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Mixture Quality Controlling In DISI Engine Fuelled with Methane

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Abstract
Fuel feeding systems play an important role in mixture formation and combustion quality inside internal combustion engine cylinders as it can determine the mixture quality. There are many types of fuel feeding systems for spark ignition engines starting with the carburetors to the direct injection (DI) systems. In the present paper compressed natural gas was injected directly inside the engine cylinder. A mathematical model based on the conservation of mass, momentum, and energy for unsteady, compressible and turbulent cold flow inside the engine cylinder was used along with the model assumptions. This model is solved numerically on a collocated finite volume grid system and the PISO algorithm is used for pressure-velocity coupling. A great increase in TKE was found.

Keywords: *Gas Direct Injection, SIE, CNG, Cold flow.*

I. INTRODUCTION

(Krishna & Malikarjuna,2011) investigated experimentally the effect of engine speed on in-cylinder tumble flows using PIV. They measured the average velocity of the flow, tumble ratio and turbulent kinetic energy (TKE). The range of engine speed was 400 to 1000 rpm. It was found that tumble ratio (TR) depends mainly on CADs and less affected by the engine speed. In addition, the turbulent kinetic energy increases directly with engine speed. Since the strength of the flow and hence the mixing characteristics are depending directly on the turbulent kinetic energy, the recommend the operation of engines at high speed. **(Mitianiec, 2005)** investigated ignition and combustion process in Spark Ignition engine with direct injection of compressed natural gas (CNG).experiments were performed using one cylinder motorcycle, 4-st engine of the type SUZUKI-Z400S equipped with gas injector. The two operation modes (stratified and homogeneous) were studied. It was found that more than one nozzle injector is needed for a good fuel distribution inside the cylinder. A piston with a bowl was recommended to direct the fuel to the spark plug. Because of higher ignition temperature for natural gas, a high voltage ignition system is required particularly for stratified charge. **(A.Rashid et al., 2010**) experimentally investigated the combustion of CNG direct injection spark ignition engine. Several injection timing parameters were investigated with a centrally located injector. It was found that late DI (120 CAD BTDC) increases the volumetric efficiency of the engine. Early injection(300CAD BTDC) was recommended to achieve a complete mixing of fuel and air at engine speeds above 4500 rpm. (**Bratta et al., 2009**) studied numerically a natural gas jet and mixture formation in DISI engines. They developed a (virtual injector) model on the basis of the calculation results. The model was capable of capturing thermodynamic and kinematic properties at the throat nozzle location with a high accuracy degree. (**Choi et al., 2015**) studied the gaseous fuel injection for CNG direct injection numerically and experimentally. Their developed model can predict the overall tendency of gas fuel injection well. The model can be used for gaseous injection using a coarse mesh which means less computational efforts. It can also be used for simulation of CNG direct injection and combustion. (**Bratta et al., 2009**) investigated Multi-Dimensional Modeling of Direct Natural-Gas Injection and Mixture Formation in a Stratified-Charge SI Engine with Centrally located Injector. The waves propagation

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phenomena which occurs within the injector have a remarkable influence on the injected mass flow rate and on the nozzle jet. It was recommended that new piston and head geometries will be effective in solving the mixture formation problems. (**Mohamad, 2010**) investigated new technique to achieve the compressed natural gas direct injection using spark plug fuel injector (SPFI). This method aims to convert the engine to a DI engine without any modification in its geometry. Results indicate that SPFI performance is slightly lower than those of port injection. This was due to the limitations in air-fuel mixing process inside the cylinder without any modifications in its geometry. (**Kalam & Masjuki, 2011**) investigated experimentally the performance of natural gas engine with direct injection. The results obtained from CNG fuel using two different systems (i.e. bi-fuel and DI) have been compared with the original gasoline engine. CNG-DI engine was found to produce slightly less power than the gasoline fuelled engine and more power compared to CNG bi-fuelled engine. The results indicate that CNG-DI reduces fuel consumption and NOx emissions but increases the CO emissions because of rich fuel regions inside the cylinder. (**Shiga et al., 2002**) studied the Combustion and Emission Characteristics of CNG-DI engine operates using Stratified Combustion mode. After comparing the results of three modes of fuel injection (single side, parallel side and opposed side injection) it was found that the best mixture stratification was achieved by the single injection. Results show that locations and number of injectors and the injection rate slightly affect the combustion and the emissions. CNG-DI can achieve the extremely lean burn (Φ=0.02). (**Zeng et al., 2006**) studied experimentally Combustion characteristics of a direct-injection natural gas engine under various fuel injection timings. Natural gas was injected at constant pressure of 8 MPa using a modified injector. It was found that the advancement of fuel injection timing causes a decrease in the volumetric efficiency because of pressure rise inside the cylinder which reduce the amount of fresh air entering the cylinder. With the advancement of fuel injection timing the equivalence ratio increases. On the other hand, retard fuel injection reduces the engine power due to low quality of mixture and combustion.

