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Effects of superheat and internal heat exchanger on thermo-economic performance of organic Rankine cycle based on fluid type and heat sources

Cheng Zhang [a, b], Chao Liu [a, *], Xiaoxiao Xu [a], Qibin Li [c], Shukun Wang [a], Xi Chen [b, d, **]

[a] Key Laboratory of Low-grade Energy Utilization Technologies and Systems, Ministry of Education, College of Power Engineering, Chongqing University, Chongqing, 400030, China
[b] Department of Earth and Environmental Engineering, Columbia University, New York, NY 10027, USA
[c] College of Aerospace Engineering, Chongqing University, Chongqing, 400030, China
[d] ICAM, School of Aerospace, Xi'an Jiaotong University, Xi'an, 710049, China

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A B S T R A C T

The study investigates the comprehensive effects of superheat and internal heat exchanger (IHX) on the thermo-economic performance of organic Rankine cycle (ORC). Exergy efficiency, net power output, and electricity production cost (EPC) are compared based on the working fluid properties and heat sources. The results indicate that under a lower heat source temperature and load, exergy efficiency of IHX-ORC does not always exceed that of simple ORC (S-ORC) when EPC is selected as an objective function, and IHX-ORC exhibits a worse economic performance than S-ORC for all fluids (R161, R1234ze, R152a, cyclopropane, butane, R123, cyclopentane, heptane, and cyclohexane). However, IHX-ORC with dry fluid achieves a better thermo-economic performance than that with wet fluid when the heat source temperature and load increase to a high level. The EPC of IHX-ORC is close to that of S-ORC with the increase in heat source temperature and load, and thus, IHX-ORC exhibits approximately 10–17% higher thermal efficiency and 5–10% higher exergy efficiency than those of S-ORC. With respect to butane and R123, the net power output exhibits approximately 22.5% and 23.5% growth, respectively. In order to evaluate the feasibility of IHX-ORC, a judgement indicator [$\alpha > 1.90625 + 0.4258\xi$] with respect to six factors is proposed.

1. Introduction

The rapid development of society is accompanied by increasing energy consumption and the deterioration in environmental pollution. The development of utilization technologies for renewable energy and enhancing conversion efficiency are the main measures to alleviate energy risk. Statistical investigations have indicated that more than 50% of the total heat generated in industry corresponds to low-grade waste heat [1]. However, it is not possible to efficiently convert low-medium temperature heat to electrical power by conventional approaches [2]. Additionally, the organic Rankine cycle (ORC) exhibits high potential and is widely used due to its simplicity, maintainability, high efficiency, and low price when compared with other waste heat recovery (WHR) applications [3]. Furthermore, local and small-scale power generation makes it easy and flexible to use waste heat on site [4]. For several decades, various studies have proposed and examined various thermodynamic cycles such as supercritical ORC, ORC with internal heat exchanger (IHX-ORC), reheat ORC, regenerative ORC, Kalina cycle, Goswami cycle, and trilateral flash cycle [2,4–6].

Although the IHX-ORC requires additional components and is a more complex system, the inclusion of internal heat exchanger (IHX) can reduce the metal usage of the boiler and condenser by reducing the heat load [7,8]. Additionally, the superheating of fluid negatively contributes to the efficiency of ORC with dry fluid, positively contributes to ORC with wet fluid, and almost does not affect the ORC with isentropic fluid [2]. Nevertheless, an experimental observation [9] indicated that liquid entrainment phenomenon was observed for ORC with superheat (SH-ORC) by using

* Corresponding author.
** Corresponding author. Department of Earth and Environmental Engineering, Columbia University, New York, NY 10027, USA.
E-mail addresses: liuchao@cqu.edu.cn (C. Liu), xichen@columbia.edu (X. Chen).

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R245fa when the superheat was 1.8 °C while the system stabilizes if the superheat increases to 8.7 °C. Therefore, superheating is necessary even for dry working fluid.

A few studies [10–13] were performed to analyze the thermodynamic performance and economic feasibility of IHX-ORC with other cycles. Studies [8,14] indicated that the inclusion of IHX always lead to improvements in the thermodynamic performance including thermal efficiency, exergy efficiency, and the outlet temperature of the heat source in a subcritical ORC. Furthermore, studies [5,15,16] revealed that there existed a threshold pressure above which the IHX did not improve the system performance. With respect to techno-economic feasibility, investigations indicated that the IHX could significantly increase the payback period [17], total heat exchange area, and total capital cost [16,18–20]. Conversely, Li [5] indicated that IHX-ORC had the close specific cost per kilowatt to the simple ORC (S-ORC) and the IHX-ORC exhibited more economic benefits when compared with S-ORC if the heat source scale increased to a high level.

The above brief review clearly indicates that the superheat and IHX are important for ORC from both the thermodynamic or techno-economic viewpoints. However, there is a paucity of studies on the comprehensive effect of superheat and IHX on ORC system. As shown in Table 1, Roy et al. [11] and Liu et al. [12] investigated the effect of superheat and recuperating on ORC system only at a constant superheat degree. Furthermore, it should be noted that Guo et al. [18] and Saleh et al. [21] advocated that a more significant improvement could be achieved if the superheat was combined with an IHX.

