Ships and Offshore Structures

ISSN: (Print) (Online) Journal homepage: https://www.tandfonline.com/loi/tsos20

Thermo-environmental analysis and performance optimisation of transcritical organic Rankine cycle system for waste heat recovery of a marine diesel engine

Mehmet Akman & Selma Ergin

SHIPS AND OFFSHORE STRUCTURES
https://doi.org/10.1080/17445302.2020.1816744

Taylor & Francis
Taylor & Francis Group

Check for updates

Thermo-environmental analysis and performance optimisation of transcritical organic Rankine cycle system for waste heat recovery of a marine diesel engine

Mehmet Akman [a] and Selma Ergin [b]

[a]Department of Motor Vehicles and Transportation Technologies, Mugla Sitki Kocman University, Bodrum, Turkey; [b]Department of Naval Architecture and Marine Engineering, Istanbul Technical University, Istanbul, Turkey

ABSTRACT

Energy efficient and environmentally friendly shipping have been primary concerns for the maritime industry. One of the alternatives to overcome these issues onboard is organic Rankine cycle (ORC) waste heat recovery system (WHRS). In this study, a transcritical ORC WHRS for a marine diesel engine is investigated at different engine operating loads by thermodynamic and environmental analysis. The engine exhaust gas is used as the waste heat source and R152a is selected as the working fluid. The energetic, exergetic and environmental parameters are analysed and the performance optimisation is conducted by using genetic algorithm. The results indicate that by employing the ORC system onboard, it is possible to increase the overall thermal efficiency of the ship power generation system by more than 2.5% and the system can save up to 678.1 tonnes CO_2 per year when the system is operated at the optimal conditions.

ARTICLE HISTORY
Received 1 December 2019
Accepted 26 August 2020

KEYWORDS
Transcritical organic Rankine cycle; waste heat recovery; marine diesel engine; energy efficiency; exergy analysis; environmental analysis

Acronyms

Nomenclature

CO_2	Carbon dioxide
CO	Carbon monoxide
DWT	Deadweight tonnage
EL	Engine load
GHG	Greenhouse gas
GWP	Global warming potential
IMO	International Maritime Organisation
LHV	Low heating value
NAV	Navigation
NO_X	Nitrogen oxide
ODP	Ozone depletion potential
ORC	Organic Rankine cycle
PGS	Power generation system
PM	Particulate matter
SFOC	Specific fuel oil consumption
SMCR	Specific maximum continuous rating
SO_2	Sulphur dioxide
SRC	Steam Rankine cycle
SWH	Service water heater
T/C	Turbocharger
VOCs	Volatile organic compounds
WHRS	Waste heat recovery system

Greek symbols

η	Efficiency
γ	Emission factor
ε	Recovered power ratio

Symbols

$\dot{m}$	Mass flow rate (kg/s)
h	Specific enthalpy (kJ/kg)
P	Pressure (kPa)
$\dot{Q}$	Heat flow (kW)
R	Emission reduction (ton)
s	Specific entropy (kJ/kg-K)
T	Temperature (K)
t	Time (s)
$\dot{W}$	Turbine, pump or electrical power (kW)
x	Specific exergy (kj/kg)
$\dot{X}$	Exergy destruction rate (kW)

Subscripts

b	Break
con	Condenser
e	Electricity
ex	Exergy
exh	Exhaust gas
f	Working fluid
fw	Fresh water
he	Heat exchanger
i	Component
nav	Navigation
oh	Operational hours
p	Pump
pgs	Power generation system
sw	Sea water
t	Turbine
th	Thermal

1. Introduction

Energy efficiency, energy economy and environmental concerns have led companies and researchers from many industries to find solutions for the ongoing problems. Waste heat recovery is a key method to overcome these concerns arose in various industries. Therefore, many studies have been conducted for the waste heat potential and recovery. Brueckner

CONTACT Selma Ergin ergin@itu.edu.tr Department of Naval Architecture and Marine Engineering, Istanbul Technical University, Maslak, Istanbul 34469, Turkey

et al. (2017) investigated the industrial waste heat potential in Germany and evaluated the exhaust gas data from German industrial emission data. Miró et al. (2017) assessed the industrial waste heat potential of the European non-metallic mineral industry. Soltani et al. (2015) investigated a multigenerational energy system fuelled with sawdust biomass and carried out the energy and exergy analysis of the system. Yağlı et al. (2016) designed a system to recover the exhaust gas waste heat of biogas fuelled combined heat and power (CHP) engine. Mossafa et al. (2016) conducted a thermo-economic analysis for the waste heat recovery system which uses geothermal fluid energy as the low-grade heat source and cold energy of LNG as thermal sink. As terrestrial applications, waste heat recovery in maritime industry is also on the agenda and researches are focusing on energetic and environmental issues. Olmer et al. (2017) reported that total fuel consumption in shipping increased from 291 million tonnes to 298 million tonnes between the years 2013 and 2015 which means the increase of CO_2 emissions by 2.5%. The shares of nitrogen oxides (NO_X) and sulphur oxides (SO_X) in the global emissions are about 15% and 13%, respectively (Mondejar et al. 2018). As measures, International Maritime Organisation (IMO) regulated the limits related with NO_X and SO_X emissions as given in Annex VI of International Convention for the Prevention of Pollution from Ships (ICPP 2015). In addition, the energy efficiency design index (EEDI) has been mandatory for new ships and the ship energy efficiency management plan (SEEMP) have become a requirement for all ships with a gross tonnage of 400 tons and above (IMO 2014). To cope with the mentioned issues, organic Rankine cycle system which uses the waste heat sources of marine engines offers promising solutions. In comparison with the classical Rankine cycle, organic working fluid is operated in the system instead of steam.

Lately, many studies have been conducted in organic Rankine cycle waste heat recovery (WHR) systems in maritime industry. Considering the energy flow of the high efficient marine diesel engines, mechanical or electrical power are produced from about 50% of total fuel energy while the remaining part is lost by cooling (MAN Diesel & Turbo 2014). Song et al. (2015) investigated the performance of ORC system which uses the jacket cooling water and exhaust gas as waste heat sources. Suarez and Greig (2013) analysed the ORC exhaust gas waste heat recovery system for a large marine diesel engine. They compared different working fluids including water in their study and reported that comparing classical Rankine cycle, ORC gives better results. Yang and Yeh (2015) examined the thermodynamic and economic performances of four different working fluids used in ORC exhaust gas waste heat recovery. Yang (2016) investigated the economic performance of ORC WHR system which uses exhaust gas, jacket cooling water, scavenge air cooling water and lubricating oil of marine diesel engine. Nawi et al. (2018) evaluated the performance of ORC exhaust waste heat recovery from marine diesel engines using bioethanol produced by three different microalgae. Song et al. (2018) analysed the transcritical CO_2 Rankine cycle integrated with organic Rankine cycle for a heat recovery system. Their study revealed that the bottoming ORC system recovered significantly the residual heat generated by the topping S-CO_2 system. Braimakis and Karellas (2018) conducted energetic optimisation of regenerative ORC configurations. Mito et al. (2018) analysed the scavenge air and exhaust gas waste heat recovery system operating at single and dual pressures for a marine diesel engine. In their study, steam was selected as the working fluid and it was reported that the scavenge air and exhaust gas WHRS operating at single pressure is more efficient than dual pressure system. Authors of the study, Akman and Ergin (2016, 2019) investigated the energetic and environmental performances of ORC WHR system for a handymax size tanker. In their studies, different ORC WHR configurations were analysed using jacket cooling water, scavenge air and exhaust gas waste heat sources.

