Comparison between Natural Gas and Diesel Fuel Oil Onboard Gas Turbine Powered Ships

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Abstract: The marine fuel plays a key role in determining the performance of marine power plants onboard ships. Many studies have pointed to the possibility of using natural gas as an alternative fuel for marine power plants. The increase number of LNG carriers worldwide and the growing of its capacity in the last decades, this had given the opportunity to increase the possibility of using natural gas as a main fuel taking into account advantage of evaporation process which occurs during the voyage for energy generation. The current paper analyzes and discusses the change of the performance of marine gas turbine power plant when using natural gas as a main fuel. The study showed that the most significant parameters related to marine gas turbine performance had some (from -1.76% to +0.97%) of those achieved by diesel fuel. Concerning fuel consumption, the paper showed that the specific fuel consumption for natural gas is lower than that of diesel by about 13.5% at the same power output. This is an important factor to push this technology forward, particularly with growing environmental problems that are caused by the conventional marine fuel.

Keywords: Marine Gas turbine, Marine diesel oil, Alternative fuels, Natural gas and gas fuelled ships.

Introduction

The use of marine gas turbines didn't start until the 1970's; the major client for marine gas turbines is the naval units where lots of naval forces
worldwide use them as main or auxiliary power generators for their ships (Woodyard, 2004). Recently, the marine gas turbine started slowly to penetrate the commercial market as auxiliary power units especially in large cruise liners as Queen Mary II, and it is anticipated to penetrate more and more in other commercial sectors as many favorable issues are convincing the ship owners to use gas turbines rather than diesel engines in new built ships. In last decade, the gas turbines found the success in two categories of commercial vessels, both relying on passengers; the fast ferries and the cruise liners. Since the year 2000, many vessels were powered by gas turbines, but most of them use the gas turbine in a combined configuration to achieve a more feasible and flexible propulsion solution due to the high fuel consumption of gas turbines (Woud and Stapersma, 2003). Marine gas turbines are available up to 50 MW but in discrete levels according to the available models. They are provided in an acoustic enclosure with dimensions usually near the standard containers size. They are directly coupled to the propeller through gearboxes but in the recent configurations, electric drives were chosen for the passenger ferries and cruise liners market. Astern motion is unavailable through the engine itself necessitating either a controllable pitch propeller or reversible gearbox and in the electric configurations the electric motor provides the astern motion (Lamb, 2004).

The major disadvantage of gas turbines in the marine applications is their low ability to work on heavy fuels, which is considered more economic than other fuel types, since most of the marine gas turbines are of the aero-derivative types derived from aircraft industry. Only few gas turbines are available now with the ability to burn heavy fuels, those engines are derived from industrial heavy duty gas turbines working basically on heavy fuels in the land based applications. For other marine gas turbines, the traditional fuel used is the Marine Gas Oil (MGO) corresponding to ISO 8217 DMA grade, which has high quality but very expensive cost (Harrington, 1992 and Wright, 2005). This make internal combustion engines more preferred than gas turbine from the view point of economic issue.

To rise above and to move a step forward to the future, new fuels have to be chosen for the fueling of marine gas turbines to have clean and efficient shipping industry (Veldhuis et al., 2005 and El-Gohary, 2006); the natural gas is the candidate for this operation. It is recommended to assess the performance of the gas turbine under this case and compare the
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results with the original diesel case. This will give a clear view of the modifications, if any, that needs to be made to a gas turbine to accommodate this gas fuel efficiently under the marine design conditions. Also it will give a prediction for the thermodynamic characteristics of the engine with the application of the mentioned fuel.

**Marine Fuel Oils and Their Problem**

Fossil fuels are a finite resource and their supply will run out at a certain time. Due to un-constant worldwide consumption in addition to the inaccurate estimate of the size of the actual global reserves, it is difficult to determine the actual time for fossil fuel to run out. But according to the current supply and demand, fossil fuels for shipping will be uneconomic. Economical factors had played the major role in the choice of diesel fuel type matching with the engine of the ship, because more than 50% of the ship's operating expenses, is generally the cost of fuel oil. Most of the world's ship-owners use degraded residue heavy fuel oil in marine power plants, for economy (Bin Lin *et al.*, 2005). The cheaper marine fuel oils, however, contain high levels of asphalt, carbon residues, sulfur and metallic compounds; the previous contents present the main source of air pollution.

