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Effect of ethanol percentage for diesel engine performance using virtual engine simulation tool

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Abstract

Exhaust emission is one of the kind emission have contribute to the greenhouse effect in the world. Ethanol is one of the best tools to fight air pollution from vehicles. From its biodegradable nature to reductions in greenhouse gas and tailpipe emissions, ethanol provides a tool to address environmental concerns without requiring an entirely new way for goods and people to get from one place to another. Ethanol contains 35% oxygen and with adding oxygen to fuel results in more complete fuel combustion, reducing harmful tailpipe emissions. This paper addresses the possibility study of ethanol percentage to changing of performance and exhaust emissions for Diesel Engine using Virtual Engine Simulation Tool AVL Boost. In this study the blend formulation between Ethanol and Diesel Fuel were E0, E2.5, E5, E7.5 and E 10. The performance of diesel engine simulated in 1,000-1,500 rpm with 0, 10, 20, 30, 40, 50 and 60 Nm engine loads. The direct blending of ethanol and diesel fuel has advantages reducing exhaust emissions CO, Soot and NOx percentages. The engine power break of pure diesel is slightly lower than those of E2.5-E10, especially for speed above than 1400 rpm. The simulation work with the same results compare to the experiments can reduce the cost of research.

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Keywords: ethanol percentage; engine simulations; diesel engine; emissions ; performance

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1. Introductions

Indonesia still relies on fossil fuels (conventional) specifically for the industrial, power generation and transportation systems. This dependence will further reduce the amount of oil reservest here. Exhaust emissions resulting from fuel combustion is one of the main sources of greenhouse (such as CO, CO2, HC) that cause global warming. To reduce dependence on fossil fuel sand reduce the influence of the environmental impact needs to be done diversification of energy sources, especially renewable and environmentally friendly. Climate change and global environmental issues caused by the development and use of energy is a consideration in the selection of alternative energy.

Climate change and global environmental issues caused by the development and use of energy is a consideration in the selection of alternative energy. Vegetable oil (vegetable oil) is one of the alternative energy sources that a lot of attention. Ethanol is commercially produced using either a wet mill or dry mill process. Wet milling involves separating the grain kernel into its component parts (germ, fibre, protein, and starch) prior to fermentation.

Many investigations are related to the influence of the blend formulation between ethanol and diesel fuel. E.T. Jimenes et al reported the physical and chemical properties of ethanol-diesel fuel blends. This study reported that using additives to avoid phase separation and to raise flash point, blends of diesel fuel with ethanol up to 15% can be used to fuel diesel engines if engine performance tests corroborate it [1]. Another investigation also reviewed ethanol and diesel blend. In this paper, the properties and specifications of ethanol blended with diesel fuel are reported. These factors include blend properties such as stability, viscosity and lubricity, safety and materials compatibility. Besides that, the effect of the fuel on engine performance, durability and emissions is also discussed in this paper. The critical factor in ensuring fuel compatibility with the engines is the formulation of additives to maintain blend stability[2]. While, P. Stage de Caro et al showed the behaviour of a diesel–ethanol mixture with two organic additives were selected for their different physico-chemical parameters. Properties directly related to engine parameters (viscosity, cetane number, heat content, volatility) and those characterising fuel quality (homogeneity, cold properties, anticorrosiveness and volatility) were explored. Fuel formulations were prepared with 2% additive and ethanol contents between 10 and 20% in volume in relation to the diesel fuel[3].

Diesel engines are the most efficient engines of all types of internal combustion engine (ICE, internal combustion engines). Current emission control has become central to the development of diesel engines, due to the use of fossil fuels would cause the climate change, which can lead to environmental damage. Many studies explain that the use of ethanol diesel related to improving performance and lowering exhaust emissions. O. Can et al investigated that the ethanol addition reduces CO, soot and SO2 emissions, although it caused an increase in NOx emission and approximately 12.5% (for 10% ethanol addition) and 20% (for 15% ethanol addition) power reductions[4]. Research on exhaust emissions in diesel engines Fuelled with Ethanol-diesel blends has also been done by H. Jinceng et al [5, 6]. However, Li et al investigated Combustion characteristics were analysed in a compression ignition engine fuelled with diesel–ethanol blends with and without a cetane number improver. The research inform that, for the same brake mean effective pressure and engine speed, the maximum cylinder pressure Pmax, the ignition delay, the premixed combustion duration, and the fraction of heat release in premixed combustion phase will increased[7]. Fumigation ethanol in a small capacity diesel engine also been investigated by B.S Chauhan et al. In this study concluded that fumigated Diesel engine exhibit better engine performance with lower NOx, CO, CO2 and exhaust temperature. And the other hand, fumigated Diesel Engine reported increase of unburned hydrocarbon (HC) emission in the entire load range. Considering the parameters, the optimum percentage was found as 15% for ethanol fumigation[8].

