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Energy Efficiency and Environmental Analysis of the Green-Hydrogen Fueled Slow Speed Marine Diesel Engine

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Abstract

Marine diesel engines are facing challenges to cope with the emission-reduction regulations set by the international maritime organization (IMO). Hydrogen fuel is one of the alternative fuels which can be used to reduce the exhaust gas emissions from ships. The current paper investigates the effect of using diesel-hydrogen dual fuels on the environmental, energetic and exergetic performance parameters of slow speed marine diesel engine. The investigation is performed using Engineering Equation Solver (EES) software package. As a case study, slow speed diesel engine has been investigated. The results obtained revealed that the energetic and exergetic parameters are influenced by engine load and hydrogen substitution percent. The exergy efficiency is increased by 3.65%, 8.20%, 13.99%, and 21.7% for the hydrogen substitution percentages of 10%, 20%, 30%, and 40%, respectively compared with the diesel engine at full load. Environmentally, CO and CO_2 emissions are reduced and NO_x emissions are increased as the hydrogen energy content increases. Dual fuel engine with input hydrogen energy fractions of 10% and 20% will comply with the required NO_x emission regulations set by IMO after using selective catalytic reduction (SCR) system. It will comply with the required regulations with relative percentages of 96.4% and 98.4%, respectively.

Keywords: *Marine diesel engine, Hydrogen, Energy and exergy analysis, Environmental aspects.*

1. Introduction

The need for alternative marine fuels is one of the most important requirements for marine diesel engines to cope with the stringent emission regulations set by the international maritime organization (IMO) [1-5]. In addition, the fast depletion of fossil fuels with their harmful environmental impacts encourages the need for renewable and clean alternative fuels [6-8]. Hydrogen fuel is one of the promising, renewable, and environmental friendly fuels for the future [9-12]. Hydrogen fuel has wide range of flammability limits, high heating value and stoichiometric air to fuel ratio and low specific gravity compared with diesel fuel as illustrated in Table 1. With these characteristics, hydrogen fuel can be used solely or combined with other fuels to improve the combustion and consequently reduce the exhaust gas emissions [13, 14]. Hydrogen fuel was proposed for spark ignition (SI) engines in many researches since 1970. The main problems associated with these applications are the reduced output power, knocking, and backfire, especially at high engine loads due to the combustion characteristics of hydrogen fuel [15-18]. Thus, hydrogen application in SI engines is limited to well-defined operating conditions [19]. On the other hand, hydrogen fuel is not currently economically feasible compared with fossil fuels due to its high cost and the lake of the infrastructure [15]. In addition to that, the relatively high auto-ignition temperature of the hydrogen fuel recommended it to operate with diesel fuel in dual fuel mode engine in order to reduce exhaust and greenhouse gases emissions [20].

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Table 1 Diesel fuel and hydrogen fuels properties [21-25]

Properties	Diesel fuel	Hydrogen gas
Density at 160 C and 1.01 bar	833-881	0.0838
Molecular weight (g/mole)	100	2
Auto ignition temperature (K)	530	858
Flammability limits (volume % in air)	0.7-5	4-75
Stoichiometric air-fuel ratio on mass basis	14.5	34.3
Net lower heating value (MJ/kg)	41.8	119.93
Flame velocity (cm/s)	30	265-325
Octane number	30	130
Boiling point (K)	436-672	20-27
Specific gravity	0.83	0.091

The performance of the diesel-hydrogen dual fuel engine (DFE) for a single cylinder diesel engine was carried out by [26]. They reported that the brake thermal efficiency increased by 3.2% using 12.5% of the supplied energy by hydrogen gas, at engine full load. The enhanced brake thermal efficiency was a result of the improved combustion efficiency. The performance and emissions of compression ignition (CI) engine fueled with hydrogen enriched diesel are simulated by [27] with numerical simulation using LOTUS Code. They concluded that high air to fuel ratios with 30%-40% hydrogen percentages results in maximizing the engine power. Nguyen *et al.*[28] studied experimentally the effect of hydrogen addition on four stroke diesel engine noise at a compression ratio of 16.7. Results indicated that the noise level is reduced at 10% hydrogen addition (on volumetric basis). The experimental analysis of adding hydrogen (H_2) and liquefied petroleum gas (LPG) on dual fuel four-stroke diesel engine efficiency and pollution was carried out by [29] reported that knocking in dual fuel engine appeared when 50% H_2 [21] and 70% LPG is adding alone. Using 30% H_2 and 40% LPG in dual fuel engine modes, the thermal efficiency increased to 17% and 6%, respectively. Pan *et al.* [20] tested two-stroke marine diesel engine mounted on a dynamometer at three loads specified in the ISO 8178-4 E3 emission test cycle and at idle. The engine is operated with ultra-low sulfur diesel oil with hydrogen added at flow rates of 0, 22 and 220 SLPM. The results show that specific reductions in CO, CO2, unburned hydrocarbon (UHC) emissions with increased NO_x emissions. The performance of DFE for a single cylinder diesel engine using different ranges of hydrogen fractions and injection timing was studied by [30]. They suggested preferable injection-timing methods to reduce smoke and NO_x emissions. Another method reported by [31] to reduce NO_x emission using supercharging in addition to controlling fuel injection timing.