II. CFD ANALYSIS

2.1 Governing Equations

Mass and momentum equations

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}\left(\rho(u_j - u_{gj})\right) = 0 \quad (1)$$

$$\frac{\partial \rho u_i}{\partial t} + \frac{\partial}{\partial x_i}\left(\rho(u_j - u_{gj})u_i\right) = \frac{\partial \tau_{ij}}{\partial x_i} - \frac{\partial p}{\partial x_i} + s_i \quad (2)$$

Where

$$\tau_{ij} = 2\mu s_{ij} - \frac{2}{3}\mu\frac{\partial u_k}{\partial x_k}s_{ij} - \bar{\rho}\overline{u_i' u_j'} \quad (3)$$

Turbulence Model equations

$$\frac{\partial}{\partial t}(\rho K) + \frac{\partial}{\partial x_j}\left(\rho K(u_j - u_{gj})\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial K}{\partial x_j}\right] + G_k - \rho\varepsilon \quad (4)$$

$$\frac{\partial}{\partial t}(\rho\varepsilon) + \frac{\partial}{\partial x_j}\left(\rho\varepsilon(u_j - u_{gj})\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + \rho C_1 S\varepsilon - \rho C_2\frac{\varepsilon^2}{K+\sqrt{\nu\varepsilon}} \quad (5)$$

Energy Equation

$$\frac{\partial}{\partial t}(\rho E) + \frac{\partial}{\partial x_j}\left[\rho(u_j - u_{gj})(E + p/\rho)\right] = \frac{\partial}{\partial x_j}\left(K\frac{\partial T}{\partial x_j}\right) - \frac{\partial}{\partial x_j}\left(\sum h_r J_r\right) + \frac{\partial}{\partial x_j}\left(u_i\tau_{ij}\right) + S_E \quad (6)$$

Species Transport equation

$$\frac{\partial \rho Y_i}{\partial t} + \frac{\partial \rho(u_j - u_{gj})Y_i}{\partial x_j} = \frac{\partial}{\partial x_j}\left(D_{eff,i}\frac{\partial Y_i}{\partial x_j}\right) + s_i \quad (7)$$

where

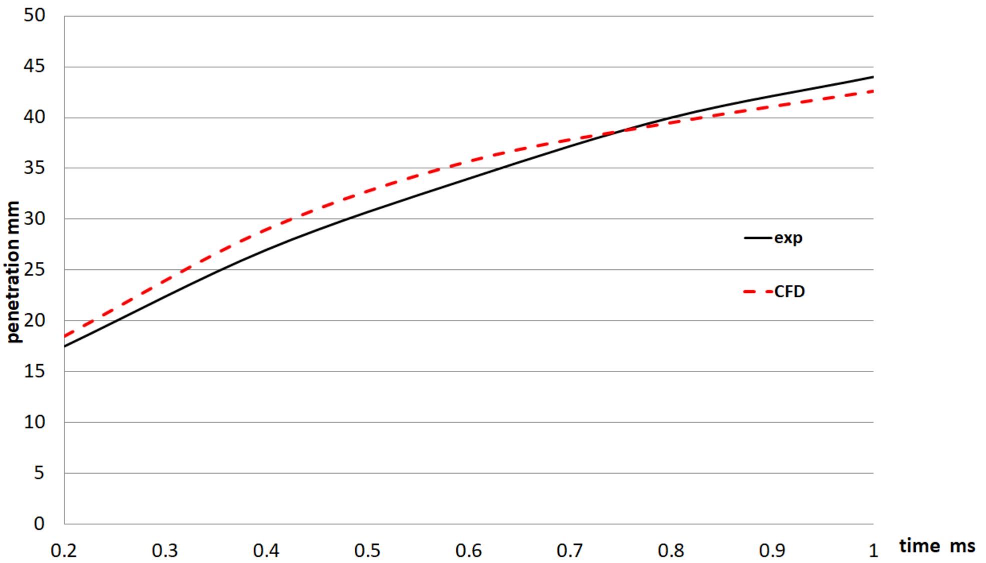
$D_{eff,i}$ is the effective diffusion coefficient and can be calculated as :