Assessments on the new engines performance and emissions obtainable are actually performed in the research and development stages using dedicated simulation tools which offer the possibility to envisage what are the best paths to be followed. By using the simulation engine can reduce the costs of research engine performance and exhaust emissions compared with laboratory tests. The previous study of injection timing for diesel engine operating with gas oil and hydrogen rich gas using AVL Boost investigated by Adrian BIRTAS et al [9]. As for Voicu et al reported a numerical simulation of the influence of injection characteristic on performance and emissions of a tractor diesel engine by using AVL[10].

Many research investigating ethanol in diesel engine, however, it was concluded that the combustion characteristics of ethanol/diesel fuel blends in diesel engine have not been clearly investigated when using the simulation software. “Virtual Engine Simulation Tool” with advanced models for accurately predicting engine performance and the effectiveness of exhaust gas after treatment devices for ethanol diesel using AVL Boost was investigated. According to a graphical programming method and with an interface containing pre-defined constituent elements of the engine, one can specify the relevant geometrical and technical features of every main part of the engine. A symbolic model of the engine is thus designed, which is physically similar to the investigated engine.

2. Methodology

2.1. Fuel preparation

"SOLAR" as the brand for diesel fuel produced by PT. Pertamina, Tbk., and four blends of ethanol with SOLAR (2.5%, 5%, 7.5% and 10%) on a volume basis, called E2.5, E.5, E7.5 and E10 respectively, were used in this experiment. The 99.6% purified ethanol and "SPAN 80" sorbitan methyl ester as surfactant, were obtained from local market. As the complementary information the properties of diesel fuel and ethanol are presented in Table 1 according to Rakopoulos et al [11]. Ethanol diesel blends were prepared by blending in a blender machine in desired dose for 15 minutes at 250 rpm to obtain the homogeneity of the blends. To maintain the stability of the ethanol-diesel blends, 1% of surfactant measured from total blends was added during blending process.

Table 1. Properties of diesel fuel and ethanol [11].

Properties of diesel fuel and ethanol	Diesel	Ethanol
Density 20°C, kg/m3	837	788
Cetane number	50	5-8
Kinematic viscosity at 40oC, mm2/s	2.6	1.2
Surface tension at 20oC, N/m	0.023	0.015
Lower heating value, MJ/kg	43	26,8
Specific heat capacity, J/kgoC	1,850	2,100
Boiling point	180-360	78
Oxygen, % weight	0	34,8
Latent heat of evaporation, kJ/kg	250	840
Bulk modulus of elasticity, bar	16,000	13,200
Stoichiometric air-fuel ratio	15.0	9.0
Molecular weight	170	46

2.2. Simulation

BOOST simulates a wide variety of engines, 4-stroke or 2-stroke, spark or auto-ignited. Applications range from small capacity engines for motorcycles or industrial purposes up to large engines for marine propulsion. A symbolic model of a diesel engine used for experimental research on a test bed was thus created. All these components need design and operational data corresponding to the operation condition which is investigated in Fig 1.

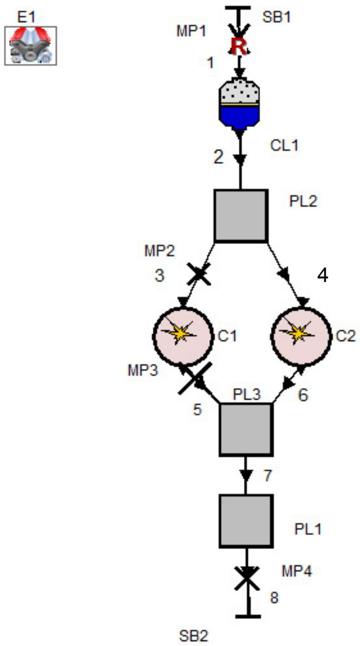

Fig. 1. The engine modelled in virtual engine thermodynamic simulation.