Although the majority of the researches about the application of hydrogen fuel are land based applications, it can also be a good option for marine applications [11, 32]. Veldhuis *et al.* [33] investigated the potential for hydrogen fuel in the marine environment from infrastructure, technical, and economical points of view for the design of high speed catamaran ship. The results show that the vessel is technically feasible and it depends on the availability of hydrogen transport chain and cost. Slow speed diesel engines are the major source for exhaust gas emissions from ships [34, 35]. Therefore, the aim of the current paper is to investigate the effect of using hydrogen gas in marine diesel engines, in dual mode of operation, on the energy and the exergy points of view. As a case study, slow speed marine diesel engine of a container vessel is investigated. In addition, the environmental analysis is included for the dual fuel engine and the NO_x emission rates are compared with the permissible IMO limits.

2. Case study

Table 2 Shop test result for the Al Dhail container ship main engine (Hyundai LTD)

Engine power (kW)	9405	18810	22572	28215	32617	33858	37620
Load (%)	25	50	60	75	86.7	90	100
Speed (rpm)	45.4	57.1	60.7	65.4	68.7	69.5	72.0
$P_{max.}$ (bar)	96	129	144	163	185	186	184
$P_{comp.}$ (bar)	61.4	88.4	100.9	122.3	144.8	145.5	147.7
SFC (g/kWh)	189.6	175	171.7	169.2	169.2	169.5	171.5
$\dot{m}_{air}$ (kg/s)	22.25	23.98	25.92	28.87	30.56	30.88	31.39
$\dot{m}_{jcw}$ (kg/s)	18.48	31.1	36.31	44	49.8	51.44	56.78
$\dot{m}_{liuboil}$ (kg/s)	30.76	56.54	66.56	81.94	94.69	98.47	110.6
$\dot{m}_{exh}$ (kg/s)	22.74	24.89	27	30.2	32.09	32.47	33.18
$T_{exh, turb,in}$ (K)	547	694	718	759	802	815	850
$T_{exh, turb,out}$ (K)	485	583	603	627	648	655	680

The engine under study is the main propulsion engine for the Al Dhail container ship with IMO number of 9732307. The ship was built in 2016 with a registered port of Marshall Islands. Her deadweight is 149, 360 ton. The main engine type is Hyundai-MAN B&W 9590 ME-C 10.2. Its maximum continuous rating (MCR) power is 37,620 kW at 72 RPM. The engine operates with ultra-low sulfur marine fuel oil (ULSFO) with 0.1% S. Table 2 shows the main shop-test performance parameters of the main engine at part load. The data presented in Table 2 is provided by the engine maker company (Hyundai LTD.) in the engine project guide and shop test results. The experimental data provided in the main engine shop tests are accredited by the Det Norske Veritas (DNV) and Germanischer Lloyd (GL) classification societies. The tests performed at ISO conditions and the specific fuel oil consumptions (SFC) are corrected to the Lower Calorific Value (LCV) of 41838.93 kJ/kg.

For the purpose of energy modeling, the relation between main engine performance parameters should be identified using shop test results. For example, Figures 1 and 2 show the specific fuel consumption and exhaust gases temperature correlations at different main engine part loads, respectively. It can be noted that, as the engine load increases, the exhaust gases temperature increase with the reduced specific fuel oil consumption. The correlations are generated using polynomial curve fitting method. They will be used to establish the energy and exergy balance sheets for the main engine. In addition, they are used to study the effect of diesel-hydrogen dual fuels for the main engine from energy, exergy and environmental points of view.

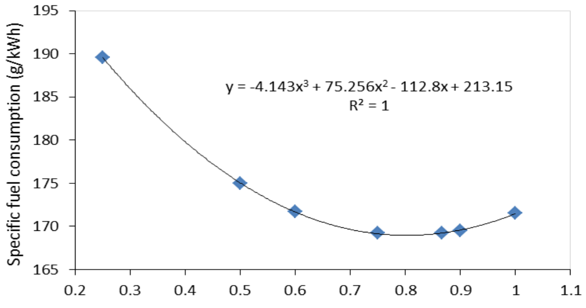

Figure 1 Correlations of the main engines specific fuel consumption

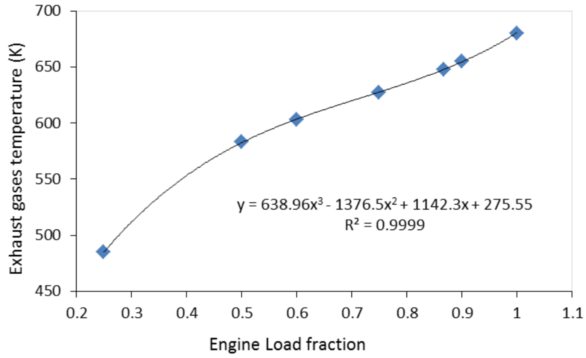

Figure 2 Exhaust gases temperature correlations

2.1 Energy and exergy modeling

Marine diesel engine can be represented by a control volume with inlet air and fuel flows and output useful power in addition to the different waste heat losses as shown in Figure 3. The main assumptions for energy and exergy analysis are steady state condition, ideal gases, and ignored potential and kinetic exergies.

