Journal of Basic and Applied Engineering Research
Print ISSN: 2350-0077; Online ISSN: 2350-0255; Volume 2, Number 6; April-June, 2015 pp. 474-480

http://www.krishisanskriti.org/jbaer.html

Simulation of Combustion in Spark Ignition Engine

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Abstract—*In an Internal Combustion Engine cyclic variations in the combustion process result into varied air fuel mixture within the cylinder, particularly in the vicinity of spark plug. These leads to incomplete burning of fuel, thus reducing the combustion efficiency and responsible for knocking. Monitoring of air fuel ratio may help in improving combustion conditions leading to engine efficiency.*
In the present work simulation studies are carried out on the in-cylinder conditions of a four stroke single cylinder spark ignition engine using the combustion model in ANSYS Fluent. The Simulation is confined to phases between IVC to EVO to reduce the computational time. The physical running conditions of the engine are recorded and used as input parameters for modeling the simulation. The simulation is run for a particular air fuel ratio by defining the mean mixture fraction and variance. The Equivalence Ratio of the engine is calculated based on its running condition, using this equivalence ratio simulation has been carried out and the peak temperature and pressure are evaluated. These pressure and temperature are compared with the real time values.

Keywords: *IVC(Inlet Valve Closed), EVO (Exhaust Valve Open) , Equivalence Ratio, Mean Mixture Fraction, Variance.*

1. INTRODUCTION

Pollution by exhaust gasses produced is growing due to increased number of vehicles. One possibility to reduce the emission is to use alternative fuels like hydrogen. It will take a long time to switch a whole economy from oil to an alternative. A solution for the short-term is to make engines cleaner and more efficient. Performing experiments on an internal combustion engine is expensive and carrying out flow measurements inside the cylinder is a complex task. Because of the technological developments and high end computers, the processes in an internal combustion engine can be modeled in detail.

Many Researchers had contributed their efforts by using simulation as a tool for studying the performance of IC engines in many directions. Maher A. R. Sadiq[1] established a simulative model for the performance parameters of spark ignition engines fueled with a range of fuels (gasoline, ethanol, or hydrogen). The mathematical and simulation model has been developed, tested, and verified against the experimental data to simulate a 4-stroke cycle of a spark ignition A good agreement was obtained between the results of the present model and the experimental results. Ashish J.Chaudhari [2] used Simulink to test the performance of spark ignition engine. The CFD results with the validated Simulink model for the engine configuration shows that at low speeds (1000 rpm), the maximum cylinder pressure prediction is about 8% higher for CFD analysis while this deviation is seen to be about 3% at higher speed (3600rpm). The Simulink model is subsequently used to test the predications of brake power and subsequently compared with the experimental results and CFD studies. Osama H. Ghazal [3] developed a simulation model for one-cylinder spark ignition engine model with port fuel injection and investigated the combustion process employed in a two zone combustion model. The empirical heat release functions are derived from the Woschni heat transfer model. Kota Sridhar [4] In his study, simulated the Intake, Compression, Expansion and Exhaust process with hexane fuel combustion. Mehrnoosh Dashti [5] used a two-zone model for simulation of the combustion process and the species including CO2, CO, H2O, H2, N2, O2 , NO and UHC were considered as exhaust gases. This analysis showed the capability of the model to predict engine performance characteristics over the various ranges of engine parameters. . Chintan R. Patel [6] Analyzed the four basic processes taking place in an S.I. engine and the values of pressure and temperature at every 2° of crank rotation are found out with the aid of certain assumptions. The simulation of suction pressure using discrete approach suggests that the pressure 30º after TDC and 30 º before BDC are of the order of 0.856 bar and 0.931 bar, respectively. This clearly indicates that the entry of hydrogen during this period will certainly offer back fire free operation of the engine. Reinhard Tatschl [7] studied the impact of the turbulent in-cylinder flow on the cycle-resolved flame propagation characteristics. Results of the application of the developed methodology demonstrate its ability to reflect the governing impact of the cycle specific in-cylinder flow field on the flame kernel formation and subsequent main combustion process. Hai-Wen Ge [8] implemented an efficient multigrid (MG) model for spark-ignited (SI) engine combustion modeling using detailed chemistry. The model is designed to be coupled with G-equation model for flame propagation (GAMUT combustion model) for highly efficient engine simulation. The model was explored for a gasoline direct-injection SI engine with knocking combustion. A two-zone MG model, which treats the unburned and burnt regions separately, was able to reproduce the results of the original GAMUT combustion model and large reduction in computer time was seen. Yang Bai [9] developed a simulation package for a port injection spark-ignition engine and included engine dynamics, vehicle dynamics as well as driving cycle selection module. The simulation results are very close to the data obtained from laboratory experiments. The PID control and fuzzy control methods have been combined into a fuzzy PID control and the effectiveness of this new controller has been demonstrated by simulation tests. The above contributions show the reliability of the simulation tools.

