Alexandria Engineering Journal (2016) 55, 1951–1960

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ORIGINAL ARTICLE

Development of a waste heat recovery system onboard LNG carrier to meet IMO regulations

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Received 20 April 2016; revised 24 July 2016; accepted 26 July 2016
Available online 25 August 2016

KEYWORDS

Waste heat recovery;
LNG carriers;
Exhaust;
Reduction

Abstract Problems resulting from gas emissions lead to increase the concern about safety and health issues with the demand to reduce the emissions from marine shipping. Marine power plants are considered as one of the greatest contributors in the pollutants around the world. Waste heat recovery systems when implemented with ship propulsion system can reduce emissions, fuel consumption and improve the overall efficiency of power generation and utilization. The present article describes the waste heat recovery technology and the potential for ship operators to lower the fuel costs, exhaust emissions, and the effect on the EEDI of the ship. The main research target is to improve the propulsion machinery efficiency of liquefied natural gas carrier using WHRS. The proposed system leads to meet the requirements and regulations set by the IMO for TIER III.

1. Introduction

Marine shipping is held responsible for environmental impacts including greenhouse gas emissions, acoustic and oil pollution. The IMO estimated theses emissions to be equal to around 4.3% of the global emissions and this ratio is expected to be tripled by the year 2020. Most of the carriers used in marine transport are using diesel, steam or gas turbine propulsion power plants. Although other renewable/hybrid propulsion systems are available they still didn't prove themselves reliable or safe to be used in variable conditions. The most common type of machinery used for propulsion is diesel and gas or steam turbine propulsion for applications where speed is critical [1,2].

The internal combustion engines are one of the main sources of pollution, the recent trend to utilize the use of fuel to the maximum potential where increasing costs in energy, increase in emissions and the fear of depletion of the natural sources of fossil fuels lead to utilization of a waste heat recovery systems to improve the overall energy efficiency [3].

About 48–51% of the total heat energy of the Internal Combustion Engine is thrown back to the atmosphere without any use which considered the main source of waste heat in marine diesel engines. The waste heat recovery system can reclaim and capture the waste heat and improve the overall efficiency of the plant. The process is considered as one of the best energy saving methods to make a more efficient usage of fuels to achieve environmental improvement as shown in Fig. 1 [4,5].

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Peer review under responsibility of Faculty of Engineering, Alexandria University.

http://dx.doi.org/10.1016/j.aej.2016.07.027
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Nomenclature

BSFC	Brake specific fuel consumption [kg/kW h]
BSEC	brake specific energy consumption
CO_2	carbon dioxide
C_P	specific heat at constant pressure [kJ/kg K]
EEDI	energy efficiency design index
HHV	higher heating value [MJ/kg]
IMO	international maritime organization
LHV	lower heating value [MJ/kg]
m_f	fuel mass flow rate [kg/s]
m_g	gas mass flow rate [kg/s]
NO_x	nitrogen oxide
Q_g	heat energy extracted from flue gas [kW]
Q_e	engine electrical energy output [kW]
Q_f	heat energy generated by combustion [kW]
Q_L	heat energy lost [kW]
Q_r	heat recovery from exhaust flue gas [kW]
SO_x	sulfur oxide
SRC	steam rankine cycle
WHRS	waste heat recovery system
TEG	thermoelectric generation

Greek symbols

η	thermal efficiency
ρ	density [kg/l]

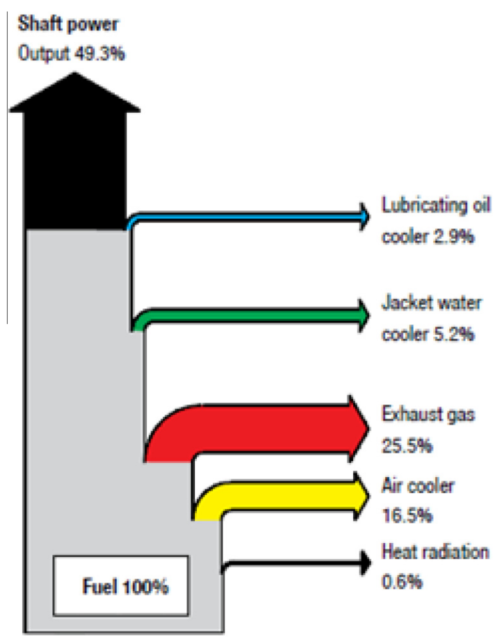

Figure 1 Heat recovery Sankey's diagram of diesel engine [4,5].

2. Waste heat recovery

Waste heat is part of the heat generated by fuel ignition process, where this heat is dumped into the ambient without any use and only partial of the heat energy generated through fuel combustion is used just for propulsion purposes. Waste heat has three main grades: low, medium and high, each of which depends on the temperature of heat supplied. The waste heat recovery systems can be classified according to the application methods as: Refrigeration, Desalination and Power Generation. Other applications are available such as Thermoelectric Generation (TEG), and Turbo-compounding. The most common types of WHRS are summarized in Table 1 [6–8].

2.1. Waste heat recovery in internal combustion engines

Waste heat temperature from an internal combustion engine is one of the rich sources of waste heat. The temperature ranges from 315 to 600 °C depending on multiple variables [9]. Generally, there are exactly four attractive sources with significant potential in an internal combustion engine where the waste heat is considered usable due to the heat flow and temperature. The most attractive among the waste heat sources of an internal combustion engine are heat of exhaust gases, heat of engine Jacket Cooling Water, heat of Lubricating Oil Cooling Water and heat of Turbocharger Cooling [10].

Table 1 Most common uses of WHRS [8].