So ship and engine designers will have to look for ways of reducing dependence on oil. This objective can be achieved through many ways such as designing ships that continue to carry large cargo at economic speeds, or by dependence on the usage of alternative fuels, the latter is the less economic solution.

The basic criteria for selecting any alternative fuel is that the fuel has to be in abundant supply or, preferably, derived from renewable sources, it should have high specific energy content, easy transportation and storage, minimum environmental pollution and resource depletion, and lastly, it should have good safety and handling properties (Yousufuddin and Mehdi, 2008). The main alternative marine fuel types may be found in two forms: liquid and gaseous fuels.

Handling of the various alternative fuels and recognizing their properties can be summarized through the weight matrix, as shown in Table (1) which may be used to determine the best fuel especially as regards of the environmental and economic issues which are the current fuels problems. The table showed that natural gas (NG) is the best
selection as a marine fuel for marine applications; this is due to its moderate cost, availability and adaptability for existing engines.

Table (1) Matrix of alternative marine fuel (Banawan et al., 2010).

<table>
<thead>
<tr>
<th></th>
<th>Ethanol</th>
<th>Methanol</th>
<th>Liquid Bio Fuel(LBF)</th>
<th>Bio-diesel</th>
<th>Hydrogen</th>
<th>Propane</th>
<th>Natural gas(N G)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Availability</td>
<td>**</td>
<td>**</td>
<td>***</td>
<td>**</td>
<td>***</td>
<td>**</td>
<td>**</td>
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<tr>
<td>Renew ability</td>
<td>**</td>
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<td>*</td>
<td>***</td>
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<tr>
<td>Safety</td>
<td>***</td>
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<td>***</td>
<td>***</td>
<td>-</td>
<td>**</td>
<td>***</td>
</tr>
<tr>
<td>Adaptability</td>
<td>**</td>
<td>**</td>
<td>***</td>
<td>***</td>
<td>**</td>
<td>**</td>
<td>***</td>
</tr>
<tr>
<td>I.M.O Exhaust Gases</td>
<td>*</td>
<td>*</td>
<td>**</td>
<td>*</td>
<td>***</td>
<td>**</td>
<td>***</td>
</tr>
<tr>
<td>Emissions limits</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Performance</td>
<td>*</td>
<td>*</td>
<td>**</td>
<td>**</td>
<td>*</td>
<td>**</td>
<td>***</td>
</tr>
<tr>
<td>Cost</td>
<td>*</td>
<td>*</td>
<td>**</td>
<td>**</td>
<td>-</td>
<td>***</td>
<td>***</td>
</tr>
</tbody>
</table>

*** Excellent ** Very good * Good - Light

Natural Gas in Marine Use

Natural gas as a fuel is well established in the urban transport and power generation sectors; that technology may transfer to the marine industry via availability of engines, systems and technical assistance. The fast increase in natural gas demands as a world energy source makes all, who are interested in energy sources; economics and environmental pollution push this new technology forward, in order to increase the possibility of using it in all applications in a more safe form, especially in the marine field. The safety considerations of using natural gas as main fuel onboard ships are considered as one of the items that affect the shifting from diesel oil to natural gas (Würsig, 2011, DNV, 2007 and IMO, 2009). As natural gas was considered a cleaner fuel, with higher energy content, this made it more suitable for all marine power plants as diesel engines, steam turbines, gas turbines and new power plants such as fuel cell (Banawan et al., 2010).

Where the marine gas turbine has the lowest environmental harm among the traditional marine power plants, the basic benefits of using natural gas as a fuel for marine gas turbine is the economic gains due to the use of less fuel cost than Marine Gas Oil (MGO). But until now the use of natural gas fuelled marine gas turbine is restricted to ship types such as Liquefied Natural Gas (LNG) carriers depending on Boil Of Gas (BOG) system (Kyrkjebø and Seatrans, 2007) and high speed passenger ships working through short voyage routes (Sandker, 2008 and Einang, 2007).
The marine gas turbines manufacturers claim that their new generation designs give ideal propulsion units for LNG carriers due to their improved fuel consumption, well-developed maintenance regimes, low emission levels and the ease of dual-fuel operation, especially when linked to an electric transmission system.

