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Simulation of IC Engines with Special Focus on Spray Models of CI Engines

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Abstract : *Spark ignited and Compression ignited internal combustion engines are the two types under the classification of IC engines based on ignition mechanism. In spark ignited IC engines the pressurized air fuel mixture will be given the spark ignition in the chamber just after sufficient fraternization where as in compression ignited IC engines the pressurized air gets ignited of the fuel spray in the chamber. In spark ignited engines, a carburetor may be used where the fuel and air gets mixed and the resulting mixture will be supplied to the chamber, or electronic controlled mechanism may be used where based on the load and required speed the concentration of fuel in the mixture will be varied whereas it is usually fixed in the carburetor based systems. In compression ignited engines, the spraying mechanism is crucial for satisfactory combustion. The injector may be moved toward the chamber and in the reverse direction. The number of nozzles and the angle of injection of fuel may be varied. In this paper, the above described phenomena are modeled and results are compared.*

Keywords - *air-fuel mixture, nozzle, injector, bowl axis.*

I. Introduction

According to history engines that may be internal combustion or external combustion are said to be designed to avoid the man and animal based transport system. Nowadays, these engines can drive large sized transportation systems like naval, air-based as well as turbo-charged roadway vehicles even with miniaturized designs of engine. Most of the transportation vehicles use either the spark ignited or compression ignited versions of internal combustion engines. But each of the types suffers from some limitations. Subsequently many hybrid engines are designed and some are in design phase [1-3].

A number of variations of spark ignited and compression ignited IC engines are proposed in the literature. In [4], supercomputing applications in IC engines are introduced. Results of three example cases are presented. The cases include motoring flow in a pent-roof engine, charge distribution in a DISC engine, and coherent flamelet combustion model. In [5], the importance of homogeneous charge compression ignition combustion of an IC engine was stated elaborately. The combustion mode transition problem of a multi-cylinder IC engine was studied. The proposed control strategies are validated using Hardware-in-the-loop simulations.

Engine thermal management was considered in [6]. The potential of manipulating combustion wall temperature for improving engine efficiency was demonstrated. The results show that optimized combustion wall temperature produces significant fuel consumption improvements at low to medium engine speed at both low and high load. In [7], the development of a homogeneous charge compression ignition free-piston engine-compressor was presented. Modeling of CI engine for control was proposed in [8]. Advanced CI engine was proposed in [9]. A CI engine for straight plant oil fuelling was presented in [10].A five-stroke engine was proposed in [11]. A review of gasoline direct compression ignition engine was presented in [12]. A non-linear model of SI engine for controller design was proposed in [13]. Sectional combustion chamber base SI engine was proposed in [14].

The combustion in IC engine is crucial for generating mechanical work out of the chemical energy. The combustion in spark ignited engines is homogeneous because of two reasons. One is the characteristics of fuel, the second the time available for the fuel to mix with the air. In compression ignited engine the case is different. The combustion is non-homogeneous because of the same reasons. In this paper, the spark ignited and compression ignited engines are studied, compared and simulated by considering different variations of the two. The rest of the paper is organized as follows. The next section deals with the features, operation, simulation and modeling of SI engine. The section III, considers the operation and modeling of CI engine. The section IV compares the SI and CI engines with the simulation results generated in sections II and III. Section V concludes the paper.

II. Simulation of SI Engine

The early spark ignited internal combustion engines have a carburetor. The carburetor is responsible to mix the fuel with air to prepare the fuel-air mixture. The spark ignited engines usually take Gasoline as fuel. The gasoline is a transparent, petroleum-derived liquid obtained by fractional distillation of petroleum. The gasoline is a high-volatile fuel and has high self-ignition temperature. The carburetor based engines mix the gasoline with air in a specific predefined ratio, before the time. The mixture will be extracted by the piston downward movement initiated by the external motor. The mixture will be entered into the chamber and then the intake valve gets closed. By the upward movement of the piston the mixture of fuel and air gets pressurized. Because of high volatile-ness and sufficient time for the fuel to get mingled with air homogeneous combustion is possible in the spark ignited engines. The compression ratio is usually set in the range of 6:1 to 10:1.

