Evaluating the thermal efficiency and net power output of ORC for three wet fluids

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Abstract
Low-grade heat can be successfully recovered from waste energy sources for power production by an organic Rankine cycle (ORC). In this work, reheat ORC has been studied with three different wet organic fluids. The performance of R1270, R152a and Cyclopropane has been modelled in an ORC operated by a source of heat of 413.15K. The reheat ORC model contains economizer, evaporator, superheater, reheater, two turbine, pump and condensers. With variable turbine inlet temperature and reheat pressure, the net power output W_{net} and thermal efficiency have been estimated. The results show that there exists an optimum temperature at the turbine inlet maximizing W_{net}. The increase of reheat pressure gives only a marginal growth in the cycle efficiency, but it rises the W_{net} with about 16.88 %, 16.26 % and 13.23 % for R1270, R152a and cyclopropane respectively.
Keywords: ORC; Thermal efficiency; Reheat pressure; Net power output.

1. Introduction

In the last decades, the alarming increase in the consumption of fossil fuels has had a profound impact on the environment, leading to a multitude of pressing environmental issues. The burning of fossil fuels has contributed significantly to global warming, resulting in climate change and its associated consequences, such as rising sea levels, extreme weather events, and habitat destruction (Singh & Singh, 2017; Subramanian *et al.*, 2023; Wuebbles & Jain, 2001). Furthermore, the release of pollutants from the combustion of fossil fuels has led to the ozone layer depletion, causing an

increased risk of unsafe UV radiation attaining the surface of Earth (de Melo Santos, de Athayde Borille, Furtado, Monteiro, & Campos-Takaki, 2023; Onwudiwe, 2023). Additionally, the emission of sulfur dioxide and nitrogen oxides has led to the formation of acid rain, which has detrimental effects on ecosystems, soil, and water bodies (Board & Council, 1983; Prakash, Agrawal, & Agrawal, 2023). The cumulative effect of these environmental challenges has necessitated the exploration of alternative energy sources and more efficient energy utilization methods.

In response to these concerns, researchers have been actively investigating the utilization of waste heat sources as a means to mitigate the environmental impact associated with conventional energy conversion processes. The ORC has emerged as a promising technology due to its ability to convert waste heat into useful power (Rahbar, Mahmoud, Al-Dadah, Moazami, & Mirhadizadeh, 2017). Compared to other thermodynamic cycles, the ORC is considered simpler and more widely used (Xu *et al.*, 2019). Its versatility and adaptability to different heat sources make it an attractive option for various applications, ranging from industrial processes to renewable energy systems.

Over the years, numerous studies have been conducted to examine and optimize the performance of ORC arrangements. Dai and Gao (Dai, Wang, & Gao, 2009) conducted a comprehensive analysis and comparison of ten different working fluids, including ammonia, R236EA, and R113, in various ORC configurations. The objective was to identify the most suitable working fluid that maximizes exergy efficiency. Through their analysis, it was established that R236EA exhibited the maximum exergy effectiveness among the evaluated working fluids. Additionally, the researchers determined that adding an internal heat exchanger (IHE) did not yield improvements in ORC performance under the same waste heat source temperature. Expanding on this research,(G. Li, 2016) focused on investigating different ORC configurations considering heat source temperature level. The study selectively examined isentropic and dry fluids and assessed the impact of internal heat exchangers and regenerative processes on system performance. The findings indicated that the incorporation of an IHE and the use of a regenerative ORC configuration led to enhanced thermal efficiency compared to other configurations.

In a thermo-economic context, Imran *et al.*(Imran *et al.*, 2014) carried out an optimization analysis on regenerative and baseline ORC systems. They considered investment costs and thermal efficiency as objective functions in order to identify the most economically viable options. The research revealed that R245fa provided the paramount marks in terms of both performance and cost-effectiveness. Furthermore, the regenerative ORC system demonstrated higher thermal efficiency compared to a conventional ORC system, which could potentially offset the higher initial investment costs over the long term.