$$D_{eff,i} = \rho D_{im} + \frac{\mu_t}{Sc_t} \quad (8)$$

2.2 Validation

The previous mathematical model was validated with a set of experimental data to determine the possibility of using the model in the next predictive studies that will be done using the engine geometry in table(1). The validation case was done with the experimental results (**Ouellett & Hill, 2000**). Simulations were compared to flow visualization studies presented by them. The flow visualization experiments consisted of schlieren photographs of transient methane jets issued from the round nozzles of a gaseous injector used in engine. The injection took place in a pressurized optically accessible. The nozzle diameter was 0.5 mm. The penetration rate was measured from the photograph with an accuracy of 0.1 ms on time. The next figure presents numerical results along with experimental results of the jet penetration along with the time. It is obvious that there is an acceptable difference between them and there is a good agreement. So, mathematical model will be used in a second validation case with a real engine conditions.

***Fig. 1:** jet penetration comparison.*



2.3 Geometrical Modeling and Meshing

The following table shows the details of the engine used in the simulation

***Table 1:** engine geometry details*

Parameter	value
bore	87.5 mm
stroke	110 mm
speed	2000 rpm
Intake port diameter	32 mm
Exhaust port diameter	30 mm
Compression ratio	10 : 1
Valve seat angle	30 deg

2.4 Injection parameters

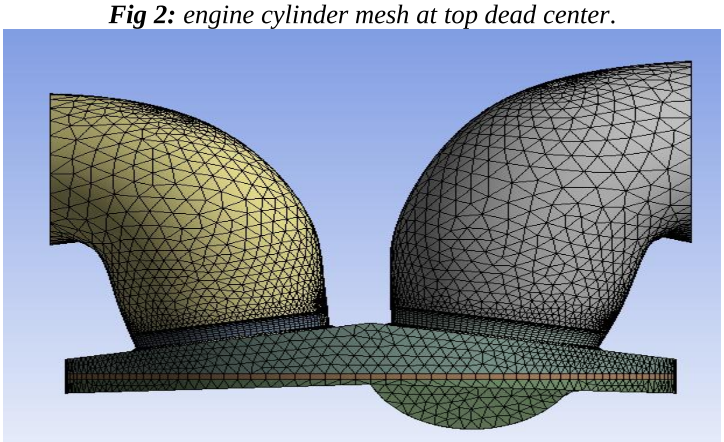
Table 2: injection details

Parameter	value
Injection hole diameter	2 mm
Start of injection	615 CAD
End of injection	655 CAD
Global Equivalence Ratio ϕ	0.666
Gas type	pure Methane CH_4
Start of injection	621.625 CAD
End of injection	655.125 CAD
Injection pressure	50 bar

2.5 Dynamic Mesh

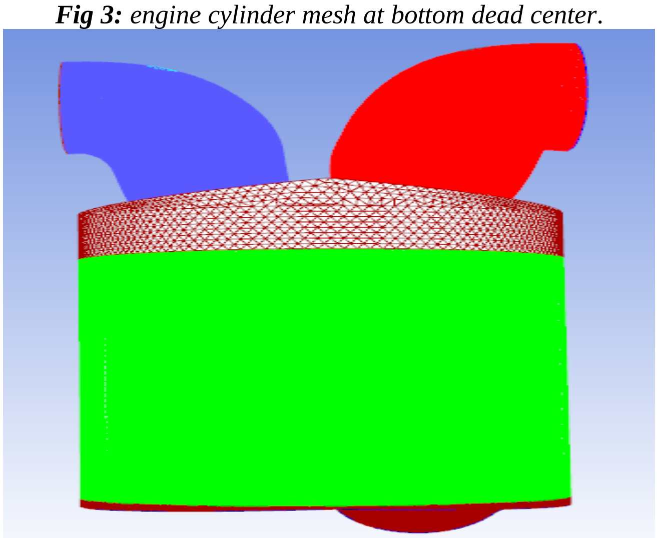
The technique used to solve was Ansys 19 commercial software (fluent IC engine) by dynamic mesh. The dynamic mesh model in ANSYS Fluent can be used to model flows where the shape of the domain is changing with time due to motion on the domain boundaries. The dynamic mesh model can be applied to single or multiphase flows (and multi-species flows). The generic transport equation applies to all applicable model equations, such as turbulence, energy, species, phases, and so on. The dynamic mesh model can also be used for steady-state applications, when it is beneficial to move the mesh in the steady-state solver. The update of the volume mesh is handled automatically by ANSYS Fluent at each time step based on the new positions of the boundaries. To use the dynamic mesh model, you need to provide a starting volume mesh and the description of the motion of any moving zones in the model. If the model contains moving and non-moving regions, you need to identify these regions by grouping them. Three groups of mesh motion methods are available in ANSYS Fluent to update the volume mesh in the deforming regions subject to the motion defined at the boundaries: Smoothing Methods, Dynamic Layering and Remeshing Methods [11,12]. The next figure shows the mesh at the beginning of suction stroke at the top dead center. At this point the number of mesh was about 842000 cells.

