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Computational Analysis of Fuel Intake System for Biogas Operated Single Cylinder 4-Stroke Spark Ignition Engine

Abstract

The improvement of flow strategy is implemented in the intake system of the engine to produce better biogas engine performance. The intake system for air-fuel mixture is studied, designed, simulated and analyzed with ANSYS-CFX and CFD-Post code. The components: the venturi and its outer body are installed to produce pressurized turbulent flow with higher fuel volume in the intake system, which is ideal condition for biogas fuelled engine.

A venturi-inlet holes mixer with three variables: number of inlet holes (4, 8 and 16); hole diameter (3, 2 and 1.5 mm) and the inlet angles (30^0, 25^0 and 20^0) are studied. In this study it is found that the combination of throat diameter 17.24 mm, convergent angle 30^0, hole diameter 3 mm and number of holes 4 is proven to improve the power output 11.5% and torque 15.5% compared to average values of the same with other combinations. However, this figure, still approximately 15% lower compared to that of gasoline mode.

Keywords: Cfd,Intake System, Venturi,Biogas

Nomenclature

D_1	inlet diameter (m)
D_2	venturi throat diameter (m)
A	venturi convergent cone angle (^{0})
d	diameter holes (mm)
n	number of holes
N	engine speed (rpm)
A_g	area of biogas flow passage (m^2)
Δp	pressure drop at venturi throat (N/m^2)
V_1	inlet air velocity (m/s)
V_2	throat air velocity (m/s)
V_{2g}	biogas velocity at throat (m/s)
m_g	biogas mass flow rate (kg/s)
m_a	air mass flow rate (kg/s)
Q	air-fuel mixture discharge (m^3/s)
BP	brake power (kW)
T	torque (Nm)
A/F	air-fuel ratio
bsfc	brake specific fuel consumption (gm/kWhr)
mep	mean effective pressure (mPa)
CV_g	calorific value of biogas (kJ/kg)
η_{th}	thermal efficiency of engine
ρ_a	density of air (kg/m^3)
ρ_g	density of biogas (kg/m^3)

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Introduction

The strategy to implement alternative fuels, in internal combustion engines is becoming the subjects of research nowadays. The reasons are driven by two factors; the environmental effects and the energy independence from petroleum based fuel. With more than one billion vehicles around the world, vehicle pollution is becoming the most significant source of air pollution. India, as one of the developing countries, has to deal with these problems too. From these facts, the priority is to find the solution for a cleaner, affordable and better quality of alternative fuels. Among the alternative fuels, biogas has been recognized

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as one of the promising alternative fuels due to its substantial benefits compared to gasoline and diesel.

However in order to adapt standard gas engines few of its components need modifications like compression ratio, valve seats, intake valve timing, etc. This implicate on the high cost that has to be spent, either by car manufacturers or the consumers. Therefore, a simpler and a cheaper method to encourage consumers to use the alternative fuelled vehicles will be a great contribution to the society[1].

Fuel intake device is one of the important components in such category and it is identified that additional research work is to be carried out in establishing a design procedure for this application. The problem under study aims to improve the biogas engine performance by implementing a suitable intake system which endorses the pressurized turbulent flow of air-fuel mixture.The pressurized flow will increase volume of the fuel; hence improve the volumetric efficiency and the turbulence will increase the homogeneity of mixture; hence improve the flame speed. The problem under consideration includes computer aided design analysis of an intake system for biogas operated 4-stroke spark ignition engine. [2]

Analysis

The fuel intake system is designed by using conventional equations and analyzed with a CFD code for evaluating performance.

Need for Analysis

Air/fuel ratio exerts a large influence on exhaust emission and fuel economy in IC engine. With increasing demand for high fuel efficiency and low emission, the need to supply the engine cylinders with a well defined mixture under entire range of operation from no load to full load condition has become more essential for better engine performance. Carburettors are in general defined as devices where a flow induced pressure drop forces a fuel flow into the air stream. But the conventional carburettors are especially designed and developed for liquid fuels like gasoline. Their main functions of conventional carburettors are carburetion and mixing the fuel with air properly apart from acceleration. But biogas being a gaseous fuel requires the intake system only for mixing the fuel with air properly.

