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ONE-DIMENSIONAL SIMULATION FOR SINGLE CYLINDER DIESEL ENGINE OPERATING WITH ETHANOL

R. Rahim, R. Mamat and M.Y. Taib

Faculty of Mechanical Engineering, Universiti Malaysia Pahang,
26300 UMP, Kuantan, Pahang, Malaysia;
Phone: +6094242287; Fax: +6094242202
E-mail: rizalman@ump.edu.my

ABSTRACT

This study intended to perform the one-dimensional simulation for single cylinder diesel engine. The one-dimensional numerical analysis of GT-Power software is used to simulate the commercial single cylinder diesel engine. The diesel engine is simulated to study the characteristic of engine performance when the engine is operating with alcohol as alternative fuel. The simulation results were compared with the data from the diesel engine operating with conventional diesel. It is found that the performance of diesel engine operating with alcohol (ethanol) demonstrate lower engine performance and engine efficiency. The simulations are conducted at full load condition for the engine operating with ethanol and conventional diesel. The simulation results show that the brake power and brake torque reduced maximum of 38.84% and 37.67% respectively for the engine operating with ethanol as compared to conventional diesel. Nowadays, the ethanol is able to compete with standard diesel and the economics have become much more favourable in it production. The decrease of low heating value resulted to increase brake specific fuel consumption and reduces the brake thermal efficiency of engine performance at full load.

Keywords: One-dimensional, Ethanol, Diesel engine, Biodiesel

INTRODUCTION

The increasing of air pollution from vehicles is at critical level since the commencement of the internal combustion engine and this phenomenon rise especially after major commercialization. The exhaust emissions generally come from the combustion of fossil fuel in vehicle engines. Diesel is toxic and may cause long term adverse effects to the aquatic environment. Biofuel are available alternative fuels which have a promising future as a substitute for conventional diesel. Suitable feedstock includes soybean, sunflower, cottonseed, corn, groundnut, safflower, rapeseed, coconut oil, palm oil, jatropha seed, tallow (animal fat) or even waste cooking oil. In the US, most biodiesel is derived from soy bean while in Europe; rapeseed is the largest source for biodiesel production. In most tropical countries in Asia such as Malaysia palm oil is the preeminent choice since it is the major vegetable oil produced in the region.

In 2003, the world total biodiesel production was around 1.8 billion liters (Bozbas, 2008). The EU countries contributed 1.5 million tons for the same year; Germany leads the production followed by France and Italy. The European Parliament and the Council of the European Union encourage the public to use biodiesel as the alternative especially for the transport sector. This is due to the fact that the transportation sector accounted for 21% of all CO_2 emissions worldwide in 2002. Currently, 95% of all energy for the transportation sector comes from fossil fuel (Kreith and Goswami, 2007). The use of biofuel in the transport sector may not be just to reduce the emissions but also to shrink the dependence upon imported energy and influence the fuel market for transport and hence to secure the energy supply.

The main benefit of biofuel is that it is 'carbon neutral'. Although the engines running on biofuel produce more CO_2 compare to conventional diesel fuels, if the analysis includes the carbon cycle, the use of biofuel actually emit less CO_2 to atmosphere(Mamat et al., 2008). The biodiesel could be used as it own or blended with conventional fossil fuel without having to make any modification to the standard diesel engines because biodiesel has comparable properties to diesel (Agarwal, 2007). A large amount of research has been conducted on biofuel to perceive the performance and the impact on emission levels to the environment. (Demirbas, 2007) suggested that the combustion of biodiesel alone provides over a 90% reduction in total unburned hydrocarbons (HC) and a 75-90% reduction in polycyclic aromatic hydrocarbons (PAHs). The results from previous experiments on biodiesel and their blends with mineral diesel in a single cylinder engine show that an increased proportion of biodiesel blend resulted in higher NOx, reduced smoke and increased brake specific fuel consumption (Chuepeng at al., 2007). The advantage of biodiesel was also reported by many researchers as renewable energy, non-toxic, biodegradable and sulphur free (Labeckas and Slavinskas, 2006; Bozbas, 2008, Mamat et al., 2009). In addition, mixing of biodiesel with standard diesel fuel improves the lubricating properties of the fuel and reduced cylinder friction (Nwafor, 2004).

