EXERGO-ECONOMIC ANALYSIS OF WASTE HEAT ORC

Jovana Radulovic
†School of Engineering, University of Portsmouth,
Anglesea building, Anglesea Road, PO1 3DJ, Portsmouth
UK
e-mail: Jovana.Radulovic@port.ac.uk

ABSTRACT

Organic Rankine Cycle is one of the most promising solutions for utilisation of waste heat. Even though several devices have been operating successfully over the last decade, the technology is far from being widely implemented, with the cost being the primary hindrance. ORC typically yields low efficiency, and there is a number of safety concerns regarding working fluids and device operation.

In the current study we have thermodynamically simulated the performance of several working fluids in an ORC system powered by waste heat. Energetic and exergetic analyses, according to the first and the second principles, show expected system outputs depending on the thermo-physical fluid parameters. Power output, thermal and exergetic cycle efficiencies and exergy destruction in individual cycle components were evaluated for a range of operational parameters. Sensitivity analysis included variation of high cycle pressure, expander inlet temperature, isentropic efficiency of the expander and the pump, and the pinch point temperature differences in the evaporator and the condenser. Economic analysis was conducted for the considered cycle designs, linking energetic and exergetic results with the component cost rates.

INTRODUCTION

The dawn of the 21st century was marked by grave environmental concerns due to rising levels of pollution. Increase of the greenhouse gases in the atmosphere is widely accepted as the primary driver of the climate change, and a number of proposed polices to address pollution emissions are already implemented. The transport sector remains a major contributor to CO_2 footprint, and numerous attempts were made to enhance the IC engine fuel economy, as the efficiency of a conventional IC engine stands at approximately 35% [1]. One of the promising solutions to tackle low energy IC engine efficiency is utilisation of the heat rejected to the exhaust. Waste Heat Recovery (WHR) allows for harnessing of the thermal energy of the exhaust to power a thermodynamic cycle and lead to additional power and electricity generation. Organic Rankine Cycle (ORC) has long been recognised as a feasible technology for WHR.

ORC technology offers a number of advantages, including utilisation of low grade heat and operation at temperatures lower than those required for steam Rankine cycles, simple start-up and maintenance, and automatic operation [2]. A number of ORC systems have already been installed and successfully operating in recent years [3]. Nonetheless, thermal efficiency of ORC is commonly lower than that of a steam Rankine cycle and modest power outputs are barriers to wider implementation of the ORC technology.

ORC configurations, operational parameters and working fluids have attracted significant research attention is the last decade. The consensus of the scientific community that a single optimal ORC working fluid cannot be identified. Majority of studies published thus far conclude that the working fluid selection depends on number of parameters, primarily on the type of the heat source. Working fluids considered are often flammable and toxic substances. Further concerns include the environmental impact (ozone depletion potential, global warming potential), stability, compatibility with cycle components as well as the cost.

ORC systems are persistently the focus of scientific attention, especially for WHR applications [4]. Numerous systems for harnessing exhaust or cooling fluid heat have been designed and analysed [5]. Both pure fluids and zeotropic mixtures have been considered [6]. For 'on-board' systems for vehicular applications, which have been gaining significant research popularity [7], device compactness is crucial. However, overall design is dictated by the system complexity and the relative size of the cycle components.

NOMENCLATURE

i	[kJ/kg]	Specific irreversibility
q	[kJ/kg]	Specific heat
T	[K]	Temperature
w	[kJ/kg]	Specific work

Subscripts

B	Boiler
C	Condenser
E	Expander
ex	Exhaust
in	Inlet
out	Outlet
P	Pump
w	Water
0	Ambient or reference

NUMERICAL MODEL

A conventional ORC set-up was considered: pump, boiler (comprised of a preheater, an evaporator and a superheater), expander and condenser (including cooling of the fluid till saturation). The thermodynamic model was based on the following assumptions: a steady-state steady-flow; negligible kinetic and potential energy losses as well as heat losses in all components and pipes; no pressure drops apart from those in the pump and the expander; no fouling inside the heat exchanger.

Schematic of a simple ORC is presented in Figure 1. The pump inlet state was always assumed to be saturated liquid at the condenser pressure. Compressed liquid at high cycle pressure is heated isobarically till superheated vapour state. Maximum cycle temperature is reached at the expander inlet. Working fluid leaving the non-isentropic expander is cooled down isobarically in the condenser. Specific work and heats are calculated as enthalpy differences between relevant fluid states at the outlet and inlet of each cycle element. Assumed isentropic efficiencies for the pump and the expander are given in Table 1.

