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EXERGEOECONOMIC ANALYSIS OF A DIESEL ENGINE

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Abstract. The method of producing power from diesel engines despite being created a long time is still quite used these days, and pollutants formation including their emission are considered one of the major sources of air pollution. This study reports the effects of the engine load on the specific cost per exergy unit of electricity. Experimental results from literature were analyzed, and using this data, combustion model based on the unburned hydrocarbons was developed. Moreover, an energy, exergy and exergoeconomic balance were carried out. Further results will be shown.

Keywords: diesel engine, pollutant emissions, exergoeconomy, thermal performance, combustion.

1. INTRODUCTION

Diesel engines are part of the axis of world industry, providing high torque, durability and economical fuel usage under a variety of conditions (Özener et al., 2014). These engines dominate sectors such as road and train transport, agricultural, military, construction, mining, maritime, propulsion and stationary electricity production (Morón-Villarreyes et al., 2007). Although the fuel demand of most diesel engines still has to be met by primitive petroleum products. Fossil fuel utilization is unsustainable, and it causes greenhouse gases (Caliskan and Mori, 2017a). As the diesel engine utilization increases air pollution increases. Therefore, it is extremely important that studies be conducted to predict the emissions of this kind of engines.

Internal combustion engines, powered with petroleum-derived fuels, emit a large number of harmful substances having negative, direct and indirect, influence on live organisms. The emissions produced by diesel engines have a serious impact on both environment and human health. The substances produced by internal combustion engines include: significant amounts of unburning hydrocarbons, nitrogen oxides, carbon oxides, carbon dioxides, sulphur oxides, particulates, aldehydes and heavy metals (Botwinska et al., 2017). Such substances are toxic to the natural environment, and they are a serious cause of smog, ground-level ozone, acid rain and also human diseases, such as asthma, coughing, or nausea (Serio et al., 2017). All of these problems have motivated the scientific society to seek for new alternatives to have decreased global warming and pollution formation (Özener et al., 2014).

Thermodynamic analysis is a well-known analysis tool for determining the characteristics of engines. First and second laws of thermodynamics are generally used for analyzing engines. First law of thermodynamics is about energy analysis. Energy analysis alone is not enough to assess the best efficiencies of engines. The analysis of second law of thermodynamics is one of the tools utilized in energy conversion systems to measure its effectiveness (Hoseinpour et al., 2017). Exergy is also known as availability, potential or quality of energy. Exergy analysis is used to understand and to calculate the real efficiencies of engines by determining their losses and destructions. It is based on both first and second laws of thermodynamics. Due to the better comparison of efficiency, the exergy analysis has been made in almost all types of thermodynamic cycles. For example, (Da costa et al., 2012) have been conducting studies related to exergetic analysis of a commercial diesel engine, with the purpose of investigating the performance characteristics when the engine is operated in dual form: natural gas and diesel. Hoseinpour et al., (2017) have presented studies about diesel engine fueled with diesel, biodiesel-diesel blends and gasoline fumigation to evaluate how these different fuels affect the energetics and exergetics balances. Khoobbakht et al., (2017) have analyzed the variation in balances of energy and exergy on a diesel engine fueled with different blends of biodiesel, ethanol and diesel fuel, investigating also operational changes in engine. Pandiyarajan et al., (2011) have used exergy analysis as a tool to measure the quantity and quality of energy extracted from a diesel engine and stored in a combined sensible and latent heat storage system.
Exergy is evaluated with a reference (dead) state. The state of reference should be the environment or immediate surroundings of the system (Caliskan and Mori, 2017a).

