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Performance enhancement of organic Rankine cycle with two-stage evaporation using energy and exergy analyses

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ABSTRACT

Organic Rankine cycle (ORC) is promising in converting the low-medium grade thermal energy into power, but the efficiency is relatively low, which is mainly due to the poor temperature matching in the evaporator. Based on the single-stage evaporation of the ORC, the two stage evaporation strategy is proposed to improve the evaporation process between the heat source and the working fluid, with the heat source segmented into sections. The two stage organic Rankine cycle (TSORC) was evaluated by the energy and exergy analyses. This paper aims to illuminate the two-stage evaporation mechanisms and optimize the cycle parameters. The results show that the two-stage evaporation enhances the evaporating temperature of the high-stage, thereby improving the evaporating process but deteriorating the expansion, condensation and pressurization processes. Overall, The TSORC tends to improve the system performance, which is at the cost of increasing the total thermal conductance, volumetric flow ratio, and the investment. A higher intermediate geothermal water temperature (IGWT) cannot effectively utilize the high-level heat source. There exists an optimal IGWT to maximize the objective function within the range of this study.

1. Introduction

The global energy shortage promotes the development of organic Rankine cycle (ORC) in heat recovery as a result of its special advantages. Compared with the Kalina cycle, the transcritical Rankine cycle, and other cycles, ORC systems have relatively simple structures, suitable working pressure and convenient operating maintenance. Therefore, the ORC technology has caused wide attention in the past three decades (Badr et al., 1984, 1985; Hung et al., 1997; Saleh et al., 2007a; Li et al., 2010; Habka and Ajib, 2015). However, the efficiency of the ORC is low, which is mainly because of the poor temperature matching in the vapor generation process.

As for improving the parameter matching of the heat source/sink and the working fluid of the ORC, Hung et al. (1997), Saleh et al. (2007a), Aljundi (2011), Hung (2001), Yari (2009, 2010), Liu et al. (2004), Arosio and Carlevaro (2003), Lakew and Bolland (2010), Pasetti et al. (2014), Brignoli and Brown (2015), and Liu et al. (2015) investigated different working fluids from their own points of view, and the specific working fluids were also recommended. However, it should be pointed out that the working fluids that the researchers recommended are only suitable for the specific working conditions, with a limited or no universality. This is because the working fluid is very sensitive to the temperatures of the heat source and heat sink, which varies from one researcher to another. Because of the randomness of the temperatures of the heat source and heat sink selected by the researchers, no working fluid has been acknowledged.

Another method that is often used to enhance the system performance of the ORC is to optimize the cycle parameters with different objective functions. For subcritical ORC, Hettiarachchia et al. (2007) used the total heat transfer area to the net power out. Roy et al. (2010) considered the power output and efficiencies. Rashidi et al. (2011) took efficiencies and specific work as the objective functions. Guo et al. (2011) showed that optimum evaporation temperature and fluids vary with screening criteria. He et al. (2012) optimized evaporating temperature and working fluids. Wang et al. (2010) optimized the thermodynamic parameters using genetic algorithm. As for the transcritical cycle, Cayer et al. (2010) and Zhang et al. (2011) compared the transcritical ORC with subcritical one.

For the ORC, the single-stage evaporation is the major factor to generate the system irreversibility under the premise of the constant pinch point temperature difference. Based on ORC, many

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Nomenclature

A	Area (m^2)
c	Specific heat (kJ/kg)
Ex	Exergy (kW)
h	Specific enthalpy (kJ/kg)
I	Irreversibility rate (kW)
K	Heat transfer coefficient (W/(m^2 °C))
M	Molar mass (kg/kmol)
m	Mass flow rate (kg/s)
P	Pressure (MPa)
Q	Heat transfer rate (kW)
r	Latent heat of vaporization (kJ/kg)
s	Specific entropy (kJ/(kg °C))
T	Temperature (K)
t	Temperature (°C)
U	Intrinsic energy (kJ)
W	Power (kW)
ΔP	Pressure difference (Pa)

Greek symbols

η	Efficiency (%)
ρ	Density (kg/m^3)
Φ	Entransy dissipation (kW K)

Subscripts

c	Condenser
cri	Critical
cw	Cooling water
e	Evaporator
ex	Exergetic
g	Generator
gw	Geothermal water
m	Mechanical
opt	Optimal
p	Pump
pp	Pinch point
s	Isentropic
t	Turbine
th	Thermal
wf	Working fluid
0	Environment
1, 2, 3, 4	State points