Fig 2: *engine cylinder mesh at top dead center*.



The next figure shows the mesh at the end of suction stroke at the bottom dead center. At this point the number of mesh was about 1100000 cells.

Fig 3: *engine cylinder mesh at bottom dead center*.

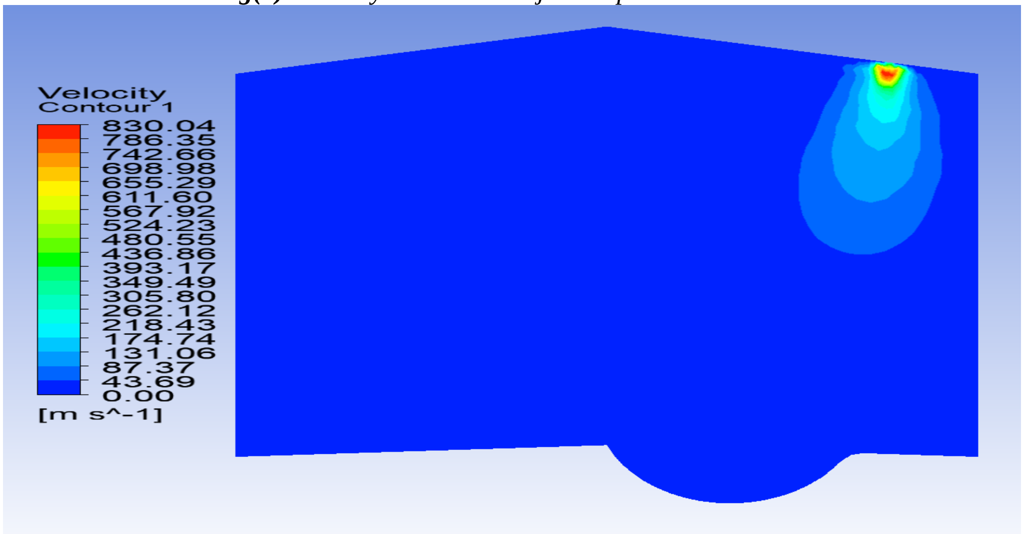


III. RESULTS AND DISCUSSION

3.1 Velocity contours at injection plane after injection:

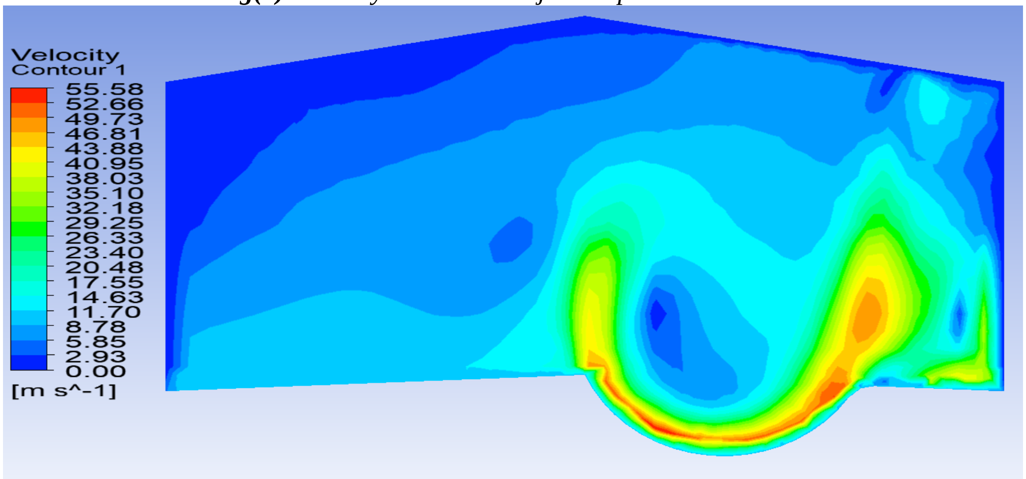
Figures(4-8) show the velocity contours at injection plane after injection start position. At fig(4) the very high velocities resulting from injection can be ssen. Velocity exceeded 800 m/s at the region neal the fuel inlet hole. So, velocity values inside the cylinder in remaining regions can be neglected. Fuel enter the cylinder forms a jet which is perpendicular to the cylinder head and parallel to the tangent of the piston curvature.