Biofuel is normally characterized by its properties of density, viscosity, low heating value, cetane number, cloud and pour points, characteristics of distillation, and flash and combustion points. Pure biodiesel or blends with mineral diesel may reduce the calorific value of the fuel thus may lead to reduced engine power and increased fuel consumption (Rakopoulos et al., 2006). The cetane numbers of biodiesel and mineral diesel are about the same but the volatility of biodiesel is slightly higher which may affect the ignition delay and increase the amount of fuel for rapid combustion and boost the combustion temperature, thus producing higher NOx levels (Labeckas and Slavinskas, 2006). Some purpose such as water pumping for a single cylinder engines are expansively used in agricultural area. The use of the diesel fuel and having only one cylinder make them an economic alternative due to their low investment cost and limited fuel consumption. As a result, these engines are important auxiliary agricultural tool for rural areas. In some cases these engines are also used for driving some vehicles (Bayrakceken et al., 2007). Alcohol maintains to get the global attention as alternative fuels despite of excess in crude oil. For a long period, as the world's crude oil supplies stop to meet worldwide consumption, it is likely that engines running on pure alcohol will become more feasible. In the short term, particularly in those countries susceptible to a shortage in crude oil supplies, contingency plans in the form of alternative liquid fuels to meet the needs of their transport and agricultural sectors are necessary. According to the extension of diesel fuel supplies, it is recently as a particular concern. The use of ethanol in compression – ignition engines has, therefore, received considerable attention with particular emphasis on adapting the fuel to meet the requirements of the engine (Ajav et al., 2000).

Hansen et al. (1989) investigated the combustion of ethanol and blends of ethanol with diesel fuel with the support of a heat release model. They observed that the effects of adding ethanol to diesel fuel were increased ignition delay, increased rates of premixed combustion, increased thermal efficiency and reduced exhaust smoke. Czerwinski (1994) used a rapeseed oil, ethanol and diesel fuel blend and compared the heat release curves with diesel fuel. He observed that the addition of ethanol caused longer ignition lag at all operating conditions. At higher and full loads, the combustion speeds were high with strong premixed phases. Ali et al. (1996) operated a Cummins N 14-410 engines on 12 fuels produced by blending methyl tallow ate, methyl soy ate and fuel ethanol with diesel fuel. The addition of ethanol to the fuel blends did not have an effect on ignition delay. The charge temperature was reported to decrease with a decrease in the diesel content of the fuel blends. Table 1 shows the typical fuel properties for diesel, ester and ethanol (Chen et al., 2008). The properties for these fuels includes boiling point, cloud point, oxygen content, carbon, hydrogen, viscosity, density, cetane number, flash point, heat value.

Table 1: Properties of diesel fuel and ethanol(Rakopoulos et al., 2007)

Fuel Properties	**Unit**	**Diesel fuel**	**Ethanol**
Density at 20°C	kg/m^3	837	788
Cetane number	-	50	5-8
Kinematic Viscosity at 40°C	mm^2/s	2.6	1.2
Surface tension at 20°C	N/m	0.023	0.015
Lower Calorific Value	MJ/kg	43	26.8
Specific heat capacity	J/kg. °C	1850	2100
Boiling point	°C	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	kg	170	46

MODEL SETUP

In general, a one dimensional (1D) simulation of an engine model consists of intake system, exhaust system, compressor and variable geometry turbocharger system (VGT), common rail fuel injection systems, exhaust gas recirculation systems, engine cylinders and valve train. The development of the single cylinder modeling in one-dimensional simulation for four-stroke direct-injection (DI) diesel engine was presented in this paper. The details of the engine parameters used in this model are described in Table 2. Figure 1 shows the diesel engine modeling using GT-POWER software. In the selected diesel engine, the intake system its have a few component, size and different data. The system was started from environment till the intake valve. All of the intake system components in the GT-POWER model are environment, intrunnerairfilter, air filter, intrunner, inport, intvalve. The components in this system need a few data to complete the data form and running the model. Engine cylinder and fuel injection system is focused in engine cylinder performance were support diesel fuel from fuel injection system, fresh air intake system and exhaust gas to exhaust system. The components, size and data must be record and inserted to the GT-POWER form. All of the engine cylinder and fuel injection system component are injector, cylinder and engine. Exhaust system is the last system in the diesel engine. The system was started from exhaust valve and finished in the environment. The GT-POWER components in the exhaust system are exhvalve, exhport, exhrunner, muffler, exhrunnerexit, and environment.

