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CFD Simulation of In-Cylinder Flow on Different Piston Bowl Geometries in a DI Diesel Engine

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Abstract

The combustion process in the diesel engine should be controlled to avoid both excessive maximum cylinder pressure and an excessive rate of pressure rise, in terms of crank angle. At the same time, the process should be so rapid that substantially all the fuel is burned early in the expansion stroke. In this direction, piston configuration plays a crucial role. Four configurations i.e., flat, inclined, central bowl, and inclined offset bowl piston have been studied. This study is concerned with the CFD analysis has been carried out on two valve four stroke diesel engine to analyze the in-cylinder air motion during suction stroke, pressure and temperature variation inside the cylinder during the compression stroke for various configurations. The engine specifications are considered from the literature. For numerical analysis, Ansys15 CFD software has been used, for meshing polyhedral trimmed cells were adopted. In-cylinder flows were analyzed by solving mass, momentum and energy equation. From this study, it is concluded that analysis has been carried out for each crank angle degree during suction and compression stroke for all the piston configurations, tumble ratio varies mainly with crank angle position. At the end of the compression stroke fuel is injected and the performance of different piston bowls are analyzed.

Keywords: DI Diesel engine; Combustion; Piston bowl configuration; In-cylinder air motion; NO_x formation.

1. Introduction

Development of fuel efficient and less polluting engine is an important goal of researcher world-wide. Recently however, heightened concern over the environmental impact of internal combustion engines has led to increasing governmental regulations regarding the emission and fuel economy performance. DI Diesel engine, having the evident benefit of a higher thermal efficiency than all other engines. The in-cylinder fluid motion in internal combustion engine is one of the most important factors controlling the combustion process. It governs the fuel air mixing and burning rates in diesel engine. The fluid prior to combustion in internal combustion engine is generated during the induction process and developed during compression stroke.

B.Murali Krishna et.al (2009) deals with experimental investigation of the in-cylinder tumble flow in an engine with a flat piston at the engine speed of 1000 rpm using particle image velocimetry. They have concluded that at 330 crank angle positions, flat piston shows an improvement of about 85 and 23% in tumble ratio and about 24 and 2.5% in average turbulence kinetic energy compared to dome and dome cavity piston. Antony et.al (2012) study is concerned with CFD analysis of in-cylinder air motion and compared with experimental results available. From there study they concluded that a central bowl on flat piston is found to be the best from the point of view of tumble ratio, turbulence kinetic energy, turbulence intensity length scale which play important role in imparting air motion. B.Murali Krishna et.al (2010) studied about in-cylinder fluid flow characteristics of single cylinder engine to see the effect of intake manifold inclination at equivalent rated engine speed engine using particle image velocimetry under various intake valve lift conditions. Based on the PIV investigation, Muralikrishna et.al concluded that 1) in-cylinder flow structure was greatly influenced by intake manifold inclination.2) it is also found that intake valve lifts with 60^{o} manifold inclination, maximum turbulence kinetic energy is highest compared to all other manifold inclination. Benny Paul et.al (2010) studied on the effect of helical, spiral, and helical-spiral combustion manifold configuration on air motion and turbulence inside the cylinder of direct injection diesel engine. The flow characteristics of these engine manifolds are examined under transient conditions using CFD code STAR-CD. After the analysis of different manifolds Benny et.al concluded that; 1) swirl ratio inside the cylinder and turbulence kinetic energy is higher for spiral manifold. B) Volumetric efficiency for spiral- helical combined manifold is 105 higher than that of spiral manifold. Baby X *et al*., have carried out an experimental investigation on in-cylinder motion, during the intake and compression strokes of a multi valve engine. Experiments were conducted on a single cylinder four valve research engine. The engine was attached with several optical accesses on cylinder liner and cylinder head. Effects of different piston bowl shapes on turbulence, flow variations and tumble distortion were analyzed. Kuleshov *et al*., has developed a model that deals with the diesel sprays and combustion. Submodels that are capable to predict the emissions like NO and soot have been implemented in the model. The model was used to handle the parameters like injection strategies, injection duration, droplet size and the bowl shape. Heywood has stated that in medium and small DI engines swirl is used to attain adequate fuel-air mixing rates. Air swirl is generated by adopting suitable changes in the design of inlet port. Friction at the cylinder wall surfaces and the turbulent dissipation of the fluid tries to reduce the angular momentum of the air entering during induction and the same is continued while the compression process is in progress. Swirl velocity can be increases by adopting suitable changes in the combustion chamber design.

