor is actually being produced with two flash steps compared to a single flash OFC for siloxanes; the two flash steps would actually be less efficient than a single one for this particular type of working fluid. Results for the single flash OFC actually showed that flash evaporation only contributed to approximately 4–5% of total exergy destruction/losses. As noted in the authors’ previous study [24], the majority of the exergy losses for siloxanes occur in the condenser because of their highly “drying” behavior that causes substantial superheat and high temperatures at the turbine exit (~190 °C for siloxanes compared to ~70 °C for aromatic hydrocarbons). Energy recovery from the turbine exhaust in the form of an additional power cycle or combined heat and power would be necessary for a siloxane OFC or ORC in order to not waste energy at this high of a temperature. In addition, siloxane working fluids may face challenges in the practical design of an inexpensive and simple turbine due to potentially very high turbine volumetric flow ratios which have been discussed in ref. [24].

Results from Fig. 6 and Fig. 7 show that the modified OFC can produce significantly more power than the double flash OFC. The modified OFC is also more attractive than the double flash OFC in terms of system simplicity because a second flash evaporator is not required. Table 4 gives some important state points for the optimized modified OFC. Note that when comparing Table 3 and Table 4, the temperature of state 3 and 7 for the double flash OFC are similar to the temperature of state 3 and 8 for the modified OFC. This suggests similar exergy content in the fluid prior to entering the high pressure and low pressure turbines. However, the vapor quality at state 7 for the double flash OFC is significantly smaller than that for the saturated vapor at state 8 for the modified OFC vapor ($q_8 = 1$). The modified OFC configuration produces more

Table 2
Important state points for the basic OFC (from Fig. 1) [24].

Fluid name	T_2^a [K]	T_3 [K]	q_3	$\dot{m}_{OFC}$ [kg/s]
Toluene	559	457	0.87	1.92
Ethylbenzene	561	453	0.89	1.90
Butylbenzene	563	452	0.94	1.85
o-Xylene	562	452	0.86	1.90
m-Xylene	561	453	0.87	1.92
p-Xylene	561	453	0.87	1.93
MD4M	557	507	1.00	2.31
D5	558	509	1.00	2.43
MD3M	552	502	1.00	2.33
D6	557	507	1.00	2.46

[a] State point 2 for the optimized basic OFC is a saturated liquid ($q_2 = 0$). This value for T_2 and $\dot{m}_{OFC}$ is applicable to the basic OFC and to the OFC with the proposed enhancements.

Table 3
Thermodynamic states for key points for the double flash OFC (from Fig. 2).

Fluid name	T_3 [K]	q_3	T_7 [K]	q_7
Toluene	494	0.66	410	0.56
Ethylbenzene	489	0.68	407	0.56
Butylbenzene	486	0.72	405	0.58
o-Xylene	489	0.64	406	0.54
m-Xylene	489	0.65	407	0.55
p-Xylene	490	0.65	407	0.55
MD4M	507	1.00	–	–
D5	509	1.00	–	–
MD3M	502	1.00	–	–
D6	507	1.00	–	–

Table 4
Thermodynamic states for key points for the modified OFC (from Fig. 4).

Fluid name	T_3 [K]	q_3	P_8^a [kPa]	T_8^a [K]
Toluene	495	0.65	180	405
Ethylbenzene	494	0.64	109.9	412
Butylbenzene	500	0.61	47.6	428
o-Xylene	488	0.64	60.9	399
m-Xylene	491	0.65	79.6	403
p-Xylene	491	0.65	81.2	403
MD4M	539	0.54	65.5	514
D5	540	0.55	195.9	514
MD3M	540	0.55	133.7	515
D6	539	0.54	91.3	513

[a] State 8 corresponds to a saturated vapor state ($q_8 = 1$).

power because all the flow is expanded through the low pressure turbine. In addition, less energy is lost in the condenser because the fluid is less superheated at the low pressure turbine exit and energy from the separated saturated liquid after flash evaporation is also utilized to produce power. This was the original motivation for the modified OFC as discussed in section 2.2.

