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Disquisition on Diesel Engine Emissions and Piston Bowl Parameters

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Abstract: To meet stringent emission norms researchers and engine developers are working tremendously all over the on diesel engine combustion and emissions. This review covers emission formation mechanisms in diesel engines and recent developments in regulations to limit diesel engine emissions especially piston bowl parameters like toroidal radius, pip region, lip and spray impingement area.

Keywords: Diesel Engine Combustion, Emission Mechanism, Piston Bowl Geometry.

I. INTRODUCTION

Diesel combustion is inherently complex phenomenon. Fuel injection into hot, moving air creates a highly non uniform, three dimensional fuel distributions, which are modified by combustion chamber boundaries. Injected fuel must mix with the intake air and combust within a few milliseconds. The engine should produce the required torque and power, at low cost to compete in the commercial market. Along with low cost its exhaust must be cleaner and within desired limits of emission norms. Objective of the review is to study detail emission mechanisms and in-cylinder combustion bowl parameters. It is evident from the literature review that piston cavity parameters such as toroidal radius, impingement region and pip inclination are playing important role in emission formation of CI engine. Cylinder charge motion is affected by different piston bowl profiles which in turns affect diesel engine emissions. The strategy developed by different researchers for low emission is discussed in this review paper.

II. DIESEL ENGINE COMBUSTION

Diesel engine is facilitated by the development of modern injection systems that are more flexible and allow significantly higher injection pressures and thus allow better spray atomization and mixture formation for higher power and more efficient combustion. The diesel engines have low ISFC (Indicated Specific Fuel Consumption) and thus low CO2 emissions due to the lack of throttling losses and also due to the higher compression ratio. With the ability to operate at low equivalence ratios, diesel engine produces low CO and HC emissions. Unfortunately, the diesel engine with high combustion temperatures suffers from high NOx and PM emissions. Reduction of these emissions without a penalty in fuel economy is the main focus of all the researchers in Diesel combustion. These emissions are strongly dependent on the distribution of fuel and its mixing with air. Spray formation and mixing are important parameters which are needed to understand the diesel combustion fundamentals for reduction of emissions. The sequence of exothermic chemical reactions between a fuel and an oxidant accompanied by the production of heat and conversion of chemical species is combustion or burning. In a combustion reaction, a compound reacts with an oxidizing element, such as oxygen or fluorine, and the products are compounds of each element in the fuel with the oxidizing element. The process releases heat energy. Example shown below

Fuel + Oxidizer → Combustion products + Heat energy
4C12H23+71O2-->48CO2+46H2O (1.1)

If the stoichiometric combustion takes place using air as the oxygen source,

Fuel + oxygen + nitrogen → water + carbon dioxide + nitrogen$C_{12}H_{23}+17.75(O_2+3.76N_2) \longrightarrow 12CO_2+11.5H_2O+66.74N_2$ (1.2)

Incomplete combustion occurs when there is not enough oxygen to allow the fuel to react completely to produce carbon dioxide and water.

III. HEAT RELEASE

The rate of heat release is defined as the rate at which heat is releases during the combustion process, predominately by releasing chemical energy of the burning fuel. Heat release can be calculated from engine data by predicting (using the I law of thermodynamics) how much heat would have to be added to the working fluid to produce a required cylinder pressure, taking into considerable changes in the combustion chamber volume. When heat release rate is calculated from measured cylinder pressure, it is termed as the “net heat release rate”. This is because it includes heat transfer to the cylinder walls, and other losses (generally 10-25% of the available chemical fuel energy is lost to the cylinder walls). The net heat release rate is given by equation below.

Net heat release rate = gross heat release rate - heat transfer to the walls = rate at which work is done on piston + change of sensible internal energy of the working fluid.

Piston bowl design can affect the rate of heat release, and the different phases of heat release in many ways. It affects the shape and magnitude of the heat release rate profile, by affecting bulk airflow and turbulence, thus affecting air/fuel mixing rates. Effective control and manipulation of the heat release rate is important to limit

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peak cylinder pressure, combustion noise, and emissions. Figure 1 show the different phases of heat release, starting from mid compression stroke.