The ongoing quest for improved efficiency and power generation has motivated our present study, which specifically focuses on the reheat ORC utilizing wet fluids operating with the same waste heat source. Our aim is to explore the influence of crucial factors such as the expander inlet temperature and reheat pressure on power output and thermal efficiency of the system. By conducting a detailed analysis and incorporating real-world operational considerations, our research aims to contribute to a deeper understanding of the performance characteristics of reheat ORC systems. Ultimately, this knowledge can guide the optimization of such systems for efficient utilization of waste heat sources in a variety of practical applications.

2. Thermodynamic analysis of ORC

2.1 Cycle design and modelling

Many researchers have investigated the influence of different system configurations on ORC performance. Li *et al.* (J. F. Li *et al.*, 2021)conducted a comprehensive study on the impact of the expander inlet temperature and reheat pressure in a regenerative ORC system. Their findings demonstrated that optimizing these parameters can significantly improve the system's thermal efficiency and net power output. The application of advanced technologies has also been explored

to enhance ORC performance. For example, the integration of an internal heat exchanger in an ORC system has been investigated by Zhang *et al.* (Zhang *et al.*, 2022). Their study demonstrated that the incorporation of an internal heat exchanger can improve the system's thermal efficiency by reducing the temperature difference between the working fluid and heat source.

A typical reheat Organic Rankine Cycle (ORC) configuration comprises various components, including a pump, condenser, evaporator, and two expanders. This arrangement is visually illustrated in Figure 1. In operation, the working fluid is initially pumped into the evaporator, where it undergoes a heat exchange process with the heat source. Consequently, the working fluid absorbs heat energy and undergoes a phase change, transitioning into a superheated vapor state.

Upon leaving the evaporator, the superheated vapor enters the first expander, commonly referred to as a turbine, where it expands, releasing its stored energy and generating power. This power generation stage is essential for harnessing the potential of the working fluid and converting it into a useful form, such as electricity. Following the expansion process, the vapor proceeds to the reheater, where it is subjected to another heat exchange with the heat source. In this stage, the vapor absorbs additional heat from the source, which increases its temperature.

Subsequently, the reheated vapor enters the second expander, referred to as a lower-pressure turbine, where it undergoes another expansion process. This turbine operates at a lower pressure than the initial expander, ensuring the efficient extraction of additional energy from the working fluid. The lower pressure in the reheater, compared to the original evaporator, allows for an optimized energy transfer process.

After the expansion in the second turbine, the working fluid exits the system and enters the condenser, where it undergoes a phase change from a superheated vapor to a saturated liquid state. This transformation occurs as the working fluid releases heat energy to a cooling medium or environment. The condensation process completes the cycle, allowing the working fluid to return to its initial state in preparation for the next cycle.

The present study assumes that (a) the system is in a steady-state. (b) There are no pressure drops in the economizer, evaporator, super heater, reheater, condenser, and the pipes. (c) no heat loss in all components. (d) The reheat pressure represents an average of the condensing and evaporating pressures (Chen, Goswami, & Stefanakos, 2010). Table 1 lists additional operating conditions which are assumed. Also, the performance of reheat ORC is studied under the framework of the first law of thermodynamics.
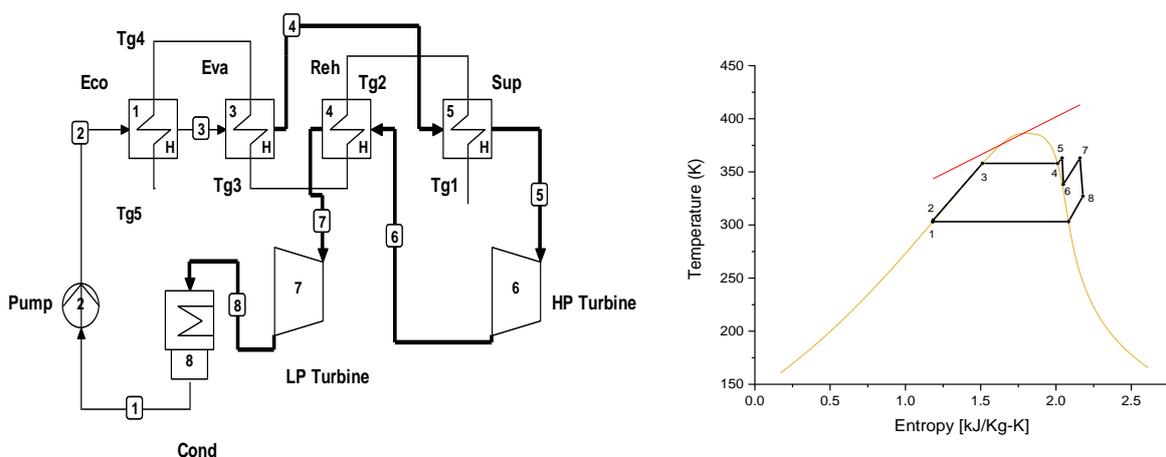
Figure 1 - Layout and (T-s) diagram of the reheat ORC system.

