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Design and Prototyping of LPG Mixer Using Computational Fluid Dynamics (CFD)

Rinson Sitanggang*

Pusat Pengembangan dan Pemberdayaan Pendidik dan Tenaga Kependidikan Bidang Otomotif dan Elektronika, Malang , Indonesia

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Abstract

The usage of natural gas is ever increasing with environment consciousness ever increasing. A general bi-fuel LPG system is analyzed on a new four-stroke engine. The mixer of LPG is to be designed for the new four-stroke gasoline engine. Mixer design is crucial to meter the flow of natural gas into the combustion chamber. With this the amount of gas can be controlled for different engine speeds and loads. To reduce the time to design computational fluid dynamics (CFD) is used to get the desired flow condition inside the mixer. Calculation is done for the initial sizing of the throat of the mixer and later is simulated to obtain the best flow characteristics. Different preliminary design is to get the best shape of the mixer. With the help of CFD the best shape is taken for fabrication. The initial calculation for sizing is based on the stoichiometry of the fuel and general fluid dynamic equations.

Keywords: Computational Fluid Dynamics, Compressed Natural Gas, Two Stroke, Venturi System, Mixer, Design

1. Introduction

Two stroke engines have a higher power to weight ratio and are simpler in design. Evolution of the four-stroke engine has seen many designs for the four-stroke engine. The intake of a four-stroke engine is based on scavenging process. LPG system is rarely used on two stroke engines. Current LPG systems only cater for four-stroke engine. In two stroke application which is also widely used in the world today there are an estimated 7-10 million four-stroke cycles in Indonesia which are motorcycle, tricycle and auto rickshaws. Four-stroke engines are characterized by very high levels of hydrocarbon (HC), carbon monoxide (CO), and particulate (PM) emissions [1]. The usage of natural gas may reduce the emmision of the two-stroke. The objective here is to overcome the problem of four-stroke emmisions by using bi-fuel [2] conversion kit that utilizes a mixer. The task is to modify the kit for a four-stroke engine. This is because natural gas is available in large quantities compared to petroleum in the world. As at January 2001, Indonesia's gas reserves stood at 97.6 trillion cubic feet (tcf). This translates to about 66.8 years of natural gas availability [3].

2. Design Development of a Simple Venturi Meter

A venturi mixer utilizes the same fluid mechanics as a standard carburetor system. That is the change in velocity causes a change in pressure in the contraction passage which in D. Ramasamy, S. Mahendran, K. Kadirgama and M.M. Noor turn effects a change in flow of the other medium or (fuel) to join and mix with the main airflow in the required proportion. The mixer throat size is selected based on the airflow capacity required to supply the engine with adequate intake air according to engine operating speed. If the throat is too small the maximum horsepower will be limited. If it is too large a poor vacuum will be created causing engine starting problem [4]. Engines that never operate at WOT should be equipped with an undersized mixer for good starting and to look for low engine speed performance. [5] found that some critical venturis require pressure drops of 9 kPa minimum to ensure constant air flow rate. So the designed model should exceed this value for stable operation of the mixer since the throat of the mixer is a venturi contraction area.

2.1. Determination of Intake Air Velocity in Intake of Engine

The engine being developed is a four-stroke engine. The intake is charged with a supercharger root type with a maximum flow rate from 0.025 kg/s at WOT to 0.004 kg/s at idling (Table2). This is based on engine simulation with the blower system.