Fluid selection, on the other hand, is one of the most important process for ORC systems. Therefore, there are also many studies on this topic. Zhu et al. (2018) investigated the ORC exhaust gas WHR system under operating conditions and they reported that R141b showed better energetic and economic performance. Hou et al. (2018) performed thermodynamic and exergoeconomic analysis for a combined transcritical CO_2 and regenerative organic Rankine cycle using zeotropic mixture. Wang et al. (2011) analysed the performance of different working fluids for waste heat recovery from the internal combustion engines and the results showed that R11, R113, R141b and R123 have better thermodynamic performance in comparison with R245fa and R245ca. Larsen et al. (2013) proposed a methodology based on the principles of natural selection to obtain the optimum working fluids for marine waste heat recovery applications. According to their results, R245fa, R236ea and RC318 are favourable based on energetic performance and safety.

The optimisation studies for ORC WHR systems are generally conducted by conventional methods. Genetic algorithm (GA) method is also a promising optimisation tool for ORC systems. Dai et al. (2009) used GA for the optimisation of ORC WHRS by using exergy efficiency as the objective function. Xi et al. (2013) investigated the performances of three different regenerative ORC configurations by using six different working fluids and optimisation was performed by GA selecting the exergy efficiency as the objective function. Riyanto et al. (2014) analysed ORC WHR system by using four working fluids and optimised the system by using GA. There are few studies on the performance optimisation of ORC WHRS with GA. Moreover in the literature, the optimisation processes are carried out at constant engine load. In this study, a transcritical ORC system is modelled for the exhaust gas waste heat recovery of a two-stroke marine diesel engine installed on a chemical/oil tanker. In the analysis, R152a is selected as the working fluid. The thermodynamic and environmental analysis are conducted and the performance parameters of the transcritical system is optimised with using the genetic algorithm where different engine loads are considered as the novel approach.

2. The ORC WHR model onboard ship

Thermal efficiencies of diesel engines have reached up to 53% (IMO 2009). Singh and Pedersen (2016) stated that a two-stroke diesel engine with 68640 kW (at 94 r/min) brake power, releases 25.5% of fuel energy as exhaust and the rest 16.5%, 5.2%, 2.9% and 0.6% of fuel energy are shared by scavenge air, jacket water, lubrication oil, heat radiation, respectively. Therefore, the amount of waste energy is very promising to recover.

Table 1. Main particulars of tanker (Akman and Ergin 2016).

Properties	Value
Deadweight	49,990 tonnes
Installed power (main engine) – MAN B&W 6G50ME	10,320 kW
Installed power (auxiliary engine)	840 kW (x2)
Navigation electric load	524 kWe
Cooling system heat dissipation	6000 kW

In this study, MAN B&W 6G50ME model two-stroke diesel engine rating 10,320 kW at 100 rpm is used for the analysis. The engine is installed on a medium-range tanker and the engine data used in this study are taken from the authors' previous analysis (Akman and Ergin 2016). Table 1 shows the main specifications of the tanker.

The temperature, mass flow rate of exhaust gas and the specific fuel oil consumption (SFOC) of the main engine at standard conditions are shown in Figure 1 (MAN 2019). The SFOC is minimum between 65% and 75% maximum continuous rating (MCR).

The change of the exhaust gas temperature after the turbocharger with the engine load is similar to the change of the SFOC with the engine load. Based on the given data, the exhaust gas heat is calculated for different engine loads. To recover the waste energy, the transcritical ORC system as given in Figure 2 is evaluated.

The analysed system consists of a super-heater, turbine, service water heater (SWH), condenser and a pump. Instead of using regeneration, reheating or combined cycle, a basic cycle integrated with a service water heater (SWH) is proposed. SWH is used after the turbine outlet of the WHR system to fulfil the ship's domestic hot water requirement which is conventionally supplied by the steam generated by auxiliary boiler. Moreover, R152a is selected as the working fluid based on its availability for transcritical analysis (Yang 2016). Nazari et al. (2016) analysed the subcritical steam cycle integrated with a transcritical ORC for recovering the waste heat of a gas turbine and they proposed R152a as the ORC working fluid from thermodynamic and exergo-economic point of view. Gao et al. (2012) also recommended R152a as the working fluid for transcritical ORC systems used in low and medium grade heat recovery. Moreover, R152a has environmentally friendly ODP and GWP values in comparison with the most fluids used in the literature. The properties of the fluid are given in Table 2.

The turbine inlet pressure and temperature of the system are increased from 4.6 to 6.6 MPa and from 388.15 to 498.15 K, respectively. To prevent the acid formation, the exhaust gas temperature after the ORC process is taken as 433.15 K (Suarez and Greig 2013). The parameters for the steady-state system analysis are given in Table 3. After the expansion process, when the turbine is operated at high inlet temperature and pressure, the outlet temperature of the turbine is quite high. Therefore, the thermal efficiency of the system is implicitly increased after the service water heating process.

3. Methodology

The ORC system is analysed and the performance parameters are evaluated. The energy balance of the system can be determined as follows.

The heat transfer from the exhaust gas is given by:

$$\dot{Q}_{in} = \dot{m}_f.(h_3 - h_2) \quad (1)$$

The cooling load in the condenser can be expressed as:

$$\dot{Q}_{out} = \dot{m}_f.(h_4 - h_1) \quad (2)$$

The pump power is given by:

$$\dot{W}_P = \dot{m}_f.(h_{2a} - h_1)/\eta_P \quad (3)$$

The turbine power can be expressed by:

$$\dot{W}_t = \dot{m}_f.(h_3 - h_{4a}).\eta_t \quad (4)$$

The generated electrical power can be expressed by:

$$\dot{W}_e = (\dot{W}_t - \dot{W}_P).\eta_g.\eta_m \quad (5)$$

The thermal efficiency of the cycle can be expressed by:

$$\eta_{th-ORC} = \frac{\dot{m}_f.(h_3 - h_{4a}).\eta_t - \dot{m}_f.(h_{2a} - h_1)/\eta_P}{\dot{m}_f.(h_3 - h_2)} \quad (6)$$