Ferguson *et al*., have tried to present the effect of design changes on the moment of inertia. They could show the swirl is proportional to angular momentum and is inversely proportional to the moment of inertia. The bowl-in-piston increases the swirl. Increases swirl is observed to be reducing the moment of inertia as the piston approaches TDC. It is found that, during compression as the piston approaches TDC it is found that, during compression as the piston approaches TDC the swirl increases and starts decreasing after TDC more or less with same pace.

Fuch *et al* in their work it has been showed that, the use of bowl in piston chamber effects the in-cylinder motions. The high swirl ratios may distribute the fuel such that it remains in the bowl. This gives a great scope to deplete almost all of the bowl oxygen during combustion. It has produced stratification of the fuel and air, and poor late diffusion burn.

Beard *et al*., have conducted experiments on various bowl shapes such as flat, W-shaped and with or without re-entrant. It is observed that the shape of the piston bowl is an important parameter to control the turbulence level and fuel air mixing rates. A small change in the bowl shape has influenced the flow parameters like swirl numbers and or turbulence intensity. This in turn modifies the combustion efficiency. It was reported that in the re-entrant region of the re-entrant region of the W-bowl shape both the swirl and the turbulence level increased at around TDC position when compared with flat bowl piston.

Francisco V. Tinaut et.al has presented a model that deals with the atomization and evaporation of transient sprays. The model was associated with the following sub-models namely primary atomization, droplet evaporation and droplet deceleration. The experimental data was extracted from the engine which was equipped with a two component phase-doppler anemometer.

McCracken et.al have proved that optimal swirl level changes as the geometry of the bowl-in-piston changes. Two different bowl-in-piston combustion chambers were taken into account. Both the bowls were tested for the same swirl levels. There was observed quantative differences in the air fuel mixing rate.

Stone et.al have tried to establish the significance of axial swirl in diesel engines, and proved that axial swirl is used in medium and small DI diesel engines for improving the fuel and air mixing rate. Axial swirl contributes for the generation of turbulence with the aid of interaction of the tangential velocity because of the inward flow produced by squish region.

From the literature survey, it is clear that in the in-cylinder fluid flow is very much dependent on the shape of the combustion chamber. However, there is a very limited study on the effect of piston bowl configuration on the in-cylinder flow characteristics. Based on these it is concluded that still there is a gap and it is proposed to fill the gap to the extent possible.

The objective of the present study is to develop a multi-dimensional fluid flow which can predict in-cylinder fluid motion behavior by using different piston bowl geometries. The parameters that are to be analyzed during suction and compression stroke are: pressure and temperature variation inside the cylinder, turbulence kinetic energy, tumble ratio, swirl ratio, cross tumble ratio, tumble intensity in a single cylinder two valve, internal combustion engine with four piston bowl configurations at an engine speed of 1000 rpm at various crank angle during suction and compression stroke.

2. GEOMETRICAL AND COMPUTATIONAL STUDIES

The engine used for CFD analysis has straight inlet and exhaust valve. The numerical predictions are calculated during suction and compression stroke from 360 to 720 CAD for four different piston geometries such as flat piston bowl, central bowl piston, offset bowl, inclined offset bowl for various crank angle degrees. The governing equations are continuity equation, momentum equation, energy equation and K-€ turbulent model equations were solved for in-cylinder fluid flow analysis. Fuel injection was located at the centre of geometry it was injected before the end of compression stroke i.e., 716 CAD with injection nozzle diameter was 0.2 mm and mass flow rate was 0.020 kg/s.

The conservative form of mass, momentum and energy conservation equations, using Einstein's summation convention over repeated indices, are then given by

Continuity equation

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

Momentum equation:

$$\frac{\partial(\rho u_i)}{\partial t} + \frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \left[\frac{\partial \sigma_{ij}}{\partial x_j}\right]$$

According to the Stokes hypothesis which assumes that the bulk viscosity can be neglected, the shear-stress tensor for a Newtonian fluid is given by:

$$\sigma_{ij} = 2\mu(T)S_{ij} - \frac{2}{3}\mu(T)S_{kk}\delta_{ij}$$

Energy equation:

$$c_p\left[\frac{\partial(\rho T)}{\partial t} + \frac{\partial(\rho u_j T)}{\partial x_j}\right] = \frac{\partial p}{\partial t} + u_j\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j}\left(K\frac{\partial T}{\partial x_j}\right) + \Phi$$