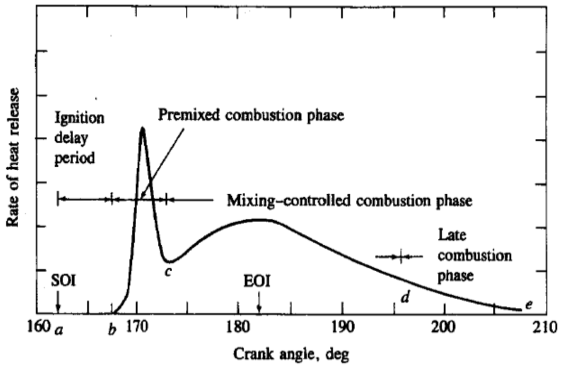

Fig1: Rate of Heat Release

(a) Start of injection (SOI)
(a-b) Delay period
(b-c) Pre-mixed combustion
(c-d) Diffusion burning
(d-e) Late Heat Release

Start of Injection (SOI)

At a pre-defined crank-angle (CA), at the end of the compression stroke (around TDC), high pressure fuel is injected into the combustion chamber. The duration of fuel injection is proportional to the engine load required. The timing of the SOI significantly affects the rate of cylinder pressure rise, by changing the cylinder volume at the time of pre-mixed combustion, and affecting the ignition delay period itself, the shortest delay period occurs when the delay period includes TDC (because the temperature of compressed air is close to its maximum). The longest delay occurs with late injection, when both cylinder temperature and pressure are lowest. The rate of pressure rise is highest with early injection, when the delay period is long; allowing large amounts of fuel to be prepared for pre-mixed combustion, and cylinder volume at ignition is at a minimum. Late injection also creates high peak pressures, as the ignition delay is long, although the effect is reduced by increased cylinder volume.

Ignition delay period

Fuel injected into the cylinder does not combust instantaneously, even though the temperature and pressure of air into which fuel is injected will be above that required for self-ignition. The time period between the start of fuel injection (usually defined as the start of injector needle lift, or a small fraction of the total needle lift), and combustion (usually defined as when the rate of heat release becomes positive or the first flame) is called the delay period. At the end of the compression stroke, and during the delay period, a small amount of heat is transferred from the hot compressed air to the cylinder walls, cylinder head, piston and other combustion chamber surfaces. Further heat transfer from the working fluid (compressed air) occurs, to heat and evaporate the injected fuel. These two effects result in the initial heat release rate being negative, as heat is given up by the working fluid.
Turbo charging of Dl diesel engines produces high air temperatures and increased pressure at the end of compression.

Rapid combustion phase

Fuel prepared to within combustible limits during the delay period, burns in a premixed combustion mode. Early ignition sites within the cylinder compress the remaining mixture already prepared to within combustible limits, raising its temperature and promoting more rapid combustion. Fuel prepared for combustion during the ignition delay period causes a very high initial heat release rate spike. This is caused by significant quantities of fuel burning in a pre-mixed condition, at high temperature and from multiple ignition sites. The magnitude of heat release depends on the amount of fuel prepared for pre-mixed combustion before ignition, and thus on the delay period, fuel injection characteristics and airflow. Piston bowl shape affects the airflow and turbulence level, and, thereby, changes the amount of fuel prepared for pre-mixed combustion.
The heat release spike caused by pre-mixed combustion is not desirable. It creates high peak cylinder pressures, requiring increased cylinder strength, and contributes significantly to diesel combustion noise and knock. Emissions of NOx also increase, because of the high peak local temperatures. Modern DI diesel engines aim to reduce the magnitude of premixed combustion heat release, by reducing the amount fuel prepared during the delay period, and by reducing the duration of the delay period.

Late Combustion phase

The last distinct phase of heat release consists of a small but steady ‘tail’ which continues well into the expansion stroke. It accounts for around 20% of the injected fuel heat release [Heywood, 1983]. Combustion late in the expansion stroke will be much cooler and less vigorous, because the air motion, availability of oxygen, and mixing rates are all reduced. Late combustion significantly affects engine out smoke emissions, and to a lesser extent emission of unburned HC.

IV. FACTORS AFFECTING COMBUSTION SWIRL

Swirl is the ordered rotation of air about the cylinder axis shown in figure 2a, and is the most important air motion in DI diesel engines. It is created by introducing the intake air into the cylinder with an initial swirling motion, or angular momentum. For comparison between different engine designs, a swirl ratio is defined as,

Swirl ratio = swirl speed (rpm) / engine speed (rpm)

Benefits of in-cylinder swirl
Swirl is used to promote more rapid mixing between the inducted charge air and fuel injected.

