

Off-design performance of a 1-kWe organic Rankine cycle (ORC) system

Chinedu K. Unamba^a, Xiaoya Li^{a,b}, Jian Song^a, Kai Wang^a, Gequn Shu^b, Hua Tian^b, Paul Sapin^a and Christos N. Markides^{a,}*

^a Clean Energy Processes (CEP) Laboratory, Department of Chemical Engineering, Imperial College London, London SW7 2AZ, UK

^b State Key Laboratory of Engines, Tianjin University, Tianjin 300072, China

* Corresponding author. Email: chinedu.unamba13@imperial.ac.uk

Abstract:

Several heat-to-power conversion technologies are being proposed as suitable for waste heat recovery (WHR) applications, including thermoelectric generators, hot-air (e.g., Ericsson or Stirling) engines, and vapour-cycle engines such as steam or organic Rankine cycle (ORC) power systems. The latter has demonstrated the highest efficiencies at low and intermediate scales and heat-source temperatures. However, ORC systems suffer a deterioration in performance at part-load or off-design conditions, and the high global warming potential (GWP) or flammability of common working fluids is an increasing concern. This paper presents the experimental investigation of a 1-kWe ORC test facility under time-varying heat-source conditions. It aims to compare the part-load performance of various architectures with different working fluids, namely: (i) R245fa, which is widely used in ORC systems, and (ii) low-GWP HFOs. The experimental apparatus is composed of a rotary-vane pump, brazed-plate evaporators and condensers, and a scroll expander with an adjustable load. An electric heater is used to provide a hot oil stream at three different temperatures: 80, 100 and 120 °C. The optimal operating conditions, i.e., pump speed and expander load, are determined for each architecture at various heat-source conditions. A maximum thermal efficiency of 2.8% is reported for a heat-source temperature of 100 °C, while a maximum net power output of 430 W is obtained for a heat source at 120 °C. An exergy analysis allows us to quantify the contribution of each component to the overall exergy destruction. The share of the evaporator, condenser and expander units remain major for all three heat-source conditions, while the exergy destroyed in the pump is negligible in comparison (below 4%).

Keywords:

Organic Rankine cycle (ORC), Experimental investigation, Part-load performance.

1. Introduction

A growing interest in improving energy efficiency through heat-to-power conversion technologies has been sparked by the escalating fuel prices and stricter environmental regulations. Organic Rankine cycle (ORC) systems are a particularly promising technology for power generation from waste-heat sources, especially for low- to medium-grade heat sources [1]. Substantial improvement has been made in ORC technologies in waste-heat recovery, solar thermal power, biomass power plants and geothermal plants. Commercial ORC systems have been successfully demonstrated for large-scale power plants by ORC manufactures such as Ormat, Turboden, BNI, UTC, Electratherm and Adoratec [6]. However, small-scale ORC systems are still in an embryo stage and no mature products appear on the market.

Several theoretical investigations are reported in the open literature, while the experimental studies on small-scale ORC systems that could generate real power are still limited. Lemort et al. [7] constructed an ORC prototype with R123 as the working fluid and focused on the performance of an open-drive oil-free scroll expander. Similar work was conducted in [8] while using R245fa as the working fluid. Pei et al. [1] reported preliminary dynamic testing results of a 3.75 kWe ORC system using R123 and a turbine. Turbine efficiency of 65% and ORC thermal efficiency of 6.8% were obtained. Li et al. [11] tested and compared a simple ORC system and a recuperative ORC system with R123 where a 10-kWe axial-flow single-stage turbine was included. In [12] and [13], a rolling-piston expander was proposed for a solar ORC system using R245fa; testing results exhibit an expander isentropic efficiency of 45.2% and an overall thermal efficiency of 4.2%. Zhang et al. [14] chose a single-screw expander for an ORC system for waste-heat recovery from a diesel engine and the maximum power output was 10.4 kWe.

As shown above, the viability of the ORC technology for small-scale energy systems is yet unproven. High-fidelity experimental data must be provided, both from laboratory-scale experiments and real-life systems, in order to assess the technical and economic potential of small-scale ORC systems. ORC systems suffer a deterioration in performance in part-load or off-design conditions, and the high global warming potential (GWP) or flammability of common working fluids is an increasing concern. Highly efficient expanders suitable for small-scale ORC systems are also a bottleneck constraint. The suitable working fluid, the expander and the cycle performance should be integrated and optimised. This paper presents the experimental investigation of a 1-kWe (i.e., about 10-kWth) ORC test facility under various heat-source conditions. The optimal operating conditions, i.e., pump speed and expander load, are determined for the basic architecture at each heat-source condition. The thermal efficiency of the system, and the effects of the expander load on its power are fully examined. These results will give references to the future real-time model-based control.