The energy balance in economizer, evaporator, super heater and reheater are:

$$\dot{m}_{wf}(h_3 - h_2) = \dot{m}_g Cp_m(Tg_4 - Tg_5) \tag{1}$$

$$\dot{m}_{wf}(h_4 - h_3) = \dot{m}_g Cp_m(Tg_3 - Tg_4) \tag{2}$$

$$\dot{m}_{wf}(h_5 - h_4) = \dot{m}_g Cp_m(Tg_1 - Tg_2) \tag{3}$$

$$\dot{m}_{wf}(h_7 - h_6) = \dot{m}_g Cp_m(Tg_2 - Tg_3) \tag{4}$$

where $\dot{m}_{wf}$ and $\dot{m}_g$ correspond to the mass flow rates of the gas and the working fluid. Cp_m is the mean specific heat capacity of the gas at constant pressure, h is the specific enthalpies of the working fluid at each state and Tg is the heat source temperature at each state.
The power of HP and LP turbines are given by

$$W_T = \dot{m}_{wf}(h_5 - h_6) + \dot{m}_{wf}(h_7 - h_8) \tag{5}$$

The heat rejection in the condenser is

$$\dot{Q}_{con} = \dot{m}_{wf}(h_8 - h_1) \tag{6}$$

The work input done by the pump is given by

$$W_{Pump} = \dot{m}_{wf}(h_8 - h_1) \tag{7}$$

The heat addition to the cycle is the sum of $\dot{Q}_{con}$, $\dot{Q}_{eva}$, $\dot{Q}_{Sup}$ and $\dot{Q}_{Reh}$ which can be expressed as

$$\dot{Q}_{in} = \dot{m}_{wf}(h_5 - h_2) + \dot{m}_{wf}(h_7 - h_6) \tag{8}$$

The net power output depends on various factors such as working fluid properties, heat source temperature, system efficiency, and other operational parameters. All this can be represented by the difference between W_T and W_{Pump}
$$W_{net} = W_T - W_{Pump} \tag{9}$$

The ORC thermal efficiency is defined by the following ratio

$$\eta = \frac{W_{net}}{\dot{Q}_{in}} \tag{10}$$

The pinch point is

$$Tg_4 = T_3 + \Delta T_{pp} \tag{11}$$

The turbine inlet temperature is

$$T_5 = T_4 + 5 \tag{12}$$

The above equations can be used to determine W_{net}, η, $\dot{m}_{wf}$ of working fluid and the hot gas temperatures.

Table 1 – Operating condition assumptions (http://www.coolprop.org/).