**Gas Turbine Thermodynamic Performance Analysis**

Figure (1) shows the working cycle processes for an ideal Brayton cycle of gas turbine engine on the temperature entropy diagram (T-S).

![Figure 1. Temperature Entropy diagram of ideal Brayton cycle.](image)

The above cycle represents the ideal cycle where no losses are considered. Schematically, the actual cycle of gas turbine engine looks like Fig. (2) where the numbers correspond to the points in the T-S diagram of Figure (1).

![Figure 2. Schematic diagram of a gas turbine engine.](image)

For marine gas turbine engines, a separate free power turbine is introduced to separate the compressor turbine from the torque variations happening due to varying propeller loading (Harrington, 1992 and El-Gohary, 2007). Thus a new point will be introduced in the cycle to represent the air condition between the two turbines. Figure (3) shows the schematic representation of the cycle with double shaft turbine.
Figure (4) shows the cycle in this. Point 4 is now moved up to the pressure line between the two turbines and point 5 is now the exhaust point after the engine.

\[ h_i = C_p i \times T_i \] (1)
The isentropic air enthalpy after the compressor ($h_2^*$) is calculated at the compression pressure ($P_2$) at constant entropy as point (1) at compressor inlet.

$$\eta_c = \frac{(h_2^* - h_1)}{(h_2-h_1)} \quad (2)$$

The pressure after the compressor is the product of the inlet pressure and the compressor pressure ratio ($R$).

$$P_2 = P_1 \times R \quad (3)$$

The compressor specific work can now be calculated by taking the difference between the air enthalpy after and before it.

$$W_C = h_2 - h_1 \quad (4)$$

After the compression process, the air enters the combustor at high pressure and temperature, enough to continue the fuel burning process taking place inside. The amount of heat added to the air from the fuel cannot be determined unless the cycle efficiency and output power are known. So it is more convenient to express this amount with respect to the unit of air mass flow. This is done by dividing the fuel calorific value ($CV$) by the total air-fuel ratio ($AF_T$), which is the product of the fuel stoichiometric ratio ($AF$) and the excess air factor ($\lambda$) (Ibrahim, 1996), thus the fuel calorific value is for unit air mass flow rather than unit fuel mass flow.

$$CV_{air} = \frac{CV}{AF_T} \quad (5)$$

$$AF_T = AF \times \lambda \quad (6)$$

This amount ($CV_{air}$) is added to the air enthalpy after the compressor in the combustor to reach the air condition before the turbine at point (3).

$$h_3 = (\eta_{comb} \times CV_{air}) + h_2 \quad (7)$$

In the previous equation, the combustion efficiency ($\eta_{comb}$) is used to express the real amount of heat added to the flow inside the combustion chamber; which is less than the theoretical amount of heat obtained from the combustion of the fuel flow by 2% due to the incompletely burnt amount of fuel (Cohen et al., 1996).

It is important to note that the air pressure inside the combustor is not constant, a small pressure drop takes place due to air friction with the
combustor walls, this pressure drop is taken to be 5% of the combustor inlet pressure ($P_2$) (Cohen et al., 1996).

$$P_3 = P_2 - \Delta P$$ (8)

The air now enters the compressor turbine which produces enough work for the compressor to continue running.

$$W_C = W_{CT}$$ (9)

$$W_{CT} = h_2 - h_1$$ (10)

Knowing that the work of the compressor turbine is the difference of the air enthalpy across the turbine, thus the air condition at point (4) after the compressor turbine can now be calculated.

$$W_{CT} = h_3 - h_4 = h_2 - h_1$$ (11)

To calculate the air pressure between the turbines at point 4, the isentropic expansion line (3-4*) must be considered to calculate the isentropic enthalpy of air after the first turbine using the turbine isentropic efficiency.

$$\eta_{CT} = \frac{h_3 - h_4}{h_2 - h_1^*}$$ (12)

Knowing that ($h_4^*$) occurs at constant entropy as point (3), the pressure at point (4) can now be known since two properties are now defined at this point; the enthalpy and the entropy.

