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# INVESTIGATION ON THE FLOW BEHAVIOUR OF A VENTURI TYPE GAS MIXER DESIGNED FOR DUAL FUEL DIESEL ENGINES

Bhaskor J. Bora[1], Biplab K. Debnath[1], Nikhil Gupta[2], Ujjwal K. Saha[3*], Niranjan Sahoo[4]

*[1] PhD Student, [3]Professor, [4]Associate Professor*
*Department of Mechanical Engineering, Indian Institute of Technology Guwahati, Guwahati 781039, India*

*[2] BTech Student, Department of Mechanical Engineering, Sardar Vallabhbhai National Institute of Technology, Surat ,Gujarat - 395007, India*

* Corresponding author email: saha@iitg.ernet.in

## ABSTRACT
The consumption of energy is increasing at staggering rate around the globe due to a rise in population and its standard of living. The energy is obtained from fossil fuels at the cost global warming. It results several harmful effects for both environmental and human welfare. These impacts can be narrowed down by using alternative renewable energy that is generated from natural resources such as biomass, solar heat and light, wind, rain, tides, waves etc. Dual fuel diesel engines allow the cultivation of alternative fuels for a cleaner combustion. Biogas, a fuel derived from biomass, mainly contains methane ($CH_4$) along with a certain quantity of carbon dioxide ($CO_2$). In a dual fuel engine, biogas can be used as a primary fuel and diesel as a pilot fuel. This requires the necessary modification in the engine that includes the installation of a gas mixer at the inlet manifold of the engine. A properly designed gas mixer for a biogas premixed charge dual fuel engine can result in a proper mixing of biogas with air. This ensures to a better combustion and higher efficiency of the engine. In this paper, the flow behavior in a newly designed gas mixer has been investigated with the help of computational fluid dynamics based software. The contours of pressure, turbulence intensity, velocity and mass fraction of $CH_4$ obtained are discussed, and compared across an existing design. It is observed that the new design of the mixer promotes a homogenous mixing of biogas with air.

**Keywords:** Gas Mixer, Biogas, Dual Fuel Engine, Computational Fluid Dynamics (CFD)

## 1. INTRODUCTION
The interest on an effective utilization of renewable energy sources to produce electrical energy has been increasing. It is not because of energy security perspective but also due to environmental concerns raised by the use of fossil fuels. However, in recent years, the demands to use fossil fuel for producing high quality energy such as electricity are growing. As a result, a significant attention has been paid to this energy source as a fuel for reciprocating internal combustion (IC) engines for power generation in rural areas. Gaseous fuels are attractive, since they have wide ignition limits and capability to form homogeneous mixtures. Moreover, gaseous fuels have high hydrogen to carbon ratios. Thus, very low emissions are possible when they are used in IC engines. Natural gas and LPG are the readily available petroleum-based fuels, while biogas and producer gas are derived from renewable sources [1]. In remote areas, the scarcity of electricity and the abundance of the sources of biogas can make it a potential energy source. From environmental point of view, biogas produces lower $NO_X$ and low soot emissions. It can be produced from cow dung and other animal wastes and also from plant matter which are renewable and abundantly obtainable in the countryside.

It is generated from the anaerobic (out of contact with air) digestion of organic matter-cow dung and leaves. The major constituents of biogas are methane ($CH_4$) and $CO_2$ with approximate concentrations of two third and one third by volume, respectively. It is produced by bacteria, which break down organic materials in the absence of air in a closed container. Biogas can be produced close to the consumption points in rural regions such as engines driving pump sets and generators [2].

Biogas has a lower ignition limit and its dual fuel combustion produces a high burning speed. This makes it easier to ignite with reduction in misfire and thereby improving emissions, performance and fuel economy. The superior energy density of biogas enables it to generate higher power output with improved torque even at lean mixtures and wide-open throttle conditions too [3]. Biogas is typically composed of 40 to 60% of flammable gases (predominantly $CH_4$). The rest consists of inert gases such as nitrogen ($N_2$) and carbon dioxide ($CO_2$) with various minor impurities including hydrogen sulphide ($H_2S$). However, its main composition varies by origin or by the conditions of digestion process, and therefore, it is hard to establish an optimum and a consistent air fuel mixture conditions for a stable engine operation.

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This problem can suitably be overcome by using a gas mixer which would provide a homogenous mixture of air and fuel gas. In an earlier work, a gas carburetor was designed and added to the intake manifold as a means to introduce biogas [4]. The designed mixer had a T- joint with the gas pipe protruding type which was fabricated according to recommendations of von Mitzlaff [5]. However, there was an inherent disadvantage of the gas mixture owing to its geometry and single point fuel injection that restricted the formation of homogenous air-biogas mixture. In this paper, a new design of a gas mixer is modeled and analyzed with the help of computational fluid dynamics (CFD). The concept of the venturimeter is used for the design of the same. The results in the form of the contours of pressure, turbulence intensity, velocity and mass fraction of $CH_4$ are compared across the existing design. The new design of the gas mixture demonstrates is found to provide an improved homogenous mixing of the fuel gas with air as opposed to the existing design.

