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Computational flow analysis in combustion chamber of CI-Engines

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Abstract

Recent years have witnessed rapid developments in the fields of IC engines. Today both the manufacturers and the customers are looking for low polluting and better efficient engines. Basically, the in-cylinder gas motion affects the performance of the engine. So, it is necessary to have a better control over in-cylinder gas motion. With the experimental techniques that are available today, it is possible to make reasonably accurate measurements to study the in-cylinder flow characteristics. But they are costly and time consuming. Over the past one and half decades, fluid dynamic based multi-dimensional models for studying fluid dynamics in-cylinder gas motion are being tried in IC engine field. The main objective of the project is the flow calculation of the compression and expansion stroke of a four-valve direct-injection. Direct engine have been carried out with different combustion chambers. In the main study, the flow characteristics inside the engine cylinder equipped with different piston configuration were compared, for this, complete calculations of the compression and expansion strokes were performed under realistic operating conditions and the ensemble-averaged velocity and turbulence flow field obtained in each combustion chamber analyzed in detail.

Keywords: compression and expansion stroke of a four-valve direct-injection, different piston configuration, combustion chamber

Introduction

The selected engine for study is one of the largely produced models of the plant-the S-325. It is a four-

Stroke three cylinder, diesel engine, suiting applications like agricultural, industrial and generator sets. It weighs about 204kilograms without starter motor, flywheel and its housing. But the typically installed engine's weight shoots up to 290 kilograms. Simpson &Co limited, dates back to about 1840, and when a Scottish man named Mr. Simpson started his trade with producing horse-less wagons or automotives which had been a lucrative business those days.

Simpson, in 1951, pioneered the concept of Diesel Engines for vehicular application in India and the core activity since then has been manufacture of wide range of Diesel Engines from 20 BHP to 120 BHP for industrial, vehicular, agricultural, generator set engines and engines for automotive and marine applications. Simpsons India limited mainly concentrates in the manufacturing of agricultural-tractor engines. Some of its largely produced models are S324, S325 and S318.

In compression ignition engines, air alone is inducted into the cylinder. The fuel (in most applications alight fuel oil, though heated residual fuel is used in marine and power-generation applications) is injected directly into the engine cylinder just before the combustion process is required to start. Load control is achieved by varying the amount of fuel injected in each cycle; the air flow at a given engine speed is essentially unchanged. There are a great variety of CI engines is used in wide range of applications- automobile, truck, locomotive, marine, power generation. Naturally aspirated engines where atmospheric air is inducted, turbocharged engines where the inlet air is compressed by an exhaust-driven turbine-compression combination, and supercharged engines where the air is compressed or blower are common.

Table 1.1: Technical Data

S. No.	Parameters	S-325
1	Type	Three cylinder, four stroke
2	Combustion	Direct injection
3	Bore	91.44 mm
4	Stroke	127 mm
5	Cubic capacity	2.5 lts
6	Compression ratio	18.5:1
7	Firing order	1,2,3
8	Tappet clearance	0.30mm(cold)
9	Atomizer setting pressure	240 kg/cm^2
10	Injection timing(B.T.D.C)	26˚
11	Fuel injection pump	MICO inline type with mechanical governor and flange mounted
12	Inlet valve opens	13˚B.T.D.C
13	Exhaust valve closes	10˚A.T.D.C
14	Tot. Inlet open period	236˚
15	Tot. exhaust open period	236˚

Related Works

Swirl

Swirl is usually defined as organized rotation of the charge about the cylinder axis. Swirl is created by bringing the intake flow into the cylinder with an initial angular momentum. While some decay in swirl due to friction occurs during the engine cycle, intake generated swirl usually persists through the compression, combustion, and expansion process. In engine designs with bowl-in-piston combustion chamber, the rotation motion set up during intake is substantially modified during compression. Swirl is used in diesels and some stratified-charge engine concepts to promote more rapid mixing between the inducted air charge and the injected fuel. Swirl is also used to speed up the combustion process in spark-ignition engines. In two-stroke engines it is used to improve scavenging. In some designs of pre-chamber engines, organized rotation about the pre-chamber axis is also called Swirl. In pre-chamber engines where Swirl within the pre-combustion chamber is important, the flow in to pre-chamber during the compression process created the rotation flow.