The mixture consists of air and gasoline. The chemical reaction during the combustion follows

$$2\ C_8\ H_{18} + 25\ O_2 \rightarrow 16\ CO_2 + 18\ H_2O \qquad (1)$$

The $C_8\ H_{18}$ corresponds to the gasoline and O_2 the air. The carbon dioxide and water vapor gets out of the chamber through exhaust valve during exhaust stroke. In carburetor based spark ignited engines the air-fuel ratio is fixed throughout the life of the engine. If some more energy output is required, throttling will be used. During throttling, the air-fuel ratio will be changed slightly, around 2% of its original value. For example, if the original air-fuel ratio is 12, then during a maximum throttling an air-fuel ratio of 11.76 may be obtained. This is done by increasing the fuel content in the mixture. Now the simulation results of a model of SI engine with carburetor will be presented. The specifications of the engine are given below.

Working Cycle: Four Stroke Cycle,
Fuel and Method of Injection: Petrol, SI, Carburation
Number of Cylinders: 1
Cooling System: Liquid Cooling
Cylinder Bore: 150mm, Piston Stroke: 180mm, Nominal Engine Speed: 1400rpm, Compression Ratio: 9,
Cylinder Head Design: Two valves

With the above specifications the engine was simulated and obtained the performance measures which are publicly available at the link http://www.fast-files.com/getfile.aspx?file=92759. The key parameters of the above engine simulation are tabulated in the table I.

Table I: Key parameters of Carburetor based SI Engine

Piston Engine Power	34.422 KW
Mechanical Efficiency	0.80592
Air-fuel equivalence ratio	1
Thermal Efficiency	0.30006
Brake Specific Fuel Consumption	0.27267 kg/kWh
Fraction of wet NOx in exh. Gas	5437.0 ppm
Specif. NOx emis. reduc. to NO2	40.233 g/KWh
Specific SO2 emission	0.0000 g/KWh
Brake Mean Effective Pressure	9.2757 bar
Indicated Mean Effective Pressure	11.509 bar
Volumetric Efficiency	0.91401
Maximum cylinder pressure	63.8 bar

In carburetor based SI engines, the compression ratio is fixed at a predefined value. Even when the load varies the work that will be produced is almost the same. During throttling a little improvement of work delivered is possible around the predefined work. In practice, because of different use of a vehicle, different loads may be driven by similar engines. But the work delivered even with variety of loads is almost same in the carburetor based SI engines. This is the major limitation of carburetor based SI engines. If the load of the engine can be sensed, the fuel-air ratio of the mixture may be varied to drive variety of loads by a single design. Electronic sensors are available in the market which has highest grade of accuracy and ease of implementation. The application of electronic sensing equipment overruled the carburetor based SI engines more than few decades ago. Using the data from the load sensors, the fuel-air ratio will be varied to drive smoothly the sensed load. This kind of spark ignited IC engines is referred as port based SI engine.

With the similar specification as that of the carburetor based SI engine, the port based SI engine was simulated and the complete results are available at the link http://www.fast-files.com/getfile.aspx?file=92759. The key parameters of the above engine simulation are tabulated in the table II.

Table II: Key parameters of Port based SI Engine

Piston Engine Power	33.158 KW
Mechanical Efficiency	0.79571
Air-fuel equivalence ratio	1
Thermal Efficiency	0.29329
Brake Specific Fuel Consumption	0.27896 kg/kWh
Fraction of wet NOx in exh. Gas	5458.4 ppm
Specif. NOx emis. reduc. to NO2	27.183 g/KWh
Specific SO2 emission	0.0000 g/KWh
Brake Mean Effective Pressure	8.9350 bar
Indicated Mean Effective Pressure	11.229 bar
Volumetric Efficiency	0.90076
Maximum cylinder pressure	67.488 bar

III. Simulation of CI Engine

In compression ignited internal combustion engines only air will be entered in to the chamber in the intake stroke. When the piston moves upward the air gets pressurized or compressed. Now the fuel, in general diesel, will be sprayed over the inlet air. The chemical formula of the normal diesel is $C_{12}H_{23}$. It may vary from $C_{10}H_{10}$ to $C_{15}H_{28}$ because of different kind of production. The diesel has high viscosity than that of gasoline. Also the viscosity of diesel increase as the temperature decreases. The combustion takes place because of spraying of the diesel. In the Si engines the mixture was prepared before the time hence a homogeneous combustion is possible. But because of lack of time, for the fuel to mingle with the air, homogeneous combustion is impossible in compression ignited engines theoretically. But the combustion will be improved if droplet size is minimized. If the size of the droplet is minimized, then the fuel drops will get into contact of air resulting better combustion. The droplets may be sprayed with different angle. Because the space on which the droplet is going to be sprayed is a 3D plane, two angles are required to find the actual angle of spray from the nozzle. Horizontal angle is denoted wit α and vertical angle is denoted by β. A sample spraying mechanism with single nozzle with $\alpha = 60^o$ and $\beta = 60^o$ is shown in figure 1. Spraying of fuel plays a crucial role in compression ignited engines. This is very crucial because of the diesel's viscosity. The combustion performance of compression ignited IC engines can be improved by

