

# Effects of critical and boiling temperatures on system performance and fluid selection indicator for low temperature organic Rankine cycles



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## ABSTRACT

The critical temperature ( $T_c$ ) and boiling temperature ( $T_b$ ) of working fluids are important selection criteria for the organic Rankine cycle (ORC) system. In this study, the ratio of  $T_b$  and  $T_c$  ( $T_{br}$ ) and the vapor expansion ratio (VER) model based on Clausius–Clapeyron equation are introduced to compare and explain their effects on the maximum net output power ( $W_{net,max}$ ) and VER of ORC. The investigation of 267 working fluids is done at four heat source temperatures ( $T_5$ ). Maximum vapor enthalpy method is proposed to determine the upper limit of the evaporation temperature, which is the optimization parameter for maximizing the net power output. At low  $T_5$  (423.15 and 473.15 K), the obvious relationships between  $W_{net,max}$  and  $T_c$  are independent of  $T_{br}$ . Therefore,  $T_c$  enables to select working fluids with high  $W_{net,max}$ . However, at high  $T_5$  (523.15 and 573.15 K),  $T_{br}$  is essential to exclude working fluids with optimum  $T_c$  ( $0.89 - 0.90T_5$ ) but low  $W_{net,max}$ . Moreover, at a given  $T_c$ , high  $T_b$  or  $T_{br}$  indicates high VER. Consequently,  $T_b$  or  $T_{br}$  is suitable to be used as the second indicator. This paper proposes the optimal combinations of  $T_c$  and  $T_b$  and the developed composite indicator for selection of working fluids.

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## 1. Introduction

The organic Rankine cycle (ORC) utilizes organic substances with low boiling points, instead of water, as working fluids. This cycle can be driven by different heat sources, including industrial waste heat [1,2], solar energy [3,4] and geothermal energy [5,6]. ORC is a promising technology for converting low-grade heat into electricity because of its advantages over the traditional steam Rankine cycle [7–10].

Working fluids are important in cycle performance and component design. The thermodynamic properties of working fluids have been studied to determine the proper criteria for selecting working fluids [11,12]. Critical temperature, which affects cycle performance, has been extensively investigated by researchers. Saleh et al. [13] and Lai et al. [14] reported that thermal efficiency in the isolated ORC increased with increasing critical temperature. Xu et al. [15] revealed that high critical temperatures

induced the  $T$ - $Q$  curves of the organic working fluid to approach that of the heat source, resulting in a decrease in integrated-average temperature difference. Different results were obtained in ORC coupled with a heat source. He et al. [16] observed that high net power output was produced by working fluids with critical temperatures close to the temperature of the fixed waste heat source. Saleh et al. [13] found that working fluids with low critical temperatures are suitable for ORC with an initial heat carrier temperature of 120 °C and a pinch point temperature difference of 10 °C. Lai et al. [14] showed that the influence of critical temperatures on thermal efficiencies was not evident for ORC with heat source temperatures of 280 °C and 350 °C. Andreasen et al. [17] indicated that working fluids with critical temperatures of half of the heat source inlet temperature produced the optimal performance for the transcritical cycle. Astolfi et al. [18] conducted economic optimization for ORC with geothermal brine inlet temperatures of 120 °C, 150 °C, and 180 °C by using the ratio of the critical temperature to the heat source inlet temperature; this study confirmed that in supercritical cycles, the use of fluids with critical temperature slightly lower than the temperature of the geothermal source generated the lowest electricity cost.

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Previous studies investigated the influence of the boiling temperature of working fluids on the ORC system performance. Wang et al. [19] investigated the performance of ORC by using the multi-objective optimization model with heat source temperatures of 100 °C–220 °C and pinch point temperature difference of 5 °C–30 °C; the results showed that boiling temperature affects the cycle performance. Mago et al. [20] found that boiling point influences the thermal efficiency of the system. Aghahosseini and Dincer [21] revealed that working fluids with high boiling temperatures resulted in low ORC irreversibility. Mago et al. [22] determined the influence of boiling temperature on the thermal efficiency of basic and regenerative ORC systems by comparing the simulation results of R113, R123, R245ca, and isobutane; in this study, working fluids with the highest boiling temperatures showed the highest thermal efficiency. Hung [23] found that working fluids with high boiling temperatures resulted in higher efficiency.

Results on studies of the relationship between the critical temperature of working fluids and the performance of ORC systems are inconsistent under different heat source temperatures and types of cycles. Research on the relationship between boiling temperature and system performance shows consistent results, that is, the thermal efficiency of ORCs increases with increasing boiling temperatures of working fluids. Although strong relationship exists between critical temperature and boiling temperature, few researchers compare their different effects on the performance of the ORC from the perspective of their relevance.