2.2. Cross Sectional Area of the Venturi Mixer

The cross sectional area is determined based on engine intake manifold diameter. The area is given as,

$$A_1 = \frac{d_1 \pi}{4} \tag{1}$$

*Email:sitanggang_rinson@yahoo.com

Since the intake manifold is given as 38 mm the cross sectional area is equal to 1134 mm^2

2.3. Inlet velocity

Based on the continuity equation mass can neither be created or destroyed, that is $\dot{m}_1 = \dot{m}_t$ and the inlet velocity is based on maximum engine speed at which the blower gives the maximum air boost at fully open throttle. This is given by [6] in equation 2,

$$\rho_1.A_1.v_1 = \rho_t.A_t.v_t \tag{2}$$

where A1 is the area at cross-section 1, and At is the area at the throat (minimum area) with assuming that the velocities are uniform across the flow area. Contraction cross section of the mixer The contraction of the venturi will cause the air velocity to rise as a linear function of the change in cross-sectional area. The air velocity at the throat cannot exceed speed of 100 m/s at maximum flow rate. This is because above this velocity the air is considered as compressible and the effects of compressibility have to be taken into account. The Mach number for air to be considered incompressible is M < 0.3. With this the maximum velocity of the throat is assumed as 100 m/s as this speed is considered incompressible limit [7]. The venturi area is calculated as equation (3). From continuity $\rho_1 = \rho_2$ so we get,

$$A_1.V_1 = A_t.V_t \tag{3}$$

$A_t = \frac{A_1.V_1}{V_t} = 1,69 \times 10^{-4} m^2$, the diameter is found accordingly and under sized value is taken [4].

$$d_t \le \sqrt{\frac{4.A_t}{\pi}} \le 0,01648 \tag{4}$$

The throat diameter is assumed as 14 mm from the equation above.

3. CFD - Model

The new design shape of the mixer is shown in Figure 1. A single inlet is chosen to enable careful fuel control with the power valve in the conversion kit.

The model is the meshed for calculation by adaptive meshing using rectangular elements in a CFD program shown in Figure 2. The amount of elements is spfecified in Table 1.

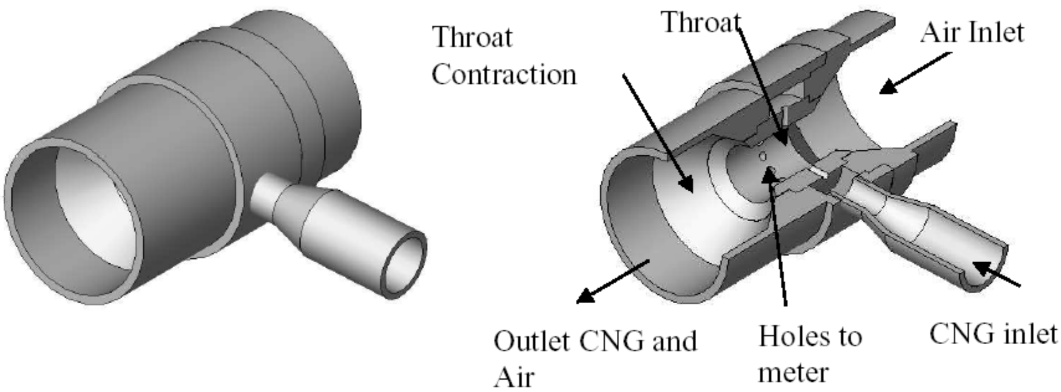

Figure 1. Mixer Shape Before Section View And After Section View

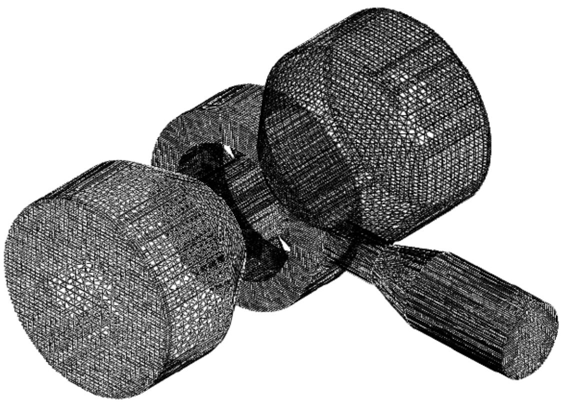

Figure 2. Meshed Model

Table 1. Characteristic of Used Fuel

Fluid Cells	34401
Solid Cells	22965
Partial Cells	26367

The air boundary condition for the inlet is calculated from the equation 4 [8] for engine volumetric efficiency.