The Exergy analysis can be applied using economic principles, called the exergoeconomic approach. Exergoeconomy (thermoeconomics) combines exergy analysis with conventional cost analysis in order to assess and improve the performance of energy systems (Abusoglu and Kanoglu, 2009). Several authors also have developed the economics analysis. Aghbashlo et al., (2017) have used thermoeconomic analysis to evaluate the performance of a DI diesel engine operating with various diesel and biodiesel blends containing different amounts of polymer waste. Caliskan and Mori (2017b) have applied the exergoeconomic analysis to investigate the effects of use of Diesel Oxidation Catalyst (DOC) and Diesel Particulate Filter (DPF) after treatment systems, integrated a diesel engine fueled with biodiesel and diesel fuel. Lee et al., (2018) have been conducting studies evaluating in an exergetic and exergoeconomic way the combination of a solid oxide fuel cell (SOFC) and an internal combustion engine, to determine measures for improving its efficiency and the cost effectiveness. Açıkkalp et al., (2014) have published works using an exergoeconomic analysis in a trigeneration system using a diesel-gas engine to suggest strategies for system optimization from the exergoeconomic parameters. Ahmadi and Dincer (2011) have presented studies about thermodynamic and thermoeconomic analysis in a combined cycle power plant (CCPP) with a supplementary firing system aimed at system optimization. Bolatturk et al., (2015) have published works conducting exergetics and thermoeconomics analysis of Turkey-based Çayırhan thermal power plant. With the support of EES package program, they could calculate the thermal and second law efficiencies and the exergoeconomic factors and determined the highest amounts of exergy losses and the highest amount of exergy loss costs.

The present work aims to develop the exergetic and exergoeconomic analysis of stationary diesel engine with maximum power of 44 kW operating with diesel and biodiesel (B7) for different loads with constant speed. The specific contributions of this paper are as follows:

- To evaluate the heat transfer of diesel engine;
- To develop the chemical species, energy, exergetic and exergoeconomic balance;
- To obtain the effect of engine load on the performance and the specific cost of power;
- To calculate the exergetic efficiency;

2. METHODOLOGY

The experiments were performance by Serio et al., (2017). The diesel engine was stabilized during 10 min under steady-state conditions. The engine speed was constant of 1800 min⁻¹ varying the load from 5 to 30 kW. The data acquisition measured the air and fuel mass flow rate and CO₂, CO, THC and NOₓ emissions. Table 1 shows the engine features.

Table 1. Diesel engine specifications (Serio et al., 2017).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Type or value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model</td>
<td>MWM D229-4</td>
</tr>
<tr>
<td>Nº of cylinders</td>
<td>4</td>
</tr>
<tr>
<td>Nº of strokes</td>
<td>4</td>
</tr>
<tr>
<td>Type of injection</td>
<td>Direct</td>
</tr>
<tr>
<td>Bore x stroke</td>
<td>102 mm x 120 mm</td>
</tr>
<tr>
<td>Total displacement</td>
<td>3.922L</td>
</tr>
<tr>
<td>Firing order</td>
<td>1-3-4-2</td>
</tr>
<tr>
<td>Maximum power at 1800rpm</td>
<td>44 kW</td>
</tr>
<tr>
<td>Aspiration</td>
<td>Natural</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>17:1</td>
</tr>
<tr>
<td>Coolant</td>
<td>Water</td>
</tr>
</tbody>
</table>

The fuel mixture used is 93% commercial diesel oil JIS#2 and 7% biodiesel with ethanol blends. The fuel properties important in this analysis is the lower heat value (LHV) of 43.2 MJ/kg. The properties of fuels are presented in Tab. 2.

Table 2. Fuels specifications (Santos et al., 2017).

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Chemical formula</th>
<th>Molar mass</th>
<th>Relative density at 20°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Commercial diesel</td>
<td>C₅₈H₁₇₇₉₅</td>
<td>136.3 kg/kmol</td>
<td>0.855</td>
</tr>
<tr>
<td>Biodiesel</td>
<td>C₁₈H₃₀O₂</td>
<td>291.8 kg/kmol</td>
<td>0.870</td>
</tr>
</tbody>
</table>
The emissions of exhaust gases change with load. The diesel engine was driving an electrical generator, so the load was measured in electric power. These values are presented in Table 3.