Where the viscous dissipation Φ is defined as:

$$\Phi = \sigma_{ij}\frac{\partial u_i}{\partial x_j}$$

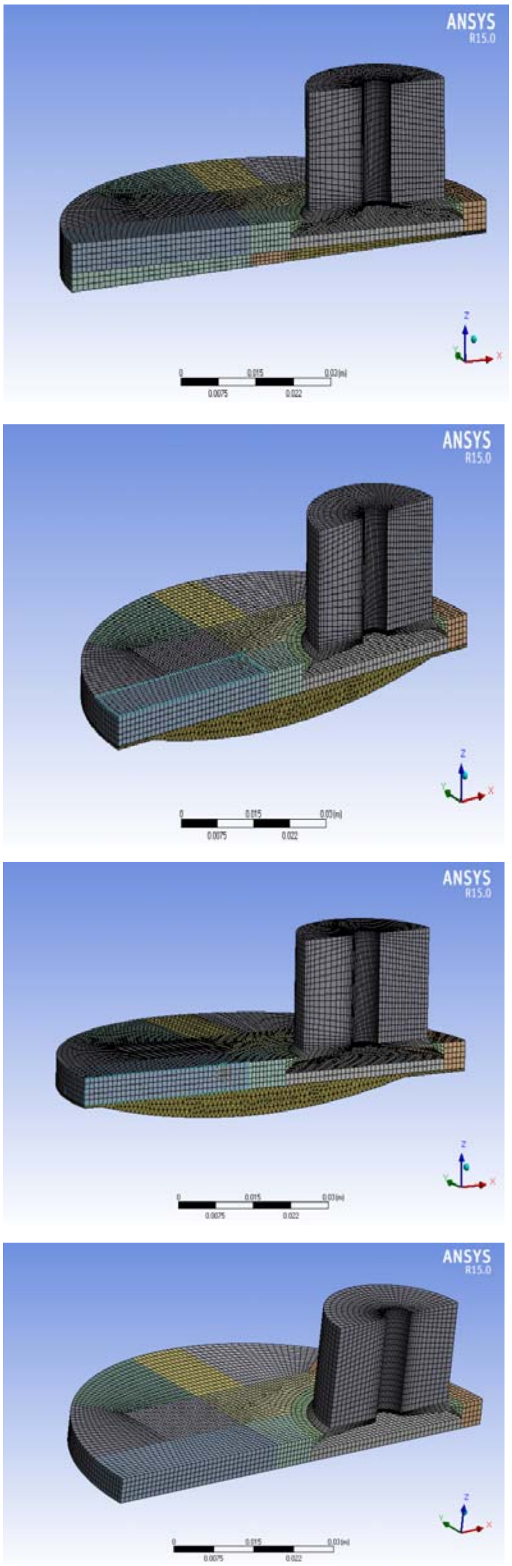


Fig. 1. Computational meshing of DI Diesel engine with different piston bowl shape configurations.

Two valve four stroke engines of four different piston bowl geometries modeling work has been done using Ansys15. Meshing of different piston bowl geometries is shown in figure 1. Numbers of cells used are 5 lakhs.

Grid independent has been done before deciding the size of the cells. Meshed quality was determined by using different parameters like orthogonal quality (ranges from 0 to 1) and. should not be less than 0.01 and Skewness (ranges from 0 to 1) and it should not exceed 0.95.

A constant pressure boundary condition is used for inlet valve. The turbulent intensity is taken as 5%, mixing length scale is 1mm, pressure at the inlet boundary at the start of injection is 0.99 bar. Frictional effects on the wall were not taken into consideration ie., the smooth wall option for turbulent flow boundary condition are used. Boundary conditions of cylinder wall are treated as No slip. Temperature of initial air during suction stroke is considered as 341 K.

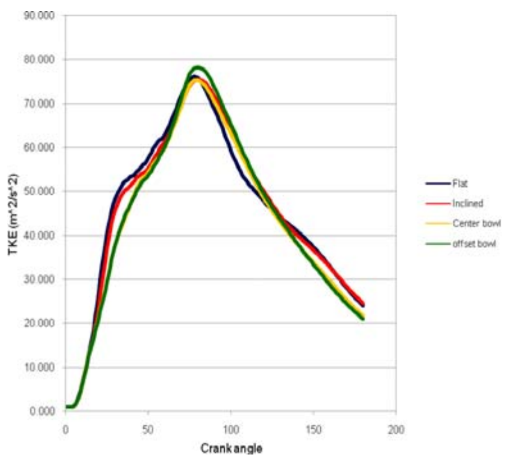


Fig. 2. Turbulence kinetic energy Vs Crank angle.

4. RESULTS AND DISCUSSIONS

4.1 Turbulence kinetic Energy

From fig. 2, Variation of TKE for various CADs during intake and compression strokes for different piston crown shapes at an engine speed of 1000 rpm. From the figures, it can be observed that overall variation of TKE with CAD is similar in all the cases of the piston crown shapes considered in this study. There is a considerable variation in TKE between 90 and 180 CAD. It may be due to the transition dynamics of intake valve which starts closing from full opening position. However, throughout the suction and compression strokes, offset bowl piston shows higher levels of TKE. This may be due to the positioning of piston bowl in the centre which gives better guidance for the jet entering the cylinder.

