EFFECTS OF PISTON BOWL GEOMETRY ON COMBUSTION AND EMISSION CHARACTERISTICS ON DIESEL ENGINE: A CFD CASE STUDY

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Abstract

The present paper investigates effect of piston bowl geometry on combustion and emissions. One of the important parameters which affect the combustion in diesel engine is piston geometry. In this study four different piston geometries (HEMISPHERICAL, OMEGA, SINGLE CURVED, DOUBLE CURVED) have been selected for comparative study. Numerical investigation has been carried out using commercially available code AVL FIRE which is a finite volume based code .Automatic mesh generation feature of code is utilized for modelling. For validation of numerical model, simulation results of baseline shape have been validated with experimentally available results. Simulation has been carried out for various piston geometries while keeping parameters such as speed, compression ratio and mass of fuel injected constant. The study has been carried out at full as well as half load .A comparative study for in-cylinder pressure, rate of heat rejection, velocity and temperature field, turbulence kinetic energy has been performed .Emission and performance parameters like SOOT, NO ,efficiency and BSFC have been studied . The results indicated that omega shaped piston works best for the current set-up. As compared to baseline shape of hemisphere, it gives 12% higher NO but 19% less SOOT. Also, the efficiency and BSFC are better for omega shaped piston.

Keywords- *OMG, SC, DC, HEMI, piston bowl, combustion, emission, CFD, AVL, simulation*

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1. INTRODUCTION

Global warming is the biggest threat faced by our mother earth in the present era. UN climate change conference 2015 held at Paris was the most discussed and major step toward this threat. A majority of the pollution across globe is caused by exhaust emissions from automobiles. Hence a lot of research is going on for the development of various strategies, methodologies that would bring down the emissions. Carbon monoxide, oxides of nitrogen and soot are the main culprits responsible for deterioration of air quality. Diesel engine due to its excellent portability and performance remains a primary choice for powering vehicles other stationary devices.

Various strategies have been developed over the time for emission reductions. Alternate fuels, that is biofuels, and blending is one of prominent field which plays significant role in emission reductions (1-3,34). Other studies suggest that fuel injection timing along with pulsed injection have significant effects on the engine emissions (4-6). Studies relating combustion in diesel engine states that swirling is a major factor affecting and promoting complete combustion which in turns reduces exhaust emissions(8). Intake pressure too affects the combustion (4).In cylinder flow is primarily a function of bowl geometry, turbulence kinetic energy and pattern of mixing of air-fuel mixture(7). Exhaust gas recirculation (EGR) is an after treatment methodology concerning combustion and reduction of emissions (9).G. Sivakumarhave used a chemically coated piston crown to study the effects on performance and emission (33). Various permutations and combinations of above factors and its effects are being studied by scientists and engineers.

Piston bowl geometry is an important factor in the mixing of air-fuel mixture in engine. Various combinations of it with other parameters have been studied. Jaichandar S et al. has studied effect of injection timing and combustion chamber geometry fuelled with biodiesel on diesel engine(10). Saito T et al have investigated effect of combustion chamber geometry on diesel engine(11). An insight to in-cylinder flow was provided by Payri F(12)et al by CFD modelling. Computational study on HCCI engine has been carried out with two different piston bowls by T. Karthikeya Sharma (32). Rakopoulos CD et al have studied piston bowl geometry and speed effects on a motored diesel engine configuration(13). But very few studies have compared different configuration of piston geometry(14,28,29).

In this study we have selected three additional piston shapes and comparison among all has been done based on various parameters. The comparative study will give us an insight into the effects of piston shapes on parameter and help in finding out such factors which will be critical to design of piston bowl geometry. MuruganSivalingamhave done the theoretical modelling and its validation by comparing combustion parameters (31).Similarly, computer modelling is another way out for our problem. In this study, comparison and analysis will be done using computer simulation. There are many commercially available codes in the market of computer simulation. We have chosen a control volume based CFD code AVL FIRE for our simulation and analysis. The detailed methodology of current study is described further.

1.1 Methodology of study

Below is the description of methodology adopted for this study. The results for half and full load diesel engine were already available. The engine specification has been mentioned in table 1. Now with the help of AVL FIRE CFD code we have prepared the numerical model using suitable initial and boundary conditions. This numerical model has been validated with experimental pressure and heat release profile which is a general trend in validation study (15,30). Since it has already been observed that piston bowl plays an important role in the combustion process and hence affects performance and emissions in diesel engine, we will be changing piston shape and doing the simulation with new configuration, successively. All the other parameters like compression ratio, injection angle, injection pressure etc. will remain constant. We have selected three more piston geometries; named as omega (OMG) shaped, single curved (SC) and double curved (DC). The baseline piston geometry is hemispherical (HEMI). Fig 1 shows pictorial representation of various shapes. Practical feasibility and a little bit of imagination is the motivation behind selecting these particular shapes. OMG and SC piston are already in use and widely studied. The curvature of SC piston is altered and one more piston shape is added to the study resulting in DC piston shape. This is a product of *curiosity and imagination*. Few studies have considered this type of shape (16). Since the wall of piston plays a significant role in the mixing and imparting turbulence, we are more interested in finding out how it affects the whole process which undoubtedly results in these types of configurations. Simulations are performed and then a comparative study between all those shapes is done. Parameters such as incylinder-pressure, rate of heat release, TKE, temp, NO, SOOT will be compared and analysed.

2. CFD MODELLING

CFD model has been prepared by means of AVL FIRE code. This is finite volume based solver. An inbuilt feature of mesh generation is employed for meshing purpose. All the shapes are symmetrical about the vertical axis. Also there are four nozzles for fuel entry to chamber. So it is beneficial in terms of computational effort to split the control volume in four parts for analysis purpose. So the analysis will be done on 90 deg cut out of total 360 control volume. After simulation will be done AVL FIRE has feature of integrating the results of single cut to the whole control volume.

Now we will be proceeding with simulation of our baseline shape of hemispherical piston. The first step in this direction will be of meshing. Number of boundary layers to be used is 2 with average cell size of 0.75 mm. This gives a total number of 32,184 cells in mesh. But we also needed to be confident about the mesh independency of results. For the same purpose we have conducted mesh independency test for few of parameters. This gives us confidence over the size and number of cells. The following table 2 shows the results of different cell size, cell numbers with their effect on variation on maximum pressure and max temperature. Parameters have been taken from full load simulation.

As we can see in above table, the variation in pressure and ROHR is below 1 % and near to 1% respectively. On further refinement of mesh there is no significant change in the values of parameters. So we can surely say that grid size of 0.75 mm and 32,184 cells will give reliable results. Below figure-2 shows mesh at TDC of all configurations to be studied.

2.1 Models Used in Simulations

Commercially available AVL FIRE code has been used for simulation. This code has been developed on finite volume approach and pressure based segregated solution algorithm. The SIMPLE (semi-implicit) algorithm is used for pressure-velocity coupled equations in the solution procedure. Calculation of boundary value is done by extrapolating method. The derivatives have been calculated by least square methodology. The turbulence and turbulent wall heat transfer is modelled by K-zeta-f model given by Hanjelic(17). Under relaxation factors used for different parameters are around 0.9 for all. Differencing scheme used for continuity equation is central differencing while for momentum MINMOD scheme has been implemented. The remaining equation of energy, turbulence and scalar quantities has been differenced by UPWIND scheme (18). Combustion phenomenon has been modelled by ECFM -3Z model(19). Well proved Extended Zeldovich and Kinetic model have been used for NOx and SOOT respectively in the combustion process (20,21). Speed and pressure of fuel affects the size of droplets. Dokowicz model which is suitable for greater Weber number applications has been employed here for evaporation. The wall spray interaction is predicted by walljet 1 model. The spray phenomenon is modelled by WAVE break-up model (20).Boundary wall layer temperature profile predictions has been carried out by Kader`s model (22). For a clearer picture, below table summarizes phenomenon and model used in AVL FIRE.