In Fig. 6 and Fig. 7, a two-phase expander with a 70% isentropic efficiency was assumed for the 2-phase OFC and modified 2-phase OFC. This isentropic efficiency corresponds to results given by other two-phase expander studies [32,33,35]. To examine the two-phase expander efficiencies necessary for viability in the OFC though, Fig. 8 and Fig. 9 show a range of hypothetical two-phase expander isentropic efficiencies for the 2-phase OFC and the resulting utilization efficiency and percent increase in power over the optimized ORC. Note that the two-phase expander isentropic efficiency is of greater importance to aromatic hydrocarbons than siloxanes for improved performance. As mentioned previously, exergy destruction in the flash evaporation process only amounted to about 4–5% of total exergy destruction/losses for siloxanes. For siloxanes, there is less potential exergy to be recovered by replacing the throttling valve with a two-phase expander. From Fig. 9, a 2-phase expander of at least a 10–15% isentropic efficiency is necessary for an aromatic hydrocarbon 2-phase OFC to generate the same power as the optimized ORC. For a siloxane 2-phase OFC, a 2-phase expander of at least a 30% isentropic efficiency is necessary to reach the same power output as an optimized ORC. In practice, the necessary two-phase isentropic expander efficiencies would need to be significantly higher than the previously listed values to justify the cost and complexity. DiPippo suggests that isentropic efficiencies of at least 75% are necessary before practical implementation of two-phase expanders into binary geothermal plants is feasible [48].

Combining the advantages of the modified OFC and the 2-phase OFC, the modified 2-phase OFC showed the greatest power output of the proposed enhancements in this study. For aromatic hydrocarbon working fluids, the modified 2-phase OFC produced approximately 76% of the theoretically available power initially in the finite thermal energy source. For the same finite thermal energy source, the modified 2-phase OFC produced approximately 20% more power than the optimized conventional ORC. Although this cycle does generate substantially more power, this needs to be evaluated in respect to the additional complexity and equipment costs. The modified OFC could be an attractive compromise between high power output and additional equipment costs. By only adding an additional low pressure turbine to the basic OFC, a 10%–12% increase in power output compared to the optimized ORC is achievable for aromatic hydrocarbons. Using a high level, order of magnitude analysis in a previous study [46], it can be shown that the optimized basic OFC requires similar heat transfer surface area in the heat exchangers as the optimized basic ORC. For ORC systems, heat exchangers by far make up a majority share of the initial system costs [49,50] and have been used as a measure of initial capital cost in thermoeconomic evaluations such as the one conducted by Schuster et al. [51]. The heat exchangers for the modified OFC would also be less expensive than the basic OFC because more power is being produced which reduces the total heat rejection rate in the condenser and subsequently decreases the necessary heat transfer area for the condenser.

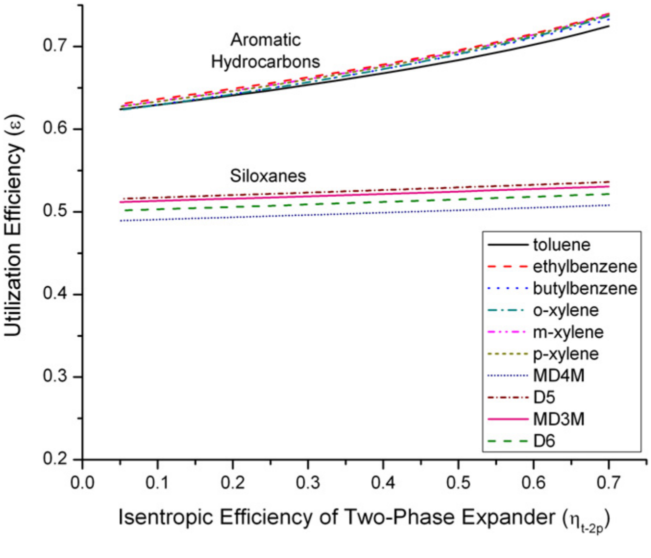

Fig. 8. 2-Phase OFC utilization efficiencies for varying 2-phase expander isentropic efficiency.