Acronyms

ALT	Atmosphere life time (yr)
GWP	Global warming potential
ODP	Ozone deletion potential
ORC	Organic rankine cycle
VFR	Volumetric flow ratio

researchers have done a lot of work to improve cycle modes in order to enhance the system performance. Kosmadakis et al. (2009), Kosmadakis et al. (2010), Wang et al. (2012), Liu et al. (2012), Zhang et al. (2013), Shu et al. (2013a, 2013b, 2014), and Yang et al. (2014) proposed a new dual-loop organic Rankine cycle (DORC) to perform better than the ORC. Mohammadkhani et al. (2014) researched two ORCs. Li et al. (2014a) put forward a parallel double-evaporator organic Rankine cycle (PDORC). Moreover, Saleh et al. (2007b) improved system performance significantly by multiple-pressure configurations. Li et al. (2015a, 2015b) presented and investigated the two stage organic Rankine cycle (TSORC) in heat recovery. However, the cycle configurations in literature (Kosmadakis et al., 2009, 2010; Wang et al., 2012; Liu et al., 2012; Zhang et al., 2013; Shu et al., 2013a, 2013b, 2014; Yang et al., 2014; Mohammadkhani et al., 2014; Li et al., 2014a; Saleh et al., 2007b) are basically in parallel to improve the system performance to some extent. However, the high-stage ORC could adversely generate much more irreversible loss, which is caused by the large temperature difference between the evaporating temperature of the high-stage and the condensing temperature. Moreover, the cycle configurations in literature (Li et al., 2015a, 2015b) are actually in serial, which is conducive to lowering the irreversibility of the vapor-generation process in the evaporator.

Based on the thermodynamics, a numerical model to optimize the TSORC is established. A dimensionless parameter, the ratio of the exergy efficiency to the production of the total thermal conductance and the volumetric flow ratio (VFR) of the TSORC to that of the ORC under the maximal net power output, $((\eta_{ex}/(\mathrm{VFR}(KA)))_{\mathrm{TSORC}}/(\eta_{ex}/(\mathrm{VFR}(KA)))_{\mathrm{ORC}}|_{W_{\mathrm{net}}=W_{\mathrm{net,max}}})$, is defined to evaluate the cycle performance. The main objective of this study is to ascertain the effectiveness of the TSORC and optimize the cycle parameters, such as W_{net}, S_g, $t_{\mathrm{gw,mid}}$, $t_{e,1}$, $t_{e,2}$, η_{th}, η_{ex}, $(KA)_e$, $(KA)_c$, and $(KA)_{\mathrm{total}}$.

2. Analysis of the TSORC system

The TSORC is the combination of the two-stage evaporation and the traditional ORC, and the biggest difference between the TSORC and ORC is that the two-stage evaporation is adopted for the TSORC whereas the single-stage is used for the ORC. A typical TSORC system for power generation can be categorized in four loop circuits according to the working media: the heat source, the working fluid of the high- and low-pressure stages, and the heat sink. The TSORC mainly consists of an evaporator of the high-pressure stage, an evaporator of the low-pressure stage, a turbine of the high-pressure stage, a turbine of the low-pressure stage, a generator, a condenser, a pump of the high-pressure stage, a pump of the low-pressure stage, a cooling tower, a cooling pump, and a hot water pump. Compared with ORC, the working fluid of the TSORC vaporizes under two different pressures, and the biggest difference of the TSORC is that the liquid working fluid flowing into the high-stage evaporator is not from the condenser but from the low-pressure stage evaporator under the saturated liquid. From the viewpoint of the structure, the low-pressure stage evaporator should be a shell-and-tube heat exchanger. Different from the traditional shell-and-tube heat exchanger, the low-pressure stage evaporator has two outlets corresponding to the overheated vapor to the turbine and saturated liquid to the high-stage evaporator, respectively.

The heat source transfers heat to the organic fluid, which absorbs heat to generate the high-pressure vapor in the evaporator of the high-pressure stage (Figs. 1 and 2, state 1") and the low-pressure vapor in the evaporator of the low-pressure stage (Figs. 1 and 2, state 1′), then the vapor flows into the turbines 1 and 2 in turn, respectively. The enthalpy is converted into shaft work to drive the generator. The vapor exited from the turbines mixes and (Figs. 1 and 2, state 2) is led to the condenser where it is liquefied by cooling water. The liquid working fluid at the condenser outlet (Figs. 1 and 2, state 3) is first pressurized by the pump 1 and flows into the evaporator a (Figs. 1 and 2, state 4′), and the liquid working fluid at the saturated state of the low-pressure is divided into two parts: a portion continues to be heated until superheated in evaporator 1, and the other part is pressurized to the high-pressure and is also heated to be heated until superheated in evaporator 2. Then a new cycle begins. The *T-s* diagram of the TSORC is shown in Fig. 2.

To simplify the analysis of the TSORC, the cycle operates steadily. The working fluid is superheated at the turbine inlet and saturated at the condenser outlet, with the kinetic change, the potential energy change, the thermal loss and the friction loss in the pipes