Abstract
Shipping is a growing sector in the global economy and its contributions to global greenhouse gas (GHG) emissions are expected to increase. Energy efficiency has always been an important factor to minimize GHG emissions. Waste heat recovery (WHR) systems can be used for both improving energy efficiency and reducing energy consumption and GHG emissions. Organic Rankine Cycles (ORC) systems are the most widely used technology for low temperature heat recovery from exhaust gases in the industrial applications. But this system is not widely used in ship applications. In this study thermodynamic performance criteria of ORC system for a container ship are developed and thermodynamic, environmental and economic effects of the system are analyzed. It can be achieved additionally 2% reduction for the ship fuel consumption by utilizing the ORC system.

Keywords
Emissions; Waste Heat Recovery; Marine Power Plant; Organic Rankine Cycle; Energy Efficiency; Container Ship

Introduction
The marine transportation sector is one of the major causes of global air pollution. Emissions from ships affect global air quality, people’s health, the marine ecology, and global warming. Carbon dioxide (CO₂), carbon monoxide (CO), particulate matter (PM), nitrogen oxides (NOₓ), and sulfur oxides (SOₓ) are the most significant pollutants emitted from marine diesel engines. Approximately 14–31%, 4–9% of the global emissions of NOₓ, SOₓ, respectively, are from marine vessels [1, 3]. According to the IMO, maritime transport emitted 1046 million tons of CO₂ in 2007, representing 3.3% of the world’s total emissions [3]. These emissions are assumed to increase by 150-250% in 2050 if no action is taken.

Annex VI of the MARPOL convention, adopted by IMO in 1997, sets limits on NOₓ and SOₓ emissions from ship exhaust gases for the prevention of air pollution from ships. The IMO has also specified Energy Efficiency Design Index (EEDI) and Ship Energy Efficiency Management Plan (SEEMP) that will reduce greenhouse gas emission. With the implementation of EEDI, ship operators should take a series of technical measures in order to reduce their CO₂ emissions. The proposed or feasible technical measures on CO₂ reduction is mainly aimed to improve energy efficiency of ships.

There are various technological improvements on ship building and propulsion systems with regard to maximization of a ship’s energy efficiency. A wide range of options for increasing the energy efficiency by changing ship design, retrofit new equipment and improving ship operation has been identified. The consumption of fuel is one of the most significant factors for CO₂ emissions of a ship. The amount of CO₂ emitted from a ship is directly related to the amount of fuel consumed by the internal combustion engine propelling the ship.

Ship energy efficiency can be improved by utilizing WHR systems. WHR systems allow ships to save energy and reduce CO₂ emissions and have the capability to improve the EEDI index. Low-grade heat energy on ships has an important potential for energy efficiency. Since conventional steam power system cannot give a better performance to recover low-grade waste heat, ORC systems can be used to recover that heat. An ORC system is the most effective technology for usage of low-grade heat sources. This technology includes a boiler, a work-producing
expansion device, a condenser and a pump like as the components of steam power plant. There are several advantages in using an ORC to recover low-grade waste heat, including economical utilization of energy resources, smaller systems and reduced emissions of CO, CO₂, NOₓ and other atmospheric pollutants [4].

Researches have started on the recovery of low-grade waste heat since the early 1980s. Boretz, J.E. evaluated ORC to produce power from the low-grade waste heat but it was not used because of some economical and safety problems in the operation of the system [5]. There have been several industrial applications like as geothermal energy, solar radiation, and waste heat from processes, nuclear power plant or conventional power plants [6-7]. HUNG, T.C. et al., have analyzed and compared the efficiencies of ORCs by using cryogens such as benzene, ammonia, R11, R12, R134A and R113 as working fluids [8-10]. LIU, B.T. et al., presented analysis of the performance ORC subjected to the influence of working fluids and have investigated the effects of various working fluids on the thermal efficiency and on the total heat recovery efficiency [11]. WEI, D. et al., have analyzed the thermodynamic performances of an ORC system under disturbances [12]. WOREK, M.W. et al., optimized the design criteria for an ORC using low-temperature geothermal heat sources [13]. They found that the performance of four working fluids that are suitable for low-temperature geothermal power cycle are investigated using an optimization criteria, the ratio of heat transfer area to net power produced, which is a good measure of total power plant cost [13]. Beyene, A. and Husband, W.W. have studied the feasibility on the low-grade heat driven Rankine cycle. They used a low toxicity, low flammability, and ozone- neutral working fluid [14].