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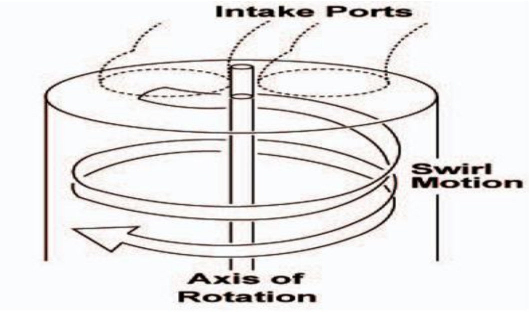

Fig.2(a): Swirl

Squish

Squish is the radially inward motion of compressed air, expelled from the gap between the piston top, and cylinder head face, as the piston approaches TDC during the compression stroke. The relative squish area is indicated by the shaded area in the top view of the piston face and Bowl-in-piston shown in figure 2b.Squish occurs in DI diesel engines because of the need to move air into the compact blow-in-piston, thereby increasing swirl. The amount of squish is given by following equation

% Squish area = area of piston top which nearly meets cylinder head/cylinder head area

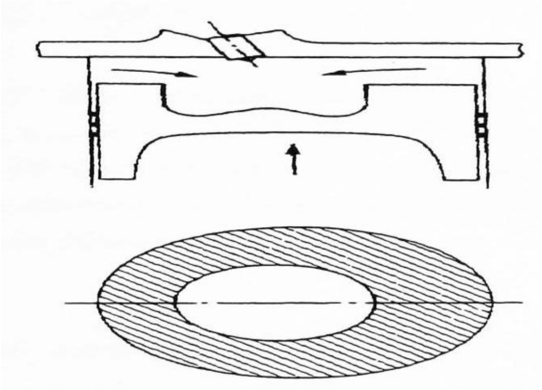

Fig.2 (b): Swirl

Turbulence

Turbulent flow is random three-dimensional fluid flow, comprising of eddies and vortices of varying sizes and lifetimes. The character of turbulent flow in an engine cylinder is a complicated combination of turbulent shear layers, re-circulating flow, and boundary layers. Turbulent airflow plays an important role in the high speed DI diesel engine, by increasing the rate of momentum, heat, and mass transfer during fuel/air mixing and combustion. Most of the initial turbulence in combustion chamber is generated by the jet-like intake flow.

Fuel Impingement

Fuel impingement on the piston bowl sides occurs shortly after fuel injection. Fuel impingement is often blamed for poor emissions of smoke and HC in many bowl design. However, combustion chambers exploit fuel impingement to control the rate of combustion.

Fuel Spray Development

Modem DI diesel engines having four valves per cylinder; inject fuel from a multi-hole fuel injector, positioned at the top Centre of the combustion chamber. Diesel sprays consist of two phase flow of a gas exterior and a liquid core. The liquid core travels many times faster than the gas exterior, which entrains air and loses momentum. As the jet leaves the nozzle, it becomes turbulent and mixes with surrounding air. The initial fuel that is injected loses momentum rapidly, as air is entrained and accelerated. Jet penetration continues until sufficient time is available for fuel/air mixing, atomization, and evaporation or the jet encounters a boundary. In Dl diesel engines, fuel will impinge on the piston bowl walls before complete evaporation. The schematic diagram in figure 3 below shows the stages in fuel jet growth and penetration.

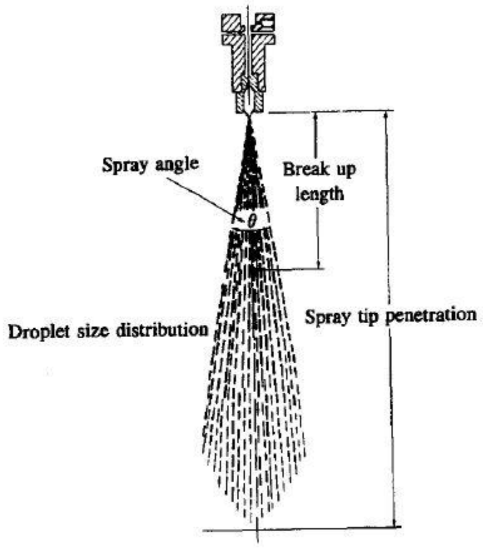

Fig 3: Schematic diagram of fuel spray growth and structure

The maximum spray tip penetration length will not be achieved in modern high speed DI diesel engines, because of the interaction of the spray with the piston bowl walls. Breakup length is the length at which the liquid spray core is considered to have broken up into droplets of various sizes.