For the expansion in the power turbine, the turbine exhausts at atmospheric pressure.

$$P_5 = P_1$$ (13)

The isentropic enthalpy at the exit of the power turbine ($h_5^*$); is determined first at the atmospheric pressure and constant entropy as point (4). Then the enthalpy at point (5) is determined using the turbine isentropic efficiency. Note that the two turbines are assumed to have the same isentropic efficiency.

$$\eta_T = \frac{h_4 - h_5}{h_4^*}$$ (14)

The work of the double shaft turbine can now be determined as the difference of air enthalpy across the turbine.

$$W_{PT} = h_4 - h_5$$ (15)
After determining the air condition at the five cycle points, the cycle performance is assessed by calculating the cycle efficiency, the peak temperatures ratio, the work ratio and the mass flow rates of air, fuel and exhaust. In order to know the different flow rates, the output power is obtained according to the power-temperature curve of the gas turbine.

The cycle efficiency ($\eta_{\text{cycle}}$) can be expressed in specific form using the specific work of the power turbine ($W_{\text{PT}}$) and the energy content in the fuel ($CV_{\text{air}}$) as follows,

$$\eta_{\text{cycle}} = \frac{W_{\text{PT}}}{CV_{\text{air}}}$$  \hspace{1cm} (16)

The peak temperatures ratio ($\varepsilon$) is the ratio between the maximum and minimum temperatures occurring in the cycle, namely the combustion temperature ($T_3$) and the inlet air temperature ($T_1$).

$$\varepsilon = \frac{T_3}{T_1}$$  \hspace{1cm} (17)

The work ratio, which is the ratio between the work of the power turbine and the total work generated from both turbines, is determined as follows,

$$WR = \frac{W_{\text{PT}}}{(W_{\text{PT}} + W_{\text{CT}})}$$  \hspace{1cm} (18)

This value measures the engine’s effectiveness, how much power is the useful power from the sum of power generated inside the engine.

The flow inside the turbines is composed of both air and fuel, but since the study is made using the air as the only working fluid, the flow rate inside the turbine is regarded to be composed of two air flows; the air from the compressor plus a quantity equals to the amount of injected fuel.

$$m'_{\text{air}} = (1 + 1/AF_T) = \frac{\text{Power}}{W_{\text{PT}}}$$  \hspace{1cm} (19)

The fuel mass flow rate is then determined according to the air to fuel ratio.

$$m'_{\text{fuel}} = m'_{\text{air}} / AF_T$$  \hspace{1cm} (20)

The specific fuel consumption is determined by dividing the mass flow rate by the output power.

$$sfc = m'_{\text{fuel}} / \text{Power}$$  \hspace{1cm} (21)
The exhaust mass flow rate is the sum of both flow rates of the air and the fuel.

\[ m_{\text{ex}} = m_{\text{air}} + m_{\text{fuel}} \]  

(22)

Which could be written as:

\[ m_{\text{ex}} = m_{\text{air}} (1 + (1/AF_T)) \]  

(23)

The above thermodynamic model is used for the assessment of using natural gas in order to compared with the original case of using diesel fuel.

**Model Validation**

In order to assess the validity of the thermodynamic model, an existing gas turbine is chosen to check for the errors obtained when comparing the gas turbine real data and the model output. The chosen engine is the LM2500 gas turbine of General Electric which has the following data as shown in Table (2).

<table>
<thead>
<tr>
<th>Table (2) LM2500 main technical data.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Power [kW/shp]</strong></td>
</tr>
<tr>
<td><strong>Sfc [kg/kWh] (MDO)</strong></td>
</tr>
<tr>
<td><strong>Efficiency [%]</strong></td>
</tr>
<tr>
<td><strong>Exhaust flow rate [kg/s]</strong></td>
</tr>
<tr>
<td><strong>Exhaust temperature [C]</strong></td>
</tr>
<tr>
<td><strong>Fuel LHV [kJ/kg]</strong></td>
</tr>
<tr>
<td><strong>Pressure ratio</strong></td>
</tr>
<tr>
<td><strong>Module weight [ton]</strong></td>
</tr>
</tbody>
</table>

After applying the model equations on the LM2500 data, Table (3) compares the results with the actual data of the engine.

