

# **Combustion Characteristics, Performance and Emissions from a Diesel Power Generator Fuelled by B7-Ethanol Blends**

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## ABSTRACT

The effects of fuel blends containing 5, 10 and 15 wt. % of anhydrous ethanol in diesel oil with 7% of biodiesel (B7) on performance, emissions and combustion characteristics of a diesel power generator are investigated. The engine was tested with its original configuration, with the fuel blends directly injected into the combustion chamber, and the applied load varied from 5 to 37.5 kW. The results were compared with standard B7 operation, and showed that in-cylinder peak pressure and heat release rate were decreased at low loads and increased at high loads with the use of ethanol. Increasing ethanol concentration caused increased ignition delay, decreased combustion duration and reduced exhaust gas temperature. The use of ethanol decreased carbon dioxide (CO<sub>2</sub>) emissions, up to 8.6% lower than B7. Carbon monoxide (CO), total hydrocarbons (THC) and oxides of nitrogen (NO<sub>x</sub>) emissions showed different behavior, depending on load and ethanol concentration.

**Keywords:** ethanol; diesel engine; combustion; emissions; power generation.

## 1 INTRODUCTION

One of the main challenges faced by internal combustion engines is to operate with reduced fuel consumption and pollutant emissions [1]. For diesel engine exhaust, for instance, the emission of oxides of nitrogen (NO<sub>x</sub>) and particulate matter (PM) is a major pollution problem [2]. Greenhouse gas emissions from the use of fossil fuels by internal combustion engines are responsible for environmental threats, including climate change and air quality degradation. Thus, many efforts have recently been spent to reduce those types of emissions [3]. This is done by modernizing mechanical components, using electronic fuel injection systems and emission control systems, and partially replacing fossil fuels by biofuels.

Ethanol is considered a potential substitute to diesel fuel, as it is renewable, has a high latent heat of vaporization, is an oxygenated fuel, sulfur-free and has high burning rate, with high potential to reduce NO<sub>x</sub> and PM emissions from diesel engines [4,5]. Brazil is a major ethanol producer, and there has been an increase in domestic production of anhydrous and hydrous ethanol [6]. The feasibility of using ethanol in diesel engines benefit companies that use diesel oil as fuel in trucks, agricultural machinery and diesel power generators, through the implementation of ethanol production plants nearby the consumption place. This can reduce the costs of transportation of diesel fuel.

The use of diesel oil-ethanol blends, ethanol fumigation, dual injection, spark ignition conversion and surface ignition are widely employed techniques to use ethanol in diesel engines [7]. Studies showed that ethanol fumigation can replace up to 50% of diesel fuel demand, dual injection up to 90%, and diesel oil-ethanol blends up to 25% [3]. One of the major limitations of using diesel oil-ethanol blends in compression

ignition engines is the low miscibility of ethanol in diesel oil for a large temperature range, due to differences in the chemical structure of these components, which may cause phase separation [8]. Lapuerta et al. [9] reported studies showing that in warm countries the solubility of pure anhydrous ethanol with diesel oil is higher. The separation may occur when the ambient temperature falls below 10°C, with high humidity or water contamination. An evaluation of the miscibility of ethanol in diesel oil showed that the presence of water in the blend favors phase separation. The authors observed that, with increasing temperature, the blends became more stable.

Zhu et al. [4] performed experiments in a 4-cylinder, direct injection diesel engine using ethanol blended with biodiesel and 0%, 5%, 10% and 15% of anhydrous ethanol. The results showed an increase of specific fuel consumption (SFC) and thermal efficiency with the use of ethanol for all tests, compared to diesel oil. A reduction of NO<sub>x</sub> emissions was observed with increasing ethanol concentration. The effect on carbon monoxide (CO) and hydrocarbon (HC) emissions depended on ethanol concentration in the fuel blend, decreasing for the blend with 5% of ethanol. The effect of ethanol in a 4-cylinder diesel engine was also studied by Park et al. [10]. The authors used 10% and 20% of anhydrous ethanol in diesel oil-ethanol blends. The results showed reduction of NO<sub>x</sub> and increase of CO and HC emissions with the use of ethanol. The high latent heat of evaporation of ethanol can reduce the in-cylinder temperature and, consequently, NO<sub>x</sub> emissions.

Hulwan and Joshi [11] studied the use of diesel oil-ethanol-biodiesel blends in a 3-cylinder diesel engine. They reported that, due to the lower cetane number and higher latent heat of evaporation of ethanol, the blends containing this fuel increase ignition delay. The lower density and viscosity of ethanol leads to better fuel atomization and air/fuel mixture, causing a rapid burning and releasing more heat. The results showed

increased SFC and slightly increased thermal efficiency with the increase of ethanol content in the fuel. It was noticed smoke reduction for all engine loads and NO<sub>x</sub> emissions decrease in low loads with the ethanol use. The authors report that several studies using ethanol-diesel oil blends show divergent results of engine performance and emissions, attributed to the different operating conditions and different engines used in each study.