2. Experimental methods

2.1. ORC testing facility

The ORC test rig (Figure 1) is composed of a rotary-vane pump, a brazed-plate evaporator, superheater, recuperator and condenser, a scroll expander with an adjustable load and a generator. A schematic diagram of the rig is shown in Figure 2. Various architectures can be achieved by involving different combinations of the heat exchangers, namely simple ORC system, recuperated ORC system, simple ORC system with superheater and recuperated ORC system with superheater.

The working-fluid mass flow rate is varied by controlling the pump rotational speed. Different heat-source temperatures can be obtained by adjusting the electric oil heater and the expander rotational speed can be adjusted by varying the resistance connected to the generator, thus changing the expander load.

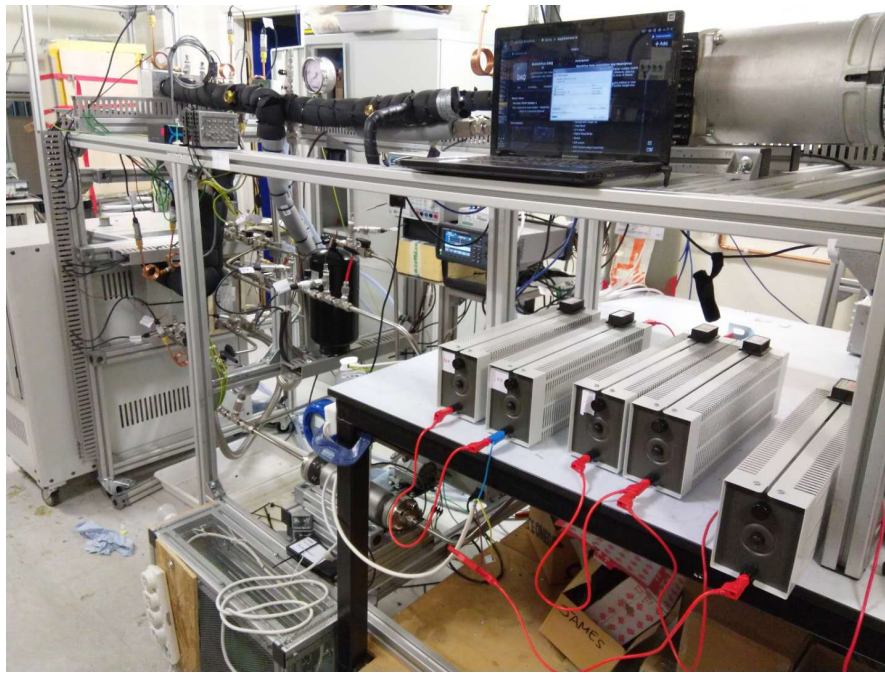


Figure 1. Picture of the ORC testing facility

The optimal operating point can be determined by adjusting the load, thereby making it possible to investigate real-time controls for time-varying heat-source conditions. In this article, a simple ORC system with the four basic components is considered and R245fa is selected as the working fluid. Table 1 summarises the operating conditions of the ORC rig, while the P&I diagram of the experimental apparatus is presented in Figure 2, where the actual cycle is highlighted in black, and the heating and cooling loops in red and blue, respectively.