### 1.1 Dual Fuel Concept

In diesel engines, the fuel is mixed with air towards the end of the compression stroke when it is sprayed into the combustion chamber at high pressure (about 200 bars). At this condition the fuel gets in touch with air at very high pressure and temperature, and starts burning instantly. However, in dual fuel mode, a definite quantity of diesel is pumped by the regular injection system. The engine, however, sucks and compresses a mixture of air and fuel gas, prepared in an external mixing device. The diesel injected actually initiates the burning of the air gas mixture. The amount of diesel fuel that is replaced by fuel gas in dual fuel mode can reach up to a maximum of 85% as compared to lone diesel run [4]. The amount of diesel replacement varies with the type of fuel gases, their composition and engine design parameters. The engine operation at partial load requires a drop of fuel gas supply, which is done by means of a gas control valve. The valve can either be operated manually or automatically by using a mechanical or an electronic system. A simultaneous fall of air supply would however decrease the suction, and hence, the compression pressure and the mean effective pressure. This would lead to a drop in power produced and efficiency. The reduction of intake air and the absence of its proper compression may affect the self ignition of diesel. The dual fuel engines should not be throttled /controlled on the air side. The air/fuel ratio of the sucked mixture can be varied by controlling the fuel gas. It is seen that, even a petit lean mixture still ignites if the diesel is atomized properly. The operating parameters of diesel mode, viz. compression ratio, injection timing etc. can also be remained unchanged in dual fuel mode.

A diesel engine when converted to work under dual fuel operation possesses certain advantages. A dual fuel diesel engine enables the use of fuel gas (obtained from biomass) having lower calorific value, and lowers the running cost of the diesel engine.

During the shortage of fuel gas, the engine can also be run with diesel fuel alone. Any contribution of fuel gas from 0 to 85 % can substitute a corresponding part of the diesel fuel without deteriorating the performance as compared to 100% diesel fuel operation. Further, due to the presence of a governor in most of the engines, the speed/power can be regulated by varying the amount of diesel fuel injection despite the fact that the gas fuel flow is remained constant.

### 1.2 Gas Mixer

For converting a diesel engine to dual fuel mode, the major modification needed is to connect a gas mixer to the inlet manifold [6]. The gas mixer is very important in dual-fuel engine, as it provides a combustible mixture of fuel gas and air in the required quantity and quality for efficient operation of the engine under all conditions [7]. According to the performance required, the flow of fuel gas can be varied. It also enables to supply a sufficient amount of air at maximum load and speed at the actual pressures of fuel gas and air. The maximum air to fuel ratio should not be less than 1.5 in order to ensure combustion even for the pilot fuel. The design of a gas mixer for a particular engine mainly depends on its rated power, specific fuel consumption, speed, and volumetric efficiency, swept volume, and manifold connection diameter [5].

## 2. EXISTING DESIGN OF THE GAS MIXER

In most of the cases, a T-junction gas mixer is used when a diesel engine is modified to a dual fuel mode [5]. The existing design of the gas mixer consists of a channel having a T-junction. It has two inlets, one each for air and gas, and an outlet for air-gas mixture (Fig. 1). The gas inlet is fixed at 90º with air inlet. The exit of the mixture is coupled to the engine intake manifold. However, this simple design creates a huge energy loss when air and gas stream collide with each other at high velocities. In order to reduce this loss of collision and to increase the additional shear mixing, recently, a gas mixer is designed and developed [4]. This gas mixer has a gas inlet inclined at 45º with the air inlet as shown in Fig. 2. However, this design is unable to overcome the asymmetrical mixing of air and gaseous fuel at the mixing region and further down stream. This finally destined the charge to become nonhomogeneous and hence, needs a close investigation.

## 3. PROPOSED DESIGN OF THE GAS MIXER

The proposed design of the gas mixer uses the concept of venturimeter and is known as ‘venturi gas mixer’ (Fig. 3). The venturi gas mixer comprises of two gas inlets, one air inlet and one air-gas outlet. It consists of a smooth contraction section and an expansion section which reduce the prominence of irreversible pressure loss. The function of the converging section is to increase the velocity of the fluid and lower its static pressure. This low pressure region will drag in more gas and enriches the turbulence mixing with air. Thus, there will be a pressure difference created b/w inlet & throat.