Yilmaz et al. [12] performed studies using diesel oil, biodiesel and ethanol blends in a diesel engine. The experiments were performed varying ethanol concentration from 3% to 25%. The results showed a small increase in engine exhaust temperature and CO emissions with the increase of ethanol concentration. NO<sub>x</sub> emissions were reduced for all loads with the increase of ethanol concentration, and unburned HC emissions was dependent on engine operating conditions and ethanol concentration, increasing for 25% ethanol at low loads and decreasing at high loads. Armas et al. [13] evaluated the emissions of a diesel engine using blends of 90% diesel oil and 10% ethanol. The authors explain that the effects of ethanol on NO<sub>x</sub> emissions do not follow a pattern and are dependent on engine operating conditions, among other factors. The results showed a small increase of NO<sub>x</sub> emissions with the use of ethanol, in comparison with diesel oil. CO emissions were reduced with the use of ethanol, especially at high loads, justified by the improvement of combustion quality through the increase of oxygen presence in the fuel. The results also showed significant reduction in PM emissions and smoke opacity, and little effect on HC and carbon dioxide (CO<sub>2</sub>) emissions.

This work analyze the performance of a 44 kW diesel power generator fuelled by blends of diesel oil and anhydrous ethanol. Tests were performed with blends containing 0%, 5%, 10% and 15% of ethanol in diesel oil, varying engine load. The

ethanol effects in performance, emissions and combustion characteristics are presented next.

## **2 METHODOLOGY**

Experiments were carried out in a production, naturally aspirated, four-stroke, and four-cylinder stationary diesel engine. The engine geometry and injection characteristics were not changed for the tests. The fuel was directly injected by a mechanically controlled system. Table 1 shows the engine main characteristics. Figure 1 shows the main measured parameters and sensor positions. The intake air mass flow rate was measured through an orifice plate, with uncertainty of  $\pm 2.3$  kg/h. Temperature was measured in the fuel tank, ambient air, inlet air, orifice plate inlet, exhaust gas and cooling water using K-type thermocouples and PT-100 sensors. The uncertainty of the measured exhaust gas temperature was  $\pm 7^\circ\text{C}$ ; the remaining temperatures were measured with an uncertainty of  $\pm 2^\circ\text{C}$ .

The inlet air humidity was measured through a thermo-hygrometer with uncertainty of  $\pm 2.5\%$  of reading and the ambient pressure was measured through a Torricelli barometer with resolution of  $\pm 1.3$  kPa. A platform balance was placed under the fuel tank to measure fuel consumption, with uncertainty of  $\pm 0.13$  kg/h. Measurement and monitoring of the electric energy generated were made by an electric transducer, with an uncertainty of  $\pm 1\%$ . Total HC emissions was measured by a heated flame ionization detector (HFID) with  $\pm 1$  ppm of resolution. NO<sub>x</sub> emissions was measured by a heated chemiluminescent analyzer (HCLD) with  $\pm 1$  ppm of resolution. CO and CO<sub>2</sub> were measured by non-dispersive infrared analyzers (NDIR), with resolution of  $\pm 1$  ppm and  $\pm 0.01\%$ , respectively.

A piezoelectric pressure transducer with  $\pm 0.5\%$  of resolution was used to obtain the pressure data the first engine cylinder. A trigger wheel with a magnetic sensor were used to synchronize the pressure data with the engine cycle. The system was set to obtain the pressure data at 0.1 crank angle intervals.

The tests were performed with diesel oil-ethanol blends using anhydrous ethanol (99.8% purity) concentrations, in weight basis, of 0% (B7E0), 5% (B7E5), 10% (B7E10) and 15% (B7E15). The blends were prepared using a digital balance with a resolution of  $\pm 1\text{g}$ . Table 2 shows the values of density, viscosity and water content for each blend. Three tests were conducted for each load applied to the engine, varying from 0 kW, to 37.5 kW, at the constant engine speed of 1800 rev/min. The readings at each load were performed only after the exhaust gas temperature and the coolant temperature were stabilized, to assure that the steady state condition was reached. A fuel filter assembly was used for each blend in order to prevent changes in results by contamination with the fuel used in the previous test. The load power and fuel consumption results were corrected to standard conditions according to ISO 3046-1:2002 standard [14].

The specific fuel consumption is defined as the fuel amount required per unit power [15,16], here calculated for dual fuel operation by:

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where SFC is the specific fuel consumption,  $\dot{m}_{B7}$  is B7 mass flow rate,  $\dot{m}_E$  is ethanol mass flow rate,  $Q_{LHV,B7}$  is the low heating value of B7 (MJ/kg),  $Q_{LHV,E}$  is the low heating value of ethanol (MJ/kg) and  $P$  is the engine output power (kW).