Use	Temperature range (°C)
Absorption refrigeration	120–140 (steam)
Adsorption refrigeration	120–140 (steam)
Dehumidification	80–85 (hot water)
Organic Rankine cycle	65
Steam Rankine cycle	530

2.2. Benefits of waste heat recovery systems

The waste heat recovery system is one of the best energy saving methods to make a more efficient usage of fuels to achieve economic and environmental improvement. The ship propulsion machinery generally runs at a constant speed for a long time during the sailing period in which it will be easier to make use of more stable waste heat on ship board. The high temperature stage of WHR was used for electric production or mechanical power while the low temperature stage was used for process feed water heating or space heating. Different techniques of waste heat recovery system must be selected according to the different characteristics and applied temperature of both the heat source and the daily life requirements aboard ships.

The benefits of waste heat recovery system on board ships may be summarized as the following:

- It is an environmentally friendly solution to reduce the ship fuel consumption.
- The quantity of energy recoverable in the exhaust gas is increased without affecting the air flow through the engine, hence, the thermal load remains constant without any change in the engine reliability.

- It can contribute in significant savings in both fuel costs and overall exhaust-gas emissions such as CO_2, NO_x, SO_x.
- Helps in meeting the regulation set by the IMO to meet the Tier III [11].

International Maritime Organization (IMO) an agency that has been formed to promote marine safety. The IMO set a limit for diesel engine depending on engine maximum operation speed, as shown in Fig. 2. Both Tier I and Tier II limits are global while Tier II standards apply only to ECA (Emission Controlled Area). The emission level for any marine propulsion engine is expected to meet the regulations of the technical code in case of Tier II and Tier III limits. The figure above represents the limit of nitrogen oxide within each Tier for variable Rated Engine speed. As shown in Fig. 2, 514 rpm is the rated engine speed that will be shown later in Section 3, the NO_x limit is almost 2.5 g/kW h for the Tier III, hence this will be the goal of the present work to reduce the emissions to meet that requirement. A mathematical model is developed for a waste heat recovery system to show the results and the effects of using such system on the LNG carrier [13,14].

3. Case study

A Liquefied Nature Gas Carrier build by STX France in 11/06 is selected to be the preset case study to explain the proposed WHRS. The ship principle dimensions and specification of the propulsion machinery are present and listed in the following paragraphs. The main target of the present work is to reduce the ship's emissions, reducing the waste heat and make the LNGC applicable with the IMO requirements for Tier III. The research is divided into multiple sections; the first one will be about developing a mathematical model of the proposed heat recovery cycle. The second section will be about acquiring the potential of recovering the waste heat from the engines [15,16].

The case study at hand is a ship with 3 dual fuel engines using liquefied natural gas as the main fuel, where before any modifications about 48% of the total energy [17,18] is wasted to the surrounding in several forms. The case is going to focus on the exhaust gas only and how to utilize it to improve the overall efficiency of the ship so it can meet the requirements of the Tier III [19].

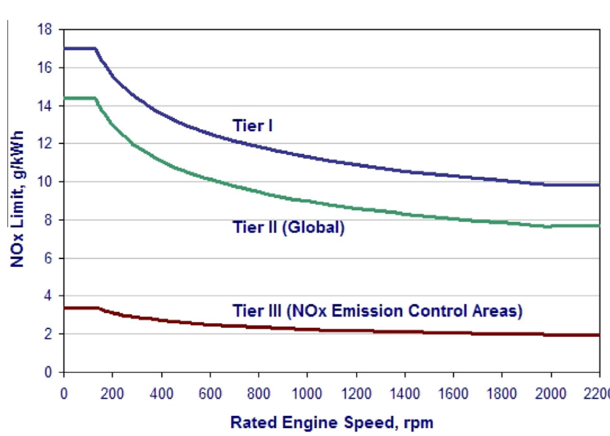

Figure 2 MARPOL Annex VI NO_x emission limits [12].

As one of the promising waste heat recovery systems the model used is a simple Rankine cycle which consists of a boiler, condenser, pump and turbine. The turbine can be considered as a type of WHRS since it is considered as a power turbine [20]. The cycle is modeled with respect to the pinch point temperature difference, turbine power output and approach temperature [21].

In the present work, the selected engine is Warsila 12V50DF for the main propulsion system and utilizing one of the WHRS (ex: Rankine Cycle), the main technical data are shown below in Table 2.

3.1. Heat balance analysis and the potential of exhaust gases

The principle equations are used to calculate and construct the heat balance sheet for the given internal combustion engine unit. The heat balance parameters and fuel properties can be predicted through the following equations using the liquefied natural gas data, where the properties are given as follows: the higher heating value is 52.225 MJ/kg, lower heating value is 47.141 MJ/kg, density is 0.466 kg/l and the specific heat at constant pressure is 1.243 kJ/kg K [23].

The discussion of ships powered by diesel engines is based on the heat balance of the engine in use. To evaluate the heat rejected by the engine to the atmosphere in the form of exhaust gases, a heat balance is carried on to determine the total heat added by the fuel and evaluate the other dependencies of the engine. The following equations were used to evaluate the total heat rejected from the engine:

$$\mathrm{BSFC} = \mathrm{BSEC}/(\mathrm{LHV} * 1000) \tag{1}$$

$$Q_e = \eta_{generator} \times \text{Heat Power} \tag{2}$$

$$\dot{m}_f = \mathrm{BSFC} * \text{Heat Power} \tag{3}$$

$$Q_f = \dot{m}_f \times \mathrm{LHV} \tag{4}$$

$$Q_L = Q_f - Q_e \tag{5}$$

$$Q_{gas} = \dot{m_{gas}} \times C_{p_{gas}} \times (t_{g_{in}} - t_{g_{out}}) \tag{6}$$

$$Q_r = Q_{gas}/Q_f \tag{7}$$

Since the case we are working with is a three identical engines with the properties that are shown in Table 2, whereas the heat recovery potential of a single engine is to be found. The overall potential from the three identical engines can be found as a percentage from the overall heat released from the three engines. Fig. 3 illustrates the heat recovery potential for one engine only as a function of the fuel flow rate. It can be concluded from the figure that there is a drastic decrease in the

Table 2 Main technical data of the ship [22].