<table>
<thead>
<tr>
<th>Table (3) Comparison of calculated and actual values for the LM2500 engine.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Calculated value</strong></td>
</tr>
<tr>
<td>-----------------------</td>
</tr>
<tr>
<td><strong>Thermal efficiency [%]</strong></td>
</tr>
<tr>
<td><strong>Specific fuel consumption [kg/kWh]</strong></td>
</tr>
<tr>
<td><strong>Exhaust flow rate [kg/s]</strong></td>
</tr>
<tr>
<td><strong>Exhaust temperature [C]</strong></td>
</tr>
</tbody>
</table>

The fuel used with the LM2500 is the marine diesel oil (MDO), in order to simplify the calculations, the fuel is approximated to C_{12}H_{26} with calorific value of 42800 kJ/kg and stoichiometric air-fuel ratio of 15.14 with maximum excess air factor of 3.11. According to the above results, the errors are found to be within a range of ±4.4% which is an acceptable value allowing to confidently use the model for the assessment of
replacing the diesel with natural gas. For the natural gas case the fuel is approximated to be 100% methane (CH\textsubscript{4}) with 50000 kJ/kg calorific and 17.39 stoichiometric air fuel ratio with maximum excess air factor of 3.13 (Tomczak, et al., 2002). Table (4) summarizes diesel oil and Natural gas properties.

Table (4) Diesel oil and natural gas properties.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Symbol</th>
<th>Calorific value [kJ/kg]</th>
<th>Stoichiometric air fuel ratio [-]</th>
<th>Max. excess air factor [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel</td>
<td>C\textsubscript{12}H\textsubscript{26}</td>
<td>42,800</td>
<td>15.14</td>
<td>3.11</td>
</tr>
<tr>
<td>Natural gas</td>
<td>CH\textsubscript{4}</td>
<td>50,000</td>
<td>17.39</td>
<td>3.13</td>
</tr>
</tbody>
</table>

**Results and discussion**

The modeling has been done assuming constant power output in both cases of natural gas. In other words, the performance of the engines is assessed on the basis of same power achieved. The software used is the EES (Engineering Equations Solver) where the thermodynamic properties of the substances under study can be easily obtained using the built-in functions and data.

The real engine’s (the LM2500) power curve is shown on Fig. (5), where the variation is assumed to be only due to the air inlet temperature. With this assumption, the type of fuel only affects the amount of heat added to the air stream, the nature of the fuel and the combustion details affect nothing.

The temperatures used in study are in the range between 10°C and 45°C which correspond to normal operating conditions for ships worldwide.

The most important parameters compared are:
- Cycle efficiency
- Specific fuel consumption
- Peak temperature
- Exhaust temperature
- Work ratio
- Air flow rate through compressor
- Combustion gases flow rates through turbines
Cycle Efficiency
It has been found from the study that natural gas gives lower efficiency than the original case of diesel for the same power output as shown in Fig.(6).

Specific Fuel Consumption
The higher calorific value for natural gas if compared to diesel oil reduces the quantity of fuel used to give the same heat output. Therefore it is observed that the specific fuel consumption for natural gas is lower than that of
diesel as shown in Fig. (7). This could be an advantage from the point of view of storage onboard the ships, but the fact is different due to the lower density of natural gas compared to the diesel fuel.

**Peak Temperature**

It was taken into consideration during the study to control the different parameters of the thermodynamic model in cases of natural gas to avoid obtaining higher combustion temperatures than in case of diesel, and this was to keep away from the need to increase the turbine blades cooling as shown in Fig. (8). Therefore, the obtained peak temperature for the natural gas is lower than that obtained in the case of diesel.
Exhaust Temperature

Due to the lower maximum temperatures achieved in the case of both gaseous fuels, the exhaust temperatures follow the same trend with lower temperatures obtained for natural gas as shown in Fig. (9).

Work Ratio

The work ratio is that ratio between the useful works developed inside the engine to the total work developed. Due to the lower efficiency for gaseous fuels, the work ratio also appeared to be inferior to that in the case of diesel as shown in Fig. (10).

