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Research Paper

# Economic research of the transcritical Rankine cycle systems to recover waste heat from the marine medium-speed diesel engine

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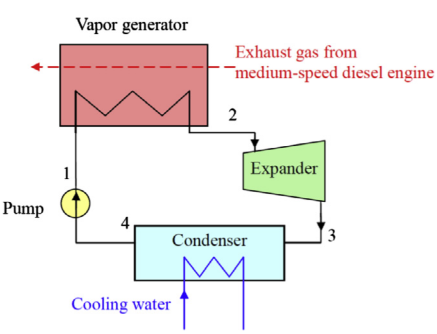
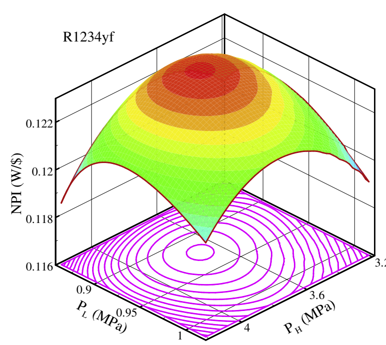
## GRAPHICAL ABSTRACT

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## ABSTRACT

The aim of this study is to investigate the economic performance of a transcritical Rankine cycle (TRC) system for recovering waste heat from the exhaust gas of a marine medium-speed diesel engine. The variation of net power output, total cost of equipments and exergy destruction are investigated for the TRC system. Furthermore, to evaluate the economic performance of energy utilization, a parameter, net power output index, which is the ratio of net power output to the total cost, is introduced of the TRC system using R125, R143a, R218 and R1234yf as working fluids. The results show that R1234yf performs the highest economic performance, followed by R143a, R125 and R218 of the TRC system. It reveals that R1234yf not only has the smallest high and low pressures of the TRC system for reducing the purchased cost of equipments, but also promotes a larger pressure ratio of the expander for generating power output among these working fluids. The comparisons of optimal pressure ratios obtained from thermodynamic and economic optimizations for these working fluids in the TRC system are also reported. In addition, an evaluation method using thermal efficiency and operating pressure ratio as parameters is proposed to assess the suitability of the working fluids of TRC system in economic analysis for waste heat recovery from the exhaust gas of a diesel engine.



## 1. Introduction

Recently, waste heat recovery has become an important topic for energy utilization due to the energy shortage and growing carbon dioxide emissions in the world. The organic Rankine cycle system has advantages to recover waste heat source to produce useful power [1–3]. However, a pinch point exists between working fluid and the heat source in the evaporator on account of the

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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 |
| $ED$ | 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 | Nusselt number |
| $P$ | pressure, MPa |
| Pr | 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 |
| $\Delta T_{con}$ | averaged temperature difference in the condenser, °C |
| $\Delta T_{mean}$ | logarithmic mean temperature difference, °C |
| $T_{r,i}$ | working fluid inlet temperature, °C |
| $T_{r,o}$ | working fluid outlet temperature, °C |
| $\Delta T_{vap}$ | 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*

| | |
|---|---|
| $\Delta$ | relative error, difference |
| $\eta$ | efficiency |
| $\mu$ | dynamic viscosity, Pa–s |
| $\rho$ | 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*

| | |
|---|---|
| EEDI | energy efficiency design index |
| GWP | global warming potential |
| IMO | International Maritime Organization |
| LMTD | logarithmic mean temperature difference |
| ODP | ozone depletion potential |
| TRC | transcritical Rankine cycle |

constant saturated temperature in evaporating process. Compared with organic Rankine cycle, the transcritical Rankine cycle, TRC, is developed without this minimal temperature difference while heating in the vapor generator. Since the vapor generating pressure is larger than the critical pressure of working fluid in the TRC system, a variable temperature profile of the working fluid can be obtained to fit with the heat source during heating process [4–6]. However, an important restriction for the TRC system application is that the critical temperature of working fluid must be lower than the temperature of waste heat source.

To select suitable working fluids for TRC system application efficiently is a basic issue. The natural material of carbon dioxide ($CO_2$), which provides superior properties for environmental protection, has been widely studied as a transcritical working fluid of the TRC system [7–14]. Yamaguchi et al. [7] and Zhang et al. [8] found that $CO_2$ was an efficient supercritical working fluid to convert solar power to electricity generation. Since the $CO_2$ cycle showed no pinch limitation in the heat exchanger, Chen et al. [9] reported that in thermodynamic analysis, a TRC system with $CO_2$ gave a slightly higher power output than the organic Rankine cycle. With the heat source and heat sink temperatures of 100 °C and 10 °C for the $CO_2$ TRC system, respectively, Cayer et al. [10] revealed that the optimal high pressure for maximizing thermal and exergetic efficiencies was about 13.5 MPa. Dai et al. [11] investigated the effects of zeotropic and $CO_2$ mixtures to improve the thermal efficiency of the TRC system. Furthermore, Velez et al. [12] pointed out that $CO_2$ TRC system had an advantage to produce power from waste heat at low temperature. However, the excessively high operating pressure of the $CO_2$ in the TRC raises equipment purchased cost significantly [13]. It may be considered as the main disadvantage in economic consideration for $CO_2$ application in the TRC system [14].

In addition, R125 is also a popular working fluid applied in the TRC systems [14–19]. From the economic performance investigation of Yang and Yeh [14], it was reported that the TRC system operated with R125 can reduce the proportion of equipment purchased cost significantly attributed to its lower operating pressures. With compared performance of $CO_2$, ethane, and R125 for a low-grade heat-source-driven transcritical cycle, Cayer et al. [15] indicated that R125 exhibited a superior performance in thermal efficiency evaluation. Furthermore, the comparisons of TRC and organic Rankine cycle systems using R125 mixture as the

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