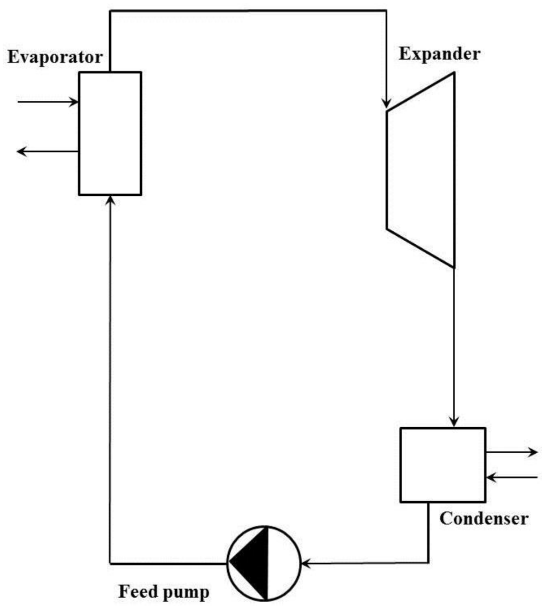


Figure 1 Schematic of a simple ORC

Modelled ORC system is powered by the waste heat of the diesel generator exhaust. We have based our analysis on Rolls Royce Field Electrical Power Supply 40 kW unit. The exhaust mass flow rate and inlet temperature were assumed according to manufacturer's specification, as stated in Table 1. Energetic and exergetic analyses are based on the first and the second principles. For a simple ORC, energy and exergy balances for the system are:

$$w_P + q_B = w_T + q_C \tag{1}$$

$$w_P + q_B\left(1-\frac{T_0}{T_{ex,in}}\right) = w_T + q_C\left(1-\frac{T_0}{T_{w,in}}\right) + i_{total} \tag{2}$$

where the temperatures of the heat source and the heat sink are approximated by the maximum temperature of the exhaust and the minimum temperature of the cooling water, respectively. Additionally, exergy balance was performed on individual ORC elements to determine the extent of exergy destruction in individual components. Detailed set of equations used is as stated in our previous works [8, 9].

Table 1 Modelling parameters assumed in the present study

Exhaust inlet temperature	673 K
Exhaust mass flow rate	0.045 kg/s
Turbine isentropic efficiency	70 %
Pump isentropic efficiency	60 %
Evaporator pinch point	20 K
Condenser pinch point	10 K
Dead state	273 K; 1 bar
Cooling water inlet temperature	278 K

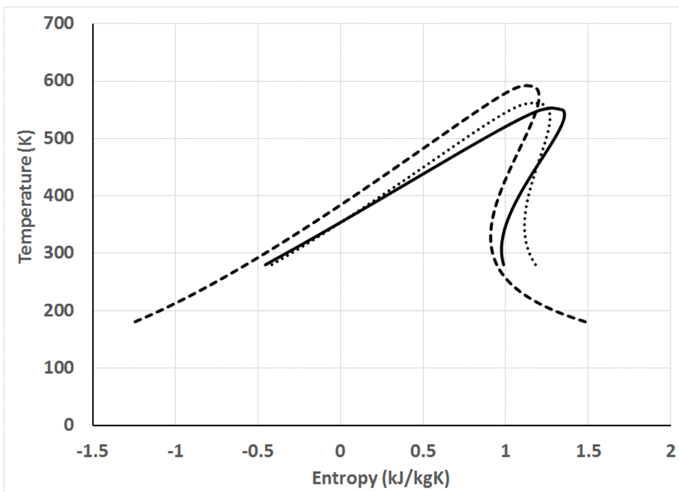


Figure 2 Saturation curves of the selected fluids: toluene (dashed line), benzene (dotted line) and cyclohexane (solid line)

The working fluid selection was based on principles outlined in the literature, including [3, 10, 11] and references within. As mentioned previously, fluid mixtures offer a number of advantages; however, in the present study only pure fluids were considered. Given the temperature of the exhaust at the inlet to the superheater, variations of the high cycle temperature were explored by altering the approach point temperature difference. A number of potential fluid candidates was considered and included in the preliminary analysis (omitted here for brevity). Results highlighted toluene, benzene and cyclohexane as promising candidates for high temperature ORC applications, which were selected for further study. Fluid properties are presented in Tables 2a and 2b. The saturation curves of the fluids are compared in Figure 2.