<table>
<thead>
<tr>
<th>Load (kW)</th>
<th>CO₂ (g/kWh)</th>
<th>CO (g/kWh)</th>
<th>THC (g/kWh)</th>
<th>NOₓ (g/kWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>1108.0</td>
<td>12.20</td>
<td>5.490</td>
<td>5.050</td>
</tr>
<tr>
<td>10</td>
<td>755.0</td>
<td>5.26</td>
<td>2.700</td>
<td>4.060</td>
</tr>
<tr>
<td>15</td>
<td>620.0</td>
<td>2.86</td>
<td>1.680</td>
<td>4.000</td>
</tr>
<tr>
<td>20</td>
<td>580.0</td>
<td>2.00</td>
<td>1.230</td>
<td>4.350</td>
</tr>
<tr>
<td>25</td>
<td>565.0</td>
<td>1.42</td>
<td>0.915</td>
<td>4.365</td>
</tr>
<tr>
<td>30</td>
<td>572.5</td>
<td>1.30</td>
<td>0.770</td>
<td>4.225</td>
</tr>
</tbody>
</table>

From the data presented, some analyzes were performed in order to evaluate the performance of system.

2.1. Chemical specie analysis

The combustion reaction was evaluated according to the chemical species. Some reaction model has been proposed by Ahmadi et al., (2011) and Ganjehkaviri et al., (2014). However, these models do not consider the unburned total hydrocarbons (THC). The combustion reaction model proposed in this paper consider THC. The species coefficients can be expressed according to as follows:

\[ x_1C_1H_2O + y_1O_2 + x_2N_2 + y_2H_2O + x_3CO_2 \rightarrow y_1CO_2 + y_2CO + y_3N_2 + y_4O_2 + y_5THC + y_6NO \]  

(1)

Where \( x_i \), \( y_i \) and \( z_i \) is the number of atoms of carbon, hydrogen and oxygen respectively. \( x_i \) and \( y_i \) is the stoichiometric coefficient of each substance.

Normally the combustion reaction is carried out for one fuel mole, however the unburned THC reduces the value to lower than one.

Where \( \lambda \) is:

\[ \lambda = \frac{m_f - m_{THC}}{M_f \cdot \dot{n}_f} \]  

(2)

and \( M_f \) is the molar mass of fuel (kg/kmol) and \( \dot{n}_f \) is the molar flow of fuel.

The parameters below were evaluated according to the chemical species:

\[ \gamma CO_2 = \lambda \cdot x_1 + yCO_2 - yCO \]  

(3)

\[ \gamma N_2 = x_2 - \frac{yNO}{2} \]  

(4)

\[ \gamma H_2O = \frac{y}{2} + yH_2O \]  

(5)

The thermodynamic model proposed is complex to solve due to the difficulty of predicting the pollutant emissions, so the emissions rate of CO, NO and THC had to be used as input data.

2.2. Energy analysis

Energy analysis is associated with the first law of thermodynamics. The first law of thermodynamics is used to determine the heat transfer involved in the analysis of the engine (Da Costa et al., 2012). The first law of thermodynamics for reacting system at steady-state system is given following according to Borgnakke and Sontag (2009).

\[ \dot{Q}_{CV} + \sum_{R} \left( h_f^0 + \Delta h^0 \right)_{in} = \sum_{P} \left( h_f^0 + \Delta h^0 \right)_{out} + \dot{W}_{CV} \]  

(6)
where \( n \) is the number of moles of the combustion reaction; \( h^0 \) is the enthalpy of formation and \( \Delta h \) is the variation of enthalpy of formation. The subscript R and P are reagents and products, respectively. The enthalpy of formation of T-T-diesel is 34661 kJ/kmol according Borgnakke and Sontag (2009).

The thermal efficiency of the diesel engine is determined based on the Low heat value (LHV) by:

\[
\eta_{DE} = \frac{W_{net}}{m_{fuel} \cdot LHV}
\]  

where \( \eta \) is the thermal efficiency, \( W_{net} \) is the net work output, \( m_{fuel} \) is the mass flow rate of fuel, and \( LHV \) is the Low heat value of the fuel.