4.2. Pressure Vs Crank Angle

From fig.3 for flat piston bowl since the turbulence ratio, swirl ratio is higher in flat piston bowl so the air fuel interaction is better. By P-Theta graph, the rate of pressure rise is higher in this case. Hence engine achieved the high peak firing pressure of 100bar which means more power we can get in flat piston bowl than other piston bowls. Cooling efficiency is also good in FPB compared to other piston bowls. Thus we can conclude that combustion efficiency of an engine is higher in flat piston bowl than in others. Usually when the temperature in combustion chamber is high (i.e. above 1400K) near TDC, then the rate of formation of NOx is higher. For inclined piston, Center & Offset Piston Bowl: Since the turbulence ratio is slightly less than flat piston bowl, we can understand the air fuel interaction is less. This results in low peak firing pressure in engine and temperature distribution is HIGHER as per Ideal gas equation. The combustion efficiency of an engine is detoriated with less turbulence ratio and swirl ratio.

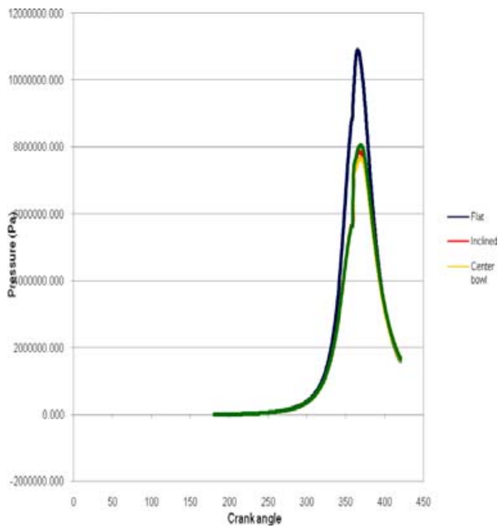


Fig. 3. Pressure Vs Crank angle.

4.3 Swirl ratio Vs Crank Angle

From fig.4, here the swirl ratio is higher for flat piston bowl at 24 Deg.CA and 90 Deg.CA, the initial mixing is having a very high swirl ratio which is considered to be good for engine this may be due to piston crown shape which is giving better air movement in the cylinder. However the magnitude of the swirl ratio is changing w.r.t crank angle degree and for center bowl, offset bowl the effect is neglible before compression stroke, whereas the effect Is high incase of flat piston bowl due to the shape of bowl.

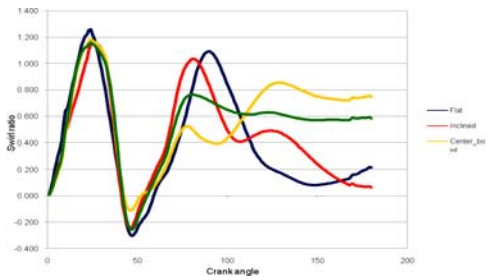


Fig. 4. Swirl ratio Vs Crank angle.

4.4 Tumble Ratio and Cross Tumble Ratio Vs Crank Angle

From Fig. 5 and 6, here the negative or positive magnitude of tumble ratio indicates the direction of the overall in-cylinder tumble flows at a given plane as clockwise or counter clockwise respectively. The reasons for this could be (i) change in the overall tumble flow pattern due to low pressure and bifurcation zones,(ii) change in piston speed with CADs, and (iii) change in the direction of the piston movement during suction and compression strokes. At 55 deg. CA we have achieved the maximum tumble ratio. Here the magnitude of tumble ratio is changing w.r.t. change in geometry of piston bowl. At 80 deg. CA Turbulent Kinetic Energy is maximum for offset piston bowl, this may be due to the positioning of piston bowl in centre which gives better guidance for the jet entering the cylinder and after that slowly the turbulent kinetic energy decreases as the piston is moving towards TDC.

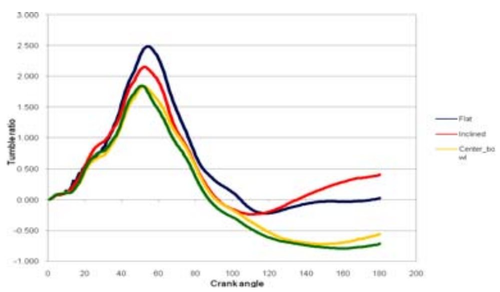


Fig. 5. Tumble ratio Vs Crank angle.