2. COMBUSTION PHENOMENA AND MECHANISM IN SPARK IGNITION ENGINE

Combustion, with rare exceptions, is a complex chemical process involving many steps that depend on the properties of the combustible substance. It is initiated by external factors such as heat, light, and sparks. The reaction sets in as the mixture of combustibles attains the ignition temperature. The combustion spreads from the ignition source to the adjacent layer of gas mixture; in turn, each point of the burning layer serves as an ignition source for the next adjacent layer, and so on. Combustion terminates when equilibrium is achieved between the total heat energies of the reactants and the total heat energies of the products. Most reactions terminate when what is called thermal equilibrium has been attained.

In a spark ignited (SI) engine, a flame front is formed which moves outwards from the ignition point, consuming the available fuel air mixture. Turbulence again plays a significant role in flame propagation, since the flame moves at the turbulent flame speed. Hence, if the turbulence levels are high, the flame front will move more rapidly to all parts of the combustion chamber. For SI engines, the rapid flame propagation avoids knock due to auto-ignition of fuel air ahead of the flame. The flame speed depends on the air fuel ratio of the mixture. If the mixture is outside of the flammability limits, usually between equivalence ratios of 0.5 and 4, the flame will not propagate and the engine will misfire. Similarly, if regions exist inside the cylinder that are outside of the flammability limits, these regions will not burn and will most likely be pushed out through the exhaust and into the atmosphere. Combustion produces a rise in pressure and temperature as the energy contained in the fuel is released and the chemical reaction is completed. The fuel combustion produces a spike in pressure and temperature as the energy contained in the fuel is released, with the production of exhaust gases. Some of the energy is radiated and convected to the cylinder walls, cylinder head, piston and the valves; and is lost. Most of the energy goes into the power stroke, where the exhaust gases expand under high pressure and push the piston down to the bottom center position. A thermodynamic energy balance shows that the energy produced due to combustion is used for work done due to expansion, while the thermal losses includes heat losses through the walls and the enthalpy of the exhaust gases at high temperature. During the subsequent exhaust stroke, the exhaust gases are pushed out through the exhaust valves, which start opening towards the end of the power stroke. This process involves formation of a high speed, high temperature jet in the gap between the exhaust valves and ports. During combustion, the fuel, which is a long chain hydrocarbon, breaks up into smaller molecules. The carbon and hydrogen contained in these molecules combine with the oxygen in the air in exothermic reactions. If the fuel air ratio is stochiometric at each location in the combustion chamber, carbon dioxide and water are formed. However, if the fuel air ratio is rich at particular locations due to inadequate mixing, the oxygen molecules are not sufficient and the combustion will be incomplete. Here carbon monoxide (CO) and unburnt hydrocarbon molecules will be produced.Thus, the combustion efficiency of the engine and pollutant formation depends on the fluid dynamics of swirl, tumble, mixing, and turbulence production during the intake and compression strokes, losses due to incomplete combustion, the heat transfer losses to the wall, and the exhaust losses. This paper explains the assessment of air fuel ratio under running condition of a 4 Stoke SI Engine (TVS FIERO-FX) and utilizing the same for simulating the compression and power strokes using FLUENT 15.0.

3. METHODOLOGY

The Simulation is run on a Single Cylinder Spark Ignition Engine with the following Specifications.