To convert the molar concentration of an exhaust gas component of interest, measured in % or ppm, into specific mass concentration per unit of energy produced, expressed in g/kW.h, the following expression is used:

$$\text{---} \text{---}$$

where  $\text{---}$  is the component composition (g/kW.h),  $\text{---}$  is the component composition (ppm  $\times 10^{-6}$  or %  $\times 10^{-2}$ ),  $\text{---}$  is the intake air mass flow rate (kg/h),  $\text{---}$  is the gas component density (kg/m<sup>3</sup>) and  $\text{---}$  is the exhaust gas density (kg/m<sup>3</sup>).

The heat release analysis allow assessing the effects of ethanol on the combustion process characteristics. The apparent net heat release rate calculated from application of the first law of thermodynamics [15] to the cylinder content is:

$$\text{---} \text{---} \text{---} \text{---} \text{---}$$

where  $\text{---}$  is the apparent net heat release rate (J/°CA),  $\text{---}$  is the ratio of specific heats,  $c_p/c_v$ ,  $p$  is the cylinder pressure (Pa),  $\text{---}$  is the crank angle (°CA) and  $V$  is the cylinder gas volume (m<sup>3</sup>).

The ignition delay is given by the time interval between the start of injection and the start of combustion [14]. The start of the injection is a known fixed value, and the start of combustion was determined from the second order derivative of cylinder pressure [17]. The combustion duration was defined as the crank angle interval between



the beginning of heat release and the accumulation of 95% of the total heat released [2,18].

### **3 RESULTS AND DISCUSSION**

Typical curves of average in-cylinder pressure and heat release rate for different ethanol concentrations are shown in Fig. 2, for engine load of 20 kW. At this load, the peak pressure was higher for B7E5 (3367 kPa), followed by B7E15 (3358 kPa), B7E10 (3350 kPa) and B7E0 (3327 kPa). For higher loads the highest peaks of pressure and heat release rates were found as the ethanol content in the fuel was increased. The air/fuel mixture formed by the fuel blends with ethanol during the premixed phase was responsible to increase the peak pressure relative to B7E0 [19]. As the engine load increases the peak cylinder pressure is increased and occurs away from the top dead center (TDC). The net heat release rate is comprised of a premixed combustion phase, followed by a diffusive combustion phase. The maximum heat release rate shows a growth up to medium loads, but decreases with approximation of the maximum loads. For all other load conditions the trend of in-cylinder pressure curves and net heat release are similar. Zhu et al. [20] reported the same behavior.

For operation with B7E15 at the load of 32.5 kW the engine showed instability. The analysis of the pressure curve collected during this test showed pressure levels below those shown for other fuels and larger ignition delay, 26.8°CA, while, for other fuels, this parameter was between 21.9°CA and 24.3°CA. Park et al. [10] also reported engine instability for operation with diesel oil and ethanol blends without mechanical modifications. The authors attributed the phenomenon to increased ignition delay, and stable engine operate was achieved by delaying the fuel injection timing. The in-

cylinder and heat release rate results here presented shows similar behavior as to those reported by Hulwan et al. [11], Zhu et al. [20], Fang et al. [21] and Mohammadi et al. [22].

Through analysis of the cylinder pressure, the ignition delay was determined, as shown by Fig. 3. Increasing ethanol concentration increased the ignition delay, justified by the lower cetane number of the ethanol blends [10,11,20]. The ignition delay is also influenced by engine load, in-cylinder temperature and pressure. With the increase of engine load there is a tendency of decreased ignition delay, since higher temperatures are obtained [11]. The shortest ignition delay, 21.7°CA, was found for 32.5 kW using B7E0, and the longest, 26.8°CA, was obtained for 37.5 kW using B7E15, being engine operation unstable at this condition. Due to the largest ignition delay, B7E15 fuel presented the highest peak of heat release rate, of 25.3 J/°CA (Fig. 2).

Figure 4 shows the combustion duration as a function of engine load and fuel. There is an increase of combustion duration with increasing engine load due the longer duration of fuel injection and air/fuel mixture formation [2]. The addition of ethanol caused a decrease of the combustion duration. This can be explained by the higher heat release rate in the premixed combustion phase caused by the increased ignition delay. In addition, the higher amount of oxygen in the ethanol blends reduces the pyrolysis process and increases oxidation during combustion, reducing the combustion duration [20].

Figure 5 shows the results for the specific fuel consumption calculated by Eq. (1). The trend of the SFC curve was not altered by the addition of ethanol. For most loads, there was an increase of SFC with increasing ethanol concentration in the blends. The differences in SFC for engine operation with ethanol is due to the lower LHV and density of the fuel blends with increasing ethanol concentration [19,23,24]. The highest

differences of SFC from B7E0 occurred for the engine load of 25 kW, increasing 5% and 8.5% when B7E5 and B7E15 were used as fuel, respectively. For B7E10 the highest increase of SFC was 6.4%, in comparison with B7E0, at the load of 10 kW.