$$\dot{m}_a = \eta_v.\rho_a.V_d.N \tag{5}$$

Where $\dot{m}_a$, a is the mass of air into the engine in kg/s. η_v is the volumetric efficiency which is assumed as 1.4 for this two stroke engine ρ_a, is the ambient air density of 1.181 kg/m^3, V_d is the engine displacement of 150 cc, and N is the engine speed in rpm. For the fuel flow rate the stoichiometry of methane is used as below. The boundary is summarized in Table 2. Methane combustion is given as below (6).

$$CH_4 + 2.O_2 + 2.\,(3,76)\,N_2 \;== CO_2 + 2.H_2O + 2(3,76)N_2 \tag{6}$$

The stoichiometry air-fuel ratio value for methane is 17.12 [2], that is can be shown in Table 2.

Table 2. Characteristic of Used Fuel

Engine Speed (rpm)	$\dot{m}_a$	$\dot{m}_f$
6000	0.025	0.00114
5000	0.020	0.00120
4000	0.017	0.00096
3000	0.012	0.00072
2000	0.008	0.00048
1000	0.004	0.00024

4. Result and Discussion

The pressure is dropping, as the air is moving across the mixer as shown in Figure 3. The maximum pressure decrease is seen at 6000 rpm. The decrease will create a vacuum for the fuel to be forced into the inlet manifold of the engine. Figure 4 indicates increasing pressure difference as the engine speed is increased. This is what is explained by [8] that as engine speed is increased the higher flow rate will create an even lower pressure in the venturi throat, which increases the fuel flow rate to keep up with greater airflow. The simulation can be seen in Figure 5 showing the pressure, mass fraction and velocity plots. Basically for all engine speeds the simulation results are same as stoichiometric values are used only the air velocity changes with the increasing engine speeds. The maximum pressure drop achieved is about 16 kPa which is more compared with 9 kPa from critical venturi systems [5].

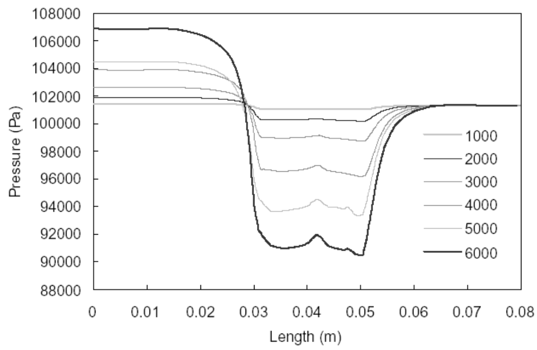

Figure 3. Effect of Pressure Drop In Venturi Mixer From The Inlet To Outlet

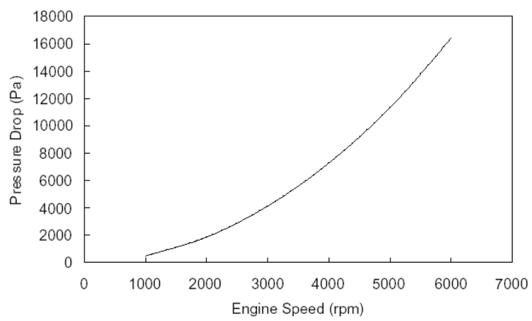

Figure 4. The Effect Of Pressure Drop In Venturi Mixer At Different Speeds

4.1. Setting Point CFD simulation Result

The point of two dimension of setting reference position was arranged to be an object of converter surface, based on the boundary demand contribution that engaged with its function of spread of fuel. The mesh of intake manifold air spread can show in Figure 6. If we investigate the contour of air flow in the intake chamber , related to the effect of designed converter setting point as engine stop , can be shown like follow Figure 8. If we investigate the contour of air flow in the intake chamber , related to the effect of designed converter setting point as engine Idle Load running , can be shown like follow Figure 7. If we investigate the contour of air flow in the intake chamber, related to the effect of designed converter setting point as engine middle load running, can be shown like follow Figure 9. If we investigate the contour of air flow in the intake chamber , related to the effect of designed converter setting point as engine full load running , can be shown like follow Figure 10.