Studies performed earlier by a number of researchers have validated codes and models listed in AVL. Tatschl`s study validate diesel combustion model for multidimensional engine simulation(23). Wieser carried out study on 3-D combustion and heat transfer models for simulations in diesel engine(24).Primary hybrid model for break-up studied by Suzzi etc. has proved to be consistent for non-evaporating and evaporating sprays(25). Again Tatschl carried out study for validates FIRE code for combustion and diesel mixture formation (26). This study gives us enough confidence to rely on the simulation results given by AVL FIRE.

3. VALIDATION OF NUMERICALMODEL

For comparative study between different configurations firstly we will need to validate our CFD model with experimental results. Experimental results were available for our baseline shape of hemispherical piston as mentioned above. As a general trend observed in other studies(16,28), in-cylinder pressure and rate of heat release profile should match with that of simulation results for validation purposes. A part of compression and expansion near the vicinity of TDC has been compared here (16) since the combustion phenomenon is predominant near TDC. In this study we will be validating pressure and heat release rate for both full load and half load conditions. Fig 3 and fig 4 shows comparison of pressure and heat release rate for full as well as half load conditions plotted against CA near vicinity of TDC.TDC is at 0 deg CA .The overlaid profiles matches closely giving us enough confidence about the numerical model along with initial and boundary conditions.

4. RESULTS AND DISCUSSIONS

Simulations were carried out for the selected piston shapes at full load. In all the simulations, parameters other than piston shape were kept constant. Hence compression ratio, initial conditions, computational models, solver methodology etc. were same at all different configurations. The cavity volume and meshing is dependent on the piston shape hence a little difference were observed. In-cylinder properties such as pressure, rate of heat release, velocity field and turbulence kinetic energy has been studied, compared and analysed for all configurations at full load. On the emission and combustion side, temperature, NO and SOOT has been taken into account.

4.1 In-Cylinder Pressure

Fig 5(a)and fig 5(b)shows the pressure profile comparison of all the configurations over a range of 60degCA BTDC to 60deg CA ATDC at full and half loading conditions. For the conventional piston shapes of hemispherical, omega and single curved the pressure is showing consistency throughout the range of CA considered at both loading condition. The peak pressure variation is under 2%. However the double curved shaped piston at full load is showing the slight variation in the profile till the TDC giving higher pressure. The partition wall between two curvatures may be attributed for this behaviour, giving a little bit more compression effect to the air. But at half load condition the peak is delayed by 1 deg. This should be due to increased delay period. The peak pressure for double curved configuration at full load is less by nearly 4% than that of hemispherical one. All the profiles are showing consistency after 5 deg CA ATDC at full load. The peaks for all different configurations at half load are less than that at full load which is general trend for pressure. The little effect on pressure is reasoned by the small size of engine, which is not giving much variation in pressure. The minimum peak of double curved shape is reasoned by slightly more cavity volume for same compression ratio.

4.2 Rate of Heat Release

Comparing rate of heat release gives us a good idea of what is happening to the combustion in cylinder. Fig 6(a)&fig 6(b) represents the heat release rate for all configuration plotted against CA of 10 deg BTDC to 20 deg ATDC for full and half loading conditions. For full load conditions, the profile for omega, hemispherical and single curved is showing similar trend. Ignition delay period and premixed combustion is similar. The peaks of all HRR are under 2-3% variation. However in the double curved configuration ignition delay period has been increased by 3 deg. This may be attributed to less temperature of fuel which is confirmed by temperature comparison. After dropping for a 2 deg, it is showing consistency over the next 3 deg giving a wider spread of profile. This is reasoned by combustion of any remaining mixture which happens to be escaped due to geometry. At half load a similar trend of profile is observed. For double curved, ignition delay period is 2 deg more than that of others. There is a sharp fall in profile of omega shaped piston. This can be reasoned with ignition happening near the cylinder wall giving a sudden fall in value. At half loading condition, we can see a greater difference for hemispherical and double curved profile, which varies from that at full loading condition. But as we know engine gives better performance at full load and hence full load results are more trustworthy.

4.3 Velocity Field

The fig-7 shows the velocity distribution at various CA ATDC. It shows the effect of piston shape. For omega and single curved shaped piston we can see a vortex forming. And in double curved shape, two vortexes are in a position to form. But the intensity is showing much variation in strength of vortexes. This is the main intention behind selecting this type of shape for study. However in this study we are not considering the variation of injection angle.

So it can be depicted that, for our set-up conditions inner half of the shape is not fully used. In the hemispherical shape, even though there can be seen formation of vortex, it is not able to impart its energy due to absence of wall near to it. The maximum velocity can be depicted for omega shaped piston.

4.4 Turbulence Kinetic Energy (TKE)

Fig8(a)& fig 8(b) shows comparison of mean TKE over CA at full and half load respectively. At full load, as we can see, omega shape is having maximum value of the TKE out of all configurations considered, for the current set-up. This justifies above results of in-cylinder pressure and heat release rate. Initially, TKE of double curved was expected to give more turbulence and hence proper fuel-air mixture mixing leading to effective combustion. But for the current set-up it is showing lesser value. The reason is attributed to the angle of fuel injection as can be seen in velocity field. Comparing between omega and single curved, it can be observed from velocity field that intensity of velocity for single curved is somehow more than omega. But the overall spread is more for omega. This reason is attributed to higher

TKE of omega than single curved shape. At half load, we observe a similar profile of TKE. However for omega shaped piston we are getting a slightly higher TKE. This should be due to the less amount of fuel giving a lighter air-fuel mixture than at full load and hence more TKE. But for single and double curved profile we are getting reduced TKE at half load than at full load. Since TKE is more sensitive to the RPM of engine we are not able to see a correctly reasoned trend among the profiles.

4.5 Temperature

Fig-9 shows variation of mean temperature over CA. It is showing highest temperature for omega shaped out of all. This confirms previous results of pressure, heat release rate and TKE. Double curved is having minimum temperature compared to others. There is a slight suppression of 1 deg temperature near TDC for double curved shape. It may be due to delayed ignition of fuel air mixture. Also the spread of temperature profile is greater just like heat release rate and then there is a sudden drop at 40 deg ATDC. The shoot up of profile for hemispherical shape may be justified by sudden burning of unburned air-fuel mixture. Fig-10 shows pictorial distribution of temperature field. It gives a realistic view of combustion process which is happening inside the combustion chamber. This gives another confirmation to all the results discussed earlier. The inside curvature of double curved shape is not utilised to full extent. Direction of fuel injection is the reason behind this particular behaviour.

4.6 Emissions: SOOT and NO

Fig 11(b)and fig 11(d)shows the NO content plotted against CA at full and half loading condition. As we know high temperature is responsible for NO formation, we are getting highest NO content in omega for full load. Formation of NO is affected by highest temperature, reaction time and oxygen content. The sharp increase in the NO is attributed to sudden rise in combustion throughout the cavity. Comparing percentage wise with maximum NO for omega shape, at full load, we are getting 12.34 %, 15.78% and 34.87% less NO for hemispherical, single curved and double curved shaped piston. Double curved piston shape is showing excellent NO characteristics for the current set-up. Compared to our baseline shape of hemispherical piston it is showing nearly 27% less NO. On the other hand, we have with us NO content at half load. This shows lower values of NO for all configurations. Comparing percentagewise with max NO of omega at half load, we see a drop of 9 % , 11 % and 40% for hemispherical, single curved and double curved respectively. Double curved shape is showing 33% lower NO as compared with hemispherical shape. This drop shows a similar trend as seen at full load.

