Thermodynamic, environmental and economic analysis of absorption air conditioning unit for emissions reduction onboard passenger ships

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**ABSTRACT**

In this paper, the waste heat of exhaust gases and jacket cooling water in marine diesel engines are analyzed to operate the absorption refrigeration unit (ARU). Thermo-economic and environmental analysis of the absorption refrigeration cycle operated with the two heat sources that use lithium bromide as an absorbent is carried out. The analysis is performed using Engineering Equation Solver (EES) software package where the thermodynamic properties of the steam and the LiBr-water mixtures are provided. The used EES code is verified by published experimental data. As a case study, a high speed passenger vessel operating in the Red Sea area has been investigated. The results show that a considerable specific economic benefit could be achieved from ARU jacket cooling water operated over that gained from main engine exhaust gases. Environmentally, applying ARU machine during cruise will reduce the annual fuel consumption for the diesel generators by 156 ton with a reduction percentage of 23%. This will reduce the exhaust gas emissions by 6.3% from the applied main engine emissions. In addition, this will result in reducing NO\textsubscript{x}, SO\textsubscript{x}, and CO\textsubscript{2} emissions with cost-effectiveness of 4.99 \$/kg, 13.18 \$/kg, and 0.08 \$/kg, respectively.

1. Introduction

The mid-range forecasted scenarios presented in the third International Maritime Organization (IMO) greenhouse gas study in 2014 show that the average annual sulfur oxides (SO\textsubscript{x}) and nitrogen oxides (NO\textsubscript{x}) emissions from ships between the years from 2007 to 2012 are 11.3 and 20.9 million tons, respectively. These emissions represent 13% and 15% of global SO\textsubscript{x} and NO\textsubscript{x} emissions of all man-created sources, respectively (IMO, 2014). In the same context, the contribution of carbon dioxide (CO\textsubscript{2}) emissions from ships accounts to about 8.0% of the global CO\textsubscript{2} emissions from various transportations means as shown in Fig. 1. It is estimated that by 2050, CO\textsubscript{2} emissions from international shipping could grow by 50% to 250%, depending on future economic growth and energy developments (Peters et al., 2013, Boden et al., 2013). Moreover, recent studies of ship emissions state that shipping-related particulate matters (PM) emissions are responsible for approximately 60,000 cardiopulmonary and lung cancer deaths annually, with most of those deaths occurring along the coasts (Berechman and Tseng, 2012; Salvatore, 2015).

In addition to the above mentioned emissions, refrigerants which used onboard vessels for air conditioning and cargo cooling
purposes consider another source of ship emissions. These refrigerants are either ozone-depleting substances like chlorofluorocarbons (CFCs) or their replacements like hydrofluorocarbons (HFCs) 1,1,1,2-tetrafluoroethane (R134a) and a mixture of pentafluoroethane, trifluoroethane and tetrafluoroethane (R404a). All these refrigerants have significant global warming potential (EC, 2017). The average annual loss of refrigerants from the global fleet, based on the latest statistics, indicates that air conditioning equipment is responsible for about 69.8% of the total loss of refrigerants; however 30.2% of this loss is cooling equipment based. Fig. 1 shows the contribution of each ship type based on the loss in refrigerants due to air conditioning. It noticed that general cargo and cruise ships contribute by considerable amounts of loss of refrigerants rather than the other various ship types (IMO, 2014).

Finally, the options applied onboard ships which reduce both the exhaust gas emissions and fuel consumption will have a good
effect from environmental and economic points of view. The importance of fuel saving options onboard ships will be increased especially with the rapid increase in marine fuel oil prices during the last few months as shown in Fig. 2 (Bunkerworld, 2018).

