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# DESIGN OF COMPRESSED NATURAL GAS MIXER USING COMPUTATIONAL FLUID DYNAMICS

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## ABSTRACT

The usage of natural gas is ever increasing with environment consciousness ever increasing. A general bi-fuel CNG system is analyzed on a new two-stroke engine. The mixer of CNG is to be designed for the new two-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

## INTRODUCTION

Two stroke engines have a higher power to weight ratio and are simpler in design. Evolution of the two-stroke engine has seen many designs for the two-stroke engine. The intake of a two-stroke engine is based on scavenging process. CNG system is rarely used on two stroke engines. Current CNG systems only cater for four-stroke engine. In two stroke application which is also widely used in the world today there are an estimated 70-100 million two-stroke cycles in Asia which are motorcycle, tricycle and auto rickshaws. Two-stroke engines are characterized by very high levels of hydrocarbon (HC), carbon monoxide (CO), and particulate (PM) emissions (Bryan, 2002). The usage of natural gas may reduce the emmision of the two-stroke. The objective here is to overcome the problem of two-stroke emmisions by using bi-fuel (Rosli, 2002) conversion kit that utilizes a mixer. The task is to modify the kit for a two-stroke engine. This is because natural gas is available in large quantities compared to petroleum in the world. As at January 2001, Malaysia’s gas reserves stood at 97.6 trillion cubic feet (tcf). This translates to about 66.8 years of natural gas availability (Gas Malaysia, 2003).

## DESIGN AND DEVELOPMENT OF A SIMPLE VENTURI MIXER

A venturi mixer utilizes the same fluid mechanics as a standard carburettor system. That is the change in velocity causes a change in pressure in the contraction passage which in 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 (Maxwell T.T, 1995). Engines that never operate at WOT should be equipped with an undersized mixer for good starting and to look for low engine speed performance. (Xinlei Wang, 2003) found that some critical venturis require pressure drops of 9kPa 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.

**Determination of intake air velocity in intake of engine**

The engine being developed is a two-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 (Table 2). This is based on engine simulation with the blower system.

**Cross sectional area of the venturi mixer**

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

$$A_1 = \frac{d_1^2 \pi}{4} \tag{1}$$

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

**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 (John K. Vennard, 1982) in equation (2),

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

where $A_1$ 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 flowrate. 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 (John D. Anderson, 1989). The venturi area is calculated as equation (3).

From continuity $\rho_1 = \rho_2$ so we get,

$$A_1 v_1 = A_t v_t \tag{3}$$

$A_t = \frac{A_1 v_1}{v_t}$ = 1.6932 x $10^{-4}$ $m^2$, the diameter is found accordingly an under sized value is taken (Maxwell T.T, 1995).