Notwithstanding the risk posed by toxicity and flammability of the proposed fluids, toluene is already successfully utilised in current ORC technologies. Effect of the maximum fluid temperature at the expander inlet was analysed in conjunction with high cycle pressure variations. High cycle pressure limit of 20 bars was imposed, as the maximum manageable parameter commonly mentioned in the literature [12].

Table 2a Basic thermodynamic properties of selected fluids

Fluid	*M*	T_b	T_c	p_c
	(kg/kmol)	*(K)*	*(K)*	*(bar)*
Cyclohexane	84.159	353.87	553.60	40.805
Benzene	78.112	353.22	562.02	49.073
Toluene	92.138	383.75	591.75	41.263

Table 2b Environmental and safety properties of selected fluids

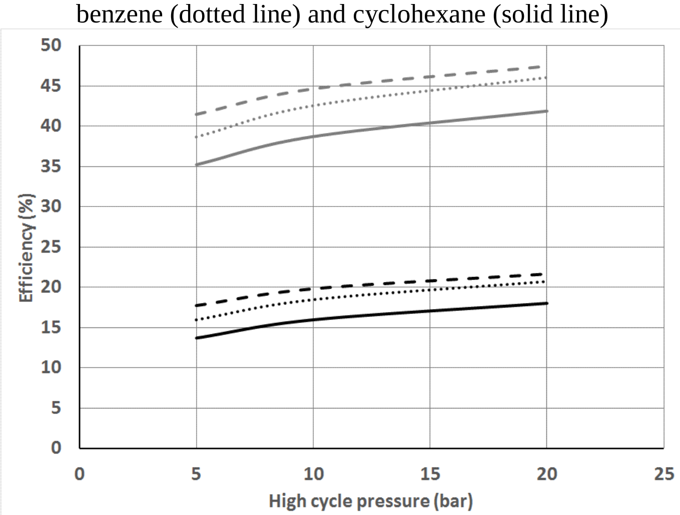
Fluid	*ODP***	*GWP****	*ASHRAE Class*****
Cyclohexane	0	Low	A3
Benzene	0	Low	B2
Toluene	0	Low	A3

** ODP: Ozone depletion potential, relative to R11;
*** GWP: Global warming potential, relative to CO_2;
**** ASHRAE Standard 34 – Refrigerant safety group classification. 1: No flame propagation; 2: Lower flammability; 3: Higher Flammability; A: Lower Toxicity; B: Higher Toxicity.

ENERGY AND EXERGY ANALYSIS

Variation of High Cycle Pressure

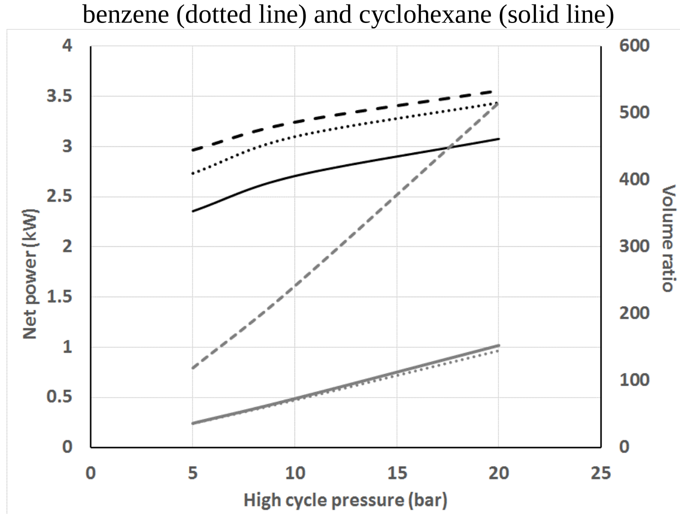
Figure 4 Thermal (black) and exergetic (grey) cycle efficiency as a function of high cycle pressure for toluene (dashed line), benzene (dotted line) and cyclohexane (solid line)



Total energy and exergy balances were performed for the system and thermal and exergetic efficiency calculated. Evaluated performance for the three fluids considered, assuming expander inlet temperature of 623K, are presented in Figure 4. Thermal performance of toluene was superior to that of the other fluids, with benzene reaching higher energetic efficiencies than cyclohexane. Analogous trends were observed for variations of the exergetic efficiency. For all fluids both energy and exergy efficiencies increased with high cycle pressures. Steeper rise of cycle efficiencies is notable between 5 and 10 bar. Between 10 and 20 bar, thermal and exergetic efficiencies increase at constant rates of approximately 0.2%/bar and 0.32%/bar, respectively. While gradients are largely similar, the greatest step increase of efficiencies were found for benzene, while the lowest percentage change of the efficiencies was in the case of toluene. Aromatics showed advanced performance compared to the cycloalkane, with thermal and exergetic efficiencies being generally above 15% and 40%, respectively, across the range of high pressures considered.