Figure 6 shows the relationship between CO<sub>2</sub> specific emissions and engine load. CO<sub>2</sub> is a natural product of hydrocarbon combustion and has a tendency to increase its volume concentration with increasing fuel consumption and proximity to the stoichiometric air/fuel ratio [15]. The combustion of ethanol blends did not change the trend of CO<sub>2</sub> emission curves. The use of ethanol caused a reduction in specific CO<sub>2</sub> emissions in all operating conditions. In general, the largest reductions, in comparison with B7E0, occurred in the higher loads, being in the ranges from 2.5 to 7.2%, 3.9 to 6.6% and 3.6 to 8.6% for B7E5, B7E10 and B7E15, respectively. The largest reductions occurred for the loads of 27.5 kW (B7E5), 32.5 kW (B7E10) and 35 kW (B7E15). The decrease of the cylinder gas temperature due to the ethanol high latent heat of evaporation inhibits the oxidation of CO, which leads to a reduction of CO<sub>2</sub> emission [25,26]. CO<sub>2</sub> reduction is also obtained by the use of fuels with low carbon-to-hydrogen ratio and high oxygen concentration [27-29], which is the case of the ethanol blends. He et al. [27] also reported reduced CO<sub>2</sub> emissions with the addition of ethanol to diesel oil.

Figure 7 shows the behavior of the specific CO emissions for the diesel oil and ethanol blends. CO formation in engines is strongly connected with combustion quality, and tends to increase with insufficient oxygen and incomplete combustion [15]. Using B7E5 CO emissions were increased up to the engine load of 30 kW, with a maximum of 14.7% at the load of 10 kW, in comparison with B7E0. From 30 kW there is a reduction of CO emissions, with a maximum reduction of 17.3% at the load of 37.5 kW. For B7E10 there was an increase in CO emissions between 5 and 20 kW, reaching 21.3% at the load of 5 kW. For the engine loads of 22.5 to 37.5 kW CO emissions reduced with

increasing engine load, reaching a reduction of 22.7%. B7E15 fuel showed an increase in specific emissions of CO in all load range, except for the load of 35 kW, decreasing 9.8%. For this fuel, the highest increase in CO emission occurred at 5 kW, of 38.7%.

The addition of ethanol to the fuel reduces the exhaust gas temperature, due to the high latent heat of evaporation of ethanol. This results in lower CO oxidation, increasing the emission rate of this component [27]. On the other hand, the higher oxygen concentration in fuel-rich zones of the combustion chamber with the addition of ethanol may lead to more complete combustion and reduce CO formation [20,23]. The results showed that these effects are dependent on engine load and ethanol concentration in the fuel. At high loads, the reduction of CO may be explained by an improvement in the quality of combustion (Fig. 7). At low loads, the cooling effect caused by the high latent heat of evaporation of ethanol is more significant, and produce longer ignition delays [4]. The low combustion temperature and the ignition delay can suppress the CO oxidation process, despite the availability of oxygen [11].

B710 fuel showed the highest range of reduction in CO emission, indicating combustion improvement with the use of ethanol (Fig. 7). In the case of B7E15, the cooling effect of ethanol may have been dominant to increase CO emission. Hulwan and Joshi [11] and Huang et al. [30] also reported CO emission reduction at high loads and increase at low loads, using diesel oil-ethanol blends.

Figure 8 shows total hydrocarbons (THC) specific emissions variation with the different engine loads tested. B7E5 fuel caused an increase in specific THC emissions in comparison with B7E0 between the engine loads of 5 kW and 20 kW, increasing 6.8% to 2.0%, respectively. From the engine load of 22.5 kW THC emissions was reduced for this fuel, reaching a reduction of 12.8% at 37.5 kW. The use of B7E10 increased THC emissions between 5 kW, by 19.1%, and 32.5 kW, by 1.4%. At the engine load of 37.5

kW, THC emissions were reduced by 9.8% for B7E10. B7E15 presented large reductions of THC emissions between 15 kW, decreasing 5.6%, and 35 kW, decreasing 28.0%.

The emission of unburned hydrocarbons is strongly associated to combustion quality. Among the effects of ethanol is the slower vaporization of the fuel, due to the high latent heat of vaporization, which tends to increase THC emissions [10,27]. The results showed that this effect was more significant at low and medium loads (Fig. 8). Yilmaz et al. [12] also presented results indicating an increase in THC emissions at low loads and reduction at high loads. However, at high loads, the blends with ethanol showed lower THC emissions than B7E0 (Fig. 8). Lower emissions of THC can be a consequence of higher fuel burn efficiency. When ethanol is added to the fuel, more oxygen is available in the combustion, improving the burning quality. Huang et al. [30], Shi et al. [31] and Sayin et al. [32], among other authors, showed results with reduced emissions of hydrocarbons by the use of blends of diesel oil and ethanol.