V. DIESEL ENGINE EMISSIONS

Unlike spark-ignited engines where the combustible mixture is predominantly homogeneous, diesel combustion is heterogeneous in nature. Emissions formed as a result of burning this uneven air/fuel mixture. Mixing of unburned hydrocarbons with oxidizing gases, high combustion chamber temperature, and adequate residence time for the oxidation process permit more complete combustion. The exhaust gases have wide range of different contents which are harmful and affect the environment as well as human beings. The most significant of these are oxides of nitrogen (NOx), particulate matter (of which smoke is a constituent), un-burnt hydrocarbons (UHC or HC) and the emission of large quantities of carbon dioxide (CO2) also cause concern. The main effects attributed to the emission of CO2, NOx and UHC are the ‘greenhouse effect’. Oxides of sulphur and particulate matter are considered responsible for local urban pollution, and can be carcinogenic. Reduction of harmful emissions from the exhaust of diesel engines has become necessary because of the introduction of legislation, following concern of the effect of urban pollution on human health, and global pollution on global warming and the environment. Limits are regularly tightened in line with the anticipated growth of the usage of vehicles, and the need for reducing the overall output of harmful emissions.

VI. FORMATION OF DIESEL ENGINE EMISSIONS

Harmful emissions in the exhaust of diesel engines occur, because the rate of combustion in the engine cylinder is too rapid for equilibrium values to be established. Reduction of harmful emissions from diesel exhaust is difficult because of the inherent dependence on the in cylinder combustion process. Further complications arise because of the conflicts of reducing one emission at the detriment of another, or reducing engine efficiency.CI engines releases all harmful emissions into atmosphere through its exhaust gases.

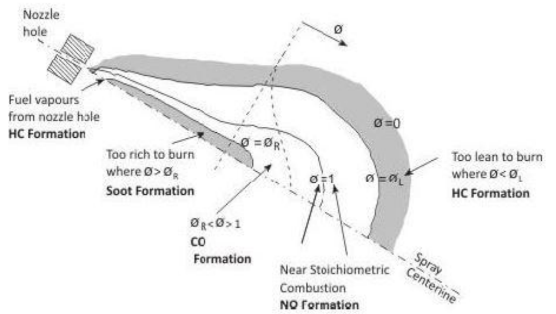

Fig 4: Schematic of a Diesel Injection Spray

A fully developed diesel spray may be considered to consist of three distinct regions based on the variations in fuel-air equivalence ratio, Ø across the cross section of the spray as seen radially outwards from the centreline of spray.

1.) A fuel rich core where fuel-air equivalence ratio is richer than the rich flammability limits i.e., $Ø > Ø_R$

2.) Flammable region in which Ø lies within the rich and lean flammability limits, i.e., $Ø_R > Ø > Ø_L$

3.) A lean flame-out region (LFOR) where Ø is lower than lean flammability limits and extends up to the spray boundary i.e., $Ø_L > Ø > 0$

Formation of NOx

NOx is formed when nitrogen and oxygen mix at a high temperature, and for sufficient time for reaction. Nitrogen is present in the intake air, and oxygen remains during combustion. NO and NO2 are collectively called NOx. However, NO is the most abundant chemical species of the two, accounting for typically 70-90% of the total concentration of NOx. Work at General Motors in the 1980s suggested that the diffusion burning region may be the primary source of NOx. Work at Sandia later showed that the premixed burning phase of the diesel combustion process occurs under very fuel rich conditions fuel-air equivalence ratio in the range of 4 under which very little NOx actually forms due to low oxygen concentrations and low temperatures and that the diffusion burning region is indeed the primary source of NOx.

There are three paths of NO formation as follows.

