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Research Article

Investigation on Diesel Engine for Airflow and Combustion in a Hemispherical Combustion Chamber

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Abstract

In the present work, computational fluid dynamics investigation on in-cylinder flow for non-reacting as well as firing condition in a DI diesel engine with different types of combustion chamber geometries has been carried out. The multi-dimensional CFD code STAR-CD is used for the simulation of air motion inside the cylinder and combustion process in a DI diesel engine. The investigation has been carried out to study the effect of combustion chamber geometries and engine speed on in-cylinder air motion under non-reacting conditions and the effect of injection timing, injection rate shaping on combustion and emission process. Based on the combustion simulation studies carried out in this investigation best possible performance and emission characteristics from a DI Diesel engine are predicted.

Keywords: *Combustion Geometry, Injection Timing.*

1. Introduction

For most of the twentieth century, Internal Combustion engines have been a relatively inexpensive and reliable source of power for applications ranging from domestic use to large scale industrial and transport applications. Their dependability, along with a seemingly inexhaustible fuel source, led to their widespread acceptance within a few decades of their introduction. However, heightened concern over the environmental impact of Internal Combustion engines has led to increasing governmental regulation regarding the emission and fuel economy performance of both Compression Ignition (diesel) and Spark Ignition Internal Combustion engines.

More stringent emission laws, along with the need to conserve the limited resources of petroleum, have increased the need for an in depth study of the process which occurs in internal combustion engines. The in-cylinder process in direct injected (DI) engines, particularly DI diesel engines are very critical. In this system, the fuel is injected at very high pressure directly into the combustion chamber. The high pressure along with high combustion gas temperatures causes the fuel spray to break up into very small droplets and vaporize. In addition, interaction of the spray jet with piston surface and cylinder wall can be of a significant source of secondary fuel break up, especially in small capacity bore diesel engines used in passenger cars. At some time following the start of fuel injection, the mixing of fuel vapor and air, coupled with sufficient high temperature, results in auto ignition. The amount of time between the start of fuel injection and auto ignition is called as ignition delay and can usually be identified as a sharp inflection point in the cylinder pressure history. The current understanding of diesel combustion is that the initial combustion occurs very rapidly, as the ignition flame kernel spreads quickly through the fuel vapor and air which had become mixed during the ignition delay period. This phase of combustion is known as premixed combustion. Following this phase, the reminder of the combustion is thought to be of controlled mixing, due to the fact that the characteristic time for combustion is much higher than that for mixing. Since the mixing in this phase can be characterized as turbulent diffusion, this second part of the combustion is known as the diffusion combustion. Throughout the combustion process, combustion products form, including two key pollutants, oxides of nitrogen and particulates, known as soot. In recent years, increasing attention has been paid to the mechanism by which NO_x and soot are formed, and in the case of soot in particular, how they are oxidized.

2. Description of present work

All of the process in diesel combustion, from ignition to pollutant formation and oxidation are strongly dependent upon the in-cylinder flow process. There is clearly need to gain better insight into the fluid

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dynamics of in-cylinder flow in Internal Combustion engines. In spark ignition engines, the flame propagation rate, and therefore the time for complete combustion is also governed by the in-cylinder fluid mechanics. In DI diesel engines, the fuel-air mixing and subsequent combustion is controlled by the flow field in the cylinder and fuel injection characteristics. Fuel spray break-up, evaporation, and fuel vapor -air mixing prior to ignition are all strongly dependent upon the local flow fields in the neighborhood of the fuel droplets.

In this work the flow analysis was carried out on open bowl combustion chamber geometry available in literature (Chen et al 1998), the results were compared with experimental data. It was observed that the flow pattern obtained were in good agreement with experimental data. Further flow analysis was conducted on Reentrant combustion chamber and Reentrant with central projection combustion chamber. The predicted flow field of the reentrant with central projection combustion was found to be of a better configuration. Hence further numerical analysis was conducted on reentrant with central projection combustion chamber to investigate effect of different injection strategies.

