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Thermal and economic analyses of a compact waste heat recovering system for the marine diesel engine using transcritical Rankine cycle

Min-Hsiung Yang *

Department of Naval Architecture and Ocean Engineering, National Kaohsiung Marine University, Taiwan, Republic of China

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ABSTRACT

The aim of this study is to investigate the economic performance of a novel compact waste heat recovering system for the marine diesel engine. The transcritical Rankine cycle is employed to convert the waste heat resources to useful work with R1234yf. To evaluate the utilizing efficiency and economic performance of waste heat resources, which are exhaust gas, cylinder cooling water and scavenge air cooling water, three operating models of the system are investigated and compared. The levelized energy cost, which represents the total cost per kilo-watt power, is employed to evaluate the economic performance of the system.

The economic optimization and its corresponding optimal parameters of each operating model in the compact waste heat recovering system are obtained theoretically. The results show that the minimal levelized energy cost of the proposed system operated in Model I is the lowest of the three models, and then are Model II and Model III, which are 2.96% and 9.36% lower for, respectively. Similarly, the CO_2 emission reduction is the highest for Model I of the three models, and 21.6% and 30.1% lower are obtained for Model II and Model III, respectively. The compact waste heat recovering system operated in Model I has superiority on the payback periods and heavy diesel oil saving over the others. Finally, the correlations using specific work of working fluid and condensation temperature as parameters are proposed to assess the optimal conditions in economic performance analysis of the system.

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1. Introduction

Lately, due to the topics of energy saving and CO_2 emissions reduction for environmental protection, waste heat recovery has become an important issue for energy utilization in the world. The organic Rankine cycle (ORC) system is an effective method to recover and reuse for waste heat sources. Wei et al. [1] reported the performance optimization of an ORC system with R245fa for exhaust heat recovery. The parameter optimizations of the ORC systems were performed by Dai et al. [2] with 10 different working fluids using a genetic algorithm. Li et al. [3] investigated the economic optimization on the ORC system for recovering the waste heat of flue gas from industrial boilers. Furthermore, a multi-objective optimization of the ORC with R134a was conducted to achieve the optimization design from both thermodynamic and economic aspects using non-dominated sorting genetic algorithm [4].

Because the ORC has lower-evaporated-temperature working fluids, it has great potential for low temperature heat sources to produce useful power, such as solar energy, geothermal energy, and the waste heat of industrial and internal combustion engine. Thermodynamic optimization of a solar ORC system was depicted by Delgado-Torres and García-Rodríguez [5]. Later, based on photovoltaic technologies, Casati et al. [6] revealed the performance of energy storage for solar ORC application. Rayegan and Tao [7] developed a new procedure to compare capabilities of various working fluids for solar energy utilization. Moreover, Guo et al. [8] demonstrated the thermodynamic performance of a novel cogeneration system consisted of an ORC power generator and a heat pump using geothermal energy. To utilize geothermal energy effectively, the performance comparisons of the ORC and the Kalina cycle were investigated by Walraven et al. [9]. An orthogonal design method was proposed by Wang et al. [10] to evaluate both of the thermodynamic performance and economic performance for geothermal resource application. In addition, for energy and green house gas savings, Campana et al. [11] investigated the effects of using ORC system to recover waste heat from intensive industries in Europe. By using ORC system, the results of recovering waste heat from the cooling water system of a large marine engine was

* Address: No. 142, Haizhuan Rd., Nanzi Dist., Kaohsiung City 81157, Taiwan, Republic of China. Tel.: +886 7 3617141x3404; fax: +886 7 3656481.

E-mail address: mhyang@webmail.nkmu.edu.tw

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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, $
c_p	specific heat, kJ kg^{-1} °C^{-1}
$CEPCI$	chemical engineering plant cost index
C_{BM}	bare module cost, $
D	diameter, m
D_h	hydraulic diameter, m
ED	exergy destruction, kW
F_P	pressure factor
F_M	material factor
g	acceleration due to gravity, m s^{-2}
h	heat-transfer coefficient, kW m^{-2} °C^{-1}
I	irreversibility, kW
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}$
Nu_r	Nusselt number
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
$T_{eg,i}$	exhaust gas inlet temperature, °C
$T_{eg,o}$	exhaust gas outlet temperature, °C
ΔT_{con}	averaged temperature difference in the condenser, °C
ΔT_{mean}	logarithmic mean temperature difference, °C
$T_{r,i}$	working fluid inlet temperature, °C
$T_{r,o}$	working fluid outlet temperature, °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 expander or pump, kW
X	equipment type
Y	the capacity or size parameter of equipment, kW or m^2

Greek symbols

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

Subscripts

con	condensation, condenser
cw	cooling water
eg	exhaust gas
exp	expander
f	liquid
g	vapor
H	high
i	inside, inlet
j	section
L	low
max	maximal
net	net
o	outside, optimization
pum	pump
r	working fluid
t	tube
th	thermal
vap	vapor generator
wall	tube wall of heat exchangers

Acronyms

CWHRS	compact waste heat recovering system
EEDI	energy efficiency design index
IMO	International Maritime Organization
LEC	levelized energy cost
LMTD	logarithmic mean temperature difference
ORC	organic Rankine cycle
RC	Rankine cycle
TRC	transcritical Rankine cycle

carried out numerically by Yang and Yeh [12]. Then, the thermodynamic and economic performances optimization of utilizing exhaust gas from a large marine diesel engine also reported [13].

Nevertheless, in the ORC as heat energy transfers from heat source to working fluid in the evaporator, a pinch point problem exists due to the evaporation at constant saturated temperature of the working fluid. This minimal temperature difference leads to an obstruction for heating process in energy conversion [14]. Alternatively, the transcritical Rankine cycle (TRC) is developed with a variable temperature profile of the working fluid which is heated in the vapor generator to obtain a better effect of heat transfer [15]. However, another important restriction for the TRC system application to point out is that the critical temperature of working fluid must be lower than the temperature of waste heat source [16]. It also becomes a basic criterion to choose the favorable working fluid. The carbon dioxide (CO_2), which possesses lower critical temperature (T_{cri} = 30.98 °C) and provides superior properties for environmental protection, has been investigated widely as a favorable transcritical working fluid to convert heat resources to electricity generation [17]. In a CO_2 TRC system, four object parameters which were energy analysis, exergy analysis, finite size thermodynamics and heat transfer area calculation were evaluated to assess the performance for low-grade stream utilization [18]. Velez et al. [19] reported that in a CO_2 TRC system, an increase up to 25% for the exergy efficiency, and up to 300% for the energy efficiency can be obtained when the inlet temperature of the turbine was risen from 60 to 150 °C. Furthermore, Dai et al. [20] depicted that maximum exergy efficiencies existed for the TRC at the corresponding optimal pressures for 7 CO_2 mixtures. Among these mixtures, CO_2/R161 was recommended for small capacity instruments and high efficiency.

However, to overcome the exorbitant critical pressure of the CO_2 (P_{cri} = 7.38 MPa), the power consumption of pump increases significantly and the purchased cost of equipment also rises obviously. It also leads to a significant disadvantage for CO_2 applied in the TRC system [21]. To seek other favorable working fluids for TRC, only simple first law analysis is not sufficient for the working fluid selection and performance comparison. Many studies

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