Acceleration and deceleration part can be served by conventional carburettor. It has also known that the inlet port design and the intake manifold configuration have a direct influence on engine performance and emissions. Based on this fact, the intake process may give a great contribution toward increasing the biogas engine performance. This study considers the novel design and analysis of the intake system as economical devices without major modification. The vehicle buyers need easier way in converting conventional fuel system into alternative fuel system with less modification. Hence, this design may fulfil the requirements[3].

Design of Intake System

The mixing device has to ensure the provision of a constant air/fuel ratio irrespective of the actual amount sucked into the engine, irrespective of the butterfly valve position. This is achieved by adequate design of the mixing device, whether a venturi mixer or a suction-pressure controlled mixing valve. A simple mixing chamber however requires a control of the fuel gas flow together with the main butterfly valve i.e. it cannot provide a constant air/fuel ratio by its design alone. Mixing chambers are the simplest devices for mixing air and fuel.

The chamber can either be a simple T-joint of two tubes or can be a chamber of a larger volume with one inlet each for air and fuel gas and an outlet for the mixture of both. However, air and fuel are not supplied in a constant ratio independent of the suction of the engine, but have to be controlled by external valves. Such mixing devices can therefore not easily be used for automatic speed and power control but can function in a fixed setting if the engine is operated at one steady condition only.

A suitable mixture for a biogas engine should be a venturi with the accelerator cone being tapered as a curve of suitable radius and the diffuser cone angle. The biogas is fed into the venturi through multiple circular ports around the throat area. Venturi mixers utilize the velocity increase and subsequent pressure reduction in a flow through a tube with a contraction. The pressure at the smallest cross-section area is a function of the air velocity, hence the air volume Fuel gas enters and mixes with the airstream at the smallest cross-section (the bottleneck).

An almost constant air/ fuel ratio is thus achieved. The area of smallest cross-section, the number of circular ports and their diameters are to be selected such that the engine performance will be optimum. With this information, an intake system or a mixer comprising of basic venturi is designed as shown in Fig. 1.

D1 - The inlet diameter remains constant= 26.5 mm = 0.0265 m.

A - Convergent angle = 20^0 to 30^0 for venturi.

D_2 - throat diameter, as thumb rule; the diameter ratio for a venturi may range at $D_2/D_1 = 0.67$ (0.65 to 0.8) which would result in a velocity ratio of $V_2/V_1 = 2.25$. Therefore D_2 should be 17.225 mm to 21.2 mm. But for three different designs with convergent angle (A) 20^0, 25^0 and 30^0 it comes exactly to 17.24 mm, 18.84 mm and 20.46 mm respectively.

n - Number of gas holes at periphery of throat, for even circulation of gas let's consider different designs with 4, 8 and 16 holes angularly evenly spaced.

d – Diameter of gas holes, based on cross-sectional area of the gas inlet A_g and number of holes, it comes 3 mm, 2 mm and 1.5 mm for number of holes 4, 8 and 16 respectively.
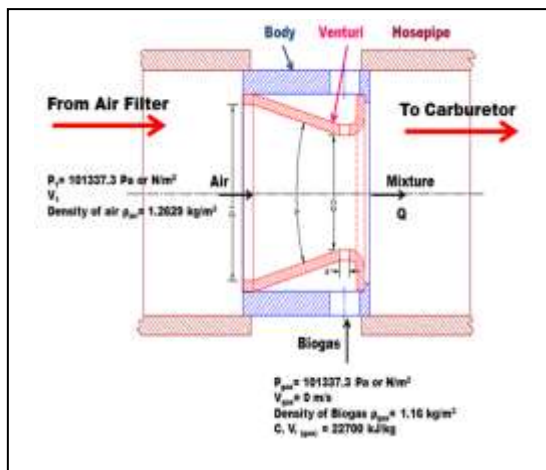
Fig. 1. Design of Intake System
Table 1. Venturi Dimensions

