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Thermoeconomic analysis of recuperative sub- and transcritical organic Rankine cycle systems

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Abstract

There is significant interest in the deployment of organic Rankine cycle (ORC) technology for waste-heat recovery and power generation in industrial settings. This study considers ORC systems optimized for maximum power generation using a case study of an exhaust flue-gas stream at a temperature of 380 °C as the heat source, covering over 35 working fluids and also considering the option of featuring a recuperator. Systems based on transcritical cycles are found to deliver higher power outputs than subcritical ones, with optimal evaporation pressures that are 4–5 times the critical pressures of refrigerants and light hydrocarbons, and 1–2 times those of siloxanes and heavy hydrocarbons. For maximum power production, a recuperator is necessary for ORC systems with constraints imposed on their evaporation and condensation pressures. This includes, for example, limiting the minimum condensation pressure to atmospheric pressure to prevent sub-atmospheric operation of this component, as is the case when employing heavy hydrocarbon and siloxane working fluids. For scenarios where such operating constraints are relaxed, the optimal cycles do not feature a recuperator, with some systems showing more than three times the generated power than with this component, albeit at higher investment costs.

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1. Introduction

The use of waste heat and of alternative sources of low- or medium-grade heat, such as geothermal or solar heat, can play a key role in decreasing the current dependence and consumption rates of fossil fuels, increasing security and decreasing emissions. Low- and medium-grade heat can be recovered to provide heating, or converted into useful power such as electricity, or a combination of the two [1]. A number of technologies exist that are suitable for the conversion of such lower-grade heat to useful power including the organic Rankine cycle (ORC), which employs different organic working fluids and their mixtures, such as hydrocarbons, refrigerants, or siloxanes [2–5]. A significant effort has been placed on the development and improvement of ORC power systems in different applications including waste-heat recovery, renewable heat (geothermal, biogas/biomass) conversion, and solar-thermal power [6–9].

The uptake of ORC technology is being handicapped by long payback periods. The power output of ORC systems can however be enhanced by employing a recuperator, a heat exchanger used to preheat the working fluid before

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evaporation using recovered heat from the working fluid after expansion. This can reduce the amount of thermal energy extracted from the heat-source stream, which increases the system's thermal efficiency. Furthermore, this decreases the heat-source stream's temperature drop within the evaporator, and thereby may in some cases relax the evaporator pinch limitations depending on where the pinch point is found inside this heat exchanger. This, in turn, may allow the ORC system to operate with higher working-fluid flowrates (until the pinch conditions are re-established), thus enabling a further increase in efficiency and power output, for the same heat-source conditions.

However, a number of questions remain unanswered regarding the introduction of a recuperator, which is an additional component that leads inevitably to higher system complexity and cost. While its addition ensures an improvement in thermal efficiency, its effect on the optimal exergy efficiency and power output are still under discussion [10,11]. The roles of the working fluid (dry, isentropic, wet) and cycle architecture (subcritical, transcritical) on the decision to include a recuperator remain unexplored. For the cases where a recuperator may indeed be beneficial, the effectiveness of the heat exchanger is also important, and the additional costs associated need to be considered.

In this work we explore the benefits and drawbacks of using recuperators in ORC systems with the aid of thermodynamic cycle analysis. The aforementioned working fluids and cycle architectures are optimized for maximum net-power generation, with particular consideration given to the heat-source characteristics and the condenser boundary conditions (cooling rates, exit temperatures). While cycles with no recuperation typically give higher exergy efficiencies, there exist cases where a combination of factors (working fluids, boundary conditions) result in recuperative cycles being optimal; we therefore extend our analysis to include the economic considerations of such cases.

2. ORC system models

2.1. External boundary conditions and working-fluid selection

In this paper, the heat source is a flue gas from an industrial cement kiln, with a flowrate of 185 kg/s at 380 °C. The heat sink is taken as cooling water at 25 °C, with a maximum temperature increase of 30 °C. Over 35 pure working fluids (see Table 1), spanning the classes of alkanes and their isomers, refrigerants, siloxanes and aromatic hydrocarbons (benzene and toluene) are considered. These working fluids, chosen to span a wide range of critical temperatures and in combination with the high heat-source temperature, are suitable for both subcritical and transcritical ORC systems. They vary in degree of 'dryness' from the very dry siloxanes and heavy hydrocarbons to the wet refrigerants such as R152a, and also including isentropic fluids such as R124 and R1234yf. The isentropic efficiencies of the pump and expander are 85% and 75% respectively while the heat exchangers' minimum temperature difference (ΔT_{\min}) is 10 °C.

Table 1: Critical properties of selected ORC working fluids.

Working fluids' class	Working fluids	Critical temperature (T_{crit} , °C)	Critical pressure (P_{crit} , bar)
Light alkanes and alkene	Propane, butane, isobutane, pentane, hexane, isohexane, heptane, propylene.	96.7, 152.0, 134.7, 196.6, 234.7, 224.6, 267, 91.1.	42.5, 38.0, 36.3, 33.7, 30.3, 30.4, 27.4, 45.6.
Refrigerants	R113, R114, R115, R12, R123, R1233zd, R1234yf, R1234ze, R124, R125, R134a, R141b, R142b, R143a, R152a, R218, R227ea, R245fa, RC318.	214.1, 145.7, 80.0, 112.0, 183.7, 165.6, 94.7, 109.4, 122.3, 66.0, 101.1, 204.4, 137.1, 72.7, 113.3, 71.9, 101.8, 154, 115.2.	33.9, 32.6, 31.3, 41.4, 36.6, 35.7, 33.8, 36.3, 36.2, 36.2, 40.6, 42.1, 40.6, 37.6, 45.2, 26.4, 29.3, 36.5, 27.8.
Heavy alkanes, siloxanes, aromatics and water	Octane, nonane, decane, D4, D5, MM, MDM, MD2M, benzene, toluene, water	296.2, 321.4, 344.6, 313.3, 346.1, 245.5, 290.9, 326.3, 288.9, 318.6, 373.9.	25.0, 22.8, 21.0, 13.3, 11.6, 19.4, 14.2, 12.3, 49.1, 41.3, 220.6.

2.2. ORC thermodynamic model

The thermodynamic model of simple subcritical ORCs is well described in literature. This consists of an energy balance across each component of the cycle. The model used here is also capable of analysing superheated and recuperated ORC systems. In recuperated cycles, the recuperator is modelled based on the amount of heat recoverable from the working fluid exiting the expander with a dimensionless parameter called the recuperative fraction (θ_{recup}):

$$\theta_{\text{recup}} = \frac{h_4 - h_{4r}}{h_4 - h_{(T_2 + \Delta T_{\min}, P_{\text{cond}})}} \approx \frac{T_4 - T_{4r}}{T_4 - (T_2 + \Delta T_{\min})}. \quad (1)$$

At $\theta_{\text{recup}} = 0$, the recuperative cycle reverts to the basic cycle with no recuperation, and when $\theta_{\text{recup}} = 1$, the maximum possible amount of heat is exchanged between the working fluid exiting the expander and that exiting the pump.

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