Technical data	
Overall length	289.6 m
Depth	26.25 m
Breadth	43.35 m
Gross tonnage	97,741 Ton
Maximum speed	20.45 knot
Main engine type	3 Engines (Wartsila 12V50DF)
Power [per single engine]	11,700 kW @ 514 rpm

heat recovery potential as the fuel flow rate is increased. Conversely, as the flow is throttled to a lower value which indicates working at a partial load the heat recovery potential rises rapidly.

The output results of the Eqs. (1)–(7) are used to construct Fig. 3 for each engine load condition is plotted. The results from the figure can be concluded as that the increasing or decreasing the engine load will result in either increasing the recovery potential of the exhaust flow gases or decreasing that value. The propulsion engine of the present study will run at partial load with fuel flow rate 1950 kg/h so the potential could be around 16–18% for the required work load and that the increase in the mass flow rate of fuel would decrease the engine load thus decreasing the amount of heat that could be recovered. From the data acquired the modeling processes of the Rankine cycle can begin with the exhaust gas value know from the engine catalog (see Figs. 4 and 5).

3.2. Rankine cycle model

- To start evaluating the WHRS it is required to evaluate the temperature profile for both gas and steam generation function.
- Two variables affect the steam temperature profile and steam generation rate are the approach point and the pinch point temperature difference (PPTD).
- For the shown figure as below, equations can be written to find the required values of the exhaust temperature and other parameters.
- A selected approach temperature of 8 °C is used in the system shown above [24].
- It is known that the exhaust gas mass flow rate is 42.3 kg/s with an exhaust temperature of 373 °C at required load condition.
- For calculation purposes the following parameters were assumed and later on iterated to find the best efficient assumptions to work on.
- The steam cycle conditions:
  1. Inlet steam is super-heated at 17 bar and 280 °C.
  2. Condenser pressure is 0.07 bar.
  3. Variable steam turbine power output (first assumption = 3 MW).

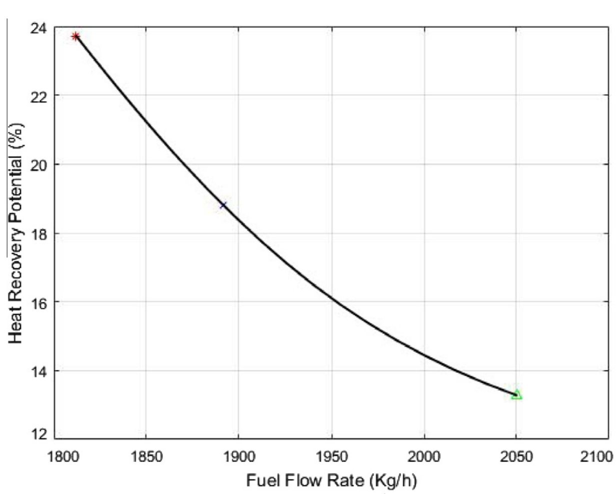

Figure 3 Heat recovery rate potential from single engine.

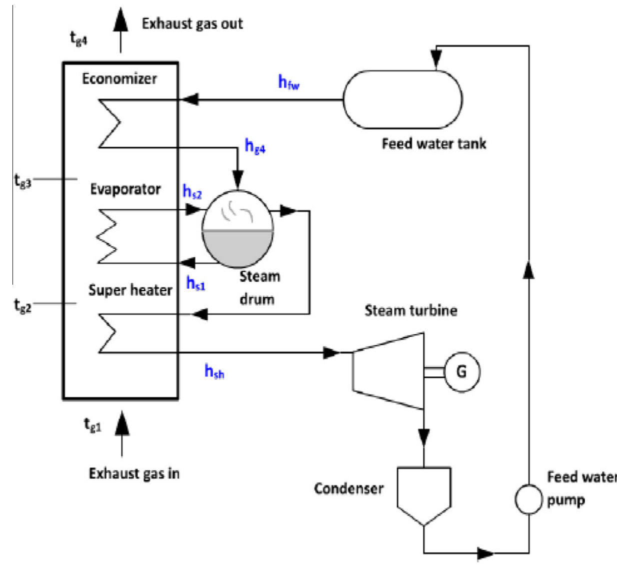

Figure 4 Simplified Rankine cycle.

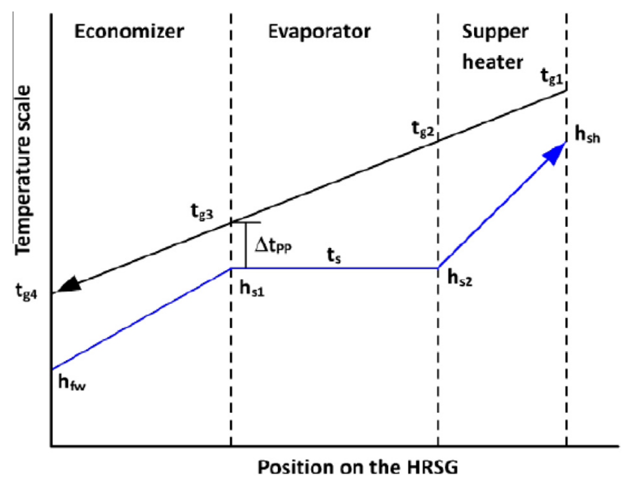

Figure 5 HRSG temperature profile.

4. Isentropic efficiency of steam turbine = 0.93.
5. Mechanical efficiency of the steam turbine = 0.98.
6. Losses to the surrounding about 2%.