where the numerator of the equation indicates the net power ($\dot{W}_{net}$) of the cycle. Recovering the waste heat and converting it into mechanical power increase the thermal efficiency of the power generation system. Then, the new PGS thermal efficiency can be calculated as:

$$\eta_{th,pgs} = \frac{P_b + \dot{W}_{net}}{Q_{LCV}.\dot{m}_{fuel}} \quad (7)$$

The exergy analysis is expressed as follows. The exergy of each state point is given by:

$$x_i = (h_i - h_0) - T_0(s_i - s_0) \quad (8)$$

where h_0 and s_0 the enthalpy and the entropy at the ambient conditions which are 101.35 kPa for the pressure (P_0) and 298.15 K for the temperature (T_0). The exergy balance is:

$$\sum_{in}^{x} - \sum_{out}^{x} = x_{des} \quad (9)$$

where x_{des} is the exergy destruction per unit mass. Then, the exergy efficiency is given as:

$$\eta_{ex} = \frac{\dot{W}_{net}}{\dot{Q}_{in}.\left(1 - \frac{T_0}{T_m}\right)} \quad (10)$$

where T_m is the average temperature of the heat source which can be expressed as:

$$T_m = \frac{T_{in} - T_{out}}{ln\left(\frac{T_{in}}{T_{out}}\right)} \quad (11)$$

The environmental analysis can be performed by calculating the recovered power ratio which is given by:

$$\varepsilon = \frac{\dot{W}_{net}}{P_b} \quad (12)$$

where P_b is the brake power at given engine load. Then, the

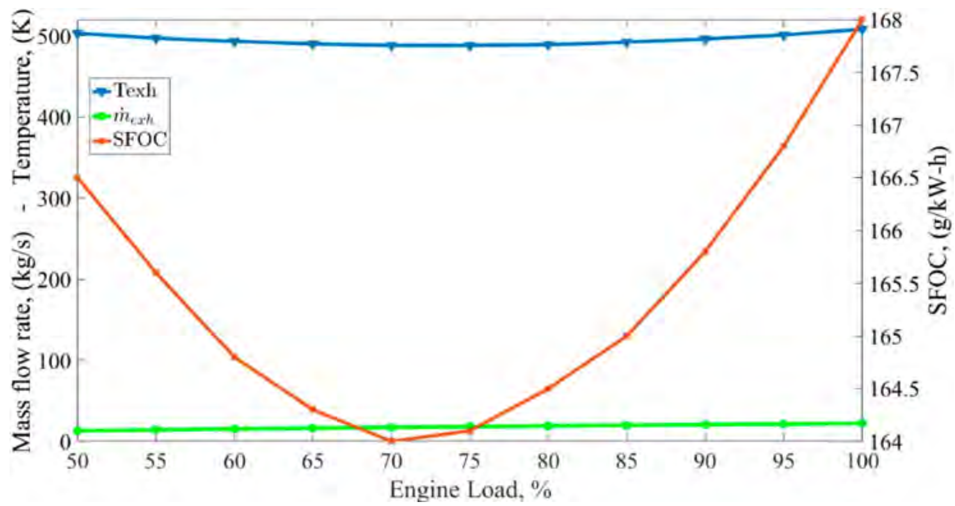

Figure 1. Exhaust gas properties and SFOC with respect to engine load (MAN 2019). (This figure is available in colour online.)

emission reduction in terms of mass can be expressed as:

$$R_i = P_b.SFOC.\varepsilon.t_{oh}.\gamma_i \tag{13}$$

where γ is the emission factor in kg per ton for the heavy fuel oil. The emission factors in kg per ton for CO_2, NO_X, SO_2, PM, CO, and VOCs are taken as 3170, 87, 54, 7.6, 7.4 and 2.4, respectively (Trozzi and Vaccaro 1998).

The exergy efficiency is selected as fitness function for GA during optimisation process. The real number vectors are used as the chromosomes which consist of the turbine inlet pressure and temperature and the average source temperature. The function is defined as:

$$\eta_{ex} = f(P, T, T_m) \tag{14}$$

The genetic algorithm (GA) configuration is given in Table 4 (Xi et al. 2013). In this method, there are three operators as the selection, crossover and mutation. The selection operator selects parents for the next generation. The crossover operator forms the new chromosomes and the mutation operator modifies these chromosomes for the converging. The flow chart for the simulation and optimisation process is given in Figure 3.

4. Results and discussions

The simulation codes are written in MATLAB 2016a and the thermodynamic properties are called from REFPROP 9.0. The energetic, exergetic and environmental parameters at different engine operating conditions are analysed and the performance optimisation is conducted. The validation study is carried out by comparing the results with the available data in the literature.

4.1. Energetic and exergetic analysis

The calculations are carried out for different turbine inlet temperature and pressure values and for different engine operating conditions to investigate the performance parameters of the ORC system. Figure 4 shows the change of the mass flow rate with the turbine inlet temperature (maximum cycle temperature) at the maximum turbine inlet pressure of 6.468 MPa. The mass flow rate decreases when the turbine inlet temperature increases. This is due to the high enthalpy difference at high temperatures. At the high engine loads, the mass flow rate also increases with the

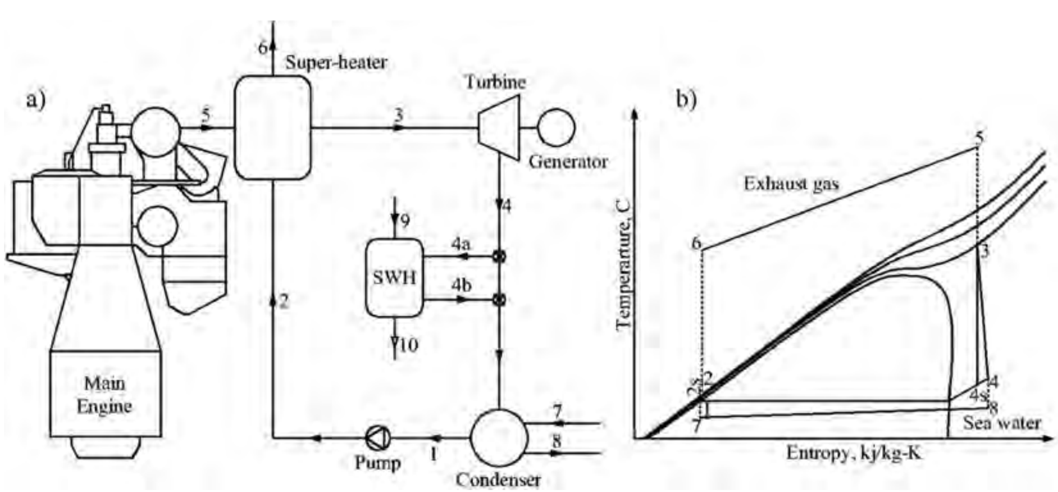

Figure 2. ORC WHR model (a) and T-s diagram (b).