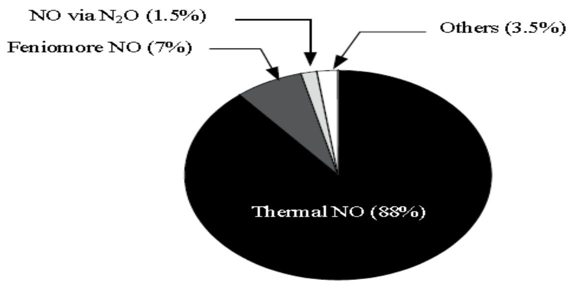

Figure 5: NO formation Paths

1.) Thermal NO (88) described by Zeldovich mechanism

2.) Fennimore NO (7%)

3.) NO via N_2O (1.5%)

Effect of Temperature on NOx Formation

The main factor affecting the amount of NOx produced is the local temperature at the point of NOx formation. Oxygen for NOx formation is supplied by the intake air. The majority of nitrogen which forms NOx also comes from the intake air, although some fuel compounds containing nitrogen can participate in NOx formation.

Mechanisms of NOx Formation (Thermal NO)

The thermal mechanism, also known as the extended Zeldovich mechanism is responsible for the majority of NOx emissions from conventional diesel engines with peak combustion temperatures in excess of 2000 K. The three chemical reactions that are important in this mechanism are:

$$O + N_2 \leftrightarrow NO + N \qquad (1.3)$$

$$N + O_2 \leftrightarrow NO + O \qquad (1.4)$$

$$N + OH \leftrightarrow NO + H \qquad (1.5)$$

The overall reaction rate for this mechanism is relatively slow and is very temperature sensitive. As a Consequence, thermal NO only appears in significant quantities well after the start of heat release. The forward rate of reaction gives this mechanism its strong temperature dependence.

N_2O Pathway

Another NO mechanism important in combustion is the N_2O pathway. At sufficiently high pressures, O atom reaction with N_2 can result in the formation of nitrous oxide through a three body reaction:

$$O + N_2 + M \leftrightarrow N_2O + M \qquad (1.6)$$

Where "M", is a third body molecule of any compound, needs to remove energy in order to complete the reaction. The combination of the Zeldovich and N_2O mechanisms is important for explaining NOx emissions from diesel engines.

Formation of Soot

Soot forms from carbon in the fuel, around the very rich regions of the injected fuel spray, when heating of the fuel by combustion gases occurs. Soot formation is most prominent at the spray tip, in free forming sprays. Figure 6 illustrates regions of soot formation in the free spray. The particulates formed are predominately composed of carbon, but contain a significant proportion of heavy HC molecules, which have condensed as soot particles at the exhaust sampling temperature. Hydrocarbons typically account for 15-45% of the total particulate mass.

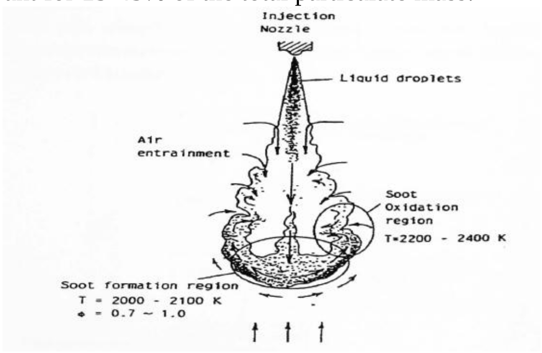

Figure 6: Schematic diagram of soot formation.

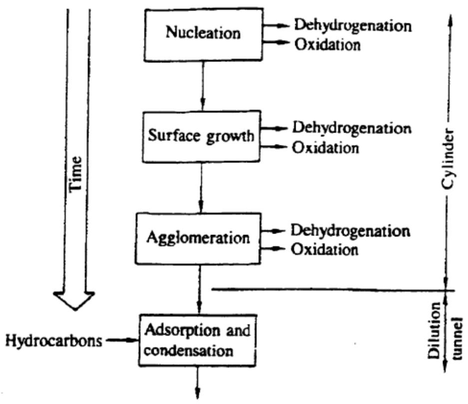

Figure 7: Schematic diagram of particulate formation, from initiation to exhaust-out emission in diesel engines.

Soot Particles Growth

The formation and growth of particulates from within the engine cylinder, to exhausting into the atmosphere, consists of several main stages. These are displayed in figure 7. Nucleation includes fast and slow routes to forming nuclei for particulate growth. Fast routes involve aromatic fuel compounds (based on benzene), which condense to form a graphite like structure. Slow routes to nucleation include pyrolysis (reactions not involving oxygen, but brought about by high temperature) of aromatics, and open chain molecules which then polymerize to form large molecules and nuclei. Particle/surface growth is the stage during which most of the solid phase material is produced. Nuclei grow by deposition of pyrolysis products onto their surface, to form small spherules of 0-50 nm diameter. Agglomeration occur when particles collide and stick together to form larger, more irregular particles, of typically 50 to 220 nm diameter. No mass is added during this stage, but the number density reduces as particles combine. Adsorption and condensation occurs as the exhaust cools. Hydrocarbon fractions from fuel and oil condense on to the soot particles, increasing particulate mass typically by 15-30%.