- By changing injection velocity Vs crank angle pattern.
- By changing number of injectors.
- By changing number of nozzles.
- By taking non-identical nozzles.
- By changing the distance between spray center and bowl axis.
- By changing the distance between sprays center and cylinder head plane.
- Alpha & beta
- By changing bowl shape.
- By selecting flat or non-flat floor of piston bowl.
- Under flat floor of piston bowl
    - By changing the Piston Bowl Depth.
    - By changing the radius of hallow chamber in periphery of the bowl.
    - By changing inclination angle of a bowl forming to a plane of the piston crown.
    - By changing top-clearance at TDC.
- Under non-flat floor of piston bowl
    - By changing In-center Piston Bowl Depth
    - By changing the radius of sphere in center of piston bowl.
    - By changing the depth of a combustion chamber in periphery.
    - By changing the radius of hallow chamber in periphery of bowl.
    - By changing inclination angle of a bowl forming to a plane of the piston crown.
    - By changing top-clearance at TDC.

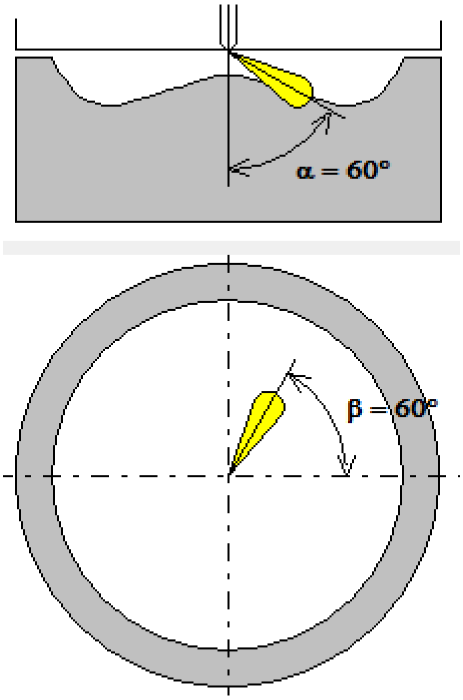

Fig. 1 Description of angle of spray

As a matter of understanding, different injection velocities are shown in figure 2.

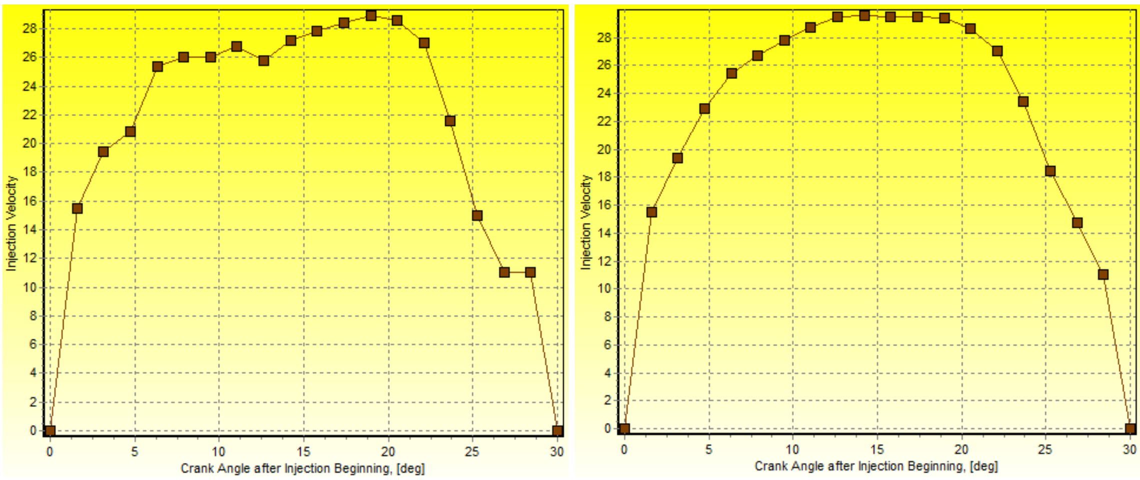

Fig. 2 Different injection velocity Vs Crank angle patterns

The injectors with different distance between spray center and bowl axis are shown in the figure 3. The injectors with different distance between spray center and cylinder head plane are shown in figure 4. Different bowl shapes for use in CI engines are shown in figure 5. Different angles of spray are shown in figure 6. Piston bowls with different Top-clearance at TDC are shown in figure 7. A number of combinations of the above are simulated and the results of three cases are presented in this paper. These cases are

Case I:

Number of nozzles: 1
Bowl shape: Pan
Top-clearance at TDC: 1mm
Distance between spray center and bowl axis: 0 mm
Distance between spray center and cylinder head plane: 0 mm
Injection velocity Vs Crank angle pattern: As shown in first part of figure 2.