Fig 11(a)and fig 11(c)shows SOOT plotted against CA at full and half loading conditions respectively. SOOT is affected by combined effect of mixing of fuel, temperature of air-fuel mixture, oxidation level combustion. For full load, minimum is SOOT for omega shape which is justified by complete combustion of air-fuel mixture due to higher TKE and temperature. A variation in the trend has been observed for hemispherical bowl. The increased SOOT content can be attributed to less turbulence and bigger cavity volume which is giving residual space for particles. Less oxidation in the mixed and late combustion phase can be reasoned for this increased SOOT. Comparing percentage wise with hemispherical at full load which is also our base shape, we are getting 8%, 15.5% and 19.8% lesser SOOT for double curved, single curved and omega shaped piston geometry. At the half load conditions we see reduced values of SOOT. Percentagewise comparison shows that we are getting 15%, 13% and 10% less SOOT for omega, single curved and double curved shape respectively. Here we also observed a wider shape of profile for single curved as compared to that at full load. This should be due to fall in combustion reaction rate giving sudden rise in un-burnt air-fuel mixture.

4.7 Efficiency and BSFC

Fig 12 shows comparison of efficiency and break specific fuel consumption at half and full load for all configurations. As we all know that full load efficiency of diesel engine is better than that at part load, here it has again been confirmed. Omega shaped piston is showing maximum efficiency of 40 % which is consistent with many of previous results like pressure, heat release, temperature. The variation in efficiency is though little at part and full load. But the variation gets reflected on BSFC. BSFC is a performance parameter which tells the effectiveness of diesel engine to utilise chemical energy of fuel into combustion. Lower the BSFC more efficient the engine is. Since for every unit of power generated we are using less fuel. Confirming to previous trends omega shape is giving least BSFC. It is a well-known fact that engine performs better at full load and hence we are getting less BSFC at full load than at half load. Fuel is not utilised properly at part load conditions. On comparing with hemispherical shaped piston, we got 9 % less, 1 % more and 6% more BSFC for omega, single and double curved shape respectively. The maximum fuel consumption is for double curved shape. It can be reasoned by the bigger cavity volume which gives larger surface for settlement of fuel which remains un-burnt. However, on combining with optimum fuel injection angle it may produce better results, which can further be explored.

5. CONCLUSION

This study aims to explore the effects of piston bowl geometry on combustion, emission and performance. This is a comparative study of four different piston shapes on parameters as described above. Following are the concluding remarks on the results.

- For the current set-up, OMG shape gives the maximum pressure while DC shape gives the minimum pressure. However the difference of pressure is not more than6%. On the other hand with HRR we found that OMG gives maximum HRR while SC gives minimum. The ignition delay of 3 deg has been observed with DC shape. So it has been concluded that with given set-up conditions OMG is optimum profile for pressure and HRR.

- Phenomenon of combustion is strongly dependent on TKE. On comparing TKE we found OMG is best suited for current configuration. DC shaped showed least TKE which was our imaginative shape. It was predicted that the double curvature would increase TKE resulting in the better combustion but results were contradictory. However variation of spray angle could result in better results for the same shape which can further be explored. HEMI and SC shape have shown intermediate results between the previous two.
- To have a better understanding of inside flow, in this study we have studies velocity vectors and temperature distribution. The results showed that because of the partition wall in DC shape, at current angle of spray, optimum mixing is not possible. So it can be concluded that mixing is dependent not alone on curvature but also the height of separating wall and spray angle.
- On emissions side NO and SOOT have been compared. OMG shape gave minimum SOOT but maximum NO on other hand our baseline shape of HEMI gave maximum SOOT. DC shape gave least NO emissions but to less temperature. Hence it can be conclude that OMG is optimum for SOOT emissions and to improve NO emission performance other options like blending of fuel can be explored.
- Efficiency and BSFC have been compared to check performance. OMG shape found out to be best performer for both parameters.
- On a concluding note, it can be said that OMG is the optimum shape for the current study. It showed better TKE results with good velocity vectors. The higher temperature resulted in reduced SOOT but increased NO. And the efficiency and BSFC is best among all.

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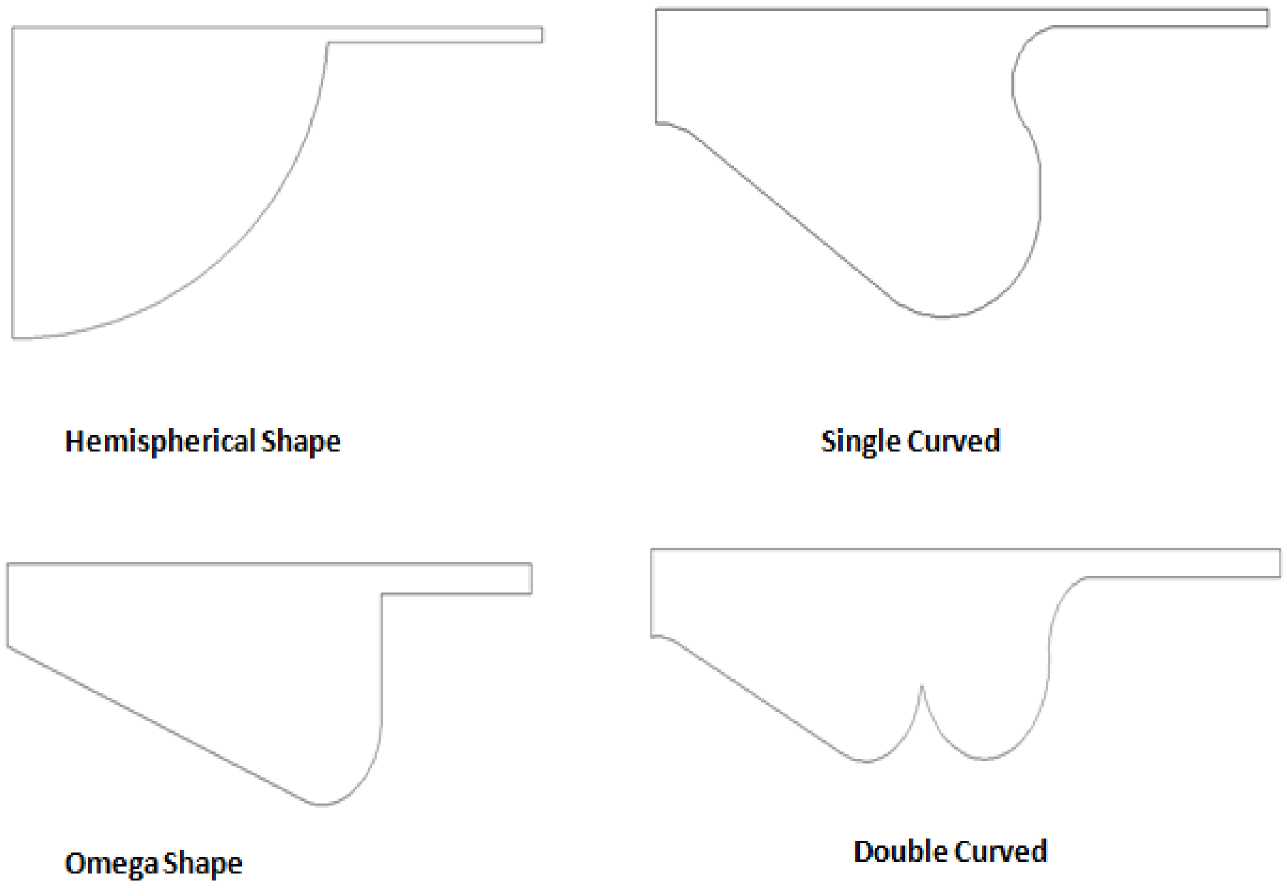
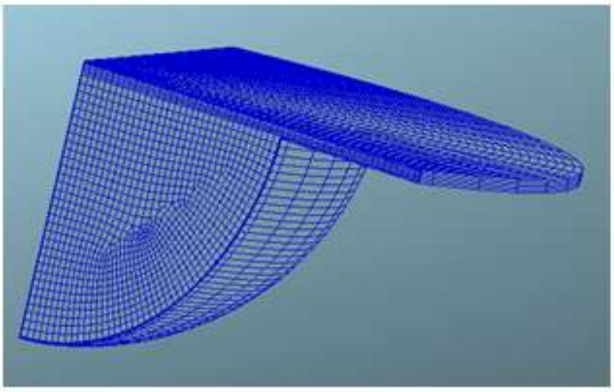
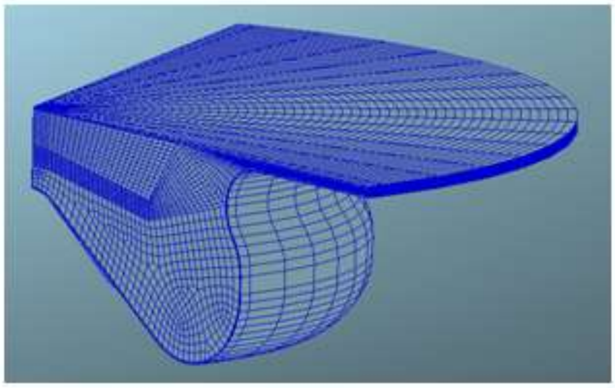