Fig(4): *velocity contours at injection plane at 630 CAD*



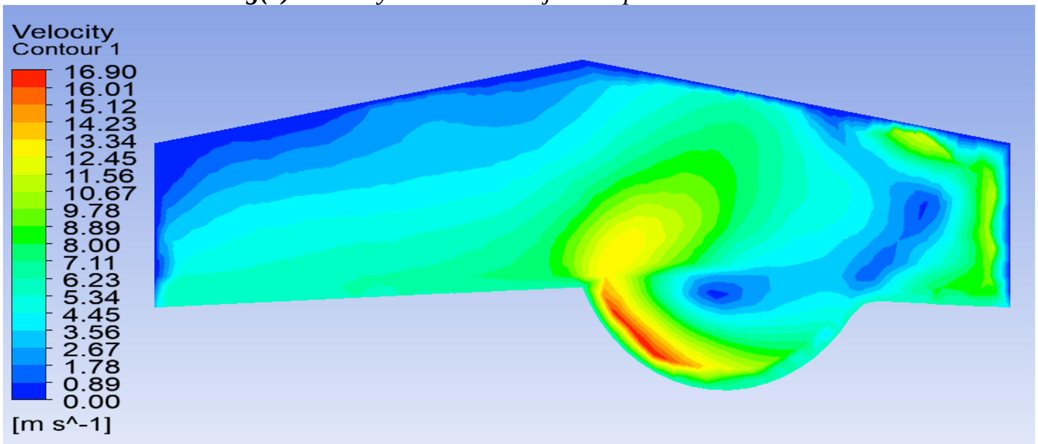
At fig(5)injection has ended. So, the velocity values decreased to about 55 m/s. the fuel jet hit the piston curvature and deflected upwards to form a large CW vortix just below the inlet valve.

Fig(5): *velocity contours at injection plane at 660 CAD*

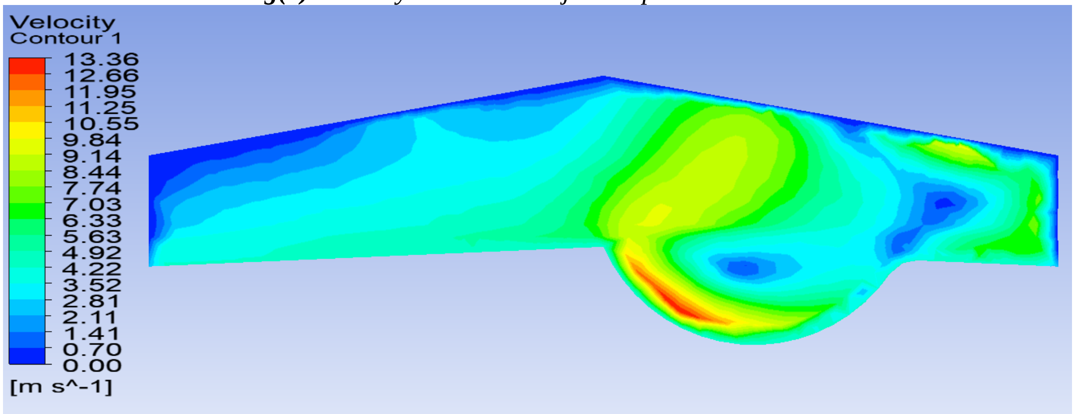


In the next three figures, the flow behaviour has almost the same characteristics. Piston movement upwards continues to comperss the mixture, increasing pressure , denisty and reducing velocity values. The CW vortix take the whole cylinder space.