$$d_t \leq \sqrt{\frac{4A_t}{\pi}} \leq 0.01468 \text{ m}$$

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

## COMPUTATIONAL FLUID DYNAMICS MODEL

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

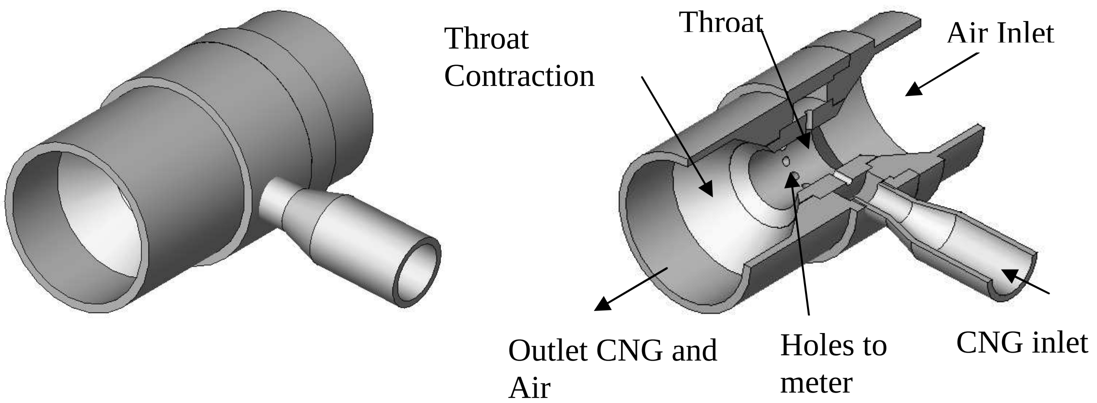



Figure 1: Mixer Shape Before Section View And After Section View

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

Table 1 Mesh Elements

| | |
|---|---|
| Fluid Cells | 34401 |
| Solid Cells | 22965 |
| Partial Cells | 26367 |

The air boundary condition for the inlet is calculated from the equation (4) (Willard, 1997) for engine volumetric efficiency.

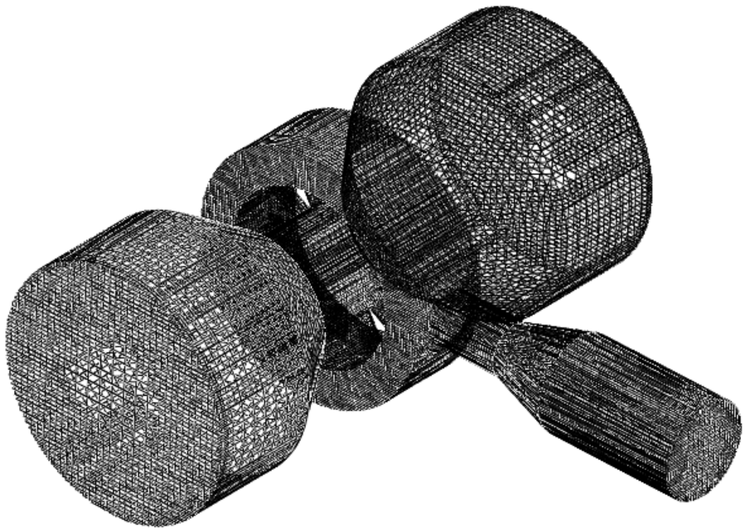


Figure 2: Meshed Model

$$\dot{m}_a = \eta_v \rho_a V_d N \tag{4}$$

Where, $\dot{m}_a$ is the mass of air into the engine in kg/s. $\eta_v$ is the volumetric efficiency which is assumed as 1.4 for this two stroke engine, $\rho_a$ 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 (5).

$$CH_4 + 2O_2 + 2(3.76)\,N_2 \rightarrow CO_2 + 2H_2O + 2(3.76)\,N_2 \tag{5}$$

The stoichiometry air-fuel ratio value for methane is 17.12 (Rosli, 2002). That is;

$$AF = \frac{\dot{m}_a}{\dot{m}_f} \tag{6}$$

Table 2: Boundary for Fuel and Air

| Engine Speed (rpm) | $\dot{m}_a$ | $\dot{m}_f$ |
|---|---|---|
| 6000 | 0.025 | 0.00144 |
| 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 |

## RESULTS 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 (Willard, 1997) 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 (Xinlei Wang, 2003).