Case II:

Number of nozzles: 7

The (β, α) of the nozzles are (0, 75), (51.43, 50), (102.86, 75), (154.29, 21), (-154.29, 35), (-102.86, 61), (-51.43, 90).

Bowl shape: Mexican Hat

Top-clearance at TDC: 1mm

Distance between spray center and bowl axis: 15 mm

Distance between spray center and cylinder head plane: 5 mm

Injection velocity Vs Crank angle pattern: As shown in second part of figure 2.

Case III:

Number of nozzles: 11 All the nozzles are identical

Bowl shape: Hesselman

Top-clearance at TDC: 5mm

Distance between spray center and bowl axis: 0 mm

Distance between spray center and cylinder head plane: 1 mm

Injection velocity Vs Crank angle pattern: As shown in first part of figure 2

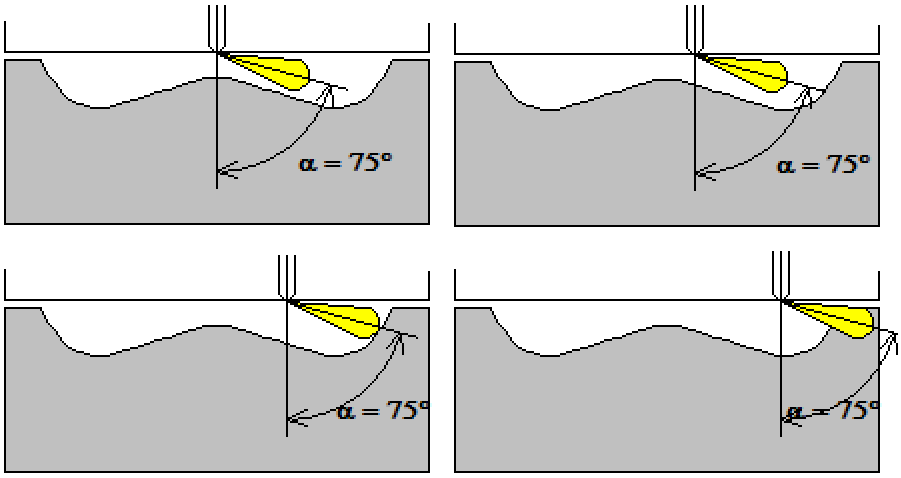

Fig. 3 The injectors with different distance between spray center and bowl axis

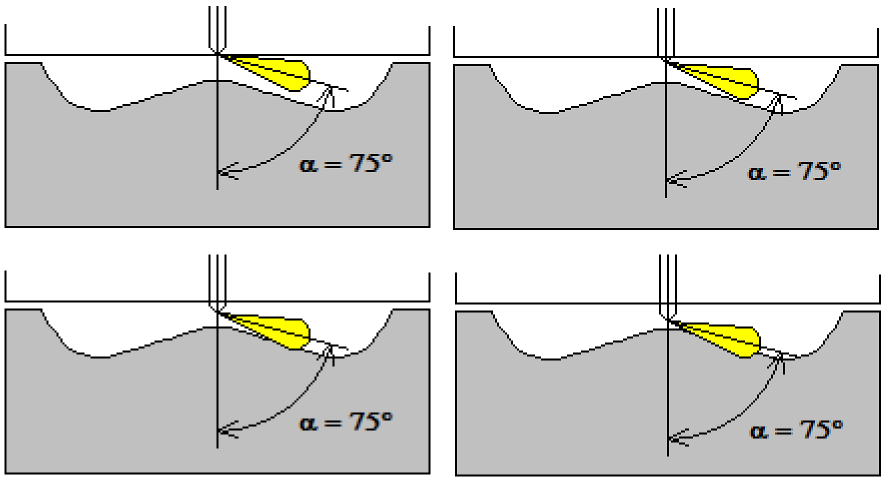

Fig. 4 Injectors with different distance between spray center and cylinder head plane