VII. PISTON BOWL DESIGNATION

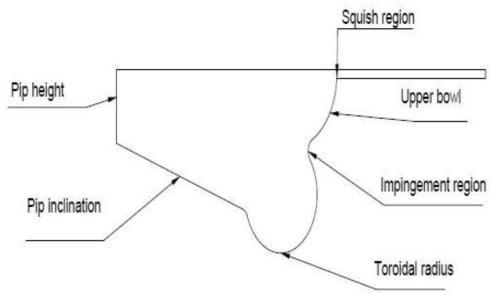

Fig 8: Piston cavity geometrical parameters

Types of Piston Bowls

These are some general types of the piston bowl shapes. The generally used type of the piston top is the center bowl in flat piston. Since the total kinetic energy is 27% more than that of the flat piston. And this total kinetic energy is highest in central bowl in all the shapes.

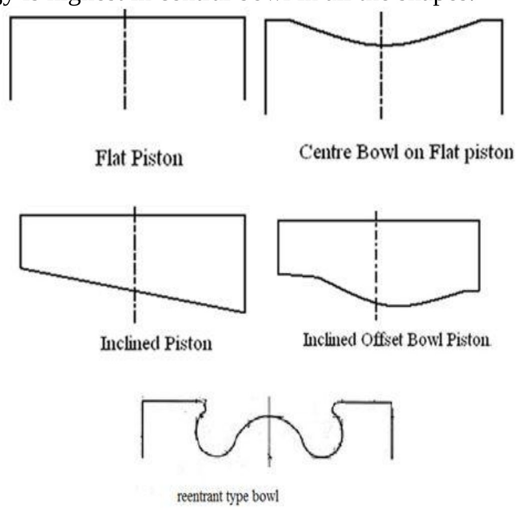

Fig 9: Piston Bowl Types

Piston Bowl Parameters

A. Throat diameter

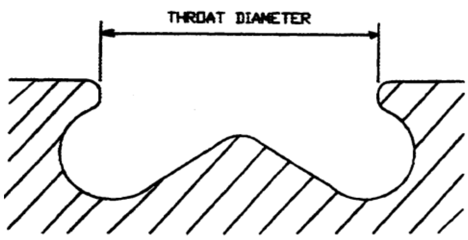

Fig 10: Piston cavity throat diameter

The diameter of the throat is defined as the minimum diameter between the piston bowl edges, near the piston top face. The ratio of the throat diameter to the maximum bowl diameter defined the amount of reentrancy of piston bowl design. High velocity airflow into the bowl, and combustion gas out of the bowl, creates large temperature gradients and high heat transfer rates to the piston bowl top surfaces. The piston bowl lip is often the hottest part of the piston bowl internal surface.

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$$\text{Amount of reentrancy} = \frac{\text{throat diameter}}{\text{maximum bowl diameter}}$$

B. Maximum diameter

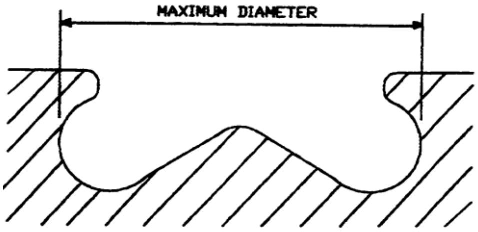

Fig 11: Piston cavity Maximum diameter

The maximum bowl diameter is defined as the largest diameter parallel with the piston face at any position through a section of the piston bowl. The ratio of the maximum bowl diameter to the bowl depth defined the piston bowl aspect ratio. The piston bowl total volume, and thus compression ratio, is largely controlled by the maximum bowl diameter. This is one of the first parameters to be set when designing a new piston bowl shape.

$$\text{Piston bowl aspect ratio} = \frac{\text{maximum bowl diameter}}{\text{bowl depth}}$$

C. Central pip

The central pip is used to occupy a volume in the Centre of the piston bowl, where the air velocity is low. Low air velocity in the Centre of the swirling flow-field results in poor air/fuel mixing rates. The central pip allowed this volume to be redistributed further from the Centre of rotation, resulting in a higher mean airflow velocity and better air/fuel mixing.