Mago, P.J. and Chamra, L.M. have performed an exergy analysis of a combined engine-ORC configuration. They illustrated that an ORC combined with an engine not only improves the engine thermal efficiency but also increases the exergy efficiency [15]. Some parametric optimization performance analysis of a waste heat recovery system using ORC has been studied in the literature [16-18]. Chacartegui, R. et al., studied low temperature ORC as bottoming cycle in medium and large scale combined cycle power plants [19]. They figured out the interest of using this alternative cycle with high efficiency heavy duty gas turbines, for example recuperative gas turbines with lower gas turbine exhaust temperature than in conventional combined cycle gas turbines [19]. Goswami D.Y. et al. reviewed of thermodynamic cycle and working fluids for the conversion of low-grade heat [20]. Wang, T. et al., reviewed researches on thermal exhaust heat recovery with Rankin cycle [21]. Krishnan, S.R. et al., examined the exhaust waste heat recovery potential of a high efficiency, low-emissions dual fuel low temperature combustion engine using and ORC [22]. Smolen. S. analyzed and simulated of a two-stage ORC for utilization of waste heat at medium and low temperature levels [23].

Waste heat recovery with ORC systems is the most widely used in the industrial applications. But this system is not widely used in ship applications. In this study the thermodynamic performance criteria of an ORC system is analyzed. An integrated ORC model for a ship power plant equipped a WHR system according to this criterion is studied. This study also includes the influences of the working fluids selection for ships.

**Thermodynamic Performance Criteria of WHR Systems for Ship Power Plants**

Waste heat is generated in a process by way of fuel combustion and then emit into the environment even though it could still be reused for some energy [24]. The strategy to recover this heat is related to the energy levels of the waste heat gases. Effective waste recovery depends on the temperature of the waste heat. It might be used three types of heat recovery classes like high temperature, medium temperature and low temperature in industrial applications. Recovery of waste heat has direct and indirect effects on the processes. Generally it affects thermal efficiency by decreasing the utility consumption, decreasing running costs, reducing air pollution, increasing the load capacity of the vessel and reducing in auxiliary energy consumption.

Marine diesel engines convert about 45–50% of the energy of fuel into mechanical work for the ship’s propulsion system and the remaining part is lost in the form of waste heat through the exhaust which can contain about 25% of the input energy [25]. The WHR system could recover part of the heat exhausted and turn it into additional power for onboard services and to supplement the propeller. Two types of WHR system applications are used on ships. One of the systems is basic waste heat recovery system that contains exhaust gas boiler for heating steam production (Fig.1a). The other WHR system contains an economizer, an evaporator and a super heater that produces a high pressure steam for a steam turbine that generates electricity (Fig.1b). It is not possible to use waste
heat energy under 160°C in those systems because low temperature corrosion occurs below this temperature level. Therefore it can recover more heat energy from the system.

FIG.1 WHR SYSTEMS FOR SHIPS (a) WASTE HEAT RECOVERY FOR HEATING STEAM PRODUCTION, (b) WASTE HEAT RECOVERY FOR POWER AND HEATING STEAM PRODUCTION AND (c) COMBINED WASTE HEAT RECOVERY USING AN ORGANIC RANKINE CYCLE

This energy is recovered by utilizing ORC system. Figure 1c indicates an ORC system for low temperature exhaust gases. In this study thermodynamic performance criteria of an integrated ORC system for a container ship are analyzed (Fig.1c). In that case the total power production can be given as following equations:

\[ W_T = W_{ME} + W_{ST} + W_{ORC} \]  (1)