The fluid flows through pipes and manifolds were simulated by one dimensional model, with flow coefficients which have to be defined as initial data. The flow was defined by continuity, momentum, and energy conservation equations; the friction coefficient and the heat transfer to the walls are variable along the pipes [12] Engine specification of data required as input for a boost model is shown in table 2.

Table 2 : The engine specification

Engine parameters	Basic data
Model and type	Fujikawa 295D, diesel four stroke
Number of valve	4
Air charge system	Naturally aspirated
Cylinder / type	2 / Vertical
Volume (cc)	1630 cc
Diameter x stroke	95 x 115 mm
Compression ratio	19 : 1
Maximum torque	96.9 Nm at 1500 rpm
Maximum power	13.5 kW at 1500 rpm
Fuel system	Direct Injection 195 bar
Inner valve seat diameters intake	38.3 mm
Inner valve seat diameters Exhaust	32.5 mm
Piston surface area	10626.93 mm2
Cylinder Head surface area	7084.65 mm2
Liner Surface Area	298.3 mm2

In this simulation, combining ethanol and gasoline percentage performed in control simulation general species setup. The fuel data input species contain of diesel, C2H5OH (ethanol), O2, N2, CO2, H2O, CO, H2,H,O, N and OH. The fuel species setting refer of mass based fraction shown in Fig 2. In this study the blend formulation between Ethanol and Diesel Fuel were E0 (Diesel), E2.5, E5, E7.5 and E 10.

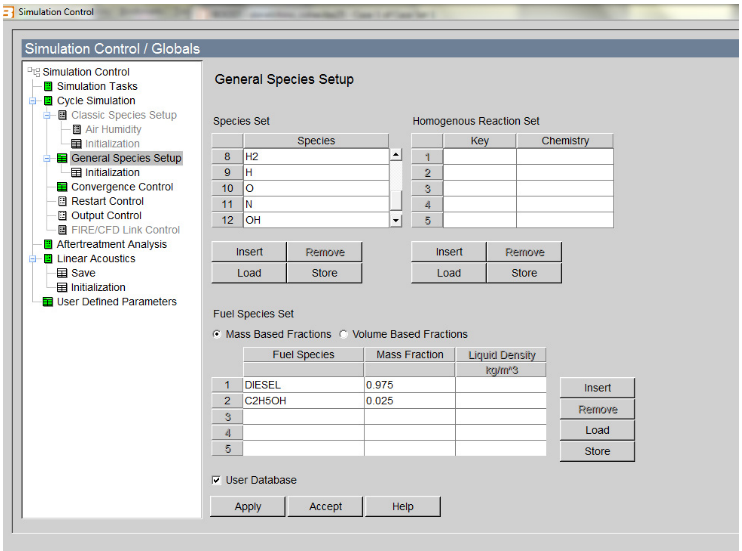

Fig.2. Fuel species setting refer of mass based fraction.

The cylinder was obviously the most complex part of the engine model and it contains the most complicated associated sub models, implying thus significantly more input data than all the other components. For modelling a multi-valve diesel engine, a pipe was connected to each valve shown in Fig 1.The branched part of the intake and exhaust port was modelled by two pipes and a junction. For this junction, the refined model should be used exclusively, as the constant pressure model causes very high pressure losses. This modelling was required only if the two valves feature different valve timings, the geometry of the runner attached to each valve is different or a valve deactivation systems was used. The engine was considered as a block of identical cylinders, exchanging successively mass and energy with the surroundings environment through the valves, according to the given succession of the cylinders operation in the AVL BOOST code[12]. The engine geometric dimensions, the piston movement, the fuel amount, the combustion characteristic and some corresponding estimates for the thermodynamic parameters of the cylinder charge when the exhaust valve opening and for the heat exchange inside the cylinder must be given as input data.