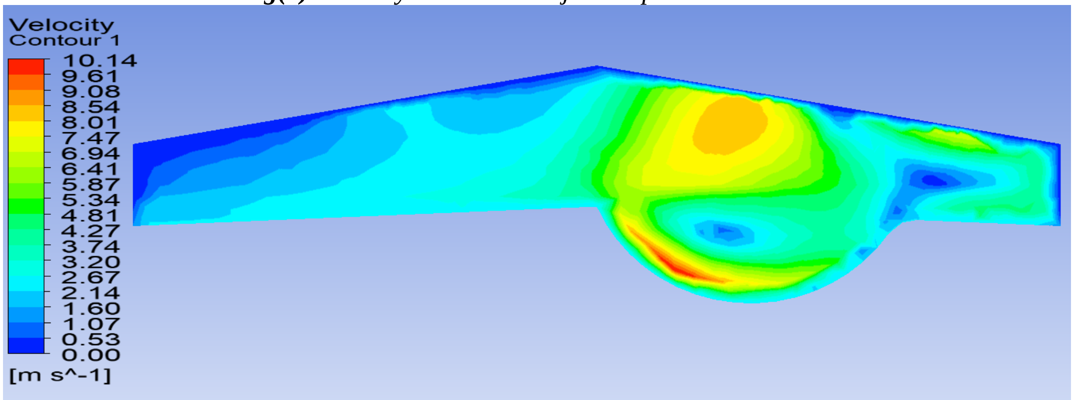
Fig(6): *velocity contours at injection plane at 690 CAD*



Fig(7): *velocity contours at injection plane at 700 CAD*



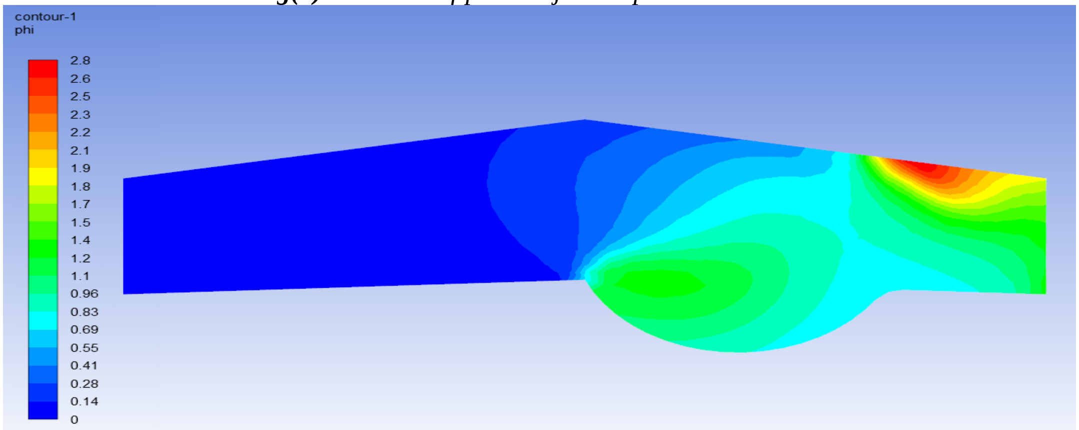
Fig(8): *velocity contours at injection plane at 710 CAD*