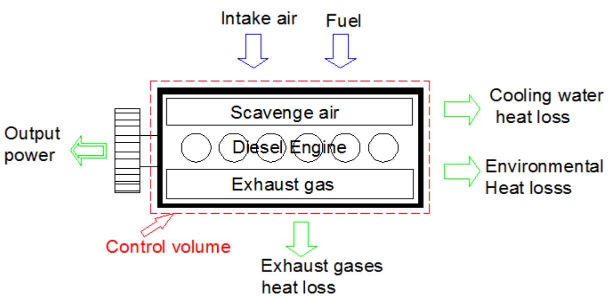

Figure 3 Marine diesel engine control volume showing input and output exergy flows

The brake thermal efficiency (η_{bth}) of the diesel engine control volume can be expressed as:

$$\eta_{bth} = \frac{\dot{Q}_o}{\dot{Q}_f + \dot{Q}_a} \tag{1}$$

where $\dot{Q}_o$ is the output shaft power in kW, $\dot{Q}_f$ and $\dot{Q}_a$ are the energy added by the fuel and air, respectively as expressed in the following Eqs. 2 and 3.

The energy added by the fuel ($\dot{Q}_f$) can be calculated as a function of the fuel mass flow rate ($\dot{m}_f$) in kg/s and the lower heating value of the used fuel (LHV) in kJ/kg as shown in Eq. 2.

$$\dot{Q}_f = \dot{m}_{fd}\, LHV_d + \dot{m}_{fH_2}\, LHV_{H_2} \tag{2}$$

The input air energy ($\dot{Q}_{air}$) is calculated using Eq. 3.

$$\dot{Q}_a = \dot{m}_a C_{p,a}(T_a - T_0) \tag{3}$$

where m_a, $C_{p,a}$, T_a, and T_o are the air mass flow rate in kg/s, specific heat capacity in kJ/kg.K, inlet air temperature, and the dead state temperature (10°C).

$C_{p,a}$ can be calculated as a function of T_a in (K) using the following equation [36].

$$C_{pa} = 1.04841 - 0.0003837 T_a + \frac{9.45378 {T_a}^2}{10^7} - \frac{5.49031 {T_a}^2}{10^{10}} + \frac{7.92981 {T_a}^2}{10^{14}} \tag{4}$$

In addition, the exergy efficiency (Ψ) of the diesel engine can be calculated as follows:

$$\Psi = \frac{\dot{E}_o}{\dot{E}_f + \dot{E}_a} \tag{5}$$

where $\dot{E}_o, \dot{E}_f$ and $\dot{E}_a$ are the exergy flow rates of the output shaft work, fuel, and air, respectively.

Exergy efficiency contributes to the sustainable developments of diesel engine [37, 38], which is directly related to the sustainability index (S_i), as shown in Eq. 6.

$$\Psi = 1 - \frac{1}{S_i} \tag{6}$$

The output shaft work exergy flow rate is equal to the output shaft work (Eq. 7). It is a function of the shaft angular velocity (ω) and the generated torque (T) [39].

$$\dot{E}_o = \dot{Q}_o = \omega T \tag{7}$$

Taking into account the chemical exergy only, the exergy flow rate of the combustion fuel ($\dot{E}_f$) can be calculated using Eq. 8 [40].

$$\dot{E}_f = \dot{m}_f \cdot e_{LHV} \cdot LHV \tag{8}$$

where LHV is the combustion fuel lower heating value. e_{LHV} is the fuel chemical exergy factor. It is a function of the mass fraction of carbon (C), hydrogen (H), oxygen (O), and sulfur (s) in the combustion fuel, as expressed using Eq. 9 [41].

$$e_{LHV} = 1.0401 + 0.1728\left(\frac{H}{C}\right) + 0.0432\left(\frac{O}{C}\right) + 0.2169\left(\frac{S}{C}\right) \times \left(1 - 2.0268\left(\frac{H}{C}\right)\right) \quad (9)$$

Table 3 presents the Stoichiometric air to fuel ratio, density and fuel chemical exergy factor of the simulated dual fuels [39, 42].

Table 3 Fuel properties used in dual marine diesel engine simulation

Fuel type	Stoichiometric A/F ratio	Density (kg/m^3)	CV_{LHV} (kJ/kg)	e_{LHV} (kJ/kg)
Diesel	14.5	800-845	41,838.93	44,582.79
Hydrogen (H_2)	34.33	0.089	119,930	116,648

The inlet air to the diesel engine exergy flow rate ($\dot{E}_a$) can be expressed as follows:

$$\dot{E}_a = \dot{m}_a\left[C_{p,a}\left(T_a - T_0 - T_0 \ln\left(\frac{T_a}{T_0}\right)\right) + RT_0 \ln\left(\frac{P_a}{P_0}\right)\right] \quad (10)$$

where P_a, R are the pressure of the intake air and the gas constant (0.287 kJ/kg K), respectively, P_0 and T_0 are the dead state pressure and temperature (101.315 kPa and10 ^oC).

