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Economic performances optimization of the transcritical Rankine cycle systems in geothermal application

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ABSTRACT

The aim of this study is to investigate the economic optimization of a TRC system for the application of geothermal energy. An economic parameter of net power output index, which is the ratio of net power output to the total cost, is applied to optimize the TRC system using CO_2, R41 and R125 as working fluids. The maximum net power output index and the corresponding optimal operating pressures are obtained and evaluated for the TRC system. Furthermore, the analyses of the corresponding averaged temperature differences in the heat exchangers on the optimal economic performances of the TRC system are carried out. The effects of geothermal temperatures on the thermodynamic and economic optimizations are also revealed. In both optimal economic and thermodynamic evaluations, R125 performs the most satisfactorily, followed by R41 and CO_2 in the TRC system. In addition, the TRC system operated with CO_2 has the largest averaged temperature difference in the heat exchangers and thus has potential in future application for lower-temperature heat resources. The highest working pressures obtained from economic optimization are always lower than those from thermodynamic optimization for CO_2, R41, and R125 in the TRC system.

1. Introduction

Exploitation of renewable energy has become an important topic due to the energy shortage and growing carbon dioxide emissions. Although the organic Rankine cycle systems, which contain lower boiling temperature working fluids, have great advantages and suitability for utilizing low-temperature heat source to produce useful power [1–7], there is a pinch point occurred between working fluid and heat source in a constant temperature boiling process of a pure fluid. This minimal temperature difference results in a largest resistance in heat transfer and causes a significant destruction in energy conversion [8]. The main difference between transcritical Rankine cycle, TRC, and organic Rankine cycle, ORC, is that the vapor generating pressure is larger than the critical pressure of working fluid for the former but is lower than the critical pressure for the latter. Therefore, a variable temperature can be achieved in vapor generator as heat energy is added to the working fluid for a better temperature profile to fit with the heat source [9]. In order to use TRC system for low-temperature heat energy conversion, it is an essential as well as important subject to seek suitable working fluids. One of the basic requirements in TRC system is that the critical temperature of working fluid must be lower than the temperature of heat source.

Geothermal energy is one of renewable resources with great potential to generate power. Due to the low-temperature applications, it will decrease the conversion efficiency and raises the equipment cost of the geothermal energy conversion systems. It becomes an important issue to maximize the energy conversion efficiency and to obtain the optimal economic performance for geothermal application. Baik et al. [10] analyzed the TRC system for the geothermal application using R125 mixtures as the working fluid to maximize the net power out with a fixed overall conductance of the heat exchangers. Their results showed that the optimized R125–R245fa mixture transcritical cycle yielded 11% more power than that of the optimized R134a subcritical cycle. Furthermore, the comparisons of optimal power output between TRC and ORC were made and reported [11,12]. They concluded that the produced power of transcritical cycle with R125 is about 14% higher than that with CO_2 [11]. The performance comparison and parametric optimizations of organic Rankine cycle and TRC for low-temperature geothermal power generation were revealed by Shengjun et al. [13]. They demonstrated that R125 in transcritical power cycle exhibited excellent economic and environmental per-