Sr. No.	D_1 (m)	D_2 (m)	A(°)	d (m)	N	A_g (m²)
1				0.0015	16	2.827x10⁻⁵
2		¹0.01724	30	0.0020	8	2.513x10⁻⁵
3				0.0030	4	2.827x10⁻⁵
4				0.0015	16	2.827x10⁻⁵
5	0.02650	²0.01884	25	0.0020	8	2.513x10⁻⁵
6				0.0030	4	2.827x10⁻⁵
7				0.0015	16	2.827x10⁻⁵
8		³0.02046	20	0.0020	8	2.513x10⁻⁵
9				0.0030	4	2.827x10⁻⁵

Software Modeling

The software "AutoCAD2011", "CATIA V5R16" and "ANSYS CFX and CFD-Post" are used for drafting, modeling and analyzing the intake system respectively. So in total nine different venturis, an outer body and two pieces of hose pipe are modeled using this tool as shown in Fig. 2.
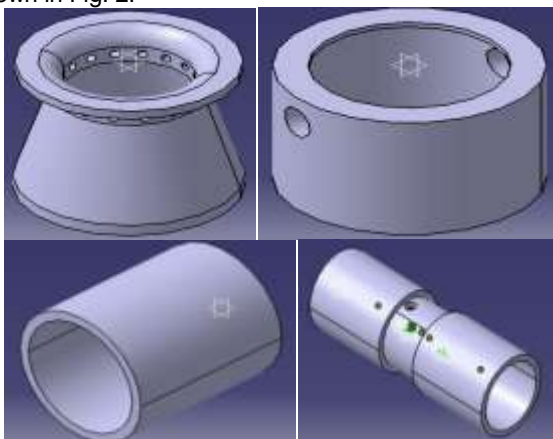

Fig. 2. A Venturi, Body, Hose Pipe and Assembly Software Analysis

CFD Stands for Computational Fluid Dynamics. It is a numerical tool to solve the equations of Fluid Dynamics by suitable methods which can capture the essential physics of the fluid. The numerical schemes that are used for discretization of the equilibrium of equations for fluid, i.e. the Navier-Stokes equations can be one out of Finite Difference Method, Finite Volume Method or Finite Element Method [4].

The general purpose CFD software ANSYS CFX and CFD-Post is used to analyze performance of designs. ANSYS CFX and CFD-Post are supported in Release 12.0 of ANSYS Workbench, which is built on a new framework while leveraging the strength of ANSYS core applications, solvers, and associated tools with a new workflow and simulation project management capability. One of the analyses is shown in Fig. 3, Fig. 4 and Fig. 5.
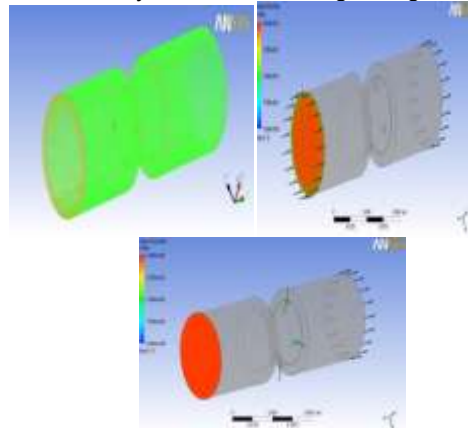