The total exergy rate is composed of physical exergy rate plus the chemical exergy rate:

\[
E_{exh} = E_{exhph} + E_{exch}
\]

and the physical and chemical exergy rate are evaluated by the following correlations:

\[
E_{exhph} = \sum n_i \cdot (h_i - h_0 - T_i \cdot (s_i - s_0))
\]

\[
E_{exch} = \sum n_i \cdot (s_i \cdot e_{i,ch} - R \cdot T_i \cdot \sum n_j \cdot \ln(y_j))
\]

Where: \( y_i \) is the molar fraction, \( e_{i,ch} \) is the standard molar chemical exergy (bejan et al., 1996), \( i \) th is the exhaust gas component. \( R \) is the universal gas constant (8.314 kJ/kmol.K). The unburned mass of THC is a mixture of gases. Due to the composition of THC is not known, the specific enthalpy of THC was assumed to be methane.

In the ambient conditions (standard reference state), the thermodynamic exergy is zero. Then, the total exergy for a fuel is exactly equal to the chemical exergy (Kumar, 2017). Thus, exergy rate of fuel \( E_{xfuel} \) is reduced to:

\[
E_{xfuel} = m_{fuel} \cdot LHV_{fuel} \cdot \varphi
\]

\[
\varphi = 1.0401 + 0.1728 \cdot \frac{H}{C}
\]

where \( \varphi \) is the chemical exergy factor of the fuel and H and C are the mass ratios of hydrogen and carbon.

Different approaches for formulating efficiencies and auxiliary costing equations have been suggested in the literature. In this work the specific exergy costing approach (SPECO) was used. It works with the fuels and products definition. The two fundamental principles (F and P principles) are formulated according to Lazzaretro and Tsatsaronis (2006). In the view of exergy analysis, destruction exergy rate happens along the whole cycle due to irreversibility (Li, et al., 2016). The exergy destruction rate is calculated considering the exergy rate of exhaust gases and the losses due to heat transfer from coolant:

\[
E_{XD} = E_{XF} - E_{Xloss} - E_{XP}
\]
The main rate of heat loss is the heat transfer to the coolant (water). Its fluid exergy rate is calculated considering the average water temperature:

\[
\dot{E}_{\text{loss}} = \dot{Q}_{\text{out}} \left(1 - \frac{T_{\text{ref}}}{T_{\text{CW}}} \right)
\]

\(T_{\text{ref}}\) is the temperature of the standard reference state \((T_{\text{ref}} = 298.73 \text{ K})\), and \(T_{\text{CW}}\) is the temperature of the cooling fluid leaving the system \((T_{\text{CW}} = 378.73 \text{ K})\) obtained from (Caliskan and Mori, 2017a).

The exergetic efficiency in each component is the ratio between the product exergy and the fuel supplied exergy according to:

\[
\varepsilon = \frac{\dot{E}_{\text{p}}}{\dot{E}_{\text{f}}}
\]

**2.4. Exergoeconomic analysis**

The thermoeconomic analysis combines the exergetic analysis with economy. The exergetic cost rate per exergy unit of each product is calculated in order to understand the cost formation process.

The analysis reveals which equipment should be possible and economically feasible for the improvement of the system. The basic thermoeconomic equation is:

\[
c_{\text{p}} \cdot \dot{E}_{\text{p}} + c_{\text{loss}} \cdot \dot{E}_{\text{loss}} = c_{\text{f}} \cdot \dot{E}_{\text{f}} + \dot{Z}
\]

\(c_{\text{p}}\) and \(c_{\text{f}}\) represent the average costs per exergy unit of product and fuel, respectively. And \(c_{\text{loss}}\) is the cost involved in heat loss, that represents the power that is wasted. This equation states that the cost rate associated with the product of the system is equal to the sum of cost rate of fuel and capital cost. The purchase cost function of the component \((Z)\) is determined by Aghbashlo et al., (2017). The capital investment of a component \((Z)\) is converted into the cost rate considering the capital recovery factor \((\text{CRF})\), for the calculation was considered that the engine works 8000 hours per year for 20 years:

\[
Z = Z_i \cdot \text{CRF} \cdot \Phi_{O&M}
\]

where \(\Phi_{O&M}\) represents the maintenance factor 1.06.