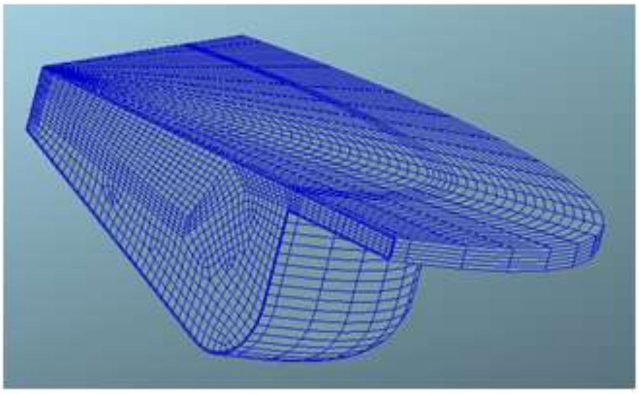
Fig 1:- Schematic Shapes of Piston



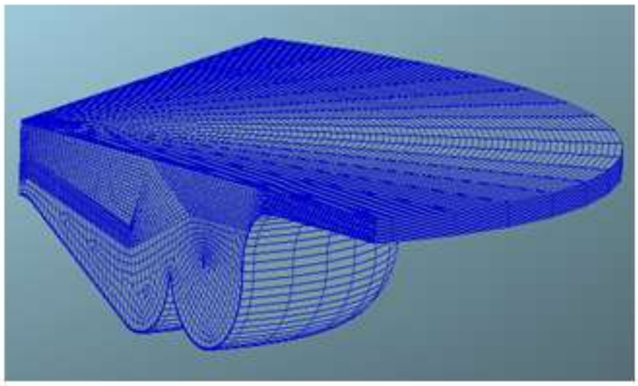
Hemispherical Shape



Single Curved



Omega Shape



Double Curved

Fig 2:- Mesh at TDC

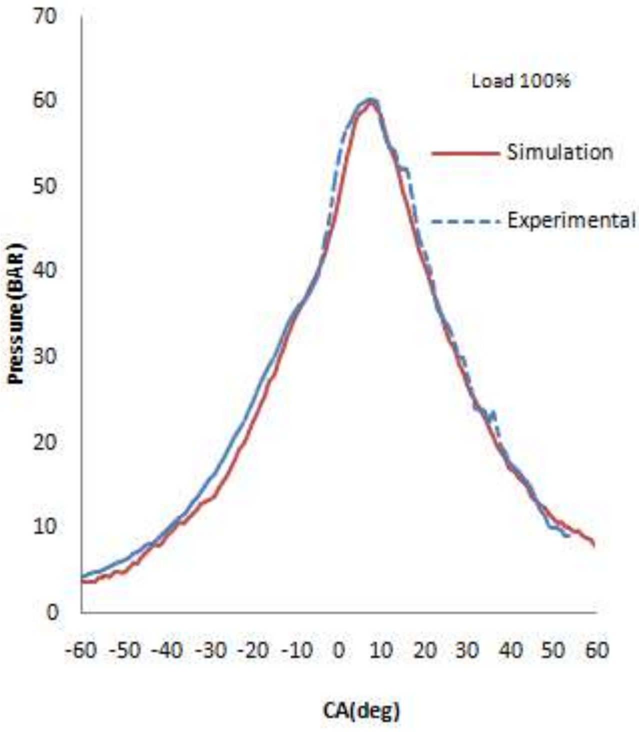


Fig 3(a)

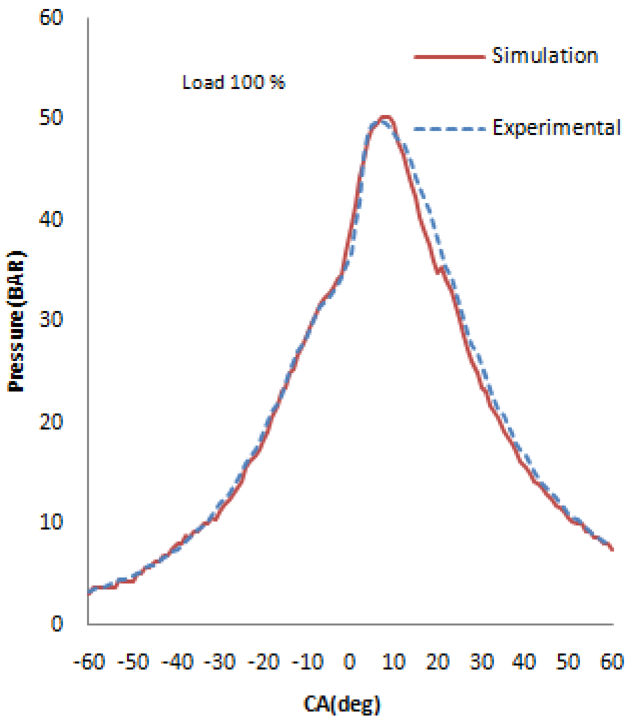


Fig 3(b)

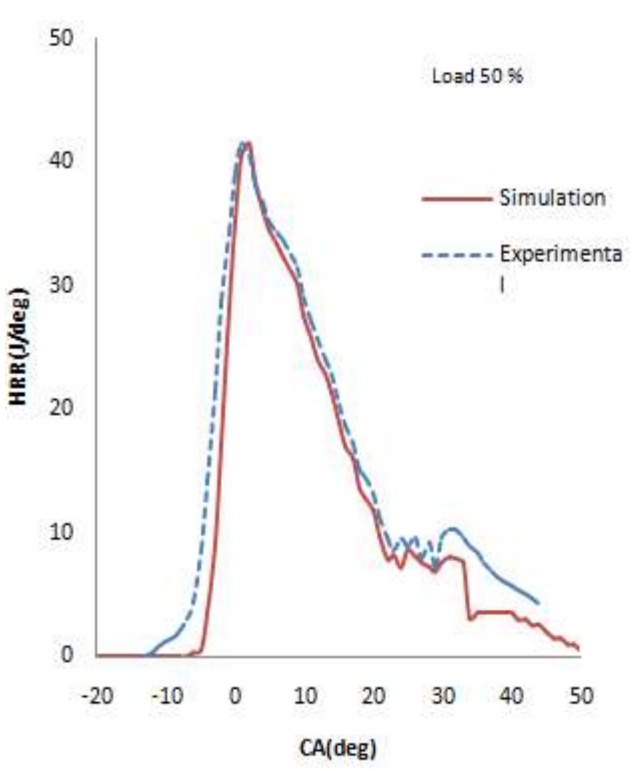
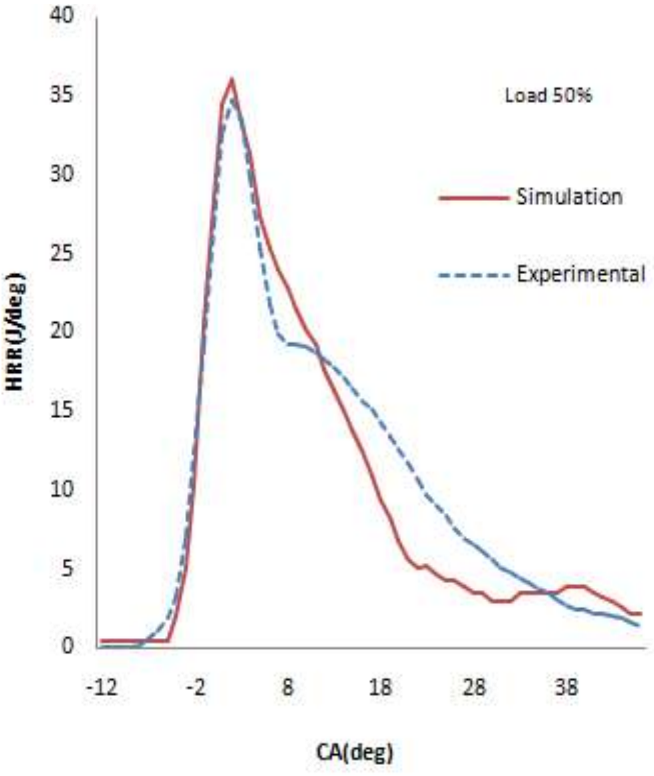