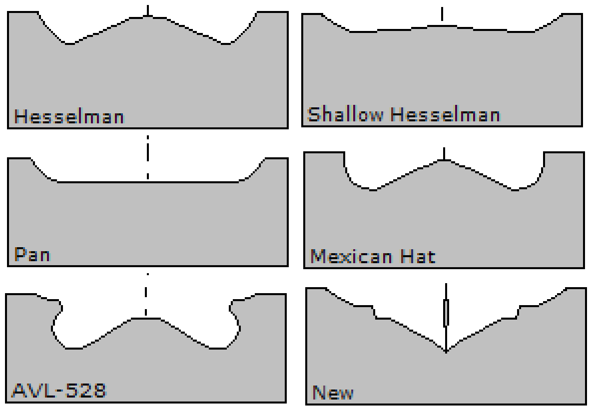

Fig. 5 Different bowl shapes

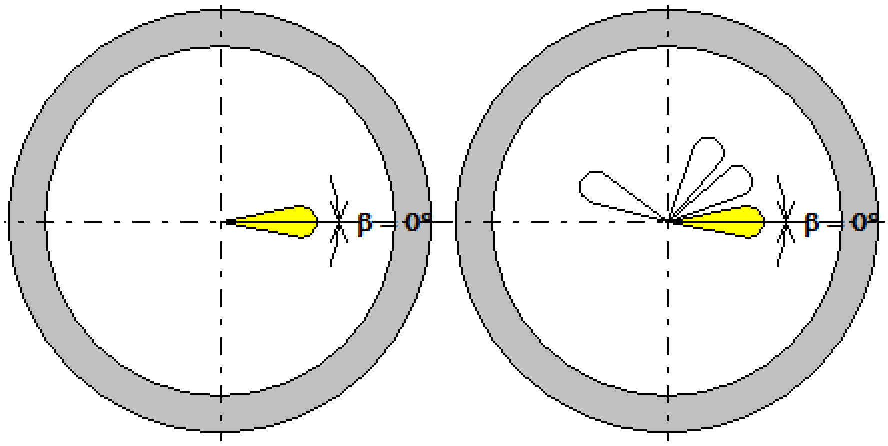

Fig. 6 Different spray angles

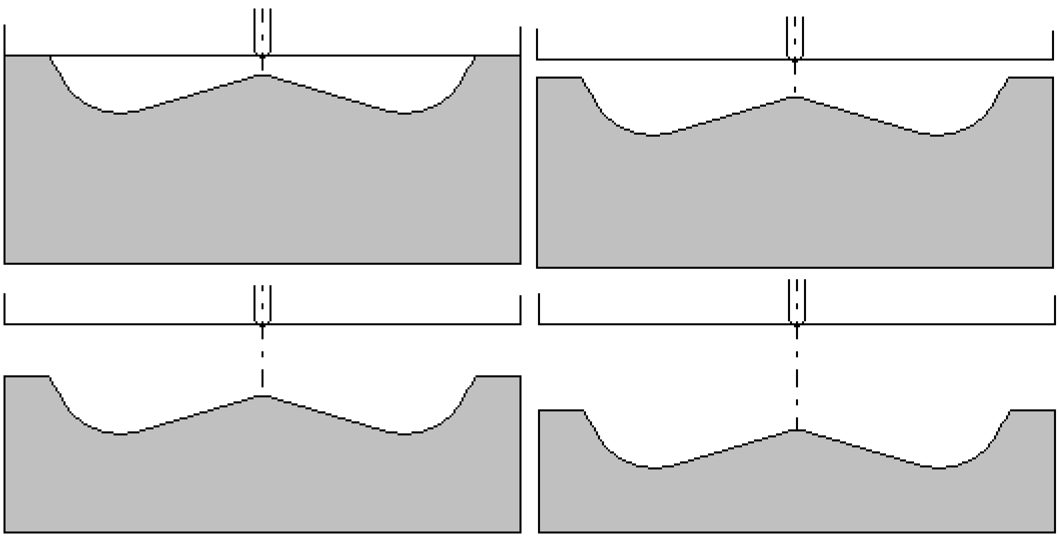

Fig. 7 Piston bowls with different Top-clearance at TDC

The specifications common to all the cases are given below.