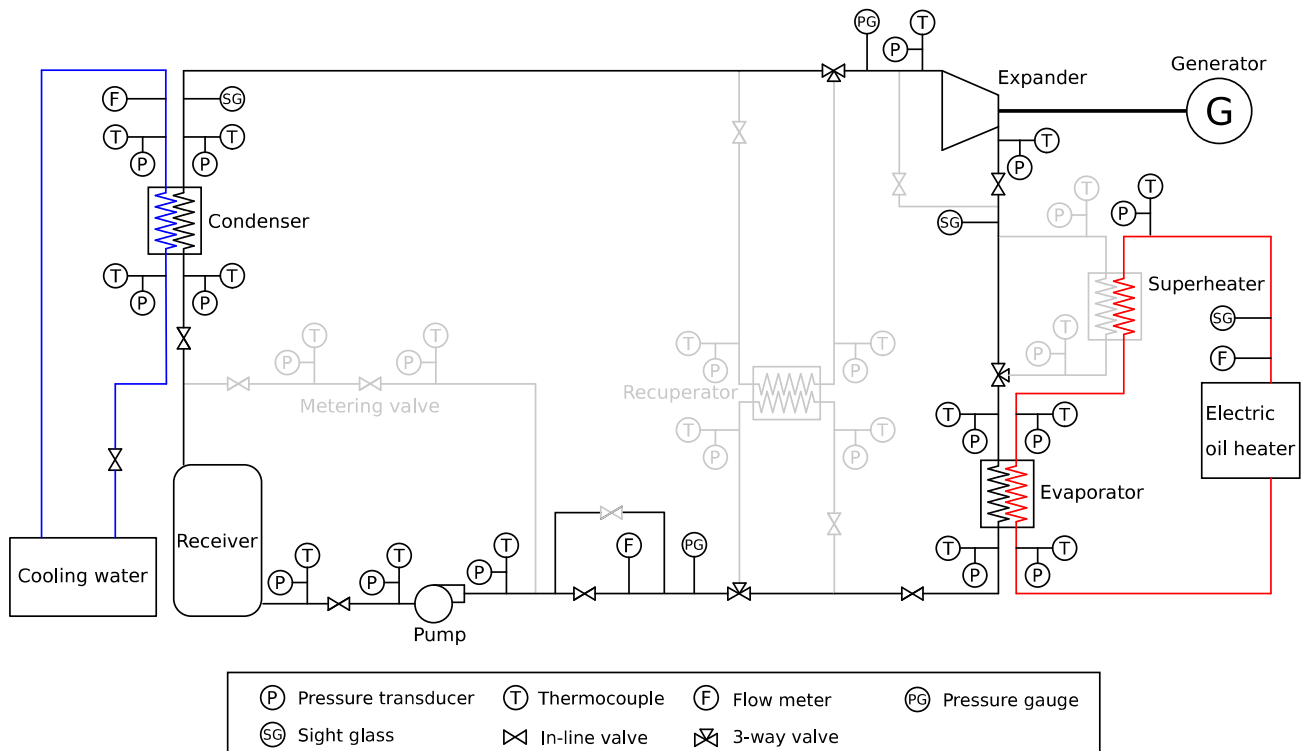


Figure 2. Piping and instrumentation (P&I) diagram of the ORC test facility.

Table 1. Operating conditions of the ORC testing facility.

Item	Properties
Working fluid	R245fa
Working fluid mass flow rate	0.04 – 0.06 kg/s
Maximum temperature operating temperature	150 °C
Maximum expander inlet pressure	13.8 bar (abs)
Expander built-in volume ratio	3.5

2.2. Heat-source and heat-sink conditions

An 18-kWth electric heater is used to provide a hot Marlotherm SM oil stream at temperatures up to 150 °C with a mass flow rate of 0.6 kg/s kept constant during the experiments. In this paper, the performance of the ORC engine is determined under varying heat-source temperatures, ranging from 80 to 120 °C. The excess heat is removed in a water-cooled condenser, with water supplied from the building tap at $T_{cw,in} = 18$ °C.

2.3. Data acquisition system (DAQ)

The ORC test facility is connected to a data acquisition system (DAQ, Keysight DAQ970) and relevant power supplies. A total of 21 pressure transducers (Omega PXM 309 with 4-20 mA output and 28-V excitation voltage) and 21 T-type thermocouples are used to monitor the fluid state at various positions, mainly the inlet and outlet of each component. 17 sensors are used to monitor the working fluid side, while the temperature in the hot and cold streams are recorded with the 4 remaining ones (see Figure 2). The mass flow rate of working fluid is measured with a high-precision Coriolis flowmeter (Krohne Optimass 6000), while the mass flow rate of oil and cooling water are measured with a turbine flowmeter and an Omega RTD flowmeter, respectively. A power analyser (Rohde & Schwarz HMC 8015) is used to measure the gross power output of the ORC system, i.e., the power generated by the scroll expander, while the power consumed by the pump is measured by an energy measurement device (Brennenstuhl PM 231E).

All the measurements are connected to the DAQ system and allow data acquisition for both steady-state and transient conditions. The sensors/devices installed can increase the data collection capability and capture various changes of the parameters such as pressure, temperature and other thermodynamic or transport properties, which enables us to better analyse the component and system performance.

3. Performance Indicators

The state of the working fluid throughout the cycle is determined from the pressure and temperature measurements, the thermophysical properties being calculated using CoolProp [15]. These data allow us to derive several performance indicators, both based on the system energy balance (first-law efficiencies) and on an exergy analysis (second-law analysis).

3.1. Overall ORC performance

As a heat engine, the first performance indicator of the ORC system is the net power output, $\dot{W}_{net}$, that is defined as the difference between the consumed power (i.e., the pumping power, $\dot{W}_p$) and the generated energy (i.e., the expander power, $\dot{W}_{exp}$):

$$\dot{W}_{net} = \dot{W}_{exp} - \dot{W}_p. \quad (1)$$

The primary energy source of a heat engine is, by definition, the heat source, which transfers energy to the working fluid in the evaporator. The thermal efficiency, η_{th} , is defined as the ratio between the net power output and the thermal input, $\dot{Q}_{evap}$:

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{evap}}, \quad (2)$$

where the heat input, $\dot{Q}_{evap}$, is calculated from a heat balance across the evaporator:

$$\dot{Q}_{evap} = \dot{m}_{wf}(h_{evap,out} - h_{evap,in}). \quad (3)$$

Finally, the rational performance of the system is reported using the exergy efficiency, η_{exg} , that is the ratio between the actual net useful power output to the ideal reversible power:

$$\eta_{exg} = \frac{\dot{X}_{out}}{\dot{X}_{in}} = \frac{\dot{W}_{net}}{\dot{X}_{in}}. \quad (4)$$

3.2. Thermodynamic losses in system components

The thermodynamic losses in the components of the ORC system are of various natures and complicated to decouple. The aim of this short analysis is not to explain the irreversibility sources in all the processes occurring through the cycle, but to quantify the contribution of each component to the overall exergy destruction, to identify the main paths for improvement. The exergy destruction rates in the 4 main components are obtained with Eqs. (5) to (8), presented below with the reference-point temperature $T_0 = T_{cw,in} = 18^\circ\text{C}$.

