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Effect of Ethanol Blended Diesel on Engine Performance and Reduction of Exhaust Gas Emission

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Abstract— **This paper presents theoretical and experimental results of diesel engine operating with diesel to analyze not only to compare the engine performance of theoretical and experimental output but also to reduce the toxic gas. The experimental research was done for the various performance parameters such as brake power, brake specific fuel consumption, and brake thermal efficiency of diesel. In experimental and theoretical engine performance results by using pure diesel, brake power increases when engine speed increases. Moreover, exhaust gas emissions of diesel and ethanol blended diesel (ED) were investigated experimentally and compared at 1400 rpm speed. By blending 5 vol% and 10 vol% of ethanol with diesel were prepared for exhaust gas emission analysis. The aim that used ethanol blended diesel is to reduce the toxic gas. Emissions were investigated by changing load conditions at constant speed (1400 rpm). Nitric oxides and carbon monoxide decreased using ethanol blended diesel while load increased at 1400 rpm speed.**

Index Terms— **Exhaust gas emission, Engine Speeds, Brake Power, Ethanol blended fuel, toxic gas**

1) INTRODUCTION

Nowadays all over the world, the usage of fuel in the automobile is dramatically increased more and more. Therefore, the resources of fossil fuel such as petrol and diesel fuel are dwindling day by day. This is a challenge to science and technology [1,2]. Limited crude oil reserves, ambient air pollution and global warming are urgent matter to investigate new environment friendly alternative and renewable energy sources for diesel engine powering. Blending of ethanol even in small quantity could give beneficial results [3,4]. The use of alcohols in mixtures with diesel is limited to anhydrous ethanol, since methanol is practically insoluble in diesel. The use of ethanol and diesel mixtures increases the ignition's lag time due to the low cetane number of ethanol. Therefore, there is limits in usage of ethanol in diesel engine [5]. The stability, density, viscosity, surface tension, specific heat, heat value, and cetane number of blends have great impact on the injection, atomization, ignition, and combustion properties, as well as cold start, power, fuel consumption, and emission characteristics of engine. The energy content of ethanol-diesel blends is lower than a typical diesel fuel [6]. Cetane number is an important fuel property for diesel engines. Cetane numbers of blended fuel depend on the amount and type of additive used in the blends. To overcome this problem, the solubilize is indispensable in ethanol-diesel blended fuel [7]. Among the alternative fuels, biodiesel has received increasing attention due to their attractive characteristics of being renewable in nature and decreasing effect on HC and CO emissions [8]. The potential of ethanol to reduce particulate emissions increases the flexibility to control NOx emissions at different engine operating conditions [9].

In this paper, an experimental investigation was carried out to assess the performance of engine parameters of a diesel engine. The objective of this study is to conduct the performance parameters of diesel in theoretically and experimentally and to reduce the exhaust gas emissions with two percentages of ethanol blended diesel.

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2) METHODOLOGY

The method of research to analyze the engine performance was conducted by theoretically and experimentally. The theoretical procedure is as follows:

(A) Admission Parameters

The operating cycle of an internal combustion engine can be broken down into a sequence of separate processes: intake, compression, combustion, expansion, and exhaust. The initial temperature and initial pressure are assumed during admission. The mean velocity of air at the inlet valve is the following equation:

$$V_{\mathrm{is}} = 2\pi RN\sqrt{(1+\lambda_{rod}^2)}\,\frac{\pi D^2}{4a_{is}} \qquad (1)$$

where V_{is} is the mean velocity of air at the inlet valve, a_{is} is the are through the valve, λ_{rod} is the ratio between the crank radius and connecting rod length. R is the crank radius and N is the crank shaft speed. D is the diameter of piston.

The pressure at the end of admission process is the following equation:

$$P_a = P_{in} - (\beta^2 + \xi)\frac{V_{is}^2}{2}\rho_{in} \qquad (2)$$

where P_a is the end of the admission process, P_{in} is the are through the valve, V_{is} is the velocity of the intake stroke. β is the coefficient showing the drop in the charge velocity and ζ_{is} is the resistance coefficient of the intake system. ρ_{in} is the density of the intake air.