As a result of continuous increasing of emissions quantity emitted by ships, IMO issued a set of regulations regarding this concern through marine pollution convention Annex six (Corbett et al., 2009; Peng et al., 2016; Tichavska et al., 2017). Regulation 14 limits SOx and particulate matter (PM) emissions, while regulation 13 and energy efficiency design indexes limit NOx and CO2 emissions, respectively (Ammar and Seddiek, 2017; El Gohary et al., 2015; Seddiek, 2015). One of the solutions for reducing ship emissions is the absorption refrigeration units (ARUs), which depend on engine heat losses (Salmi et al., 2017; Cao et al., 2015). The heat balance of marine diesel engine, the most used prime mover in ships (Eyring et al., 2005; Kosmas and Acciaro, 2017), shows approximately more than 50% of the fuel input is lost in heat losses including exhaust gases (25.5%), fresh water cooling (5.2%), charge air cooling (16.5%), oil cooling (2.9%), and radiation (0.6%) (Salmi et al., 2017; Cao et al., 2015, Ouadha and El-Gotni, 2013). Three of these heat losses, at different temperature levels, can be used as a source for ARU: exhaust gas (300–600 °C), charge air (200 °C), and jacket cooling water (80–100 °C) (Ouadha and El-Gotni, 2013). These heat loses will depend mainly on the cycle of operation and engine speed.

Traditionally, vapor compression refrigeration systems are the common onboard ships. These systems are powered using electric energy (Seddiek et al., 2012). Although they have the advantages of high coefficient of performance (COP) and low purchase price, the use of these systems will be phased out due to their contribution to the greenhouse effect and the depletion of the ozone layer (Riffat and Qiu, 2004; Wu et al., 2018). Regulation 12 of the Marine Pollution Prevention Convention (MARPOL) Annex VI, issued by IMO, states that: new installations containing ozone-depleting substances are prohibited on all ships from January 2020 (IMO, 2009). On the other hand, absorption cooling is an environmentally friendly cooling method. It uses a number of refrigerant–absorbent pairs. The most common ones are LiBr–water and ammonia–water. They offer a good thermodynamic performance (Cao et al., 2015) and can be operated in single stage and double stage coolers depending on the input heat source. Ammonia absorption cooling system can cool down to −60 °C (Táboas et al., 2014). Lithium bromide absorption coolers mainly can be used in air conditioning to cool down above zero Celsius (Ouadha and El-Gotni, 2013).

The objective of the current paper is to study the thermodynamic, economic and environmental effects of using absorption refrigeration for reducing both the running costs of the air-conditioning and the harmful emissions during ship cruise. High speed passenger ship is investigated, as a case study for application.

2. Thermodynamic analysis

A simple single-effect absorption refrigeration cycle consists of eight components: a generator, a condenser, an evaporator, an absorber, a pump, two throttle valves, and a heat exchanger (HE) as shown in Fig. 3. The working fluid is a mixture of water and lithium-bromide. The generator provides the cycle with the heat ($Q_\text{g}$) to evaporate the water from the water-LiBr solution to high pressure steam (7). The produced steam flows to the condenser where thermal energy ($Q_\text{c}$) is rejected to a cooling medium. A throttle valve reduces the steam pressure (8) and allows it to return to the vapor phase (9). The evaporator represents the cooling capacity ($Q_\text{e}$) of the absorption machine. The exit low pressure steam of the evaporator (10) is absorbed into the lithium-bromide strong solution coming from the generator (6) through rejecting heat energy ($Q_\text{a}$) to a cooling medium. The absorber (1) and condenser (8) output temperatures may have the same value depending on the design process of the absorber and condenser cooling cycle (Hong et al., 2017).
et al., 2010; Yan et al., 2013; Arshi Banu and Sudharsan, 2018; Buonomano et al., 2018). The solution pressure at the exit of the absorber (1) is raised using a pump (2), before it returns to the generator (3) for a new cycle. The solutions leaving the generator (4) and the absorber (1) are referred to as strong and weak solutions, respectively, with reference to the percentage of lithium-bromide (ASHRAE, 2009). The solution heat exchanger preheats the weak solution (2–3) using part of the heat energy of the boiling strong solution (4–5) leaving the generator which improves the cycle efficiency.