Expander exergy destruction rate

$$\dot{I}_{exp} = \dot{m}_{wf}[h_{exp,in} - h_{exp,out} - T_0(s_{exp,in} - s_{exp,out})] - \dot{W}_{exp}. \quad (5)$$

Evaporator exergy destruction rate

$$\dot{I}_{evap} = T_0[\dot{m}_{wf}(s_{evap,out} - s_{evap,in}) - \dot{m}_{hs}(s_{hs,evap,out} - s_{hs,evap,in})]. \quad (6)$$

Condenser exergy destruction rate

$$\dot{I}_{cond} = -T_0[\dot{m}_{wf}(s_{cond,out} - s_{cond,in}) - \dot{m}_{cw}(s_{cw,cond,out} - s_{cw,cond,in})]. \quad (7)$$

Pump exergy destruction rate

$$\dot{I}_p = \dot{W}_p - \dot{m}_{wf}[h_{p,out} - h_{p,in} - T_0(s_{p,out} - s_{p,in})]. \quad (8)$$

4. Results

As described in Section 2, the experimental facility presented in this paper can be operated in different configurations, namely with or without recuperator or additional superheater. In this paper, only the performance of the basic configuration (i.e., made of the 4 main components: pump, evaporator, expander and condenser) is investigated. Steady-state results are obtained for a set of 3 different heat-source temperatures, T_{hs} : 80, 100 and 120 °C, the aim being to investigate the off-design performance of the ORC engine. Two operating parameters – the pump speed and expander load – are varied, so as to determine the optimum operating conditions for each value of T_{hs} . These parametric studies and the corresponding results are presented in the first Section 4.1. The optimum configurations are then compared in Section 4.2, where the variations of the ORC performance are explained using both first-law and second-law analyses.

4.1. Influence of operating parameters

For given heat-source and heat-sink conditions, and at a given working-fluid mass flow rate, changing the load on the expander (referred to as the expander electrical load, R_{exp}) changes the torque applied to the expander shaft, thus influencing its rotational speed, which has an impact not only on the expander performance but also on the overall system as it impacts the pressure and temperature levels throughout the cycle. The loss mechanisms occurring in an ORC system are complicated to predict accurately with comprehensive models. In this article, the optimum working points of the system are determined empirically.

Figure 3 presents the measured net power output, $\dot{W}_{net}$, as a function of the load imposed on the expander for a heat-source temperatures of 100 °C. For each pump speed, an optimum expander load is determined, for which the net power output – that is the difference between the power output through the vapour expansion and the pumping power – is maximal. Figure 3 also shows an increase of the net power output for increasing pump speed, hence working fluid mass flow rate. Yet, increasing the mass flowrate further eventually leads to a decrease of the net power output, which can be observed from the performance map plotted in Figure 4 (a), from which the optimal working conditions, i.e., optimal pump speed and expander load, can be determined. The contour plots are made of isometric curves ranging from the lowest to the highest net power output measured with a set heat-source temperature of 100 °C. The maximum net power reported in Figure 4 (a) is 259 W, while the maximum net power outputs for the heat source at 80 and 120 °C are 162 and 430 W, respectively.

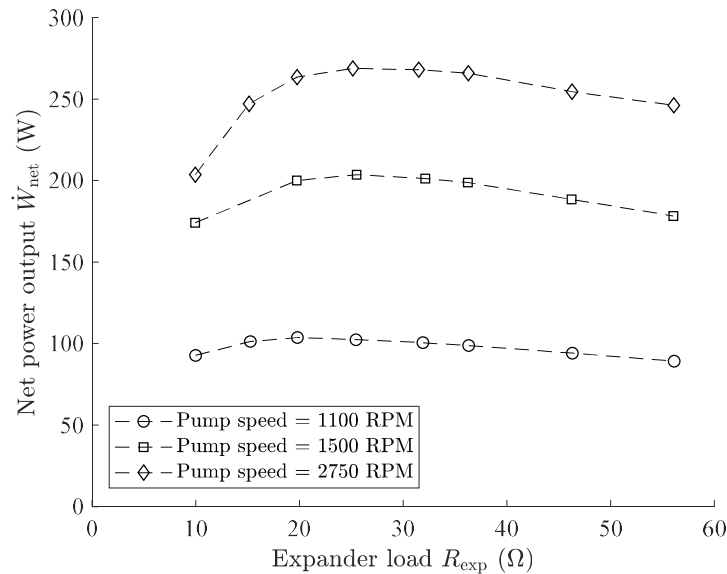


Figure 3. ORC net power output for various pump speeds and expander loads, for a heat-source temperature of 100 °C and heat-sink temperature of 18 °C.