The temperature at the end of admission process is the following equation:

$$T_a = \frac{T_{in} + \Delta T + \gamma_{res} \times T_{res}}{1 + \gamma_{res}} \quad (3)$$

where T_a is the temperature of the admission process, T_{in} is the temperature of the ambient intake process γ_{res} is the coefficient of residual gas.

The volumetric efficiency of engine is the following equation:

$$\eta_V = \frac{r_c}{r_c - 1} \times \frac{P_a}{P_{in}} \times \frac{T_{in}}{T_a(1 + \gamma_{res})} \quad (4)$$

where P_a is the pressure of the admission process, P_{in} is the pressure of the ambient intake process γ_{res} is the coefficient of residual gas. r_c is the compression ratio.

B. Compression Parameters

During compression the temperature and pressure of the mixture increase. Higher compression ratios and thermodynamic parameters at the end of the compression process result in higher expansion ratios and better utilization of the heat.

The pressure at the end of compression is the following equation:

$$P_{com} = P_a (r_c)^{n_1} \quad (5)$$

$$T_{com} = T_a (r_c)^{n_1 - 1} \quad (6)$$

where T_{com} is the temperature of the compression process, P_{com} is the pressure of the compression process n_1 is the polytrophic index.

C. Combustion Parameters

Combustion is a complicated physical and chemical process whose initiation, development and completeness depend on the features and velocities of the chemical reactions, the conditions of heat and mass exchange in the flame zone and heat transfer to the walls. Combustion takes place in the gaseous phase.

The maximum combustion temperature is obtained by the following equation:

$$\frac{\xi_z H_{LCV}}{M_1(1+\gamma_{res})} + \frac{(U_{com} + \gamma_{res})}{1+\gamma_{res}} + 8.314\lambda T_{com} = \mu(U_z^{"} + 8.314 T_z) \quad (7)$$

The maximum combustion pressure is obtained by the following equation:

$$P_z = \lambda P_{com} \quad (8)$$

where, ξ_z is heat utilization coefficient, H_{LCV} is the lower calorific value of fuel (MJ/kg), U_{com} is the internal energy of 1 kmol of air at the compression temperature, U_{com} is the internal energy at temperature t_{com}, of 1 kmol of combustion products (MJ/kmol), U''_z is the internal energy of 1 kmol of combustion products at the temperature of end of combustion process (MJ/kmol), T_z is the maximum combustion temperature at the end of combustion process (K) and λ is the ratio of pressure increase (P_z / P_{com}).

D. Expansion Parameters

During the process of expansion, also called the power or working stroke, the heat energy evolved by fuel combustion is converted into mechanical work. The fuel continues to burn intensively during the initial phase of the expansion process.

Pre-expansion pressure coefficient is calculated by the following equation:

$$\rho = \frac{\mu}{\lambda} \times \frac{T_z}{T_{com}} \quad (9)$$

The pressure at the end of expansion is calculated by the following equation:

$$P_e = \frac{P_z}{\delta^{n_2}} \quad (10)$$

The temperature at the end of expansion is obtained by the following equation:

$$T_e = \frac{T_z}{\delta^{n_2 - 1}} \quad (11)$$

where $\delta = \dfrac{r_c}{\rho}$ is the ratio of subsequence expansion process parameter. n_2 is the polytropic index at expansion process.

E. Indicated Parameters

The indicated work cannot be completely transmitted from the engine shaft to the consumer, since some of it is spent for friction in moving parts and for driving the auxiliary mechanisms.

The mean indicated pressure of the cycle is calculated by the following equation:

$$P_{i,d} = P_a \times \frac{r_c^{n_1}}{r_c - 1}\left[\lambda(\rho - 1) + \frac{\lambda \times \rho}{n_2 - 1}\left(1 - \frac{1}{\sigma^{n_2 - 1}}\right) - \frac{1}{n_1 - 1}\left(1 - \frac{1}{r_c^{n_1 - 1}}\right)\right] \quad (12)$$

The mean indicated pressure of an actual cycle is obtained by the following equation:

$$P_i = \phi_1 \times P_{i,d} \quad (14)$$

where, ϕ_1 is the rounding off coefficient.