Table 2. Properties of the working fluid, R152a (NIST 2010).

Properties	Value
Molecule weight, g/mol	66.05
Normal boiling point, K	249.13
Critical temperature, K	386.41
Critical pressure, kPa	4516.8
Max. applicable temperature, K	500
ODP	0
GWP (100 years)	6

turbine inlet temperature. Since, the amount of heat transfer is high from the super-heater.

Figure 5 presents the change of the pump power with the turbine inlet temperature for the pressure of 6.468 MPa. The pump power also decreases with the increasing turbine inlet temperature. The high operating pressures require high pumping powers. When the temperature of the working fluid increases the mass flow rate decreases. This results in low pumping power, as expected. At the high engine loads, the mass flow rate increases. This means the system needs more pumping power. The pumping power has the highest value as 102.27 kW at the turbine inlet temperature of 388.15 K and at full load. The corresponding mass flow rate is calculated as 13.19 kg/s. The regular operation load of the main engine is about 80% to 85% of MCR so that the required pumping power is lower.

When the turbine inlet temperature and pressure values are high, as expected, the mechanical power generation is also high. However, the exhaust gas temperature after turbocharger limits the turbine inlet temperature based on the pinch point. The change of the turbine power with the turbine inlet temperature is presented in Figure 6. According to Figure 6, the turbine power is remarkably high at high engine loads and at the turbine inlet temperatures. At 100% MCR, 391.17 kW turbine power is gained at 6.468 kPa and at 498.15 K operating conditions.

The mechanical power is converted to the electrical power by the ORC WHRS. The produced electrical powers at different turbine inlet temperatures and at changing engine loads are shown in Figure 7. There are two service generators onboard the ship for supplying the electrical power requirement of navigation and cargo handling. As can be seen from Figure 7, when the engine is operated at its regular load (85% MCR), the ORC WHR system supplies up to 242.73 kW which can remarkably

Table 3. The parameters used in this study (Yang and Yeh 2015; Grljušić 2015).

Section	Parameter	Value
Condenser	Sink temperature	298.15 K
Turbine	Isentropic efficiency (η_t)	0.85
Pump	Isentropic efficiency (η_p)	0.85
Working fluid	Condensation temperature	308.15 K
Exhaust gas	Pressure drop	3 kPa
Heat exchangers	Efficiency (η_{he})	98%
	Pressure drop	2% of P_{max}
	Pinch point temperature difference	5 K
Generator	Mechanical efficiency (η_m)	96%
	Generator efficiency (η_g)	95%
Main engine	Operational hours	5796 h
Fuel	LHV	42,700 kj/kg
Service water	Inlet temperature ($T_{fw,in}$)	298.15 K
	Outlet temperature ($T_{fw,out}$)	323.15 K

Table 4. Configuration of the genetic algorithm.

Parameter	Value
Chromosome	[P_i, T_i, $T_{m,i}$]
Population size	200
Crossover probability	0.4
Mutation probability	0.2
Elite count	20

decrease the generator load. At full load, the system can generate 331.29 kW electrical power.

The thermal efficiency of the ORC WHR system is presented in Figure 8. When the turbine inlet temperature and pressure increase, the thermal efficiency of the system also increases. However, at lower turbine inlet temperatures based on the quality of the fluid after the expansion process, thermal efficiency values are lower as well. The selected operational points of the cycle are in the transcritical region therefore the thermal efficiency becomes high. The maximum thermal efficiency is obtained as 16.78% at 498.15 K and 6.468 MPa as can be seen in Figure 8.

After ORC WHRS integration, the thermal efficiency of the power generation system rises up to 50.41% as can be seen in Figure 9. According to the figure, high efficiency values are obtained between 60% MCR and 80% MCR when the ORC system is operated at above 420 K and 6.468 MPa. It should be noted that although the generated power is higher at heavy loads than the light loads, the thermal efficiencies are lower at high load points.

The exergy efficiency of the ORC system is also the function of temperature and pressure of the cycle. Moreover, the source temperature directly affects the exergetic performance so that the minimum temperature differences are aimed during optimisation. In Figure 10, the exergy efficiencies change between 24.82% and 46.06% at 85% MCR. At high turbine inlet temperatures, the recovered power is high which increases the exergy efficiency.

Exergy destruction is an inevitable phenomenon during the energy conversion process. It is the function of the entropy generation and the ambient temperature. According to Figure 11, the high temperature difference between the source inlet and outlet during the super-heating process results in low exergy efficiency. The exhaust gas temperatures are low for the partial loads so as expected, the exergy efficiencies are high at partial loads. Higher temperature difference results in higher entropy generation which causes the higher exergy destruction, as well. In Figure 11, the exergy destruction in the components of the ORC WHR system at different engine loads are presented. The results are for the turbine inlet temperature and pressure values of 473.15 K and 6.86 MPa, respectively. According to this figure, the super-heater is the component which has the maximum exergy destruction as 256.79 kW at full load. High temperature differences at high engine loads cause high exergy destruction as well.

4.2. Environmental analysis

The ORC system recovers the waste heat energy to convert it into the mechanical power. Taking into account of main engine brake power at different engine loads, the recovered power ratio

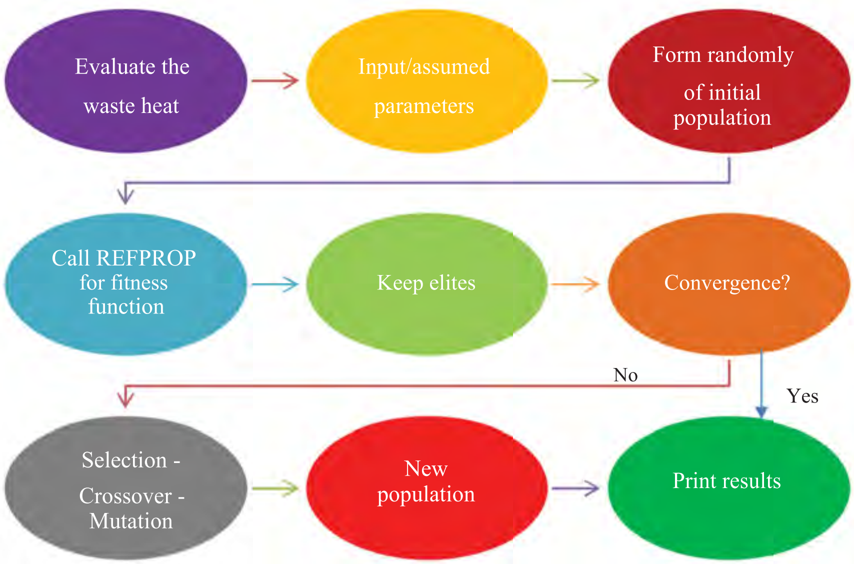

Figure 3. Flow chart of the simulation and optimisation process. (This figure is available in colour online.)

is presented in Figure 12. According to this figure, at about 50% MCR, the recovered power ratio is maximum as 3.82%. Although, the power change by the engine load is about linear, the change of the SFOC by load is curvilinear which effects the amount of waste heat transferred to the system. The recovered power ratio is minimum at about 75% MCR. However, the loads between 60% MCR and 75% MCR are more efficient and economical during operation.