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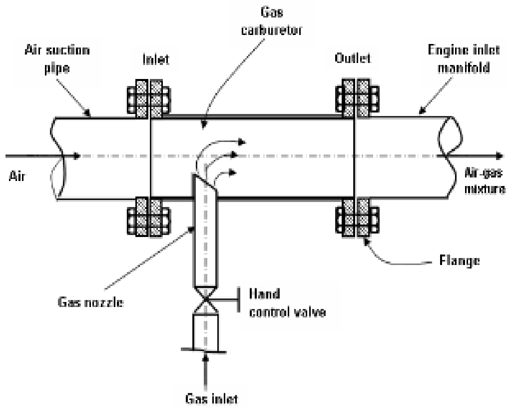


Fig. 1 T-junction gas mixer proposed by von Mitzlaff [5]


Fig. 2 T-junction gas mixer developed by Sahoo [4]

Using Bernoulli equation, this pressure difference can be correlated to the rate of flow as:

$$P_1 - P_2 = \frac{V_2^2 - V_1^2}{2}\rho \qquad (1)$$

Applying continuity equation,

$$A_1 V_1 = A_2 V_2 = Q \qquad (2)$$

$$\Delta P = \frac{A_1^2 - A_2^2}{A_1^2 \times A_2^2}\rho Q \qquad (3)$$

From equation (3), it can be understood that, pressure drop is inversely proportional to the square of the area when all other parameters being kept constant. Hence, at the throat region the pressure and velocity are minimum and maximum, respectively. There are four main designing parameters namely converging angle, diverging angle, nozzle angle and $\beta$ (ratio between the diameters of throat and inlet manifold). The values of converging angle, diverging angle, nozzle angle and $\beta$ are considered as 20°, 5°, 35°and 0.46 respectively as shown in (Fig .4). These geometric values are dictated by our past simulation studies. The throat diameter, inlet manifold and biogas inlet diameter are 16 mm and 34.3 mm and 8 mm respectively. According to Stewart *et al.*, the length of the diverging section should be 10 times the inlet manifold diameter [8]. However, considering the diverging angle of 5° the length of the same is found to be 220 mm with respect to the manifold diameter of 34.3 mm. All this design parameters of the gas mixer is based on a diesel engine with rated power of 3.5 kW. The volumetric efficiency and speed of the engine are considered as 90% and 1500 rpm. The maximum diesel substitution by biogas in this engine is taken as 80% [4].

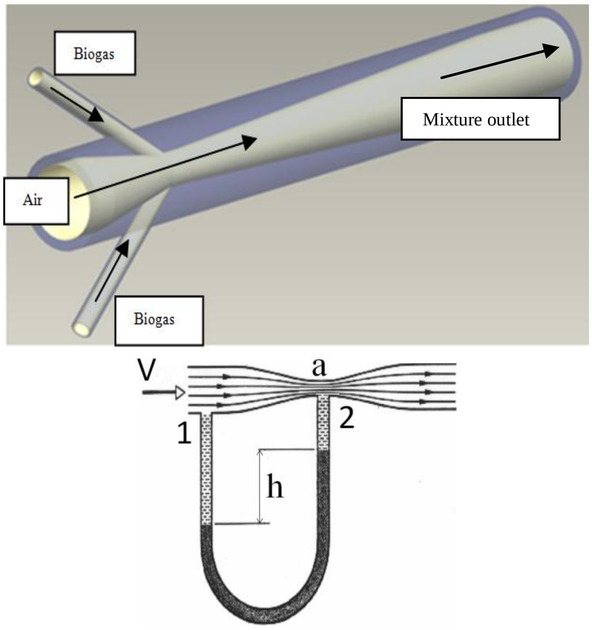


Fig. 3 Proposed design of venturi gas mixer

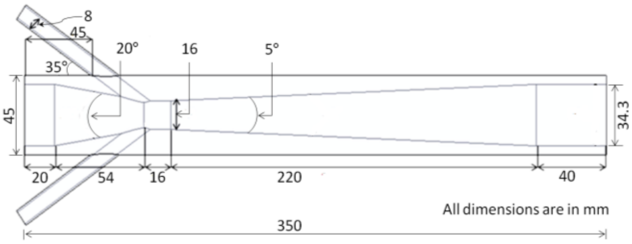


Fig. 4 The schematic diagram of the venturi gas mixer

## 4. METHODOLOGY

The CFD analysis has been carried out in FLUENT software, which is available in ANSYS Workbench 14. The analysis has been performed for the existing gas mixtures with gas nozzles inclined at 45° and 90°, as well as the proposed design of gas mixer. The design of the gas mixers are first modeled in Pro/Engineer and then analyzed in Fluent. The governing equations of the dynamics of air and gas mixture are the conservation and the thermodynamics laws. Simulation is based on the turbulence model of the non-stationary 3D flow. Gases are considered as compressible viscous fluid. The standard $k$-$\varepsilon$ model is used to solve the flow problems reigning inside the gas mixer for the three inlets; one is air inlet and the other two for fuel biogas inlet. The equations governing the inflow model, including the conservation equations of mass, momentum and energy are summarized in the conservative form of the Navier-Stokes equations [9, 10]:

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$$\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial x_i}(\rho\, u_i)=0 \quad (4)$$

$$\frac{\partial}{\partial t}(\rho u_i)+\frac{\partial}{\partial x_i}(\rho u_i u_j)+\frac{\partial P}{\partial x_i} = \frac{\partial}{\partial x_j}(\tau_{ij}+\tau_{ij}{}^{R})+S_{ij}, i=1,2,3 \quad (5)$$

$$\frac{\partial}{\partial t}(\rho\, H)+\frac{\partial}{\partial x_j}(\rho\, u_i H)=\frac{\partial}{\partial x_i}(u_j(\tau_{ij}+\tau_{ij}{}^{R})+q_i) +\frac{\partial P}{\partial t}-\tau_{ij}{}^{R}\frac{\partial u_i}{\partial x_j}+\rho\varepsilon+S_i u_i+Q_H \quad (6)$$

$$H=h+\frac{u^2}{2} \quad (7)$$

where '$S_i$' is an external force per unit mass, 'h' is the thermal enthalpy, '$Q_H$' is a heat source or sink per unit volume, '$S_{ij}$' is the viscous shear stress tensor, '$q_i$' is the diffusive heat flux. The subscripts are used to denote summation over the three coordinate directions. The mathematical form of the energy equation is [8]:

$$\frac{\partial}{\partial t}(\rho\, E)+\frac{\partial \rho u_i}{\partial x_i}(E+\frac{P}{\rho}) =\frac{\partial}{\partial x_i}(u_j(\tau_{ij}+\tau_{ij}{}^{R})+q_i)-\tau_{ij}{}^{R}\frac{\partial u_i}{\partial u_j}+\rho\varepsilon+S_i u_i+Q_H \quad (8)$$

$$E=e+\frac{u^2}{2} \quad (9)$$

where "e" is the internal energy.

For an ideal gas at constant specific heat ratio, $\gamma=\frac{C_P}{C_V}$ the pressure is given by the state law of perfect gases:

$$P=\rho RT=(\gamma-1)\rho \quad (10)$$

### 4.1 Model of Turbulence

The turbulence modeling is found to have more significance while modeling combustion in IC engines [9]. In fact, turbulence directly affects the mixing, the mixture homogenization and combustion in an engine. The adequate prediction of the turbulence behavior is necessary for a better comprehension of these phenomena in order to improve engine performances and to reduce emissions. In the majority of the multidimensional computer codes developed till date have numerous key characteristics of velocity. These are directly related to the scales of turbulence in the models corresponding to the admission, combustion, and transfer of heat and so on [9, 10]. These processes would be modeled correctly if the modeling and the prediction of turbulence are also precise. In this paper, turbulence model is considered according to the k- ε turbulence model [10, 12, 13]. The Reynolds-stress tensor used in this model is defined as [10]:

$$\tau_{ij}{}^{R}=\mu_t(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}-\frac{2}{3}\delta_{ij}\frac{\partial u_k}{\partial x_k})-\frac{2}{3}\rho k\delta_{ij} \quad (11)$$

'$\delta_{ij}$' is equal to unity when i=j, and zero otherwise, 'μ' is the dynamic viscosity coefficient, '$\mu_t$' is the turbulent eddy viscosity coefficient and 'k' is the turbulent kinetic energy. In the frame of the k-ε turbulence model, '$\mu_t$' is defined using, turbulent viscosity factor '$f_\mu$'.

$$\mu_t=\frac{f_\mu C_\mu \rho K^2}{\varepsilon} \quad (12)$$

### 4.2 Boundary and Initial Condition for the Model

The boundary conditions as shown in Fig. 3 are used to represent the flow in the mixer for the CFD analysis. If the boundary conditions are not appropriately defined, the turbulence model will not match for the flow at high speeds [13]. After defining proper boundary conditions, the design of the gas mixer is imported to Fluent for carrying out further simulation. The intensity of turbulence is specified as 3% [9]. The air is assumed to be a perfect gas, and its properties are obtained from the kinetic theory of gases. The species model used to calculate various species formation during combustion is non–premixed combustion model. Initial velocity conditions of biogas and air are considered as 20 m/s and 5.82 m/s; respectively [4].