Fig 4(a)



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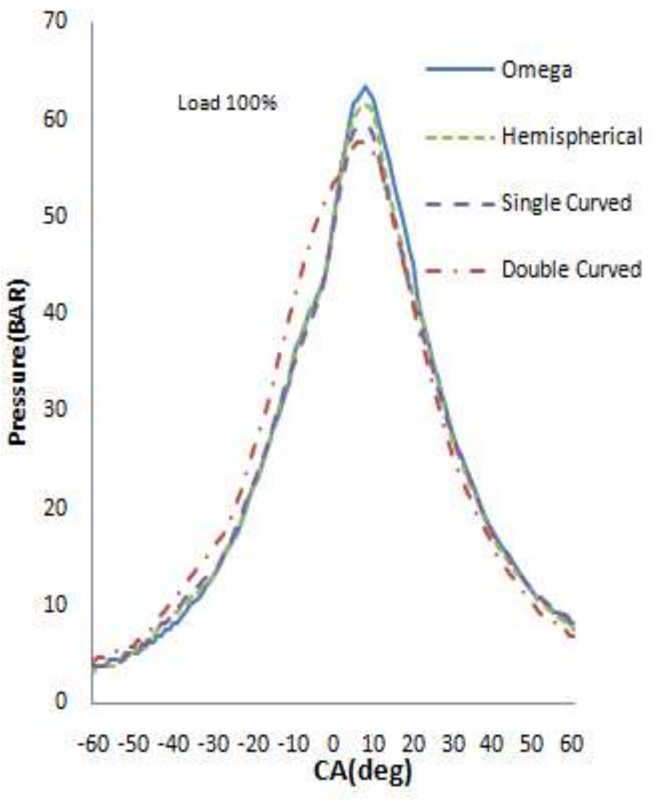

Fig 5(a)

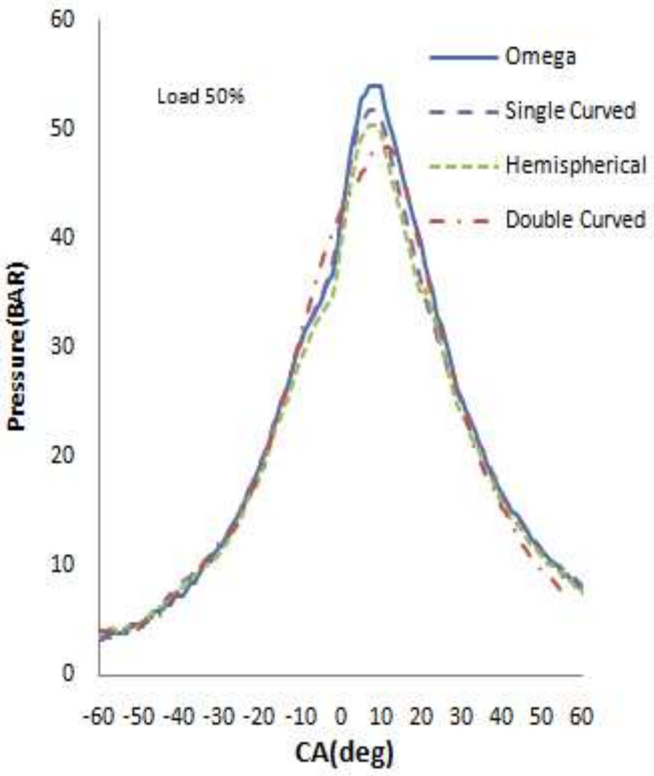

Fig 5(b)

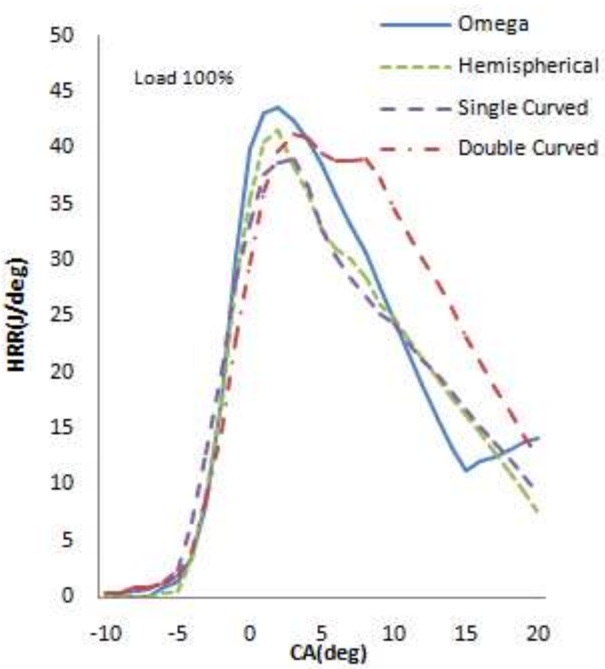

Fig 6(a)

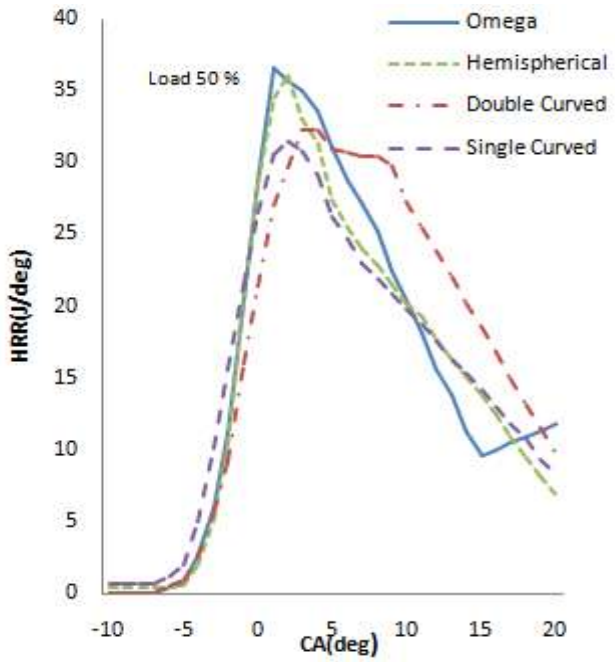

Fig 6(b)