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Nomenclature

A_{con}	heat-transfer area of condenser, m^2
A_{vap}	heat-transfer area of vapor generator, m^2
B_1, B_2	bare module factor of equipment
C	cost, $
C_1, C_2, C_3	pressure factor of equipment
C_P	purchased equipment cost, $
CEPCI	chemical engineering plant cost index
C_{BM}	bare module cost, $
D	diameter, m
D_h	hydraulic diameter, m
F_P	pressure factor
F_M	material factor
f	Darcy friction factor
g	acceleration due to gravity, m s^{-2}
h	heat-transfer coefficient, kW m^{-2} $°C^{-1}$
i	enthalpy, kJ kg^{-1}
k	thermal conductivity, kW m^{-1} $°C^{-1}$
K_1, K_2, K_3	coefficients of equipment cost, $
L_t	thickness of tube wall, m
M	molecular weight of working fluid, g $mole^{-1}$
m	mass flow rate, kg s^{-1}
N	section number of the heat exchangers
NPI	net power output index, W $\$^{-1}$
P	pressure, MPa
P_r	Prandtl number
Q	heat transfer rate, kW
Re	Reynolds number
s	entropy, kJ kg^{-1} $°C^{-1}$
T	temperature, °C
$T_{r,i}$	working fluid inlet temperature, °C
$T_{r,o}$	working fluid outlet temperature, °C
$T_{w,i}$	geothermal source or cooling water inlet temperature, °C
$T_{w,o}$	geothermal source or cooling water outlet temperature, °C
ΔT	temperature difference between inlet and outlet of the heat exchanger, °C
ΔT_{con}	averaged temperature difference in the condenser, °C
ΔT_{mean}	logarithmic mean temperature difference, °C
ΔT_{vap}	averaged temperature difference in the vapor generator, °C
U	overall heat-transfer coefficient of the heat exchanger, kW m^{-2} $°C^{-1}$
v	specific volume, m^3 kg^{-1}
W	power of the turbine or pump, kW
X	equipment type
Y	the capacity or size parameter of equipment, kW or m^2

Greek symbols

Δ	relative error, difference
η	efficiency
μ	dynamic viscosity, Pa s
ρ	density, kg m^{-3}

Subscripts

b	bulk
con	condensation, condenser
cw	cooling water
f	liquid
g	vapor
ge	geothermal source
i	inside, inlet
j	section
max	maximal
net	net
o	outside, optimization
pum	pump
r	working fluid
t	tube
th	thermal
tot	total
tur	turbine
vap	vapor generator
w	geothermal source, or cooling water
wall	tube wall of heat exchangers

Acronyms

GWP	global warming potential
ODP	ozone depletion potential
TRC	transcritical Rankine cycle

formance and could maximize utilization of the geothermal energy. Although the thermal efficiency of R125 in TRC is 46.4% lower than that of R123 in ORC, it provides 20.7% larger recovery efficiency. To maximize the plant efficiency for a geothermal power plant, Walraven et al. [14] illustrated the performance of thermodynamic cycles for power production from low-temperature geothermal heat sources. They found that due to the low heat source temperatures, a low condensation temperature was very important for performance improvement. Furthermore, Chen et al. [15] investigated the performance of TRC system using zeotropic mixtures as working fluids to turn the low-grade heat into power.

Carbon dioxide (CO_2), which is non-flammable, non-toxic, low cost and abundance in nature, and provides superior environmental protection, has been extensively considered as a supercritical working fluid by a number of researchers [12,16–18] in energy conversion power system. Cayer et al. [16] numerically analyzed a CO_2 transcritical power cycle for a low temperature source. In their study, energy analysis, exergy analysis and calculation of the heat exchangers area were considered as the major methodologies. Thermo-economic analysis of a CO_2 TRC system was numerically reported by Li et al. [17] using the low-temperature geothermal source with the temperature ranging from 90 °C to 120 °C. Furthermore, TRC system using CO_2 blending with low GWP working fluids for low-grade heat energy recovery was investigated by Dai et al. [19]. However, having 6–12 MPa operating pressures, the TRC system with CO_2 raises safety and high cost concerns. They may be considered as the disadvantages for CO_2 application. Le et al. [20] showed that higher exergy efficiencies were achieved for the system with zeotropic mixtures than with pure CO_2 due to the more suitable thermal match in the heat transfer process. Among their selected supercritical working fluids, the highest optimal thermal efficiency of the TRC system was always obtained with R152a in both basic (11.6%) and regenerative (13.1%) configurations. In addition, a transcritical CO_2–R1270 cascade system for cooling and heating applications was theoretically analyzed by Dubey et al. [21]. Their result showed that transcritical cycle with CO_2–propylene gave better system performance than subcritical cascade cycle. Velez et al. [22] thermodynamically analyzed the TRC system with CO_2 for power generation from waste heat energy. They reported that an increase up to 300% for the