The recovered power ratio also means fuel saving and emission reduction. Figure 13 shows the emission reduction with respect to the engine load at 498.15 K and 6.468 MPa turbine inlet conditions.

According to Figure 13, the fuel saving is high at heavy loads which means emission reduction is high as well. The fuel consumption of the main engine is 10,552 tonnes of heavy-fuel oil per 5796.2 operational hours for a year when the engine is operated at full load. This value at 85% MCR is 8809 tonnes/year. With the ORC system, the fuel saving in tonnes per year is calculated as 260.41 tonnes at the same operating case. According to Figure 13 for 85% MCR, up to 829.9 tonnes of CO_2, 1.94 tonnes of CO, 22.78 tonnes of NO_X, 14.14 tonnes of SO_2, 0.63 tonnes of VOCs and 1.99 tonnes of PM can be reduced per year. The emissions of the main engine without ORC system at 85% MCR is found to be 27,924 tonnes of CO_2, 65.19 tonnes of CO, 766.36 tonnes of NO_X, 475.67 tonnes of SO_2, 21.14 tonnes of VOCs and 66.95 tonnes of PM per year at full load.

4.3. The effects of SWH on the performance of ORC

As can be seen from Figure 2(a), the SWH uses the energy of the fluid after the expansion process. For the maximum turbine inlet pressure of 6.468 MPa and maximum temperature of 498.15 K, the calculated temperature after the expansion process is 401.12 K. To employ the SWH increases the thermal efficiency of the ORC WHRS by decreasing the cooling load of the condenser. It should be noted that the SWH is activated when the turbine outlet temperature is above 333.15 K. Figure 14 shows the change of thermal efficiency with respect to turbine inlet pressure for the ORC WHRS when the SWH is active and inactive.

According to Figure 14, in comparison with the condition that the SWH is inactive, it is possible to increase the thermal efficiency of the ORC WHRS about 20% at 498.15 K and 85%

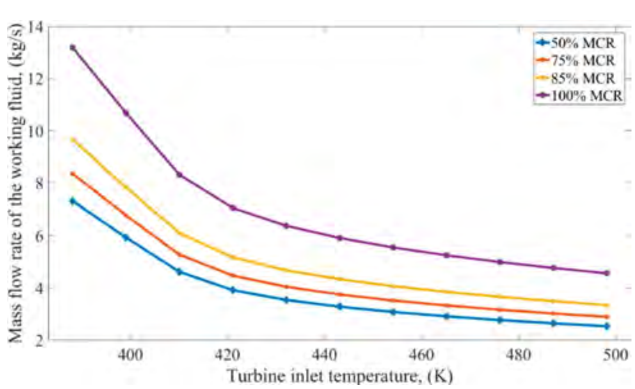

Figure 4. The change of mass flow rate of the working fluid with the turbine inlet temperature. (This figure is available in colour online.)

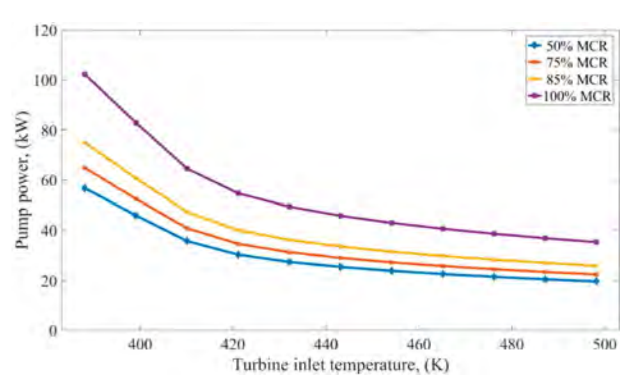

Figure 5. The change of pump power with the turbine inlet temperature. (This figure is available in colour online.)

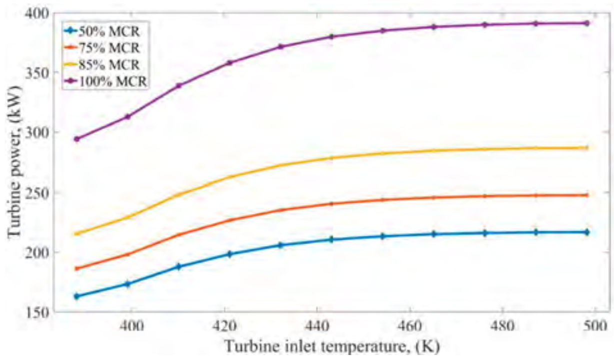

Figure 6. The change of turbine power with the turbine inlet temperature. (This figure is available in colour online.)

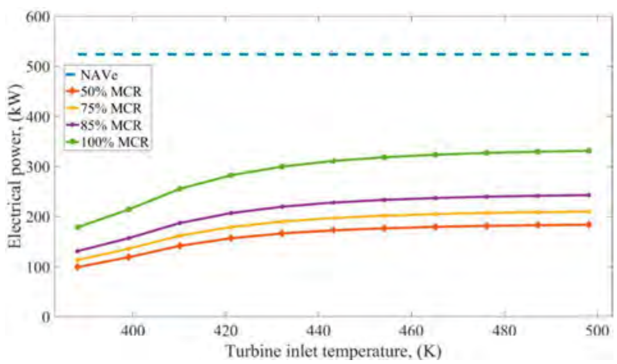

Figure 7. The change of electrical power output with the turbine inlet temperature. (This figure is available in colour online.)

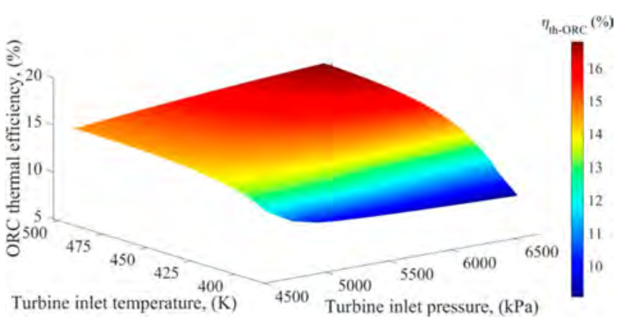

Figure 8. The change of thermal efficiency with turbine inlet temperature and pressure. (This figure is available in colour online.)

MCR. Moreover, the system generates 1.8 t/h hot water for domestic applications at the same conditions. On the other hand, the entropy generation in the condenser is decreased which means that the exergy efficiency of ORC WHRS can be increased approximately by 16%.