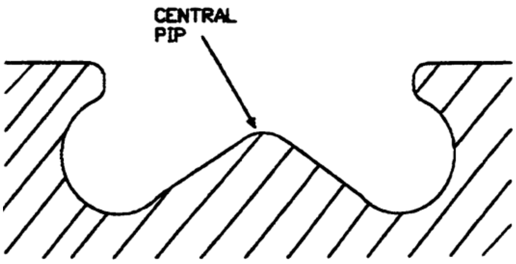

Fig 12: Piston cavity central pip

D. Depth

Fig 13: Piston cavity Depth

Piston bowl depth is defined as the maximum depth from the face of the piston, to the bottom of the main toroidal radius (in conventional bowl designs). Maximum the bowl depth is closely linked with the need to provide a certain depth of spray impingement area, and is often balanced with the volume occupied by the piston pip.

E. Main toroidal radius

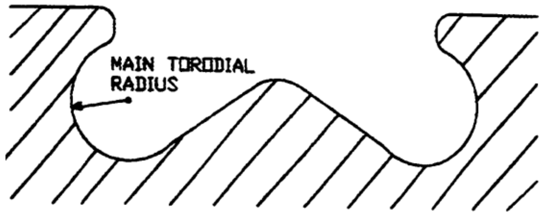

Fig 14: Piston cavity toroidal radius

The majority of combustion occurs in the main toroidal radius volume.

F. Impingement Area

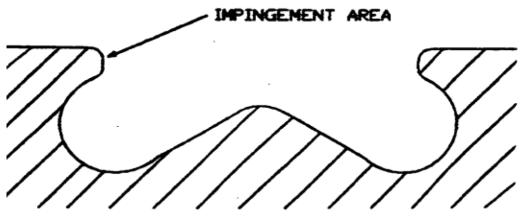

Fig 15: Piston cavity Impingement area

The area at the side of the piston bowl where high velocity fuel impinges during fuel injection is known as the impingement area. The impingement area is most significant in the early stages of fuel injection and combustion. It can affect the ignition delay and initial rate of pressure rise, by controlling the amount and composition of fuel air mixture prepared for initial combustion.

VIII. EFFECT ON EMISSIONS DUE TO BOWL PARAMETERS

Many researchers investigated the effect of each piston cavity parameter on combustion, emission and charge motion. Some of them are discussed here. The review of the literature focuses on the study of the existing correlations between parameters and diesel engine emissions.

a) Effect of Toroidal radius

The influence of toroidal radius and lip shape on diesel engines performance and emissions as follows. Author observed that larger toroidal radius in the cavity helps to increase engine performance through enhanced fuel air mixing. Higher peak cylinder temperature and lack of fuel rich regions are also responsible for less soot emission, this cause to reduce soot emission.

b) Effect of Lip shape.

In combustion analysis to study effect of lip shape found that lip deliberately splits the fuel in upper and lower bowl. First stage of combustion is taking place in upper bowl and then in lower bowl, this helps to move the gases with lower oxygen concentration further up into the bowl. But in case of conventional single bowl cavity combustion starts in single bowl and whatever fuel is injected in later are forced to mix with the hot, lower oxygen concentrated gases from the first injection, causing additional soot to form. From this conventional bowl limits the amount of oxygen exposed to the injected fuel. However, in the stepped-bowl the fuel is injected at an angle that allows oxygen to be located on either side of the injected fuel. This allows for better air entrainment.

c) Effect of Pip inclination

The pip shape activates the swirl and vortex which is the flow in the combustion space, and in the exemplary embodiment, the mixing of the fuel and the air flowing into the combustion space is improved and the mixing ratio can be greatly increased.

d) Effect of Impingement Position

The effect of impingement point and lip shape on combustion as follows. In the cavity with lip the wall jet is divided into two regions: one region above one below the lip. Below the lip the spray is semicircular rather than circular. With the round lip the main spray flows smoothly along the concave surface of the lip and the loss of kinetic energy is small. As shown in figure 16. This low energy loss produces a larger spray volume at the bottom. The unburned HC emission improves at high loads in a reentrant cavity with round lip in comparison with a cavity without lip.