From the known mass flow rate of the exhaust stream a detailed analysis of the heat exchanger was performed and the total power output of the considered cycles was evaluated, presented in Figure 5. The trends are comparable to the cycle thermal performance, with toluene achieving larger net power output than the other two fluids. The shape of the power curves is also similar to the efficiency graphs, showing a linear rise of the net power with the high cycle pressure. It is worth noting that cyclohexane showed greater incremental increase of the net power with pressure increase of 0.037 kW/bar.

Figure 5 Net power (black) and expander volume ratio (grey) as a function of high cycle pressure for toluene (dashed line), benzene (dotted line) and cyclohexane (solid line)



Notwithstanding the importance of the net power and thermal and exergetic efficiencies, other elements have to be taken in account when assessing the feasibility of an ORC system. The size of the expander is a crucial metric, which has a significant effect on the overall price of the module. We have evaluated the expander volume ratio (ratio of specific volumes of the working fluid at the outlet and inlet expander states). As expected, high cycle pressure has drastic effect on the expander size, as shown in Figure 5. For all fluids, the volume ratio increased linearly with the high cycle pressure across the tested range. However, hugely different values were found for toluene compared to the other two fluids, with the gradient being almost four times higher per bar of pressure increase. Volume ratio for benzene and cyclohexane expanders were almost identical, with that for benzene being marginally lower. Whilst toluene has over-performed other fluids in all aspects of thermal and exergetic efficiency, as well as the cycle power production, remarkably high volume ratio is an issue. It would be beneficial to have a smaller expander from both design and cost perspective and volume ratio of toluene expander is a concern.

Figure 6 Total irreversibilities as a function of high cycle pressure for toluene (dashed), benzene (dotted) and cyclohexane (solid)

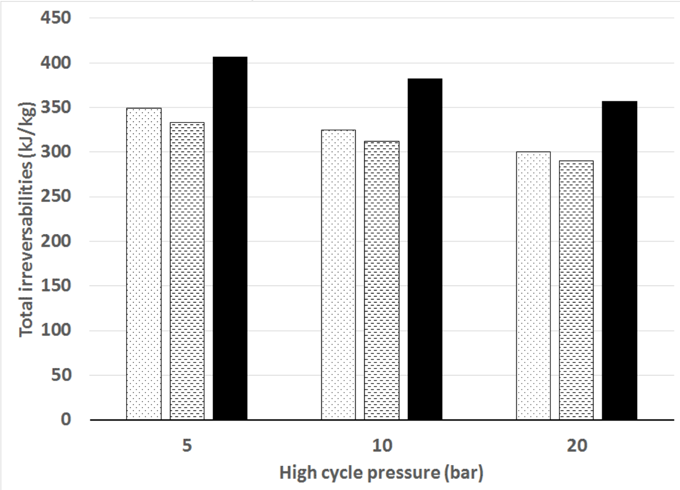
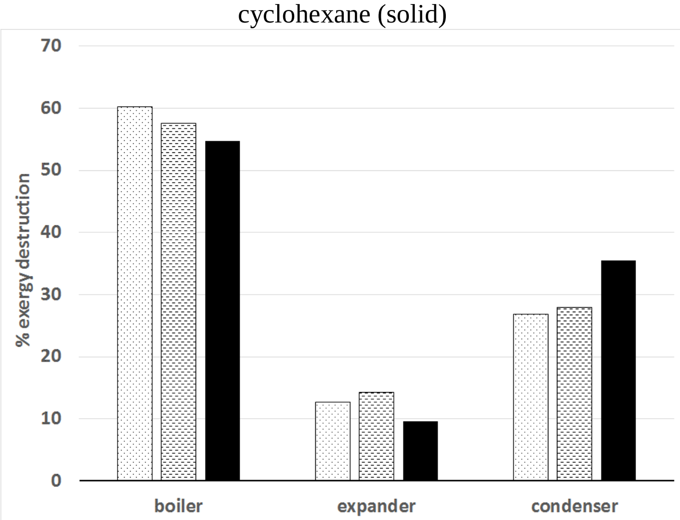