A similar map is also obtained for the thermal efficiency, drawn in Figure 4 (b). The maximum thermal efficiency reached with a 100 °C heat-source is 2.8%, which corresponds to an exergy efficiency of 13.7%. It is worth noting that the operating conditions corresponding to the maximum net power differ from those that correspond to the maximal thermal efficiency, as a higher mass flowrate also increases the heat input.

The overall thermal (and exergy) efficiency is limited by the condenser cooling capacity and a high net positive pump suction head that was chosen to avoid detrimental cavitation effects, but which incurs large subcooling degree at the outlet of the condenser (18 °C for the measured optimum configuration in Figure 4 (b)), decreasing further the condenser and overall performance.

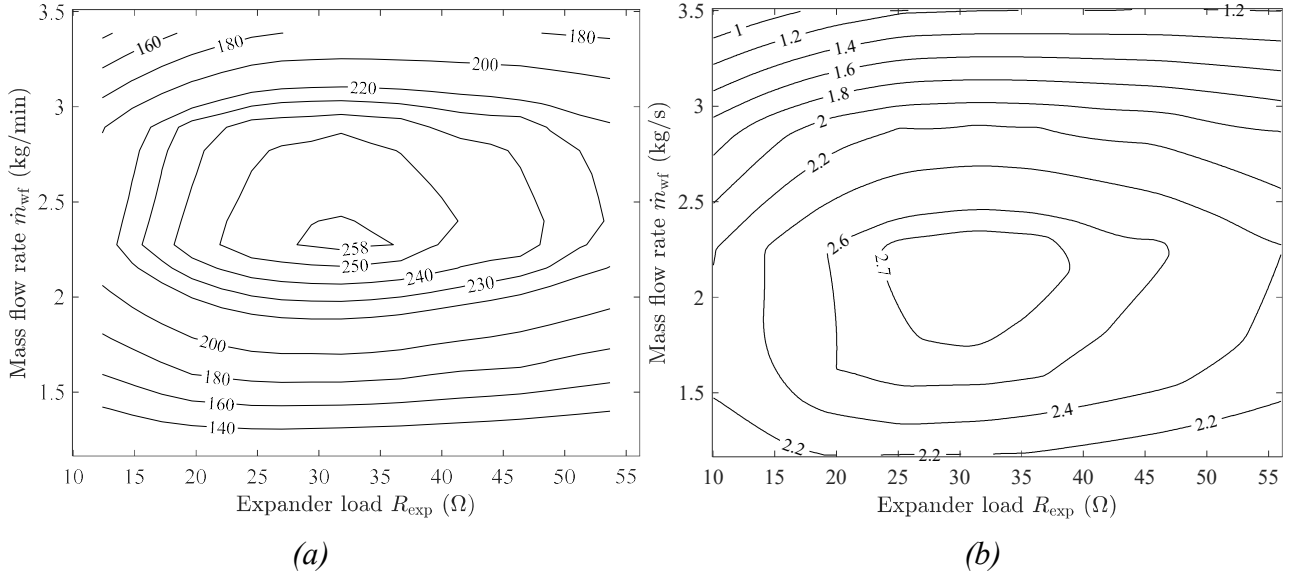


Figure 4. ORC (a) net power output (in W) and (b) thermal efficiency (in %) part-load performance maps, for a heat-source temperature of 100 °C and heat-sink temperature of 18 °C.

4.2. Part-load performance of the ORC engine

4.2.1. Thermodynamic diagram

The temperature-entropy (T - s) diagrams of the three optimum configurations determined for the different heat-source temperatures are shown in Figure 5.

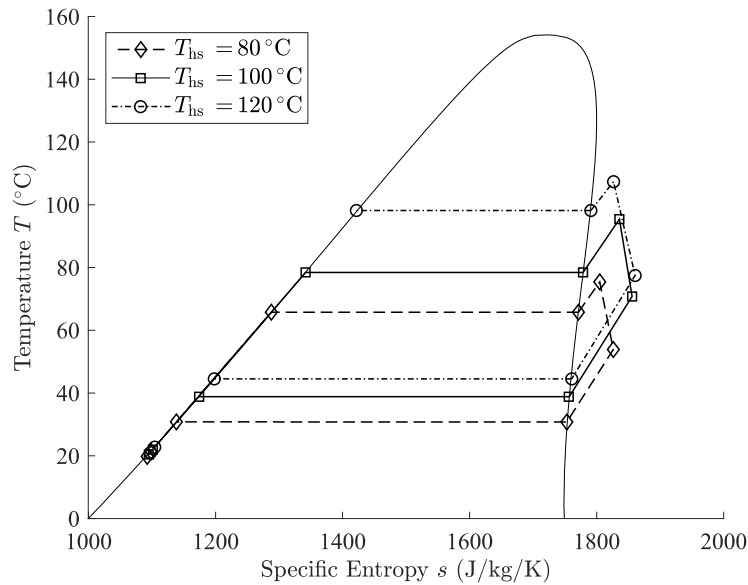


Figure 5. Temperature-entropy diagrams of the ORC engine operating at three different heat-source temperatures. Each cycle is obtained setting optimum operating parameters.