The indicated specific fuel consumption (isfc), the indicated thermal efficiency (η_{ith}) and the indicated power (IP) are calculated by the following equations:

$$\text{isfc} = \frac{3600 \times \eta_V \times \rho_{in}}{P_i \times \alpha \times a_{th}} \quad (15)$$

where, η_V is the volumetric efficiency.

$$\eta_{ith} = \frac{3600}{\text{isfc} \times H_{LCV}} \quad (16)$$

$$\text{IP} = \frac{P_i \times N \times V_d \times n}{30 \times \tau} \quad (17)$$

where, n is the number of cycle, τ is engine torque.

(F) Effective Parameters

Power of an engine transmitted from the crankshaft to the consumer is known as the brake power.

Mechanical losses of without supercharged diesel engine is obtained by the following equation:

$$P_{mech} = (A + BV_p) \quad (18)$$

where, A and B are the coefficients values which is depending on the S/D ratio of engine type.

Brake mean effective pressure is given by:

$$P_b = P - P_{mech} \quad (19)$$

Mechanical efficiency (η_{mech}), brake specific fuel consumption (bsfc), brake thermal efficiency(η_{bth}) and brake power (BP) can be calculated by the following equations [10]:

$$\eta_{mech} = \frac{P_b}{P_i} \tag{20}$$

$$\text{bsfc} = \frac{\text{isfc}}{\eta_{mech}} \tag{21}$$

$$\eta_{bth} = \eta_{ith} \times \eta_{mech} \tag{22}$$

$$\text{BP} = \text{IP} \times \eta_{mech} \tag{21}$$

3) EXPERIMENTAL SET-UP AND CONDITIONS

The experiment was carried out to conduct the performance of diesel fuel at various engine speeds such as 1400, 1600, 1800 and 2000 rpm and to investigate emissions of ethanol blended fuels at 1400 rpm speed. The content of ethanol is 0%, 5% and 10% by volume. The figure 1. as shown in below is that ethanol was blended in diesel fuel.

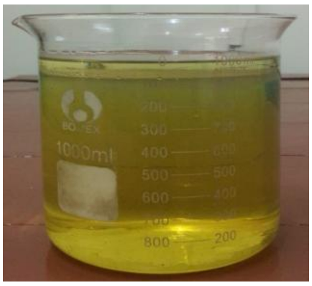

Fig. 1. Sample of ethanol and diesel blend

Table1. Specification of Zh1100A Diesel Engine

No.	Model	Zh1100A	Unit
1	Engine Type	Single cylinder, Four stroke	-
2	Combustion system	Swirl	-
3	Cylinder bore	110	Mm
4	Piston stroke	115	Mm
5	Compression ratio	20:1	-
6	Maximum Power	13.4	kW
7	Cooling Method	Water cool	-
8	Starting Method	Hand Starting	-

In the table.1 shows the specification of Zh1100A diesel engine. Model Zh1100A diesel engine was used without any engine modifications. Emission investigation were carried out for six different load conditions. In this experiment, the six load conditions were 5, 10, 15, 20, 25, and 30 kg. Firstly, the engine was run by using normal diesel. And then, the experiments were done various blended ethanol diesel fuel.

Then, the engine was tested by using 5% and 10% ethanol blended diesel. The results were measured by making the same procedure as diesel. For each experiment, three times were performed to obtain an average value of the experimental data. The remaining fuel in the engine was removed before running with another fuel blends.

To analyze the exhaust gas emission, Testo 340 exhaust gas analyzer is used in the experimental condition. This flue gas analyzer is designed and developed by TESTO company for reliable measurement of flue gas and the particles. The gas type that can be measured Sulfur Dioxide (SO2), Nitrogen Oxide (NOx), Carbon Monoxide (CO) and oxygen (O2). It uses 10 hr. Li-Ion rechargeable battery. O2 measuring range is 0 to 21 percent and the accuracy is +/- 0.2 percent. CO measuring range: 0 to 4,000 ppm. Accuracy: +/-200 ppm (0 to 400 ppm); +/-5 percent of reading (401 to 1,000 ppm); +/-10 percent of reading (1,001 to 4,000 ppm). Draft measurement range: -4 to 16"H_2O (998 to 3,985 Pa). CO_2 (calculated): 0 to CO_2 max. Temperature measuring range: - 40 to 1,832 °F. Efficiency: 0 to 100 percent. Operating temperature range: 23 to 113 °F. Storage temperature range: -4 to 122 °F.