Vibe 2 zone was used for combustion model in this paper. The Vibe function is a very convenient method for describing the heat release characteristics shown in calculation 1, 2 and 3[12].

$$\frac{dx}{d\propto} = \frac{a}{\Delta\propto_c}.(m+1).y^m.e^{-a.y^{(m+1)}} \quad (1)$$

$$dx = \frac{dQ}{Q} \quad (2)$$

$$y = \frac{\propto-\propto_0}{\Delta\propto_c} \quad (3)$$

where, Q total fuel heat input; $\propto$ crank angle; $\propto_0$start of combustion; $\Delta\propto_c$combustion duration; m shape parameter; a Vibe parameter a = 6.9 for complete combustion

It is defined by the start and duration of combustion, a shape parameter 'm' and the parameter 'a'. These values can be specified either as constant values or dependent on engine speed (in rpm) and engine load (expressed as BMEP in bar). In diesel engines, the combustion characteristic depends strongly on the capabilities of the fuel injection system, compression ratio and the charge air temperature. For accurate engine simulations the actual heat release characteristic of the engine the following standard values for diesel engine with naturally aspirated was m parameter= 0.4.

In vibe 2 zones instead of one mass averaged temperature, two temperatures (burned and unburned zone) are calculated. In this research of ethanol diesel blend, the predicts the knocking characteristics of the engine, provided the actual rate of heat release is described properly by the Vibefunction specified.

Instead the first law of thermodynamics is applied to the burned charge (equation 4)and unburned charge (equation 5) respectively[12].

$$\frac{dm_b u_b}{d\propto} = -p_c \frac{dV_b}{d\propto} + \frac{dQ_F}{d\propto} - \sum \frac{dQ_{Wb}}{d\propto} + h_u \frac{dm_b}{d\propto} - h_{BB,b} \frac{dm_{BB,b}}{d\propto} \tag{4}$$

$$\frac{dm_u U_u}{d\propto} = -p_c \frac{dV_u}{d\propto} + \frac{dQ_F}{d\propto} - \sum \frac{dQ_{Wu}}{d\propto} + h_u \frac{dm_B}{d\propto} - h_{BB,u} \frac{dm_{BB,u}}{d\propto} \tag{5}$$

in which ; index b burned zone ; index u unburned zone. The term$h_u \frac{dm_b}{d\propto}$covers the enthalpy flow from the unburned to the burned zone due to the conversion of a fresh charge to combustion products.

The heat transfer to the combustion chamber walls can be evaluated using various computational submodels (e.g. Woschni 1978, Woschni 1990, Hohenberg, Lorenz, Model AVL 2000). In this simulation study Woschni's 1978 correlation was selected with the following calculation relations (equation 6)[12]:

$$Q = hA\,(T_g - T_w) \tag{6}$$

with: h = heat transfer coefficient; A = exposed combustion chamber surface area ;T_g = temperature of the cylinder gas ; T_w = cylinder wall temperature.

The engine friction in this software using Patton, Nitschke and Heywood calculation where the friction losses associated with the main bearings, the valve train, piston group and auxiliary components. The model was originally developed for fully warmed up engine conditions.

The total FMEP is calculated as follows (equation 7)[12]:

$$FMEP_{TOT} = (FMEP_{CS} + FMEP_P + FMEP_{VT} + FMEP_{AUX} + FMEP_{IP}) \left(\frac{V_{Toil}}{T_{oil} - 90^o\,C}\right)^{0.24} \tag{7}$$

in which $FMEP_{CS}$ = the crankshaft mean effective pressure; $FMEP_P$ = the reciprocating mean effective pressure; $FMEP_{VT}$ = valve train mean effective pressure : $FMEP_{AUX}$ = Auxiliary loss mean effective pressure. The last term takes the effect of a changing oil viscosity (as a function of oil temperature) into account (Shayler et al.[13]).

The NOx formation model implemented in this software is based on Pattas and Häfner [14] meanwhile the CO formation model is based on Onorati et al. [15]. For the soot formation model, this software is based on Schubiger et al. [16].