3. Specifications of Engine

Table 1 Engine Specifications

Number of cylinders	One
Bore	79.5 mm
Stroke	95.5 mm
Connecting rod length	144 mm
Intake valve dia	35 mm
Exhaust valve dia	30.5 mm
Bowl dia	37.0 mm
Compression ratio	19.5:1
Engine speed	1000 rpm

The computational fluid dynamics code, STAR-CD has been used in this study. Flow conditions inside the cylinder are predicted by solving momentum, continuity and energy equations.

4. Analysis

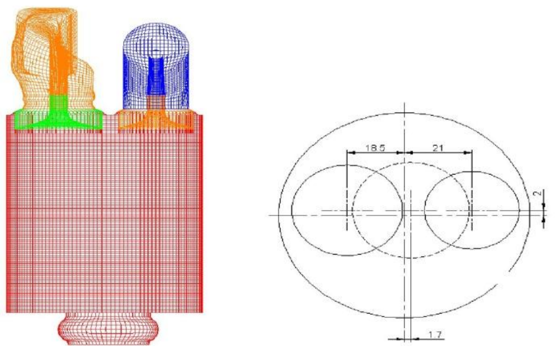

Fig.1 Computational domain of open bowl combustion chamber geometry

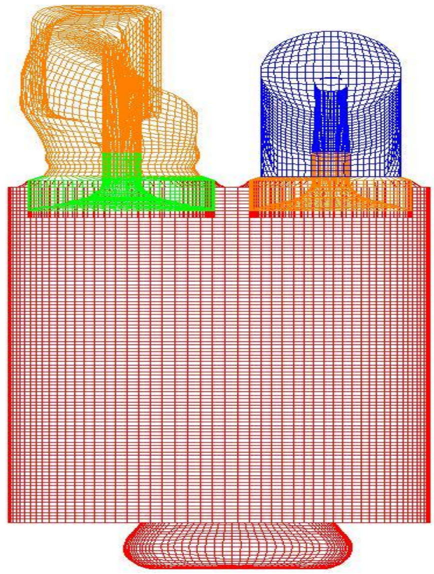

Fig.2 Computational domain of reentrant combustion chamber geometry

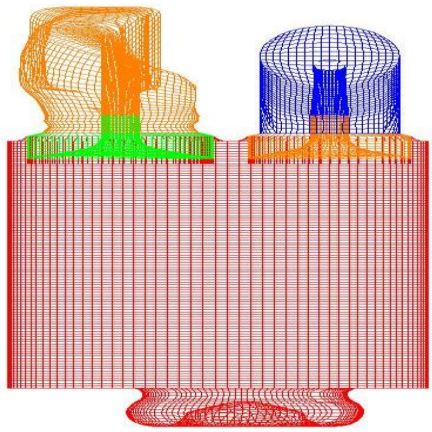

Fig.3 Computational domain of reentrant with central projection combustion chamber geometry

4.1 Simulation of Combustion Process

The simulation studies was carried out for various reacting condition to investigate the effect of early injection, EGR, multiple injection and injection rate shaping on the combustion performance and emission characteristic of the engine. The predicted flow field variation of the computational domain reentrant with central projection bowl chamber has been used for combustion simulation.

4.2 Effect of Injection Timing and EGR

In order to study the effect of early injection and EGR on the combustion performance and emission characteristic of the engine, three injection timing , start of Injection (SOI) 12 deg bTDC, 16 deg bTDC and 20 deg bTDC are considered and three EGR fraction 0% EGR,10% EGR and 20% EGR are considered for this study.