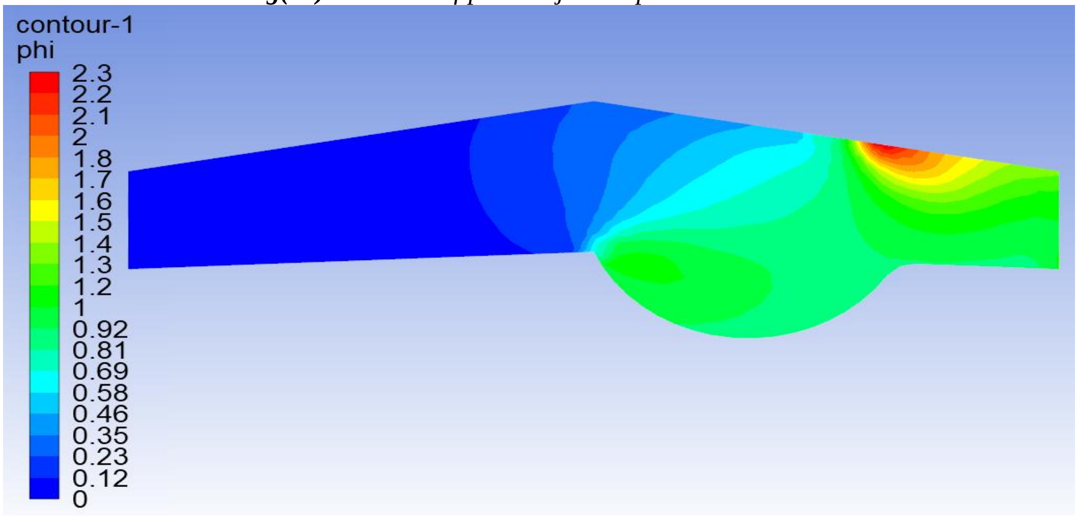
3.2 Contours of the equivalence ratio Φ at injection plane before ignition:

Figures(9-12) show the contours of equivalence ratio Φ at vertical and horizontal planes before ignition point.Fig(9) shows briefly the concept of the stratified charge. The concentration of the fuel in the right part specially in the top right corner make a very rich regions then the mixture become leaner at the left side of the cylinder to be almost only air at the left corner of the cylinder.

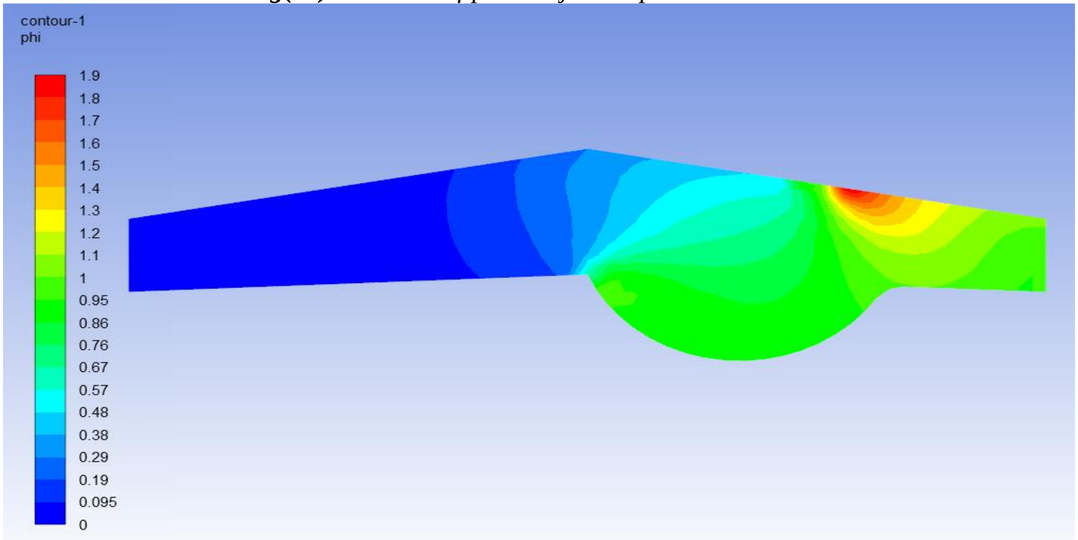
Fig(9): *contours of phi at injection plane at 690 CAD*



