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Off-design comparison of subcritical and partial evaporating ORCs in quasi-steady state annual simulations

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Abstract

The subcritical ORC (SCORC) is considered the industry standard due to its simple configuration, acceptable efficiency and low costs. However, it is known that alternative ORC configurations have the potential to increase efficiency. A cycle modification which closely resembles the SCORC is the partial evaporating ORC (PEORC), where a two-phase mixture of liquid-vapour enters the expander instead of superheated vapour. In theoretical studies at design conditions, higher power outputs are achieved for the PEORC compared to the SCORC. This work aims to go a step further by investigating the performance of the SCORC and PEORC under time-dependent operating conditions. A direct comparison between the SCORC and PEORC is made for identically sized systems using as input the waste heat stream of a waste incinerator plant and the changing ambient conditions. Performance maps of both cycle configurations are compiled and the benefit of an expander operating at variable speed is briefly discussed. The results indicate that for the specific case under investigation, the PEORC has an increased annually averaged net power output of 9.6% compared to the SCORC. Use of annually averaged input conditions results in an overestimation of the net power output for both the SCORC and PEORC, and furthermore, the relative improvement in power output for the PEORC is reduced to 6.8%. As such, the use of time-averaged conditions when comparing cycle architectures should preferably be avoided.

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Keywords: organic rankine cycle; subcritical ORC; partial evaporating ORC; cycle architectures; simulation; quasi-steady state

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1. Introduction

Many of the applications where organic Rankine cycles (ORCs) are deployed experience time-varying heat stream characteristics. Examples of these mass and temperature variations can amongst others be found in the cement industry (drying processes) [1], transportation sector (mobile combustion engines) [2] and steel industry (electric arc furnaces, cokes ovens) [3, 4]. These variations add complexity to the assessment of the real power output of the organic Rankine cycle. Therefore, when comparing the well-established subcritical ORC (SCORC) to alternative thermodynamic cycles, this off-design operation needs to be included in the performance evaluation. While it is known that alternative ORC architectures can provide increased power output from the same waste heat stream, most of the studies in the literature are conducted assuming fixed design conditions [5-7]. This is also the case for the promising partial evaporating cycle (PEORC).

The PEORC is almost identical to the standard SCORC with the difference that two-phase liquid-water is supplied at the expander inlet. Performance increases of up to 11% are marked for medium temperature heat sources (250 °C) [6]. In addition, the changes in technology merely concern advancements in expander technology, the operation of other components remains identical to subcritical operation. Volumetric expanders are considered suitable for two-phase expansion, yet limited experimental results are available in literature [8]. For example, Smith et al. [9], performed measurements on twin-screw expanders with inlet vapour qualities from 50% to 25%. They showed isentropic efficiencies ranging between 40% and 80%, respectively. Li et al. [10] studied a rolling piston-type two-phase expander reporting isentropic efficiencies of 58.7%. A trend, indicating reduced isentropic efficiencies under two-phase expansion for high efficiency single phase twin-screw machines, was shown by Öhman and Lundqvist [11]. They also propose a simplified model that relates the performance between superheated and two-phase inlet conditions.

This work aims to fill the gap that the PEORC is solely investigated and compared at design conditions. To address this, quasi-steady state simulations are performed using as as input the waste heat profile of a waste incinerator plant and the varying ambient conditions. Detailed off-design component models were used to compile off-design performance maps for both the SCORC and PEORC. These models were previously calibrated and validated in another work by the authors [12]. Operation as PEORC is achieved by adapting the expander model equations according to information available in open literature.

Nomenclature

$\dot{m}$	mass flow rate, kg/s
N	rotational speed, RPM
p	pressure, Pa
$\dot{Q}$	heat transfer rate, W
T	temperature, °C
$\dot{W}$	power, W
x	vapour fraction, -
Abbreviations	
NTU	number of transfer units
ORC	organic Rankine cycle
PEORC	partial evaporating organic Rankine cycle
SCORC	subcritical organic Rankine cycle
Greek symbols	
ϵ	isentropic efficiency, -
Subscripts and superscripts	
cf	cold fluid
hf	hot fluid
in	Inlet
out	outlet
wf	working fluid