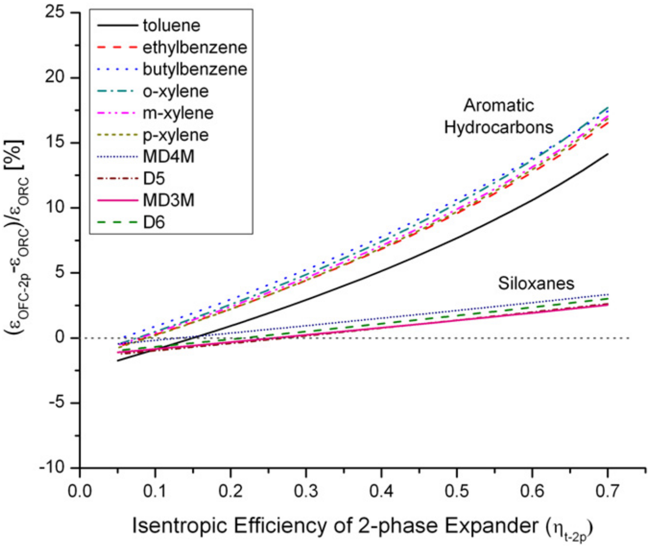

Fig. 9. 2-Phase OFC percent difference in utilization efficiency compared to the optimized ORC for varying 2-phase expander isentropic efficiency.

5. Conclusions

Several enhancements to the basic Organic Flash Cycle (OFC) were proposed to improve power output from a specific flow rate of a given finite thermal energy reservoir. Some of the sources of inefficiency in the basic OFC configuration included irreversibilities generated by the flash evaporation process and the high superheat at the turbine exit. The BACKONE, Span–Wagner, and REFPROP equations of state were used to model the working fluid thermodynamic properties. Ten aromatic hydrocarbon and siloxane working fluids were reexamined for the OFC with proposed enhancements such as double flashing, two-phase expanders, and using energy in the separated liquid in a low pressure turbine which also reduced vapor superheat prior to the condenser. Results showed that the modified OFC using aromatic hydrocarbons can produce approximately 10% more power output than the optimized ORC. Combining this technique with presently available two-phase expander technology would yield approximately 20% greater power output than the optimized ORC for aromatic hydrocarbons. Results for the enhanced OFC for siloxanes were less promising as increases in power output ranged from only 2–4%. Based on results from this study, further research into other possible modifications of the OFC seems warranted for aromatic hydrocarbon working fluids. Further research on the OFC utilizing low temperature resources may also have potential applications in fields such as non-concentrating solar thermal energy.

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Nomenclature

a	Helmholtz free energy [J/kg]
c_p	Isobaric heat capacity [J/kg K]
d_i, g_i, n_i, p_i, t_i,	Equation of state specific constants
h	Enthalpy [J/kg]
I_{exp}	Number of exponential terms in equation of state
I_{pol}	Number of polynomial terms in equation of state
$\dot{m}$	Mass flow rate [kg/s]
q	Vapor quality
s	Entropy [J/kg]
T	Temperature [K]
W_{net}	Net power [W]

Greek symbols

α	Total reduced Helmholtz energy
α^0	Ideal Gas component of Helmholtz energy
α^R	Residual component of Helmholtz energy
δ	Reduced density
ε	Utilization efficiency
ρ_0	Fluid specific characteristic density [kg/m^3]
τ	Inverse reduced temperature
χ	Exergy [J/kg]

Subscripts and superscripts

d	Dead state
in	Inlet
l	Liquid
s	Source

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