The heat lost with the exhaust gases can be expressed as a function of the average specific heat capacity ($C_{p,exh}$) and temperature (T_{exh}) of the exhaust hot gases as expressed in Eq. 11 [43, 44].

$$\dot{Q}_{exh} = (\dot{m}_a + \dot{m}_f)C_{p,exh}(T_{exh} - T_0) \quad (11)$$

The exhaust hot gas exergy flow rate is the summation of its chemical ($\dot{E}_{exh,ch}$) and physical exergy ($\dot{E}_{exh,ph}$) flow rates as follows [41]:

$$\dot{E}_{exh} = \dot{E}_{exh,ph} + \dot{E}_{exh,ch} \quad (12)$$

The exhaust hot gas physical exergy flow rate can be calculated using the following equation [41]:

$$\dot{E}_{exh,ph} = (\dot{m}_a + \dot{m}_f)\left[C_{p,exh}\left(T_{exh} - T_0 - T_0 \ln\left(\frac{T_{exh}}{T_0}\right)\right) + RT_0 \ln\left(\frac{P_{exh}}{P_0}\right)\right] \quad (13)$$

where P_{exh} is the exhaust gases pressure and $C_{p,exh}$ is the summation of the mass fraction (Y_i) and specific heat capacity ($C_{p,i}$) of exhaust hot gas components (i) as expressed in Eq. 14.

$$C_{p,exh} = \sum_i Y_i C_{p,i} \quad (14)$$

In addition, the exhaust gases chemical exergy can be calculated using the following equation [45]:

$$\dot{E}_{exh,ch} = (\dot{m}_a + \dot{m}_f)n\left(\sum_i x_i e_i + R_1 T_0 \sum_i x_i \ln(x_i)\right) \quad (15)$$

where n is the number of moles per kilogram of exhaust gases, x_i, e_i are the molar fraction and the standard chemical exergy in kJ/mole of each exhaust gas component, and R_1 is the universal gas constant per mol (0.008314 kJ/mol K). The different values of e_i and C_p of the exhaust gas species are calculated using EES program and the results are validated by the references [36, 46, 47].

The heat rate transferred to the sea water coolers of the diesel engine ($\dot{Q}_{cw}$) can be calculated using Eq. 16 [48]. It is the summation of the engine jacket ($\dot{Q}_{jcw}$), charge (inlet) air ($\dot{Q}_{ch.a,cw}$), lubricating oil ($\dot{Q}_{lub.oil}$) coolers heat.

$$\dot{Q}_{cw} = \dot{Q}_{jcw} + \dot{Q}_{lub.oil} + \dot{Q}_{ch.a,cw} = \dot{m}_w C_w (T_{w,o} - T_{w,i}) \quad (16)$$

where $\dot{m}_w$, C_w, $T_{w,o}$, and $T_{w,i}$ are the mass flow rate, specific heat capacity (4.18 kJ/kg K), outlet and inlet temperatures of the cooling water.

In addition, diesel engine cooling water exergy loss rate ($\dot{E}_{cw}$) can be computed as:

$$\dot{E}_{cw} = \dot{m}_{cw} C_{p,w}\left[T_{w,o} - T_{w,i} - T_0 \ln\left(\frac{T_{w,o}}{T_{w,i}}\right)\right] \quad (17)$$

The heat loss to the environment ($\dot{Q}_{env}$) from diesel engine can be calculated using the difference between total inputs and outputs energies as expressed in the following equation:

$$\dot{Q}_{env} = (\dot{Q}_{add} + \dot{Q}_{air}) - (\dot{Q}_o + \dot{Q}_{exh} + \dot{Q}_{cw}) \quad (18)$$

The exergy loss rate of the heat transfer to the environment can be assessed as a function of the diesel engine working environment temperature (T_{env}) as follows:

$$\dot{E}_{env} = \dot{Q}_{env}\left(1 - \frac{T_0}{T_{env}}\right) \quad (19)$$

Finally, the amount of the destructed exergy due to system irreversibility can be calculated using the exergy balance equation for the diesel engine control volume which can be expressed as:

$$\dot{E}_{dest} = (\dot{E}_a + \dot{E}_f) - (\dot{E}_{exh} + \dot{E}_o + \dot{E}_{cw} + \dot{E}_{env}) \quad (20)$$

From environmental point of view, the emission quantity per trip for the ship ($m_{em,trip}$) can be computed using the following equation:

$$m_{em,trip} = \sum T \cdot P_{i,j} \cdot L_{i,j} \cdot F_{i,j} \quad (21)$$

where, T, P, and L are the engine running time, output power, and load. F is the emission factor for the combusted fuel in kg/kWh, i and j are the pollutant and fuel types.

For dual diesel and hydrogen fuels, the emission factor for the dual hydrogen engine (F_{DHE}) can be calculated using Eq. 22.

$$F_{DHF} = F_{DE} + (X_{H_2} \cdot F_{HE}) \quad (22)$$

where F_{DE} and F_{HE} are the emission factor for the diesel and the hydrogen engines, respectively. X_{H2} is the hydrogen percent in hydrogen-diesel dual fuel engine. The emission factor for slow speed marine diesel engine firing ultra-low sulfur fuel (0.1%S) are 17 g/kWh , 588.79 g/kWh, 1.4 g/kWh, 0.5 g/kWh for nitrogen oxides (NO_x), carbon dioxide (CO_2), carbon monoxide (CO), hydrocarbon (HC) emissions, respectively [49].

3. Results and discussion

In this section, the energy, exergy and environmental results of the diesel-hydrogen dual fuel engine are presented. Energy and exergy results are based on the fraction of each component calculated by dividing its individual value to the air plus fuel input values. In addition to the energy and exergy assumptions included in the section 2.1, the results of the diesel and dual fuel engines are calculated at the same output power. Engineering Equation Solver (EES) software package is used to solve the energy and exergy equations.