Where, \( W_{ME}, W_{ST} \) and \( W_{ORC} \) are the power produced by the main engine (M/E), the steam turbine (ST) and the ORC turbine (T), respectively. The main engine power can be defined as:

\[ W_{ME} = F \eta_{ME} \]  (2)

Where, \( F \) is the energy of fuel to supply to the main engine and \( \eta_{ME} \) is the efficiency of the main engine. The fuel energy is given by:

\[ F = m_{fuel} LHV \]  (3)

Where, \( m_{fuel} \) and LHV are the mass flow rate and the low heat value of the fuel, respectively. The power of the steam turbine (ST) and the organic ranking cycle turbine (T) can be defined as:

\[ W_{ST} = m_s (h_{in,ST} - h_{out,ST}) \]  (4)

\[ W_{ORC} = m_{orf} (h_{in,T} - h_{out,T}) \]  (5)

Where, \( m_s, m_{orf}, h_{in,T}, h_{out,T}, h_{in,ST} \) and \( h_{out,ST} \) are mass the flow rate of steam, the mass flow rate of organic working fluid, the enthalpies of input and output in the ORC turbine and the steam turbine, respectively. Also, \( \dot{Q}_u \) is the utilized heat in the waste heat recovery system of a power plant can be detailed as follows:

\[ \dot{Q}_u = \dot{Q}_{ST} + \dot{Q}_{ORC} \]  (6)

Where, \( \dot{Q}_{ST} \) and \( \dot{Q}_{ORC} \) are the heat utilization from the steam turbine and the ORC turbine, respectively. Also \( \dot{Q}_u \) is considered as a different parameter supplied from the fuel heat energy because it should be defined as a different heat load. There are performed as following equations:

\[ \dot{Q}_{ST} = m_{air} c_p \left( T_{in} - T_{out} \right) \]  (7)
\[
\dot{Q}_{\text{ORC}} = \dot{m}_{\text{exh}} c_{p_{\text{exh}}} (T_{\text{in},1} - T_{\text{out},2})
\]  

(8)

Where, \(\dot{m}_{\text{exh}}\) and \(c_{p_{\text{exh}}}\) are the mass flow rate of exhaust gases and specific heat of exhaust gases, respectively. \(T_{\text{in}}\) is the inlet temperature of exhaust gas boiler, \(T_{\text{out},1}\) and \(T_{\text{out},2}\) are the outlet temperature from an economizer 1 and economizer 2 (see Fig.1c). On the other hand, the useful heat \(\dot{Q}_u\), rejected from the power plant in inlet temperature \(T_{\text{in}}\) and outlet temperature \(T_{\text{out},2}\) which is higher than the environmental temperature \(T_0\). By using Eqs. (7), (8) in Eq. (6), it becomes

\[
\dot{Q}_u = \dot{m}_{\text{exh}} c_{p_{\text{exh}}} (T_{\text{in}} - T_{\text{out},2})
\]  

(9)

The specific heat of the exhaust \((c_{p_{\text{exh}}})\) gas is calculated as a function of the inlet temperature by the following equation [26]:

\[
c_{p_{\text{exh}}} = (0.991615) + \left(6.99703 \times 10^{-1}\right)T_{\text{in}} + \left(2.712987 \times 10^{-1}\right)T_{\text{in}} - \left(1.22442 \times 10^{-6}\right)T_{\text{in}}^2
\]  

(10)

**Artificial Thermal Efficiency**

An alternative criterion of performance is the artificial thermal efficiency \((\eta_a)\). The artificial thermal efficiency is given by [26]:

\[
\eta_a = \frac{W_f}{F - \frac{\dot{Q}_u}{(\eta_b)_H}}
\]  

(11)

Where, \((\eta_b)_H\) is the exhaust gas boiler efficiency.