The cost of the exergy destruction rate is defined as:

\[
\dot{C}_{\text{D}} = c_{\text{f}} \cdot \dot{E}_{\text{D}}
\]

The total cost rate was determined as follows:

\[
\dot{C}_{\text{TOT}} = \dot{Z} + c_{\text{f}} \cdot (\dot{E}_{\text{D}} + \dot{E}_{\text{loss}})
\]

The exergoeconomic factor is defined as:

\[
f = \left(\frac{Z}{Z + c_{\text{f}} \cdot (\dot{E}_{\text{D}} + \dot{E}_{\text{loss}})}\right) \times 100
\]

The relative cost difference was calculated as follows:

\[
\eta_k = \left(\frac{c_{\text{p}} - c_{\text{f}}}{c_{\text{f}}}\right)
\]

As the purpose of the engine is to generate power, all costs associated with ownership and operation must be included in the cost of power. The exergy losses are covered by the supply of additional fuel to the engine. The auxiliary equations of the cost balance are formulated using the fundamental principles (fuels and products principles), as discussed by Lazzaretto and Tsatsaronis (2006), as follows:
Table 4. Auxiliary equations in exergoeconomic analysis.

<table>
<thead>
<tr>
<th>Auxiliary equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>( C_i = c_i \cdot E_i )</td>
</tr>
<tr>
<td>( C_{\text{air}} = 0 )</td>
</tr>
<tr>
<td>( c_{\text{fuel}} = 0.80 \left( \frac{\text{USD}}{\text{kg}} \right) ) (Aghbashlo et al., 2017)</td>
</tr>
<tr>
<td>( c_{\text{exh}} = \frac{C_{\text{air}} + C_{\text{fuel}}}{\dot{E}<em>{\text{air}} + \dot{E}</em>{\text{fuel}}} )</td>
</tr>
<tr>
<td>( c_{\text{power}} = \frac{c_{\text{fuel}} \cdot \dot{E}<em>{\text{fuel}} + \dot{Z} - c</em>{\text{exh}} \cdot \dot{E}<em>{\text{exh}}}{\dot{E}</em>{\text{power}}} )</td>
</tr>
<tr>
<td>( c_{\text{loss}} = c_{\text{fuel}} )</td>
</tr>
</tbody>
</table>

Where \( C_i \) is the cost rate at each stream.

3. RESULTS

In order to validate the model of combustion reaction which consider the unburned hydrocarbon some texts were carried out. The work (Caliskan and Mori, 2017b) contains the chemical formula of the fuel \( (C_{14.01}H_{25.00}) \), the mass flow rate of air and exhaust gases, the air and fuel temperatures and the mass flow emissions. The diesel engine was experimentally analyzed in three loads for diesel JIS #2 and constant speed of 1800 rpm. The input data were the chemical formula of the fuel and the emission rates of CO, THC and NO\(_x\), and the output data was the CO\(_2\) mass flow rate. The results are shown in Tab. 5.

Table 5. Validation of combustion model with the work (Caliskan and Mori, 2017b).

<table>
<thead>
<tr>
<th>Torque (N.m)</th>
<th>( \dot{\omega} ) (kW)</th>
<th>CO(_2) model (g/h)</th>
<th>CO(_2) measured (g/h)</th>
<th>Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>106.1</td>
<td>0.9996</td>
<td>20.00</td>
<td>16505</td>
<td>16455</td>
</tr>
<tr>
<td>198.9</td>
<td>0.9997</td>
<td>37.49</td>
<td>27270</td>
<td>27221</td>
</tr>
<tr>
<td>254.9 full</td>
<td>0.9998</td>
<td>48.06</td>
<td>33188</td>
<td>33146</td>
</tr>
</tbody>
</table>

The error between the model and the measured data of CO\(_2\) is very low. Then it can be concluded that the model of combustion reaction in Eq. (1) can predict the chemical species with low error.