The first noticeable effect of the heat-source temperature increase is to increase the evaporating pressure, which should lead to an increase of the pressure ratio across the expander. However, the $T-s$ diagrams show that the condensing pressure increases also significantly with the heat-source temperature. This effect, detrimental to the overall performance of the engine, is due both: (i) to the limited capacity of the condenser to reject heat to the water stream, held constant at 18 °C during the experiments (the heat is released in the condenser at an increased pressure for higher heat-source temperatures, which increases both the mismatch between the temperature profiles and the exergy destruction rate), and (ii) to the 3.5 built-in volume ratio of the scroll expander that constrains the ratio between evaporating and condensing pressures.

The off-design performance of the ORC engine is thus expected to be significantly hindered by the limited condensing capacity, when increasing the heat-source temperature.

4.2.2. Thermal efficiency

The optimal thermal efficiency is reported in Figure 6 as a function of the working fluid mass flow rate for the three heat-source temperatures investigated in this study. First, it is worth noting that the overall performance is low, even at 100 °C, for which the global optimum was found, with a maximum thermal efficiency measured is 2.8%, while in similar conditions, endoreversible thermodynamics predicts a Chambadal-Novikov efficiency of 11% [16], which is an upper bound of the useful work that can be recovered from a heat source, accounting for the irreversibility of heat transfer. This is due to the many losses occurring throughout the cycle, both in components and in the piping. In other words, the possibility to switch between different ORC architectures was favoured, at the expense of system compactness, hence of system performance.

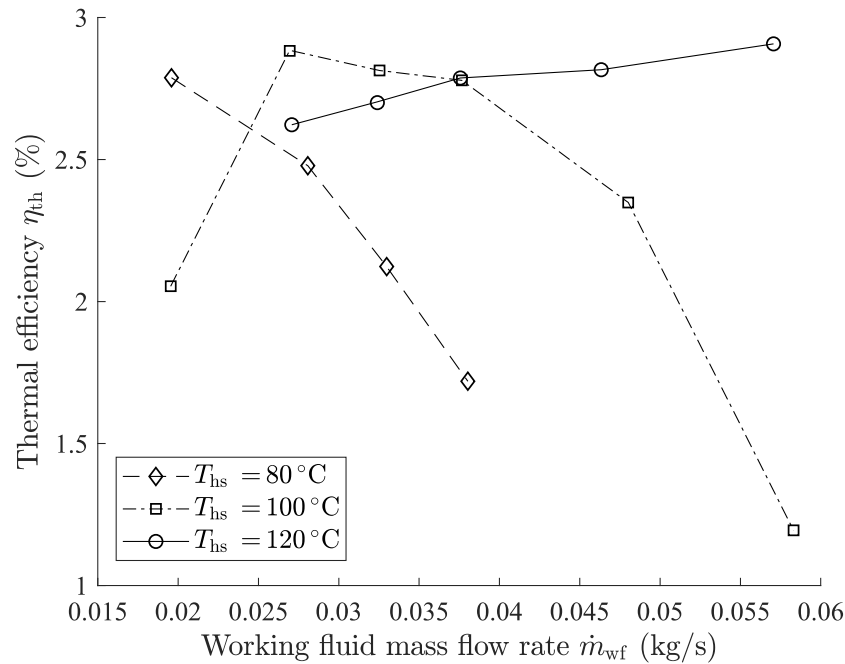


Figure 6. ORC thermal efficiency as a function of mass flow rate for various heat-source conditions.

4.2.3. Exergy destruction

The first-law analysis proposed in the previous subsection provides information at the system level. This subsection aims at determining the contribution of each component to the overall performance deterioration, using a second-law analysis. The exergy destruction in a single component quantifies the wasted work potential in a specific process (e.g., the gas expansion) due to thermodynamic irreversibility. In other words, it points out the lost opportunities to generate useful work.

Figure 7 reports the exergy destruction in each component for the three optima obtained at the three heat-source temperatures investigated. For an increasing heat-source temperature, it is observed that the expander, evaporator and condenser shares to the overall losses are predominant, while the contributions of exergy destruction in the pump is negligible in the three cases (which lies between 3

and 4%). The large proportion of the exergy loss in the evaporator and condenser units is due to the mismatch between the temperature profiles of the heat-transfer fluid (i.e., the Marlotherm SM oil in the evaporator and the cooling water in the condenser) and of the working fluid in the heat exchanger.