The formation of oxides of nitrogen is strongly dependent on the temperature of the gas inside the cylinder, the oxygen concentration and the residence time of the gases at high temperatures [15]. The ethanol addition to diesel oil causes several effects that act one against another to form these components. The low cetane number and the large amount of oxygen in the ethanol blends cause high peak heat release rate, as shown previously, which can increase the temperature within the cylinder. On the other hand, ethanol high latent heat of evaporation and low flame temperature tend to reduce the gas temperature within the cylinder. The behavior of the NO<sub>x</sub> formation is dependent on the intensity of these effects during combustion, and therefore a function of the engine operation characteristics [30].

Figure 9 shows NO<sub>x</sub> specific emissions variation with engine load and fuel. In general, the fuel blends containing ethanol exhibited reduction or no variation of specific NO<sub>x</sub> emissions for all loads applied. B7E5 and B7E10 fuels presented differences within the uncertainty range, in comparison with B7E0. B7E15 fuel showed significant reductions, with maximum reduction of 10.4% at the engine load of 35.0 kW. Among the blends tested, B7E15 contains the highest ethanol concentration, and operation with a high engine load requires a high fuel amount injected. Thus, the cooling effect of ethanol at this condition inhibited NO<sub>x</sub> formation.

#### **4 CONCLUSIONS**

Ethanol added to B7 increased ignition delay and reduced combustion duration. Pressure and heat release rate were reduced at lower loads and increased at medium and high loads. B7 containing 15% of ethanol increased fuel consumption up to 17.9%, and reduced CO<sub>2</sub> and NO<sub>x</sub> emissions up to 8.6% and 10.4%, respectively. CO emission was increased for low to medium engine loads and reduced up to 22.7% at full load for B7 with 10% of ethanol. THC emissions were generally increased at low loads and reduced at high loads, with a maximum reduction of 28.0% for operation with 15% of ethanol.

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4 – Combustion duration variation with diesel oil-ethanol blends.

Figure 5 – Specific fuel consumption variation with diesel oil-ethanol blends.

Figure 6 – CO<sub>2</sub> specific emissions variation with diesel oil-ethanol blends.

Figure 7 – CO specific emissions variation with diesel oil-ethanol blends.

Figure 8 – THC specific emissions variation with diesel oil-ethanol blends.

Figure 9 – NO<sub>x</sub> specific emissions variation with diesel oil-ethanol blends.

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Table 2 – Properties of diesel oil-ethanol blends.

Table 1 - Diesel engine and generator details.

EQUIPMENT	PARAMETER	TYPE OR VALUE
ENGINE	Cycle	Four strokes
	Diesel oil injection	Direct
	Bore × stroke	102 × 120 mm
	Number of cylinders	4, in line
	Total displacement	3.922 L
	Intake system	Naturally aspirated
	Rated power	44 kW
GENERATOR	Number of poles	4
	Voltage	220 V
	Number of phases	3
	Rated power	55 kVA
	Frequency	60 Hz

Table 2 - Properties of diesel oil-ethanol blends.

PROPERTY	FUEL				METHOD
	B7E0	B7E5	B7E10	B7E15	
DENSITY @ 20°C (kg/m <sup>3</sup> )	842 ± 1	839 ± 1	836 ± 1	833 ± 1	ASTM D4052
VISCOSITY @ 40°C (mm <sup>2</sup> /s)	3.0 ± 0.1	2.6 ± 0.1	2.4 ± 0.1	2.3 ± 0.1	ASTM D7042
WATER CONTENT (mg/kg)	103 ± 2	256 ± 4	357 ± 6	427 ± 7	ASTM E203