2. Simulation model and calibration

The modelling choices and equations are detailed in the PhD of Lecompte [12]. In the cited work, an extensive discussion on the solution strategy is provided. In this section, the modelling scheme is only briefly introduced. The implemented model of the heat exchanger is a hybrid approach between the finite volume model [13] and the moving boundary model [14]. The main benefit of the moving boundary model is the fast calculation time, while the finite volume model show increased accuracy for complex geometries and for fluids with thermophysical properties which cannot be assumed constant during heating while omitting phase transitions. Finite volume models however are at the expense of an increased computational time. Initially, the heat exchanger is discretised in $N = 58$ segments. If there is a phase transition, an additional segment is inserted. If $N = 0$ the model is equal to a moving boundary model. These segments are interconnected according to the geometry of the heat exchanger. The P-NTU [15] correlations are used to determine the heat transfer in each segment. The convective heat transfer correlations for plate heat exchanger are taken from: Martin [16] (single-phase), Han et al. [17] (two-phase evaporation) and Han et al. [18] (two-phase condensation).

The expander model is taken from Declaye et al. [19] and Lemort et al. [20], while the pump model uses characteristic curves from Manente et al. [21]. The extrapolation to two-phase inlet conditions is done according to Öhman and Lundqvist [11]; see Eq. 7 and Eq. 8. These equations relate the isentropic efficiency of superheated or saturated inlet conditions to the isentropic efficiency for two-phase expander inlet. The cut-off point is found at $\epsilon_{1-\mathrm{ph,pk}} = 0.6$. For lower values of $\epsilon_{1-\mathrm{ph,pk}}$ the isentropic efficiency for two-phase inlet is increased over the isentropic efficiency under single-phase inlet. They attribute this to a decrease in leakage, because of the sealing of leakage paths by the liquid phase. However, they state that the available test data in literature is scarce and that the actual behaviour of the two-phase expansion is unknown. In this work, the isentropic efficiency of the expander is always lower than 60% so we prefer to analyse the worst-case scenario without taking into account the possible sealing of leakage paths.

$$\epsilon_{2-\mathrm{ph,pk}} = \epsilon_{1-\mathrm{ph,pk}} + \psi_{2-\mathrm{ph}}(1 - x_{in})/10 \quad (1)$$

$$\psi_{2-\mathrm{ph}} = -0.15\epsilon_{1-\mathrm{ph,pk}} + 0.09 \quad (2)$$

The proposed models above are all semi-empirical and thus calibration on experimental data is required. An 11-kWe organic Rankine cycle set-up was used to perform experimental measurements to characterize the off-design performance. The experimental set-up is a scaled-down version of a real commercial ORC system designed for low heat source temperatures (between 80 °C and 150 °C) and uses R245fa as the working fluid. This set-up includes a twin-screw expander, centrifugal pump and plate heat exchangers for the evaporator and condenser. The calibration and subsequent validation is extensively described in the PhD of Lecompte [12]. Details about the experimental setup and the data reduction techniques can be found there. The sampling plan includes heat source mass flow rates in the range of 1.5 kg/s to 3 kg/s, cold water volumetric flow rates in the range of 7 m^3/h to 14.5 m^3/h and heat source temperature levels in the range of 110 °C to 120 °C.

The validation results show a closed heat balance of evaporator and condenser with a maximum deviation between secondary and primary heat flow rate of ±5%. The only input parameters to the cycle model are the pump and expander rotational speeds. The important dependent parameters are the evaporation pressure, the condensation pressure and the working fluid mass flow rate. All three predicted parameters show a maximum deviation of less than ±1% from the measured value. The modelled net power output deviates less than ±2% from the measured value. In general, this is a satisfactory result that gives confidence in using these models in further analysis. It is important to note that the validation has only been done for superheated vapour at the expander inlet. The extrapolation model of Öhman and Lundqvist [11] is used to predict the performance of the expander in two-phase flow regime.

3. Quasi-steady annual simulations

The PEORC and SCORC are compared based on quasi-steady state annual simulations. Steady-state operation is imposed at each moment but temporal variations of the boundary conditions are accounted for. In this work, the changing boundary conditions are the time-varying properties of the waste heat stream and cooling loop (i.e. changing ambient conditions). Scaled data from an ORC connected to a waste incinerator plant is used as the input (MIROM, Roeselare, Belgium). Corresponding profiles are shown in Figure 1, where missing data correspond to maintenance periods. It can be seen that during actual operation there are large fluctuations of the input boundary conditions.