Lately, the air circulation system presents the third part, where automatically fan cells will suction the air either from outside (atmosphere) or from circulation branches (ship’s cabins), and passing it thought the chilled water pipes where the cooling load is absorbed. Adjusting generator heat source temperature could be achieved through using a control unit governor on the flow rate of engine exhaust gas passing inside the heat exchanger or through three way valve in case of jacket cooling water heat source.

It is very important to calculate the required heat for the ARU generator. This should be compared with the heat gained by fresh water in case of exhaust gases heat source or the available jacket cooling water. Starting with exhaust gases heat source, applying the heat transfer concept between the exhaust side and the water side, the heat transferred from the exhaust gases stream to the fresh water stream ($Q_{exh}$) can be calculated using Eq. (1).

$$Q_{exh} = m_{exh} C_{p,w} (T_1 - T_2)$$

The maximum exhaust-heat transfer ($Q_{exh,max}$) in the counter flow heat exchanger at any configuration can be calculated as follows

$$Q_{exh,max} = m_{exh} C_{p,w} (T_1 - T_4)$$

The heat gained in the fresh water ($Q_{FW}$) to be used in the ARU generator from the exhaust gases and the heat exchanger outlet temperature ($T_3$) can be calculated from Eq. (3).

$$Q_{FW} = m_w C_w (T_1 - T_4) = Q_{exh,max} \epsilon$$

where ($\epsilon$) is the counter flow heat exchanger effectiveness, ($N_u$) is the number of transfer unit, and ($R_c$) is the heat capacity rate ratio. The values $N_u$ and $R_c$ can be estimated using the thermodynamic equations listed by Holman (2002).

$$\epsilon = \frac{1 - R_c e^{-N_u(1-R_c)}}{1 - R_c}$$

On the other hand, the heat gained from jacket cooling water ($Q_{jcw}$) can be calculated using Eq. (5).
\[ \dot{Q}_{cw} = \dot{m}_{cw} C_w (T_h - T_a) \]  

(5)

2.1. Case study

One of the high speed ferries, operating at the Red sea area, is the Riyadh catamaran ferry. The ferry connects the two ports Dubai in the Kingdom of Saudi Arabia and Safaga port in Egypt. The main technical data of this ferry can be shown in Table 1 (Austral, 2008). The increased numbers of these ships sailing in the Red Sea area have improved the maritime transport in this area. These improvements have increased the number of voyages per year and consequently the amount of the gaseous emissions (Seddiek, 2016; Seddiek et al., 2012). These emissions lead to harmful impacts on the environment in this area.

The proposed onboard absorption air conditioning system will consists of three main parts as shown in Fig. 3. The first part presents the necessary arrangements required to achieve the required inlet hot water temperature for ARU that could be obtained by using either the engines exhaust gas heat losses for heating a separately fresh water cycle or using main engine jacket cooling water directly.

The second part is the Li-Br ARU, which will provide the required cooling load in form of very low temperature. Generally, there are three configurations of Li-Br system: single effect, double effect, and half effect. The main differences among the previous systems are COP value and heating range. Most of researches pointed that single effect type is suitable for waste heat application (ASHRAE, 2009).

2.2. Thermodynamic modeling

In order to conduct a performance evaluation of LiBr-water absorption cooler, some assumptions and initial values must be considered. The required initial values are the evaporator capacity \((\dot{Q}_e)\), the generator, condenser, evaporator, and absorber output temperatures and the effectiveness of the solution heat exchanger. The main assumptions for the thermodynamic model are: the cycle is a steady state system, the refrigerants at the evaporator and condenser outlets are assumed as saturated vapors and saturated liquids, respectively, flow restrictors are adiabatic, heat loss to the surroundings is negligible, and there are no pressure losses in the pipes and heat exchangers.