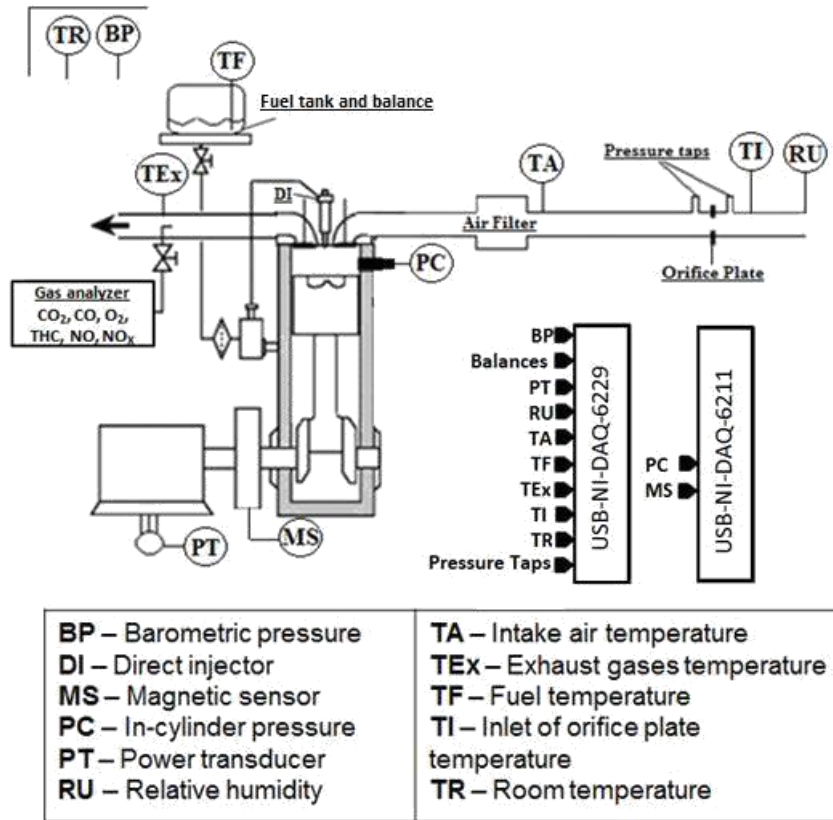


Figure 1 – Schematics of the experimental apparatus.

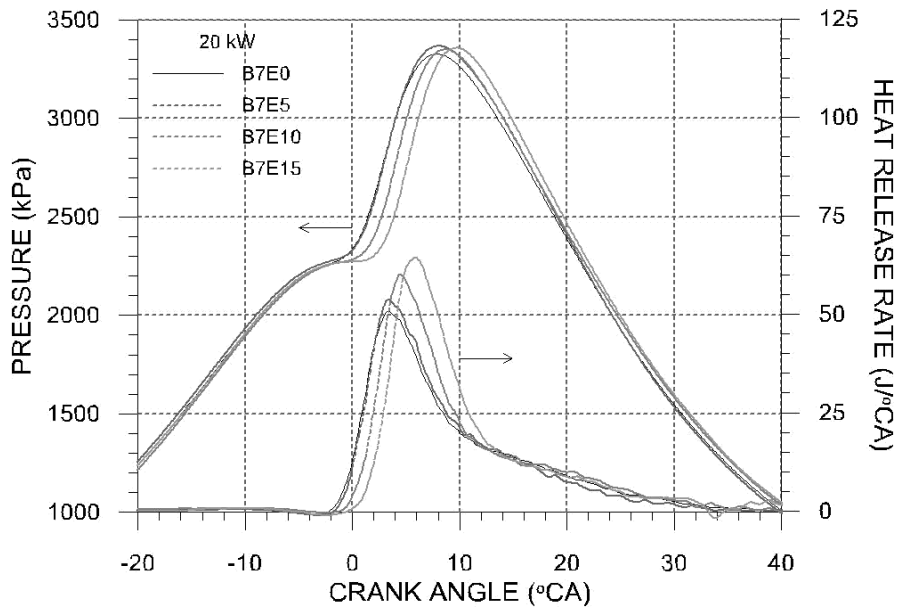


Figure 2 – Typical curves of in-cylinder pressure and heat release rate.



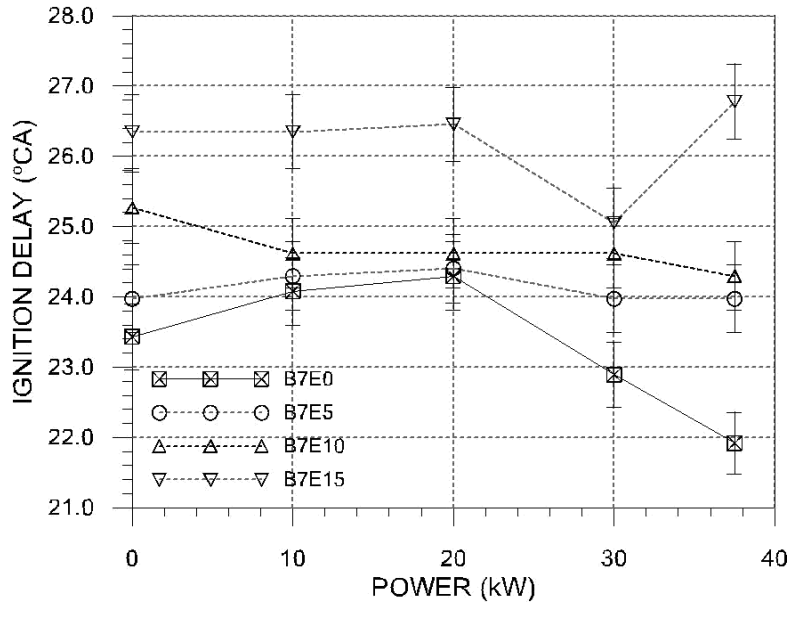


Figure 3 – Ignition delay variation with diesel oil-ethanol blends.