Fig. 3. Mesh Generation and Boundary Conditions

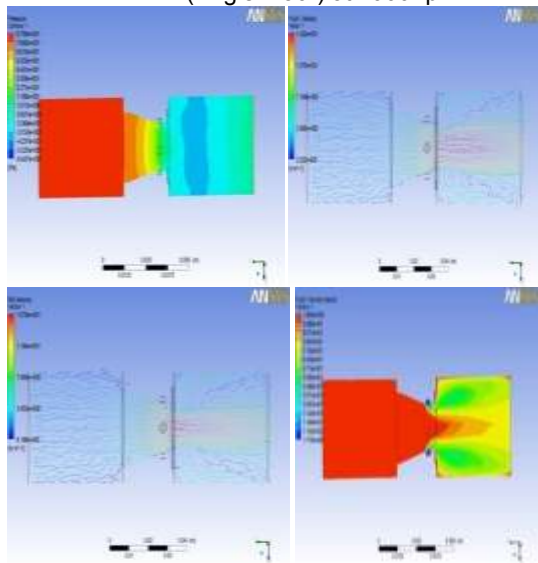Post Processing: Results for D_2= 0.01724 m (Angle = 30^0) at 2000 rpm

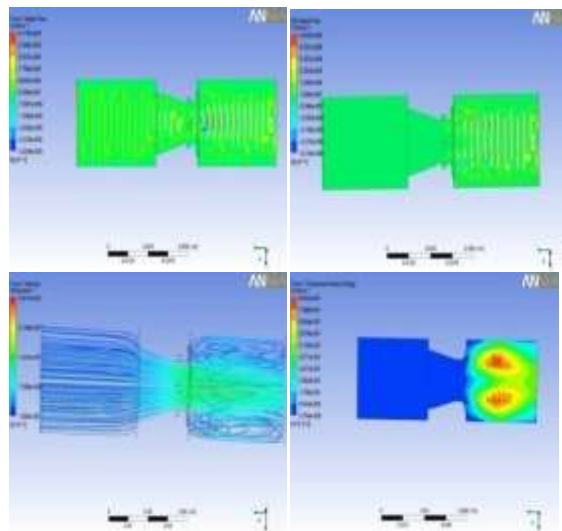**Fig. 4. Pressure Contour, Air and Biogas Velocity Vector, Air Volume Fraction**

Fig. 5. Air and Biogas Mass Flow Contour, Velocity Streamlines and Turbulent Kinetic Energy Results

The results for D_2 = 0.01724 m (A=30^0), 0.01884 m (A=25^0) and 0.02046 m (A=20^0) with 4 holes of diameter 3 mm at N = 2000 rpm, 4000 rpm, 6000 rpm and 8000 rpm have been found as shown in Appendix A. Also for results for D_2 = 0.01724 m (A=30^0) with 8 holes of diameter 2 mm and 16 holes of diameter 1.5 holes have been found. The summary of these results is given in Table 2 to Table 7.

Table 2. Results with 3 mm x 4 holes (Angle= 30^0, 25^0 & 20^0)

Sr. No	N rpm	$\Delta P= P_2-P_1$ N/m^2			V_1 m/s		V_2 m/s			V_{2g} m/s			3-D streamline Velocity m/s			Turbulence KE m^2/s^2		
		All	$^1D_2$	$^2D_2$	$^3D_2$	*All*	$^1D_2$	$^2D_2$	$^3D_2$	$^1D_2$	$^2D_2$	$^3D_2$	$^1D_2$	$^2D_2$	$^3D_2$	$^1D_2$	$^2D_2$	$^3D_2$
1	2000	-76	-56	-41	4	10.8	9.0	7.5	11	9	7	16	12	8	0.85	0.095	0.087
2	4000	-302	-220	-160	8	21.5	18.0	15.0	20	17	14	28	20	15	2.71	0.85	0.35
3	6000	-712	-490	-386	12	32.0	27.0	23.0	28	25	21	41	30	20	6.38	2.71	1.055
4	8000	-1210	-876	-605	16	43.0	36.0	31.0	38	33	28	55	40	30	7.34	6.38	4.71