Table 2 Boundary and initial conditions

Intake manifold	
Type	Constant pressure
Turbulent intensity	5%
Length scale	1 mm
Temperature	363K
Exhaust manifold	
Type	Constant pressure
Turbulent intensity	5%
Length scale	1 mm
Temperature	400K
Initial conditions in intake manifold	
Pressure	1.2 bar
Temperature	363K
Turbulent intensity	5%
Length scale	1 mm
Initial conditions in exhaust manifold	
Pressure	1.3 bar
Temperature	400K
Turbulent intensity	5%
Length scale	1 mm
Initial conditions in the cylinder	
Pressure	Constant pressure
Temperature	450K
Turbulent intensity	5%
Length scale	1 mm

4.3 Simulation for multiple injections

Simulation was carried out to study the effect of Multiple injections on combustion performance and emission characteristic of the engine under consideration. Four types of multiple injection system were considered for this study is shown in Table 4.3. The start of injection is 12 deg BTDC fixed for all cases and the injection duration is 20 deg. The pilot and main injection pulses are separated by dwell period of 6 degrees CA. The pilot injection quantity is varied from 0% to 50%. And the main injection quantity is varied from 100% to 50%.

5. Results and discussions

The DI diesel engine performance is greatly influenced by in-cylinder fluid dynamics. In particular, in-cylinder fluid dynamics contribute to the fuel-air mixing, which is one of the most important factors for controlling the fuel burning rate. It significantly affects the ignition delay, magnitude of the premixed combustion, magnitude and timing of the diffusion combustion and finally the emissions of oxides of nitrogen and soot. There are two characteristics that define the in-cylinder flow fields, i.e bulk fluid motion and turbulence. One parameter that characterizes the bulk motion is the swirl. Swirl in diesel engine is known to be an important parameter that affects the mixing of fuel jet. Swirl that is generated during the intake stroke as a result of intake port geometry and its orientation is called as induction swirl. Swirl that is generated during the compression stroke as result of combustion chamber geometry and it's orientation is called as compression swirl.

In internal combustion engines, the fuel evaporation and mixing processes are strongly influenced by the turbulent nature of the in-cylinder flow. The gradient of velocity in the mean flow is one of the major reasons for turbulent fluctuations. The air-jet created by flow during the intake and compression process interacts with the cylinder wall and moving piston to generate large scale rotating flow, both in the vertical and as well as in horizontal planes. The behavior of the in-cylinder turbulent flow can be characterized by monitoring the kinetic energy and the integral length scale variation of turbulent eddies that contribute to turbulence production during intake and compression processes. In-cylinder flow characteristics at the time of fuel injection and subsequent interactions with fuel sprays and combustion are the fundamental considerations for the engine performance and exhaust emissions of a diesel engine.

5.1 Validation of Mean Velocity during Suction Stroke

The numerical simulation was conducted at an engine speed of 1000 rpm for the purpose of validation. Figures 5.2 to 5.4 show the validation of velocity along the X direction (U) for the crank angles 50, 90 and 120 respectively. Positive value indicates flow towards the intake valve and negative value indicates that the flow is away from the intake valve and towards the exhaust valve.

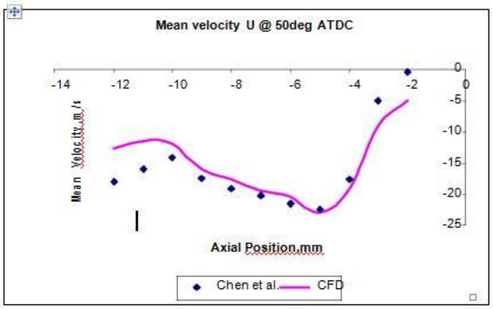

Fig.4 Mean velocity U at 50 degree ATDC

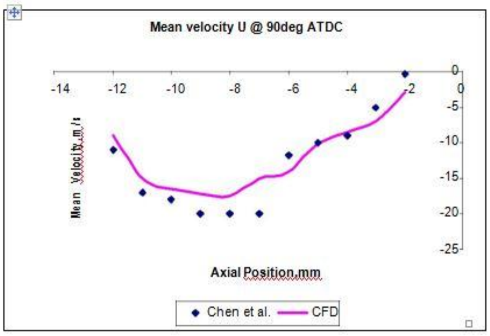

Fig.5 Mean velocity U at 90 degree ATDC

5.2 Effect of combustion chamber geometry

Simulation is performed out to study the effect of combustion chamber geometry on flow structure. Three combustion chamber geometries are considered, namely