3.1 Energy analysis results

The movement of energy flows in and out of marine diesel engines is an important factor in studying its performance. Using these flows, energy balances and efficiency can be easily determined which play an important role in the economic decisions for marine power plants selection. Sankey diagram is one of the visual energy flow balance representing tools. Figure 4 shows the Sankey diagram of the case study ship main engine energy flows. The chemical energy in the supplied marine fuel is converted in diesel engine in to useful output mechanical energy to operate ship propeller and thermal energy losses. In addition, Figure 5 shows the energy flow balance of diesel and dual fuel engines using different hydrogen fuel percentages ranges from 10% to 40%. It can be noted that the brake thermal efficiency increases with the increased hydrogen substitution percent due to the improved combustion efficiency of the mixture hydrogen and diesel fuels.

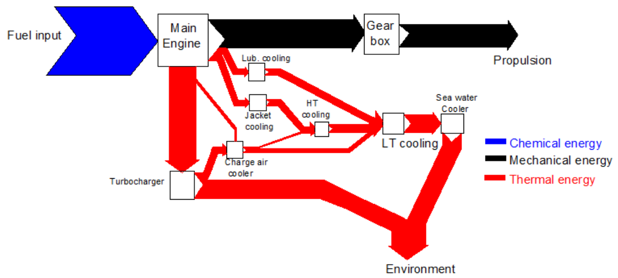

Figure 4 Sankey diagram of Al Dhail container ship main engine energy flows

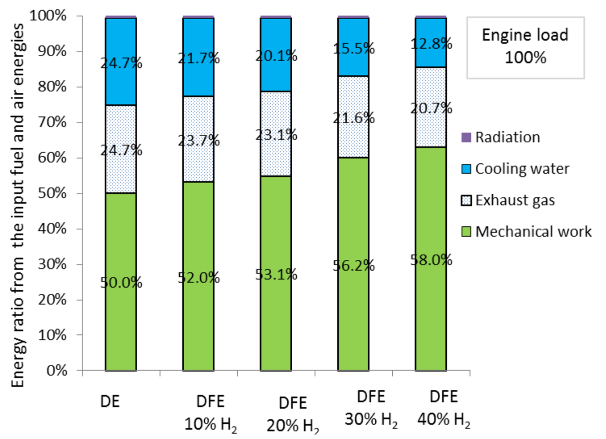

Figure 5 Energy balance of DE and DFE at full load using different hydrogen substitution percentages

The percentages of each energy component of the diesel energy balance diagram will be changed when using dual hydrogen-diesel fuels. Hydrogen substitution percentages in the current study is assumed to be increased up to 40% to prevent engine knocking problems occurred at high hydrogen percentages above 50% [21, 29]. Figure 6 shows the effects of hydrogen fuel substitution percentages and engine load on brake thermal efficiency of the dual fuel engine. Brake thermal efficiency of the marine diesel engine increases with increasing the engine load fraction except at full load (72 rpm). This is because of the improved fuel energy magnitude converted to brake power which subsequently increases the engine brake thermal efficiency, at increased engine load for different hydrogen substitution percentages. However, brake thermal efficiency is reduced at engine full load due to the increased fuel consumption and the inverse relation between fuel consumption and the brake thermal efficiency Eq. 1.

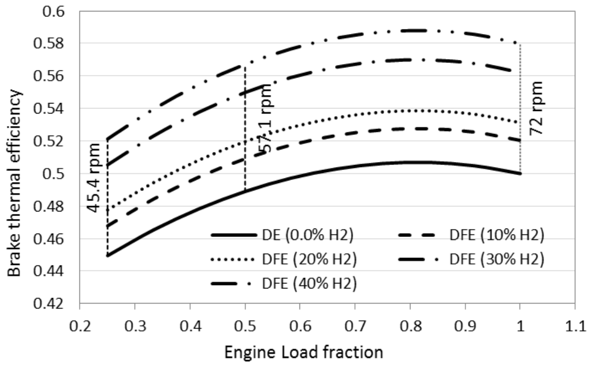

Figure 6 Dual fuel diesel engine brake thermal efficiency at part loads

Exhaust gases heat losses are the major heat loss source from marine diesel engine. Figure 7 shows the effect of substitution of hydrogen gas percentages in dual fuel engines on the exhaust gases temperature at three engine loads. All dual fuel engines with the different hydrogen substitution percentages have higher exhaust gases temperature than diesel engine at all engine loads. This is because of the increased engine efficiency when adding hydrogen gas to the diesel engines as a result of improved combustion efficiency due to favorable hydrogen gas flammable properties as shown in Table 1. In addition, the higher the main engine loads the higher the exit exhaust temperature as a result of the increased fuel consumption at high loads as illustrated in Figure 1.