In order to facilitate the evaluation of system's parameters, the current work was done by a Matlab code which was written to help with the parameters' calculation. Where the model implemented for the steam cycle has several governing equations which can be summarized from Eqs. (8)–(24).

$$W_{pump,in} = \dot{m}_{steam} \times (h_2 - h_1) \tag{8}$$

$$Q_{in} = \dot{m}_{steam} \times (h_3 - h_2) \tag{9}$$

$$W_{turbine,out} = \dot{m}_{steam} \times (h_3 - h_4) \tag{10}$$

$$Q_{out} = \dot{m}_{steam} \times (h_4 - h_1) \tag{11}$$

$$W_{net} = W_{turbine} - W_{pump} \tag{12}$$

$$\eta_{cycle} = \frac{W_{net}}{Q_{in}} \tag{13}$$

$$\eta_P = \frac{w_s}{w_a} = \frac{h_{2s} - h_1}{h_{2a} - h_1} \quad (13)$$

$$\eta_T = \frac{w_a}{w_s} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}} \quad (14)$$

$$W_{pump,in} = \dot{m}_{steam} \times (h_{2a} - h_1) \quad (15)$$

$$W_{turbine,out} = \dot{m}_{steam} \times (h_3 - h_{4a}) \quad (16)$$

$$\Delta t_{pp} = t_{g3} - t_s \quad (17)$$

$$\dot{m}_g \times C_p \times (t_{g1} - t_{g3}) = \dot{m}_{st} \times (h_{sh} - h_{appr}) \quad (18)$$

$$P_{st} = \dot{m}_{st} \times (h_{sh} - h_{out}) \times \eta_m \quad (19)$$

$$h_{out} = h_{sh} - \eta_{is} \times (h_{sh} - h_{out}, is) \quad (20)$$

$$\dot{m}_g \times C_p \times (t_{g1} - t_{g4}) \times (1 - h_l) = \dot{m}_{st} \times (h_{sh} - h_{fw}) \quad (21)$$

$$\dot{m}_g \times C_p \times (t_{g3} - t_{g4}) = \dot{m}_{st} \times (h_{appr} - h_{fw}) \quad (22)$$

$$\dot{m}_g \times C_p \times (t_{g2} - t_{g3}) = \dot{m}_{st} \times (h_{s2} - h_{appr}) \quad (23)$$

$$\dot{m}_g \times C_p \times (t_{g1} - t_{g2}) = \dot{m}_{st} \times (h_{sh} - h_{s2}) \quad (24)$$

4. Results and discussions

From the data acquired above, it can be found that the modifications to the SRC can improve the overall plant efficiency with recovery almost 16% of the wasted heat from the heat lost out of the engine exhaust gases, further improvements can be employed but they're still in testing phase and not available on the market. There is other important issue that was discussed earlier in the introduction section which is the reduction in the emission so it can meet the regulation requirements of the IMO and improving the EEDI. The attained new ship EEDI is a measure of the ship energy efficiency (g/t nm) and is calculated by the following formula:

$$\text{EEDI} = \frac{P \times \text{SFC} \times C_f}{\text{DWT} \times V_{ref}} \quad (25)$$

where P is 75% of the rated installed shaft power, SFC is the specific fuel consumption of the engines, C_f is the emission of carbon dioxide rate based on the type of fuel, DWT is the ship deadweight tonnage and V_{ref} is the carrier's speed at design load. The calculated EEDI is based on the regulations which apply to carriers including gas, bulk and general cargo ship. Fig. 6 shows a fitting curve for the EEDI values at different DWT according to the reference values for the ship before applying the WHRS and after our approach to implement the WHRS onboard of the ship. The curve of the EEDI for the ship drastically decreases as the DWT is increased, whereas the values changed from 60 to almost 50 g CO_2/t nm). It can be concluded from the figure that the EEDI was reduced by almost 17% of its initial value at a DWT of 100,000 till reaching DWT of 400,000 (see Fig. 7).

There are several methods for reducing emission in general but the two most common types are either slow steaming or waste heat recovery where the main goal is to reduce the fuel consumption per trip. The first to do is to find the equivalent emission reduction where a new factor called emissions factor has to be applied. The factor has been introduced by the IMO to correspond to the factors used by IPCC in order to ensure a harmonized factor used by different parties and still follow a fixed protocol. To summarize all, before the introduction of the fuel consumption the values are dependent on the distance traveled at certain speed and time, the values for the reduced emission can now be estimated as a factor of the reduced fuel consumption as follows:

$$\text{Reduction} = \frac{\text{Emission Factor} \times \text{Fuel Consumption}}{\text{Transport Work}} \quad (26)$$

From the general equation shown in Eq. (26), a pattern was implemented in the Matlab code to calculate the percentage of emission reduction for carbon dioxide and nitrogen oxide before and after installing the waste heat recovery system. Where the main results to be expected is a declination in the value of emission after installing the waste heat recovery due to the reduction in the fuel consumption which means that the fuel efficiency is increased by inspecting Fig. 8, it can be concluded that there is a significant emission reduction in the environmental pollutants achieved, and of the two pollutants, the greatest reduction achieved was in the NO_x and then followed by CO_2 emission, where the reduction percentage in both the CO_2 & NO_x is 16.88% and 36.28%. However the system was designed at a certain pressure and pinch point temperature difference. The low gas velocity increases risk of soot deposits, however, appropriate pinch point was selected for the turbine to achieve higher steam yield, higher energy efficiency and higher thermal efficiency the pinch point should be as low as possible hence the gas temperature will increase.

As shown in Fig. 9, the potential of superheated steam is very low due to relative low temperature and mass flow rate. When the three engines are under different operating conditions (25, 50, 75, 100)% of the full load condition. As

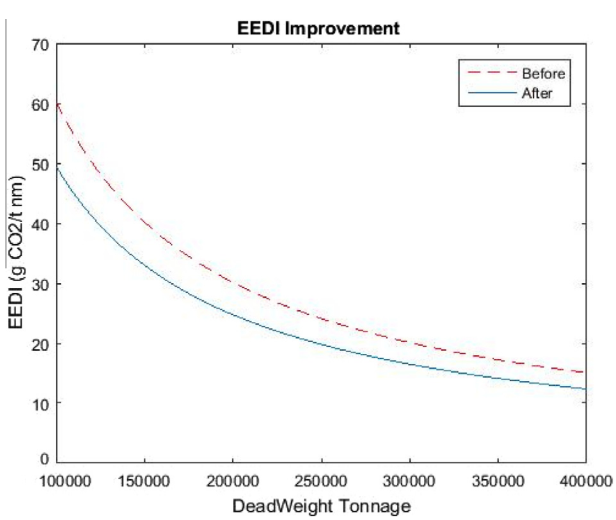

Figure 6 EEDI improvement.