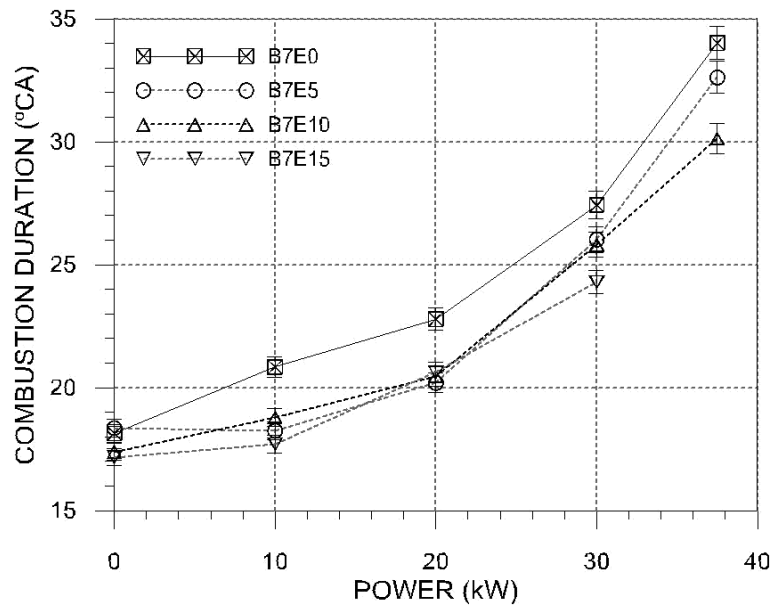


Figure 4 – Combustion duration variation with diesel oil-ethanol blends.

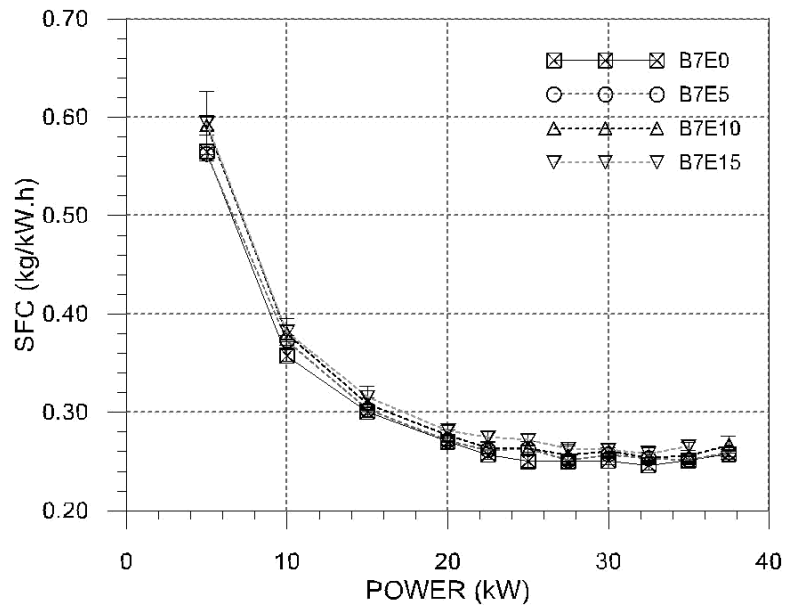


Figure 5 – Specific fuel consumption variation with diesel oil-ethanol blends.

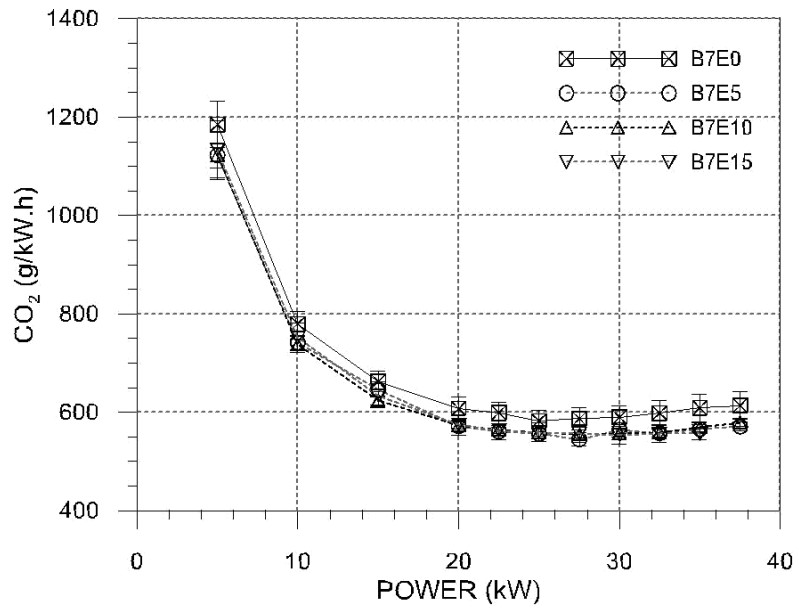


Figure 6 – CO<sub>2</sub> specific emissions variation with diesel oil-ethanol blends.