Based on the above assumptions, mass and energy balance equations can be applied to the absorption cycle taking into account the operating conditions at each point. The indexes presented in Fig. 3 are used to nominate the state points in the ARU cycle. Applying energy balance equations to the absorber, the condenser and the evaporator gives the following expressions:

\[ \dot{Q}_e = \dot{m}_h h_{10} + \dot{m}_{sw} h_6 - \dot{m}_{sw} h_1 \]  

(6)

\[ \dot{Q}_c = \dot{m}_h (h_7 - h_8) \]  

(7)

\[ \dot{Q}_c = \dot{m}_h h_{10} - \dot{m}_h h_9 \]  

(8)

The ratio of the solution mass flow rate leaving the absorber to the vapor mass flow rate entering the condenser is called the cycle circulation ratio \((F)\).

\[ F_c = \frac{\dot{m}_{sw}}{\dot{m}_h} \]  

(9)

The temperatures of the solution heat exchanger exit points can be determined from the following energy balance and heat exchanger effectiveness \((\varepsilon_{HE})\) Eqs. (10) and (11).

\[ \dot{m}_{sw} h_2 + \dot{m}_h h_4 = \dot{m}_{sw} h_3 + \dot{m}_h h_5 \]  

(10)

<table>
<thead>
<tr>
<th>Table 1</th>
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<tr>
<td>Technical data of Riyadh high speed passenger ship.</td>
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<td>Ship’s item</td>
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<tr>
<td>-------</td>
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<tr>
<td>Ship Name</td>
</tr>
<tr>
<td>Type</td>
</tr>
<tr>
<td>IMO number</td>
</tr>
<tr>
<td>Year of built</td>
</tr>
<tr>
<td>Flag</td>
</tr>
<tr>
<td>Passengers/crew</td>
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<tr>
<td>Main Engine</td>
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<tr>
<td>Maximum Continuous Rating</td>
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<tr>
<td>Service Speed</td>
</tr>
<tr>
<td>Fuel Consumption at 90% MCR</td>
</tr>
<tr>
<td>Generating Sets</td>
</tr>
<tr>
<td>Air conditioning cooling capacity</td>
</tr>
<tr>
<td>Number of trips per year</td>
</tr>
</tbody>
</table>
The coefficient of performance (COP) measures the refrigeration cycle performance. It can be calculated using Eq. (12).

\[
\text{COP} = \frac{Q_g}{Q_g + W_p}
\]  

(12)

where \( Q_g \) is the heat input to the generator. It can be determined from the energy balance for the generator using Eq. (13).

\[
Q_g = \dot{m}_g h_4 + \dot{m}_p h_2 - \dot{m}_w h_1
\]  

(13)

\( W_p \) is the power required to pump the exit absorber low pressure (PL) solution side to the generator high pressure (PH) side.

\[
W_p = \dot{m}_w V_p (P_{PH} - P_{PL})
\]  

(14)

Mass and energy equations for the LiBr-water absorption cycle are analyzed using Engineering Equation Solver (EES) software under steady-state operations. It has a built in functions for the thermodynamic properties of the steam and the LiBr-water mixtures. The thermodynamic properties of the steam and the LiBr-water mixtures for the current EES model are verified using the cases provided in (Pátek and Klomfar, 2006, 2009). In addition, current modeling system of the LiBr-water absorption cycle using EES program is validated by the theoretical and experimental data provided by (Florides et al., 2003). The main input parameters are: the cooling capacity of 11 kW, the temperatures of the generator, absorber, condenser, and evaporator are 90 °C, 34.9 °C, 30 °C, and 6.7 °C, respectively, strong and weak mass fractions of 60% and 55.5% respectively, and heat exchanger effectiveness of 60%. Table 2 shows the results for the two studies. The maximum difference percentage between the results was 1.13% for the cycle COP.

2.3. Environmental and economic modeling

The reduction in emission quantity after applying ARU (\( ER_{ARU} \)) per year, during cruise, can be calculated using Eq. (15).

\[
ER_{ARU} = Q_{El} T_s F_e
\]  

(15)

where \( Q_{El} \) is the saved electric power in kW at cruise, \( T_s \) is the sailing hours per year, and \( F_e \) is the emission factor for the engine in (kg/kWh).