**Fuel Energy Saving Ratio (FESR)**

Fuel Energy Saving Ratio (FESR) is another useful criterion. FESR is defined as the ratio of the savings to the fuel energy required in power plants. FESR directly measures the extent of fuel savings and can be formulated as [26]:

\[
FESR = \frac{\Delta F}{(\eta_b)_H + (\eta_O)_C}
\]  

(12)

Where, \(\Delta F\) is the saved fuel energy, \((\eta_b)_C\) is overall efficiency of the system and \((\eta_b)_H\) is exhaust gas boiler efficiency. \(\Delta F\) can be defined as:

\[
\Delta F = \left(\frac{\dot{Q}_u}{(\eta_b)_H}\right) + \left(\frac{W_f}{(\eta_O)_C}\right) - F.
\]  

(13)

The first term at the right side of the Eq.13 can be evaluated as follows:

\[
\frac{\dot{Q}_u}{(\eta_b)_H} = \left(\frac{\dot{Q}_{ST}}{\eta_{b,1}} + \frac{\dot{Q}_{\text{ORC}}}{\eta_{b,2}}\right)
\]  

(14)

Where, \(\eta_{b,1}\) and \(\eta_{b,2}\) are effectiveness parameters of the exhaust gas boiler heat exchangers and the ORC heat exchangers, respectively. It is described in Eqs.15 and 16:

\[
\eta_{b,1} = \varepsilon_{\text{exh},1}
\]  

(15)

\[
\eta_{b,2} = \varepsilon_{\text{exh},2}
\]  

(16)

In this study it is assumed that the effectiveness of, \(\varepsilon_{\text{exh},1}=0.90\) and \(\varepsilon_{\text{exh},2}=0.80\). The second term at the right side of the Eq.13 can be evaluated as follows:
\[
\frac{\dot{W}_T}{(\eta_o)_{c}} = \left( \frac{\dot{W}_{ME} + \dot{W}_{ST} + \dot{W}_{ORC}}{\eta_{ME} \eta_{ST} \eta_{ORC}} \right)_{c}
\]

Where, \( \eta_{ST} \) and \( \eta_{ORC} \) are the efficiency of the steam turbine and the ORC turbine, respectively. In this study it is assumed that the main engine efficiency is \( \eta_{ME} = 0.50 \), the steam turbine efficiency is \( \eta_{ST} = 0.44 \) and the ORC turbine efficiency is \( \eta_{ORC} = 0.52 \).

**Energy Utilization Factor (EUF)**

Energy Utilization Factor (EUF) is an indication of overall efficiency of a cogenerated ship power plant with an integrated ORC system. It is defined as the ratio of useful energy to the heat supplied by the fuel. EUF can be formulated as [26]:

\[
EUF = \left( \frac{\dot{W}_T + \dot{Q}}{\dot{F}} \right)
\]

**Cogeneration Efficiency**

The cogeneration efficiency is a performance criterion. The cogeneration efficiency considers the quality differences between work and heat. Therefore it is conceivable performance criterion for evaluating the energy efficiency. It can be formulated as [26]:

\[
\eta_{CG} = \left( \frac{\dot{W}_T + \dot{E}_H + \phi(\dot{Q}_{U} - \dot{E}_H)}{\dot{F}} \right)
\]

Where, \( \phi \) is a constant whose value changes between 0<\( \phi <1 \). It is assumed that \( \phi \) is equal to 0.27 in this study. \( \dot{E}_H \) is the exergy rate of the heat which is released from the exhaust gas boiler outlet temperature (\( T_{out,2} \)) to the atmospheric temperature (\( T_0 \)). The heat exergy rate of the system can be calculated as the following equation:

\[
\dot{E}_H = \dot{Q}_U \left( 1 - \frac{T_{out,2}}{T_0} \right)
\]