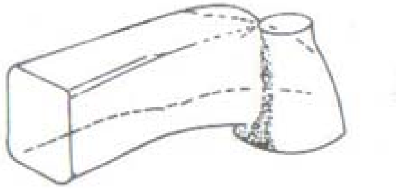

Fig 1: Air swirl movement

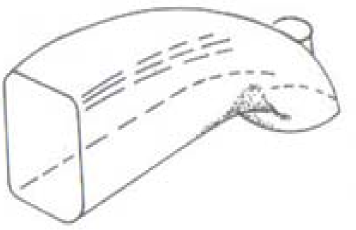

Fig 2: Air swirl movement

Types of Diesel Combustion Systems

Diesel engines are divided into two basic categories according to their combustion chamber design: (1) Direct injection (DI) engines, which have a single open combustion chamber, into which the fuel is injected directly; (2) Indirect-injection (IDI) engines, where the chamber is divided into two regions and the fuel is injected into the “pre-chamber” which is connected to the main chamber (situated above the piston crown) via a nozzle or one or more orifices.

Direct Injection Systems

In largest size engines, where mixing rate requirements are least stringent, quiescent direct-injection systems are used. The momentum and energy of the injected fuel jet are sufficient to achieve fuel distribution and rates of mixing with the air. Additional organized air motion is not required. The combustion chamber shape is usually a shallow bowl in the piston crown and a central multi-hole injector is used.

As the engine sizes decreases, increasing amounts of air swirl are used to achieve faster fuel-air mixing rates. Air swirl is generated by suitable design of inlet port; the swirl rate can be increased as the piston approaches the TDC by forcing the air towards the cylinder axis, into a bowl-in-piston type of combustion chamber. There are two types of design DI engines with swirl in common use. The first one is the DI engine with swirl having a centrally located multi-hole injector nozzle. Here the design goal is to hold the amount of liquid fuel impinges on the piston cup walls to a minimum. The second design is the M.A.N M system with its single-hole-fuel-injector nozzle; oriented so that, most of the fuel is deposited on the piston bowl walls. These two types of designs are used in medium size (10-15cm bore) diesels and with increased swirl, in smaller-size (8-10 cm bore) diesels.

Indirect-Injection Systems

Inlet generated air swirl, despite amplification produced in piston cup, and has not provided sufficiently high fuel-air mixing rates for small high-speed diesels such as those used in automobiles. Indirect- injection or dual chamber injection have been used instead, where the vigorous charge motion required during the fuel intake is generated during the compression stroke. Two broad classes of IDI systems are (1) swirl chamber systems and (2) pre-chamber systems. During compression, air is forced from the main chamber above the piston into the auxiliary chamber through the nozzle or orifice (or set of orifices). Thus, towards the end of compression, a vigorous flow into the auxiliary chamber is set up. In swirl chamber systems the connecting passage and chamber are shaped so that this flow within the auxiliary chamber rotates rapidly.

Fuel is usually injected into the auxiliary chamber at lower injection-system pressure than is typical of DI systems through a spindle as a single spray. Combustion starts in the auxiliary chamber; the pressure rise associated with the combustion forces fluid back to the main chamber where the jet issuing from the nozzle entrains and mixes with the main chamber air. A glow plug is normally used as a cold-starting aid, located mostly at the right-side of the pre-chamber. The plug is heated prior to starting the engine to ensure ignition of fuel early in the engine cranking process.

Photographic studies of engine combustion

High-speed photography at several thousand frames per second has been used extensively to study diesel engine combustion. Some of these studies have been carried out in combustion chambers are very close to those used in practice, under normal engine operating conditions. Sequences of individual frames from movies provide valuable information on the nature of the combustion process in different types of diesel engine. Figure shows four different chamber geometries that have been studied photographically. These are: a quiescent chamber typical of diesel engines in the 3 to 20 dm^3/cylinder displacement used or industrial, marine, and rail traction applications; a smaller high-speed DI engine with swirl and four fuel jets centrally injected; an M.A.N. M DI system; and (d) a Ricardo Comet V swirl chamber IDI system.

The combustion sequences were recorded on the color film and show following features: Fuel spray(s). The fuel droplets reflect light from spot lamps and define the extent of liquid fuel spray prior to complete vaporization

Premixed Flame: These regions are of too low a luminosity to be recorded with exposure level used. The addition of copper additive dope of the fuel gives these normally blue flames a green color bright enough to render them visible.