ARU can be considered as one of the emission reduction measures. The annual emission cost-effectiveness for applying ARU onboard the ship (\( ACE_{ARU} \)) can be calculated for the reduction in each type of pollutants emissions using Eq. (16) (Ammar and Seddiek, 2017; ICF, 2009b).

\[
ACE_{ARU} = \frac{C_{tu}}{ER_{ARU}}
\]  

(16)

where \( C_{tu} \) is the total annual cost of ARU machine which includes the summation of the capital, operating and maintenance costs in ($/year).

Total annual installation cost of ARU system (\( C_{tu} \)) depends upon the annual initial money recovery (AMR), installation cost (\( C_{ins.} \)), the annual maintenance and operating costs (\( C_{m&o} \)) and the heat exchanger cost unit cost (\( C_{HE} \)) in case of exhaust gas operated ARU. \( C_{tu} \) can be calculated using Eq. (17).

\[
C_{tu} = AMR + C_{ins.} + C_{m&o} + C_{HE}
\]  

(17)

The annual initial money recovery (AMR) when applying ARU depends on the unit cost (UC), the expected ship age after applying the absorption system (n), and the interest rate (i) (Hunt and Butman, 1995). AMR can be calculated using Eq. (18).

\[
AMR = UC \times \frac{i(1 + i)^n}{(1 + i)^n - 1}
\]  

(18)

Fuel saving due to using ARU instead of CRU during sailing (\( m_{fs} \)) depends on saved electric power during the trip (\( Q_{El} \)) in (kW), specific fuel consumption for the diesel generator (\( b_{edg} \)) in (kg/kWh) and the sailing hours per year (\( T_s \)). Thus, \( m_{fs} \) can be calculated using Eq. (19).

<table>
<thead>
<tr>
<th>Table 2</th>
<th>Cycle validation.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
<td>Present study</td>
</tr>
<tr>
<td>COP</td>
<td>0.696</td>
</tr>
<tr>
<td>Generator (kW)</td>
<td>14.35</td>
</tr>
<tr>
<td>Condenser (kW)</td>
<td>10.90</td>
</tr>
<tr>
<td>Absorber (kW)</td>
<td>13.44</td>
</tr>
<tr>
<td>Pump (kW)</td>
<td>0.289</td>
</tr>
</tbody>
</table>
The Fuel cost saving ($\text{Cfs}$) will depend on three main factors including: the amount of fuel saved, fuel prices ($\text{Cf}$ in $$/kg), and fuel price yearly change ($\text{PI}$). This can be expressed using the following equation:

$$\text{Cfs} = \text{m}_{\text{BA}} \cdot \text{Q}_{\text{E}} \cdot \text{T}_{\text{fs}}$$

(19)

3. Results and discussion

The results include the effect of main engines heat losses and the specifications of ARU on the viability of using the absorption refrigeration system onboard the case study. In addition, environmental and economic results for applying the two the waste heat recovery operated ARU machines onboard the selected case study is discussed.

3.1. Thermodynamic results

3.1.1. Waste heat and ARU analysis

From the technical data of the case study, Carrier absorption unit (16LJ11) with nominal cooling capacity of 264 kW (Carrier, 2016) can be used onboard to cover the required 250 kW refrigeration load. The required mass flow rates for the hot and cooling water streams for the selected model are 10.4 kg/s and 17 kg/s respectively. The inlet and outlet temperatures are in the range of 95–75 °C and 29.4–38.4 °C, for the hot and the cooling water streams, respectively. The output child water mass flow rate is 11.4 kg/s with inlet and outlet temperatures of 12.27 °C and 6.7 °C, respectively. The required hot water with mass flow rate of 10.4 kg/s and a hot water temperature difference in either one of the ranges from 95 °C to 85 °C or from 85 °C to 75 °C. Therefore, the required waste heat source should provide Carrier absorption unit (16LJ11) with the required hot water temperature and mass flow rate. The absorption machine is assumed to be operated only during the cruise period of 8 h.

