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Thermo-economic performance analyses and comparison of two turbine layouts for organic Rankine cycles with dual-pressure evaporation

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ABSTRACT

Multi-pressure evaporation organic Rankine cycle involves two or more evaporation processes with different pressures. Compared to the conventional single-pressure evaporation type, the multi-pressure evaporation type can significantly reduce the exergy loss in the endothermic process, and its widely used in the heat–work conversion of low and medium temperature (<350 °C) thermal energy is promising. The turbine layout of the multi-pressure evaporation type has two typical forms: the separate turbine layout and induction turbine layout. Turbines in two layouts may exhibit considerable differences in the geometric and operating parameters. Selecting a suitable turbine layout is crucial to improve the system thermo-economic performance. While, the thermo-economic performance variations and comparison of two turbine layouts remain indeterminate for various operating conditions. This study was based on the one-dimensional efficiency model and purchased equipment cost model of the radial-flow turbine. The thermo-economic performance of two turbine layouts was analyzed and compared for nine pure organic fluids. Effects of two-stage evaporation pressures on the thermo-economic performance of two turbine layouts were also studied. Results show that the total power output of the induction turbine layout can increase by 0.3–5.4%, and its specific investment cost is lower for most of operating conditions and the maximum decrement is 34.2%, compared to the separate turbine layout. The decrement in the specific investment cost decreases as the high-stage evaporation pressure increases, and it generally increases as the low-stage evaporation pressure increases. Moreover, the total power output is larger, the thermo-economic advantage of the induction turbine layout is generally greater.

1. Introduction

The organic Rankine cycle (ORC) is a common heat–work conversion technology which is based on the principle of the Rankine cycle and generally uses organic fluids as working fluids [1]. ORCs exhibit considerable potential in the efficient heat–work conversion of low and medium temperature (<350 °C) thermal energy. Tchanche et al. [2] found that ORCs could be used for the heat–work conversion of geothermal energy, solar thermal energy, biomass energy, ocean thermal energy, and waste heat recovery; and had advantages of high efficiency, simplicity, and a wide installed capacity range. Results of Velez et al. [3] showed that ORCs had advantages of a wide applicable heat source temperature range. Our previous work [4] made improvements for the conventional ORC system, and results indicated that ORCs were relatively flexibility and stability. Moreover, Basaran and Ozgener [5] found that ORCs could be safety enough by selecting the suitable working fluid.

Achieving a higher heat-work conversion efficiency is always an important goal for ORC systems. The cycle type of an ORC system significantly affects its heat–work conversion efficiency [6]. Conventional cycle types of ORC, such as the subcritical and transcritical cycles [7], are generally based on the single endothermic pressure. These cycle structures are simple; however, the temperature match between the heat source fluid and working fluid is generally unsatisfactory [8], mainly due to the pinch point temperature difference limitation and the working fluid isobaric endothermic characteristics. Le et al. [9] found that the exergy loss in the endothermic process could exceed 40% of the total exergy loss for ORC systems using pure fluids and zeotropic mixtures. Results of Baral et al. [10] showed that the exergy loss in the endothermic process accounted for 42% of the total exergy loss for a small-scale ORC system. In addition, characteristics of various low and medium temperature heat sources considerably vary [11]. The adaptability of ORCs with the single endothermic pressure is generally poor for various characteristic heat sources.

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Nomenclature

c	absolute velocity (m s^{-1})
$\bar{c}$	ratio of absolute velocity
D	diameter (m)
$\overline{D}$	ratio of wheel diameter
f	friction loss coefficient
h	specific enthalpy (kJ kg^{-1})
h^*	stagnation enthalpy (kJ kg^{-1})
l	height of blade (m)
$\dot{m}$	mass flow rate (kg s^{-1})
p	pressure (MPa)
Re	Reynolds number
s	specific entropy (J kg^{-1} K^{-1})
T	temperature (°C)
u	peripheral velocity (m s^{-1})
$\bar{u}$	ratio of peripheral velocity
W	power output (kW)
w	relative velocity (m s^{-1})
$\overline{w}$	ratio of relative velocity
Δh	variation of specific enthalpy (kJ kg^{-1})