Air Mass Flow Rate

By nature, the stoichiometric air to fuel ratio for gaseous fuels is higher than that of diesel. So, the air flow rates for the cases of natural gas is higher than the case of diesel as shown in Fig. (11). Also, the excess air factors used for the gaseous fuels are higher than that of diesel and this introduces another increase in the amount of air entering the engine. This is the main reason for the lower efficiency of gaseous fuel engines, because more air is used to keep the temperature in a reasonable range thus losing a part of the heat generated by the fuel combustion to decrease the maximum temperature. This also means that bigger compressor may be needed to achieve the same required performance for all the cases.
Exhaust Mass Flow Rate

Following the increase in air mass flow rates for natural gas, the exhaust also is increased but by a smaller percentage if compared with the diesel case, this smaller increase is due to the lower fuel flow rates for both gaseous fuels. Value of exhaust mass flow rate can be determined from equation (23).

Table (5) presents a numerical comparison between diesel oil and natural gas as fuels for the mentioned marine gas turbine.
Table (5) Main study results for the two fuels at 15°C ambient temperature.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Diesel</th>
<th>Natural gas</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Efficiency</strong></td>
<td>0.3798</td>
<td>0.3788 (-0.26%)</td>
</tr>
<tr>
<td><strong>Specific fuel consumption [kg/kWh]</strong></td>
<td>0.2169</td>
<td>0.1876 (-13.5%)</td>
</tr>
<tr>
<td><strong>Max. temp. [K]</strong></td>
<td>1485</td>
<td>1474 (-0.74%)</td>
</tr>
<tr>
<td><strong>Exhaust temp. [K]</strong></td>
<td>812</td>
<td>806 (-0.74%)</td>
</tr>
<tr>
<td><strong>Work ratio</strong></td>
<td>0.443</td>
<td>0.4388 (-0.97%)</td>
</tr>
<tr>
<td><strong>Air flow rate [kg/s]</strong></td>
<td>70.84</td>
<td>72.32 (+2.09%)</td>
</tr>
</tbody>
</table>

**Conclusion**

It is found from the study that gaseous fuels give good performance if compared to the diesel fuel. To achieve this performance, the engine’s compressor and turbines need to be modified to accommodate the different flow rates. Although, it is expected that without modifications the performance of gaseous fuels will not be too much lower than the performance of diesel and this is proved by the wide use of natural gas in electric generation stations using gas turbines. On the other hand, the paper showed that one of the main advantages of using natural gas with marine gas turbine is the reduction in the value of specific fuel consumption by 13.5%. To extend the present work, the future work in this field will include the capability of using other marine alternative fuel such as Hydrogen supported by a study of the problems facing this application onboard ships and the comparative study among the various marine fuel oils from the view point of power plant performance.

**Abbreviations**

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>BOG</td>
<td>Boil Of Gas</td>
</tr>
<tr>
<td>LNG</td>
<td>Liquefied Natural</td>
</tr>
<tr>
<td>DNV</td>
<td>Det Norsk Veritas</td>
</tr>
<tr>
<td>MGO</td>
<td>Marine Gas Oil</td>
</tr>
<tr>
<td>IMO</td>
<td>International Maritime</td>
</tr>
<tr>
<td>NG</td>
<td>Natural Gas</td>
</tr>
<tr>
<td>LBF</td>
<td>Liquid Bio Fuel</td>
</tr>
<tr>
<td>AF&lt;sub&gt;T&lt;/sub&gt;</td>
<td>Total air fuel ratio</td>
</tr>
<tr>
<td>C&lt;sub&gt;ex&lt;/sub&gt;</td>
<td>Exhaust heat</td>
</tr>
<tr>
<td>CV&lt;sub&gt;a&lt;/sub&gt;</td>
<td>Energy content in</td>
</tr>
<tr>
<td>Cp</td>
<td>Specific heat</td>
</tr>
<tr>
<td>S</td>
<td>entropy</td>
</tr>
<tr>
<td>T&lt;sub&gt;1&lt;/sub&gt;</td>
<td>Inlet air</td>
</tr>
<tr>
<td>T&lt;sub&gt;3&lt;/sub&gt;</td>
<td>Combustion</td>
</tr>
<tr>
<td>T&lt;sub&gt;a&lt;/sub&gt;</td>
<td>Air Ambient</td>
</tr>
</tbody>
</table>