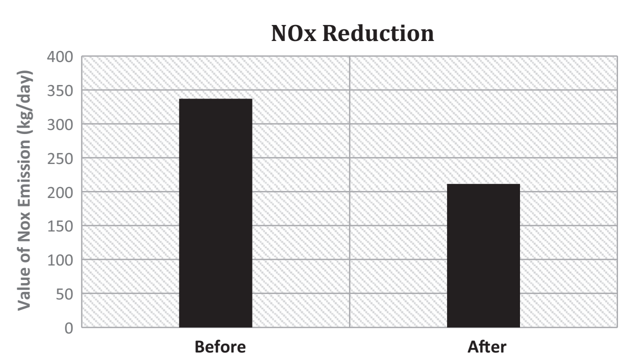

Figure 7 NO_x emission reduction before and after using WHRS.

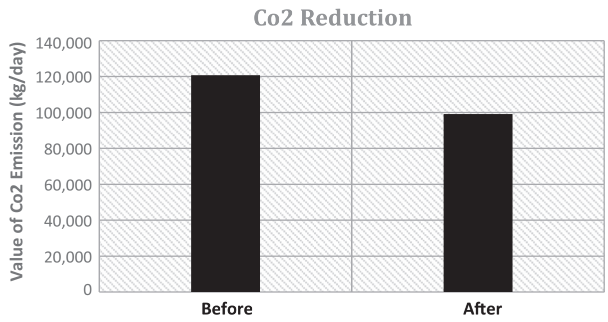

Figure 8 CO_2 emission reduction before and after using WHRS.

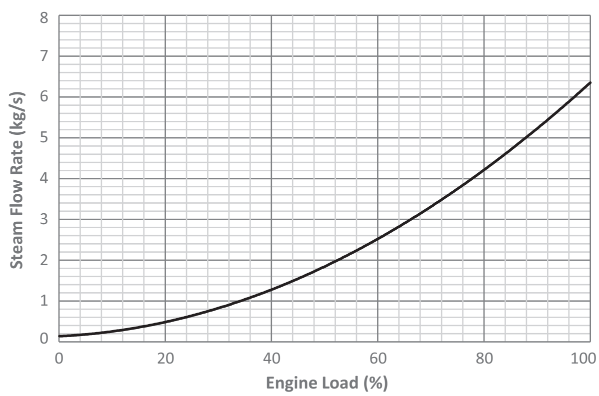

Figure 9 Superheated steam yield function of engine load.

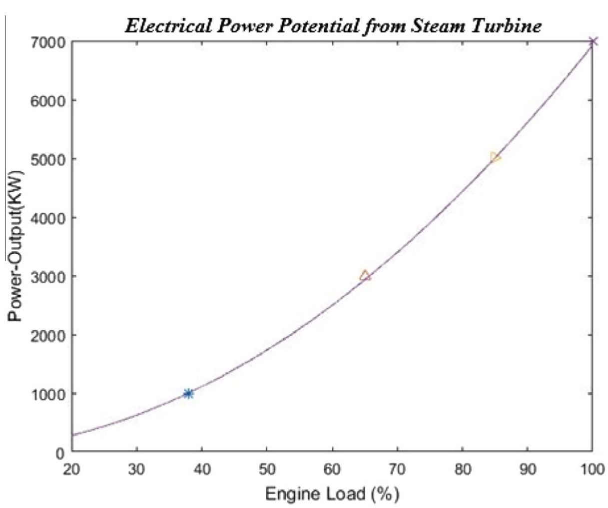

Figure 10 Steam turbine electrical power yield function of engine load.

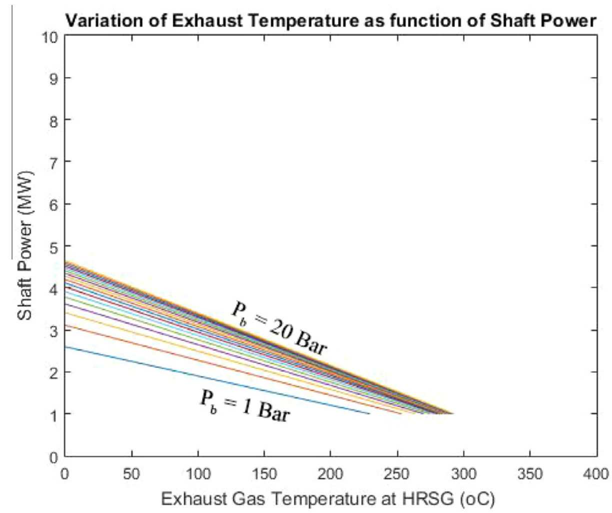

Figure 11 Variation of exhaust temperature function Pshaft [50%] (maximum and minimum boiler pressure 1–20 bar).

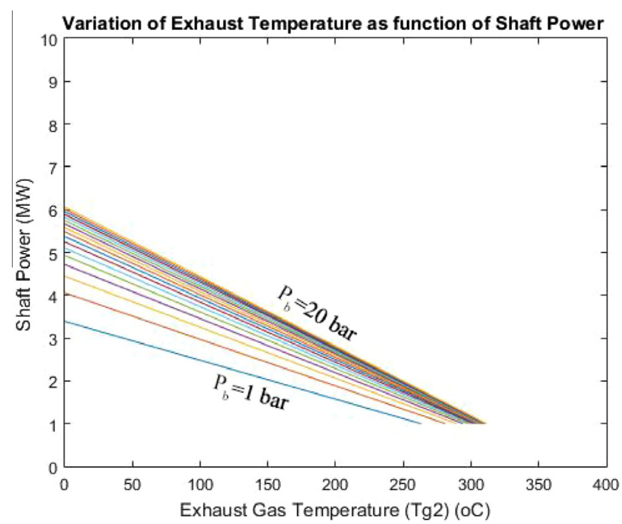

Figure 12 Variation of exhaust temperature function Pshaft [75%] (maximum and minimum boiler pressure 1–20 bar).

the engines load keep on going up the yield increases sharply as the exhaust valve is in fully open condition.