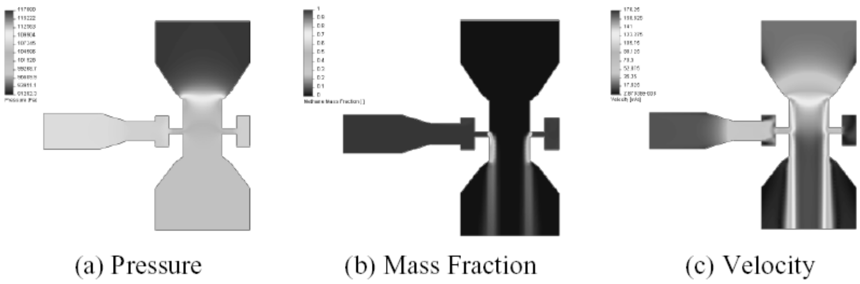

Figure 5. Simulation results of the mixer at 6000 rpm.

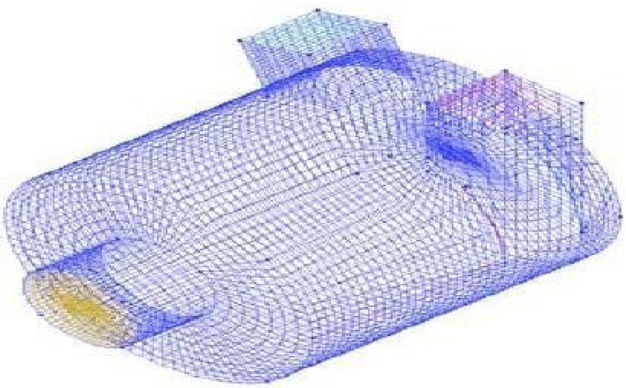

Figure 6. Mesh of the Intake Manifold Air Spread

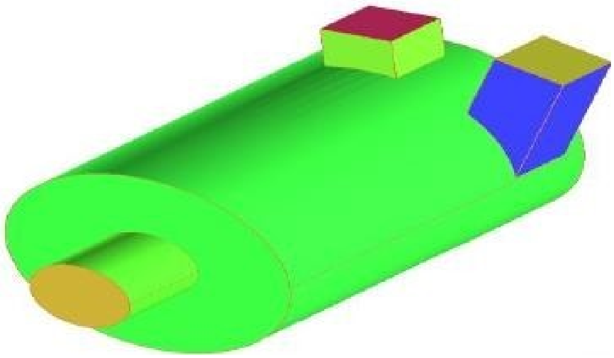

Figure 8. Flow Distribution as Engine Stop

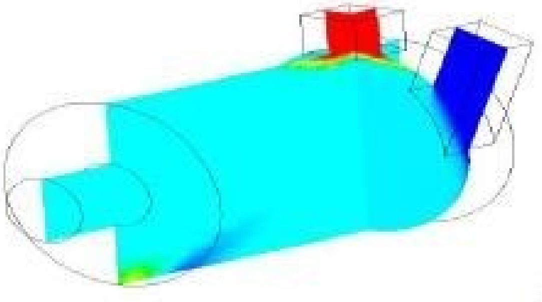

Figure 7. Flow Sistribution as Engine Idle Load Running

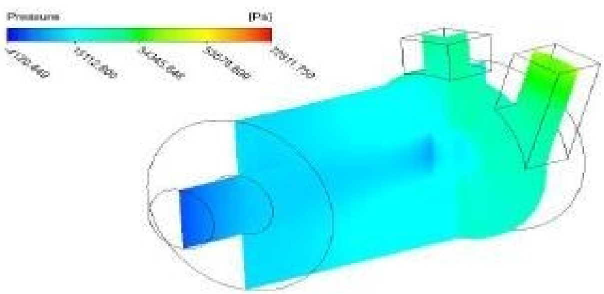

Figure 9. Flow Distribution as Engine Middle Load Running

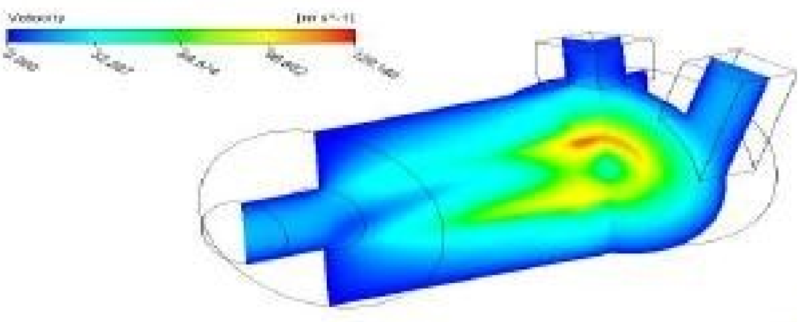

Figure 10. Flow Distribution as Engine Full Load Running

4.2. Performance Thermodynamics within combustion Result

After the simulation using MatLab 7.3 were excessed more than 5 time with initial condition to zero and boundary condition 1000 – 6000 RPM of engine speed versus maximum value of 100% of volumetrically efficiency (Full Load), the result can be shown as follow Figure 11.

For torque, that excessed more than 5 time with initial condition to zero and boundary condition 1000 – 6000 RPM of engine speed versus value of engine torque 12 Newton Meter of volumetrically efficiency (Full Load Setup), the result can be shown as follow Figure 12.

For that thing excessed more than 15 time with initial condition to zero and boundary condition 1000 – 6000 RPM of engine speed versus value of engine Power 10 KW Meter of volumetrically efficiency (Full load setup), the result can be shown as follow Figure 13.

For that thing excessed more than 15 time with initial condition to zero and boundary condition 1000 – 6000 RPM of engine speed versus value of engine combustion pressure 20 Bar of volumetrically efficiency (Full load setup), the result can be shown as follow Figure 14.

The quality of combustion can be representative by producing value of lamda (A/F) ratio, that can enhance by converter when it works in maximum load. Figure 15 shows the result.