2.3. Engine experiment

A two cylinder direct injection stationary diesel engine was selected for the experiment. The basic data of the engine were presented in Table 2. The experiment was carried out by set-up the diesel engine on an engine test bed that. Fuel balance, emission meter, pressure sensor, crank angle sensor, and temperature sensor for intake and exhaust manifold were also installed. To manage the rpm and engine loads the engine was coupled with eddy current dynamometer. Fuel balance used for fuel consumption measurement and flow of air intake was measured using hotwire anemometer. Meanwhile, the pressure sensor and crank angle sensor were combined to measure the indicated mean effective pressure (IMEP)[17]. The schematic diagram for experimental set up is presented in Fig. 3.

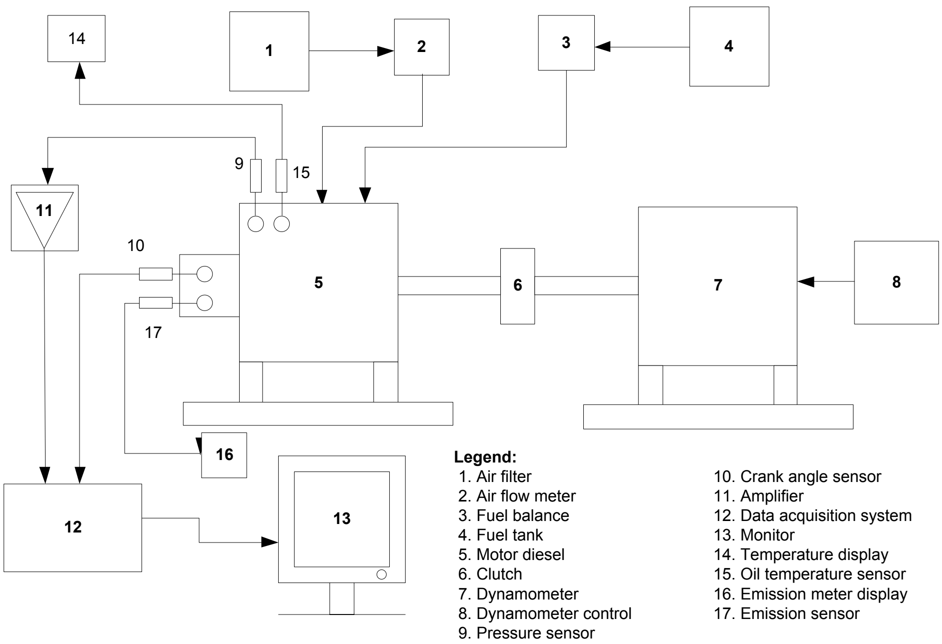

Fig. 3. Schematic diagram for experimental set-up[17].

2.4. Experimental procedure

The experiments were conducted using fuel test that was prepared at 1500 rpm. The loads were set on 0, 10, 20, 30, 40, 50 and 60 Nm. Then, the parameters of the engine in every operated condition were recorded i.e. fuel consumption, air intake consumption, oil engine temperature, air intake and exhaust temperature, cooling water temperature both intake and outlet of radiator, and emission of CO, HC and smoke. The IMEP and fuel consumption data were recorded at least twice in each test. Although each measurement tools and sensors were well calibrated, they always have accuracy and measurement uncertainty corresponding to their specification[17].

3. Result and Discussion

3.1. Performance Analysis

The variations of engine power break with variation of speed for diesel fuel and blended ethanol from E0, E2.5, E5, E7.5 and E 10 was presented in Fig.4.Fig 4a shows the engine power break of Virtual Engine Simulation Tool AVL Boost using different blended fuel under various speeds. It seems that the engine power break is quite insensitive above than 1400 rpm. On the contrary, below 1400 rpm the engine power break is sensitive. However, the engine power break of pure diesel (E0) is slightly lower than those of E2.5-E10, especially for speed above than 1400 rpm. This phenomenon is also clearly presented in Fig 4b which discusses the rate of heat release (ROHR) at the 1480 rpm. This figure shows that rate of heat release E2.5-E10 higher than E0.Maximum engine power was measured at 1400 rpm as 11.15 kW for diesel fuel compare to 11.62 kW for E25. The result shows that maximum engine power for diesel engine is slightly lower the ethanol blended. The maximum power of E2.5 increased with four percentages. The combination of changes in the combustion process was contributed from the physical and chemical differences of fuel structure of ethanol and diesel fuel shown in Table 1.