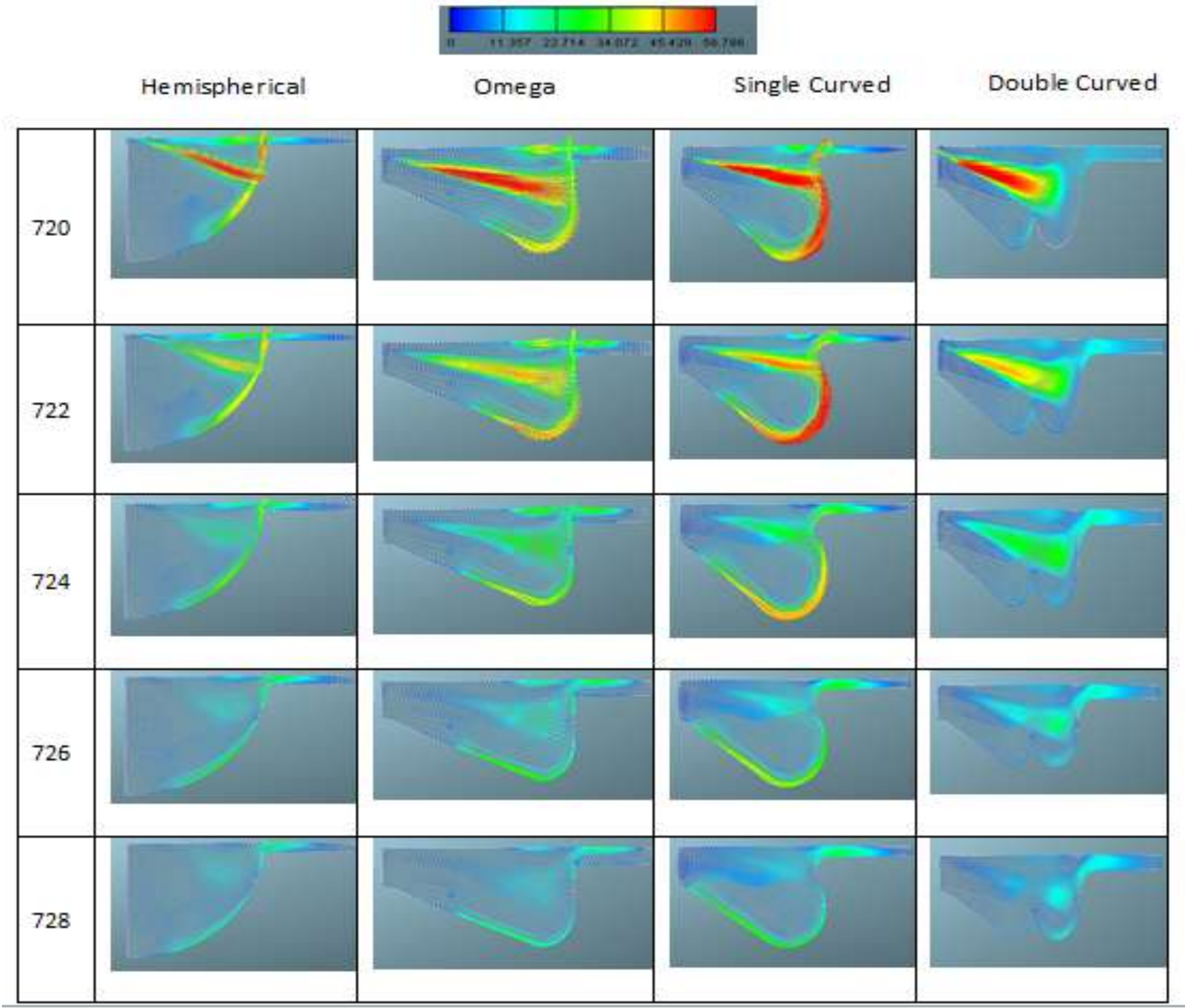

Fig 7: Velocity Distribution

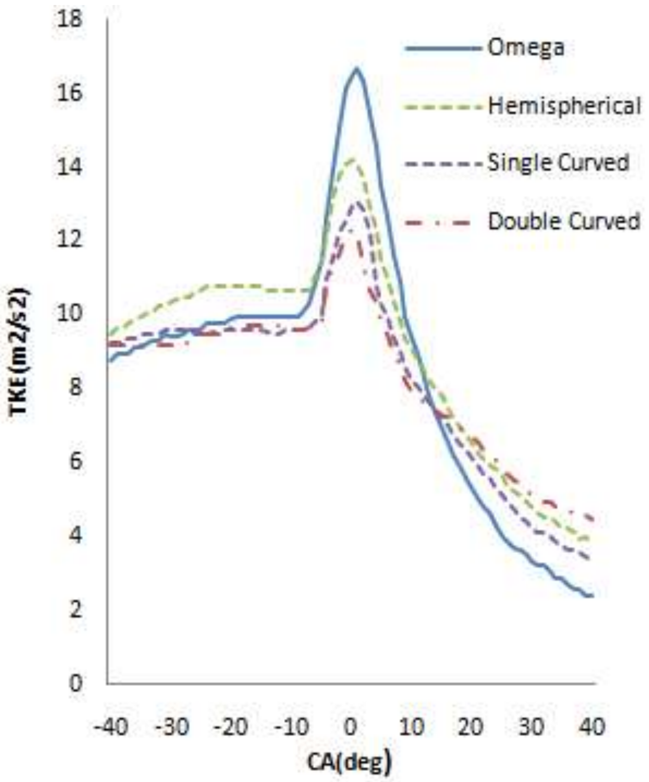

Fig 8(a)

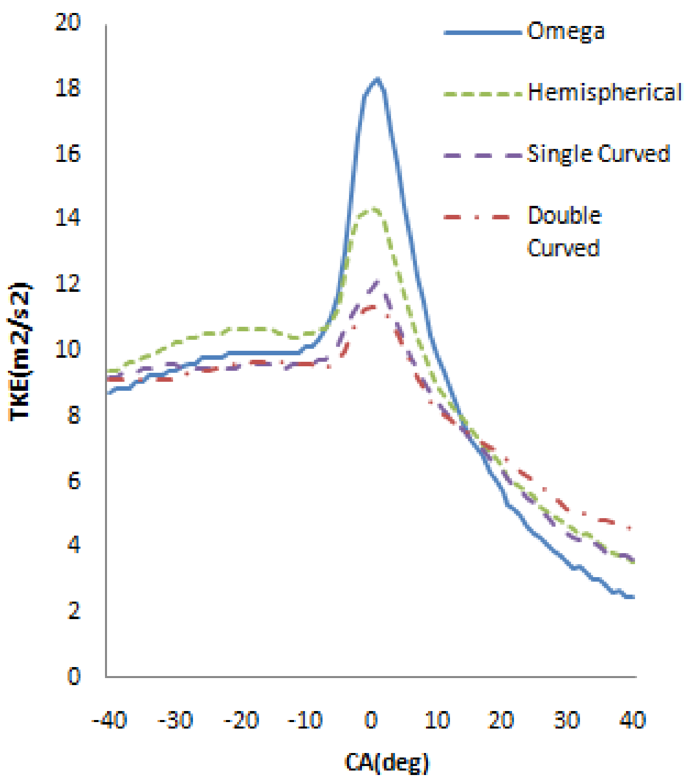

Fig 8(b)