Using Eqs. (1)–(6), the main engine waste heat energy streams of the case study and the ARU cycle coefficient of performance can be calculated. The two streams are the exhaust gases and the jacket cooling water heat losses. The exhaust gases represents 25.5% of the total 16,093 kW input power with exhaust mass flow rate of 8.7 kg/s and a temperature of 460 °C (MTU, 2014). These gases can provide the absorption unit (16LJ11) with the required hot water temperature in the range of 95 °C to 85 °C and mass flow rate of 10.4 kg/s. Therefore, the available generator power will be 343.7 kW. The outlet exhaust gases temperature from the heat exchanger will be 415 °C, assuming heat exchanger effectiveness of 60% (Balghouthi et al., 2008, Ibrahim et al., 2018).

On the other hand, the heat transferred to the jacket water cooling is 1091 kW at a mass flow rate of 26.11 kg/s and output temperature of 85 °C with temperature difference of 10 °C (MTU, 2014). This hot water stream can provide the absorption unit (16LJ11) with the required hot water temperature in the range of 85 °C to 75 °C and mass flow rate of 10.4 kg/s. This will provide the ARU generator with the same required power as the exhaust gases heat source. The current paper aims at assessing the effect of the different operating parameters on the ARU unit using exhaust and jacket cooling water heat sources. In addition, the expected total life cycle for the two ARU units will be evaluated in order to make an economical decision based on both initial and maintenance costs.

Although ARU consists of many components, three components play a role in its capability and usage onboard ships, mainly; condenser, absorber and evaporator. It is noted that, the effect of both the condenser and the absorber have nearly similar trends. This
trend agrees with the published paper (Mortazavi et al., 2010). The effect of the condenser and evaporator temperatures on the performance of the water-LiBr refrigeration cycle is studied as follows:

### 3.1.2. Effect of ARU condenser temperature

The effect of the condenser temperature on the performance of the water-LiBr refrigeration cycle is generated by changing $T_c$ and $T_g$, and simultaneously maintaining $T_a$ at 30 °C, and $T_e = 6.7$ °C. Fig. 4 shows the effect of condensation temperature on the cycle COP and the required generator power for both exhaust gases and jacket cooling water heat sources. The required generator power increases significantly with increasing condensation temperature. Its maximum values ranges from 325 kW to 315 kW for the generator operated on jacket cooling water and exhaust gases, respectively. As the condensation temperature increases at constant evaporator temperature, the refrigeration mass flow rate will be increased which is direct proportional to the required generator power (Eq. (13)). The In contrast, the cycle COP decreases with the increase of condenser temperature. This decrease in the cycle COP agrees with the experimental results of (Palacín et al., 2011).

The variation of the absorber power and strong solution mass fractions at the different output condensation and generation temperatures is illustrated in Fig. 5. As the condensation temperature increases, the strong solution mass fraction decreases, while the absorber power increases. The strong solution mass fraction is increased by an average of 6.8% when the generator operates on the exhaust gases heat source and condensation temperature of 20 °C compared with that of jacket cooling heat source. This is due to the increased exit generator temperature from 80 °C to 90.5 °C. On the other hand, the absorber power decreases by approximately 0.7% when the solution mass fraction increases by 6.8% at the same condensation temperature. This is due to the increased strong solution mass flow rate from 0.237 kg/s to 0.30 kg/s.

### 3.1.3. Effect of ARU evaporator temperature

The influence of the evaporator temperature on the performance of the water-LiBr refrigeration cycle is studied, in this section. This analysis is made by varying $T_e$ and $T_g$, and at the same time keeping $T_a$ and $T_c$ at 30 °C. The temperature of the refrigerant leaving the evaporator is varied from 5 °C to 25 °C at the two generator temperatures of 90.5 °C and 80 °C. Fig. 6 shows the effect of the evaporation temperature on both the circulation ratio and the required pump power. The circulation ratio and the required pump power reduce as the evaporator temperature increases. The required pump power of the jacket cooling water operated generator is lower than that of the exhaust gas. As a result of this, the economic operation of the absorption cooling machine is improved with the increased evaporator temperature especially when the temperature at which the available heat source is low. This agrees with the results published by (Wonchala et al., 2014).