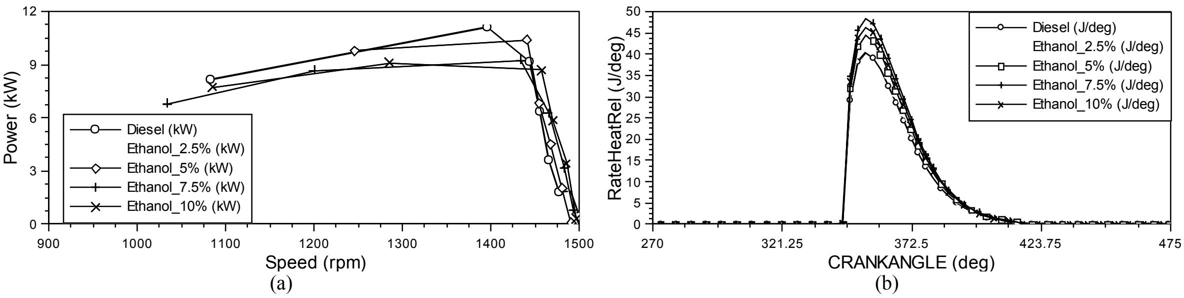

Fig. 4. Effect ethanol addition to; (a) engine power breakand; (b)rate of heat release.

Comparison engine power brake result between simulation and experiment shown in Fig 5a.In the is presented the same trend between engine power break simulation result and experiment result. It seems slightly difference engine power brake result between the simulation and experiment at many sampling points above1400rpm. It showed that the simulation work with the same results compare to the experiments can reduce the cost of research.

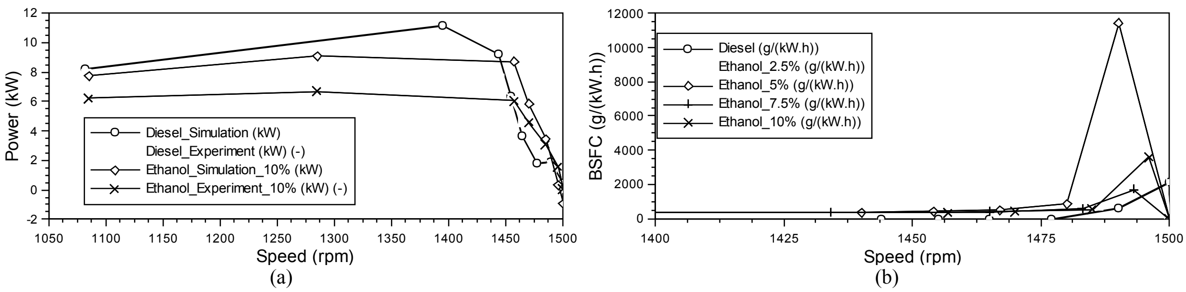

Fig. 5. (a) Comparison engine power brake between simulation and experiment setup and;(b) Effect ethanol addition to brake specific consumption.

The brake specification fuel consumption (BSFC (g/kWh)) is defined as the ratio of the rate of fuel consumption (g/h) and brake power (kW). In Fig 5b indicated that the variation of the BSFC with speed (rpm)) for diesel fuel and blended ethanol. For all of fuels tested at range 1475 rpm – 1500 rpm minimum BSFC was obtained at 490 rpm as 450 g/kWh for diesel fuel, 1000 g/kWh for E7.5% and 2000 g/kWh for E10. From the result of BSFC, we argued that the rate of brake specific consumption is depend to the lower heating value where the ethanol lower heating value lower than diesel fuel as seen on Table 1. The heating value (qc) is defined as the heat transferred out of a system during combustion when the initial and final states of the products and reactants are at the same temperature [18]. In other words, the heating value indicates how much energy is contained in a fuel. Increasing the heating value of a fuel will increase the power output of an engine [19].

The variations of CO produced with a speed during the test for diesel fuel and blended ethanol with diesel fuel is displayed in Fig 6a. It reported that CO was measured at below 1400 rpm for E2.5 and E5 higher than diesel fuel. Meanwhile CO measured for E 7.5 and E 10 seem lower than diesel fuel on wide range speeds. Since ethanol has less carbon than Diesel fuel and its oxygen content increases the oxygen to fuel ratio in the fuel rich regions, the CO emissions are generally reduced at full load because of the increased air-fuel ratio and more complete combustion.