**Reducing Exhaust Gas Emission and Air Pollutant**

Exhaust gas emissions reduce with WHR systems, by increasing the efficiency of the overall system. The exhaust gas emission saving (\( \Delta\dot{m}_{e,ch} \)) can be calculated as follows:

\[
\Delta\dot{m}_{e,ch} = \left( \frac{\dot{m}_{e,ch}}{\dot{m}_{fuel}} \right) \cdot \Delta\dot{m}_{fuel}
\]

Where, \( \Delta\dot{m}_{fuel} \) is the mass rate of fuel saving in the power plant, \( \dot{m}_{fuel} \) and \( \dot{m}_{e,ch} \) are the mass rates of fuel consumption and exhaust gases that are inlet parameters in the exhaust gas boiler. The amount of air pollutants is reduced by reducing fuel consumption. The amount of air pollutants can be calculated according to the following equation [27]:

\[
P_i = \Delta\dot{m}_{fuel} \cdot EF_i
\]

Where, \( P \) is the pollutant, \( i \) is the pollutant type (CO\(_2\), CO, NO\(_x\), SO\(_x\), VOC and PM), EF is the emission factors of the pollutant type.

**Working Fluids**

Because the working fluid is an important part of an ORC system, fluid selection is one of the most important contributors to overall cycle performance. Some important criteria for the determination of fluids should be taken into consideration for use in ORC systems. The selected fluid should not only exhibit favorable physical, chemical, environmental, safety and economic properties such as low specific volume, viscosity, toxicity, flammability, ozone depletion potential (ODP), global warming potential (GWP), atmospheric lifetime (ALT) and cost, but also contribute to favorable process attributes such as high efficiency or moderate pressure in heat exchangers [28].
GWP is an index that determines the potential contribution of a chemical substance to global warming. This value is used to compare the abilities of different greenhouse gases to trap heat in the atmosphere. It compares the amount of heat trapped by a certain mass of the gas in question to the amount of heat trapped by a similar mass of carbon dioxide. GWP value is expected to be low in terms of environmental criteria. The ODP is a relative value that indicates the potential of a substance to destroy ozone gas as compared with the potential of chlorofluorocarbon-11 (CFC-11) which is assigned a reference value of 1 [29].

The consideration of safety requirement of organic working fluid mainly includes toxicity, flammability, and chemical stability. For the identification of the safety level of a working fluid, the ASHRAE 34 safety classification can be used. Toxicity and flammability are the two key parameters used by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) to indicate the safety level of an organic working fluid [30]. Table 1 indicates safety classification matrix for working fluids.

<table>
<thead>
<tr>
<th>Table 1: Safety Classification of Working Fluids</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flammability</td>
</tr>
<tr>
<td>High</td>
</tr>
<tr>
<td>Low</td>
</tr>
<tr>
<td>Non-flammable</td>
</tr>
</tbody>
</table>

According to ASHREA standards the safety class code A indicates a “Lower Toxicity” specification and the safety class code B indicates a “Higher Toxicity” specification for organic fluids. As a similar, a class number code of 1 shows a “Non-flammability” specification, a class number of 2 shows a “Lower-flammability” specifications and a class number of 3 shows a “Higher-flammability” specification for organic fluids. Several working fluids which are the most commonly used as fluids for ORC systems in the literature are shown in Table 2 [20, 30].

<table>
<thead>
<tr>
<th>Table 2: Specifications of Working Fluids</th>
</tr>
</thead>
<tbody>
<tr>
<td>------------------</td>
</tr>
<tr>
<td>R600</td>
</tr>
<tr>
<td>R601a</td>
</tr>
<tr>
<td>R600a</td>
</tr>
<tr>
<td>R601</td>
</tr>
<tr>
<td>R290</td>
</tr>
<tr>
<td>R11</td>
</tr>
<tr>
<td>R114</td>
</tr>
<tr>
<td>R134a</td>
</tr>
<tr>
<td>R245fa</td>
</tr>
<tr>
<td>R123</td>
</tr>
</tbody>
</table>

An important criterion in the selection of liquid for ship ORC systems is that they are safe, non-toxic and harmless for the environment. The best result for environment is provided A3 class working fluids. However these fluids have high-flammability feature. The best choosing for non-flammability features are B1 Class fluids. But these fluids have a toxic effect that is the most serious problems in terms of health. Ensuring safety in maritime applications is the most important issue to be considered. Since A1 safety class fluids have non-flammability and non-toxicity features, they are acceptable for ORC system applications of ships.