Figure 7 Percentage of exergy destruction in individual cycle elements for toluene (dashed), benzene (dotted) and cyclohexane (solid)



In addition to the highest exergetic efficiency of all fluids tested, total cycle irreversibilities were lowest for the toluene cycle. Slightly higher irreversibilities were found for benzene. All fluids show reduction in irreversibilities with increase in high cycle pressure. Interestingly, both benzene and cyclohexane irreversibilities decrease at the same rate of 2.5 kJ/kg/bar within the 10-20 pressure range. Furthermore, exergy destruction analysis was performed for individual cycle components, with results presented in Figure 7 for systems operating at high cycle pressure of 10 bar. Pump exergy destruction was found to be negligible for all systems, typically around 0.24% of total irreversibilities, and was therefore omitted from the graph. As expected, heat addition in the boiler is the largest contributor to exergy destruction, in the range of 50-60% for all fluids. The largest boiler exergy destruction was observed for benzene and the lowest for cyclohexane. Conversely, the greatest exergy destruction in the condenser was found for cyclohexane, while the lowest contribution to irreversibilities was seen for benzene condenser. Interestingly, toluene expander caused the largest exergy destruction.

Variation of Expander Inlet Temperature

Figure 8 Thermal (black) and exergetic (grey) cycle efficiency as a function of max cycle temperature for toluene (dashed line), benzene (dotted line) and cyclohexane (solid line)

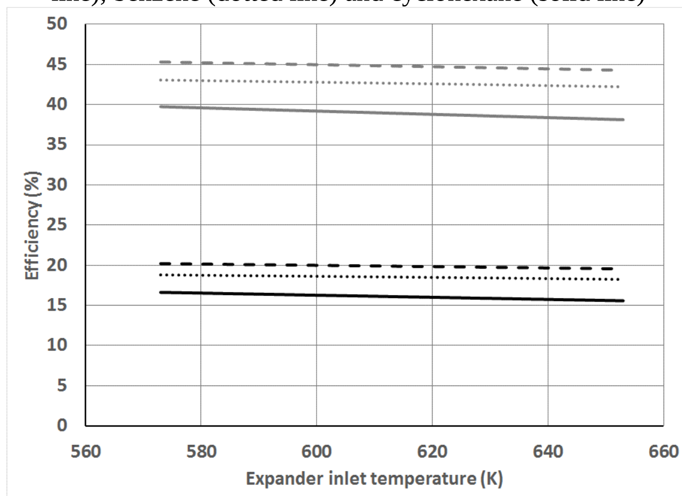
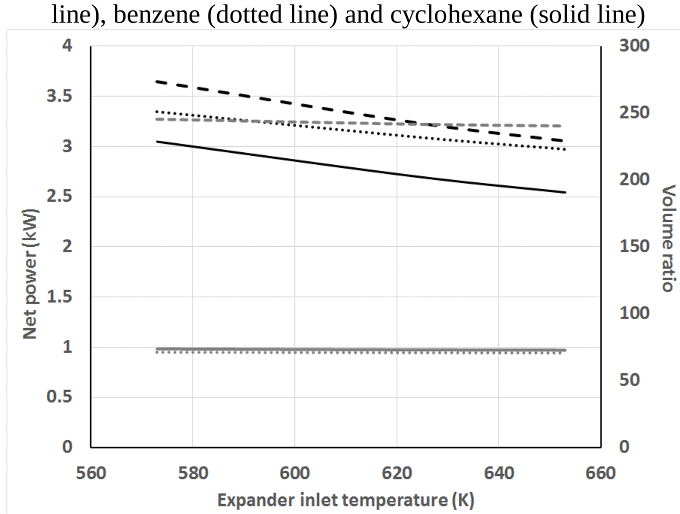


Figure 9 Net power (black) and expander volume ratio (grey) as a function of max cycle temperature for toluene (dashed line), benzene (dotted line) and cyclohexane (solid line)