Table 2: Detail parameter of a single cylinder diesel engine

No	**Description**	**Value**
1	Fuel	Diesel
2	Engine Type	Single cylinder
3	Displacement	406.6 cc
4	Piston pin offset	1 mm
5	Number of Cylinder	1
6	Bore	86.0 mm
7	Stroke	70 mm
8	Connecting Rod Length	118.1 mm
9	Compression Ratio	17.3
12	Maximum Intake Valve Open	7.095 mm
13	Maximum Exhaust Valve Open	7.095 mm
14	Intake Duration	252 CAD
15	Exhaust Duration	291 CAD

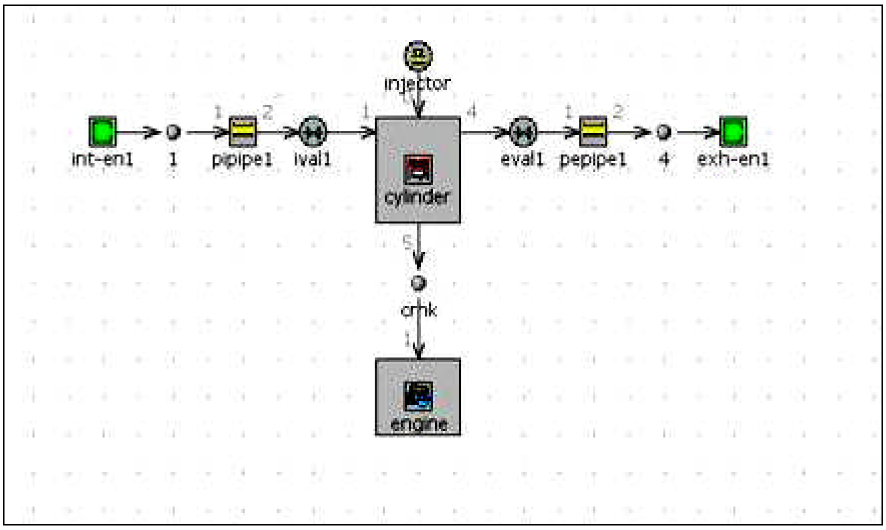

Figure 1: GT POWER single-cylinder diesel engine

The diesel engine modeling using GTPOWER computational model consist of the intake system model, engine cylinder and fuel injection system model, and exhaust system model. The intake system, the engine cylinder and fuel injection system were connected in which the intvalve in the intake system and cylinder in the engine cylinder and fuel injection system. All of this diesel engine components connected by orifficeconn. If the work was finish its can developed the diesel engine modeling using GTPOWER software as shown in Figure 1. The data is very important for building an engine model. A list of information that is required to build a model in GTPOWER is included in library. Not every item will be required for all models, and sometimes additional information will be needed, but the list is generally a good starting point. If the model is being built at an early design stage, determining optimal values for some of the items listed may be the purpose of the simulation. If this is the case, those particular attributes should be defined as parameters and run for a series of cases to determine an optimal value. Data required in engine characteristics are compression ratio, firing order, inline or V configuration, V-angle (optional), 2 or 4 stroke. Data required in cylinder geometry are bore, stroke, connecting rod length, pin offset, piston TDC clearance height, head bowl geometry, piston area and head area. Data needed in intake and exhaust system is geometry of all components. Data in throttles are throttle location and discharge coefficients versus throttle angle in both flow directions. Data in fuel injectors are location and number of injectors, number of nozzle holes and nozzle diameter, injection rate, fuel type and LHV. Data in intake and exhaust valves are valve diameter lift profile, discharge coefficient, valve lash. Data in ambient state are pressure, temperature and humidity. Performance data can be very useful when tuning a model after it has been built. The preparation was important before the model simulation was running. The preparations are to review the completed model, run setup, case setup, plot requests and plot setup. All of the parameters in the model will be listed automatically in the setup and each one must be defined for first case of the simulation. Commonly, computation time can be reduced in steady state simulations by planning the order of the simulations and utilizing the initialization state in run setup. Cycle and/or time plots may be requested by selecting the appropriate plot from the plot options folder within each part. All plots requested in individual parts will be stored regardless of whether the user chooses to use plot setup. If the model has been prepared for simulation, the GT-POWER simulation may be started and this will start the simulation running. A window wills be progress of simulation in the form of scrolling text. Once the input has been read successfully, the simulations will begin, and occasional reports of the progress will be given.