**Modeling of an ORC System for the Ship Power Plant**

The merchant fleet broadly consists of bulk carriers, tankers, and container ships. Those ships are significant individual contributors to air pollution. Because container ships operate at relatively higher speeds than bulkers and tankers, they need more propulsion power. Therefore fuel oil consumption of the container ship is more excessive than other ships. A container ship equipped a WHR system was selected for this study. The specifications of the container ship and the main engine are shown in Table 3.
TABLE 3 THE SPECIFICATIONS OF THE CONTAINER SHIP, THE MAIN ENGINE AND THE BOILER SYSTEM

<table>
<thead>
<tr>
<th>Specification of the ship</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Length overall</td>
<td>295</td>
<td>m</td>
</tr>
<tr>
<td>T.E.U.</td>
<td>4 200</td>
<td>-</td>
</tr>
<tr>
<td>Breadth</td>
<td>32</td>
<td>m</td>
</tr>
<tr>
<td>Draught</td>
<td>12.6</td>
<td>m</td>
</tr>
<tr>
<td>Dead-weight</td>
<td>55 000</td>
<td>tons</td>
</tr>
<tr>
<td>Service speed</td>
<td>25</td>
<td>knots</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Main engine data parameters of the ship</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type of engine</td>
</tr>
<tr>
<td>Cylinder bore</td>
</tr>
<tr>
<td>Piston stroke</td>
</tr>
<tr>
<td>Number of cylinders</td>
</tr>
<tr>
<td>Engine speed</td>
</tr>
<tr>
<td>Main engine power</td>
</tr>
<tr>
<td>SFOC</td>
</tr>
<tr>
<td>FOC</td>
</tr>
</tbody>
</table>

Input data parameters in the boiler system of the ship

| Mass rate of exhaust gases | 66.67 kg/s |
| Inlet temperature          | 650 K      |
| Outlet temperature         | 436 K      |
| Power of steam turbine     | 1600 kW    |

The main engine’s loading conditions changes from 25% to 100% of the engine’s maximum continuous rating (MCR). In real operational conditions of a ship the basic parameter determining the engine load level is the sailing speed. Within the load range, the engine produces exhaust gases of such parameters (the amount and temperature), that a sufficient amount of heating energy for safe and effective ship operation is provided. Any changes of the engine working parameters caused by exhaust gas changes shall affect WHR and ORC system working parameters. In this study it is assumed that the main engine loading condition is 85% that is standard optimization point.

![FIG. 2 WASTE HEAT RECOVERY SYSTEM (a) FOR POWER AND HEATING STEAM PRODUCTION WITH BASIC SYSTEM, (b) FOR ADDITIONAL POWER PRODUCTION WITH AN ORC ON THE CONTAINER SHIP POWER PLANT](image_url)
The ship has a WHR system producing steam for power and heating. Figure 2a shows the WHR system of the ship. The exhaust gas boiler is the main part of the WHR system. The boiler consists of the economizer, the evaporator, and the super heater. The WHR system extracts heat energy from the exhaust gas by heating, evaporating and superheating water in heat exchangers in the stack. The feed water is pumped by the feed water pump into the water/steam drum. The heating medium is the saturated water contained in the drum, which is pumped by the economizer circulating water pump. The exhaust gas boiler outlet temperature is not less than 160°C to avoid sulfur corrosion in the economizer outlet. The saturated steam is advanced into the super heater section of the boiler. The superheated steam enters into the steam turbine stages of turbo generator, where it expands producing mechanical power and driving the electric generator. The condensed steam is then pumped into the feed water tank (hotwell).