Fig(10): *contours of phi at injection plane at 700 CAD*

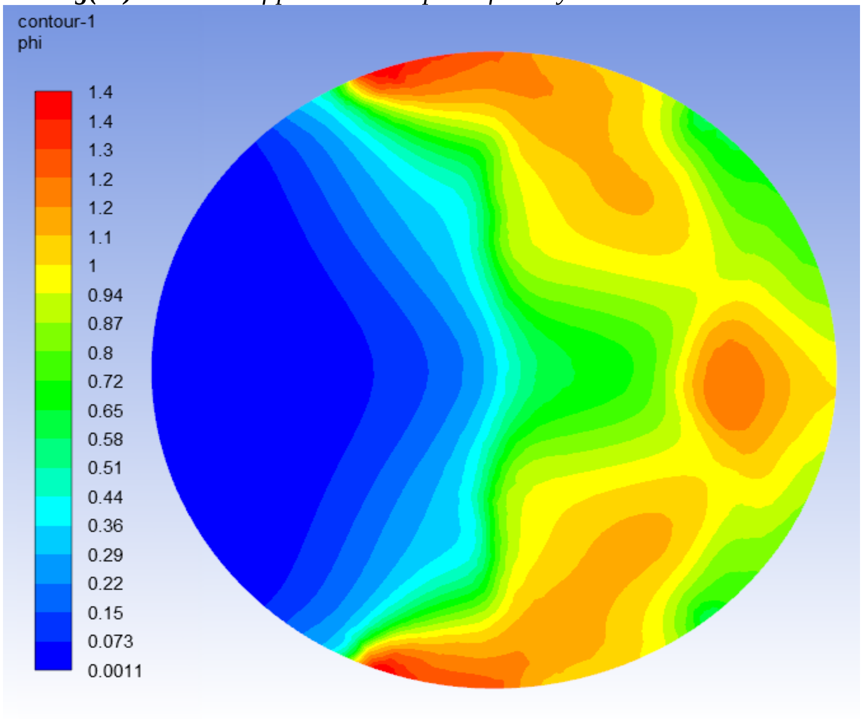


Fig(11): *contours of phi at injection plane at 710 CAD*



At fig(12) it can be noticed that the fuel continues to spread in the cylinder towards the left side. Mixing process at this CAD is better than the previous positions because of the time available for mixing.

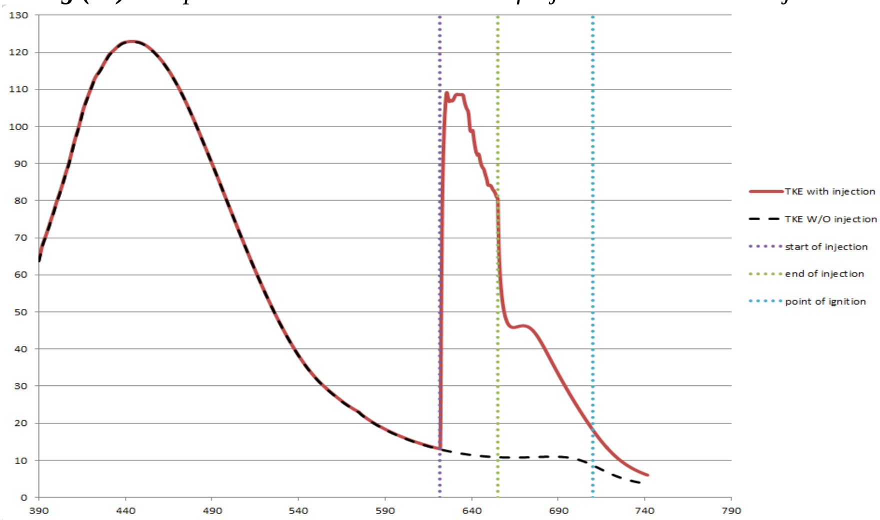
Fig(12): *contours of phi at 10 mm plane from cylider head at 710 CAD*



3.3 Effect of injection on TKE

The next figure indicates the change in TKE due to injection. At the start of injection, TKE jumps to reach level near its maximum value during suction process. Then it decreases steadily at injection duration because of the piston upwards movement. Then it dramatically decrease at the end of the injection because of the vanish of the fuel jet. But it still higher than its value in case of no injection. TKE jumps with an increase of 107 % of its original value in case of no injection at the ignition point.

Fig (13). *comparison between TKE in case of injection and without injection*



IV. CONCLUSION

In this paper, a CFD analysis has been carried in a DISI engine to study in-cylinder flows and air-fuel interactions. From the analysis, the following conclusions are drawn:

1) The lean limit *A/F* was improved greatly by direct gas injection system

2) The lean limit *A/F* was extended by not only increasing the direct injection ratio but also by setting the injection timing more close to TDC. This means that stratified mixture is effective in extending the lean limit *A/F*.

3) The percentage of increase in the turbulent kinetic energy due to Gas DI inside the SIE reached 107 % , which mean a better mixing before combustion.
4) The curved piston plays an important role in the mixing process and gas motion inside the engine cylinder to reach the spark plug region.
5) The most appropriate position for the spark plug between the two intake valves near the injection hole.

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