In the present work, the diesel engine was experimentally analyzed in six loads and constant speed of 1800 rpm. The results of CO\(_2\) modeled and its associated error are shown in Tab. 6.

Table 6. Validation of combustion model with the work (Serio et al., 2017).

<table>
<thead>
<tr>
<th>Load (%)</th>
<th>( \dot{\omega} ) (kW)</th>
<th>CO(_2) model (g/kW.h)</th>
<th>CO(_2) measured (g/kW.h)</th>
<th>Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>11.36</td>
<td>5</td>
<td>1212</td>
<td>1108</td>
<td>8.58</td>
</tr>
<tr>
<td>22.73</td>
<td>10</td>
<td>756.3</td>
<td>755</td>
<td>0.17</td>
</tr>
<tr>
<td>34.09</td>
<td>15</td>
<td>652.5</td>
<td>620</td>
<td>4.98</td>
</tr>
<tr>
<td>45.45</td>
<td>20</td>
<td>583</td>
<td>580</td>
<td>0.52</td>
</tr>
<tr>
<td>56.82</td>
<td>25</td>
<td>554.6</td>
<td>565</td>
<td>-1.88</td>
</tr>
<tr>
<td>68.18</td>
<td>30</td>
<td>546.3</td>
<td>572.5</td>
<td>-4.80</td>
</tr>
</tbody>
</table>

The error to predict the chemical species of CO\(_2\) is high. The lack of the fuel chemical composition in work (Serio et al., 2017) reduced the model accuracy.

The input data of work (Serio et al., 2017) were evaluated. Considering the CO, NO and THC emissions the heat transfer was calculated by Eq. (6) and the thermal efficiency of engine was calculated by Eq. (7). Table 7 shows the ratio between the heat transfer and the power for different loads.
Table 7. Energy balance of diesel engine of maximum power of 44 kW.

<table>
<thead>
<tr>
<th>Load (%)</th>
<th>( \dot{w} ) (kW)</th>
<th>( m_{\text{air}} ) (kg/h)</th>
<th>( m_{\text{fuel}} ) (kg/h)</th>
<th>( T_{\text{exh}} ) (C°)</th>
<th>( Q ) (kW)</th>
<th>( \eta ) (%)</th>
<th>( Q/W ) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>11.36</td>
<td>5</td>
<td>140.3</td>
<td>2.7234</td>
<td>220</td>
<td>10.07</td>
<td>15.30</td>
<td>201.4</td>
</tr>
<tr>
<td>22.73</td>
<td>10</td>
<td>140.4</td>
<td>3.3826</td>
<td>260</td>
<td>10.56</td>
<td>24.64</td>
<td>105.6</td>
</tr>
<tr>
<td>34.09</td>
<td>15</td>
<td>139.5</td>
<td>4.3596</td>
<td>310</td>
<td>11.13</td>
<td>28.67</td>
<td>74.19</td>
</tr>
<tr>
<td>45.45</td>
<td>20</td>
<td>140.0</td>
<td>5.1840</td>
<td>350</td>
<td>12.75</td>
<td>32.14</td>
<td>63.76</td>
</tr>
<tr>
<td>56.82</td>
<td>25</td>
<td>141.6</td>
<td>6.1560</td>
<td>405</td>
<td>14.55</td>
<td>33.83</td>
<td>58.2</td>
</tr>
<tr>
<td>68.18</td>
<td>30</td>
<td>141.9</td>
<td>7.2756</td>
<td>437</td>
<td>19.21</td>
<td>34.36</td>
<td>64.02</td>
</tr>
</tbody>
</table>

At low loads the losses by heat transfer are higher than the power, and the load increases, both power and heat transfer increase. However, the increase of power is higher than the heat transfer. Therefore, the losses due to heat transfer are reduced and the thermal efficiency increases. In the load of 56.82% there was a point outside the curve, that occurred due to a measurement error in this load. The maximum power was not achieved, that happened due to high temperature of exhaust gases, which passes through a valve of exhaust gas recirculation (EGR). However, the effect of EGR was not considered in this work.