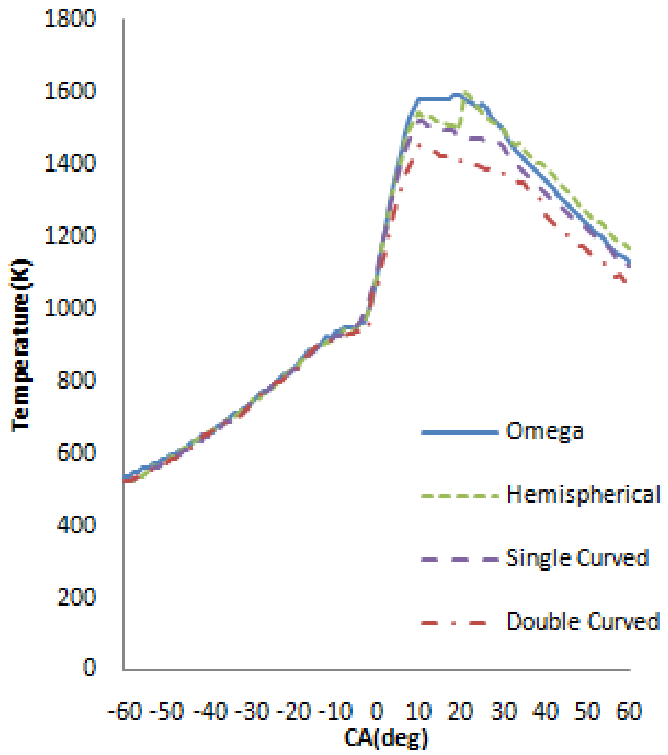

Fig 9

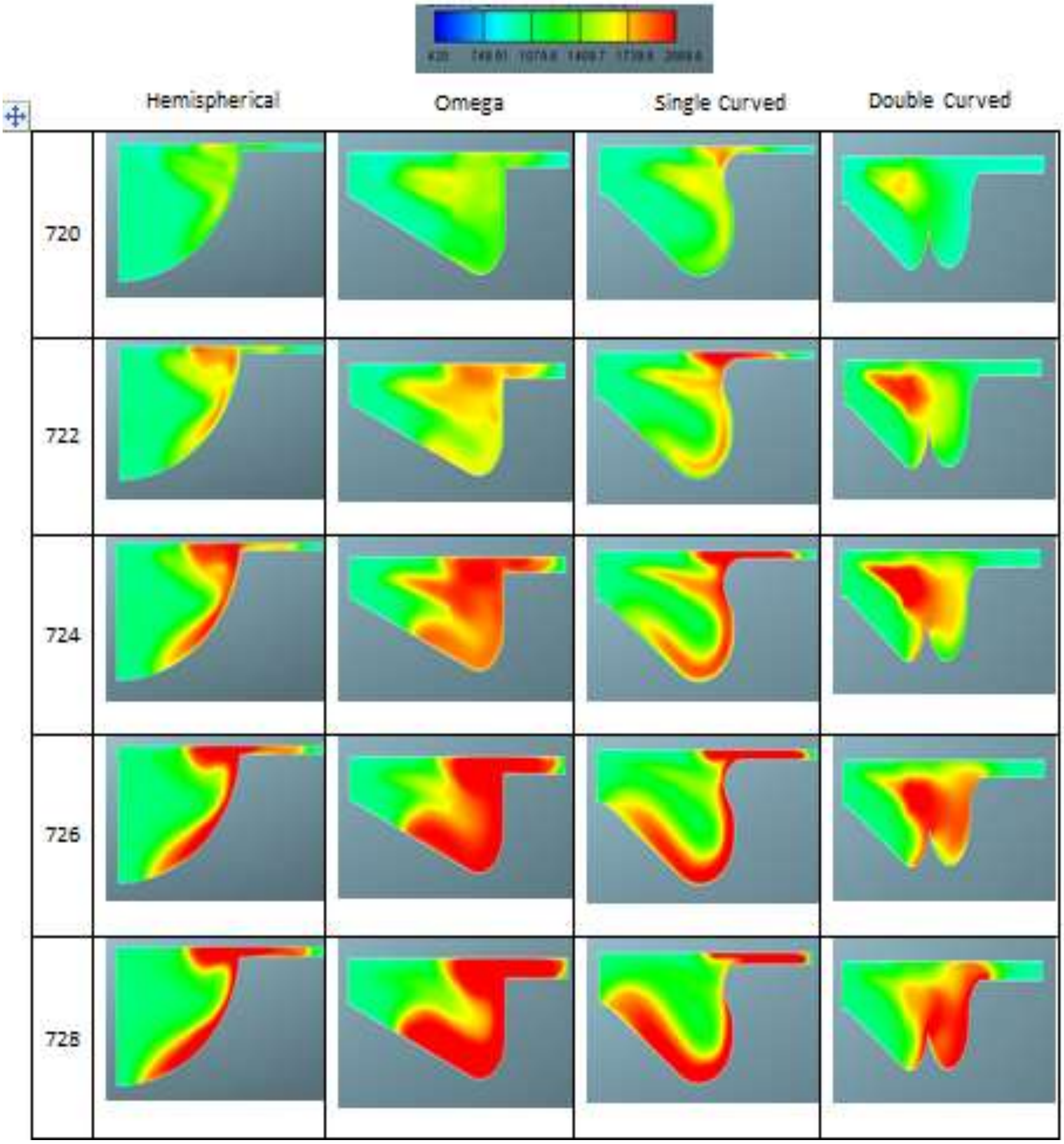

Fig 10: Temperature Distribution

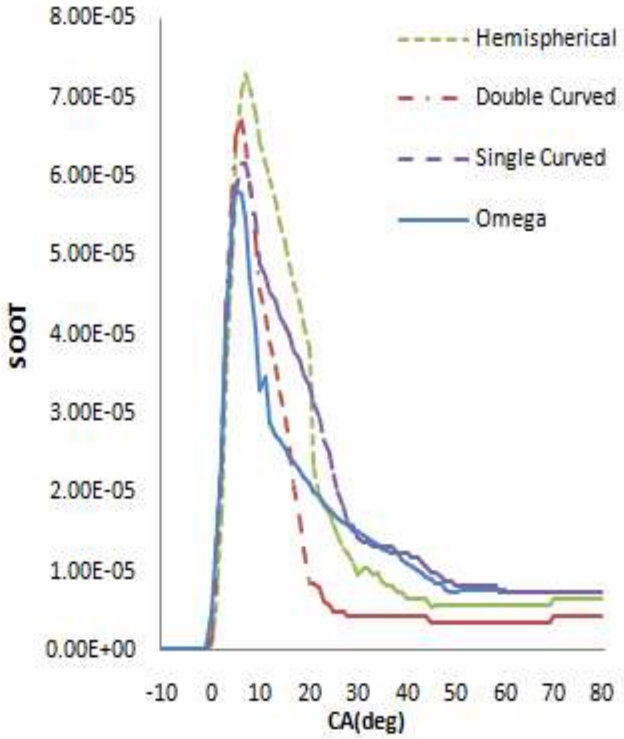

Fig 11(a)

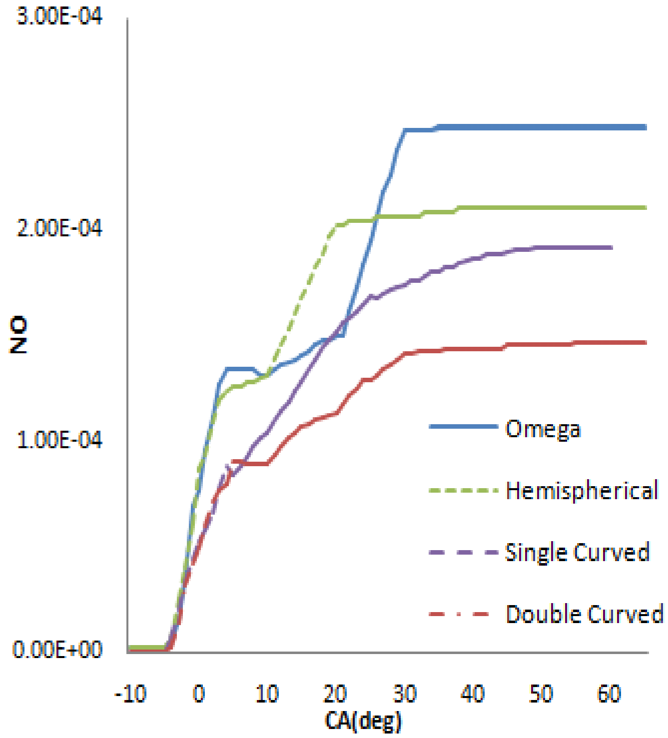

Fig 11(b)