Heat source temperatures (K)	413.15
Hot gas mass flow rate	14
Specific heat for hot gases (kj/kg-K)	1.1
Condensing temperature (K)	303.15
LP and HP turbine efficiency	0.8
Efficiency of the pump	0.7
Pinch temperature difference (K)	8

2.2 Working fluid selection

Extensive research efforts have been dedicated to enhancing the performance of ORC systems. Researchers have explored different working fluids, configurations, and optimization techniques to maximize their efficiency and power output. Among the various types of working fluids that can be used in an ORC system. The selection of a suitable working fluid depends on numerous factors. According to (Imran *et al.*, 2014), fluids can be classed in three types isentropic dry, and wet fluids depending on the slope of saturation vapour on (T-s) diagram.

Figure 2 shows the three types of fluids in a (T–s) diagram dT/ds. For environmental reasons, some working fluids have been abandoned, and some others are going to be phased out as shown in (Imran *et al.*, 2014). In this work, only three wet fluids are considered. Table 2 and Figure 3 show the critical properties of the selected fluids and the working fluid candidate in (T-s) diagram, respectively. All the data in Table 2 comes from Cool prop (G. Li, 2016).

Table 2 - Properties of the three wet fluids.

Working fluid	M (kg/kmol)	T_{cr} [K]	P_{cr} [MPa]
R1270	42.08	364.21	4.56
R152a	66.05	386.41	4.52
Cyclopropane	42.08	398.30	5.58
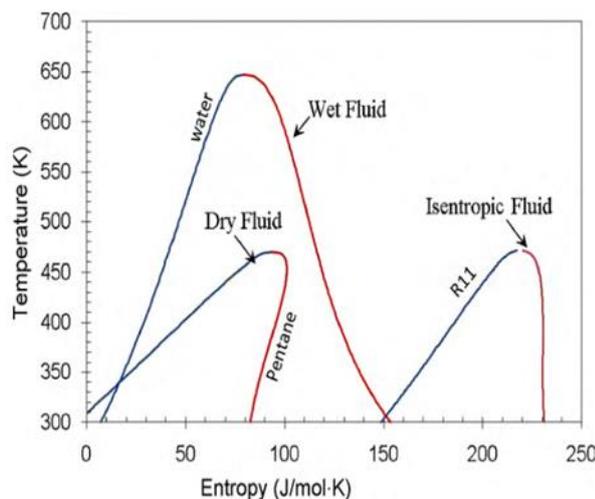
Figure 2 - Dry, isentropic and wet working fluids (Imran *et al.*, 2014).
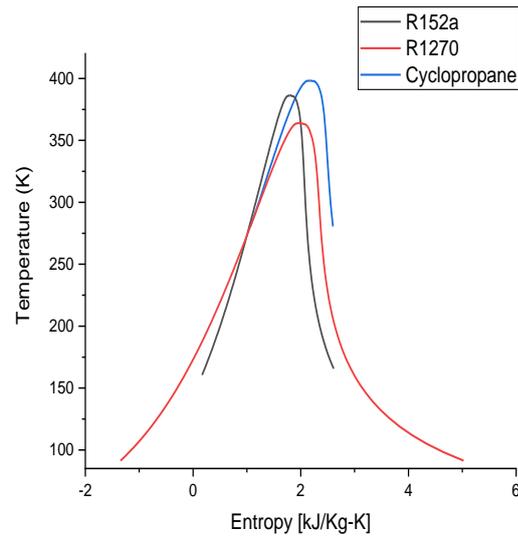
Figure 3 – (T-s) diagrams of the selected working fluid.

3. Results and discussion

3.1. Effect of turbine inlet temperature

In this section, we set the evaporation temperature variation from 323K to $T_{cr} - 10$ to limit the cycle being a subcritical cycle and to keep appropriate difference between the temperatures of the hot gas and the working fluid. The turbine inlet temperature is the evaporation temperature plus 5K to ensure that the exit of the working fluid from the HP turbine lies in the super-heated vapor region.

Figure 4 illustrates that the net power output first increases with the turbine inlet temperature and then decrease after reaching its maximum at turbine inlet temperature of 363K. Differently, the net power output of R1270 increases and reaches its maximum at turbine inlet temperature of 358 K with net power output of 108.63 kW.