4.5 Velocity Vector Contours

Distribution of pressure and temperature at various crank angle degrees are shown in figure compression stroke for flat piston. The increment of time step is taken 0.5crank angle degree. Piston starts from bottom dead center about 541degrees and the maximum pressure reaches at 716 degree. The pressure increases from 542 crank angle degree to 716 crank angle degree and reaches its maximum value at 716 crank angle degrees .The Maximum pressure at the end of compression is 85bar and temperature is 1030 K.

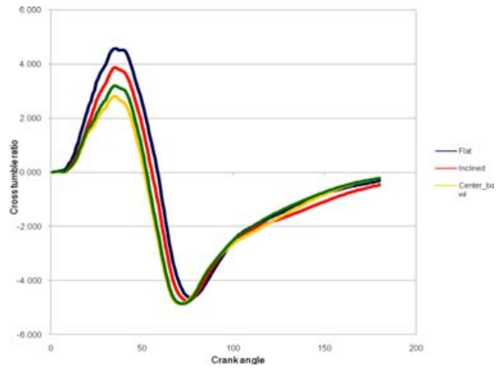


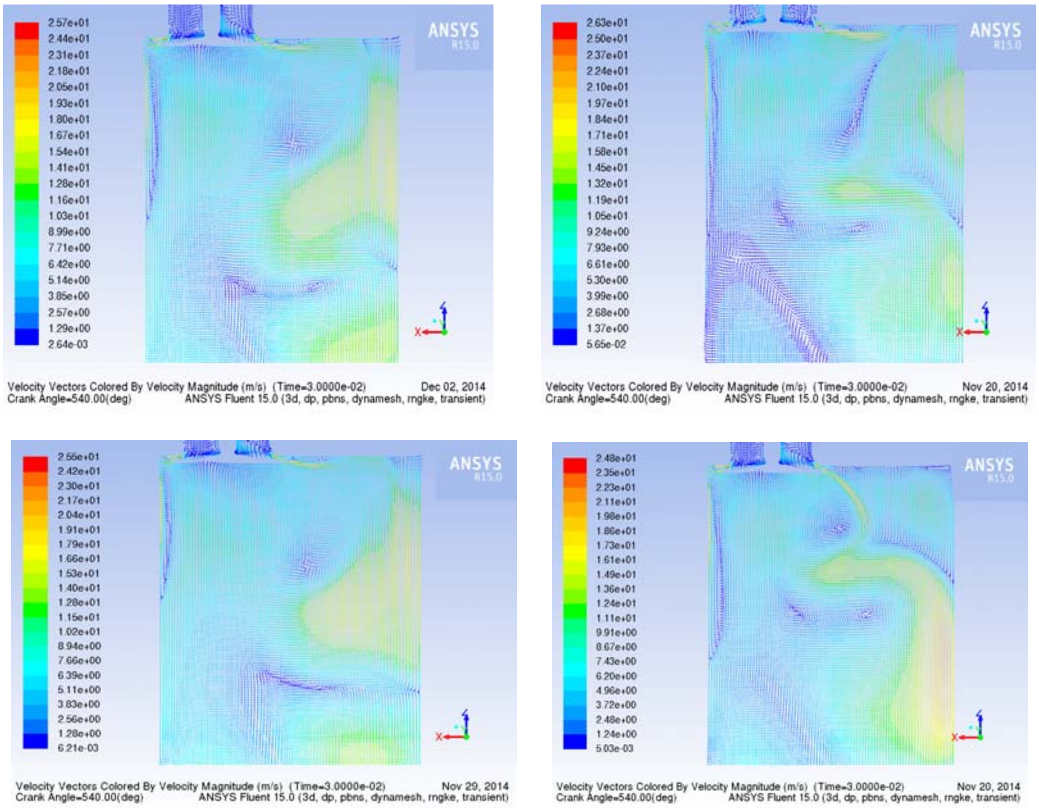
Fig. 6. Cross Tumble ratio Vs Crank angle.

Oxides of nitrogen are formed within the combustion chamber due to the dissociation of the molecular oxygen and nitrogen at the peak combustion temperatures and persist during expansion and exhaust in non-equilibrium amounts. The maximum NO levels are formed with air-fuel ratio of about 10% above the stoichiometric. The chemical nature of fuel has no effect on the yield of NO. At the end of the crank angle degree(during expansion stroke) rate of NO_x for flat(0.0003083 Kg-mol/m^3-s), inclined (0.0002596), central bowl(0.0002263), offset bowl(0.0003010).