Engine Performance Parameters

In this section, some basic parameters that commonly used to characterize engine operation are developed. These include the mechanical output parameters of work, torque, and power; the input requirement of air, fuel and combustion; efficiencies; and emission measurement of engine exhaust(Heywood, 1988, Pulkrabek, 2004).

***Volumetric Efficiency*:** Volumetric efficiency is used as an overall measure of the effectiveness of a four stroke cycle engine and its intake and exhaust system as an air pumping device. The equation use for volumetric efficiency is expressed as Eq. (1) (Pulkrabek, 2004):

$$\eta_\upsilon = \frac{\dot{m}_a}{\rho_a V_{disp} N/2} \tag{1}$$

where ρ_a = the inlet air density.

$\dot{m}_a$ = the steady-state flow of air into the engine

V_{disp} = displacement volume

***Brake Engine Torque*:** Torque is a good indicator of an engine’s ability to do work. It is defined as force acting at a moment distance and has units of N-m or lbf-ft. Torque (τ) is related to work by Pulkrabek (2004):

$$2\pi\tau = W_b = (bmep)V_d \;/ n \tag{2}$$

where W_b = brake work of one revolution
V_d = displacement volume
n = number of revolutions per cycle

For a four-stroke cycle engine that takes two revolutions per cycle,

$$\tau = (bmep)V_d / 4\pi \tag{3}$$

***Brake Power*:** Power is defined as the rate of work of the engine. If n = number of revolutions per cycle and N = engine speed, then brake power is expressed as Eq. (4) Pulkrabek (2004):

$$\begin{aligned} \dot{W} &= WN / n \\ \dot{W} &= 2\pi N \tau \\ \dot{W} &= (1/2n)(mep)A_p \overline{U}_p \\ \dot{W} &= (mep)A_p \overline{U}_p / 4 \end{aligned} \tag{4}$$

where W = work per cycle
A_p = piston face area of all pistons
$\overline{U}_p$ = average piston speed

Brake Thermal Efficiency: Brake thermal efficiency (η_{bth}) is the ratio of energy in the brake power, (bp), to the input fuel energy in appropriate units (Ganesan, 2003). Solving for thermal efficiency as per below:

$$\eta_{bth} = \frac{\text{bp}}{\text{Mass of fuels} \times \text{calorific value of fuel}} \tag{5}$$

***Brake Mean Effective Pressure*:** Mean effective pressure is a good parameter for comparing engines with regard to design or output because it is independent of both engine size and speed. If brake work is used, brake mean effective pressure is obtained:

$$bmep = w_b / \Delta v$$
$$bmep = 2\pi n \tau / V_d$$

where $\Delta v = v_{bdc} - v_{tdc}$

Brake Specific Fuel Consumption: Brake power gives the brake specific fuel consumption:

$$bsfc = \dot{m}_f / \dot{W}_b \tag{6}$$

where $\dot{m}_f$ = rate of fuel flow into engine

RESULTS AND DISCUSSION

In this study, the effects of ethanol and diesel fuel on the engine performance were being discussed. The tests were performed by varying the engine speed starting from 500 rpm until 4000 rpm with the increment of 500 rpm. The variation of the engine performance that had been discussed were engine air flow, volumetric efficiency, brake power, brake thermal efficiency, brake engine torque, brake mean effective pressure, brake specific fuel consumption and in-cylinder pressure. The results from the simulation model were compared the trends between the diesel and ethanol fuel. Figure 2 shows the effect of engine air flow with respect to engine speed. It can be seen that the trend for this fuel were similar for diesel fuel and ethanol. The engine air flow increase as increasing of the engine speed. The stoichiometric air-fuel ratio (AFR) for ethanol is 40% lower than diesel fuel. Therefore, the engine air flow operating with ethanol is slightly lower due to inducted less air as compared to diesel fuel.