1. Open Bowl.
2. Reentrant bowl.
3. Reentrant with central projection.

5.2.1 Swirl ratio inside the cylinder

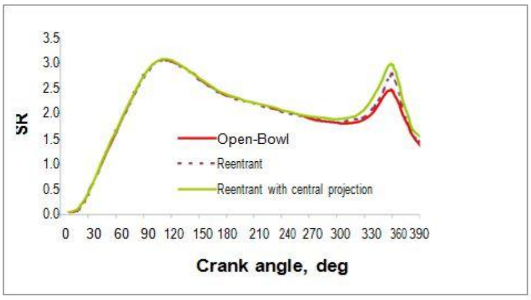

Fig.6 Swirl Ratio vs. Crank angle for different combustion chamber

The swirl is generated during the early stroke of intake and the maximum value is reached in between 110 - 120 deg CA ATDC at which the piston reaches its maximum instantaneous speed. After maximum instantaneous speed, the discharge velocity into the cylinder decreases and swirl drops slowly during the rest of the intake stroke. During the suction stroke the swirl ratio for all three combustion chamber geometries are almost identical, because the swirl was generated by the intake port geometry. As a result, the combustion chamber geometry does not influence the flow during the intake stroke significantly. Since the intake port geometry is helical in shape for all three cases, it generates high swirl during early stage of valve opening and it reaches maximum of 3.08 at full valve opening. The losses due to friction are low during suction stroke and it increases the swirl. After the starting off valve closing (120 deg ATDC), there is a reduction of mass flow rate of the incoming air and the friction between the wall and air inside the cylinder decreases the swirl ratio. This decreasing trend continues in the first part of the compression stroke. Early during the first stage of compression stroke, between BDC and 80 deg CA after BDC the stratified swirl structure obtained at the end of the intake stroke is maintained. However, as the compression advances in the second stage between 80 deg CA after BDC and TDC the axial upwards flow induces a gradual increase of the swirl velocity in the top part of the piston. However, when piston approaches TDC, swirl is enhanced as the flow accelerates in preserving its angular momentum within the piston bowl. The maximum swirl was found at 360 deg CA i.e. at compression TDC.

Table 3 Mass average swirl ratio for three different combustion chamber geometries

Combustion chamber geometry	Maximum swirl ratio during suction stroke	Maximum swirl ratio during suction stroke
Open bowl	3.07	2.49
Reentrant	3.07	2.78
Reentrant with central projection	3.08	2.97

It is observed that, reentrant chamber with central projection shows a swirl ratio of 19% higher than base line bowl. The projection at the center of the reentrant bowl reduces the effective diameter of the chamber, thereby increasing the angular momentum, which results in higher swirl ratio. After compression TDC, during the expansion stroke, there is a reverse squish as the flow exits from the piston bowl and wall friction contributes to the sudden fall in the swirl velocity.

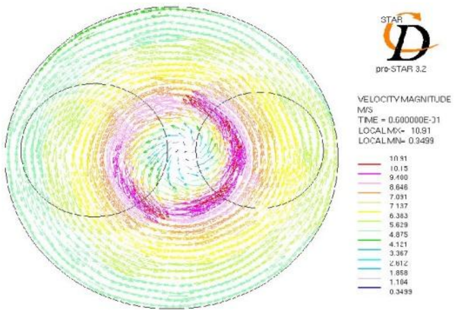

Fig.7 Velocity vector in the bowl entry at 360 deg CA ATDC Open bowl

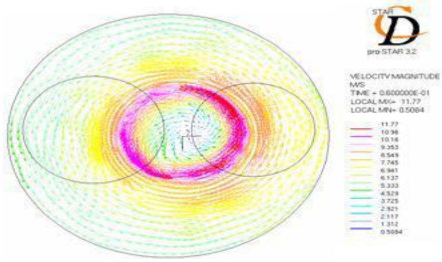

Fig.8 Velocity vector in the bowl entry at 360 deg CA ATDC – Reentrant

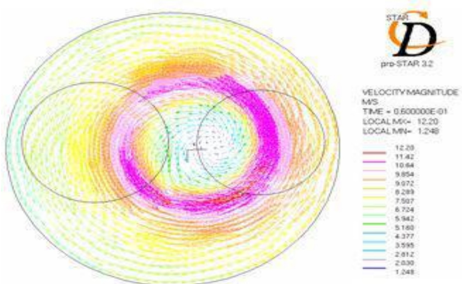

Fig.9 Velocity vector in the bowl entry at 360 deg CA ATDC –Reentrant with central projection.