5. CONCLUSION

From this investigation, it is observed that CFD computational study is important to understand in-cylinder flow structure during suction and compression stroke on a single cylinder DI Diesel engine with different piston bowl geometries are summarized as follows:

1) Piston bowl position plays a predominant role in the air pattern inside the cylinder.
2) The turbulence intensity and length scale are higher for flat bowl piston.
3) Considerable variation in turbulence kinetic energy due to the transition dynamics of intake valve which starts closing from full opening position, however throughout the suction and compression stroke, offset bowl piston shows high level of TKE.
4) For flat piston bowl, since turbulence ratio, swirl ratio is higher so the air fuel interaction is better. By p-theta graph, the rate of pressure rise is higher. Hence engine achieved high peak firing pressure of 100 bar, which means more power can be achieved in flat piston bowl than other configurations.



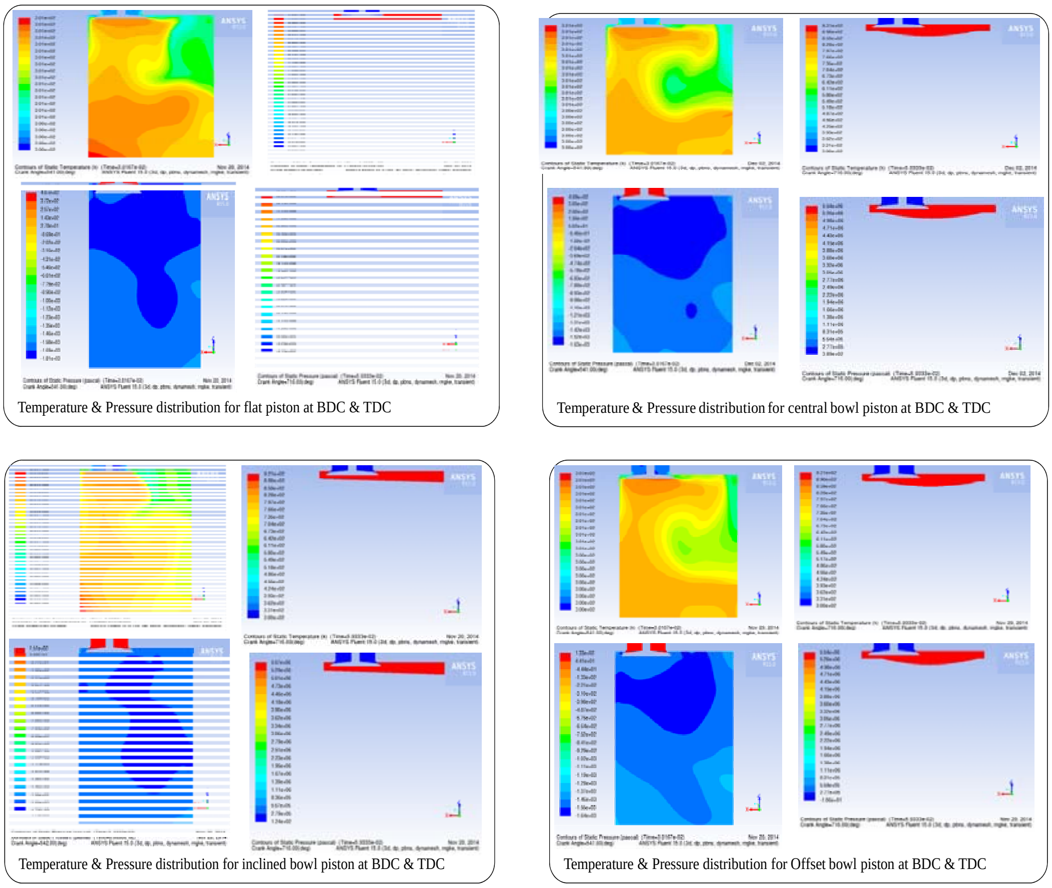


Fig. 7. Temperature and pressure distribution of various piston configurations.