The turbine inlet temperature of R1270 is lower because of its lower critical temperature. It can be observed that it is not always the case that a higher turbine inlet temperature will result in a greater net power output. While increasing the turbine inlet temperature can potentially lead to higher power generation, there are other factors that come into play, and there may be practical limitations or diminishing returns.

Figure 5 shows the thermal efficiency variation in terms of the turbine inlet temperature. It is clear that for all working fluids, the thermal efficiency increases with the increase of turbine inlet temperature, while the cyclopropane presents the maximum thermal efficiency compared to R152a and R1270.
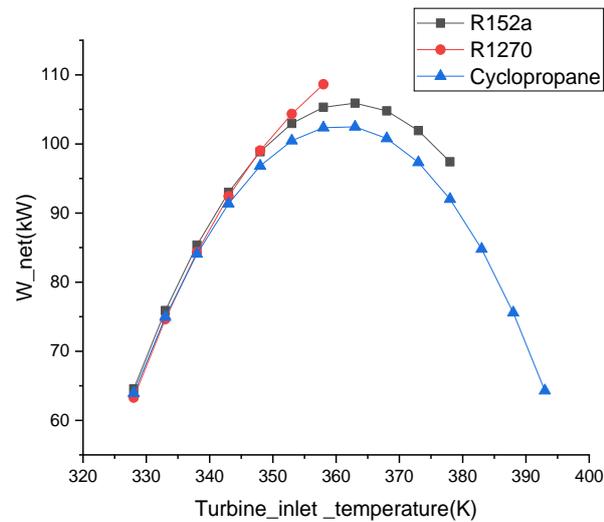
Figure 4 - Net power output with turbine inlet temperature.
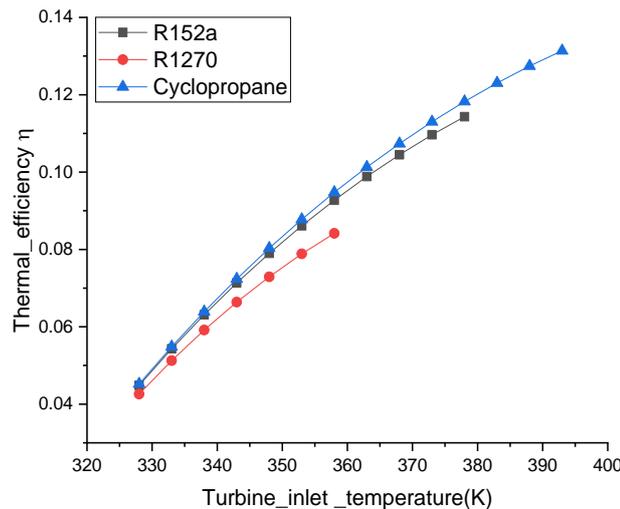
Figure 5 - Thermal efficiency with turbine inlet temperature.

3.2 Effect of reheat pressure

In this section, we set the evaporation temperature as 358K and the maximum reheat pressure is $P_{ev} * 0.9$.

