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Parametric optimization and thermodynamic performance comparison of single-pressure and dual-pressure evaporation organic Rankine cycles

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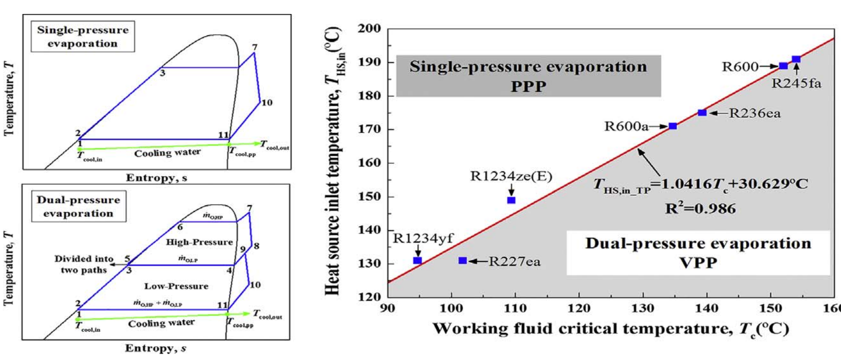
HIGHLIGHTS

- Single-pressure and dual-pressure evaporation ORCs using pure fluids are studied.
- Optimized evaporation pressures and evaporator outlet temperatures are obtained.
- System performance for the 100–200 °C heat sources is analyzed and compared.
- Net power outputs of dual-pressure evaporation ORCs can increase by 21.4–26.7%.
- A quantitative criterion is provided to assess the optimal cycle type.

GRAPHICAL ABSTRACT



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ABSTRACT

Dual-pressure evaporation organic Rankine cycle (ORC) involves two evaporation processes with different pressures, and can significantly reduce the exergy loss in the heat absorption process compared with conventional single-pressure evaporation ORCs. However, the applicable heat source temperatures of dual-pressure evaporation ORCs and the effects of the working fluid thermophysical properties on the applicable conditions remain indeterminate. Optimal cycle parameters for various heat source temperatures also need to be studied. Solving these questions is crucial for the application and promotion of dual-pressure evaporation ORCs. This study focuses on a typical dual-pressure evaporation ORC driven by the 100–200 °C heat sources without a limit on the outlet temperature. Nine pure organic fluids were selected as working fluids. Evaporation pressures and evaporator outlet temperatures of the single-pressure and dual-pressure evaporation ORCs were optimized, and their optimized system thermodynamic performance was compared. Results show that the applicable heat source temperature range of the dual-pressure evaporation ORC ($W_{net,dual} > W_{net,single}$) generally increases as the working fluid critical temperature increases. The upper limit of the applicable heat source temperatures ($T_{HS,in\,TP}$), working fluid critical temperature and pinch point temperature difference generally conform to a linear relation. For the heat source temperature below $T_{HS,in\,TP}$, the maximized net power output of the dual-pressure evaporation ORC is larger than that of the single-pressure evaporation ORC. Furthermore, the increment generally increases as the heat source temperature decreases, and the maximum increments are 21.4–26.7% for nine working fluids. For the heat source temperature above $T_{HS,in\,TP}$, the dual-pressure evaporation ORC is unbefitting.

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Nomenclature

g	gravitational acceleration (9.8 m s^{-2})
H	pressure head (m)
h	specific enthalpy (kJ kg^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})
p	pressure (MPa)
Q	heat flow rate (kW)
s	specific entropy (J kg^{-1} K^{-1})
T	temperature (°C)
W	power (kW)
ΔT	temperature difference (°C)

Greek symbols

η	efficiency

Subscripts

1–11	state points shown in Figs. 1 and 2
c	critical state
cond	condensation or condenser
cool	cooling water
dual	dual-pressure evaporation
e	evaporation or evaporator
HAP	heat absorption process
HP	high-pressure stage
HS	heat source fluid
HS, 1	heat source fluid at the high-pressure stage outlet
HS, 2	heat source fluid at the low-pressure stage outlet
in	inlet
ip	inflection point
LL	lower limit
LP	low-pressure stage
max	maximum
min	minimum
net	net output
O	organic working fluid
opt	optimal or optimized
out	outlet
P	feed pump
pp	pinch point
single	single-pressure evaporation
sup	superheating or superheated
sv	saturation vapor curve
sys	system
T	turbine
TP	transition point
UL	upper limit