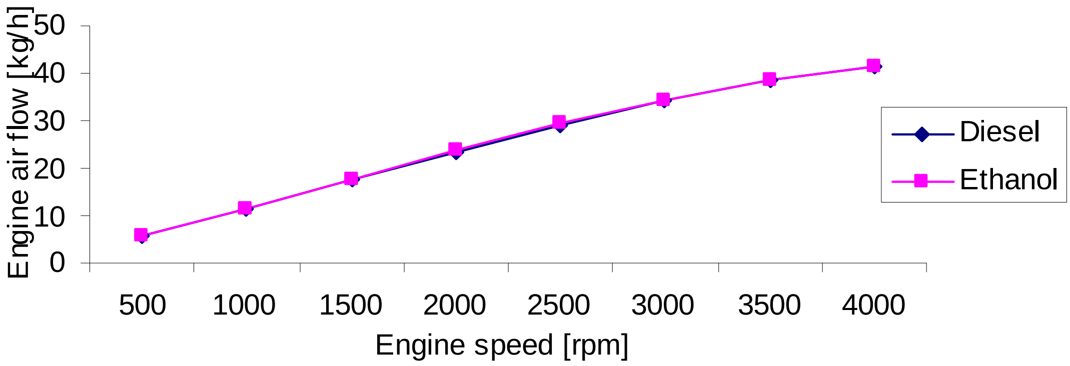

Figure 2: Effect of engine speed variation on engine air flow

Figure 3 shows the effect of volumetric efficiency with respect to engine speed. It can be seen that the maximum efficiency achieved at 2000rpm of engine speed. Engine with higher volumetric efficiency will generally be able to run at higher speeds and produce more overall power due to less parasitic power loss moving air in and out of the engine. The turning point for the volumetric efficiency occurs at the engine speed 2500rpm, so volumetric efficiency decrease sharply as the engine speed increase further. The sharp decrease happens because of higher speed is accompanied by some phenomenon that have negative influence on η_υ. These phenomenon's include the charge heating in the manifold and higher friction flow losses which increase as the square of engine speed.(Rahman et al., 2009)

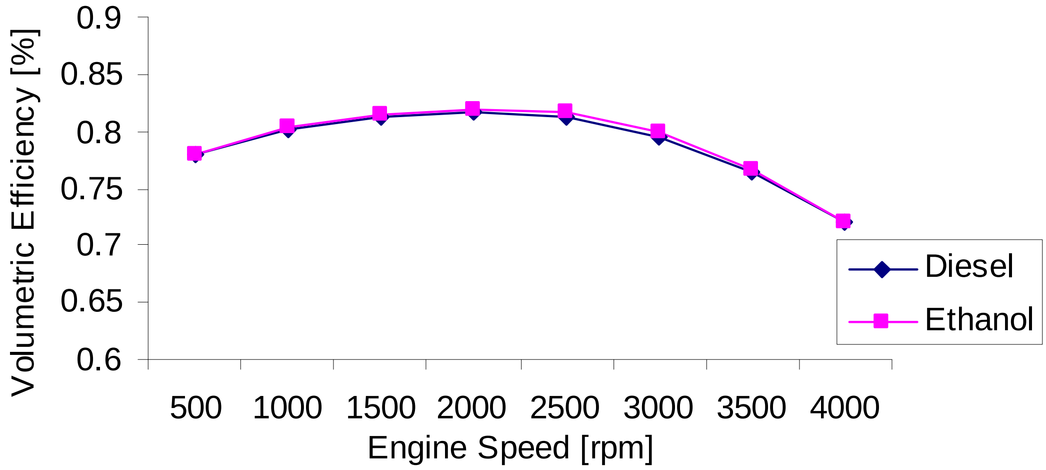

Figure 3: Variation of volumetric efficiency against engine speed

The variation of brake power for diesel and ethanol with engine speed is shown in Figure 4. The figure illustrates the engine outputs at full load. It was observed that the injection delay of diesel fuel was longer than that of ethanol and the ignition delay decreases with the increase in brake power. A close resemblance occurred at low speed representing small discrepancy in output between the fuels. However, at higher speed, a clear gap appeared between the diesel fuel and ethanol. The maximum brake power for diesel and ethanol are 6.86 kW and 4.20 kW respectively. The maximum reduction brake power recorded was about 38.84% at the highest speed. It is well known that the heating value of the fuel affects the power of an engine. The lower energy level of the ethanol fuel causes some reductions in the engine power when it is used in diesel engines without any modifications (Ozer et al., 2004).