Table 1: Specification of the Engine

Item	Specification
Make	TVS FIERO FX
Volume of Engine	147cc
Compression Ratio	9.4
Brake Horse Power	8.8 KW
Cylinder Bore	58mm
Stroke Length	72mm

Theoretical Average Peak Temperature in the cylinder is about 1800K [10]

3.1 Process Compression in Ansys Fluent

IC Engine simulations require process compression tools and automation to reduce problem setup time, automate solution runs, and post-processing of large data sets. In the past, geometry and meshing, solution, and post-processing were performed in different software running independently, with no interaction between them. This meant that each simulation had to be set up completely from the beginning, even when simple design changes were made. With a complex problem setup, any simple user error at any stage has the potential to derail the entire simulation. Thus the previous process is inherently time consuming and error prone. Process compression and automation can only be accomplished in an integrated environment where the software at each step is aware of the overall goals of the simulation and shares a common problem description. ANSYS Workbench provides an ideal integrated environment with powerful tools for geometry, meshing, CFD solvers, and post-processing available on a common platform.

Table 2: Input Parameters for Simulation

Parameter	Value
Connecting Rod Length	144.3mm
Crank Radius	45mm
Engine Speed	2000rpm
Valve Lift	0.2mm
Injection Position (X,Y,Z)	0,-2E-5,-0.00012 (meters)
Injection Axis (X,Y,Z)	0,-0.6562,-0.7545 (meters)
Spark Location (X,Y,Z)	0,6,-0.001 (meters)
Spark Energy	0.2J

Table 3: Boundary Conditions

Boundaries	Temperatures
Inlet Valve	300K
Exhaust Valve	300K
Internal Piston	500K
Cylinder Head	300K

3.2 Design Modeler

The engine model is developed using the geometry and mesh of Cold Flow Simulation inside an SI engine that was available in ANSYS. The existing model is modified using the actual engine specifications by defining the input parameters as shown in the Table 2. Since the simulation is confined to the phases between IVC to EVO, the motion of the valves is not required hence they are omitted and the model before and after decomposition is as shown in the Fig. 1.

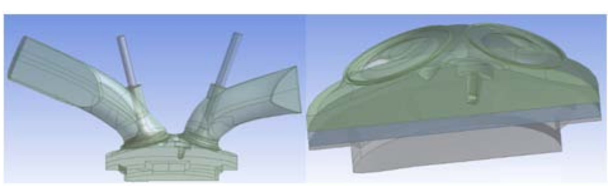

Fig. 1: (a) Before Decomposition (b) After Decomposition

3.3 Meshing

The decomposed geometry is used to generate the mesh. The goal of the IC Engine meshing tool is to minimize the effort required to generate a mesh for the IC Engine specific solver. It uses the named selection created in the decomposition to identify different zones and creates the required mesh controls.

The meshing process for **Full Engine IVC to EVO** combustion simulation is almost similar to the cold flow meshing process. Since only the chamber part needs to be meshed the mesh settings are slightly altered. The combustion chamber after meshing is shown in the figure:2

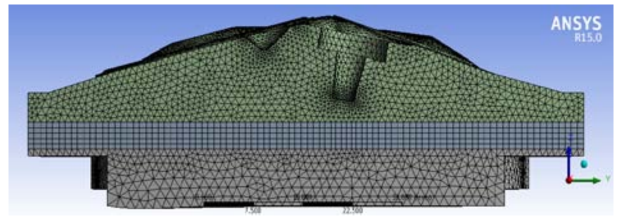

Fig. 2: Meshed View of the Combustion chamber

3.4 Solver Setup

In the solver set up we define the engine type and the species model based on our requirement and we define the combustion chemistry in which G-Equation is defined as premix combustion model. Injection parameters are set to Rosin Rammler distribution for keeping the injection close to practical. The spark location and the Boundary conditions are defined as per the data provided in the Table 2 and Table 3. In order to check the progress of combustion process at various time steps during the simulation temperature and Pressure monitor have been defined along with some preliminary monitors to check the swirl and tumble inside the cylinder. In order to define the air fuel ratio at which the combustion has to be simulated the mean mixture fraction has to be defined.