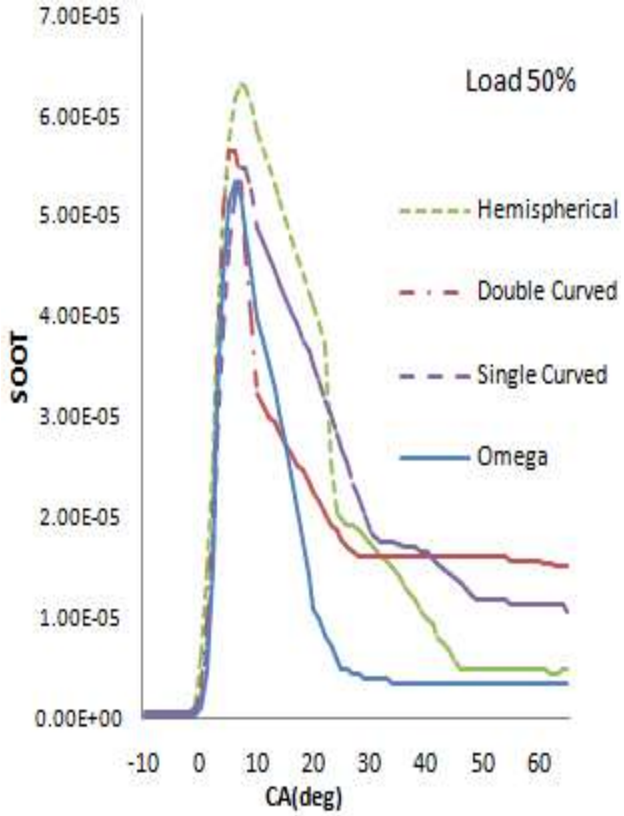

Fig 11(c)

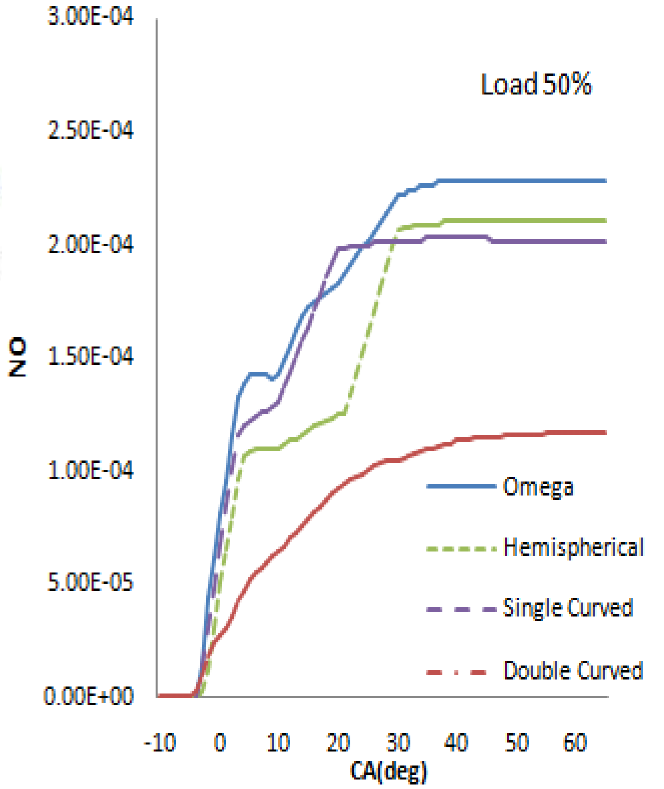

Fig 11(d)

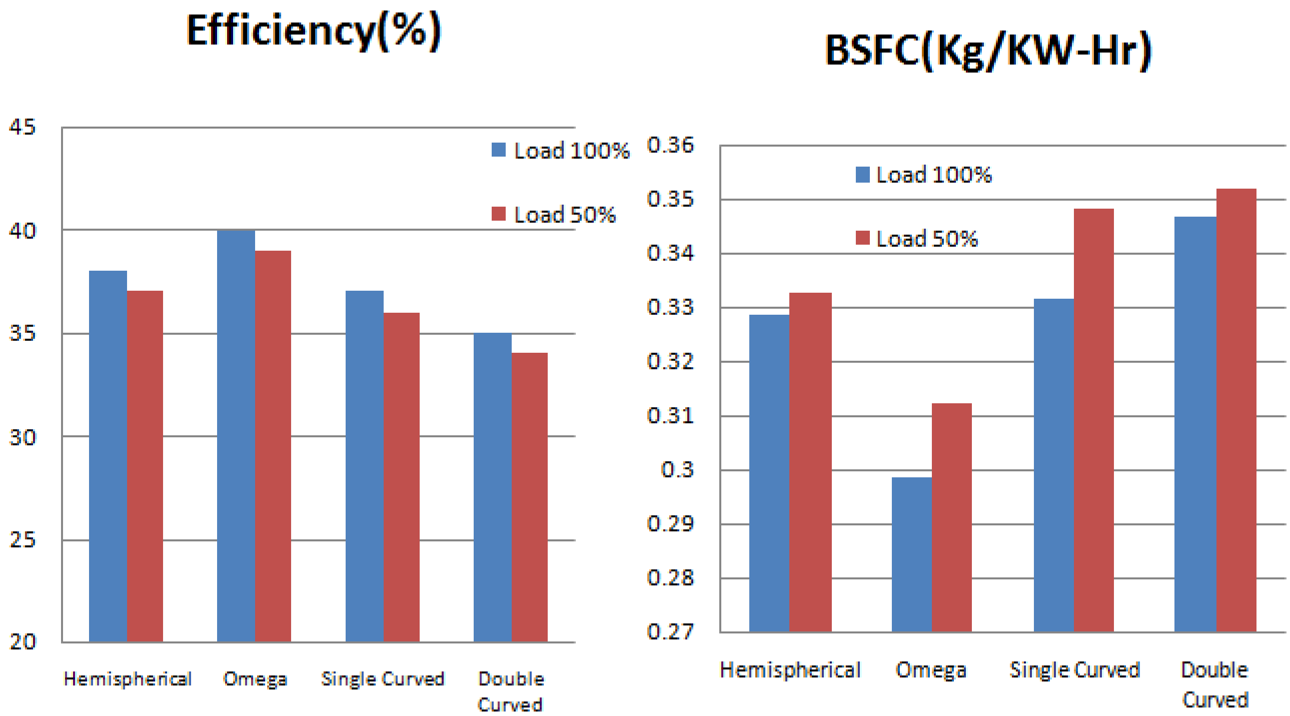

Fig 12: Efficiency and BSFC

Table 1: Engine Specifications

General Specs	Kirloskar (Model -TV1) , Four strokes, compression ignition, Constant Speed vertical, Water cooled, direct injection
Number of Cylinder	4
Bore X Stroke	87.5 mm X 110 mm
Rated output	5.2 kW
Compression ratio	17.5: 1
Rated speed	1500 rpm
Piston Bowl	Hemispherical
Number of injectors	4

Table 2: Mesh Independency Test

Cell Size	Cell numbers	Max Pressure(Bar)	% Variation	Max HRR(J/deg)	% Variation
1 mm	20,568	58.91	-1.88	38.12	-8.23
0.9 mm	23,896	59.09	-1.58	38.96	-6.21
0.8 mm	29,324	59.45	-0.98	39.56	-4.77
0.75 mm	32,184	60.04(Baseline)	0	41.54	0
0.7 mm	36,612	60.06	+0.03	42.02	+1.15
0.65 mm	40,440	60.07	+0.033	42.10	+1.34

Table 3 Model Summary

Phenomenon to be modelled	**Model**
Breakup	WAVE
Wall interaction	Walljet1
Turbulence	K-zeta-f
Evaporation	Dukowics
NO	Extended Zeldovich
SOOT	Kinetic Model