4.4. Comparisons of the analysed ORC with SRC

The analysed ORC WHRS works at high temperature and pressure conditions where the steam Rankine cycle also seems an option. The critical temperature and pressure of water are 647.1 K and 22,064 kPa (NIST 2010), respectively. Therefore, the operating conditions for the transcritical R152a correspond to the subcritical conditions for water. According to the results, at the same initial conditions, transcritical ORC WHRS has better performance comparing to the steam Rankine cycle as shown in Table 5.

According to Table 5, in comparison with ORC, SRC generates 17.7% less power and 39.7% less efficient in thermo-environmental point of view. The reasons for this lower performance are due to low quality of steam after the expansion process and high condensation pressure. Using the condensation temperature as 308.15 K for SRC, the corresponding condensation pressure is obtained as only 5.62 kPa which requires vacuuming process. In addition, the turbine inlet pressure range used in ORC, corresponds saturated steam at temperatures from 530 K to 554 K (NIST 2010) which means that SRC is more appropriate for the high quality waste heat recovery.

The costs of transcritical organic Rankine and steam Rankine cycles depend on many thermal parameters such as heat source temperature, pinch point temperature difference, thermodynamic properties of working fluids, operating temperature and pressure, which directly affect the component sizes and power capacities of the WHRS (Yang 2016; Andreasen et al. 2017; Noroozian et al. 2019). Studies in the literature show that ORC is also more cost-effective comparing to steam Rankine cycle in low or medium grade waste heat recovery applications. The vapour in steam cycle is generally above 500 °C (Garg et al. 2014) which is too high for most ORC working fluids of which maximum applicable temperatures are below 500 °C (NIST 2010). In comparison with ORC, steam Rankine cycle requires large turbines based on the low pressure

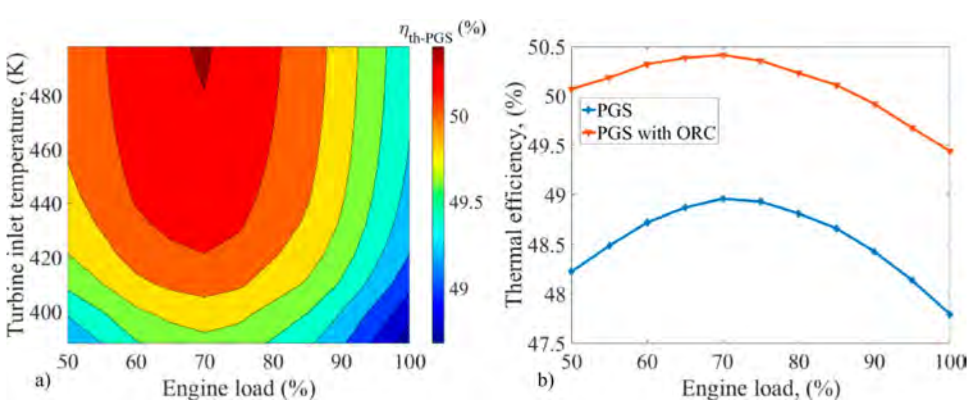

Figure 9. The change of thermal efficiency with engine load for the PGS (a–b). (This figure is available in colour online.)

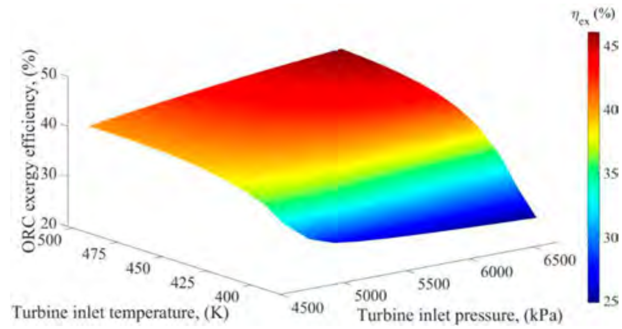

Figure 10. The change of exergy efficiency with turbine inlet temperature and pressure. (This figure is available in colour online.)

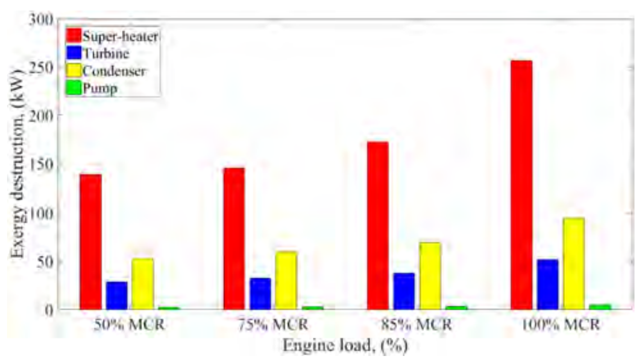

Figure 11. The exergy destruction rates of ORC components with respect to the engine load. (This figure is available in colour online.)

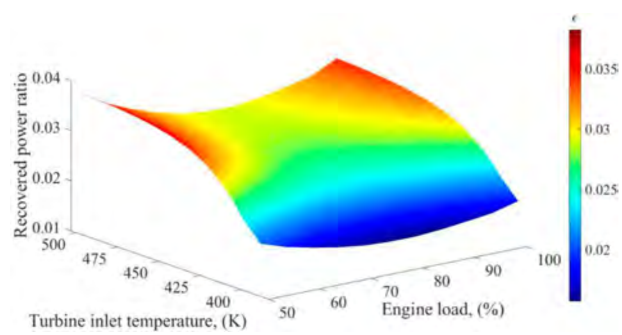

Figure 12. The change of recovered power ratio with turbine inlet temperature and engine load. (This figure is available in colour online.)

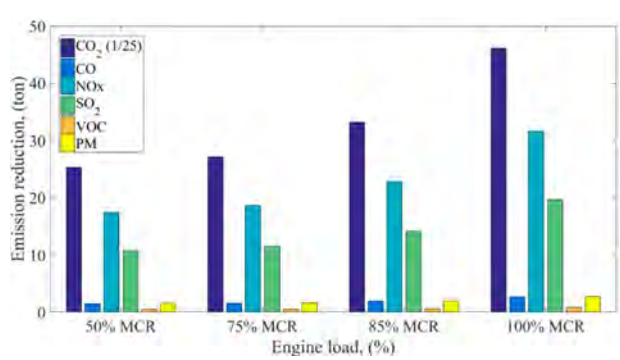

Figure 13. The emission reduction by ORC WHR system with respect to engine load. (This figure is available in colour online.)