Working Cycle: Four Stroke Cycle,
Fuel and Method of Injection: DI Diesel
Number of Cylinders: 1
Cooling System: Liquid Cooling
Cylinder Bore: 150mm, Piston Stroke: 180mm, Nominal Engine Speed: 1400rpm, Compression Ratio: 18,
Cylinder Head Design: Two ports

The complete simulation results of the above cases are made available at the link http://www.fast-files.com/getfile.aspx?file=92759. The spray models of the above three cases are shown in the figures 8 to 13.

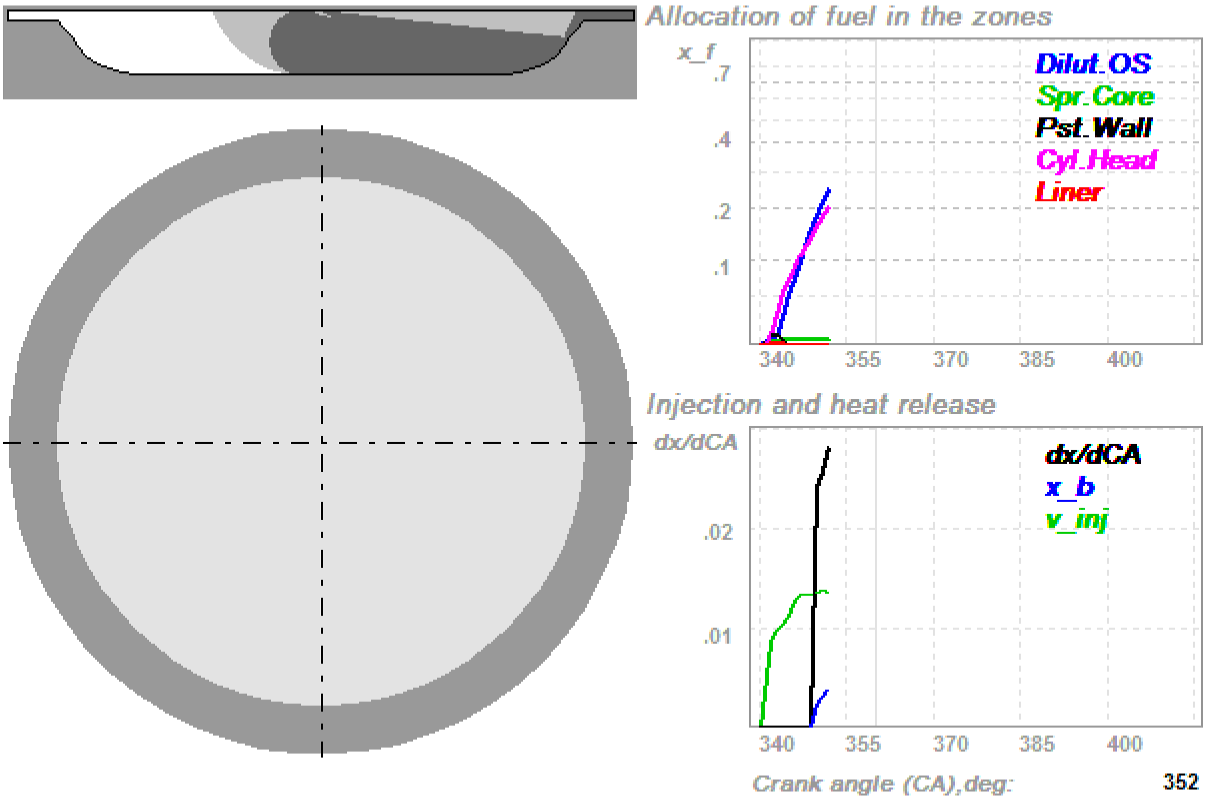

Fig. 8 Fuel spray visualization of Case – I at Crank angle 352°

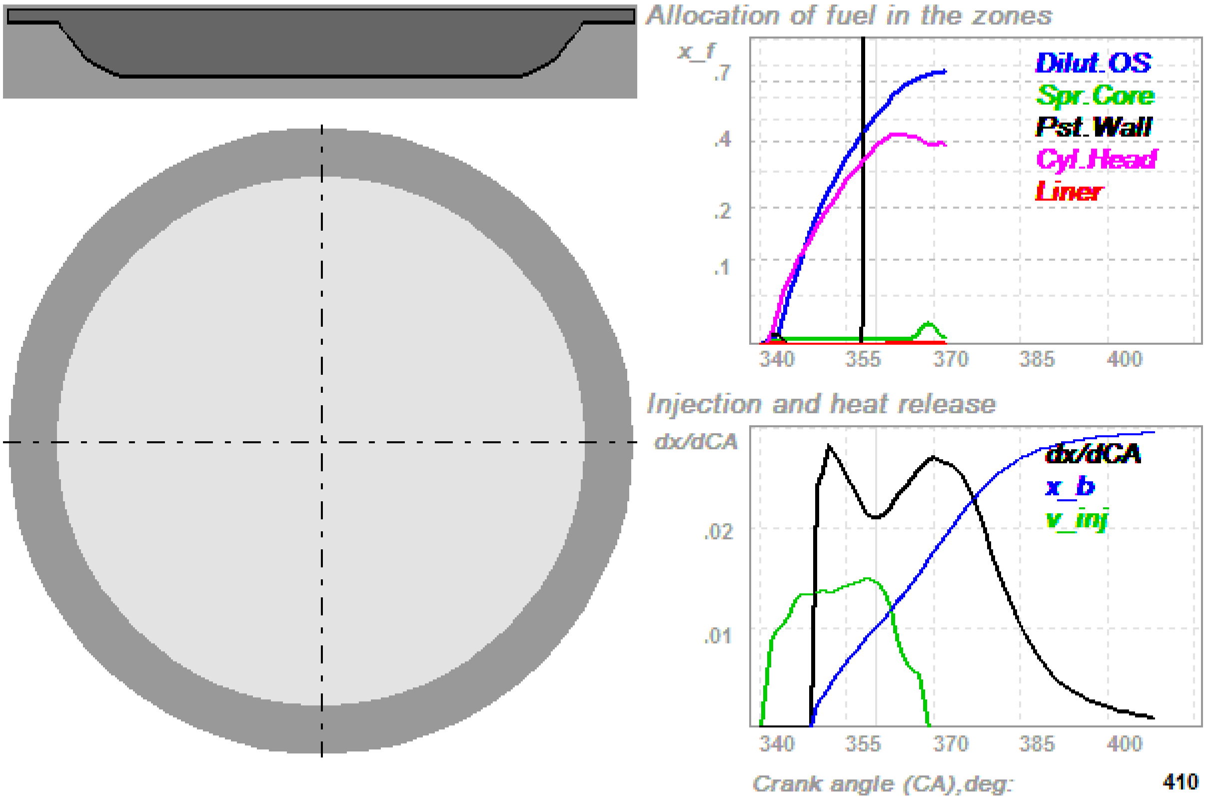