Diffusion Flame: The burning high-temperature carbon particle in this flame provides more than adequate luminosity and appears as yellow- white. As the flame cools, radiation from the particles changes color through orange to red.

Over-Rich Mixture: The appearance of a brown region, usually surrounded by a white diffusion flame, indicates an excessively rich mixture region where substantial soot particle production has occurred. Where this fuel-rich soot–laden cloud contacts unburned air there is a hot white diffusion flame.

Figure shows a sequence of photographs from one combustion unit event of single spray, burning under conditions typical of a large quiescent DI engine. The fuel spray is shown penetrating into the chamber. Ignition occurs at -8 degree in the fuel-air mixture left behind on the edge of the spray not from the injector. The flame then spreads rapidly (-7 degree) along the outside of the spray to the spray tip. Here some of the fuel, which has had a long residence time in the chamber, burns with a blue-green low-luminosity flame. The flame engulfing the remainder of the spray is brilliant white-yellow from the burning of the soot particles which have been formed the fuel-rich spray core. At this stage (1˚), about 60% of the fuel has been injected. The remainder is injected into this enflamed region, producing a very fuel-rich zone apparent as the dark brown cloud (11˚).This soot cloud moves to the outer region of the chamber (11˚ to 20˚); white-yellow flame activity continues near the injector, probably due to combustion of ligaments of fuel which issued from the injector nozzle as the injector needle was seating. Combustion continues well into the expansion stoke (31˚c).

This sequence shows that fuel distribution is always highly non-uniform during the combustion process in this type of DI engine. Also the air which is between the individual fuels sprays of the open-chamber diesel mixes with each burning spray relatively slowly, contributing to the poor air utilizations with this type of combustion chamber.

Figure shows a combustion sequence from the DI engine with swirl. The inner circle corresponds to the deep bowl in the piston crown, the outer circle to the cylinder liner. the fuel sprays first appear at-13˚.at-7˚ they have reached the wall of the bowl; the tips of the sprays have been deflected slightly by the anticlockwise swirl. The frame at-3˚ shows the first ignition. Bright luminous flame zone are visible, one on each spray. Out by the bowl walls, where fuel vapor has been blown around by the swirl, larger greenish burning regions indicating the presence of premixed flame can be seen. The fuel downstream of each spray is next to ignite, burning yellow-white due to the soot formed by the richer mixture. Flame propagation back to the injector follows extremely rapidly and at TC the bowl is filled with flame. At 5˚ ATC the flame spreads out over the piston crown toward the cylinder wall due to combustion produced gas expansion and the reverse squish flow. The flame then spreads rapidly (-7 degree) along the outside of the spray to the spray tip. Here some of the fuel, which has had a long residence time in the chamber, burns with a blue-green low-luminosity flame. The flame engulfing the remainder of the spray is brilliant white-yellow from the burning of the soot particles which have been formed the fuel-rich spray core.