Figure 2b shows an ORC system which can be installed in the ship. An ORC uses a working fluid with a lower boiling point than water in order to recover the low-temperature heat source. The ORC system consists of an economizer, a heat exchanger, an expander, a condenser, and a pump (Fig.2b).

Results and Discussion

The results are categorized by thermodynamic performance results, environmental results and economical results. The results are given as following subsections.

Thermodynamic Performance Results

It is examined performance analysis of an ORC system on the container ship. This study is performed for ten different working fluids which are shown in table 2. It is assumed that \( T_{\text{sat,2}} \) is 410 K. The comparison of performance for the fluids in the ORC system on the container ship is shown in Table 4.

<table>
<thead>
<tr>
<th>Working Fluids</th>
<th>( W_{\text{net}} ) (kW)</th>
<th>( \eta_{\text{ORC}} )</th>
<th>( \Delta h_{\text{m}} ) (kg/s)</th>
<th>( \Delta h_{\text{f}} ) (kg/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R600</td>
<td>732</td>
<td>0.54</td>
<td>0.89</td>
<td>0.030</td>
</tr>
<tr>
<td>R601a</td>
<td>1014</td>
<td>0.70</td>
<td>1.23</td>
<td>0.042</td>
</tr>
<tr>
<td>R600a</td>
<td>694</td>
<td>0.54</td>
<td>0.84</td>
<td>0.285</td>
</tr>
<tr>
<td>R601</td>
<td>1002</td>
<td>0.69</td>
<td>1.22</td>
<td>0.041</td>
</tr>
<tr>
<td>R290</td>
<td>321</td>
<td>0.39</td>
<td>0.39</td>
<td>0.013</td>
</tr>
<tr>
<td>R11</td>
<td>833</td>
<td>0.55</td>
<td>1.12</td>
<td>0.038</td>
</tr>
<tr>
<td>R114</td>
<td>852</td>
<td>0.55</td>
<td>1.14</td>
<td>0.039</td>
</tr>
<tr>
<td>R134a</td>
<td>350</td>
<td>0.35</td>
<td>0.47</td>
<td>0.016</td>
</tr>
<tr>
<td>R245fa</td>
<td>739</td>
<td>0.49</td>
<td>0.98</td>
<td>0.033</td>
</tr>
<tr>
<td>R123</td>
<td>869</td>
<td>0.60</td>
<td>1.06</td>
<td>0.036</td>
</tr>
</tbody>
</table>

Table 4 indicates that the net power range is 321 kW and 1014 kW, the thermal efficiency range is 0.39 and 0.70. Maximum power and thermal efficiency are supplied by R601a and R601. These amounts for R601a are equal to 1014 kW and 0.70, respectively. R601a and R601 are the A3 safety class fluids. Although the maximum power in the ORC system is produced by A3 safety class fluids they should not be used on ships since they have a higher flammability feature. B1 class fluids have a toxic effect that is the most serious problems in terms of health. The most important issue in maritime applications is ship safety. Since A1 safety class fluids have non-flammability and non-toxicity features, they are acceptable for ORC system applications of ships.

The performance results for A1 class fluids are shown in Table 5. The result shows that R11, R114, and R134a are three optimal working fluids among these working fluids for the ship. It is observed that Energy Utilization Factor (EUF) could achieve as high as 70% while Fuel Energy Saving Ratio (FESR) is found to be 27%. It can be seen that R114 has the highest EUF, the highest efficiency and the highest FESR. The EUF on the WHR system in the ship is found 0.68 and on the integrated ORC is found 0.704. The FESR in the systems applied with ORC is found 0.267 and without ORC is found 0.150. The artificial thermal efficiency (\( \eta_{\text{a}} \)) and the cogeneration efficiency (\( \eta_{\text{CG}} \)) is found to be 65% and 53%, respectively.
The amount of fuel saving for the fluids is shown in Table 6. It is assumed that the ship has 300 cruising days per year. The maximum amount of fuel saving is achieved from R114 organic fluid. It can be achieved approximately 2% fuel saving by utilizing the ORC system.