$^1D_2$- 0.01724 m; $^2D_2$- 0.01884 m; $^3D_2$- 0.02046 m

By using above result from software and the following equations the remaining terms have been found. The sample calculations are as [5,6,7],

$$m_g = \rho_g \times A_g \times V_{2g(act)} \qquad (1)$$

$$Q = A_1 \times V \qquad (2)$$

$$m_a = \rho_a \times Q \qquad (3)$$

$$BP = m_g \times \eta_{th} \times CV_g \qquad (4)$$

$$T = \frac{BP \times 60 \times 1000}{2\pi N} \qquad (5)$$

$$\frac{A}{F} = \frac{m_a}{m_g} \qquad (6)$$

$$bsfc = \frac{m_g \times 3600}{BP} \qquad (7)$$

$$mep = \frac{BP \times 60 \times 1000}{V_s \times N} \quad (8)$$

Table 3. Results with 3 mm x 4 holes (Angle= 30^0, 25^0 & 20^0)

Sr. No	N (rpm)	Q (m^3/s)	m_g ($\times 10^{-4}$ kg/s)			m_a ($\times 10^{-4}$ kg/s)	BP (kW)			A/F Ratio		
	All	All	$^1D_2$	$^2D_2$	$^3D_2$	All	$^1D_2$	$^2D_2$	$^3D_2$	$^1D_2$	$^2D_2$	$^3D_2$
1	2000	2.2×10^{-3}	2.90	2.27	1.95	28.4	1.97	1.55	1.33	9.8:1	12.5:1	14.6:1
2	4000	4.4×10^{-3}	5.85	4.87	3.57	56.76	3.98	3.32	2.43	9.7:1	11.7:1	15.9:1
3	6000	6.6×10^{-3}	8.12	6.82	5.20	85.14	5.53	4.64	3.54	10.5:1	12.5:1	16.4:1
4	8000	8.8×10^{-3}	10.72	9.42	8.77	113.52	7.30	6.42	5.97	10.6:1	12.1:1	13.0:1

$^1D_2$- 0.01724 m; $^2D_2$- 0.01884 m; $^3D_2$- 0.02046 m

Table 4.Results for Average Values with 3 mm x 4 holes

Sr. No	D_2 (m)	A (^{0})	T (Nm)	A/F Ratio	bsfc (gm/kWhr)	mep (mPa)
1	0.01724	30	9.11	10.2 : 1	529	0.60
2	0.01884	25	7.60	12.2 : 1	528	0.50
3	0.02046	20	6.23	15.0 : 1	528	0.41

Table 5. Results for D_2= 0.01724 m with 3 mm x 4 holes, 2 mm x 8 holes and 1.5 mm x 16 holes

Sr No	N rpm	$\Delta P = P_2 - P_1$ N/m^2			V_1 m/s	V_2 m/s			V_{2g} m/s			3-D streamline Velocity m/s			Turbulence KE m^2/s^2		
	All	1	2	3	All	1	2	3	1	2	3	1	2	3	1	2	3
1	2000	-76	-71	-70	4	10.8	10.5	10.0	11	10	9	16	15	15	0.85	0.80	0.7
2	4000	-302	-295	-292	8	21.5	21.0	20.0	20	18	16	28	26	26	2.71	2.6	2.5
3	6000	-712	-702	-699	12	32.0	31.0	31.0	28	25	22	41	39	38	6.38	6.2	6.0
4	8000	-1210	-1200	-1195	16	43.0	41.0	41.0	38	34	30	55	52	51	7.34	7.1	6.9

1- 3 mm x 4 holes; 2- 2 mm x 8 holes; 1- 1.5 mm x 16 holes

Table 6. Results for D_2= 0.01724 m with 3 mm x 4 holes, 2 mm x 8 holes and 1.5 mm x 16 holes

Sr No	N (rpm)	Q (m^3/s)	m_g ($\times 10^{-4}$ kg/s)			m_a ($\times 10^{-4}$ kg/s)	BP (kW)			A/F Ratio		
	All	All	1	2	3	All	1	2	3	1	2	3
1	2000	2.2×10^{-3}	2.90	2.92	2.95	28.4	1.97	.99	2.01	9.8:1	9.7:1	9.6:1
2	4000	4.4×10^{-3}	5.85	5.26	5.25	56.76	3.98	.58	3.57	9.7:1	10.8:1	10.8:1
3	6000	6.6×10^{-3}	8.12	7.30	7.22	85.14	5.53	.97	4.91	10.5:1	11.7:1	11.8:1
4	8000	8.8×10^{-3}	10.72	9.93	9.84	113.52	7.30	.76	6.70	10.6:1	11.4:1	11.5:1