## 5. RESULTS AND DISCUSSION

The CFD analyses for the three different gas mixer designs are discussed in this section. The designs investigated are the existing gas mixtures with gas nozzles inclined at 45° and 90° along with the proposed venturi gas mixer. The analyses are executed on the basis of pressure, turbulence intensity, velocity, mass fraction of $CH_4$ at different cross sectional planes. The planes are made at equidistant from the entry of biogas into the air stream to find out the optimum design.

### 5.1 Pressure Analysis

Fig. 4 shows the pressure contours of existing T-junction design gas mixer with single nozzle inclined at angle of 45°, 90° and venturi gas mixer. Figs. 5A and B display that, the existing T-junction gas mixers with single nozzle inclined at an angle of 45° and at an angle 90$^o$ give very less pressure drop at the entry. This hampers in effective fuel suction. As a result, the mixing of the fuel gas and air in the gas mixer is not appropriate. In addition, the mixing of the fuel biogas and air is non-uniform as there is a pressure drop at one side of the current T-junction gas mixers with the single nozzle. On the other hand, for the venturi gas mixer, the mixing of the fuel biogas and air would be proper if the pressure drop is uniform throughout the throat section of the gas mixer. However, in the proposed venturi type design as shown in Fig. 5C, the pressure drop at the entry at the throat area is found to be higher as compared to existing gas mixer. This enhanced pressure drop owing to the geometry of the venturi type mixer provides in better suction of the biogas intruding through the small pipes [11]. This led to the superior mixing of the fuel biogas and air which would result a proper combustion of air fuel mixture. Similar patterns of pressure distributions of venturi gas mixer are found in the literature [12, 13].

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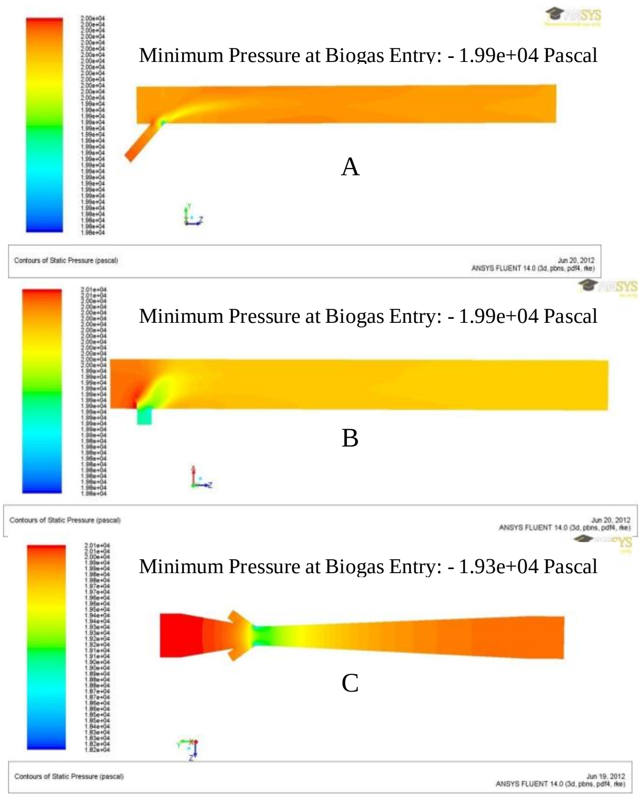


Fig.5 (A) Pressure contour of existing T-junction design gas mixer, single nozzle inclined at an angle of 45°, (B) Pressure contour of existing T-junction design gas mixer, single nozzle inclined at an angle of 90°(C) Pressure contour of venturi gas mixer

### 5.2 Turbulence Intensity Analysis

The contours of turbulence of existing T-junction design gas mixer with single nozzle inclined at angle of 45°, 90° and venturi gas mixer are shown in Fig. 6. In order to have a better combustion, the biogas and air requires to be mixed at a molecular level. The turbulence enhances the mixing of biogas and air. On analyzing the turbulence contour of the various designs, the existing T-junction gas mixers with single nozzle inclined at an angle of 45° and at an angle 90$^{o}$ as illustrated in Fig. 6A and B, exhibits a non-uniform turbulence which results in non-uniform mixing of the biogas and air mixture. Fig. 6C expresses the turbulence contour of the venturi type gas mixture. It can be clearly understood from the figure that a wide zone of uniform turbulence is generated at the diverging segment of the gas mixture. This will cause a healthier mixing of biogas and air, which results a homogeneous combustion and reducing the effective consumption of fuel as obtained by researchers [9, 13].