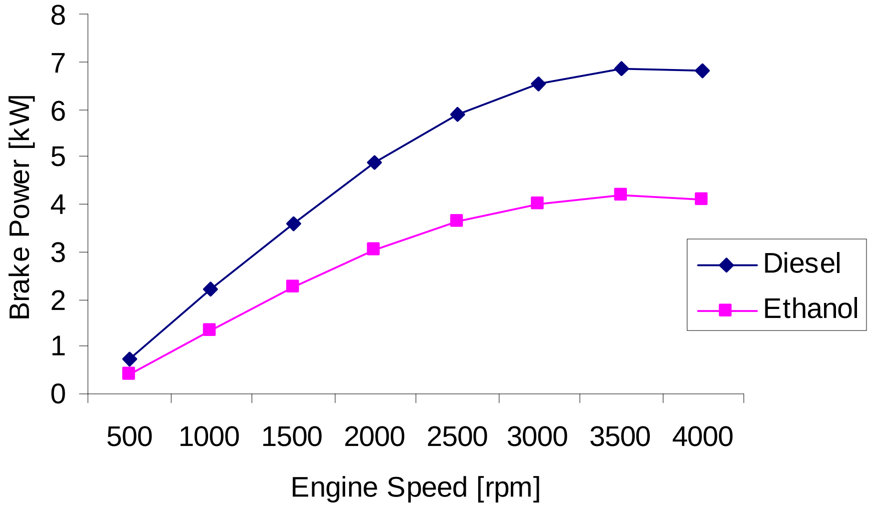

Figure 4: Variation of brake power for diesel and ethanol with engine speed

Figure 5 shows the variation of brake thermal efficiency with engine speed. It is a good measure in assessing how efficiently the energy in the fuel was converted to mechanical output. They generally show similar trends and closely resemble one another. The brake thermal efficiencies for ethanol were lower compared to diesel fuel. The maximum reductions were about 1.2% and 0.9% for the engine speed 1500rpm and

2000rpm, respectively. Brake thermal efficiency lower for ethanol due to lower calorific value compared to diesel fuel. Increasing the ethanol amount in the fuel blend decreased the brake thermal efficiency.(Rakopoulos et al., 2008)

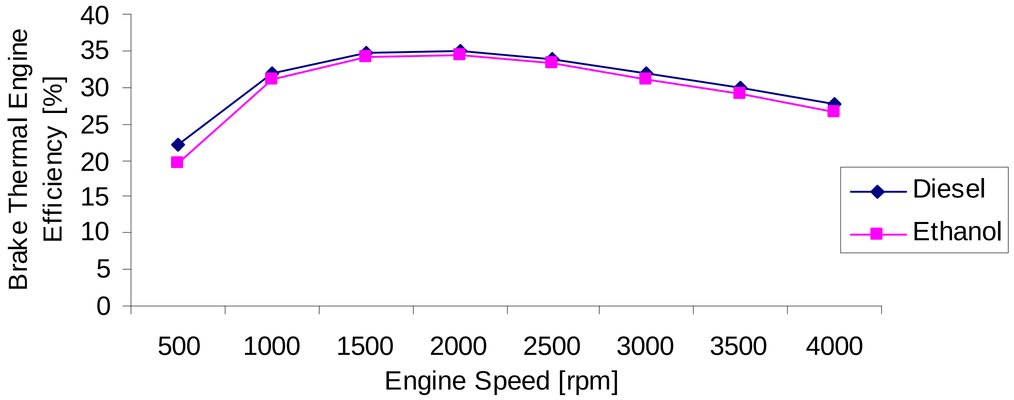

Figure 5: Effect of engine speed variation on brake thermal efficiency

The effect of diesel fuel and ethanol on brake engine torque for various speeds is shown in Figure 6. The torque is a function of engine speed (Abu-Zaid, 2004). At low speed, torque increases as the engine speed increase, reaches a maximum and then, as engine speed increase further, torque decreases as shown in Figure 6. The torque decreases because the engine is unable to ingest a full charge of air at the higher speed. From Figure 6, it is clearly shown the gap between diesel fuel and ethanol. The maximum reduction brake engine torque recorded was about 37.67% when engine speed achieved at 2000 rpm.