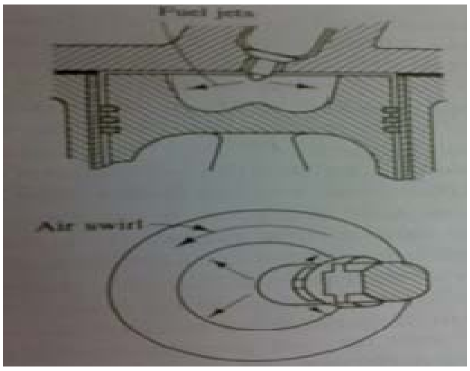

Fig 3: Hemisphere bowl design combustion

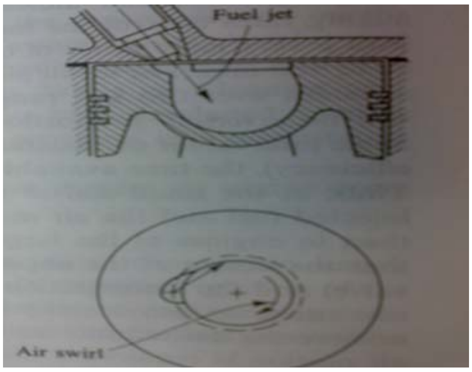

Fig 4: Hemisphere bowl design combustion

Design Calculations

Diesel Cycle Calculations

Inlet temp T_1 $=20^o = 273+20=293$ K

Pressure P_1 =1atm = 1.013bar

Compression ratio =18.5

Cut of ratio (P) =2.4

$$T_2 = T_1 (r)^{y-1}$$
$$= 293 * (18.5)^{1.3-1}$$
$$T_2 = 703.1k$$

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}}$$

$$P_2 = \left(\frac{703.1}{293}\right)^{\frac{1.3}{1.3-1}}$$

$$P_2 = 44.97bar = P_3$$

$$\rho = \frac{V_3}{V_4} = 2.4$$

$$T_3 = T_1 (r)^{y-1}$$
$$= 293 * (18.5)^{1.3-1} * 2.4$$
$$T_3 = 1687.45k$$
$$T_4 = T_1 P^{Y}$$
$$= 293 * (2.4)^{1.3}$$
$$T_4 = 914.4k$$

$$\frac{P_4}{P_3} = \left(\frac{T_4}{T_3}\right)^{\frac{\gamma}{\gamma-1}}$$

$$\frac{P_4}{44.97} = \left(\frac{914.4}{1687.45}\right)^{\frac{1.8}{1.8-1}}$$

P_4 = 3.16bar

Table 2: Pressure and Temperature Values at Various Stroke

Temperature	Pressure
T_1 = 293K	P_1 = 1.013bar
T_2 = 703.1K	P_2 = 44.97bar
T_3 = 1687.45K	P_3 = 44.97bar
T_4 = 914.4K	P_4 = 3.16bar

Swirl - Inlet Velocity Calculations

Coefficient of Swirl and Air Velocity

$$C_s = \frac{8T}{\dot{m} v_0 B}$$

$$\frac{m_f}{m_a} = 0.069 \qquad m_f = 0.176$$

$$m_a = \frac{0.176}{0.069} = 2.55\text{kg/s}$$

Toroidal Piston Bowl (Directed Type)

$$C_s = 0.034$$

$$C_{s1} = \frac{8T}{\dot{m} v_{01} B}$$

$$= \frac{8 * 122 * 10^3}{2.55 * v_{01} * 91.4}$$

$$= 0.034$$

$$v_{01} = 123.16 \text{ m/s}$$

Hemispherical Piston Bowl (**Shallow Ramp Type)**

$$C_{s2} = 0.136$$

$$C_{s2} = \frac{8T}{\dot{m} v_{02} B}$$

$$= \frac{8 * 122 * 10^3}{2.55 * v_{02} * 91.4}$$

$$= 0.017$$

$$v_{02} = 246.32 \text{ m/s}$$

Swirl Ratio Calculations

$$R_s = \pi \eta_v B L \left[\int_{\theta_1}^{\theta_2} (A_v C_D) C_{s1} d\theta \Big/ \left[\int_{\theta_1}^{\theta_2} (A_v C_D) d\theta\right]^2 \right]$$

$$A_v = \pi L_v D_v$$

$$= \pi * 25.4 * 0.15$$

$$A_v = 11.969\text{mm}^2$$

$$A_v C_D = 11.969 \text{ x } 0.85 = 10.174\text{mm}^2$$

Toroidal Piston Bowl (Directed Type)

$$(A_v C_D) C_{s1} = 0.3459$$

$$R_{s1} = \pi * 0.82 * 91.4 * 127 \left[\int_{-0.226}^{3.63} 0.3459 d\theta \Big/ \left[\int_{-0.226}^{3.36} 10.174 d\theta\right]^2 \right]$$

$$= 25.91$$

Air swirl analysis of toroidal with direct type inlet port design for piston at bdc

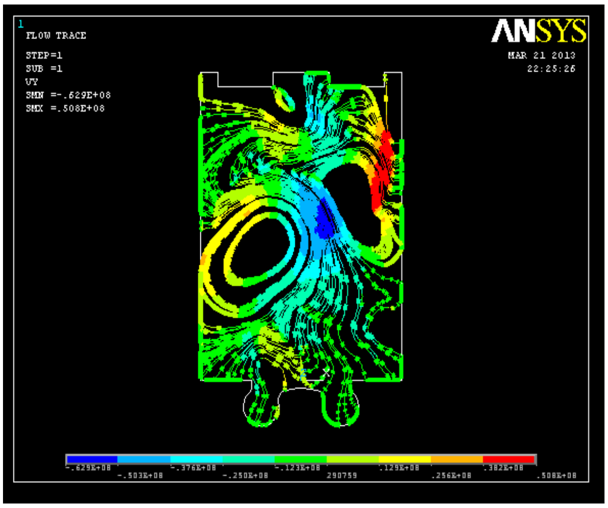

Fig 5: Air Swirl Pattern of Toroidal with Direct type Inlet valve Design – Particle Flow.