Greek symbols

α	absolute velocity angle (°)
β	relative velocity angle (°)
δ	tip clearance (m)
ζ	loss coefficient
η	efficiency
μ	viscosity (Pa s)
ρ	density (kg m^{-3})
τ	blockage factor
φ	nozzle velocity coefficient
ψ	rotor blade velocity coefficient
Ω	degree of reaction

Subscripts

1	rotor blade inlet
2	rotor blade outlet
B	exhaust pipe
c	critical state
cond	condensation
e	evaporation
endo	endothermic
f	friction loss
HP	high-pressure stage
ITLF	induction turbine layout form
LP	low-pressure stage
m	average
O	organic fluid
s	isentropic
STLF	separate turbine layout form
T	turbine
u	peripheral

Abbreviations

CEPCI	Chemical Engineering Plant Cost Index
GWP	Global Warming Potential
ITLF	Induction Turbine Layout Form
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
PEC	Purchased Equipment Cost
PTORC	Parallel Two evaporator Organic Rankine Cycle
SIC	Specific Investment Cost
STLF	Separate Turbine Layout Form
STORC	Series Two evaporator Organic Rankine Cycle

The ORC with multi-pressure evaporation is that involves two or more evaporation processes with different pressures and an isobaric condensation process. Multi-pressure evaporation ORCs increase the number of optimizable cycle parameters (e.g., evaporation stages; the pressure, working fluid mass flow rate, and superheat degree of each evaporation stage); and thus, the endothermic process of the cycle can be designed more suitably to adapt to the exothermic characteristics of the heat source fluid compared with that of the conventional single-pressure evaporation ORC [12]. Consequently, multi-pressure evaporation ORCs can significantly increase the power output or system efficiency by reducing the exergy loss in the cycle endothermic process and increasing the adaptability to various characteristic heat sources.

Several studies have proven the thermodynamic advantages of multi-pressure evaporation ORCs. For example, Li et al. [8] proposed the cycle structures of the series two evaporator organic Rankine cycle (STORC) and parallel two evaporator organic Rankine cycle (PTORC), and then compared their thermodynamic performance with that of the single-pressure evaporation ORC. Results showed that the net power output of the STORC system increased by 6.5–9.0% and that of the PTORC system increased by 3.3–4.5%, compared to the single-pressure evaporation ORC system [8]. Walraven et al. [13] compared the thermodynamic performance of the subcritical ORCs with one or more pressure levels, transcritical ORC, and Kalina cycle for 100–150 °C geothermal heat sources, and results showed that the subcritical ORC with multi-pressure evaporation could obtain the best thermodynamic performance. Sadeghi et al. [14] compared the thermodynamic performance of the single-pressure evaporation ORC, PTORC and STORC based on ten zeotropic mixtures; results indicated that the net power output of the STORC system could increase by 34.3% compared with that of the single-pressure evaporation ORC system. Manente et al. [15] focused on the five specific heat source temperatures with eight pure working fluids, and compared the thermodynamic performance of single-pressure and multi-pressure evaporation ORC systems. Results showed that the net power output of the multi-pressure evaporation ORC system could be 29% higher than that of the single-pressure evaporation ORC system [15]. Our previous work [16] also compared the thermodynamic performance of single-pressure and dual-pressure evaporation ORC systems for heat sources of 100–200 °C with nine pure organic fluids. Results indicated that the net power output of the dual-pressure evaporation type could increase by 21.4–26.7% compared to the single-pressure evaporation type, and the increment generally increased as the heat source temperature decreased [16].

The turbine is a typical and widely used expander in ORC systems. It is also a crucial component in achieving the heat–work conversion. Results of Li et al. [8] showed that the exergy loss of turbines could exceed 36% of the total exergy loss for the multi-pressure evaporation ORC system, and the maximum was nearly 42.3%. Wang et al. [17] also found that the exergy loss of turbines could exceed 32% of the total exergy loss for a multi-pressure evaporation ORC system. In addition, results of Lecompte et al. [18] indicated that the purchased equipment cost (*PEC*) of the turbine was the highest compared to other components in an ORC system. The same conclusion (the *PEC* of the turbine was the highest) was also found by Nazari et al. [19]. Moreover, Shu et al. [20] even found that the *PEC* of the turbine accounted for nearly 50% of the total *PECs* for an ORC system. However, existing studies on multi-pressure evaporation ORCs mainly focused on the optimization of the cycle endothermic process or the system performance comparison with other cycle structures of ORC. Few studies have focused on the