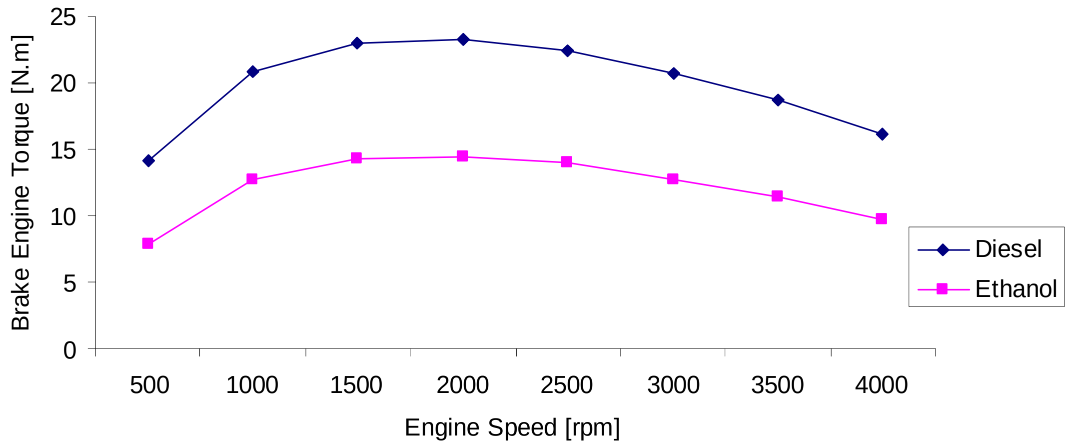

Figure 6: Variation of brake engine torque against engine speed

Figure 7 shows the variation of brake mean effective pressure against engine speed. They generally show the similar trends for both fuels. The maximum reduction of the brake mean effective pressure recorded was about the same with the brake engine torque.

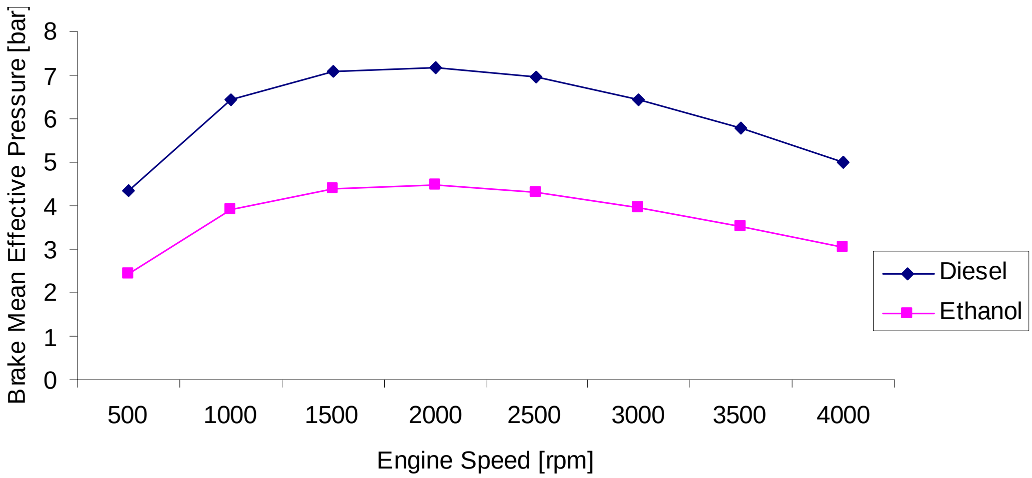

Figure 7: Variation of brake mean effective Pressure against engine speed

Figure 8 shows the effect of engine speed variation on brake specific fuel consumption (BSFC) for diesel fuel and ethanol. They generally show similar trends for both fuels. The minimum BSFC (241.484 g/kWhr) was obtained for diesel, (388.882) g/kWhr for ethanol. As shown in Table 1, the cetane number for diesel fuel is highest than ethanol. Table 1 shows that the diesel fuel has the higher low calorific value (43 MJ/kg) compared to ethanol (26.8 MJ/kg). From Figure 8 and Table 1, it was observed that the higher BSFC for ethanol was due to longer ignition delay. It is caused by their lower cetane number. The decreasing of the cetane number, increase the ignition delay. Increasing the ethanol amount in the fuel blend increased the brake specific fuel consumption (Rakopoulos et al., 2008).