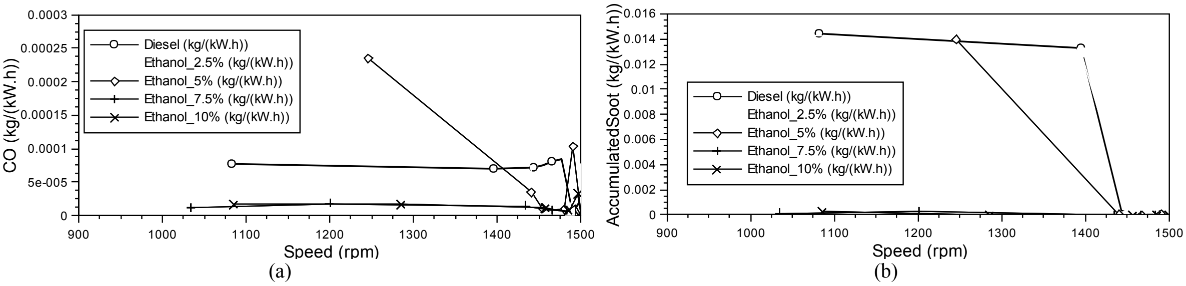

Fig. 6. Effect ethanol added to; (a) CO emission produced; (b) Soot emission.

The soot level was decreased with the addition of ethanol at all range speed depict in Fig. 6b. The addition of ethanol to diesel fuel naturally reduces the amount of soot in the fuel. Therefore, lower soot emissions are expected with ethanol–Diesel fuels. The particulate emissions were consistently reduced with increasing quantity of oxygenated fuel. Oxygenated fuel is nothing more than fuel that has a chemical compound containing oxygen. It is used to help fuel burn more efficiently and cut down on some types of atmospheric pollution.

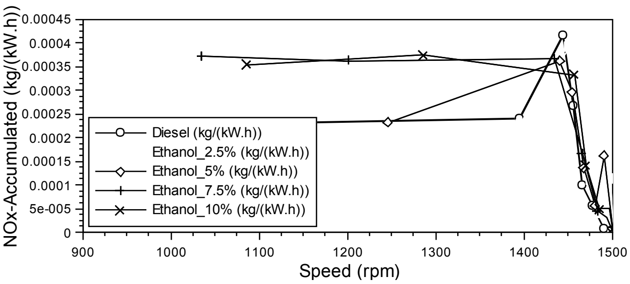

Fig. 7. Effect ethanol added to NOx emission.

Ethanol addition shifts the engine speed where peak NOx emission occurs from 1400 to 1500 rpm illustrated in Fig 7. The NOx emission of the 10% ethanol–Diesel mixture is higher between 1000–1400 rpm and lower after 1450 rpm than that of Diesel. The NOx variation of the 15% ethanol–Diesel emulsion with respect to engine speed shows similar trends with that of 10% ethanol. However, the NOx emission of the 15% ethanol addition is higher than that of Diesel fuels at all speeds.

The explanation is that ethanol–Diesel fuel emulsions cause high NOx because of the low cetane number of ethanol. Low cetane number means increased ignition delay and greater rates of pressure rise, resulting in higher peak cylinder pressures and higher peak combustion temperatures. The higher peak temperature increases NOx formation.

4. Conclusion

In this paper conclusion, the simulations performed that the engine power break of pure diesel (E0) is slightly lower than those of E2.5-E10. This phenomenon is also clearly presented the rate of heat release (ROHR). In other hand increasing the rate of brake specific consumption is depend to the lower heating value of ethanol where the ethanol lower heating value lower than diesel fuel. For the carbon emission, ethanol has less carbon than Diesel fuel and its oxygen content increases the oxygen to fuel ratio in the fuel rich regions, the CO emissions are generally reduced at full load because of the increased air-fuel ratio and more complete combustion. Elsewhere the particulate emissions were consistently reduced with increasing quantity of oxygenated fuel. While the explanation is that ethanol–Diesel fuel emulsions cause high NOx because of the low cetane number of ethanol. The simulation work with the same results compare to the experiments can reduce the cost of research.

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