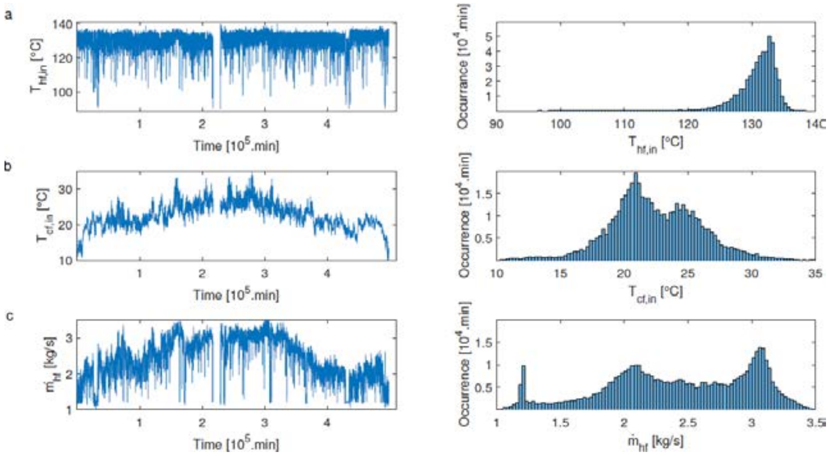

Fig. 1. Temporally varying boundary conditions used as inputs to the quasi-steady state simulations: a) $T_{\mathrm{hf,in}}$, b) $T_{\mathrm{cf,in}}$, and c) $\dot{m}_{\mathrm{hf}}$.

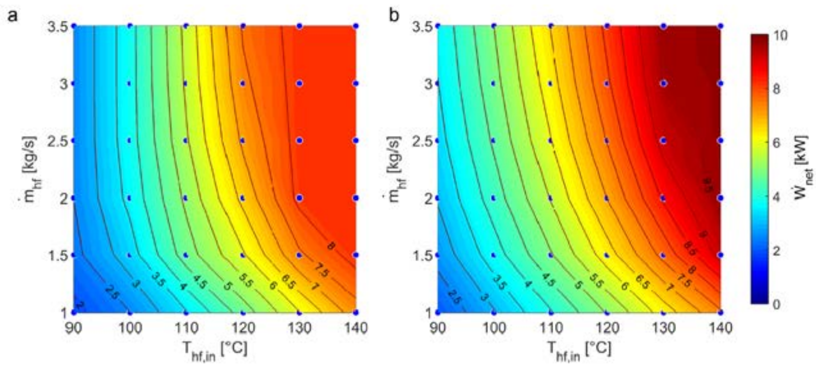

Fig. 2. Off-design performance maps at $T_{\mathrm{cf,in}}$ = 30 °C for: a) the SCORC, and b) the PEORC.

To efficiently characterize both ORC architectures, performance maps are compiled [22]. These maps are derived by linear interpolation from a full factorial experiment with three factors and six levels. The factors are the input boundary conditions $\boldsymbol{T}_{\mathbf{cf,in}}$, $\dot{\boldsymbol{m}}_{\mathbf{hf}}$, $\boldsymbol{T}_{\mathbf{hf,in}}$. The levels are respectively: [10, 15, 20, 25, 30, 35] °C, [1, 1.5, 2, 2.5, 3, 3.5] kg/s and [90, 100, 110, 120, 130, 140] °C. A constant volume flow rate of 13.5 m³/h water is imposed in the cooling loop of the dry cooler. An example of such a map for $\boldsymbol{T}_{\mathbf{cf,in}}$ = 30 ° is shown in Figure 2a for the SCORC and Figure 2b for the PEORC. The shape of the surfaces is fairly identical. However, for low temperature and high mass flow rate, the SCORC reaches a maximum platform. This is due to the isothermal evaporation which limits heat transfer to the cycle. Furthermore the PEORC clearly indicates increased net power output in all working conditions.

The effect of the third variable $T_{\mathrm{cf,in}}$ is shown in Figure 3 for both the SCORC and PEORC. Plots for five boundary conditions are made in order not to overload the figure. These points include the four corner boundary points from Figure 2 and a central point. For each of the operating points a similar trend is noted. While $T_{\mathrm{cf,in}}$ increases from 10 °C to 35 °C the net power output reduces the most under low heat input conditions (up to 68% for the SCORC and 67% for the PEORC) and the least under high heat input conditions (up to 33% for the SCORC and 30% for the PEORC). The explanation is found in the behaviour of the volumetric expander. For low heat input conditions, the evaporation pressure decreases and thus induces greater over-expansion losses. These losses increase further with increasing $T_{\mathrm{cf,in}}$. An improvement would be to have expanders with variable built-in volume ratio or to deploy variable speed expanders to increase the pressure in the system.