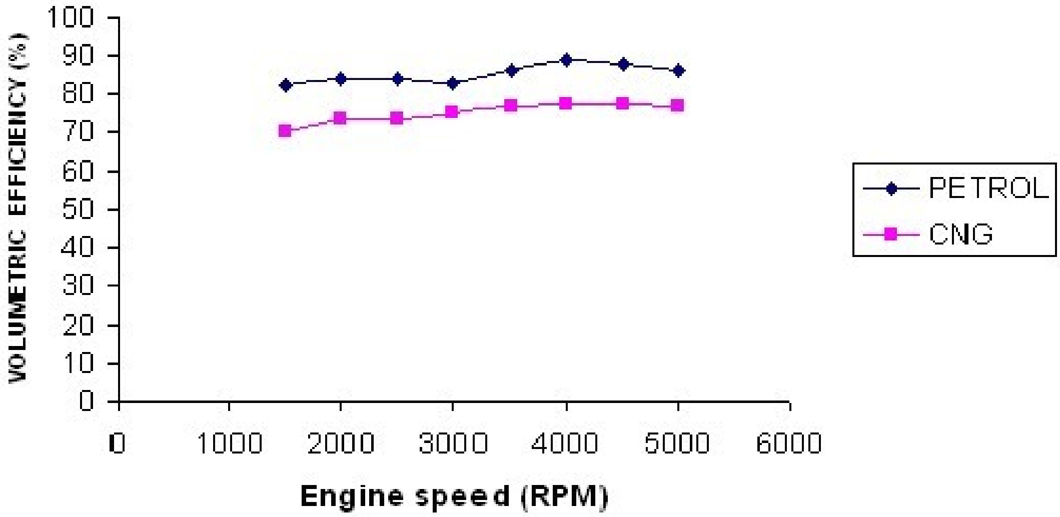

Figure 11. Volumetric Efficiency VS Engine Speed

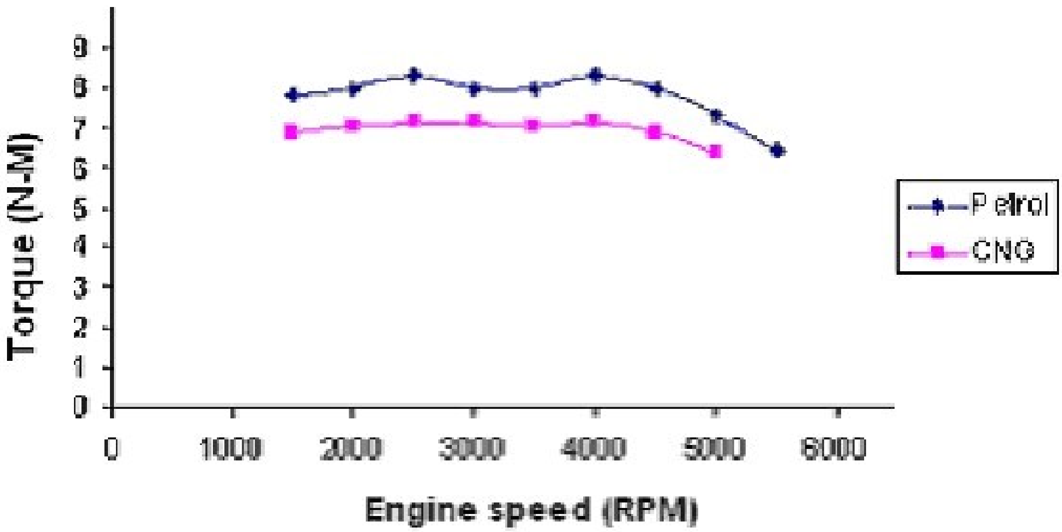

Figure 12. Engine Torque vs Engine Speed

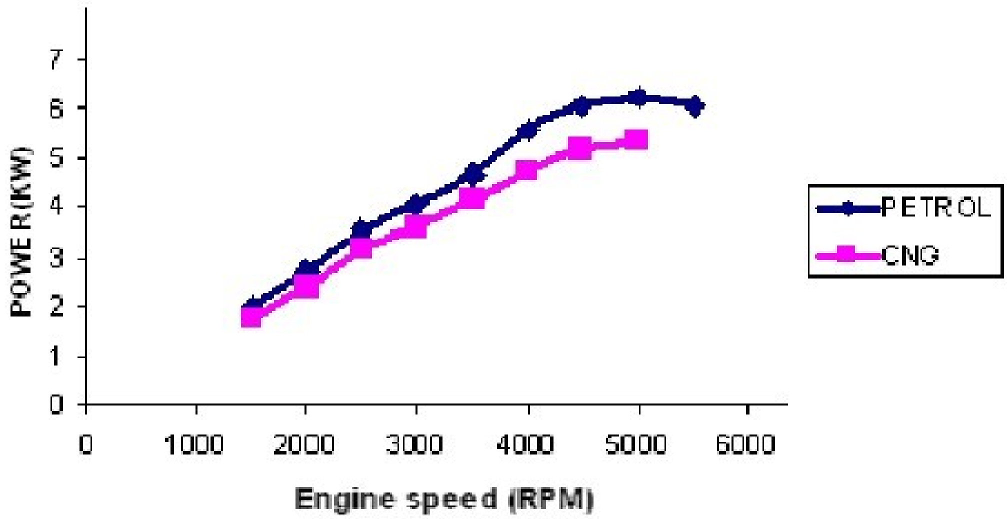

Figure 13. Engine Power vs Engine Speed

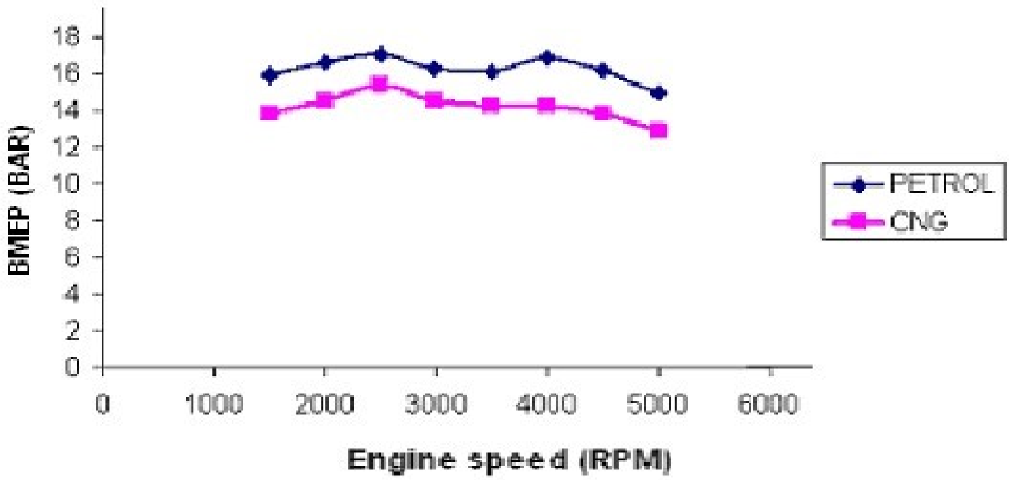

Figure 14. BMEP vs Engine Speed

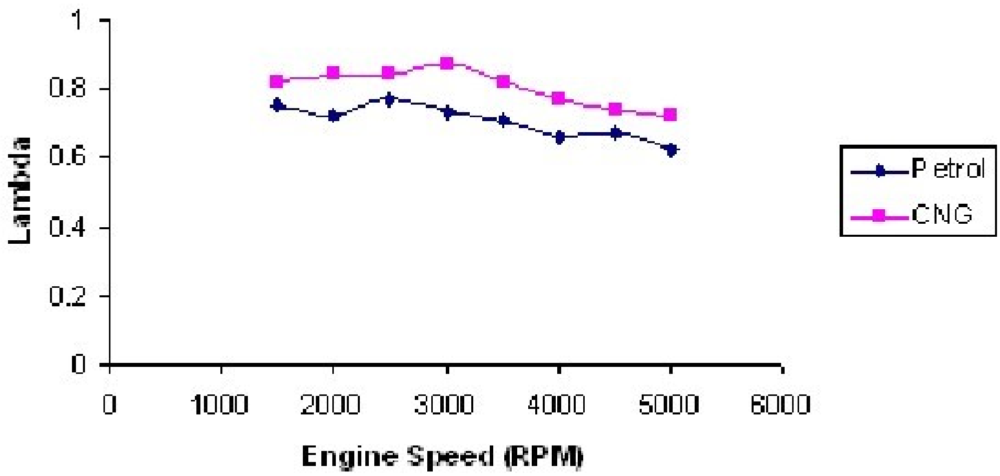

Figure 15. BMEP vs Engine Speed

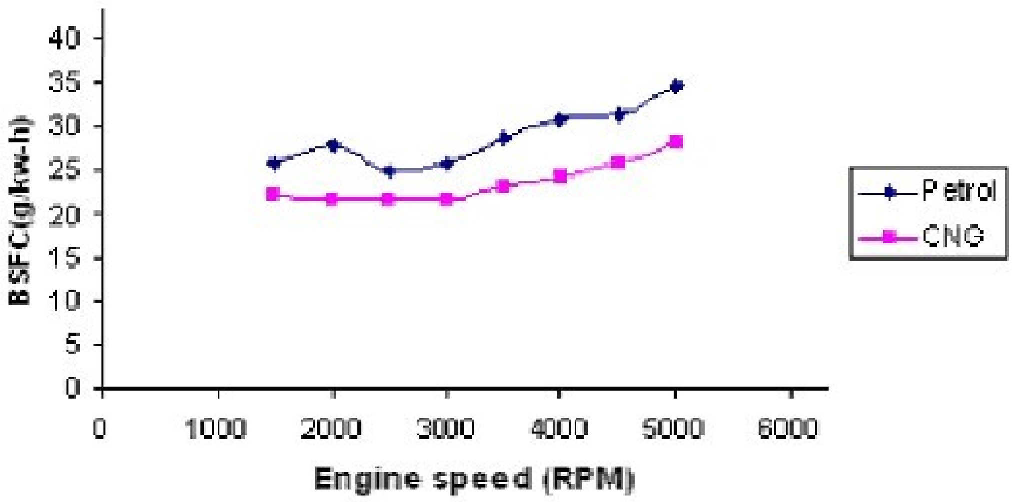

Figure 16. BSFC vs Engine Speed