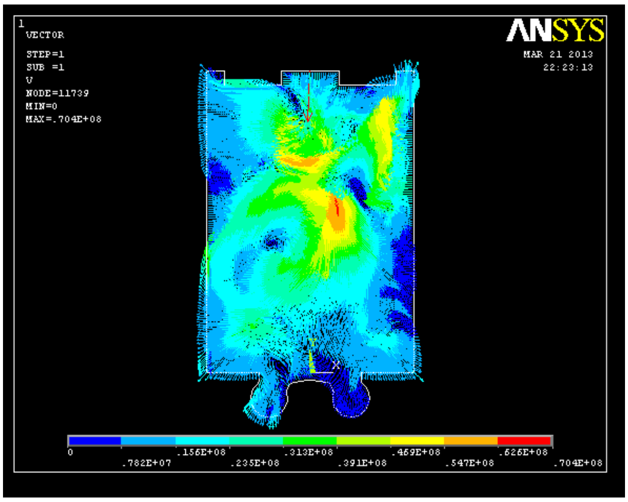

Fig 6: Air Swirl Pattern of Toroidal with Direct type Inlet valve Design-Vector Plot

Air swirl analysis of hemispherical with direct type inlet port design for piston at bdc

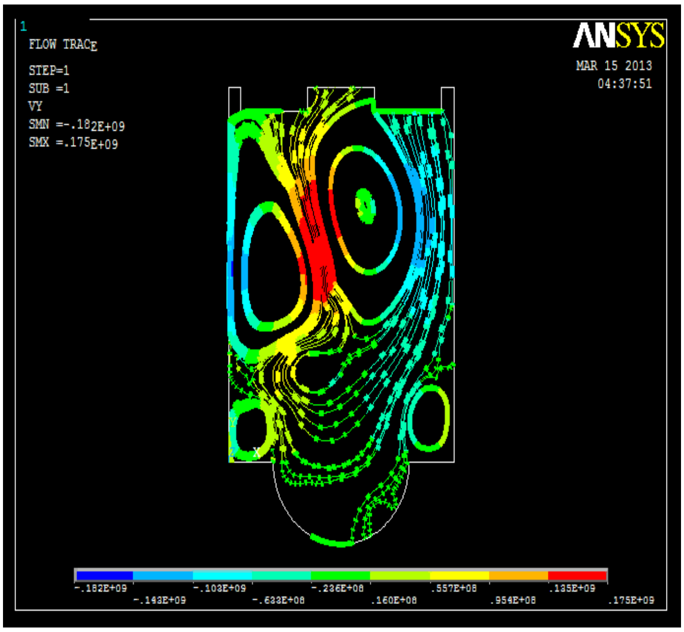

Fig 7: Air Swirl Pattern of Hemispherical with Direct type Inlet valve Design – Particle Flow

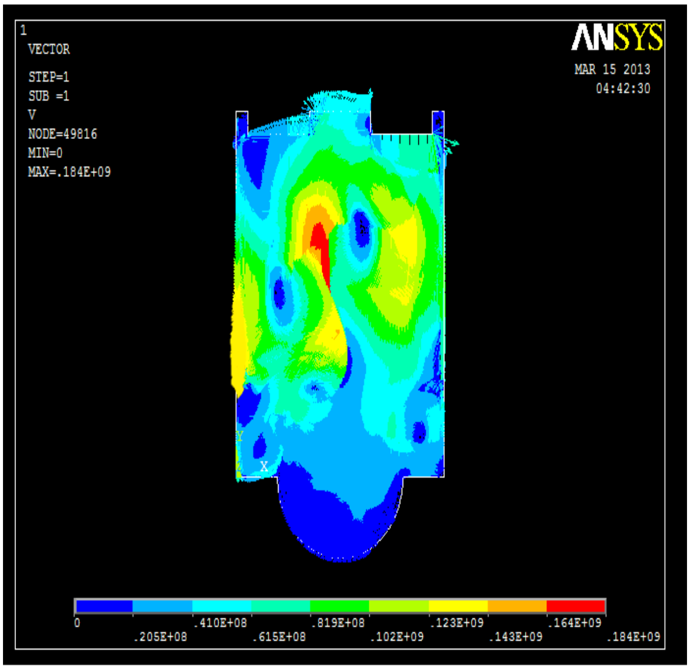

Fig 8: Air Swirl Pattern of hemispherical with Direct type Inlet valve Design-Vector Plot