### 5.3 Velocity Analysis

The contours of velocity and velocity vectors of existing T-junction design gas mixer with single nozzle inclined at angle of 45°, 90° and venturi gas mixer are shown in Fig. 7. Velocity of the mixing species is important from the point of view of fuel air transport which comprises an integral part of combustion. The study of the velocity contours and velocity vectors at entry for every single design revealed that only a solitary narrow velocity stream of the biogas is formed in case of the existing T-junction gas mixers with single nozzle inclined at an angle of 45° and at an angle 90$^{o}$ as shown in Figs. 7A and B. However, as shown in Fig. 7C, for the venturi type gas mixer, the velocity profile of the biogas demonstrates the formation of a wider velocity stream. Moreover, the close observation for the existing T-junction gas mixers with single nozzle portray that, there are zones where the velocity is low adjacent to the mixing zone. The low velocity zone will increase the chances of eddy formation and loss of energy. On the other hand, in case of venturi type gas mixer, there are no such zones. Hence, there will be an enhanced mixing of the biogas with air in case of venturi type gas mixer. This is in good agreement with the results of CNG-air gas mixer [12]. The velocity contour obtained by Gorjibandpy and Sangsereki with 12 holes CNG-air venturi gas mixture also closely matches with the findings of the present study [13].

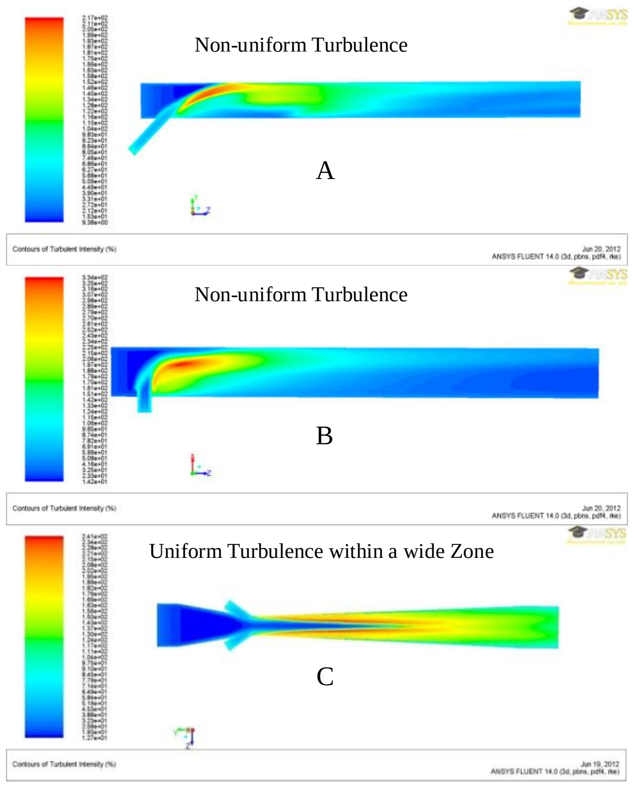


Fig.6 (A) Turbulence contour of existing T-junction design gas mixer, single nozzle inclined at an angle of 45° (B) Turbulence contour of existing T-junction design gas mixer, single nozzle inclined at an angle of 90°, and (C) Turbulence contour of venturi gas mixer