Abbreviations

GWP	Global Warming Potential
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
PPP	Pinch Point occurs at the Preheater inlet
PTORC	Parallel Two evaporator Organic Rankine Cycle
STORC	Series Two evaporator Organic Rankine Cycle
VPP	Pinch Point occurs at the Vaporization bubble point

1. Introduction

To reduce CO_2 emission and weaken greenhouse effect are global challenges. Renewable energy utilization and waste heat recovery are recognized key approaches to reducing CO_2 emission and weakening greenhouse effect. Exploring the efficient utilization technology for renewable energy and waste heat resources has attracted the attention worldwide. The organic Rankine cycle (ORC) is an important and promising heat–power conversion technology that has been widely used in the renewable energy utilization (e.g., solar thermal, geothermal, and biomass energies) and waste heat recovery (e.g., internal combustion engine exhaust, industrial flue gas, and hot processed liquids) around the world [1–9]. ORC is based on the principle of the Rankine cycle and uses organic fluids as working fluids [8,10]. ORC has advantages of stability, flexibility, safety, as well as wider applicable ranges of the heat source temperature and installed capacity; compared to other heat–power conversion technologies utilizing the low and medium temperature (< 350 °C) thermal energy [2,3,9,11–15].

The more efficient utilization of the renewable energy and waste heat resources is the primary goal for the design and optimization of ORC systems. The design and optimization of the system is based on the cycle type of ORC, which significantly affects its energy utilization efficiency [8,10,16,17]. For the conventional subcritical ORCs, although the cycle concept is simple, their performance (e.g., the thermal efficiency, exergy efficiency, or net power output) is generally considerably lower than the theoretical ceiling for a given heat source and heat sink. Therein, the exergy loss during the finite temperature difference heat transfer between the working fluid and heat source fluid is generally the largest, and can exceed 40% of the total exergy loss [18–20]. Reducing the exergy loss during the finite temperature difference heat transfer between the working fluid and heat source fluid is crucial to increase the energy utilization efficiency for a conventional ORC system.

Conventional subcritical ORCs are generally based on the single evaporation pressure. The temperature match between the working fluid and heat source fluid is generally poor due to the pinch point temperature difference limitation and the working fluid isobaric heat absorption characteristics, which results in the considerable exergy loss [8,17–24]. Specifically, when the local heat capacity rate is not well matched between the working fluid and heat source fluid, a large temperature gradient will exist between them, which significantly increases the exergy losses. In other words, to be more direct viewing and vivid, when the working fluid of the subcritical ORC absorbs the heat from the heat source fluid, the curve of the working fluid is a polyline shape in the *T-s* diagram, whereas the curve of the heat source fluid is generally almost linear, which results in a poor temperature match and significantly increases the exergy losses. Furthermore, the heat release characteristics of various heat sources considerably vary [2,3,17,25,26], and the adaptability of conventional subcritical ORCs based on the single evaporation pressure is insufficient to meet the demand of temperature match [17,22,23,25]. Improving the temperature match between the working fluid and heat source fluid is necessary to achieve a high energy utilization efficiency.

Several scholars have attempted to introduce zeotropic mixtures into the ORC system to increase the heat–power conversion efficiency, because the zeotropic mixture has a varying phase change temperature [5,7,13,27–32]. However, the temperature match between the working fluid and heat source fluid remains unsatisfactory, though the temperature match in the condensation process can be improved [5,7,29]. The limitations of the pinch point temperature difference and the working fluid isobaric heat absorption characteristics (e.g., the curve of the working fluid is the polyline shape in the *T-s* diagram) still exist for zeotropic mixtures [33]. Using the transcritical ORC is also an important approach to increasing the heat–power conversion efficiency [8,10,17,34–36]. For the transcritical ORC, the working fluid temperature increases continuously in the vapor generator, and that may provide a better temperature match between the working fluid and heat source fluid, compared to the conventional subcritical ORCs [10,17,35]. Therefore, the exergy loss in the cycle heat absorption process will be