The mean velocity vectors in the bowl entry at 360° ATDC for different combustion chamber configurations. The predicted velocity is low for the open bowl configuration compared to the other two reentrant bowl configurations. The intensity of maximum velocity at the end of compression stroke is observed in Reentrant with central projection bowl chamber geometry and a strong recirculation is sustained due to this bowl configuration. This helps in better fuel air interaction, which may lead to better combustion and performance.

5.3 Turbulent kinetic energy inside the cylinder

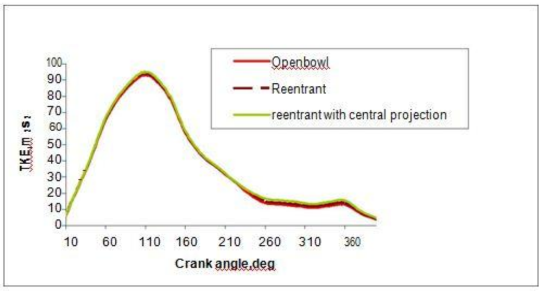

Fig.10 Mass averaged turbulent kinetic energy for different combustion chamber geometries.

After starting off valve closing (120 deg CA) ATDC, the TKE decreases with respect to the piston movement. This is due to the mass flow reduction caused by the closing of valve. The mass averaged turbulence kinetic energy during rest of the intake and compression strokes is similar for all the three cases. From the Figure 5.17 to 5.19 lower values could be expected for open bowl and reentrant chamber, since the squish effect in these combustion chambers is smaller. However the turbulent kinetic energy for the reentrant with central projection chamber at compression TDC and early stage of expansion stroke slightly higher. Thus reentrant with central projection combustion chambers seem to conserve their turbulent energy better.

5.4 Effect of engine speed

Simulation is carried out to study the effect of engine speed on flow parameters inside the cylinder.

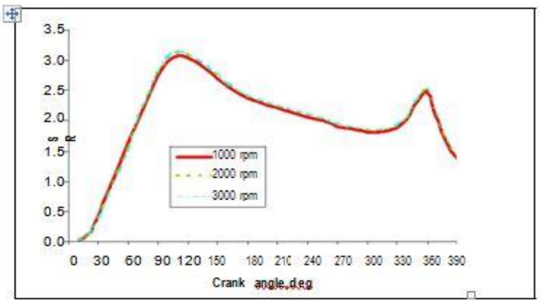

Fig.11 Mass average swirl ratio for different engine speed – open bowl combustion chamber

The flow in the cylinder during the intake and compression stroke has been analyzed, and a comparison is made for all the three combustion chamber geometries at different engine speeds (1000, 2000, and 3000 rpm) with an intake pressure and temperature of 2.01 bar and 323 K respectively.

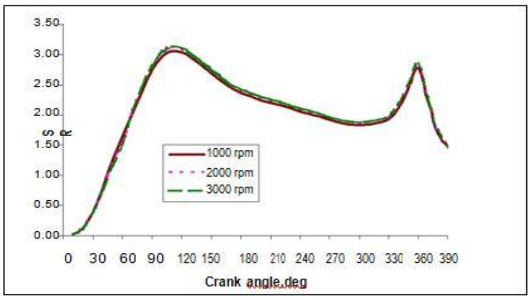

Fig.12 Mass average swirl ratio for different engine speed – reentrant combustion chamber