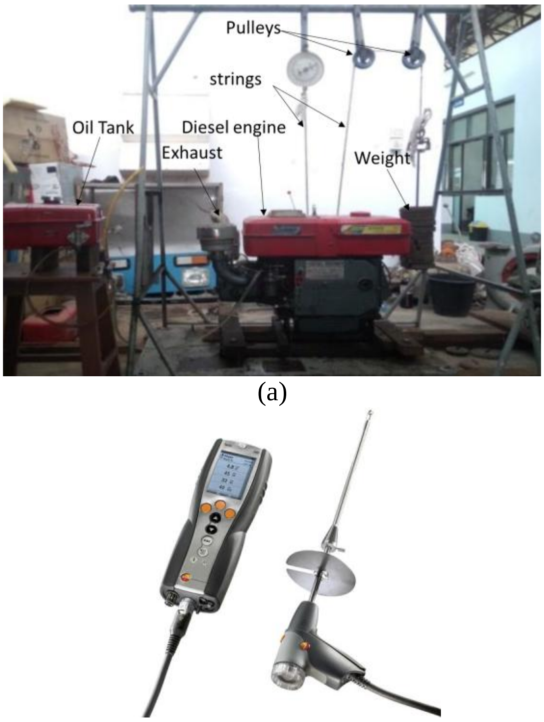

Figure 2. (a) Diagram of experimental setup and (b) Exhaust Gas analyzer (Testo 340)

4. RESULTS AND DISCUSSION

Performance parameters of diesel that was calculated by theoretically are compared with the experimental results at various speeds for 1400, 1600, 1800, 2000 rpm as shown in Fig. (3,4,5). In these three figures, theoretical and experimental results are compared.

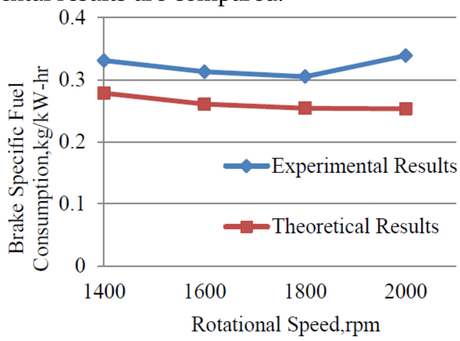

Fig. 3. Brake specific fuel consumption versus engine speed for diesel fuel

As shown in Fig. 3, theoretical calculation results of brake specific fuel consumption decrease while the engine speed is decreased until 1800 rpm. However, at 2000 rpm, the brake specific fuel consumption for experimental work is increased when the theoretical result is still constant.

Brake thermal efficiency is increased when engine speeds are increased in theoretical results. In experimental results, brake thermal efficiency is increased at 1400, 1600, and 1800 rpm but brake thermal efficiency is dramatically decreased at 2000 rpm as shown in Fig. 4.