The output power from the steam turbine at the exhaust gas boiler pressure varies with varying the engines load. The results of this variation are shown in Fig. 10 which emphasis the variation in the output power of the power turbine with different engine loads. Therefore, the electrical power potential for the chosen WHRS can be obtained. There is a variation in ratios of electrical power yield at other exhaust gas boiler working pressure to the chosen pressure which is 17 bar with the main engine loads. It is clear that this is the most appropriate pressure to work to achieve the full potential for the full-range of engine loads with our chosen pinch point.

Figs. 11–13 indicate the effect of the exhaust gas temperature variations at the heat recovery steam generator (HRSG) as a function of various input temperature. Several boiler pressures were used while varying the shaft power as the exhaust gas temperature at HRSG increases. At varying load showing the variation of the shaft power output as function of the increasing output temperature thus it can be found that increasing the engine load would result in higher shaft power and thus utilizing that output energy. While running the engine at various loads 50%, 75% and 100%, the results indicated that setting the boiler pressure to 20 bar would increase the overall shaft power by almost 25%, 28% and 31% respectively for the various engine loads. While working at lower pressure

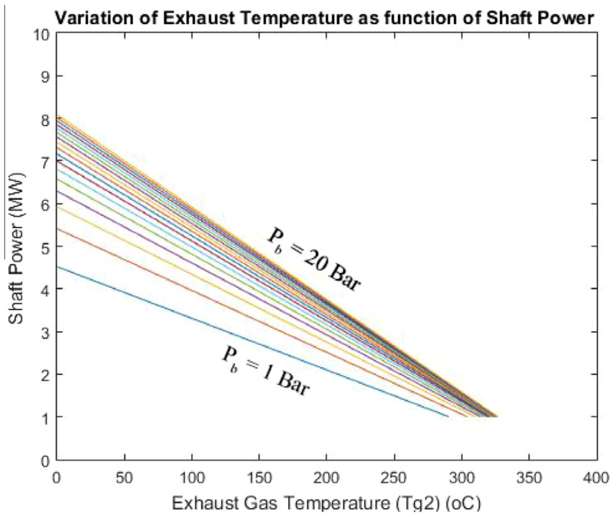

Figure 13 Variation of exhaust temperature function Pshaft [100%] (maximum and minimum boiler pressure 1–20 bar).

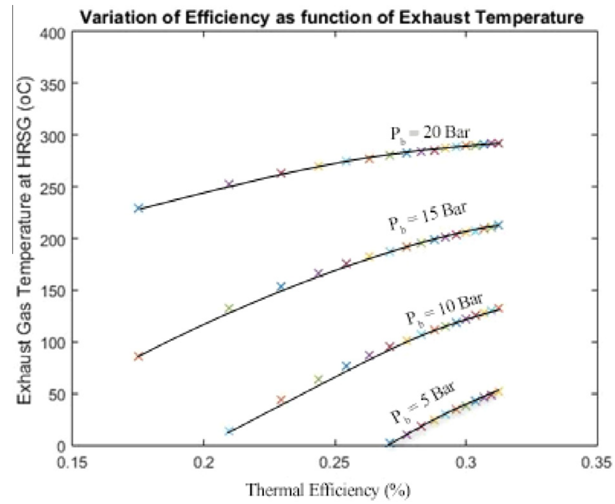

Figure 15 Variation of efficiency as function of T (exhaust) (variable boiler pressure 5–20 bar).

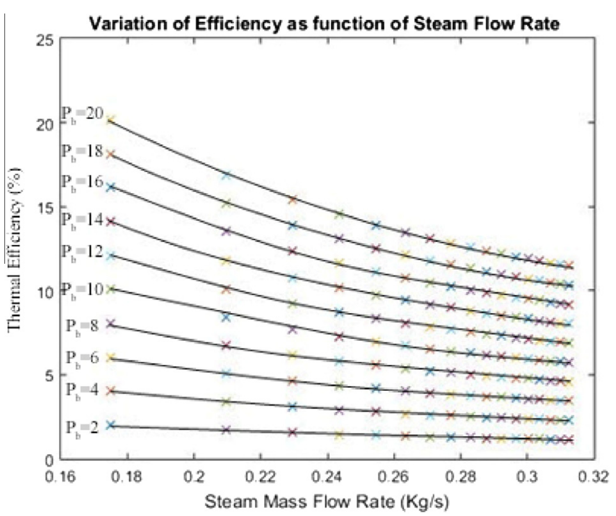

Figure 14 Variation of efficiency as function of steam flow rate (variable boiler pressure 2–20 bar (step 2 bar)).

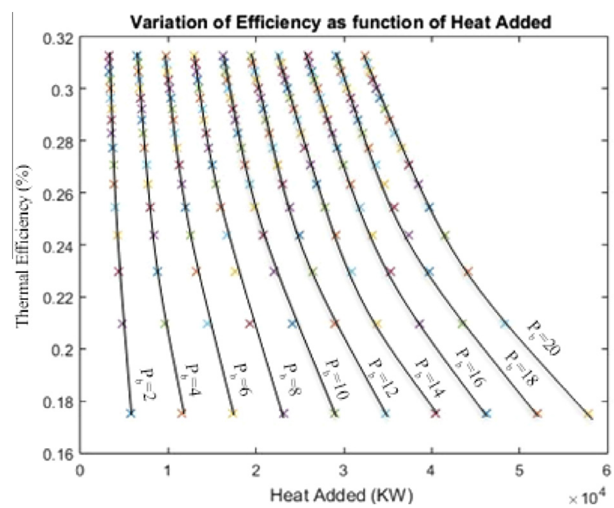

Figure 16 Variation of thermal efficiency as function of heat added ($Q_{exhaust}$) (variable boiler pressure 2–20 bar).

it causes a decrease in the variation of the exhaust temperature and the shaft power. An optimum working condition is desired upon the selection of the type of boiler and plant configuration. The results included in Figs. 11–13 also indicated that working at higher may exceed the limitations of the material and working fluid optimum operation temperature which is dependable on the working fluid either its water or organic.