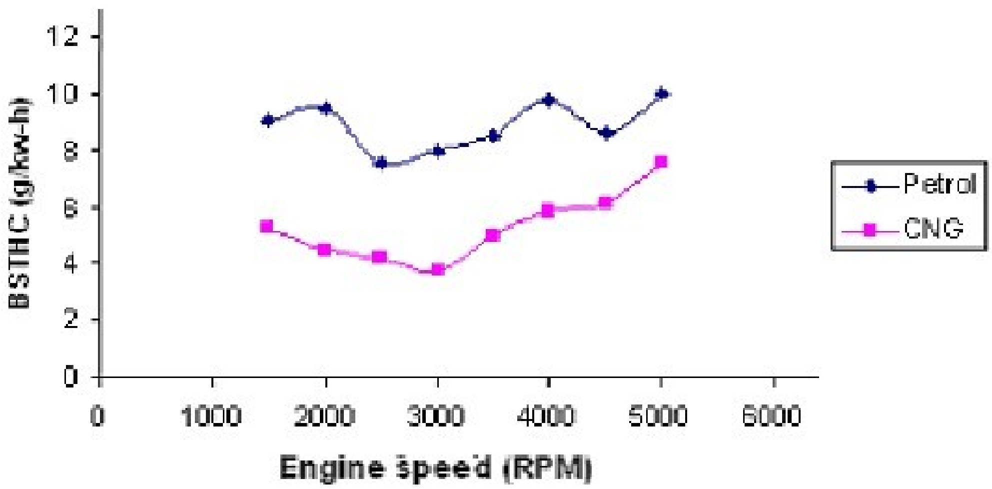

Figure 17. BSTHC vs Engine speed

The quality of combustion can also be representative by BSFC , that can enhance by converter. If the spread of the fuel arranged by converter as full load running. Figure 16 shows the result. The quality of combustion can also be representative by BSTHC , that can enhance by converter. If the spread of the fuel arranged by converter as full load running. Figure 17 shows the result.

5. Conclussion

- As the pressure drop is enough for the venture mixer , this model has to be further validated. The validation will be done on a test rig where the flow rate will be applied for the typical engine conditions. The results obtained here will help in optimization of the gas mixer before the prototype is fabricated and tested in a real engine.
- Using proposed mixer the torque and power of the engine were increasing significantly among kind of engine speed.

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