In this study, the thermodynamics properties and ORC performance of 267 working fluids at heat source inlet temperatures of 423.15, 473.15, 523.15, and 573.15 K were investigated. The different effects of critical temperature and boiling temperature on the maximum net output power ( $W_{\text{net,max}}$ ) and the vapor expansion ratio (VER) were compared and explained using the ratio of  $T_b$  and  $T_c$  ( $T_{b/c}$ ) and the VER model. Finally, this paper proposes the optimal combinations of  $T_b$  and  $T_c$  and the developed composite indicator based on  $T_c$  and  $T_b$ .

**2. Materials, system and assumptions**

**2.1. Materials**

According to the Montreal protocol, chlorine-containing working fluids should be eliminated [24]. Based on the slope of the temperature entropy curve to be infinity, positive, or negative,

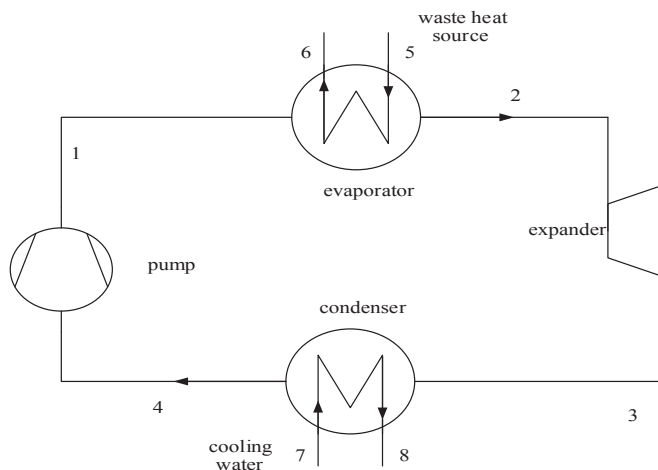


Fig. 1. Schematic diagram of the ORC.

working fluids are generally classified into isentropic, dry, or wet, respectively [22]. Unlike wet fluids, dry and isentropic fluids are superheated after isentropic expansion and no liquid droplets are formed at the turbine outlet. Appendix A shows the basic thermodynamic properties of the preselected organic fluids.

**2.2. System description**

The layout of a basic ORC system without internal heat exchanger for waste heat recovery is shown in Fig. 1. The components of the ORC include an evaporator driven by low-grade waste heat, an expander, a water- or air-cooled condenser, and a working fluid pump. In the evaporator, the working fluid is heated by waste heat and becomes high pressure vapor, which expands in the expander and converts heat energy into work. The exhaust vapor is then condensed by cooling water or air in the condenser. The condensed working fluid is pressurized in the pump and enters into the evaporator for the next cycle.

**2.3. Limitation of evaporation temperature**

Superheating in an ORC slightly increases the thermal efficiency but decreases the exergy efficiency [25]. Hence, the saturated Rankine cycle was investigated in this study. The ORC system becomes unstable near the critical point of the working fluid; therefore, the reasonable maximum operating pressure or temperature must be determined. Drescher and Bruggemann [26] suggested that pressure difference between the maximum evaporator pressure and the critical pressure should be 0.1 MPa. Delgado–Torres and Garcia–Rodriguez [27] reported that the temperature difference between the maximum evaporator temperature and the critical temperature should be 10 °C–15 °C. However, a fixed pressure or temperature interval may be unsuitable in determining the distance because of the diversity of working fluids. Rayegan and Tao [4] established the maximum entropy method (s-max); this technique requires two steps to obtain the result (the maximum entropy point on the  $T$ – $S$  diagram is first obtained, then the higher limit is increased to attain 1% mass fraction of the liquid across the turbine) but is unsuitable for isentropic fluids. To make this process faster and universal, we propose a new method based on the

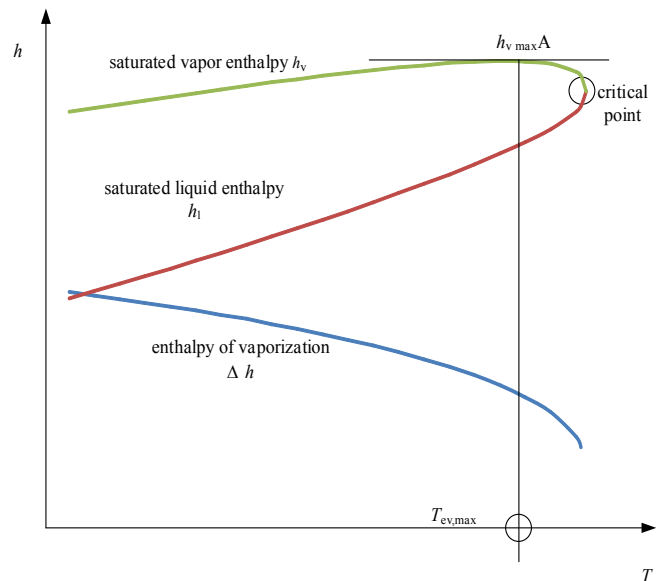


Fig. 2.  $h$ – $T$  diagram of butane.

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