Conversely, in Fig. 14 there is a variation in Rankine cycle thermal efficiency with the steam flow rate as a function of boiler pressure. The Rankine cycle thermal efficiency is dependable on the steam mass flow rate that passes by the system during a short period, although the variation is from almost 0.18 till almost 0.32 kg/s exceeding this limit may cause a counterproductive effect since the contact time of steam will be less per unit time which may indicate a reduction in the thermal efficiency. The figure indicates a drastic reduction in the thermal efficiency for boiler pressure from 12 to 20 bars where almost 7% reduction can be clearly seen for boiler pressure of 20 bar. While for lower pressures from 2 to 10 bar the change in thermal efficiency is very miniscule which indicated a less desired effect. To sum all up, the steam mass flow rate in the cycle increases with increasing the boiler pressure. Also, the figure shows that increasing the mass flow rate leads to lowering the thermal efficiency of the overall cycle.

Fig. 15 shows the variation of the cycle thermal efficiency as a function of various exhaust gas temperature inlets where increasing of inlet flue gas temperature to the exhaust gas boiler would increase the thermal efficiency. The thermal efficiency increases as the exhaust gas temperature increases for lower pressure but as the boiler pressure limit increases till 20 bar, it can be found that the variation is almost tiny. Fig. 16 shows the variation in thermal efficiency according to the variation of the heat added, where with the increase in the heat added at a constant pressure the thermal efficiency is decreased due to the increase in the work done on the

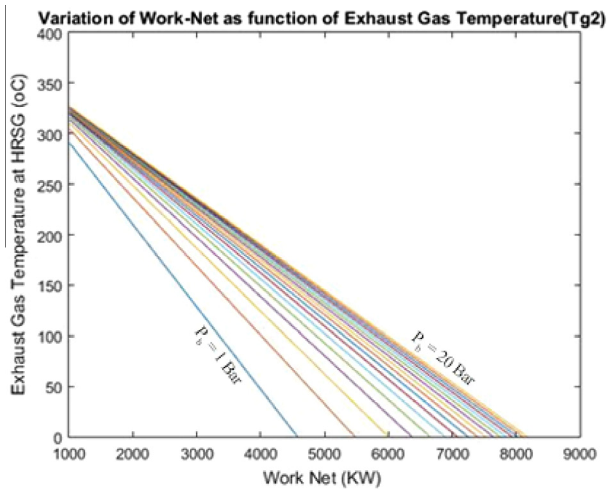

Figure 17 Variation of network as function of exhaust temp (variable boiler pressure 1–20 bar).

system. Conversely, for Fig. 17 the increase in the work net will lead to a decrease in the exhaust temperature which may indicate an increase in the efficiency of the thermal system excluding all the thermal losses that may affect heat transfer characteristics. Fig. 18 shows the relationship of the mass flow rate of the Rankine cycle with the exhaust gas temperature, and normally as the mass flow rate increases the exhaust temperature should decrease which means that all the heat from the exhaust gas has been absorbed by the Rankine cycle which is clearly shown in the figure till reaching the almost 7 kg/s at which the temperature of the exhaust will be near zero which is not true; however, the design conditions are limited with the heat transfer limitation due to material, working fluid and contact time for the fluid and the exhaust gas. Fig. 19 shows the Sankey's diagram before and after the installation of the waste heat recovery system, where almost 56.73% were wasted into the atmosphere in the form of exhaust gases, lubrication and others but with the installation of the proposed system recovered 16.1% of that value which leads to the 40.63% in total is the heat lost due to exhaust gases, lubrication and others.

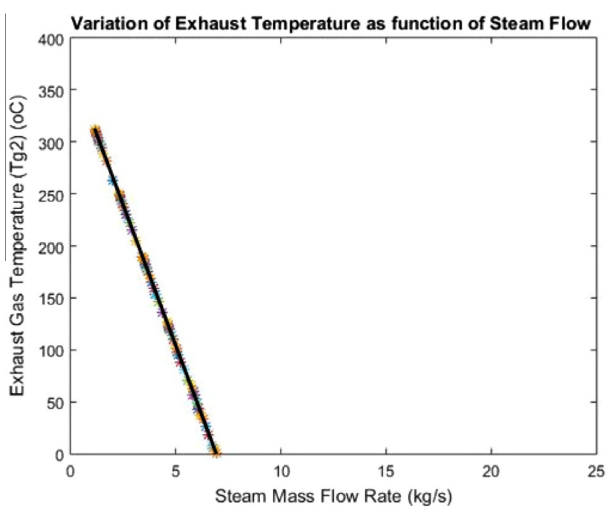

Figure 18 Exhaust temperature function of steam flow.

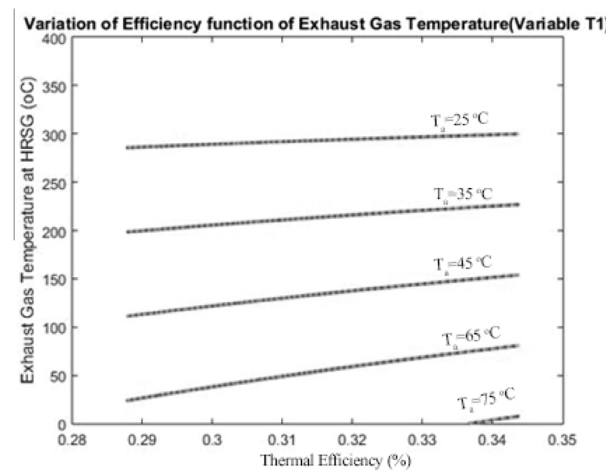

Figure 20 Thermal efficiency as function of Tgas (100% engine load).