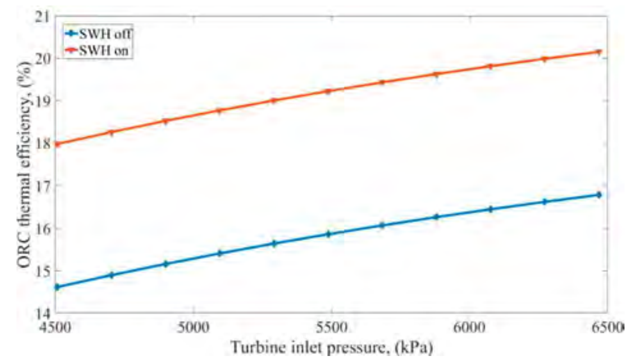

Figure 14. The change of thermal efficiency for the ORC WHRS based on SWH operation. (This figure is available in colour online.)

Table 5. Performance comparisons of the organic and steam Rankine cycles.

Cycle	Engine Load (%)	P_1 (kPa)	T_3 (K)	$\dot{W}_{net}$ (kW)	η_{th-ORC} (%)	η_{ex-ORC} (%)	ε (%)
SRC	85	793.85	498.15	136.88	8.81	24.2	1.56
ORC	85	793.85	498.15	226.99	14.62	40.1	2.58

operation and low fluid density which increase the system size, additional water treatment equipment to prevent the water erosion, which make the SRC system economically and technically not feasible for the low grade waste heat recovery applications onboard ships (Andreasen et al. 2017; Saadon and Islam 2019). Moreover, it was reported that the maintenance cost of ORC systems is cheaper than that of SRC (Vanslambrouck et al. 2012) which also promotes the usage ORC for maritime applications.

4.5. Thermo-environmental optimisation

Using the upper and lower boundaries of chromosome genes, the optimisation of exergy efficiency is performed. The results are presented in Table 6. As expected, the optimal operational region of the integrated ORC WHR system is between 60% MCR and 75% MCR.

Table 6. Optimal performance parameters of the ORC WHR system.

Parameter	Value	Parameter	Value
P_1	793.85 kPa	$\dot{W}_t$	247.86 kW
P_2	6600 kPa	$\dot{W}_{net}$	225.48 kW
T_2	312.11 K	$\dot{W}_e$	209.92 kW
P_3	6468 kPa	η_{ex}	46.39 %
T_3	498.15 K	$\dot{X}_{des,sh}$	145.71 kW
P_4	793.85 kPa	$\dot{X}_{des,t}$	32.99 kW
T_4	401.13 K	$\dot{X}_{des,c}$	60.27 kW
s_1	0.12114 kj/kg − K	$\dot{X}_{des,p}$	3.21 kW
s_2	0.12152 kj/kg − K	ε	0.0291
s_3	0.23800 kj/kg − K	R_{CO_2}	678.1 tonnes
s_4	0.24183 kj/kg − K	R_{CO}	1.58 tonnes
$\dot{m}_f$	2.887 kg/s	R_{NO_x}	12.7 tonnes
η_{th-ORC}	16.79 %	R_{SO_2}	18.61 tonnes
$\eta_{th_{PGS}}$	48.93 %	R_{VOC}	0.51 tonnes
$\eta_{th_{PGS+ORC}}$	50.17 %	R_{PM}	1.63 tonnes
T_m	467.28 K	$\dot{m}_{fw}$	2.95 kg/s
$T_{exh,in}$	503.15 K	$T_{fw,in}$	298.15 K
MCR	70%	$T_{fw,out}$	323.15 K
$\dot{W}_P$	22.38 kW		

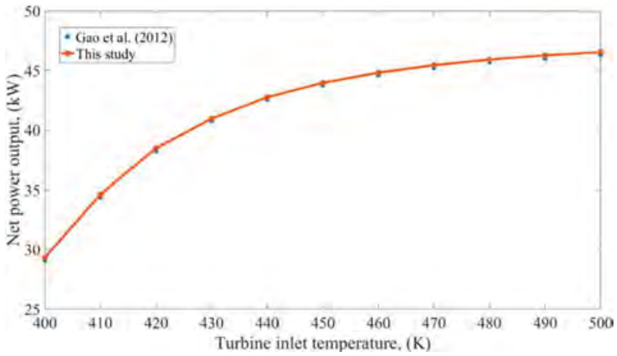

Figure 15. Comparisons of the net power output with the previous study. (This figure is available in colour online.)

4.6. Validation study

The net power output calculated for the transcritical organic Rankine cycle and the net power output values presented in the study of Gao et al. (2012) are compared in Figure 15 for the validation of the results. During the validation study, the turbine inlet pressure is 6.6 MPa. Turbine isentropic efficiency and pump isentropic efficiency are taken as 0.85 and 0.7 respectively. The source inlet and outlet temperatures are 533 and 333 K, respectively. The mass flow rate of the reference system is 1 kg/s. The study shows that the results are consistent with the results of Gao et al. (2012) and the differences are found to be less than 1%.

5. Conclusions

The energetic, exergetic and environmental analysis of a transcritical ORC exhaust gas waste heat recovery system is investigated at different engine operating conditions and the performance parameters of the system are optimised with using the genetic algorithm. R152a is used as the working fluid. According to the results, the following conclusions are obtained:

- The amount of the exhaust gas waste heat and its energy quality due to its high temperature are very high. Therefore, the ORC WHR system can work in the transcritical region with an appropriate working fluid.
- The optimisation study shows that the power generation system integrated with ORC WHRS should be operated between about 70% MCR and 75% MCR of the main engine to maximise its exergy efficiency and to minimise its fuel oil consumption.
- The ORC WHR system can fulfil about 43% of the navigation electrical load when the system is operated at the optimal working conditions. On the other hand, at full load, up to 63.2% of the navigation electrical load can be supplied from the ORC WHRS.
- It is possible to increase the power generation system efficiency up to 2.53% when the ORC system is operated at optimal working conditions and more than 2.9% of mechanical power can be recovered.
- The ORC WHR system provides the fuel saving and remarkably decreases the emissions. It is possible to save 225.18 tonnes of fuel from 7210 tonnes of fuel consumed per year at optimal working conditions.
- The ORC WHR system at optimal working conditions can reduce total 678.1 tonnes of CO_2 emissions out of 24,504 tonnes of CO_2 emitted per year.

Disclosure statement

No potential conflict of interest was reported by the author(s).

ORCID

Mehmet Akman http://orcid.org/0000-0002-6274-2742
Selma Ergin http://orcid.org/0000-0001-8343-2455

References

Akman M, Ergin S. 2016. Thermodynamic analysis of organic Rankine cycle for waste heat recovery system of a ship. GMO SHIPMAR, Proceedings of 1st International Congress on Ship and Marine Technology, Istanbul. p. 1–10.

Akman M, Ergin S. 2019. An investigation of marine waste heat recovery system based on organic Rankine cycle under various engine operating conditions. Proc I Mech E Part M J Eng Mari Environ. 233(2):586–601.

Andreasen JG, Meroni A, Haglind F. 2017. A comparison of organic and steam Rankine cycle power systems for waste heat recovery on large ships. Energies. 10:547.