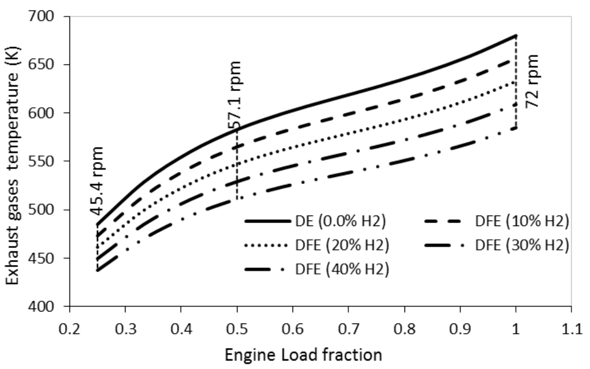

Figure 7 Effect of diesel and dual fuel engines loads on the exhaust gases temperature different hydrogen percentages

3.2 Exergy analysis results

Exergy balance of marine diesel engine helps in improving its performance through identifying the type and magnitude of the available energy fractions in the lost energy flows that can be used by one of the waste heat recovery systems. The exergy balance for the case study engine at full load is shown in Figure 8 using hydrogen substitution percentages ranges from 10% to 40%. The exergy ratio of each component is calculated by dividing each exergy value by the summation of the input fuel and air exergies. Exergy destructed due to the processes irreversibility (like mechanical friction, heat transfer, mixing, and combustion) is the highest fraction of the lost exergies shown in Figure 8. It is reduced from 39.1% for the diesel engine to 37.2%, 34.6%, 31.7%, and 27.6% for dual fuel engine with hydrogen percentages of 10%, 20%, 30%, and 40%, respectively. Exhaust gases and cooling water exergy ratios have the next effective contributions in the lost exergies, respectively. These lost energy sources can be further used to produce a useful energy to increase the exergetic efficiency of the diesel engine. The highest exergy ratio for the studied diesel and dual fuel engines in all cases is the mechanical shaft work. A comparison between Figures 5 and 12 reveals that the input fuel exergy efficiency value is higher than its corresponding energy efficiency value since the quality or the exergy of the fuel is higher than its heating or energy value.

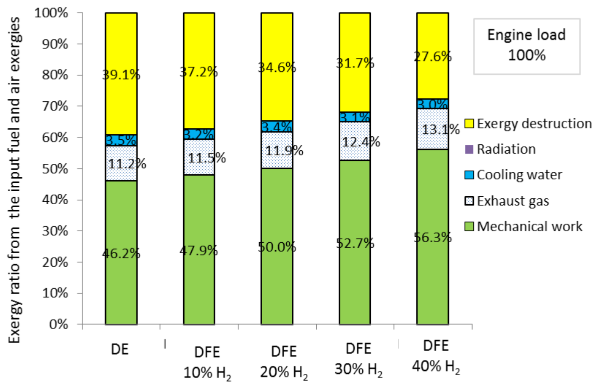

Figure 8 Exergy balance of DE and DFE at full load using different hydrogen substitution percentages

The exergy efficiency of the DFE depends on the engine load and hydrogen substitution percent as shown in Figure 9. Exergy and thermal efficiencies of the diesel engine have the similar trend but with alternative values. As the engine load increases, its exergy efficiency will be improved due to the efficient convergence of the supplied fuel to mechanical shat work (Eq. 5). The engine exergy efficiency is improved by increasing the hydrogen percent in dual hydrogen-diesel engine as a result of the better combustion efficiency of the diesel hydrogen fuels. The exergy efficiency of the DE and DFE ranges from 40% to 56% for the investigated hydrogen percentages from 10% to 40%. At full engine load the exergy efficiency increased from 46.2% to 47.9%, 50%, 52.7%, and 56.2% for the DFE with hydrogen substitution percentages of 10%, 20%, 30%, and 40%, respectively.

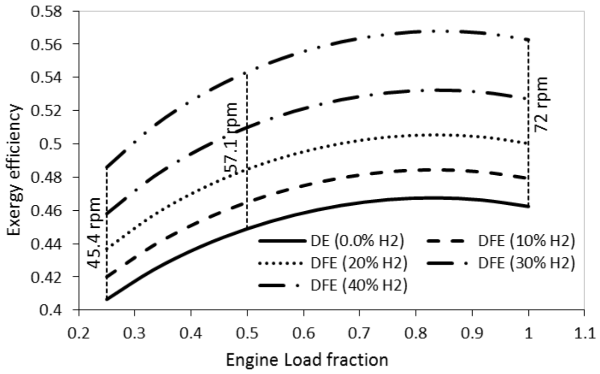

Figure 9 Exergy efficiency of the DE and DFE as a function of hydrogen fuel percent at different engine loads

Exergy efficiency and the sustainability index of the diesel engine are in direct relation as expressed in Eq. 6. It takes into account the effect of the first and the second laws of the thermodynamics to study the renewability and sustainability of the energy conversion in diesel engines. The higher the sustainability index value of the diesel engine, the lower its environmental impact. Fortunately, using dual hydrogen-diesel fuel engine improves the sustainability index values of the diesel engine as shown in Figure 10. The diesel engine sustainability index values range from 1.7 to 1.8 for engine load percentages of 25% and 100%, respectively. These values are ranged from 1.9 to 2.2, respectively using DFE with 40% hydrogen substitution percent.