Mean Mixture Fraction (f) = $\frac{\varphi}{\varphi + r}$; where φ= Equivalence ratio; r = Air Fuel Ratio

Based on the Experimental Results, the actual A/F ratio at which the engine is running is found to be 17.43; The range of Equivalence ratio (φ) is 0.2 to 1.4 [11].

4. RESULTS AND DISCUSSIONS

The simulation of the combustion process is set from IVC to EVO, for a total of 3520 time steps and for each time step a total number of 50 iterations have been set. For the given set of equations the simulation is run and the results have been plotted in the form of graphs and contours as shown in the figures below.

Fig. 3 shows the mass average static temperature plot is about1700K. The temperature plot show that the current model plots are within the range of real time engine . Figures 4 show the swirl and tumble ratio with respect to crank angle and is found to be uniform throughout the cycle. Fig. 5 to Fig. 19 show the velocity magnitude inside the cylinder for 250 crank angles i.e) from inlet valve close to exhaust valve open.

In actual , inlet valve closes at 250^{o} and exhaust valve opens at 495^{o}. Since the simulation is started at IVC the crank angle at IVC is taken as 0^{o} and EVO at 250^{o}. The temperature plot shows 9% variation from that of the theoretical value.

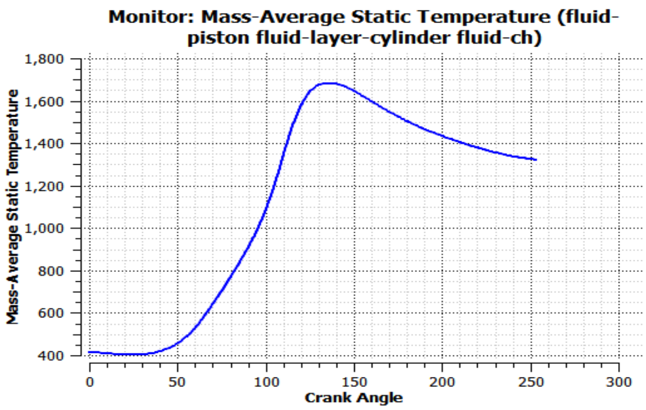

Fig. 3: Temperature Plot

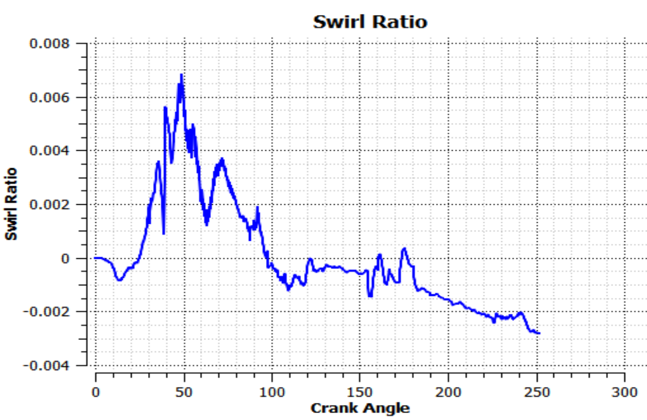

Fig. 4: Swirl ratio

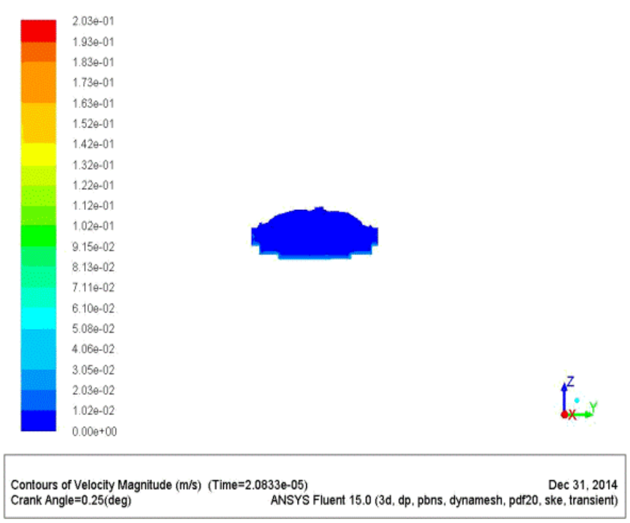

Fig. 5 Velocity Magnitude at 0.25 CA

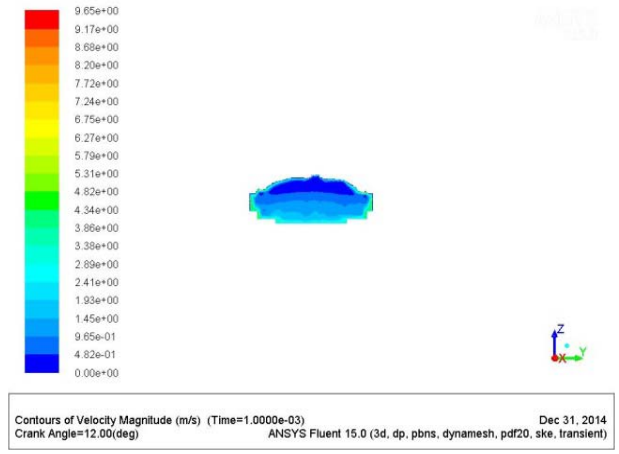

Fig. 6 Velocity Magnitude at 12 CA

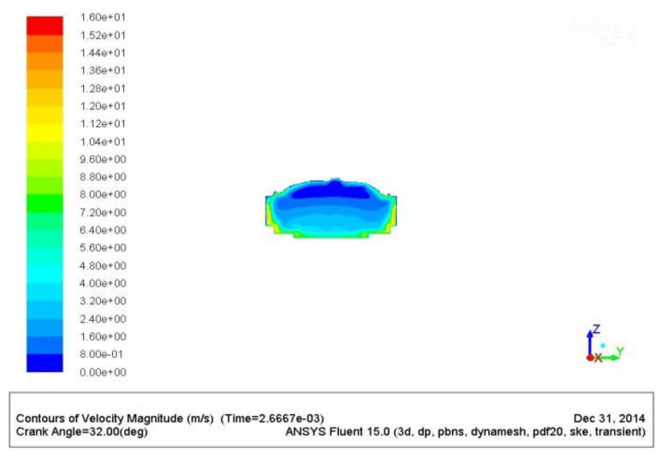

Fig. 7 Velocity Magnitude at 32 CA

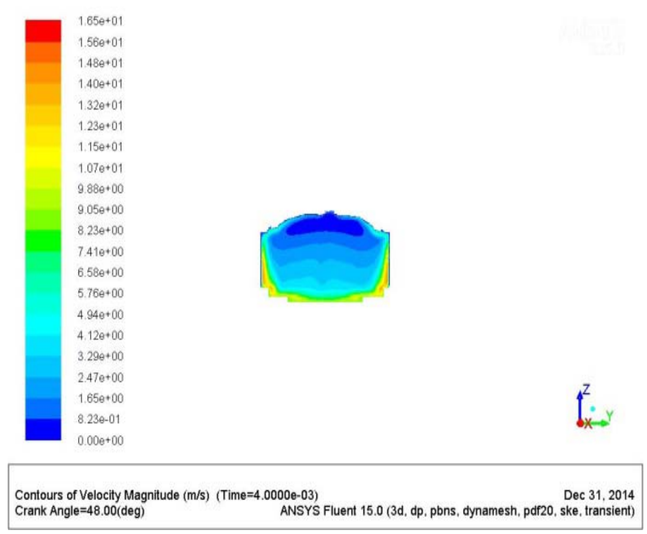

Fig. 8 Velocity Magnitude at 48 CA

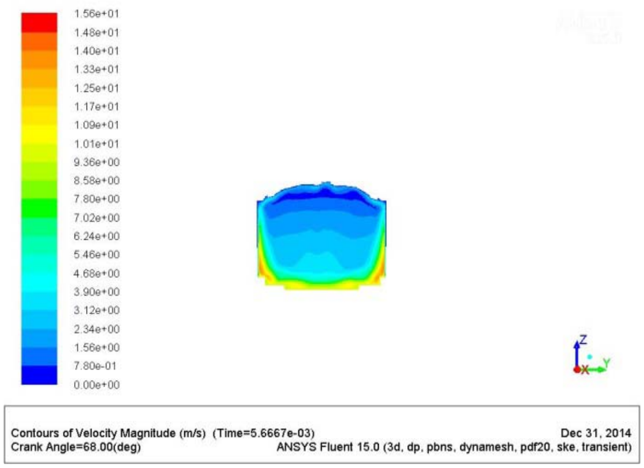

Fig. 9 Velocity Magnitude at 68 CA

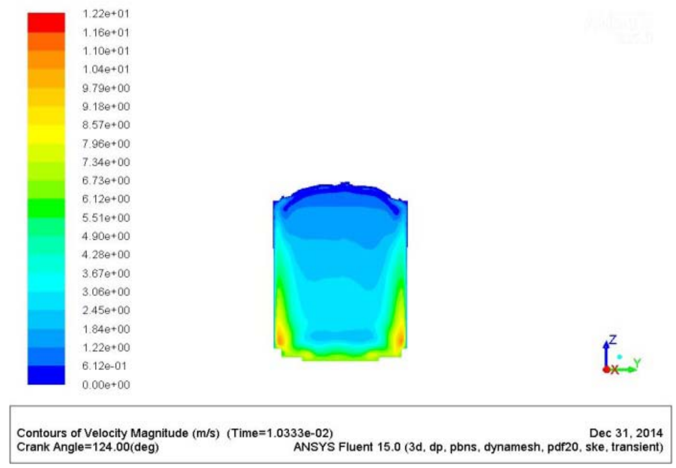