1- 3 mm x 4 holes; 2- 2 mm x 8 holes; 1- 1.5 mm x 16 holes

Table 7. Results for Average Values with D_2 = 0.01724 mm

Sr. No	D_2 (m)	A (^{0})	d x n (mm x no)	T (Nm)	A/F Ratio	bsfc (gm/kWhr)	mep (mPa)
1	0.01724	30	3 x 4	9.11	10.2 : 1	529	0.60
2	0.01724	30	2 x 8	8.5	10.9 : 1	528	0.55
3	0.01724	30	1.5 x 16	8.4	10.9 : 1	528	0.55

Graphs

The graphs are plotted with the results and compared with the standard graphs of IC Engines. Based on graphs of results some significant findings are noted and final conclusions are drawn. During this study few limitations and scope for future study are also sensed(8-9).
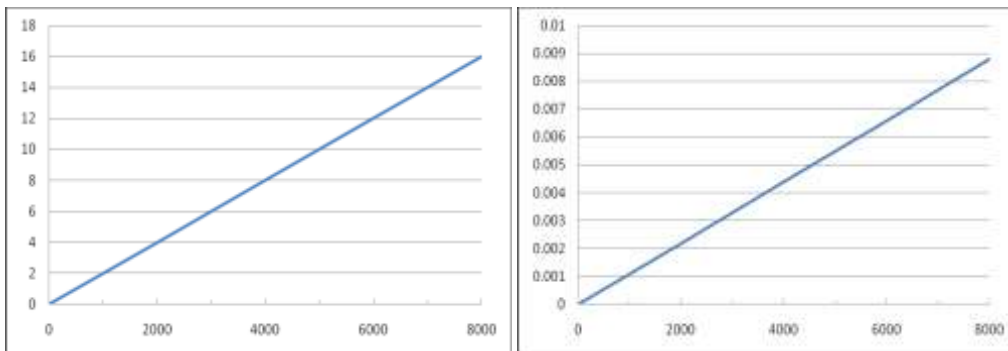
Fig. 6. Inlet Velocity vs. Engine Speed and Discharge vs. Engine Speed
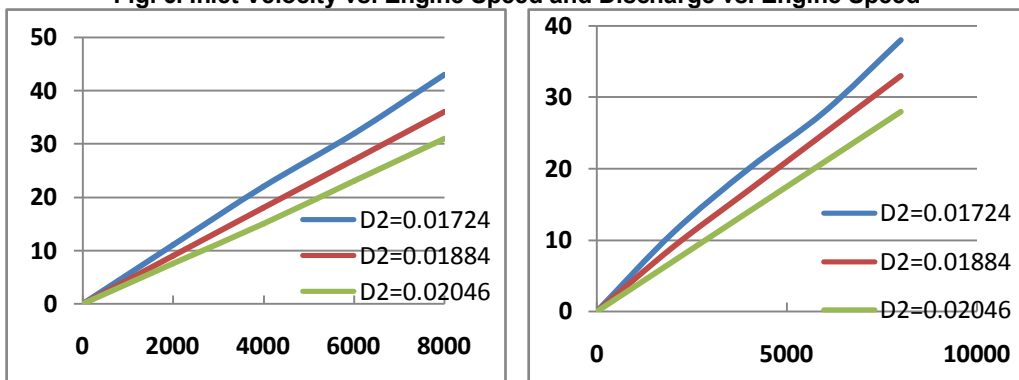