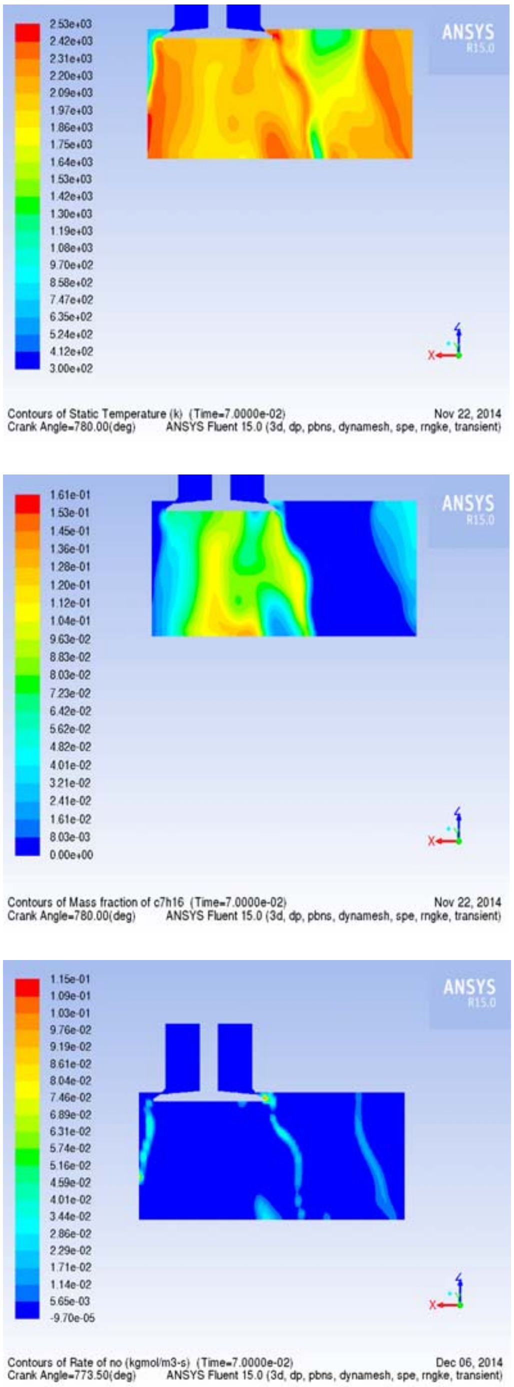


Fig. 8. Contours of temperature, mass fraction, rate of NO_x for flat bowl.

- In sum after thorough analysis of results from different piston bowls, flat piston bowl is giving the best results in terms of power, efficiency, peak cylinder pressure and temperature.
- SINCE the swirl ratio and turbulence ratio is higher for flat piston bowl, which is considered to be good for engine this is due to piston crown shape which is giving better air fuel interaction inside the cylinder.

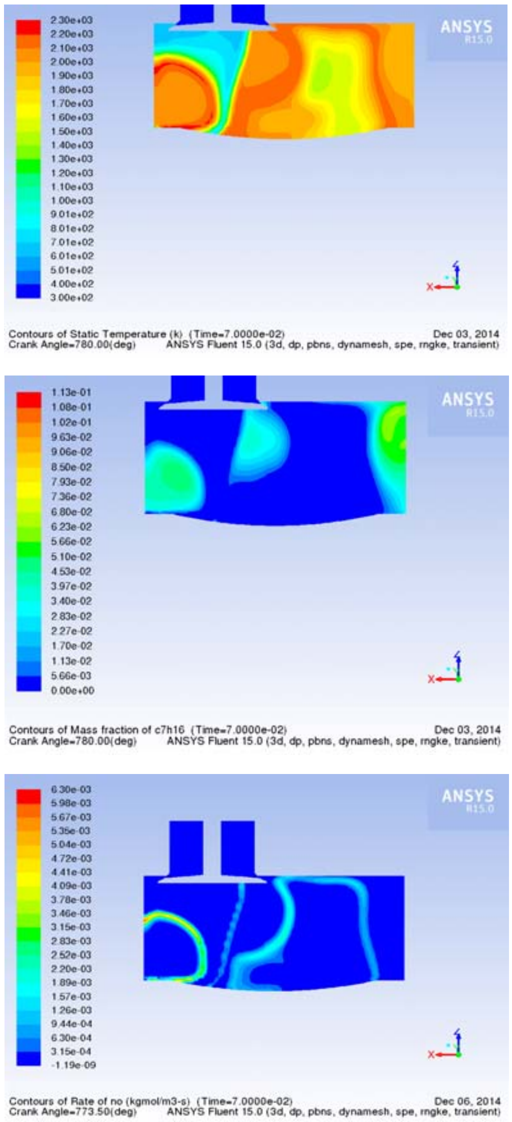


Fig. 9. Contours of temperature, mass fraction, rate of NO_x for central bowl.

- In order to cut down the cost of additional hardware like Exhaust Gas Recirculation (EGR) to control the Nox, we can control by optimizing combustion chamber design and injections timing strategies.
- For automotive market like Indonesia is using high sulphur content diesel fuel currently, the norms are EURO2 which does not require EGR to control Nox, in such cases we can implement my strategy to optimize the engine by piston bowl designs.
- Hence we can say using flat piston bowl as per this engine specification is giving better results compared to other piston bowl designs.
- Thus we can conclude that combustion efficiency of an engine is higher in flat piston bowl than other configurations.