When the evaporation temperature is increased maintaining both $T_e$ and $T_a$ at 30 °C, the mass fraction of the absorber power and weak solution mass fractions will be reduced keeping strong solution mass fractions at the same value as shown in Fig. 7. The decrease in the power and mass fraction approaches their lowest values when the evaporation temperature reaches its condensation temperature. In addition, both the strong and weak solutions mass flow rate starts with a high value, and they will be reduced by 55.77% and 67.09%, respectively at the end of the considered range of the evaporator's temperature, for the generator exit temperature of 80 °C. These reduction percentages changes to be 46.65% and 60% when the exit generator temperature is increased to 90.5 °C, respectively. The results also indicate that the absorber powers for the jacket cooling water and the exhaust gases heat sources are nearly the same at high evaporator temperatures starting from 15 °C.

![Fig. 5. Effect of the condensation temperature on the absorber power and strong solution mass fraction.](image-url)
Fig. 6. Effect of the evaporation temperature on both the circulation ratio and pump power.

Fig. 7. Effect of the evaporation temperature on the solution mass fractions and the absorber power.

Fig. 8. Comparison between the present emissions quantity of M/V Riyadh and IMO limits.
3.2. Environmental and economic results

Technically, using ARU will lead to saving in the required generator power, and consequently reduce both fuel consumption and emissions. For the current case study, ARU will save fuel consumption by 156 ton/year, with fuel price of 600 $/ton (Bunkerworld, 2018). This saving will result in dispensing with one of the diesel generators during ship cruise. The emission factors for the high speed diesel engines operated with marine diesel oil (MDO) with 1.0% sulfur are 10.81 g/kWh, 4.1 g/kWh, 0.3 g/kWh, 645 g/kWh, and 0.2 g/kWh for NOx, SOx, PM, CO2, and HC emissions, respectively during cruise (Banawan et al., 2010, ICF, 2009a). From these factors, emission rates in kg/min during trip for each main engine can be calculated as shown in Fig. 8. For the case study, SOx and NOx emission rates are 0.3936 kg/min and 1.038 kg/min, respectively, during cruise mode for each main engine. These rates should be compared with IMO 2020 and IMO 2016 (Tier III) rates of 0.24 kg/min and 0.287 kg/min, respectively. The same can be done for the maneuvering and stand by modes. From Fig. 8, both NOx and SOx emission rates of the high speed diesel engine will not be combatable with the new IMO emission limits during ship cruise. This will highlight on the importance of the ARU to meet the stringent IMO regulations.

Environmental benefits of the ARU unit are clear when comparing the yearly emissions reduction in ton/year after applying ARU unit, as shown in Fig. 9. The yearly emissions from the main engine are 102.7 ton/year, 38.87 ton/year, 2.981 ton/year, 6115 ton/year, and 1.987 ton/year for NOx, SOx, PM, CO2, and HC, respectively. Applying ARU unit during cruise mode will reduce these emissions by 6.48 ton/year, 2.46 ton/year, 0.18 ton/year, 387 ton/year, and 0.12 ton/year, respectively.