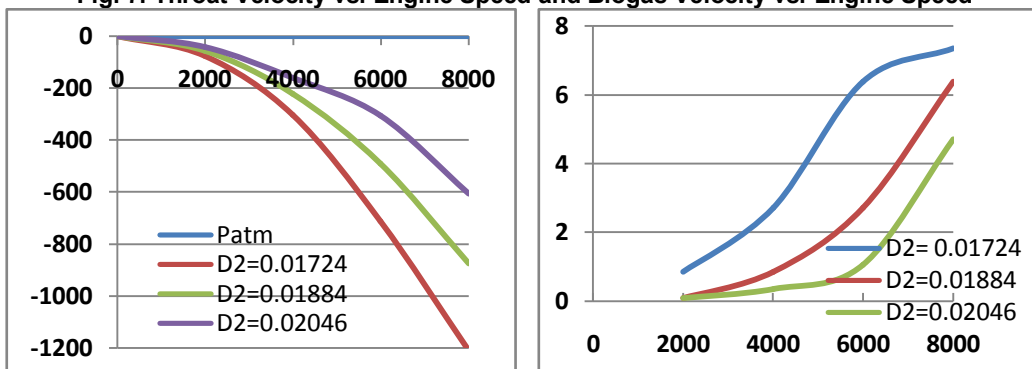**Fig. 7. Throat Velocity vs. Engine Speed and Biogas Velocity vs. Engine Speed**

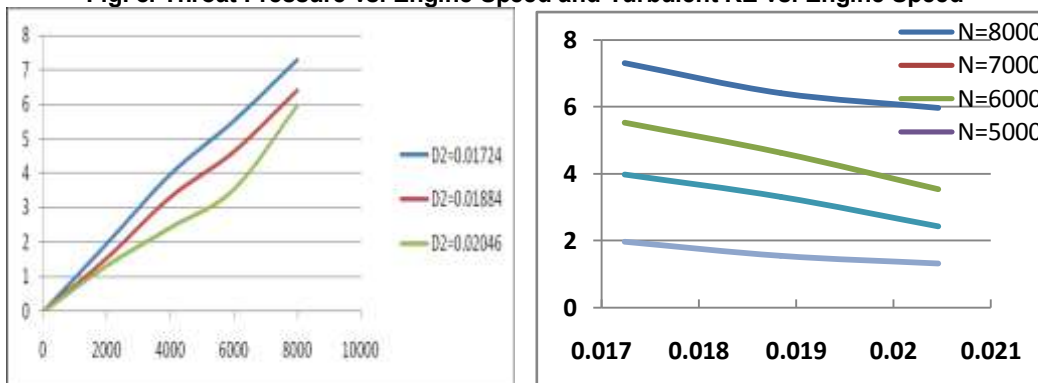**Fig. 8. Throat Pressure vs. Engine Speed and Turbulent KE vs. Engine Speed**

Fig. 9. B.P. vs. Engine Speed and B.P. vs. Throat Diameter

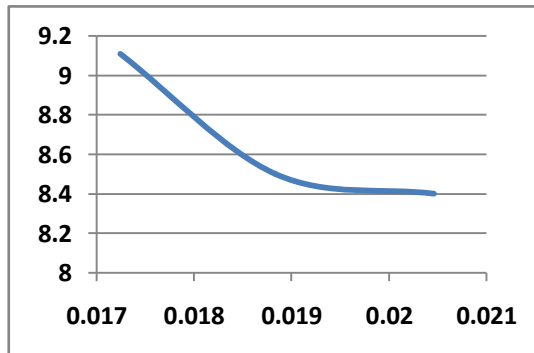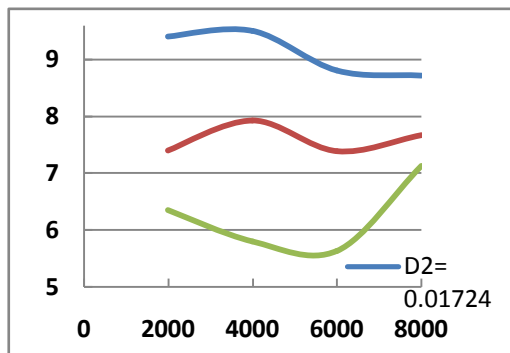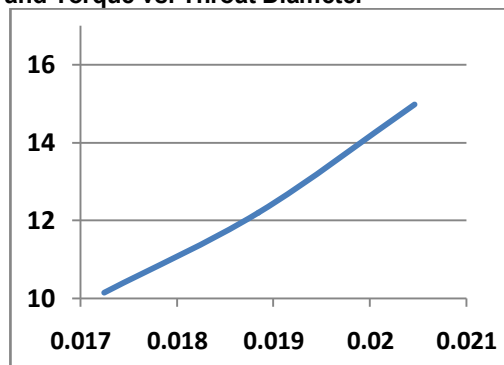**Fig. 10. Torque vs. Engine Speed and Torque vs. Throat Diameter**