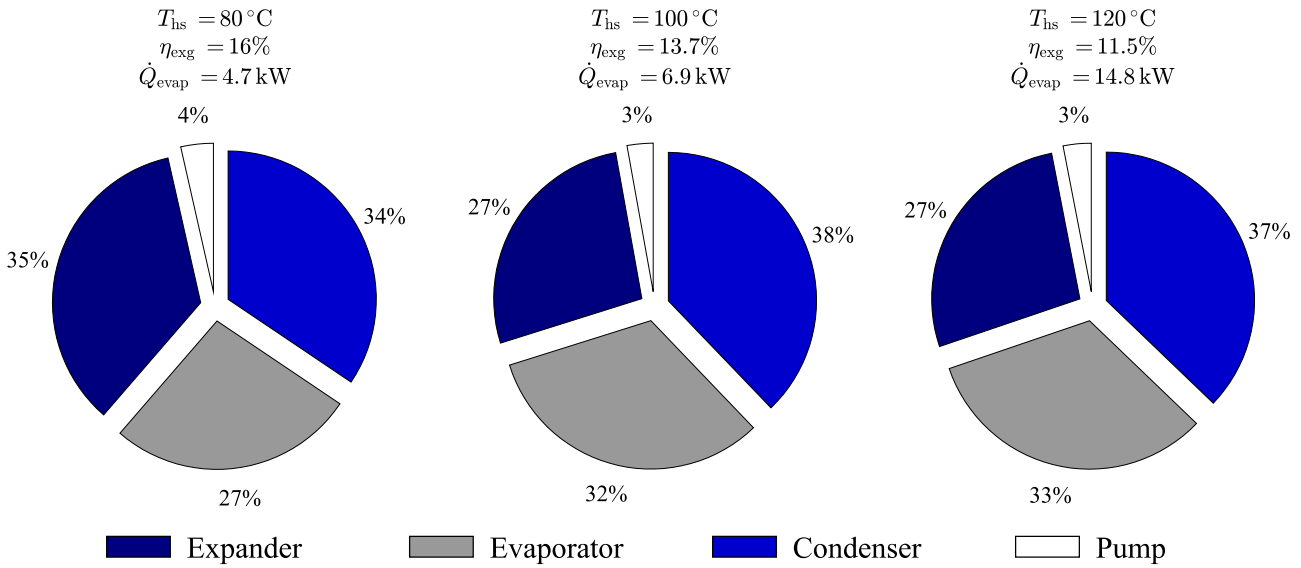


Figure 7. Exergy destruction pie charts showing the contribution of each component to the overall exergy destruction at different heat-source temperatures, with optimum operating parameters.

The expander contributes also significantly to the overall exergy destruction. The performance of scroll expanders is adversely affected by four main loss mechanisms that are: (i) radial and flank internal leakage, (ii) heat-transfer losses, (iii) friction losses, and (iv) under- or over-expansion losses that occur due to a mismatch between the built-in and actual pressure ratios across the expander [17]. For a pump speed set at 2500 RPM and a heat-source temperature at 100 °C (that is the point presented in Figure 7), the measured (electrical) expander isentropic efficiency is 28%, for a pressure ratio of 4.42 (while the expander has a built-in volume ratio of 3.5). A further investigation of the expander performance with varying mass flow rates, pressure ratio and superheat levels are required to determine the optimum operating point for this component.

5. Conclusions

A thermodynamic analysis of an ORC engine is presented, in which R245fa is used as the working fluid. Heat is extracted from an 18-kWth heat source and converted to work by a 1-kWe scroll expander coupled to a generator with a magnetic clutch, while rejecting heat through a water-cooled condenser. Neither the expander nor the integrated pump-motor unit require any lubrication, which avoids many oil management issues. In addition to the four main components (pump, expander, condenser, evaporator), the apparatus is fitted with an optional recuperator and superheater.

The ORC engine is operated at various pump speeds and the electrical load connected to the generator is varied in order to determine the best operating parameters for each heat-source condition, ranging from 80 to 120 °C (delivered at a constant oil mass flow rate of 0.6 kg/s).

System performance is determined based on two indicators. A maximum thermal efficiency of 2.8% is achieved for a heat source at 100 °C, and a maximum power output of 430 W is reported for a heat source at 120 °C. The ORC engine performance at off-design conditions is found to be significantly reduced, even when increasing the heat-source temperature.

An exergy analysis is conducted to determine the contribution of each component to the overall exergy destruction with optimum working conditions. It is shown that the evaporator, condenser and expander units remain the three main components where the exergy is destroyed, with contributions respectively ranging from 27 to 33%, 34 to 38%, and 27 to 35%.