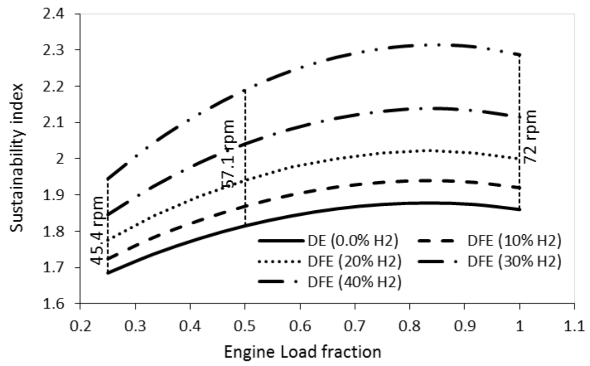

Figure 10 Effect of the DE and the DFE load fractions of the sustainability index

The second highest exergy loss source in diesel engine exergy balance Eq. 21 is the exhaust gases. The effect of DE and DFE load fractions on the exhaust gases exergy efficiency is shown in Figure 11. For the same output power of DE and DFE, the fuel consumption of DFE will be reduced and consequently the exhaust gases mass flow rate will be reduced due to the improved brake thermal efficiency. The load fraction of the DFE will be lower than that of the DE. Therefore, the exhaust gases energy and exergy ratios will be higher for DFE compared with the DE ratios as shown in Figure 11. As the hydrogen substitution percent increases, the DFE load fraction will be decreased compared with DE load and the exhaust gases exergy ratio will be increased. The exhaust gases exergy ratio ranges from 11.2% to 13% for the DE and DFE with 40% hydrogen substitution percent, respectively.

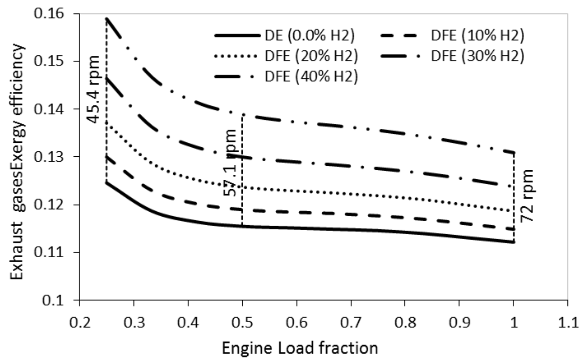

Figure 11 Effect of engine load on the exhaust gases exergy ratio using different hydrogen percentages

3.3 Environmental analysis results

The effects of using different hydrogen energy contents in diesel-hydrogen dual fuel engine are analyzed to evaluate its influence on the exhaust gas pollutants. Hydrogen gas has higher flame speed propagation than that of diesel fuel [50, 51]. Therefore, substitution of diesel fuel with hydrogen gas in DFE will enhance the premixing of the combustible mixtures due to the increased hydrogen to carbon ratio (H/C). This will increase the combustion efficiency. The higher the combustion efficiency, the lower carbon monoxide (CO) and carbon dioxide (CO_2) emissions will be produced which agrees with the published researches [52-55]. Figure 18 shows the relative change in DFE emissions compared with DE at different hydrogen percentages. The specific CO_2 emissions are reduced from 588.8 g/kWhr (DE) to 482.0 g/kWhr, 398.7 g/kWhr, 288.5 g/kWhr, and 224.6 g/kWhr for the DFE with hydrogen percentages of 10%, 20%, 30%, and 40%, respectively. This is equivalent to the reduction percentages of 18.2%, 32.3%, 51.0%, and 61.8%, respectively. In addition, CO emissions will be reduced by 57.3%, 62.6%, 67.0%, 68.3%, respectively. On the other hand, hydrogen gas increases the probability of the complete combustion in DFE. This will lead to higher in-cylinder temperature and peak pressure [56-60]. NO_x emission formation depends on the oxygen concentration, reaction duration and the in-cylindrical temperature rise [55, 61]. As a result of the increased in-cylinder temperature, NO_x emissions will be increased with the hydrogen enrichment percentages and this agrees with the pervious literature [52-54]. Figure 12 shows the increased NO_x emissions from 17.0 g/kWhr to 36.8 g/kWhr, 37.7 g/kWhr, 45.0 g/kWhr, and 55.1 g/kWhr for the DFE with hydrogen percentages of 10%, 20%, 30%, and 40%, respectively. The percentages of this increase are 117%, 121%, 165%, and 223%, respectively compared the pure diesel engine.

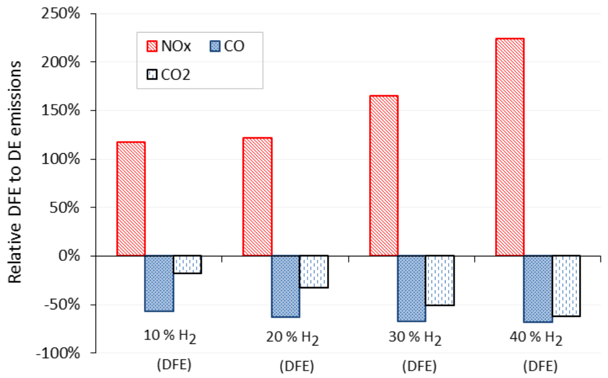

Figure 12 NO_x, CO, and CO_2 emissions from DFE compared with pure diesel engine

From Figure 12, it can be noticed that NO_x emissions will be increased for the DFE. In order to comply with the international maritime organization (IMO) regulations, NO_x emission rates should be compared with IMO Tier III limits for the year 2016 [5, 62, 63]. DE and DFE NO_x emission rates can be calculated using the emission factors for the slow speed diesel engine operated with ultra-low sulfur fuel (0.1%S) [49] and DFE emission rates as a function of hydrogen percentages expressed in Eq. 22. These rates should be checked with the required IMO limits. Figure 13 illustrates the different NO_x emission factors for DE and DFE using different hydrogen percentages. The rate of NOx emissions increased from 9.06 kg/min for the case study DE to 19.66 kg/min, 20.07 kg/min, 24.01 kg/min, and 29.32 kg/min for DFE using input hydrogen energy percentages of 10%, 20%, 30%, and 40%, respectively. These rates reveals that NO_x emission rates for both DE and DFE (10% hydrogen) are higher than that of IMO limits by 344.3% and 864.2%, respectively. Therefore, these rates should be decreased to comply with the required IMO regulation for NO_x emissions Tier III.

