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Organic Rankine cycle for power recovery of exhaust flue gas

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HIGHLIGHTS

- Three types of working fluids are selected for ORC using exhaust flue gas.
- The mixture that matches with heat sink has the greatest efficiency.
- The mixture that matches with heat source has the lowest superheat degree.
- There does not exist a working fluid that satisfies all the indicators.

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ABSTRACT

To study the effects of different working fluids on the performance of organic Rankine cycle (ORC), three working fluids, a mixture that matches with heat source, a mixture that matches with heat sink and a pure working fluid, are selected in this paper. Thermodynamic models were built in Matlab together with REFPROP, with which, the physical properties of the selected working fluids can be acquired. Heat source of the ORC system is the exhaust flue gas of boiler in a 240 MW pulverized coal-fired power plant. Some indicators such as thermal efficiency, inlet temperature of expander, superheat degree, mass flow, volumetric flow, and exergy destruction distribution, as well as the influence of recuperator are studied. The analytical results show that the mixture that matches with heat sink has the greatest efficiency and the mixture that matches with heat source has the lowest superheat degree. The rate of heat exchanged in recuperator to that in evaporator has a maximum value with evaporating pressure. There exists no optimal working fluid for all indicators (thermal efficiency, heat exchanger area, mass flow and volumetric flow etc.). An appropriate working fluid should be chosen by taking both investment cost and power generating benefits into account. The cost-benefit ratio of the proposed ORC plant was evaluated either.

1. Introduction

Exhaust flue gas in coal-fired power plants is a rich low-temperature waste heat resource. The temperature of the exhaust flue gas of a pulverized coal-fired power plant is usually between 120 °C and 130 °C, and exergy loss of it is the biggest among the losses of power generating unit.

Organic Rankine cycle (ORC) has been demonstrated to be a promising technology for low-temperature heat recovery [1,2]. It is expected to be a practical way to recover the exhaust flue heat in coal-fired power plants with the matched evaporating temperature to heat source. Literature review found that the applications of the ORC were mainly focused on solar energy [3,4], geothermal energy [5,6], waste heat from diesel engine [7,8] and industry [9]. Among the existed studies, the interests included that of selection of appropriate working fluids for different heat sources, optimal design of cycle process for a better performance and the design of main parts of ORC, such as expander and heat exchangers [10]. The working fluids include pure and mixed fluids. Lots of researches have been done on pure working fluids with their stable and more easily available thermo physical properties [11,12]. Zeotropic mixed working fluids have sloped phase-change profiles that may match well with heat source and heat sink in the cycle, which can improve the system performance [8,13–16]. In addition, mixtures have advantages over pure working fluids in aspects of safety, environment friendliness [17], and volumetric flow rate [18]. But for the reasons of the separation in evaporating process, unknown thermal properties and abundant combinations, mixtures haven't been studied and applied widely.

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Nomenclature

A	area, m^2
c_p	specific heat at constant pressure, kJ^{-1} kg K
E	exergy, W
h	enthalpy, J kg^{-1}
I	exergy loss, W
m	mass flow rate, kg s^{-1}
Q	heat, W
T	temperature, °C
U	heat transfer coefficient, w m^{-2} K^{-1}
w	power, W

Greek symbols

Δ	difference
η	efficiency, %

Subscripts

0	ambient
con	condenser
ev	evaporator
exg	exergy
exp	expander
f	working fluid
i	section
is	isentropic
in	inlet
net	net output
out	outlet
p	pump
pin	pinch
rec	recuperator
s	heat source
t	in theory
the	thermal
w	cooling water

The heat source temperature of ORC can cover the range of 80°C–300 °C while selecting suitable working fluids [11]. The temperature of the exhaust flue gas of circulating fluidized bed power plant can be as high as 150 °C, and for off-design conditions this temperature varies accordingly. In the present study, 130 °C is selected as the heat source of ORC for studying the power recovery from exhaust flue gas in a typical pulverized coal-fired power plant. Three types of working fluids are selected. The corresponding performances of ORC are compared based on the pinch model of evaporator [19].

2. System description and thermodynamic modeling

ORC is composed of an evaporator, an expander, a condenser and a feed pump. Fig. 1 shows the schematic diagram of ORC with and without recuperator. Exhaust flue gas enters the evaporator at state point 5 and exits at state point 6 having heated the working fluid to vapor phase. Recuperator is introduced in Fig. 1(b), where it can absorb heat that is recovered from the working fluid after expansion. T–S diagrams for ORC are shown in Fig. 2. Fig. 2(a) is a supercritical cycle and Fig. 2(b) is a subcritical cycle.

The equations for different components and processes of ORC are described as follows.

The expander: Assuming non-isentropic expansion process in the expander, the enthalpy of working fluid in the outlet of expander and the work output of expander can be given as,

$$h_{f,2} = h_{f,1} - \left(h_{f,1} - h_{f,2,t}\right)\eta_{\mathrm{is}} \tag{1}$$

$$w_{\mathrm{exp}} = m_f\left(h_{f,1} - h_{f,2}\right) \tag{2}$$

where $h_{f,2,t}$ is the enthalpy of working fluid in the outlet of expander under the ideal condition, η_{is} is the isentropic efficiency meaning the ratio of brake power to isentropic power requirement.

The exergy loss of expander can be given as,

$$I_{\mathrm{exp}} = E_{f,1} - E_{f,2} - w_{\mathrm{exp}} \tag{3}$$

where E_i is the exergy of state point, i, and can be calculated as,

$$E_i = m[(h_i - h_0) - T_0(s_i - s_0)] \tag{4}$$

in which, the subscript 0 indicate the environmental state. In the present study, the environmental temperature is assumed to be at 20 °C at the atmospheric pressure.

The condenser: The overall heat transfer is presented by,

$$Q_{\mathrm{con}} = m_f\left(h_{f,2} - h_{f,3}\right) = m_w\left(h_{w,8} - h_{w,7}\right) \tag{5}$$

The exergy loss of condenser can be given as,

$$I_{\mathrm{con}} = E_{f,2} + E_{w,7} - E_{f,3} - E_{w,8} \tag{6}$$

The pump: The working fluid is compressed to the top pressure in this process. The power consumption of the pump can be expressed as,

$$w_p = m_f\left(h_{f,4} - h_{f,3}\right)\Big/\eta_p \tag{7}$$

where η_p is the pump efficiency meaning the ratio of fluid power to brake power.

The exergy loss of pump can be given as,

$$I_p = w_p + E_{f,3} - E_{f,4} \tag{8}$$

The evaporator: This is the heat absorption process at constant pressure. The heat transfer between the heat source and working fluid can be expressed as,

$$Q_{ev} = m_f\left(h_{f,1} - h_{f,4}\right) = m_s\left(h_{s,5} - h_{s,6}\right) \tag{9}$$

The exergy loss of evaporator can be given as,

$$I_{\mathrm{ev}} = E_{f,4} + E_{s,5} - E_{f,1} - E_{s,6} \tag{10}$$

The recuperator: If the internal heat recovery is considered in the cycle, parts of the heat of exhaust working fluid from expander can be used to preheat the inlet working fluid of the evaporator. In the recuperator, the liquid working fluid discharged from pump is heated from $T_{f,4}$ to $T_{f,4'}$, as well as the working fluid exhausted from the expander is cooled from $T_{f,2}$ to $T_{f,2'}$. The heat transfer in recuperator can be expressed as,