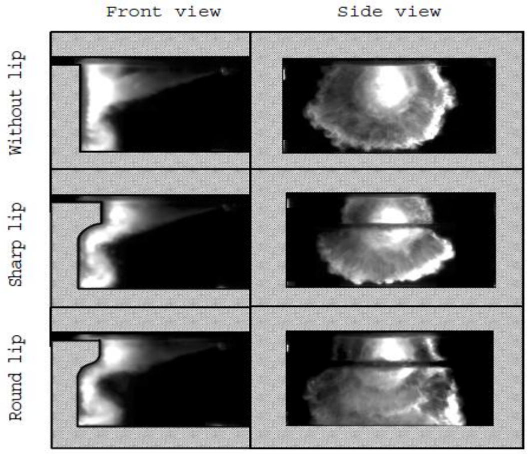

Fig 16: Lip shape and spray distribution

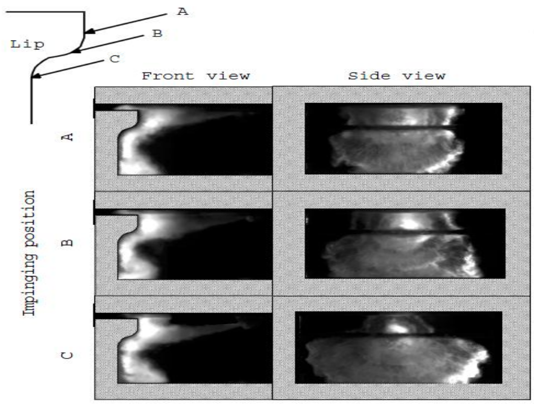

Fig 17: Impingement point and fuel distribution

The impinging position on the cavity wall has a significant effect on the fuel distribution and combustion. Photographs show that the amount of spray spreading above the lip decreases and that below the lip it increases as the position of impingement changes from "A" to "C". When impingement occurs at "C", where most of the spray impinges below the lip and the movement of the wall jet in the upward direction is restricted by the lip. With impingement on the lip corner the combustion pressure and temperature become a little higher than at other positions of impingement, and the combustion rate becomes a little faster. This assisted in the reduction in HC and smoke with a small penalty in NOx emission.

e) Effect of cavity depth

The shallow piston bowl geometry results in lower efficiency because fuel reaches the cylinder liner at all values of the spray angle α. Fuel in such zones reaches the cylinder liner. Shallow piston bowls provide advantage only in large engines with a high distance between the sprayer and the cylinder liner as well as in engines with the high boosting pressure because the high density of air reduces the spray penetration distance and as result of this, fuel does not reach the cylinder liner. Deep piston bowls have advantages if the engine has a low boosting pressure and fuel sprays have the long penetration length due to the small air density and the long injection duration.

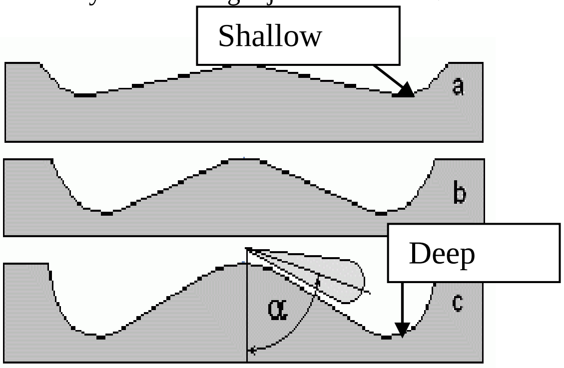

Fig 18: Bowl depth

The review suggests that deep bowl combustion chamber; emissions are improved by strong swirl and comparatively small number of nozzle opening. In the shallow dish Chamber, a relatively large number of nozzle openings give better emissions, however large strong swirl causes over-swirl phenomena, resulting in poorer emissions and fuel consumption.

IX. CONCLUSION

Conclusion made from above research is as follows.

1) Diesel engine emissions are mostly affected by in-cylinder combustion parameters.

2) Diesel piston bowl parameters such as spray impingement position i.e. lip area, toroidal radius in which main combustion occurs, pip height and pip inclination which changes flow pattern and swirl rate in combustion chamber are main factors affecting diesel engine combustion and emissions.

3) Hence by optimizing piston bowl parameters emissions can be reduced to greater extent which is more cost effective.

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