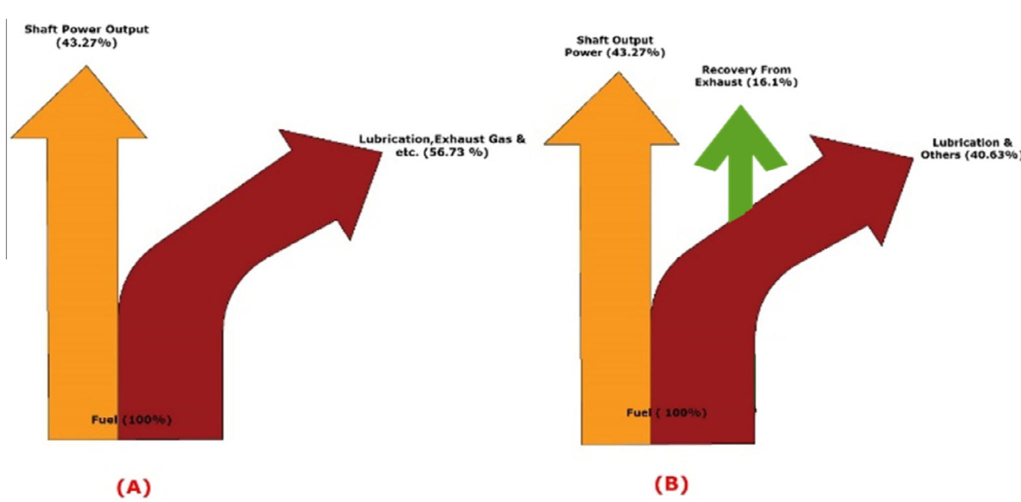

Figure 19 Sankey's diagram before and after improvements.

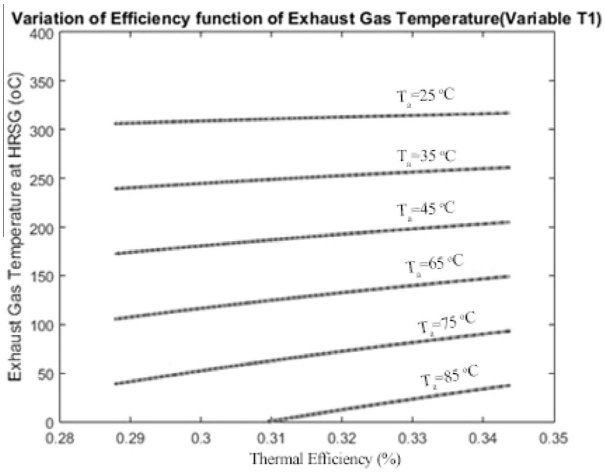

Figure 21 Thermal efficiency as function of Tgas (75% engine load).

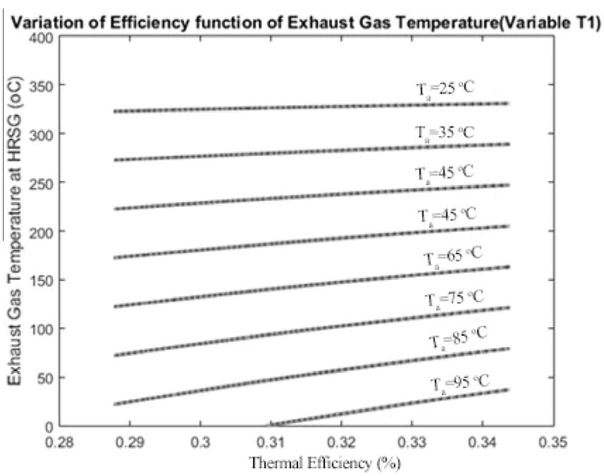

Figure 22 Efficiency as function of Tgas (50% engine load).

Figs. 20–22 indicate the variation of thermal efficiency as function of exhaust gas temperature for temperatures between 25 and 85 °C while maintaining the engine load at 100%, 75% and 50%. The results shown in the three figures indicated a general increase in the thermal efficiency with the decrease in exhaust gas temperature while changing the engine load contributed in limiting the available gas temperature over the HRSG.

5. Conclusions

The switch to LNG as a fuel source involves more than just changes in vessel specifications. It is a long process that requires a significant investment in areas such as infrastructure development to transport and supply LNG fuels. The present study investigated the potential of developing a waste heat recovery system for onboard usage on a LNGC with dual fuel diesel propulsion system. A single source of waste heat which has been exploited is the exhaust gas stream. The proposed system is a single loop Rankine cycle where the main fluid used is water while investigating the potential of using organic fluids as a substitute for water. The WHRS proved to be one of the cheapest and easiest methods to recover some of the energy lost to the ambient after the combustion process is done. There are many promising systems that can be used to reduce the emission which leads to air pollution and also the reduction in the heat loss from the engine which can lead to improving the overall system's efficiency. There is an advantage of using modifications applied to the WHRS but also there is a down side to that which is the high initial cost that leads to more investment required leading to reduction in fuel.

The results show the dependency on the initial parameters that initiate difference in flow of steam and pressure difference such as the pinch point temperature difference and the approach such parameters should be considered when designing a waste heat recovery system to achieve the best possible matching of the components and hence achieve a better overall plant efficiency, fuel reduction and lowering the emissions to the lowest possible values. The proposed system was examined for two main goals which is to comply with the IMO regulations for emission levels for TIER III and to improve the EEDI. The potential of reduction was found for the exhaust to be 36.28% for Nitrogen Oxide and 16.88% for Carbon diode while maintain a 5–10% reduction in EEDI at 400,000 DWT. Furthermore, a saving of 16.1% was achieved by installing the WHRS where there is a potential to increase the percentage of heat recovery by using a complex system i.e. dual loop Rankine cycle up-to 24% of the initial value. It is clear from the above and the results that the overall performance of the studied system reveals it to be a viable technology.

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