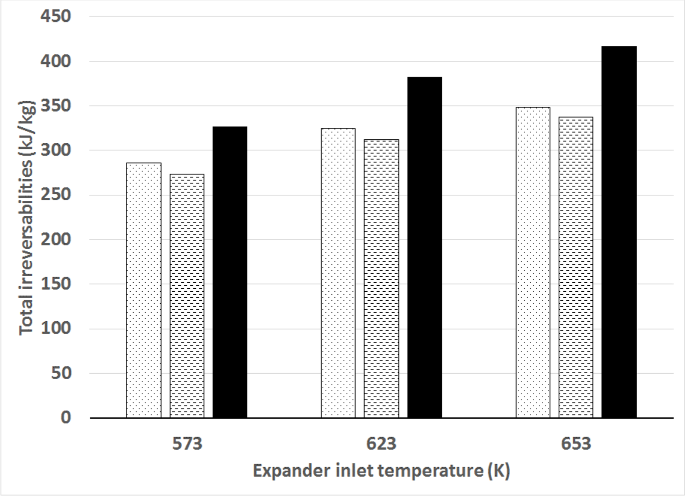


By setting the high cycle pressure at 10 bar, energetic and exergetic analysis was carried out on system with variable temperature of the superheated fluid at the expander inlet. Considering the approach point temperature difference in the 20-100K range, the maximum cycle temperature was varied between 573K and 653K. Energy and exergy efficiencies for the three fluids are presented in Figure 8. Toluene has again showed supremacy over benzene and cyclohexane, reaching both higher energy and exergy efficiencies. A gentle decline in the cycle performance with increasing temperature of the fluid at the expander inlet is obvious. Variation of the maximum cycle temperature had minor effect on the toluene cycle efficiency, whereas the largest decrease of thermal and exergy performance was noted for cyclohexane system of 0.13%/10K and 0.2 %/10K, respectively.

While the efficiency was decreasing slightly with the increase in the maximum cycle temperature, the effect on the cycle net power output was more prominent. Higher approach point temperature difference in turn allows for more heat available to be transferred to the fluid; hence, the mass flow rate increases. Simultaneously, lower temperature of the fluid at the expander inlet limits the expander work output. The overall effect is dominated by the former, and in the power output is substantially decreased. While toluene and cyclohexane yielded the highest and the lowest power performance, respectively, interestingly, both fluids showed similar response trend to the maximum temperature increase. Benzene system shows least susceptibility to power reduction due to expander inlet temperature increase, at the rate of 0.046kW/10K. It should be noted that the optimal power output was found for lower expander inlet temperature and higher boiler pressures.

Total irreversibilities (Figure 10) follow the same pattern as observed for high cycle pressure variations. The largest extent of exergy destruction is found in the cyclohexane cycle, whilst toluene one had the lowest irreversibilities. For all three fluids increase of the fluid temperature at the expander inlet led to increase of irreversibilities, with the largest rate of exergy destruction noticed for cyclohexane. Benzene and toluene had similar rate of entropy generation as the maximum cycle temperature increased.

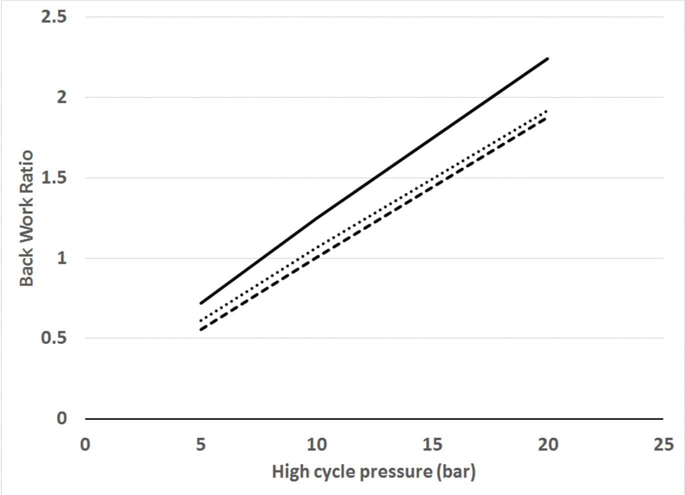
Figure 10 Total irreversibilities as a function of high cycle pressure for toluene (dashed), benzene (dotted) and cyclohexane (solid)