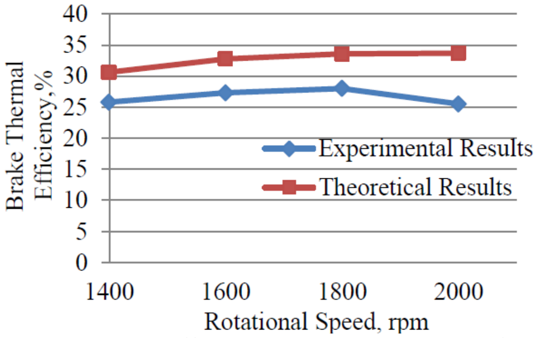

Fig 4. Brake thermal efficiency versus engine speed for diesel fuel

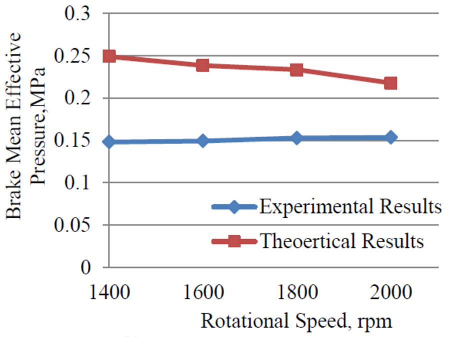

Fig. 5. Brake mean effective pressure versus engine speed for diesel fuel

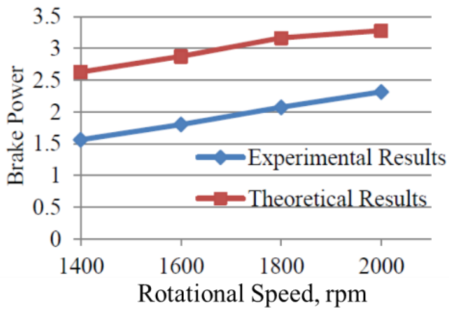

Fig. 6. Brake power versus engine speed for diesel fuel

In Fig.5, theoretical results of brake mean effective pressure are higher than experimental results. The experimental results for various speeds is nearly constant, however, the theoretical results are linearly decreased according to increment of engine speed.

In fig. 6, theoretical results of brake power is higher than experimental results of brake power. The brake power for theoretical and experimental are linearly increased as the speed is increased. The difference between the theoretical and experimental results of brake power for diesel is greater than 1 kW at various speeds.

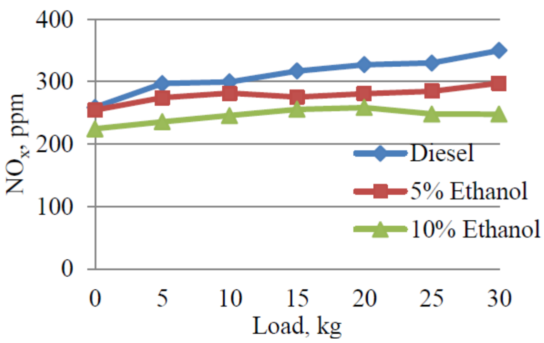

Fig. 7. Comparison of NOx with various load condition for diesel and ethanol blended fuel at 1400 rpm

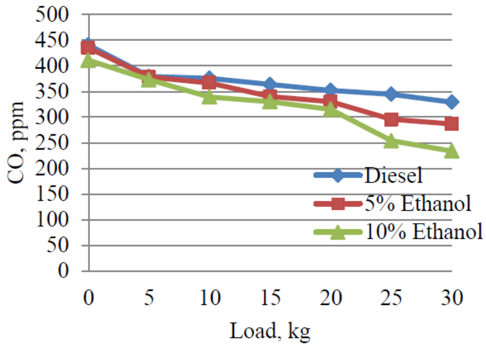

Fig. 8. Comparison of CO with various load condition for diesel and ethanol blended fuel at 1400 rpm

The comparison of effect on exhaust gas emission using ethanol blended diesel fuel and pure diesel are shown in Fig. 7 and Fig. 8. The exhaust gas emission concentration is denoted in part per million (ppm). In experimental investigation of emissions, carbon monoxide (ppm) and nitric oxides (ppm) of 5 vol% and 10 vol% ethanol is lower than diesel.

In Fig. 7, the NO_x(ppm) is linearly increased as the load is increased from 0 kg (no load condition) to 30 kg. At 30 kg load, the difference among three NO_x result is large. In Fig. 8, the experimental results of Carbon dioxide (CO) emission of diesel, 5vol% blended ethanol, and 10vol% blended ethanol are described. The concentration of CO is sharply decreased for all three fuels according to decreasing in load.

According to the experimental results, the concentration of NO_x and CO using 10 vol% blended ethanol is the lowest among three results. Therefore, the 10 vol% ethanol blended diesel fuel is best for reduction of toxic gas in Diesel engine.

5) CONCLUSION

In this paper, the comparison results of brake power, torque, brake specific fuel consumption, brake mean effective pressure, and brake thermal efficiency with theoretically and experimentally results were presented. Experimental studies were carried out with a single cylinder, four strokes, water cool, direct injection diesel engine. In the comparison of engine performance, most of the parameters that calculated theoretically is higher than that of experimentally. However, Brake power increased in both theoretical and experimental results. And also, the engine was tested by using normal diesel, 5vol% and 10vol% ethanol blended diesel to investigate the reduction of toxic gas emissions with six different load conditions at the same speed. When ethanol percentages are higher, CO and NOx concentrations are lower. Therefore, in reduction of toxic gas such as CO and NO_x, the ethanol blended diesel fuel can be used. Moreover, this diesel engine can be run 5vol% and 10vol% ethanol without any engine modifications.

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