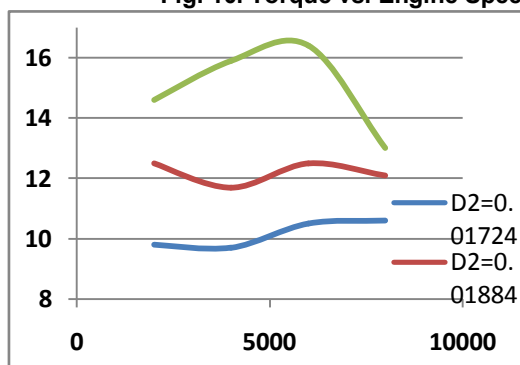
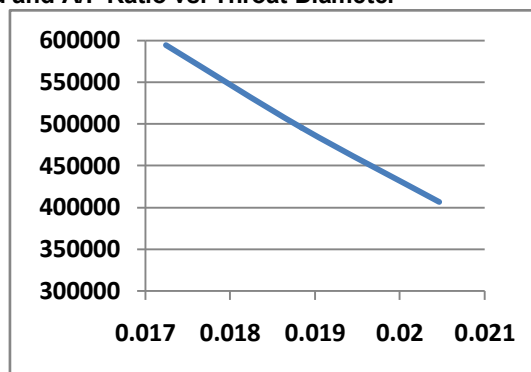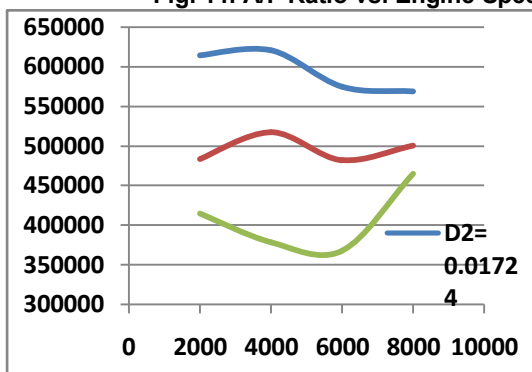**Fig. 11. A/F Ratio vs. Engine Speed and A/F Ratio vs. Throat Diameter**

Fig. 12. Mep vs. Engine Speed and mep vs. Throat Diameter

Conclusion

The study has demonstrated the possibility of implementation flow management strategy in improving the biogas engine performances, through a system called advanced intake system. This approach is simpler and cheaper compared to improvement on combustion chamber. The research provided an example how the flow management strategy may improve the engine performance without major modification. It is shown in the study that the intake system is capable in increasing the engine performance. The combination of throat diameter 17.24 mm, convergent angle 30^0, hole diameter 3 mm and number of holes 4 produced as much as 8 to 15 % increase in power output while 7 to 22 % increase in torque as compared to average power output and torque of few other combinations. The overall implementation of this advanced intake system is proven to improve the power output 11.5% and torque 15.5% compared to average values of the same of other combination. Besides producing cleaner emission, the biogas operation also reduces operational cost on the vehicle. It promises a green and sustainable source of energy. In the overall, the study has provided a simpler, cheaper and effective alternative to improve the biogas engine performance by implementing a pressurized and turbulent mixture proper A/F ratio.

Acknowledgements

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