Figure 6 presents the effects of the variation of the reheat pressure on the net power output, where it increases with the increase of the reheat pressure, and R1270 has the maximum net power output with a value of 114.96 kW, while R152a and cyclopropane have maximum values of 111.97 kW and 107.08 kW respectively.
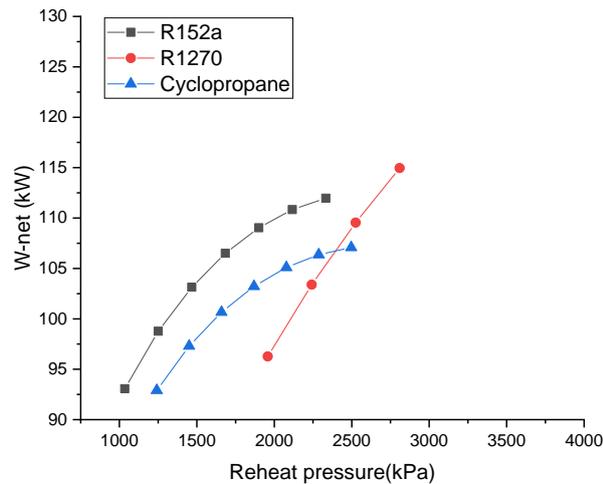
Figure 6 - Net power output with reheat pressure.
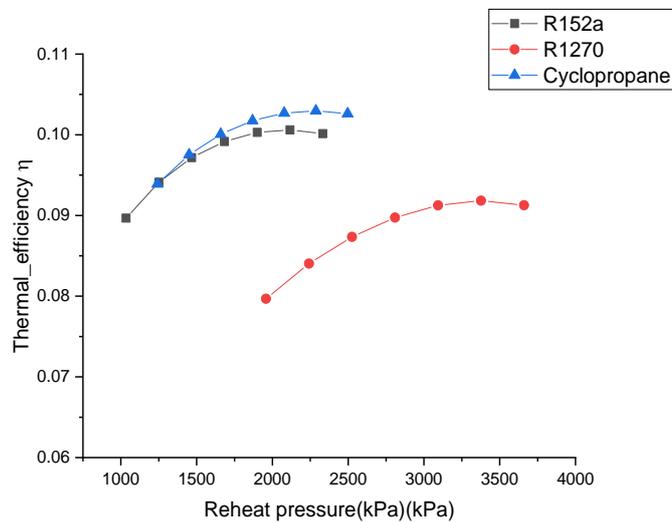
Figure 7 - Thermal efficiency with reheat pressure.

Figure 7 presents the effects of the variation of the reheat pressure on the thermal efficiency. For all working fluid, there is a marginal augmentation in thermal efficiency, for R152a, there is an increase with about 1.04 %, for R1270 and cyclopropane the increase is about 1.16 % and 0.87 % respectively. From Figs. 6 and 7 it can be concluded that the increase of reheat pressure gives only a marginal growth in the cycle efficiency, but it rises the net power output by making the use of higher pressures possible and keeping the quality of working fluid at turbine exhaust remains in the vapor region.

4. Conclusion

In this study, the effects of turbine inlet temperature and reheat pressure on the net power output and thermal efficiency of a reheat ORC system were investigated. Three wet working fluids, namely R1270, R152a, and cyclopropane, were analyzed under a waste heat source temperature of 413.15 K. The findings of the study are as follows.

For the net power output, it was observed that there exists an optimum turbine inlet temperature that maximizes the output. The optimal turbine inlet temperatures were found to be 358 K for R1270, 363 K for R152a, and 363 K for cyclopropane. These temperatures were identified as the points at which the net power output reaches its maximum value. At higher or lower turbine inlet temperatures, the net power output begins to decrease. Therefore, to achieve the highest net

power output, it is crucial to operate the reheat ORC system with the specific optimal turbine inlet temperature for each working fluid.

On the other hand, regarding thermal efficiency, the study revealed that higher turbine inlet temperatures result in higher thermal efficiencies. Unlike the net power output, there was no specific optimum temperature identified for maximizing thermal efficiency. Instead, as the turbine inlet temperature increases, the thermal efficiency of the system also increases. This observation implies that higher turbine inlet temperatures enhance the overall energy conversion efficiency of the reheat ORC system.

Regarding the reheat pressure, it was found that increasing the reheat pressure only leads to a marginal improvement in the efficiency of the cycle. However, it does have a significant impact on the net power output. The increase in reheat pressure resulted in net power output increases of approximately 18.69 kW for R1270, 18.90 kW for R152a, and 14.17 kW for cyclopropane. Therefore, while the efficiency gains were modest, the higher reheat pressures significantly contributed to boosting the overall power generation capacity of the reheat ORC system.

These findings highlight the importance of optimizing the turbine inlet temperature and reheat pressure in a reheat ORC system to achieve the desired balance between net power output and thermal efficiency. By carefully selecting the operating parameters, it is possible to maximize power generation while maintaining a satisfactory level of energy conversion efficiency.

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