Fig. 12 Velocity Magnitude at 124 CA

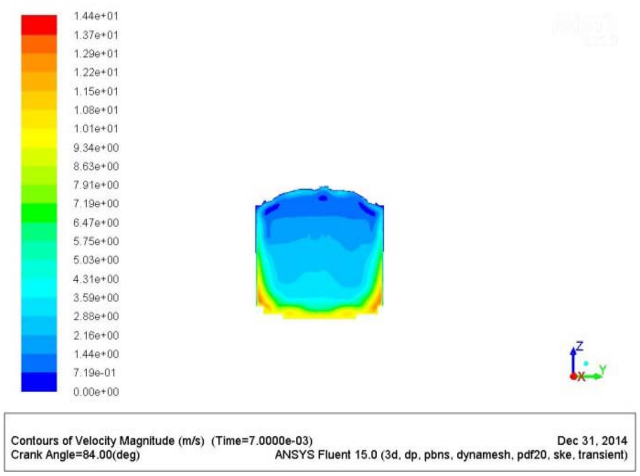

Fig. 10 Velocity Magnitude at 84 CA

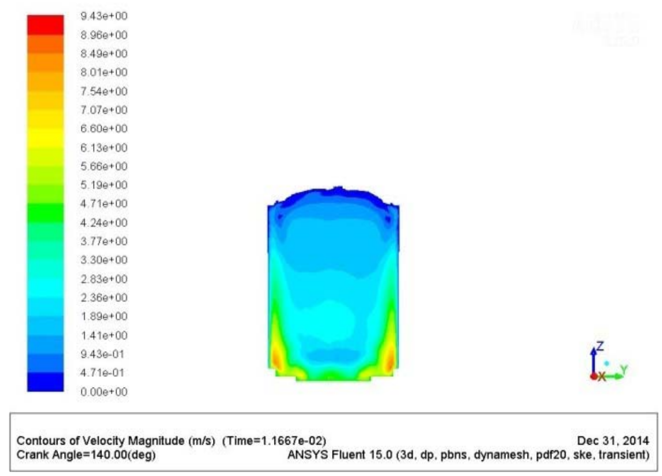

Fig. 13 Velocity Magnitude at 140 CA

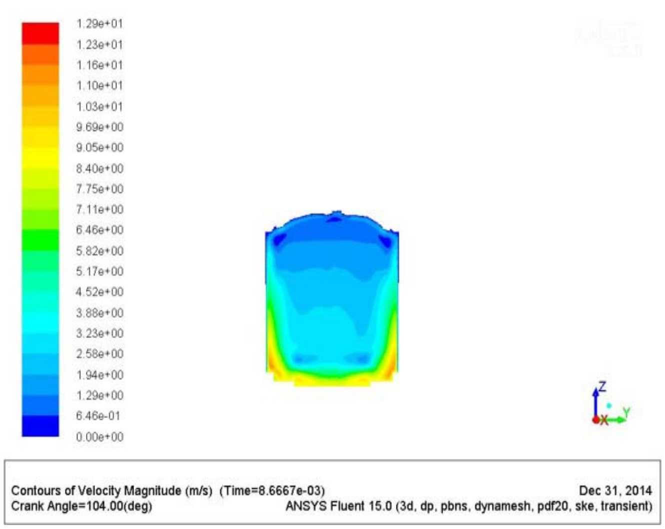

Fig. 11 Velocity Magnitude at 104 CA

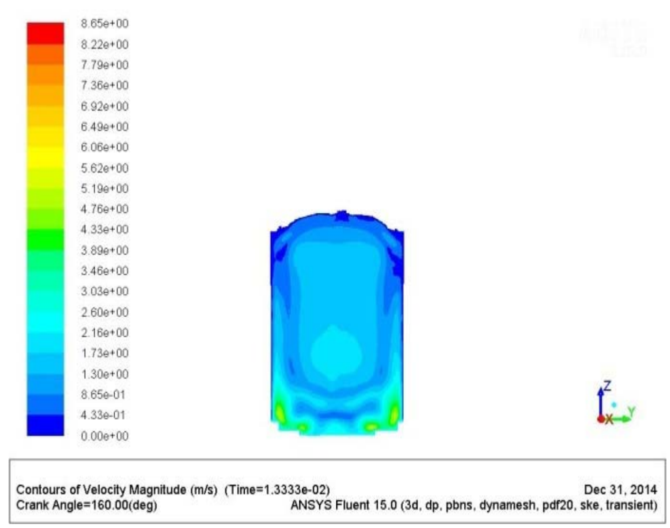

Fig. 14 Velocity Magnitude at 160 CA

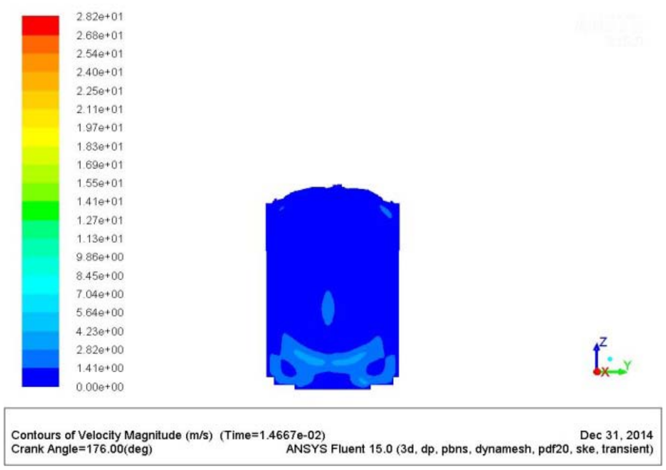

Fig. 15 Velocity Magnitude at 176 CA

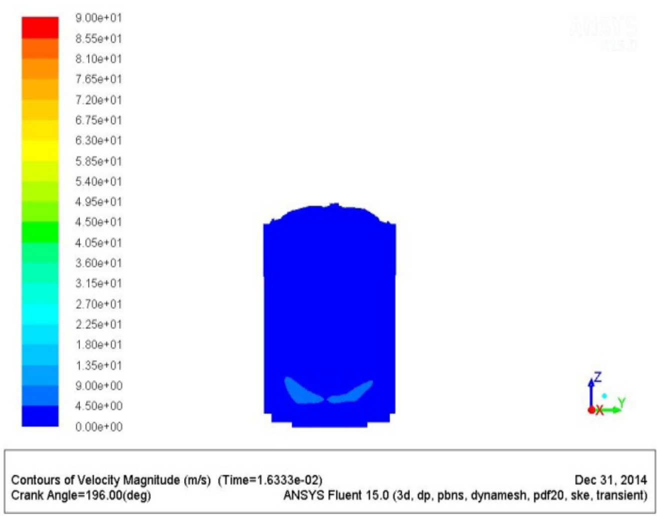

Fig. 16 Velocity Magnitude at 196 CA

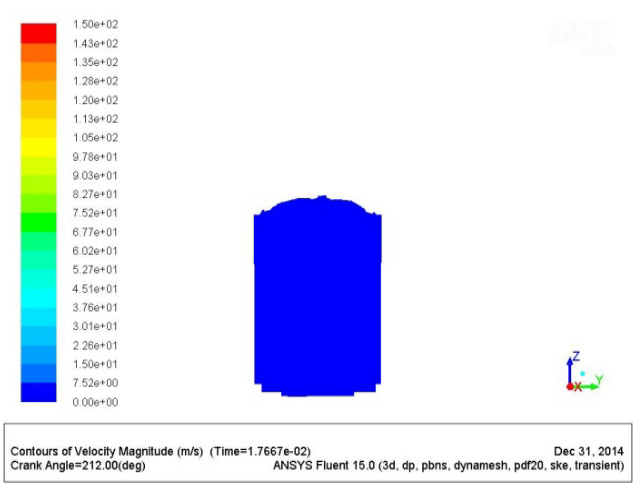

Fig. 17 Velocity Magnitude at 212 CA