Air swirl analysis of hemispherical with shallow ramp type inlet port design for piston at bdc

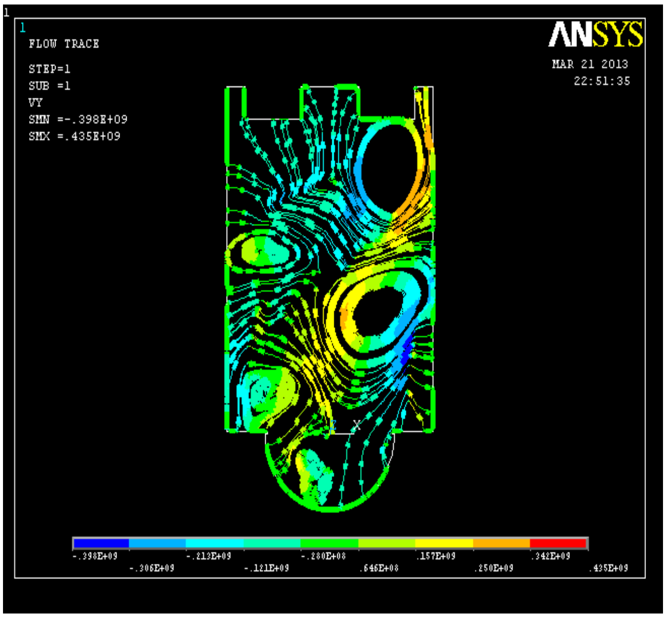

Fig 9: Air Swirl Pattern of Hemispherical with Shallow ramp type Inlet valve Design-Particle Flow

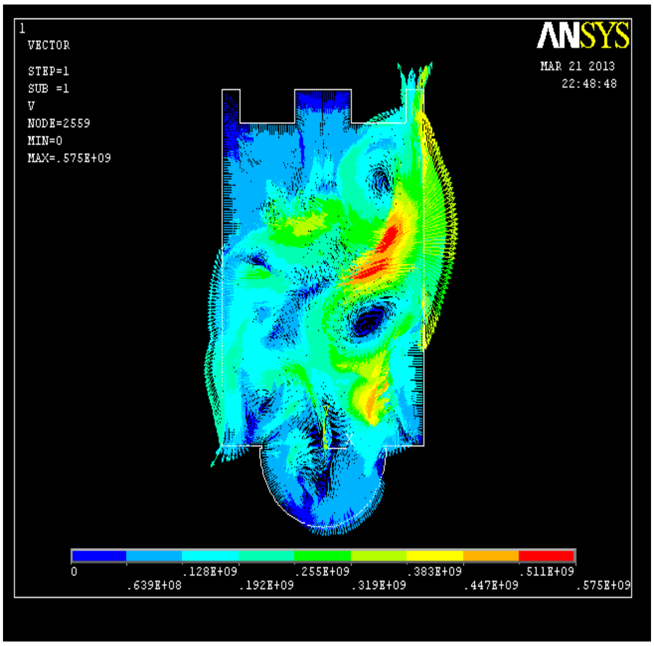

Fig 10: Air Swirl Pattern of Hemispherical with Shallow ramp type Inlet valve Design-Vector Plot

Conclusion

A comparative analysis of air and fuel flow inside the combustion chamber of a particular model DI diesel engine (S325) at two specific piston positions namely Top Dead Centre (TDC) and Bottom Dead Centre (BDC) was made and results viz., the fuel flow pattern through the compressed air at TDC i.e., the extent of Spray Penetration in the piston bowl volume and the air flow pattern through the old and newly designed inlet valves when the piston is at BDC, effected through Ansys-Flotran/ Cfd.

Hence the advantageous features of the new design changes in combustion chamber are discovered and tabulated in the calculation section, through which a new and more efficient engine model which would satisfy the required need-elimination of black smoke is kept for the industry's foresight and review.

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