Pump and Expander analysis

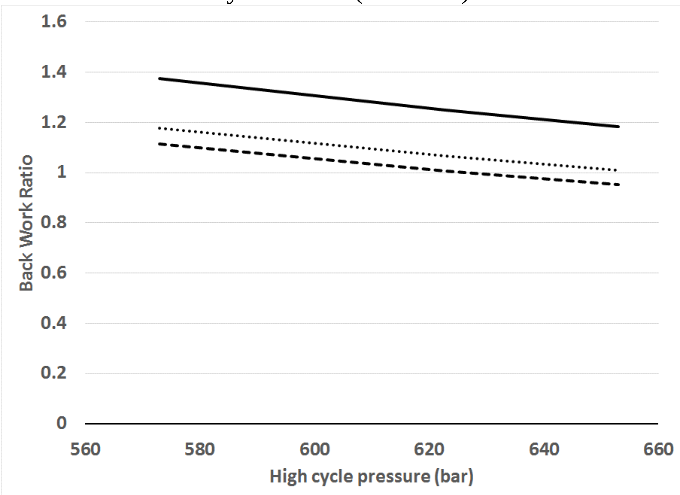
Pump is often a somewhat neglected element of ORC systems. When organic fluids, and refrigerants in particular, are employed in ORC systems, the pump consumes an amount of electricity, which may not be negligible, especially for a small ORC. In order to determine the power input to the pump, relative to the power produces, a back work ratio (the ratio of the power supplied to the pump and the power output from the expander) is often utilised. Additionally, the pump affects the design of the device as well as the cost of the system [13]. Back work ratio for considered fluids in the range of specified operational parameters is presented in Figure 11 (for expander inlet temperature kept fixed 623K) and Figure 12 (for cycle high pressure kept fixed 10 bar). The evaluated back work ratio are almost all below 2% as the pressures considered in this study are relatively moderate. Cyclohexane pump required larger power, and the consumption is augmented with the increase of the high cycle pressure. Similar trend is noticed for variation of the expander inlet temperature. Maximum fluid temperature did not affect the pump power consumption, yet back work ratio is decreasing due to lower specific expander work. Higher power demand of the cyclohexane pump is confirmed.

Figure 11 Back work ratio as a function of high cycle pressure for toluene (dashed line), benzene (dotted line) and cyclohexane (solid line)



Variation of maximum fluid temperature had a negligible impact on the expander volume ratio; thus the pressure ratio dictates the required expander size, which has a prominent effect on the overall cost. Given the evaluated power output in the 2-4 kW range, a micro-scale expander is needed. Scroll expanders are an established choice, although vane expanders could also be considered. Both expander types are of relatively lower cost compared to large scale ones [14], yet very few are commercially available at present. Cost around $2000/kWe are reported in the literature [15, 16]. Cost of the small-scale systems can also be extrapolated from [3], showing low power WHR ORC systems tend to be require larger investments.

Figure 12 Back work ratio as a function of expander inlet temperature for toluene (dashed line), benzene (dotted line) and cyclohexane (solid line)



All tested fluids perform better energetically and exergetically at expander inlet states characterised by low temperature and high boiler pressure. However, when high boiler pressures are achieved, the effect of expander inlet temperature is secondary. Additionally, high cycle pressure selection has a critical impact of the volume ratio, and consequently on the expander size. Sizing requirements for benzene and cyclohexane expanders are manageable, especially at lower pressures [17]. Energetic and exergetic efficiency, as well as the power output, are higher for benzene ORC, but there are other parameters to be considered. Compared to similar systems, in terms of power output and low operational boiler pressures, fluids considered here reach significantly higher efficiencies [18].

CONCLUSION

Organic Rankine cycle can be successfully utilised in waste heat harnessing applications. A simple ORC, powered by high-temperature diesel generator exhaust, was designed. Three working fluids were considered: toluene, benzene and cyclohexane. High cycle pressures and expander inlet temperature were varied to examine the effect of operational parameters on cycle energetic performance. Cycle net power, back work ratio, and expander volume ratio were analysed. Exergetic efficiency and irreversibilities in individual cycle components were evaluated.

Toluene showed superior energetic performance, achieving the highest power outputs. Exergetic efficiency was also larger for toluene cycle compared to other fluids. However, significant volume ratio is a downside of the toluene expander. Benzene and cyclohexane expanders require smaller volumes, especially at lower boiler pressures. Compared to cyclohexane, benzene achieves improved efficiencies, and has better back work ratio. While the effect of the high cycle pressure is the governing consideration, benzene cycle showed less sensitivity to the effect of the expander inlet temperature.

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