**Nomenclature**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
<th>Unit</th>
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<tbody>
<tr>
<td>AF&lt;sub&gt;T&lt;/sub&gt;</td>
<td>Total air fuel ratio</td>
<td></td>
</tr>
<tr>
<td>C&lt;sub&gt;ex&lt;/sub&gt;</td>
<td>Exhaust heat</td>
<td></td>
</tr>
<tr>
<td>CV&lt;sub&gt;a&lt;/sub&gt;</td>
<td>Energy content in</td>
<td>kJ/kg</td>
</tr>
<tr>
<td>Cp</td>
<td>Specific heat</td>
<td>kJ/kg</td>
</tr>
<tr>
<td>S</td>
<td>entropy</td>
<td>kJ/</td>
</tr>
<tr>
<td>T&lt;sub&gt;1&lt;/sub&gt;</td>
<td>Inlet air</td>
<td>°K</td>
</tr>
<tr>
<td>T&lt;sub&gt;3&lt;/sub&gt;</td>
<td>Combustion</td>
<td>°K</td>
</tr>
<tr>
<td>T&lt;sub&gt;a&lt;/sub&gt;</td>
<td>Air Ambient</td>
<td>°C</td>
</tr>
</tbody>
</table>
Comparison Between Natural Gas and Diesel Fuel Oil Onboard...

CV | Calorific heat | kJ/kg | W_C | Total compressor | kJ/
h | enthalpy | kJ/kg | W | Compressor work | kJ/
LC | Lower calorific | kJ/kg | W_P | Power turbine | kJ/
m_f | Fuel mass flow | Kg/s | W_R | Work ratio | -
m_a | Air mass flow rate | Kg/s | W_T | Total work | kJ/
m_ex | Exhaust mass flow | Kg/s | η_cy | Cycle efficiency | -
R | Compression | - | - | η | Peak temperatures | -
sfc | Specific fuel | g/kW. | λ | Excess air factor | -

References


مقارنة بين الغاز الطبيعي والوقود الديزل على ظهر السفن التي تعمل بالتربيبات الغازية

محمد مرسي الجوهری (201) ، و إبراهيم صادق صديق (203)

1 قسم الهندسة البحرية ، كلية الدراسات البحرية ، جامعة الملك عبد العزيز، جدة - المملكة العربية السعودية

2 قسم الهندسة البحرية ، كلية الهندسة ، جامعة الإسكندرية

3 قسم تكنولوجيا الهندسة البحرية ، الأكاديمية العربية للعلوم والتكنولوجيا والنقل

المستخلص. يلعب الوقود الدور الرئيسي في تحديد أداء المحركات البحرية على السفن. العديد من الدراسات أشارت إلى إمكانية استخدام الغاز الطبيعي كوقود بديل عن الديزل في المحركات البحرية. ومع تزايد عدد ناقلات الغاز المسال عالمياً وتنامي حمولتها في الفترة الأخيرة لتصبح إلى 530 ألف متر مكعب حمولة صافية أعطى الفرصة لتزايد إمكانية استخدام الغاز الطبيعي على تلك السفن مستفيدة بنسبة التخمير التي تحدث في الشحنات أثناء الرحلة في توليد الطاقة. الورقة البحثية الحالية تحلل وتناقش المقارنة في أداء محركات الترتيبات الغازية البحرية عند استخدام الغاز الطبيعي. أوضحت الدراسة أن معدلات الأداء سجلت بعض التغييرات في حدود (0.97% to 1.67%) عن تلك التي يحققها الوقود الأصلي. كما بنيت الدراسة إمكانية حدوث انخفاض في الاستهلاك النوعي للوقود بمقدار 13.5% عند نفس قيمة القدرة المعتادة وهو ما يعتبر عملاً مهمًا للدفع بتلك التكنولوجيا إلى الأمام وبخاصة في ظل تنامي المشاكل البيئية التي يسببها الوقود التقليدي.