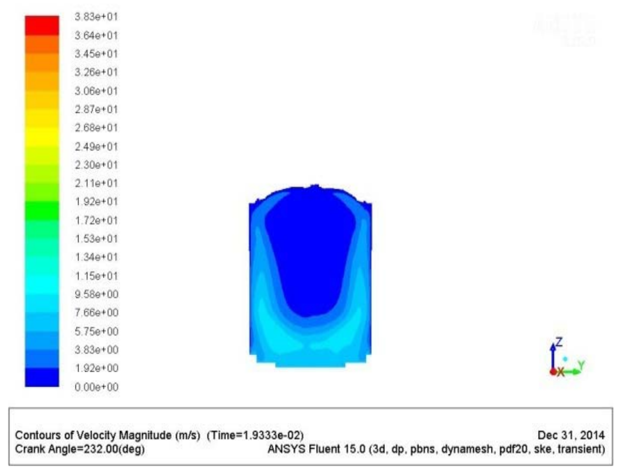

Fig. 18 Velocity Magnitude at 232 CA

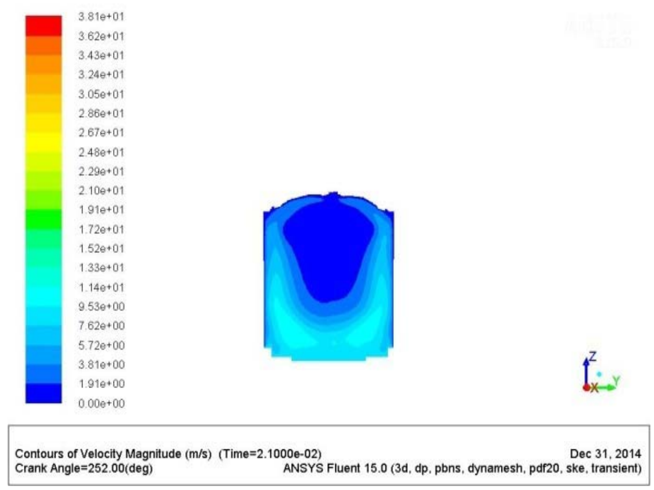

Fig. 19 Velocity Magnitude at 252 CA

5. CONCLUSION

The combustion simulation for a single cylinder spark ignition engine is run successfully and the results show that the combustion model chosen is suitable for analyzing the effect of air fuel ratio on the combustion efficiency of the engine. Further analysis can be done at different equivalence ratios to identify the correct proportion of air fuel ratio at which combustion is optimum.

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