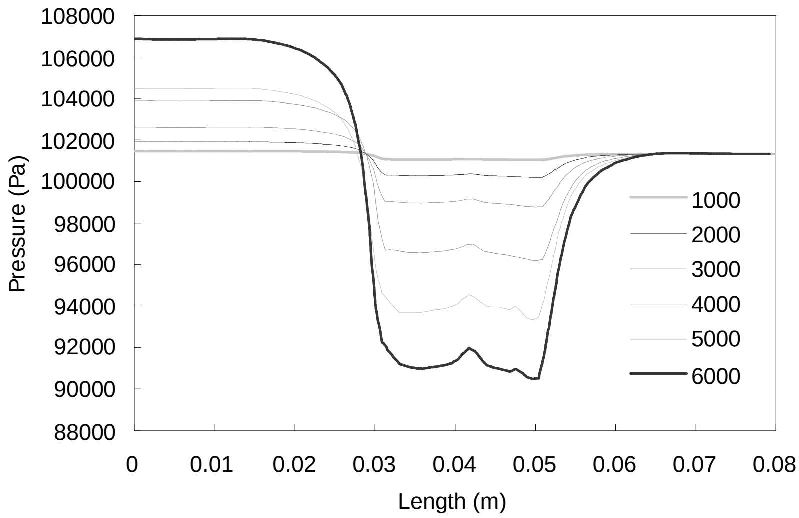



Figure 3: Effect of Pressure Drop In Venturi Mixer From The Inlet To Outlet

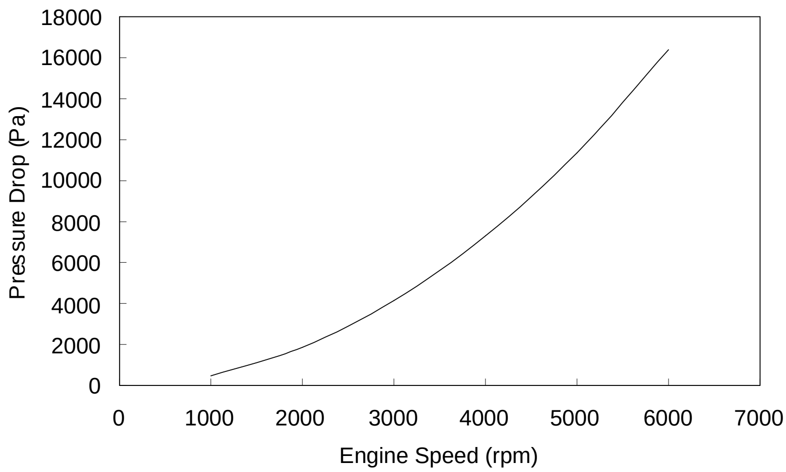
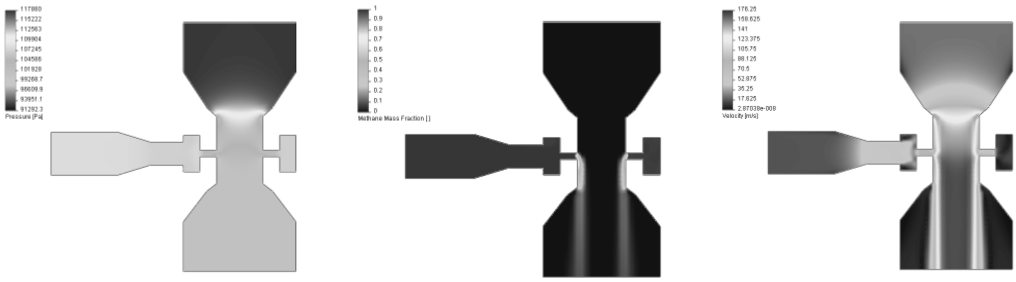



Figure 4: The Effect Of Pressure Drop In Venturi Mixer At Different Speeds



(a) Pressure (b) Mass Fraction (c) Velocity

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

## CONCLUSIONS

As the pressure drop is enough for the venturi, 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.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## Nomenclature

| | |
|---|---|
| A, a | constants |
| $C_p$ | specific heat at constant pressure $J.kg^{-1}.K^{-1}$ |
| D | mass diffusion coefficient $m^2.s^{-1}$ |
| g | acceleration due to gravity $m.s^{-2}$ |
| k | thermal conductivity $J.m^{-1}.K^{-1}$ |
| $t'$ | time s |
| t | dimensionless time |
| $\dot{m}_a$ | mass of air into the engine in kg/s |
| $V_d$ | engine displacement |
| $N$ | engine speed in rpm |
| $A_1$ | area at cross-section 1, $m^2$ |
| $At$ | area at the throat, $m^2$ |
| P | Pressure, Pa |

*Greek symbols*

| | |
|---|---|
| $\mu$ | coefficient of viscosity Pa.s |
| $\nu$ | kinematic viscosity $m^2.s^{-1}$ |
| $\rho$ | density of the fluid $kg.m^{-3}$ |
| $\eta_v$ | is the volumetric efficiency |