The thermal efficiency and exergetic efficiency are shown in Fig. 2.

![Figure 2](attachment:image2.png)

Figure 2. Diagram of thermal efficiency and exergetic efficiency versus load percentage

Both efficiencies increase with the load and the maximum efficiency is around the full load. The reason for this is that the engine was designed to work at full load. For larger loads, the increase in power is higher than the increase of exergy destruction, thus the thermal and exergetic efficiencies are better.

Figure 3 shows the effect of load on the total exergy loss rate and the exergy destruction rate.

![Figure 3](attachment:image3.png)

Figure 3. Total exergy loss and exergy destruction rate as affected by load
The total exergy loss is composed of the exergy destruction plus the exergy loss rate and the exergy rate of exhaust gases. The values of the exergy destruction rate ranged from a minimum value of 34.93 kW to a maximum of 77.31 kW. The values of the total exergy loss rate ranged from a minimum value of 39.35 kW to a maximum of 88.47 kW. The values of exergy loss rate and the exergy rate of exhaust gases are not as significant as the exergy destruction rate. As the load increases, the exergy destruction rate and total exergy loss rate increase. However, the exergetic efficiency increases, which occurs due to the increased power output.

The results of the thermoeconomic analysis of the system are shown in Figure 4, which indicate the relation between the fuel cost rate and the load percentage.

![Figure 4. Diagram of fuel cost rate versus load percentage](image)

The fuel cost rate varied from a minimum value of 2.179 US$/h to a maximum value of 5.822 US$/h. The fuel cost rate is proportional to the mass flow rate, due to the load is directly proportional to the mass flow rate of fuel.

Figure 5 shows the effect of load on the electricity cost rate per exergy unit.

![Figure 5. Effect of load on the specific cost of electrical power](image)

The cost rate of electricity per exergy unit was evaluated by Eq. (16), which is the average costs per exergy unit of fuel (cp). It ranged from a maximum of 113.3 US$/MJ to a minimum of 49.78 US$/MJ. This reduction in cost with increasing load occurs due to the fact that with increasing load, the exergetic efficiency increases as showed at Fig 2. In this way, there is more energy being converted from the combustion, increasing the power.

The total cost rate per load is shown in Fig. 6.
The total cost rate was calculated by Eq. (19), which is summarized of cost rate of component and the cost of the total exergy loss. Its values increase with enhancing engine load and varies from a minimum of 2.04 US$/h to a maximum of 5.38 US$/h. As load increases, total cost rate increases, due to the increase of total exergy loss rate. Its behavior is similar to the total exergy loss rate.

Figure 7 shows the effect of load on the exergoeconomic factor. The exergoeconomic factor was calculated according to Eq. (20). Their values ranged between 0.156% and 0.0717%, decreasing with the load. It compares the component costs with the total cost rate composed of exergy destruction and exergy loss. As the load increases, the total cost rate and the exergy destruction increase. The low value of exergoeconomic factor means that the total cost rate is higher than the component cost. The component cost is negligible in relation to the total cost rate.
Figure 8 shows the relation of relative cost difference with load percentage.

![Diagram of relative cost difference with load percentage](image)

Figure 8. Diagram of relative cost difference with load percentage

The relative cost difference was calculated according to Eq. (21). Their values change from 7.302 to 2.647. This variable reveals the relative increase in the average cost per exergy unit between fuel and product of component. It is useful for optimizing a component indicating a potential of improvement. Its value should be minimized. The reduction of the values with the load means that the engine is being optimized. In high loads, the exergetic efficiency is high and specific cost of electricity is low.