Environmental Results

The amount of air pollutants is reduced by reducing fuel consumption. The amount of the reduced air pollutants can be calculated with Eq. (22). Emission factors (EF) of low speed marine diesel engine are shown in Table 7 [31].

The amount of reduced emissions from the ship with ORC is illustrated in Table 8. The largest reduce is carried out by R114 working fluid in the ORC system. The amount of reduced CO₂ is 3200 tons as annually.

Economic Results

Fuel costs comprise a significant proportion of operating costs. For large ships such as container ships, the fuel expenses constitute about 30-55% of the total operational costs [32]. There are positive effects of the ORC system application on economic gains due to saving fuel consumption. Economic gain achieved by using ORC is shown in Table 9 as the daily, monthly and yearly. In this study the unit cost of fuel is assumed as 630 US$/ton averagely [33]. The best result is obtained as 636 thousand dollar per year from the R114 organic working fluids.

Conclusion

The marine transportation is one of the major contributors to climate change and global air pollution. Large ships, particularly container ships, bulk carriers, cruise ships and tankers are significant contributors to air pollution. The IMO has put in place EEDI that represents the amount of CO₂ emitted per mile and per amount of good
transported. It is mandatory for a ship owner to respect this index. The amount of CO₂ emitted from a ship is directly related to the amount of fuel consumed by the engines of the ship. WHR systems allow ships to save energy and reduce CO₂ emissions and have the capability to improve the EEDI index.

In this study, it is proposed an ORC system for the container ship, and thermodynamic, environmental and economic effects of the system are analyzed. Thermodynamic performance criteria of ORC system are developed in order to compare thermodynamic and environment and economic performance of ten typical working fluids. Fluid selection for the ORC is an important issue and is very dependent on the target application, on the working conditions and even on the criteria taken into account. Thermodynamic performance, environmental protection, and safety requirement have to be considered simultaneously when selecting working fluids. However it is difficult to find an ideal organic working fluid which has good thermodynamic performance, zero ODP and low GWP and can meet non-toxic, non-flammable, and non-explosive requirement.

ORC system is proposed for the recovery of the ship’s main engine exhaust gas energy, and ten working fluids are analyzed for this purpose. The result shows that R114, R11 and R134a are three optimal working fluids among these working fluids. It can be achieved additionally 2% reduction for the ship fuel consumption by utilizing the ORC system. In addition to fuel savings, by increasing the efficiency of the overall system, the CO₂ emissions are reduced and therefore the EEDI is improved.

**NOMENCLATURE**

\[ W \]  
Power (kW)

\[ h \]  
Specific enthalpy (kJ/kg)

\[ m \]  
Mass flow rate (kg/s)

\[ q \]  
Heat flow rate (kJ/s)

\[ T \]  
Temperature (K)

\[ F \]  
Fuel energy (kW)

\[ cp \]  
Specific heat (kJ/kgK)

\[ E \]  
Exergy rate (kW)

**ACRONYMS**

ORC  Organic Rankine Cycle

WHR  Waste Heat Recovery

EUF  Energy Utilization Factor

FESR  Fuel Energy Saving Ratio

IMO  International Maritime Organization

EEDI  Energy Efficiency Design Index

SEEMP  Ship Energy Efficiency Management Plan

ODP  Ozone Depletion Potential

GWP  Global Warming Potential

MCR  Maximum Continuous Rating

**REFERENCES**


Cengiz Deniz was born in Turkey in 1965. He received the Diploma in marine engineering from Merchant Marine Academy, Istanbul in 1986 and the MSc and PhD degrees in maritime transportation engineering from Istanbul University in 1994 and 2000, respectively. He worked as second and first engineer in ships. He has been working in Department of Marine Engineering, Istanbul Technical University, Maritime Faculty, since 1992. He is currently an Associated Professor in the Department of Marine Engineering. His research interests include ship power system, marine diesel engines, air pollution from ships, international maritime conventions.