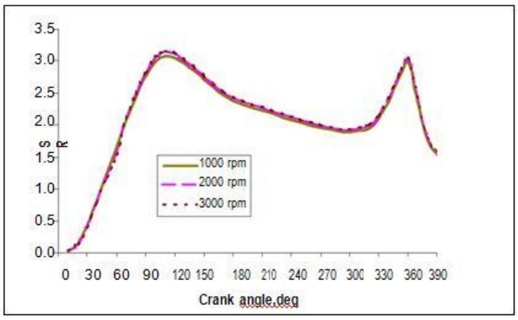

Fig.13 Mass average swirl ratio for different engine speed – reentrant with central projection combustion chamber

During the early stage of suction stroke, the swirl ratio decreases with increase in engine speed. This is probably because the effect of intake air jet decays the angular momentum. Afterwards swirl ratio increases with increase in engine speed. This is due to the fact that there is a reduction in decay in the angular momentum which results in increase in swirl ratio. The maximum swirl ratio during suction stroke is in between 110 -120 deg CA at which the piston reaches its maximum instantaneous speed. After the starting of valve closing (120 deg ATDC), there is a reduction of mass flow rate of incoming air and the friction between the wall and air inside the cylinder decreases the swirl ratio. The trend continues until the piston nears the TDC. The maximum swirl ratio is obtained during compression stroke when piston reaches the TDC. The swirl ratio is increases with increase in engine speed during compression stroke, but the variation of swirl ratio is marginal. From the above results it is concluded that the engine speed has no appreciable effect on swirl ratio.

5.5 Simulation of injection process

The investigation is extended to reacting condition to study the effect of injection timing, EGR, Multiple injections and injection rate shaping on the engine performance and emissions. The fuel taken for this analysis is Dodecane ($C_{12}H_{26}$) as its properties are closer to Diesel fuel. For simulating combustion first a chemical reaction scheme is defined, which is a single step global reaction. A proper reaction rate mechanism based on Eddy break up model is also specified. Based on the shell auto ignition model the combustion is initiated and it proceeds according to the Eddy brake up model.

Table 4 Injection profile

Nozzle opening Pressure	280 bar
Start of Injection	12 deg BTDC
Injection Duration	20 deg
Number of holes	3
Hole diameter	0.280 mm
L/D ratio	3.51

5.6 Effect of Injection Timing on Combustion

Simulation is carried out to study the effect of injection timing on combustion performance and emission characteristic of the engine operating at a speed of 1000 rpm. A single step injection profile is considered for study. The start of injection (SOI) is 12, 16 and 20 deg. BTDC.

5.7 In-Cylinder Pressure and Temperature

The effect of injection timing on cylinder average combustion pressure and temperature. The peak pressure obtained for three-injection timing is summarized in Table. From the graph the peak pressure increases with advancing the injection timing. The peak pressure of 13.8% and 5.9% is obtained when the fuel is injected at 20° BTDC and 16° BTDC respectively. The pressure and temperature obtained at 6° ATDC plotted in Figures 5.25 to 5.30.

Table 5 Comparison of Peak pressure and temperature for various injection timing

Start of Injection, deg BTDC	Peak Pressure, bar @ deg CA	Peak Temperature, K @ deg CA
20	115@364.3	1197 @ 373.8
16	107 @ 365.2	1185 @ 375
12	101 @ 366.3	1169 @ 380