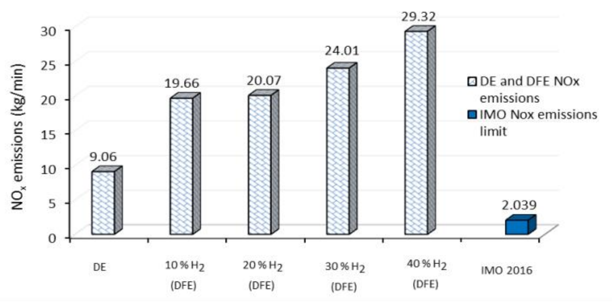

Figure 13 Effect of hydrogen substitution percentages on NO_x emissions compared with IMO limits

Two methods can be used to reduce NO_x emissions from marine diesel engines either using after treatment devices or engine modifications. Engine modifications includes direct water injection, EGR (exhaust gas recirculation), and humid air motors [64]. Selective Catalytic Reduction (SCR) is the most common after-treatment device. It uses urea or ammonia with a specific catalyst to reduce NO_x emissions by 90% [5, 65, 66]. The use of SCR system will reduce DE NO_x emissions to 1.9 kg/min and DFE to 2.007 kg/min, 2.401 kg/min, and 2.932 kg/min using hydrogen percentages of 10%, 20%, 30%, and 40%, respectively. Figure 14 shows the relative NO_x emission reduction percentages compared with IMO Tier III limits. DFE with hydrogen percentages of 10% and 20% would comply with the IMO NO_x regulations with relative percentages of 96.4% and 98.4%, respectively after using SCR system.

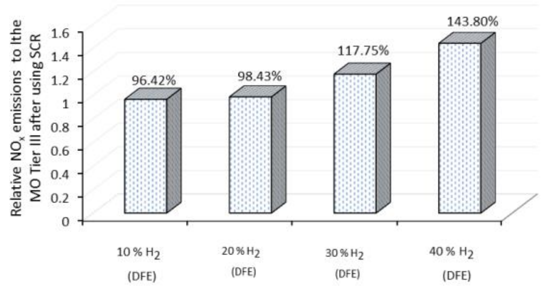

Figure 14 The relative NO_x emission reduction to IMO Tier III using SCR after treatment method

Conclusions

The energetic, exergetic, and environmental analysis of a two stroke marine diesel engine were studied using diesel-hydrogen dual fuels at different engine loads with various input hydrogen fractions. The main conclusions from the case study can be summarized as follows:

- From an energy point of view, the brake thermal efficiency of the dual fuel engine increased from 50% for the pure diesel fuel to 52.0%, 53.1%, 56.2%, and 58.0% using input hydrogen energy fractions of 10%, 20%, 30%, and 40%, respectively. The exhaust gases temperature for the dual fuel engine are reduced due to the improved combustion efficiency of the dual fuels. As the input hydrogen energy fraction increase, the energy lost in the exhaust gases and cooling water decrease as a result of the reduction in the specific fuel consumption and the improved thermal efficiency of the dual fuel engine.
- From an exergy point of view, the exergetic efficiency and sustainability index values are increased by 21.7% and 22.9%, respectively for the dual fuel engine (DFE) with 40% input hydrogen energy fraction compared with pure diesel engine. This increase in exergy is due to the reduced destruction exergy of the diesel engine (DE) from 39% to 27.6% for the DFE with 40% hydrogen, respectively and the increased mechanical shaft exergy from 46.2% for the diesel engine to 56.3% for the DFE with 40% input hydrogen energy. In addition, the exergy ratio of the exhaust gases from the input exergy are increased from 11.2% for the DE to 11.5%, 11.9%,

12.4%, and 13.1% for the DFE with input hydrogen energy percentages of 10%, 20%, 30%, and 40%, respectively. Conversely, the cooling water exergy ratio is reduced from 3.5% for the DE to 3% for the DFE (40% hydrogen).

- From an environmental point of view, the reduction percentages in CO and CO_2 emissions for the DFE, with 40% input hydrogen energy, are 68.3% and 61.8%, respectively. In addition, the percentages of the increase in DFE-NO_x emissions are 117%, 121%, 165%, and 223% for the input hydrogen energy percentages of 10%, 20%, 30%, and 40%, respectively, compared the pure diesel engine. Selective catalytic reduction method can be used to reduce NO_x emissions by 90%. After this reduction, DFE with input hydrogen percentages of 10% and 20% would comply with the required international NO_x regulations with relative percentages of 96.4% and 98.4%, respectively.

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