Therefore, the objective of this study is to investigate the comprehensive effects of the superheat and IHX on ORC system based on the working fluid properties and heat sources. Three ORC configurations are investigated as follows: S-ORC, SH-ORC, and IHX-ORC. With respect to different ORC configurations, the thermal efficiency, exergy efficiency, net power output, and electricity production cost (EPC) are compared under the conditions of different heat source temperatures and loads. Subsequently, the effects of working fluid type, heat source temperature domains, and heat loads on the system thermo-economic performance are investigated. Furthermore, the effects of superheat and pinch point temperature difference in IHX on the IHX-ORC system economic performance are analyzed. Finally, a judgement indicator α with respect to heat source temperature, heat load, and working fluid properties is proposed to evaluate whether a working fluid is feasible for IHX-ORC system.

2. Modeling methodology

2.1. System description

In this section, three ORC configurations are introduced. The schematic diagram of S-ORC, SH-ORC, and IHX-ORC is shown in Fig. 1. Furthermore, the components and procedures are described.

2.1.1. Simple organic Rankine cycle

Superheat is necessary to avoid liquid droplet impingement in expander during expansion for wet fluids. Typically, the dryness fraction at the outlet of an expander is kept above 90% [7]. The working fluid absorbs heat from the exhaust flue gas in the evaporator and subsequently continuously vaporizes into saturated vapor (dry and isentropic fluid) or overheated vapor (wet fluid). Subsequently, the high-pressure vapor flows into the expander to expand and convert into shaft work. Following this, the expanded superheated vapor flows into the condenser wherein the vapor is condensed to saturated liquid by the cooling water. The saturated liquid is pumped into the evaporator again to continue the next cycle. Fig. 2 (a) and (b) show the *T-s* diagram of S-ORC.

2.1.2. Organic Rankine cycle with superheat (SH-ORC)

In order to compare the difference in thermo-economic performance between IHX-ORC and SH-ORC, SH-ORC is selected as a reference system. The SH-ORC exhibits the same components as S-ORC. Nevertheless, the working fluid of evaporator outlet is heated to overheated state with a higher superheat degree as shown in Fig. 2 (c) and (d).

2.1.3. Organic Rankine cycle with internal heat exchanger (IHX-ORC)

The superheat degree of vapor at outlet of expander maintains a high level when dry fluid or higher superheat is selected, and this leads to a significant waste of energy to the heat sink. An increase in the temperature of the working fluid discharged from the outlet of expander is introduced to the inlet of the low-pressure side of IHX. Conversely, the low temperature albeit high pressure working fluid discharged from the pump is introduced to the inlet of high pressure side of IHX. Thus, the excess heat is recovered from the exhausted vapor to the fluid supply to avoid energy wasting. In IHX-ORC, the IHX is combined with superheat as shown in Fig. 3 (a) and (b).

2.2. Cycle modeling

The heat source temperature and mass flow rate ranges are 150–270 °C and 5–50 kg/s, respectively. In order to avoid the acid dew point , the outlet temperature of flue gas exceeds 82 °C [28]. The cooling medium is domestic water and its inlet temperature is set as 20 °C. The optimal condensing temperature exhibits only slight variations with different heat source temperatures and working fluids [29]. Therefore, the condensing temperature is assumed as 30 °C in the modeling. In order to search for the optimal thermodynamic parameters of ORC system, the temperature difference ranges of 0–30 °C, 5–30 °C, and 5–15 °C are set for the pinch point temperature in the evaporator, condenser, and IHX, respectively. Similarly, the superheat degree range is 0–50 °C. With respect to the turbine and pump, the isentropic efficiencies are 0.8 and 0.75 [20], respectively. In the economic model, the annual operating time, life cycle time, and annual loan interest rate are assumed as 8000 h, 20 years, and 5% [29], respectively. The grid price of state grid company is in the range of 0.03–0.152 $/kWh in China [30]. Thus, the grid electricity price is set as 0.15 $/kWh in the study. The given conditions and ORC parameters are shown in Table 2. Furthermore, other assumptions are as follows [22,23]: The ORC system is under the stable state. The pressure drop in the pipes is neglected and the heat losses in the components are neglected. Additionally, working fluids at the condenser outlet correspond to saturated liquid.

The thermo-economic models for different ORC configurations are listed in Table 3. K_1, K_2, K_3, B_1, B_2, C_1, C_2 and C_3 are fitting cost coefficients for different equipment in the economic models, and the values are given in Table 4. An evaporator is a finned tube heat exchanger, and a condenser is a shell-and-tube heat exchanger in the heat exchanger design. The heat exchanger process in the evaporator is divided into the following three regions: preheating, evaporation (two-phase), and superheating regions. The heat exchanger process in the condenser is divided into the following two regions: cooling and condensation (two-phase) regions. The correlations between heat-transfer coefficient and pressure drop for the different regions of heat exchangers are shown in Table 5 and Table 6, respectively. A more detailed description of the heat exchanger design is found in a previous study [30].