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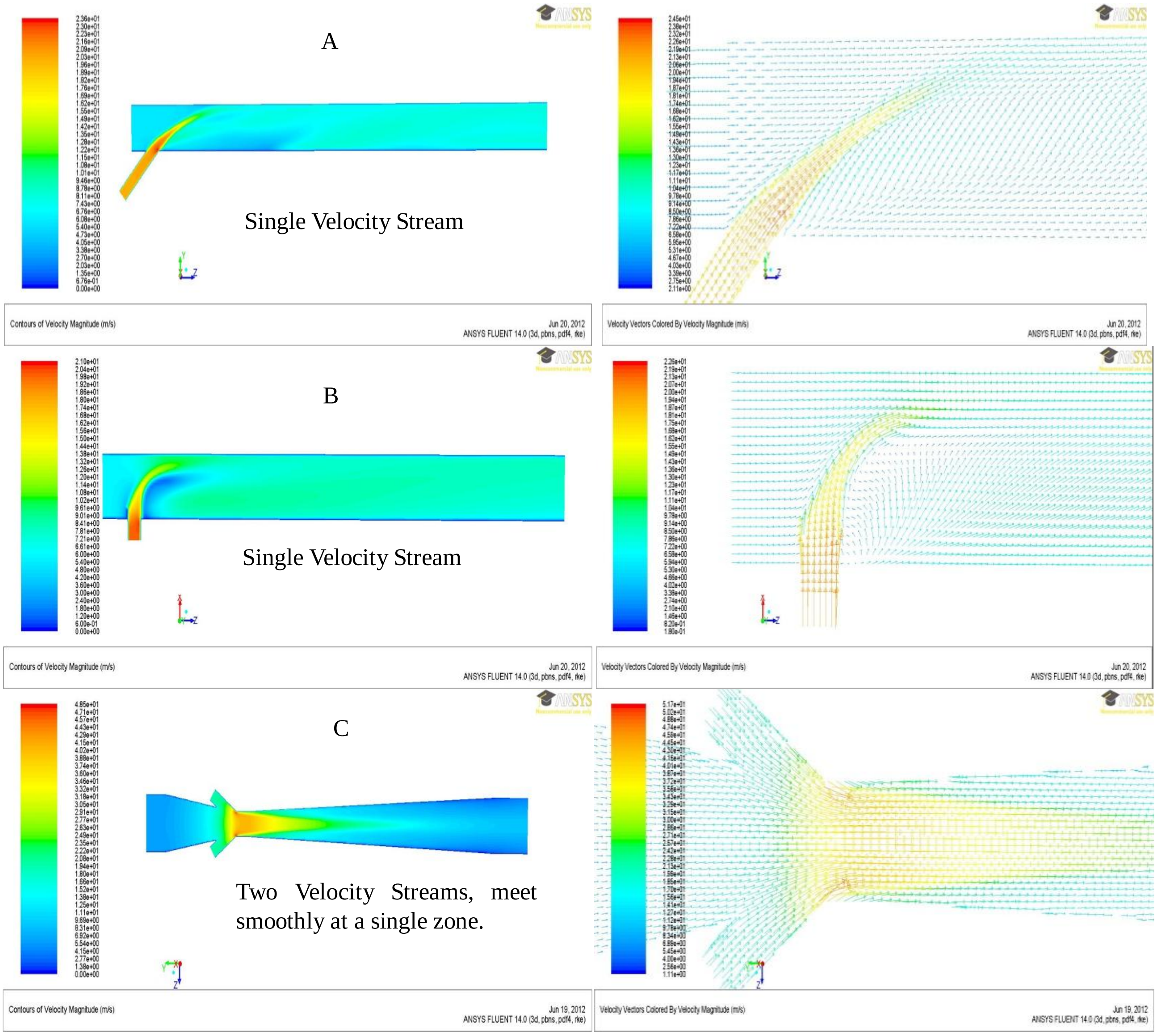


Fig.7 Velocity and Velocity Vector contour of (A) existing T-junction design gas mixer, single nozzle inclined at an angle of 45°(B) existing T-junction design gas mixer, single nozzle inclined at an angle of 90° (C) venturi gas mixer

### 5.4 Analysis of Mass Fraction of $CH_4$

Fig. 8 expresses the contours of $CH_4$ for the existing T-junction design gas mixer with single nozzle inclined at angle of 45°, 90° and venturi gas mixer. For superior combustion, the mixing of the fuel biogas and air should be homogeneous and uniform. The study of the $CH_4$ contours in case of venturi gas mixer indicates that at the throat there are two distinct stream of biogas which slowly disappears in the diverging section and the distribution of biogas turn out to be more uniform as shown in Fig. 8C. Figs. 8A and B illustrate that In case of existing design with single nozzle, the $CH_4$ distribution along the span of the gas mixer is not even, with one side being rich and other side being lean. This trend indicates that the mixing of biogas with air is made more uniform and homogeneous in case of venturi gas mixer than the existing T-junction gas mixers with single nozzle inclined at an angle of 45° and at an angle $90^o$. The contours in Fig. 9 demonstrate the mixing of the biogas with air along the length of all the three types of gas mixer. These contours are produced by making cross section planes at equidistant length starting from entry of the gas mixer. The red and blue zone represents the biogas and air at the inlet in the cross section plane 1. Figs. 9(A) and (B) demonstrate the cross section planes from 1 to 6 for existing T-junction design gas mixers with single nozzle. There are zones where there is no mixing between air and fuel biogas. However, in case of venturi gas mixer, there is a gradual mixing of air and biogas to take place. The mixing neighborhood is progressively develops even though moving from one plane to other as presented in Fig. 9(C). This recognizes an appropriate mixing amongst the biogas and air. Thus, it is clear that the mixing is healthier in case of venturi gas mixer than the designs proposed by von Mitzlaff or Sahoo.