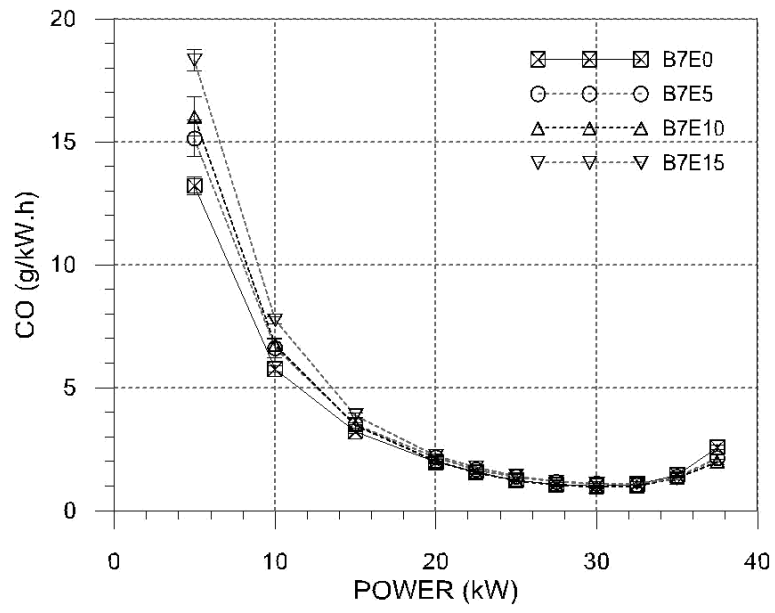


Figure 7 – CO specific emissions variation with diesel oil-ethanol blends.

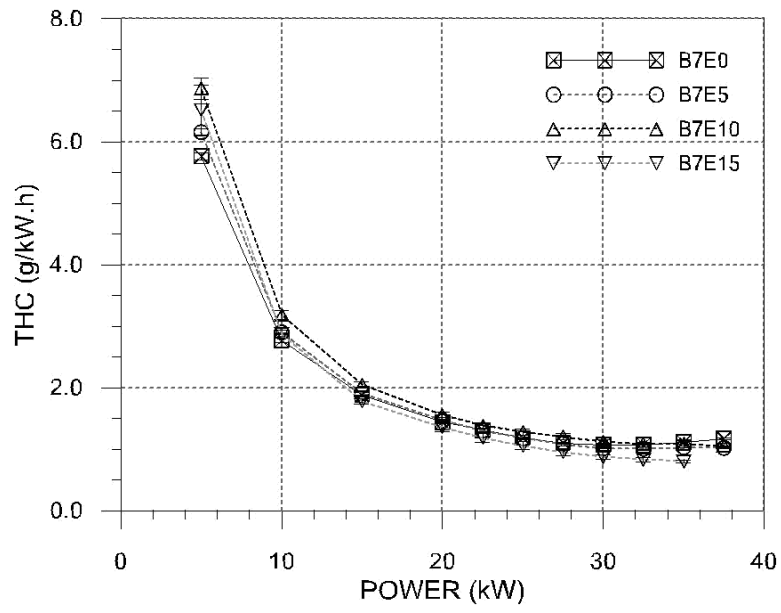


Figure 8 – THC specific emissions variation with diesel oil-ethanol blends.

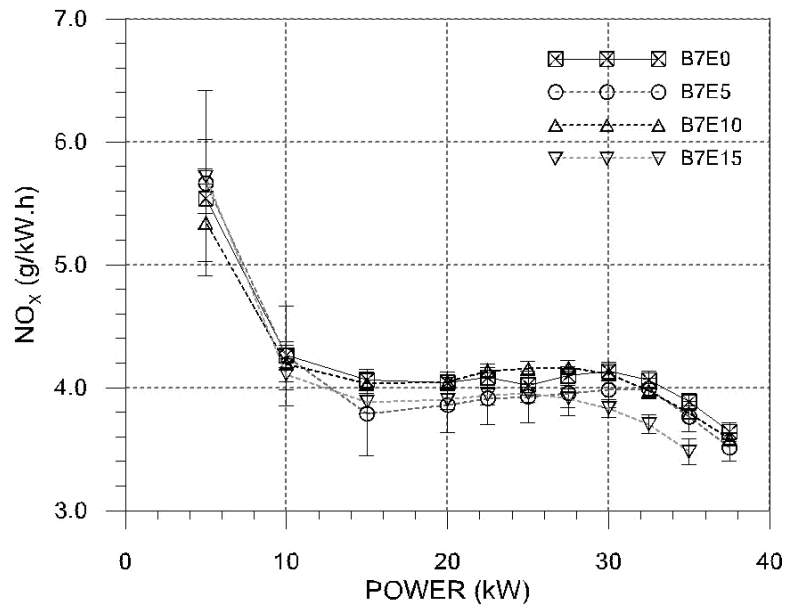


Figure 9 – NO<sub>x</sub> specific emissions variation with diesel oil-ethanol blends.