4. CONCLUSIONS

The thermodynamic performance of a diesel engine fueled with biodiesel was calculated for different loads through an exergoeconomic analysis. The calculated parameters changed significantly with the load variation, which influenced the final cost of the flows. The lowest unit cost of exergy was calculated as 49.78 USD / kW for the load of 30 kW, where for the same load, the exergy efficiency of the engine was the highest recorded (27.43%). With this study it can be concluded that the production of power from the diesel cycle is quite feasible. A suggestion for future studies is that the environmental impact of the cycle is also calculated and is interesting if this analysis was done for a cycle where the data regarding the fuel used were known in order to minimize the final error.

5. ACKNOWLEDGEMENTS

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6. REFERENCES


6. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.
CERTIFICATE OF PRESENTATION

This is to certify that

ARTHUR BARBOSA

Presented the Article

EXERGOECONOMIC ANALYSIS OF A DIESEL ENGINE

in the 17th Brazilian Congress of Thermal Sciences And Engineering

25 – 28 November 2018, Águas de Lindóia, SP BRAZIL

Erick Franklin and Marcelo Ribeiro
Chairs of ENCIT2018
Prezada coordenação do curso de Engenharia Mecânica da UFRN,

Declaramos para os devidos fins que o trabalho intitulado EXERGOECONOMIC ANALYSIS OF A DIESEL ENGINE, aceito para publicação e apresentação no 17th Brazilian Congress of Thermal Sciences and Engineering – ENCIT 2018, que ocorrerá em Águas de Lindóia – SP entre os dias 25 e 28 de novembro de 2018, deverá ser apresentado por Arthur Marinho Barbosa (Matrícula: 20170008626) aluno regular do curso de Engenharia Mecânica. Autorizamos e afirmamos que não iremos apresentar o trabalho referido.

Eduardo José Cidade Cavalcanti

Prof. Dr. Eduardo José Cidade Cavalcanti

Matheus Seabra Rodrigues Lima

Matheus Seabra R. Lima
Ao(s) treze dia(s) do mês de dezembro do ano de dois mil e dezoito, às nove horas e trinta minutos, na Sala 414 CTEC, neste Campus Universitário, instalou-se a banca examinadora do Trabalho de Conclusão de Curso do(a) aluno(a) ARTHUR MARINHO BARBOSA, matrícula 20170008626, do curso de Engenharia Mecânica. A banca examinadora foi composta pelos seguintes membros: EDUARDO JOSÉ CIDADE CAVALCANTI, orientador; SANDI ITAMAR SCHAFER DE SOUZA, examinador interno; THIAGO CARDOSO DE SOUZA, examinador interno. Deu-se início à abertura dos trabalhos pelo EDUARDO JOSÉ CIDADE CAVALCANTI, que após apresentar os membros da banca examinadora, solicitou a (o) candidato (a) que iniciasse a apresentação do trabalho de conclusão de curso, intitulado “EXERGOECONOMIC ANALYSIS OF A DIESEL ENGINE”, marcando um tempo de trinta minutos para a apresentação. Concluída a exposição, EDUARDO JOSÉ CIDADE CAVALCANTI, orientador, passou a palavra aos examinadores para arguirem o(a) candidato(a); após o que fez suas considerações sobre o trabalho em julgamento; tendo sido APROVADO, o(a) candidato(a), conforme as normas vigentes na Universidade Federal do Rio Grande do Norte. A versão final do trabalho deverá ser entregue à Coordenação do Curso de Engenharia Mecânica, no prazo de 7 dias; contendo as modificações sugeridas pela banca examinadora e constante na folha de correção anexa. Conforme o que regem o Projeto Político Pedagógico do Curso de Engenharia Mecânica da UFRN, o(a) candidato(a) não será o aprovado(a) se não cumprir as exigências acima.

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