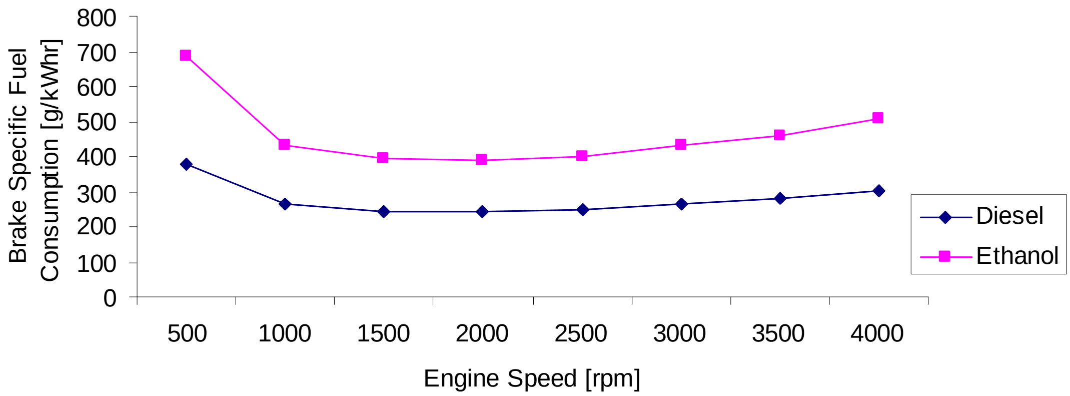

Figure 8: Effect of engine speed variation on brake specific fuel consumption

Figure 9 shows the effect of engine speed variation on in-cylinder pressure for diesel fuel and ethanol. The results show that the peak pressure for diesel is higher compared with ethanol. It is found that the ignition delay for diesel is lower due to highest cetane number, resulting in high peak in-cylinder pressure. The graph line of pressure rise and drop suggest that the shorter ignition delay occurs and it thus responsible for the early start of combustion with the engine operating. If combustion starts too early in the cycle, the work transfer from the piston to the gases in the cylinder at the end of the compression is too large. If the combustion starts too late, the peak cylinder pressure is reduced, and the expansion stroke work transfer from the gas to the piston decreases (Alla, 2002).

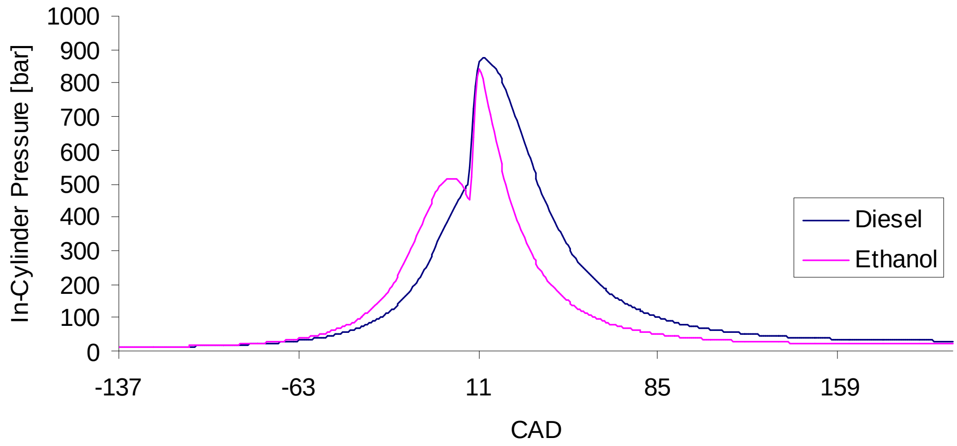

Figure 9: Effect of CAD variation on In-Cylinder Pressure

CONCLUSION

The effect of using the different fuels and their impact on the engine performance of a single cylinder diesel engine has been investigated and the conclusions can be summarized as follows:

1. The decrease in brake thermal efficiency because of the lower low heating value. Among the two fuels of ethanol and diesel, the lower brake thermal efficiency was obtained from ethanol.
2. The brake specific fuel consumption increased mainly due to the lower low heating value. The brake specific fuel consumption for ethanol was higher than that of diesel fuel.
3. It is found that the ignition delay for ethanol was increased due to the lower cetane number, thus resulting in low peak in-cylinder pressure.
4. The lower energy content of ethanol caused some increment in brake specific fuel consumption of the engine depending on the percentage of ethanol in blend.

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