A comprehensive model of all loss mechanisms in the different components is required to predict accurately and systematically the off-design and part-load performance of ORC engines. Generating high-fidelity data for such systems, such as the preliminary results presented in this article, is key to a further understanding of the significant loss mechanisms and their interactions. This experimental apparatus was built with this aim in mind, but also to allow future investigations of the transient performance of different architectures subject to time-varying conditions

Acknowledgments

This work was supported by the UK Engineering and Physical Sciences Research Council (EPSRC) [grant number EP/P004709/1]. Data supporting this publication can be obtained on request from cep-lab@imperial.ac.uk.

Nomenclature

h	specific enthalpy, J/kg
$\dot{I}$	exergy destruction rate, W
$\dot{m}$	mass flow rate, kg/s
$\dot{Q}$	heat flow rate, W
s	specific entropy, J/kg/K
T	temperature, K
$\dot{W}$	power, W
$\dot{X}$	exergy flowrate, W

Greek symbols

η	efficiency
--------	------------

Subscripts and superscripts

0	reference (or dead) state
cond	condenser
cw	cooling water
evap	evaporator
exg	exergetic
exp	expander
hs	heat source
in	inlet
net	net
out	outlet
p	pump
th	thermal
wf	working fluid

References

- [1] Pei G., Li J., Li Y., Wang D., Ji J., Construction and dynamic test of a small-scale organic Rankine cycle. Energy 2011;36:3215-23.
- [2] Freeman J., Hellgardt K., Markides C.N., An assessment of solar-powered organic Rankine cycle systems for combined heating and power in UK domestic applications. Applied Energy 2015;138:605-20.

- [3] Chatzopoulou M.A., Simpson M., Sapin P., Markides C.N., Off-design optimisation of organic Rankine cycle (ORC) engines with piston expanders for medium-scale combined heat and power applications. *Applied Energy* 2019;238:1211-236.
- [4] Ramos A., Chatzopoulou M.A., Freeman J., Markides C.N., Optimisation of a high-efficiency solar-driven organic Rankine cycle for applications in the built environment. *Applied Energy* 2018;228:755-65.
- [5] White M.T., Oyewunmi O.A., Chatzopoulou M.A., Pantaleo A.M., Haslam A.J., Markides C.N., Computer-aided working-fluid design, thermodynamic optimisation and thermoeconomic assessment of ORC systems for waste-heat recovery. *Energy* 2018;161:1181-98.
- [6] Kang S., Design and experimental study of ORC (organic Rankine cycle) and radial turbine using R245fa working fluid. *Energy* 2012;41:514-24.
- [7] Lemort V., Quoilin S., Cuevas C., Lebrun J., Testing and modeling a scroll expander integrated into an organic Rankine cycle. *Applied Thermal Engineering* 2009;29:3094-102.
- [8] Chang J., Chang C., Hung T., Lin J., Huang K., Experimental study and CFD approach for scroll type expander used in low-temperature organic Rankine cycle. *Applied Thermal Engineering* 2014;73:1444-52.
- [9] Wu Z., Pan D., Gao N., Zhu T., Xie F., Experimental testing and numerical simulation of scroll expander in a small-scale organic Rankine cycle system. *Applied Thermal Engineering* 2015;87:529-37.
- [10] Quoilin S., Lemort V., Lebrun J., Experimental study and modeling of an organic Rankine cycle using scroll expander. *Applied Energy* 2010;87:1260-8.
- [11] Li M., Wang J., He W., Wang B., Ma S., Dai Y., Experimental evaluation of the regenerative and basic organic Rankine cycles for low-grade heat source utilization. *Journal of Energy Engineering* 2013;139:190-7.
- [12] Zheng N., Zhao L., Wang X., Tan Y., Experimental verification of a rolling-piston expander that applied for low-temperature organic Rankine cycle. *Applied Energy* 2013;112:1265-74.
- [13] Wang X., Zhao L., Wang J., Zhang W., Zhao X., Wu W., Performance evaluation of a low-temperature solar Rankine cycle system utilizing R245fa. *Solar Energy* 2010;84(3):353-64.
- [14] Zhang Y., Wu Y., Xia G., Ma C., Ji W., Liu S., Yang K., Yang F., Development and experimental study on organic Rankine cycle system with single-screw expander for waste heat recovery from exhaust of diesel engine. *Energy* 2014;77:499-508.
- [15] Bell I.H., Wronski J., Quoilin S., Lemort V., Pure and Pseudo-pure Fluid Thermophysical Property Evaluation and the Open-Source Thermophysical Property Library CoolProp. *Industrial & Engineering Chemistry Research* 2014;53(6):2498-2508.
- [16] Novikov, I.I., The efficiency of atomic power stations (a review). *Journal of Nuclear Energy* 1958;7:125–128.
- [17] Lemort, V., Legros, A, Positive displacement expanders for Organic Rankine Cycle systems. In *Organic Rankine Cycle (ORC) Power Systems* (pp. 361-396). Woodhead Publishing. 2017.