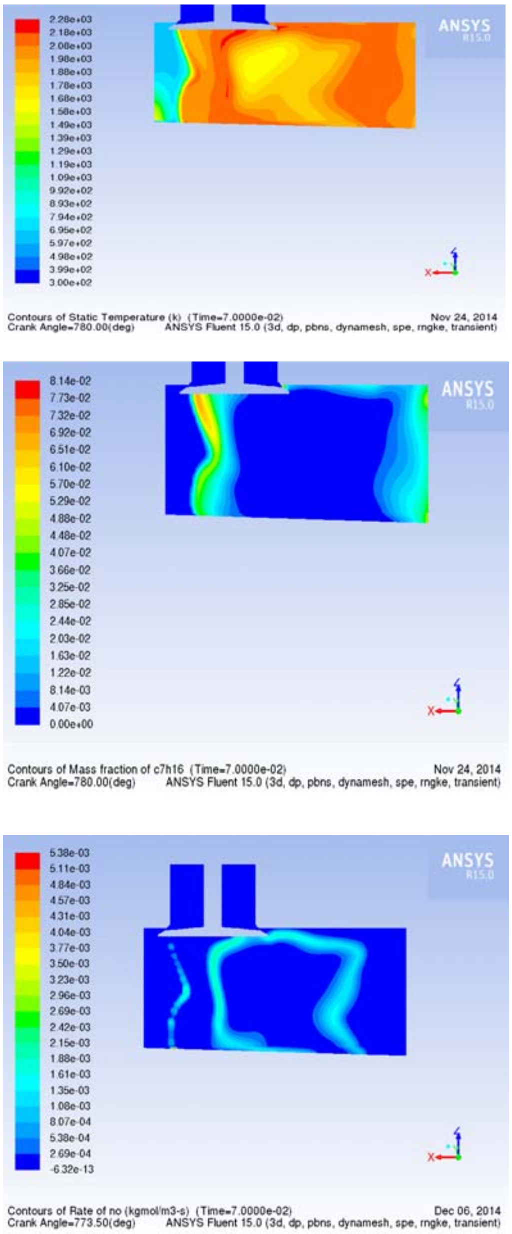


Fig. 10. Contours of temperature, mass fraction, rate of NO_x for Inclined bowl.

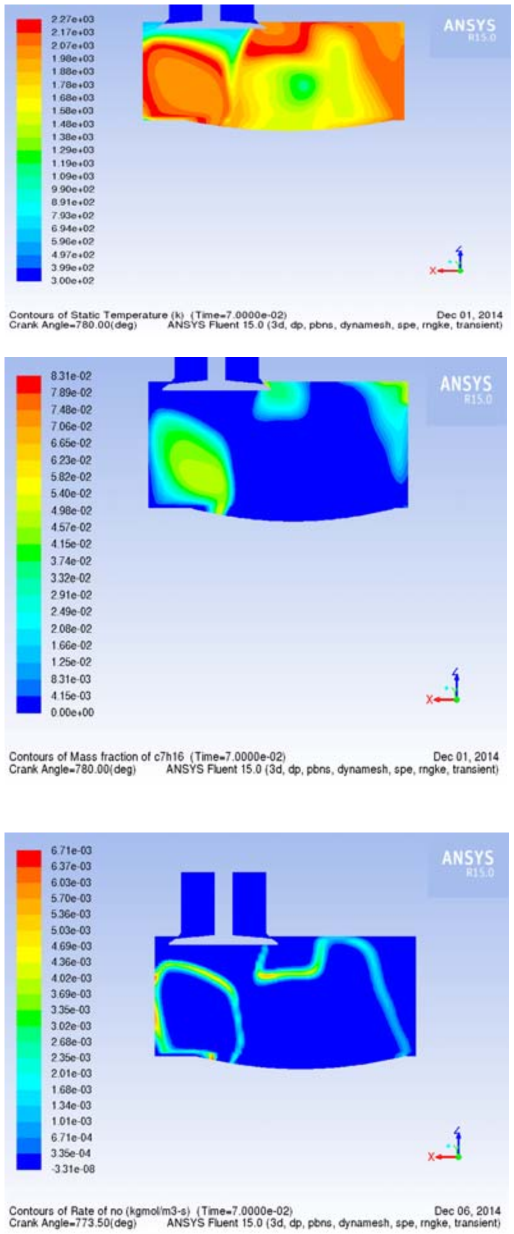


Fig. 11. Contours of temperature, mass fraction, rate of NO_x for offset bowl.

Future Scope

- Using multiple injection strategy i.e. pilot injection, main injection and post injection, I will study the effects on performance characteristic.
- By varying the main injection timing before TDC and after TDC, how much it effects emissions will be carried out.
- To do performance test with prototype of designed piston bowls which were used for simulation.
- To compare the experimental results with simulated results of all the above piston bowls.

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