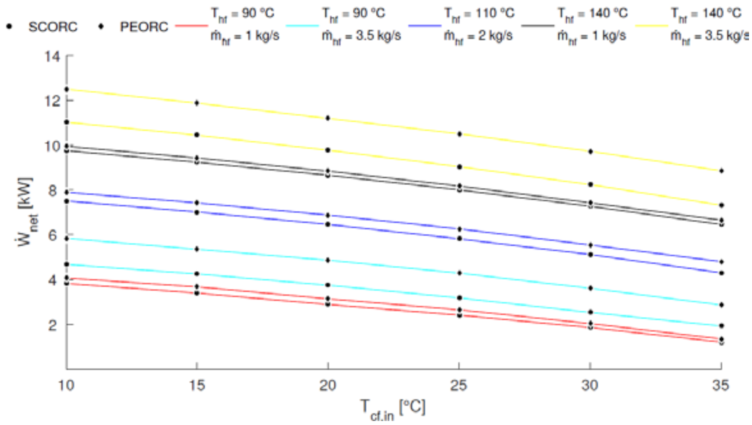

Fig. 3. Effect of $T_{\mathrm{cf,in}}$ on the net power output for the SCORC and PEORC with varying $T_{\mathrm{hf,in}}$ and $\dot{m}_{\mathrm{hf}}$.

Finally, full annual simulations are performed based on the above performance maps. Figure 4 shows the net power output of the SCORC and PEORC systems over the whole year. In this case, the increased mass flow rate of the hot fluid offsets the effect of a decrease in the net power output due to the increased cold fluid temperatures during the summer. Overall, the PEORC shows an increased net power output compared to the SCORC. The average net power output over the year for the SCORC is 9.08 kW compared to 9.96 kW for the PEORC. This corresponds to an increase of yearly generated net electricity of 9.6%. It is interesting to note that when working with time averaged profiles of $T_{\mathrm{cf,in}}$, $T_{\mathrm{hf,in}}$ and $\dot{m}_{\mathrm{hf}}$ the results are different. The net power output for the SCORC and PEORC are now respectively 9.39 kW and 10.03 kW. The power output during a year is thus overestimated for both the PEORC and SCORC but also the relative performance increase for the PEORC has been reduced to only 6.8%. It is therefore crucial to account for the specific temporal changes when comparing cycle architectures.

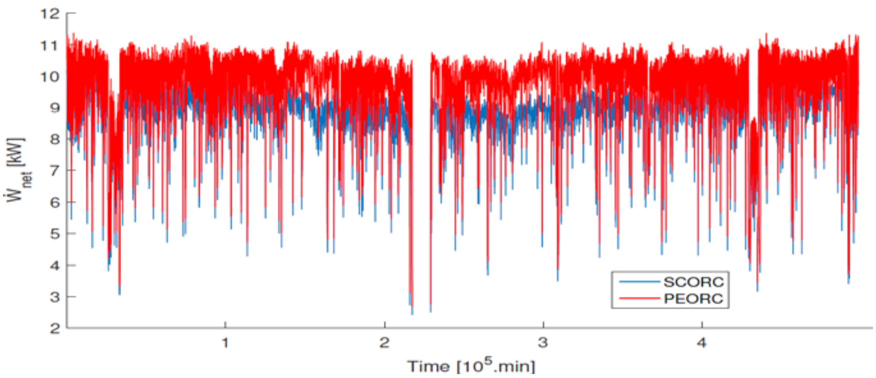

Fig. 4. Quasi-steady annual simulations showing the net power output for the SCORC and PEORC.

4. Conclusions

This work aimed to investigate the off-design operation of PEORC and SCORC systems in quasi-steady state simulations. A direct comparison between the SCORC and the PEORC is made for identically sized systems using as input the waste heat stream of a waste incinerator plant and the varying ambient conditions. Performance maps of both cycle configurations were compiled and the benefit of an expander operating at variable speed was briefly considered. The results indicate that for the specific case under investigation, the PEORC shows an increased annually averaged net power output of 9.6% compared to the SCORC. Furthermore, the use of annually averaged input values result in an overestimation of the net power output from both the SCORC and PEORC systems, while the relative improvement in power output predicted for the PEORC is reduced to 6.8%. As such, the use of time-averaged values when comparing cycle architectures should preferably be avoided.

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