Economically, the application of the ARU unit can be judged from the annual installation costs and its recovery period. The capital cost of the ARU unit ranges from 500 $/kW to 700 $/kW with installation costs of 12% from this cost. The annual operating and maintenance costs are 8 $/kW and 0.008 $/kWh, respectively (Gupta et al., 2016; Seddiek et al., 2012; Florides et al., 2003). For the application of carrier absorption unit (16LJ11), the capital and installation costs are $218,817 and $206,976 with yearly fuel saving of 52,000 $/year for exhaust gas and jacket cooling water operated ARU units, respectively. The heat exchanger costs are the only added cost for the exhaust gas operated ARU unit compared with jacket cooling water. It adds extra expenses for the total ARU costs but it improves the performance of the generator compared with using exhaust gases directly (Seddiek et al., 2012). The initial heat exchanger cost is $10,572 which presents 5.7% of the capital costs of the exhaust gas operated ARU unit. The total life cycle cost (LCC) of the two ARU units depends on the ship age and the remaining working years after the installations. Fig. 10 shows the LCC for exhaust gas and jacket cooling water operated ARU units over 18 years of operation, assuming the average ship age 28 years (Ammar and Seddiek, 2017; Mikelis, 2008). While the capital cost of the ARU unit is $184,800 for jacket cooling water operated ARU unit, the capital cost of the exhaust gas operated ARU unit is $195,372 which represents 54.92% of the LCC for this unit. The operating and maintenance costs are $38,016 and $91,238, respectively representing 11.3% and 27.1%, respectively of the LCC for jacket cooling water operated ARU unit. Therefore, evaluating the total LCC will help in purchasing decisions for the ARU unit. The decision will be based on the unit which minimizes energy and maintenance costs.

In addition, the economic decision for applying ARU unit should consider the time required for the money recovery. Fig. 11 shows the yearly fuel saving cost and the annual cost for the capital cost recovery with the pay back periods for the exhaust gas operated ARU unit. The pay back periods should be compared with the available economic life of the ship. For the case study, the annual costs for capital money recovery are 57,723 $/year and 54,600 $/year for the exhaust gas and jacket cooling water operated ARU units, respectively, at i = 10%, and 6 years payback period. Moreover, the fuel saving costs has to be considered in order to evaluate the total economic benefits for applying ARU unit on board the case study. The annual fuel saving costs at the end of the 18 years of ship operations will be $133684, at 2% fuel price increment.

On the other hand, calculating the annualized cost-effectiveness for each reduction in emissions will show extra eco-environmental benefits for applying ARU unit for the main engine. Fig. 12 evaluates the cost-effectiveness for each emission reduction. The
lower the annual emission cost-effectiveness, the higher the economic benefit for reducing the emission parameter. The effectiveness costs for reducing SOx emissions are 13.94 $/kg and 13.18 $/kg for exhaust gas and jacket cooling water operated ARU units, respectively, as shown in Fig. 12. NOx emissions will be reduced by 648 ton/year with cost-effectiveness of 5.28 $/kg and 4.99 $/kg, respectively. The most economic cost-effectiveness option for applying ARU unit is for the reduction of CO2 emissions. It is reduced by 387 ton/year with cost-effectiveness of 0.08 $/kg.
4. Conclusions

The application of water-LiBr absorption cooling machine has been introduced for applying onboard ships. As a case study, high speed passenger vessel operating in the Red Sea area has been investigated. The application is discussed from thermodynamic, environmental and economic points of view. The main conclusions can be summarized as follows:

- From thermodynamic point of view, both exhaust gas and jack cooling water heat sources can provide the required heat for the ARU generator to cover the required cooling capacity of 250 kW.
- From an environmental point of view, the application of ARU machine will save fuel consumption by 156 ton/year. This will reduce NOx and SOx emissions by 6.48 ton/year and 2.46 ton/year, respectively. The highest emission reduction is in CO2 emissions by 387 ton/year.
- From an economical point of view, using exhaust gases and jacket cooling water waste-heat sources for ARU machines will annually save $133,684 at the end of the expected its life cycle with 2% fuel price increment. Based on the total life cycle cost analysis, jacket cooling water operated ARU unit is more economical than that of exhaust gases with capital and installation costs of $218,817 and $206,976, respectively. It will lead to a reduction in SOx, NOx and CO2 emissions with cost-effectiveness of 13.18 $/kg, 4.99 $/kg, and 0.08 $/kg, respectively. Both systems can be considered as an economic option for the newly built ships or currently operated ships with available payback period of six years.

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