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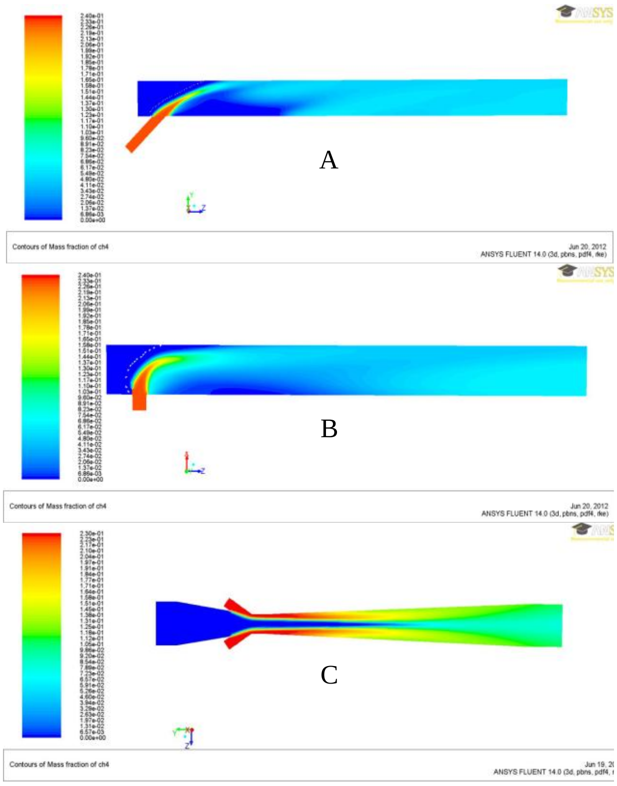



Fig.8 $CH_4$ contour of (A) existing T-junction design gas mixer, single nozzle inclined at an angle of 45° (B) contour of existing T-junction design gas mixer, single nozzle inclined at an angle of 90°(C) venturi gas mixer

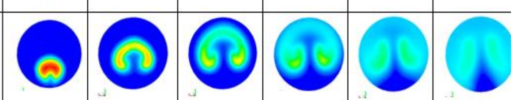
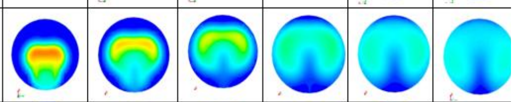
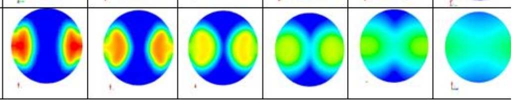
| Models | Plane 1 | Plane 2 | Plane 3 | Plane 4 | Plane 5 | Plane 6 |
|---|---|---|---|---|---|---|
| 1.Existing Design(45°) A |  | | | | | |
| 2.Existing Design(90°) B |  | | | | | |
| 3.venturi (θ1=19°, θ2=5°, β=0.45, θn=35°) C |  | | | | | |

Fig.9 Contours showing the mixing of biogas with air at equidistant length from the entry of the gas mixers

## 6. CONCLUSIONS

The T-junction gas mixer as proposed by von Mitzlaff creates a large energy loss when air and gas stream strike with each other at high velocities. In order to cut this loss of collision and to increase shear mixing, a modified T-junction gas mixer has recently been tested. However, this modified mixer is found to show an asymmetrical mixing of air and gaseous fuel, and needs further attention. Hence, a venturi gas mixer for the dual fuel diesel engine is proposed, and its flow behavior is studied with the help of FLUENT software.

In the present work, CFD analysis of the proposed venturi type gas mixer is carried out on the basis of pressure, turbulence intensity, velocity, mass fraction of $CH_4$. Results obtained are compared with those of earlier designs by von Mitzlaff and Sahoo. It is seen that the venturi type design provides a higher pressure drop at the throat region and ensures a better suction of biogas which ultimately boosts the mixing. This design further provides the flow to be converged at the throat, and thereafter, scatters in the diverging section covering a larger volume with a uniform concentration of $CH_4$. As the cross sectional area increases in the diverging section, the mixing zone of biogas, especially $CH_4$ gradually covers a larger area. The difference between maximum and minimum concentration reduces, thereby ensuring more even distribution of $CH_4$. Thus, the venturi type gas mixer seems to provide far better mixing of fuel biogas and air than the existing T-junction gas mixers developed earlier. Currently, the work is underway to develop a physical model of the mixer and to study its performance in an engine test bed.

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## NOMENCLATURE

*Notations*

| | |
|---|---|
| $A$ | Area ($m^2$) |
| $P$ | Pressure (Pascal) |
| $Q$ | Mass flow rate (kg/s) |
| $T$ | Temperature (K) |
| $V$ | Velocity( m/s) |

*Greek symbols*

| | |
|---|---|
| $\varepsilon$ | Dissipation rate ($m^2/s^3$) |
| $\rho$ | Density of fuel gas –air mixture ($kg/m^3$) |
| $\tau_{ij}$ | Shear stress tensor ($N/m^2$) |

*Subscripts*

| | |
|---|---|
| *1* | Inlet of the Gas mixer |
| *2* | Throat of the Gas mixer |