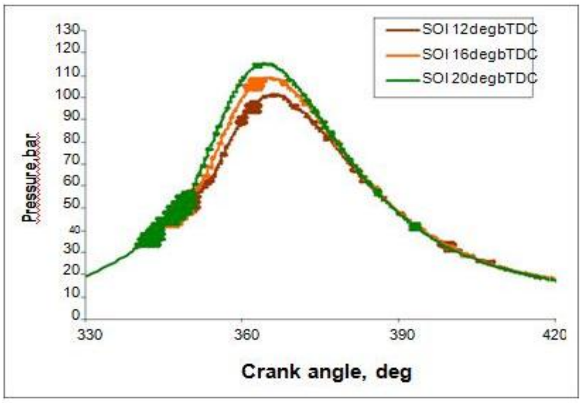

Fig.14 Cylinder average pressure for various injection timing

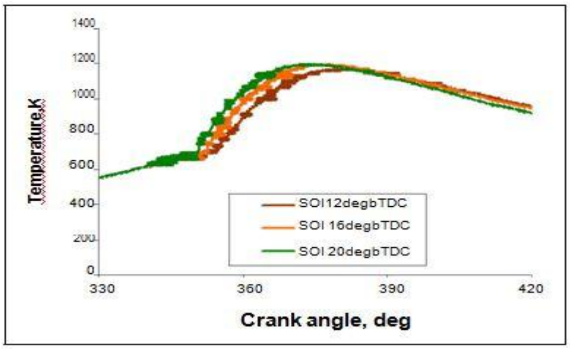

Fig.15 Cylinder average temperature for various injection timing

It is observed that, the peak temperature is 2.3% and 1.3% higher with reference to the base line results (12° BTDC) obtained for the injection timings of 20° BTDC and 16° BTDC respectively. This is due to the fact that, as the injection timing is advanced, ignition delay increases. Further, pressure and temperature inside the cylinder are not sufficient to ignite the fuel. Since a large amount of evaporated fuel is accumulated during the ignition delay period, there is a rapid combustion due to which pressure shoot up. It was also observed that the occurrence of peak pressure and temperature advances with early injection.

5.8 Heat release rate

Table 6 Ignition Delay, Peak heat release rate, combustion duration for various injections

Start of Injection, deg BTDC	Ignition delay, deg	Peak Heat Release Rate, J/deg	Combustion Duration, deg
20	10.3	178	49.4
16	8.2	125.2	50.2
12	6.6	97.1	55.5

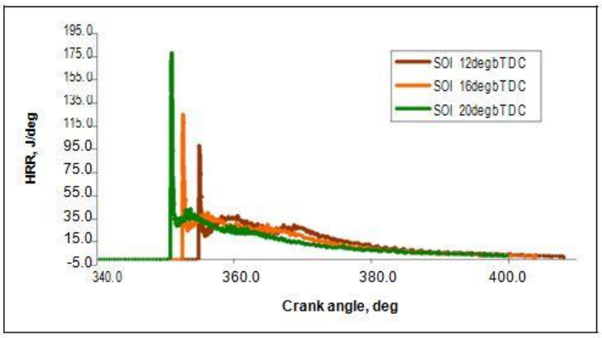

Fig.16 Heat release rate for various injection timing

The advanced injection of 20 deg BTDC and 16 deg BTDC shows higher peak heat release rate as compared to base line case of 12 deg. BTDC. Advanced injection leads to longer ignition delay which results in larger accumulation of charge before the start of combustion. Longer ignition delay is due to lower values of pressure and temperature inside the cylinder during the initial period of fuel injection at advanced injection timings. The longer ignition delay leads to rapid burning rate and the pressure and temperature inside the cylinder rises suddenly. Hence, most of the fuel burns in premixed mode causing higher peak heat release rate and shorter combustion duration. Whereas the in baseline case of 12 deg BTDC the ignition delay is short causing accumulation of relatively less amount of evaporated fuel. Shorter ignition delay is due to the fact that pressure and temperature inside the cylinder during the initial period of fuel injection is high. The shorter ignition delay shortens the mixing time, which leads to slow burning rate and slow rise in pressure and temperature. Hence, most of the fuel burns in diffusion mode rather than in premixed mode resulting in lower peak heat release rate, longer combustion duration.

Conclusions

1) Maximum swirl ratio 3.07 occurs at maximum valve lift position during intake stroke and 2.98 at the end of compression stroke.
2) Reentrant with central projection combustion chamber geometry shows maximum swirl ratio at TDC as compared with the other two combustion chamber geometry configurations. This will result in better combustion characteristics.
3) Peak pressure increases with early injection timing. Peak pressure is 109 bar and 115 bar when the fuel is injected at 16˚ bTDC and 20˚ bTDC respectively.
4) Occurrence of peak pressure also advances with early injection. Since the combustion duration decreases with early injection, it is an indication of HCCI mode of operation.
5) The reentrant with central projection combustion chamber is optimum design compared with other two combustion chamber geometry which gives better air motion.

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