Braimakis K, Karellas S. 2018. Energetic optimization of regenerative organic Rankine cycle (ORC) configurations. Energy Convers Manage. 159:353–370.

Brueckner S, Arbter R, Pehnt M, Laevemann E. 2017. Industrial waste heat potential in Germany – a bottom-up analysis. Energ Effic. 10:513–525.

Dai Y, Wang J, Gao L. 2009. Parametric optimization and comparative study of organic Rankine cycle (ORC) for low grade waste heat recovery. Energy Convers Manage. 50:576–582.

Gao H, Liu C, He C, Xu X, Wu S, Li Y. 2012. Performance analysis and working fluid selection of a supercritical organic Rankine cycle for low grade waste heat recovery. Energies. 5:3233–3247.

Garg P, Srinivasan K, Dutta P, Kumar P. 2014. Comparison of CO_2 and steam in transcritical Rankine cycles for concentrated solar power. Energy Procedia. 49:1138–1146.

Grljušić M, Medica V, Radica G. 2015. Calculation of efficiencies of a ship power plant operating with waste heat recovery through combined heat and power production. Energies. 8:4273–4299.

Hou S, Cao S, Yu L, Zhou Y, Wu Y, Zhang F. 2018. Performance optimization of combined transcritical CO_2 recompression cycle and regenerative organic Rankine cycle using zeotropic mixture fluid. Energy Convers Manage. 160:187–200.

International Convention for the Prevention of Pollution from Ships. 2015.

International Maritime Organization. 2009. MEPC 59/Inf. 10.

International Maritime Organization, Third Greenhouse Gas Study. 2014. [accessed 2019 Feb 15]. http://www.imo.org/en/OurWork/Environment/PollutionPrevention/AirPollution/Documents/Third%20Greenhouse%20Gas%20Study/GHG3%20Executive%20Summary%20and%20Report.pdf.

Larsen U, Pierobon L, Haglind F, Gabrielli C. 2013. Design and optimization of organic Rankine cycles for waste heat recovery in marine applications using the principles of natural selection. Energy. 55:803–812.

MAN Diesel & Turbo. 2014. Thermo efficiency system for reduction of fuel consumption and CO_2 emission.

MAN Diesel & Turbo. 2019. CEAS engine calculations [accessed 2019 Oct 22]. http://marine.man.eu/two-stroke/ceas.

Miró L, McKenna R, Jäger T, Cabeza LF. 2017. Estimating the industrial waste heat recovery potential based on CO_2 emissions in the European non-metallic mineral industry. Energ Effic. 11:427–443.

Mito M, Teamah M, El-Maghlany WM, Shehata AI. 2018. Utilizing the scavenge air cooling in improving the performance of marine diesel engine waste heat recovery systems. Energy. 142:264–276.
Mondejar ME, Andreasen JG, Pierobon L, Larsen U, Thern M, Haglind F. 2018. A review of the use of organic Rankine cycle power systems for maritime applications. Renew Sustain Energy Rev. 91:126–151.
Mossafa AH, Mokarram NH, Farshi LG. 2016. Thermo-economic analysis of combined different ORCs geothermal power plants and LNG cold energy. Geothermics. 65:113–125.
Nawi ZM, Kamarudin SK, Abdullah SR, Lam SS. 2018. The potential of exhaust waste heat recovery (WHR) from marine diesel engines via organic Rankine cycle. Energy. 166:17–31.
Nazari N, Heidarnejad P, Porkhial S. 2016. Multi-objective optimization of a combined steam-organic Rankine cycle based on exergy and exergo-economic analysis for waste heat recovery application. Energy Convers Manage. 127:366–379.
NIST Standard Reference Database. 2010. Reference fluid thermodynamic and transport properties REFPROP, version 9.0.
Noroozian A, Naeimi A, Bidi M, Ahmadi M. 2019. Exergoeconomic comparison and optimization of organic Rankine cycle, trilateral Rankine cycle and transcritical carbon dioxide cycle for heat recovery of low-temperature geothermal water. Proc Inst Mech Eng A J Power Energy. 233(8):1068–1084.
Olmer N, Comer B, Roy B, Mao X, Rutherford D. 2017. Greenhouse gas emissions from global shipping, 2013–2015. ICCT.
Riyanto H, Martowibowo S, Maksum H. 2014. Application of genetic algorithm optimization for organic Rankine cycle waste heat recovery power generation. Technical report.
Saadon S, Islam SMS. 2019. A recent review in performance of organic Rankine cycle (ORC).
Singh DV, Pedersen E. 2016. A review of waste heat recovery technologies for maritime applications. Energy Convers Manage. 111:315–328.
Soltani R, Dincer I, Rosen MA. 2015. Thermodynamic analysis of a novel multigeneration energy system based on heat recovery from a biomass CHP cycle. Appl Therm Eng. 89:90–100.
Song J, Li X, Ren X, Gu C. 2018. Performance analysis and parametric optimization of transcritical carbon dioxide (S-CO_2) cycle with bottoming organic Rankine cycle (ORC). Energy. 143:406–4016.
Song J, Song Y, Gu C. 2015. Thermodynamic analysis and performance optimization of an organic Rankine cycle (ORC) waste heat recovery system for marine diesel engines. Energy. 82:976–985.
Suarez S, Greig A. 2013. Making shipping greener: ORC modelling under realistic operative conditions. Low Carbon Shipping Conference, London.
Trozzi C, Vaccaro R. 1998. Methodologies for estimating air pollutant emissions from ships. Techno report methodologies for estimating air pollutant emissions from transport, RF98.
Vanslambrouck B, Broek MV, Gusev S, Paepe M. 2012. Efficiency comparison between the steam cycle and the organic Rankine cycle for small scale power generation. Renewable Energy World Conference & Expo North America.
Wang EH, Zhang HG, Fan BY, Ouyang MG, Zhao Y, Mu QH. 2011. Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery. Energy. 36:3406–3418.
Xi H, Li M, Xu C, He Y. 2013. Parametric optimization of regenerative organic Rankine cycle for low grade waste heat recovery using genetic algorithm. Energy. 58:473–482.
Yağlı H, Koç Y, Koç A, Görgülü A, Tandiroğlu A. 2016. Parametric optimization and exergetic analysis comparison of subcritical and supercritical organic Rankine cycle (ORC) for biogas fuelled combined heat and power (CHP) engine exhaust gas waste heat. Energy. 111:923–932.
Yang M. 2016. Optimizations of the waste heat recovery system for a large marine diesel engine based on transcritical Rankine cycle. Energy. 113:1109–1124.
Yang M, Yeh R. 2015. Thermo-economic optimization of an organic Rankine cycle system for large marine diesel engine waste heat recovery. Energy. 82:256–268.
Zhu Y, Li W, Sun G, Li H. 2018. Thermo-economic analysis based on objective functions of an organic Rankine cycle for waste heat recovery from marine diesel engine. Energy. 158:343–356.