Fig. 9 Fuel spray visualization of Case – I at Crank angle 410°

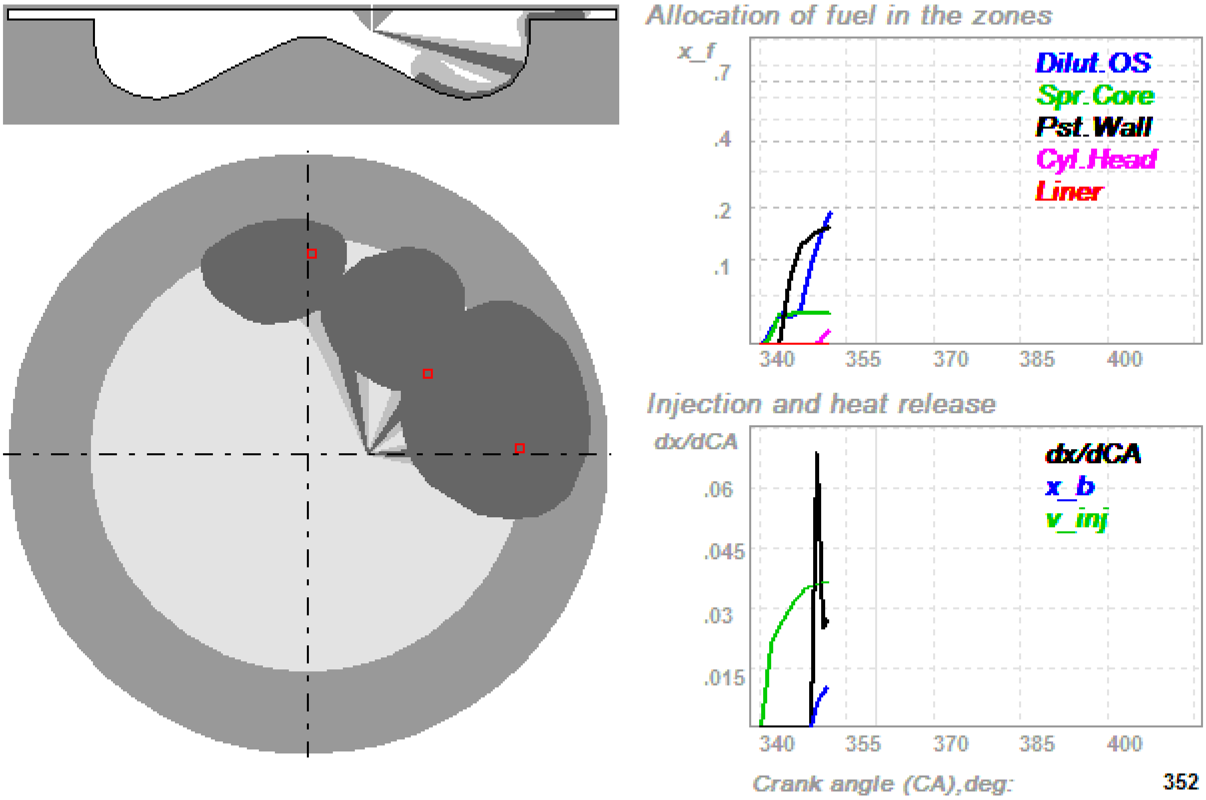

Fig. 10 Fuel spray visualization of Case – II at Crank angle 352^o

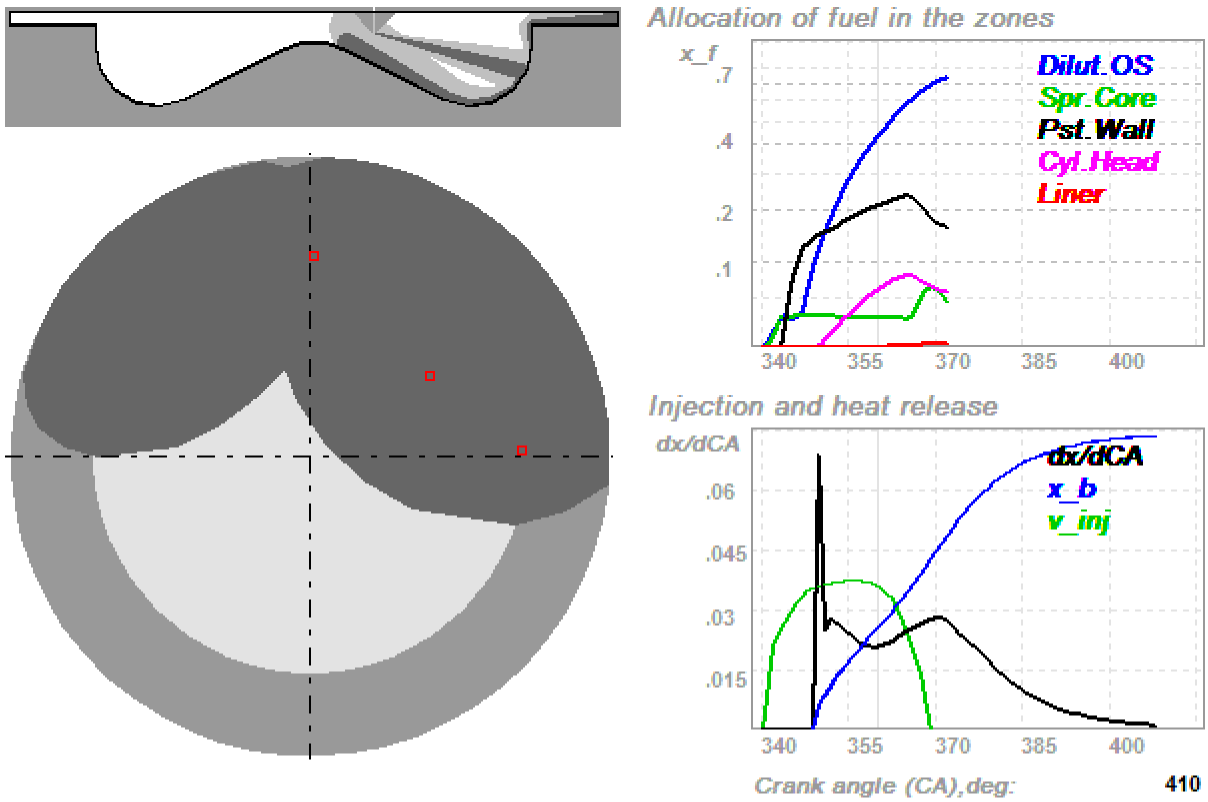

Fig. 11 Fuel spray visualization of Case – II at Crank angle 410^o

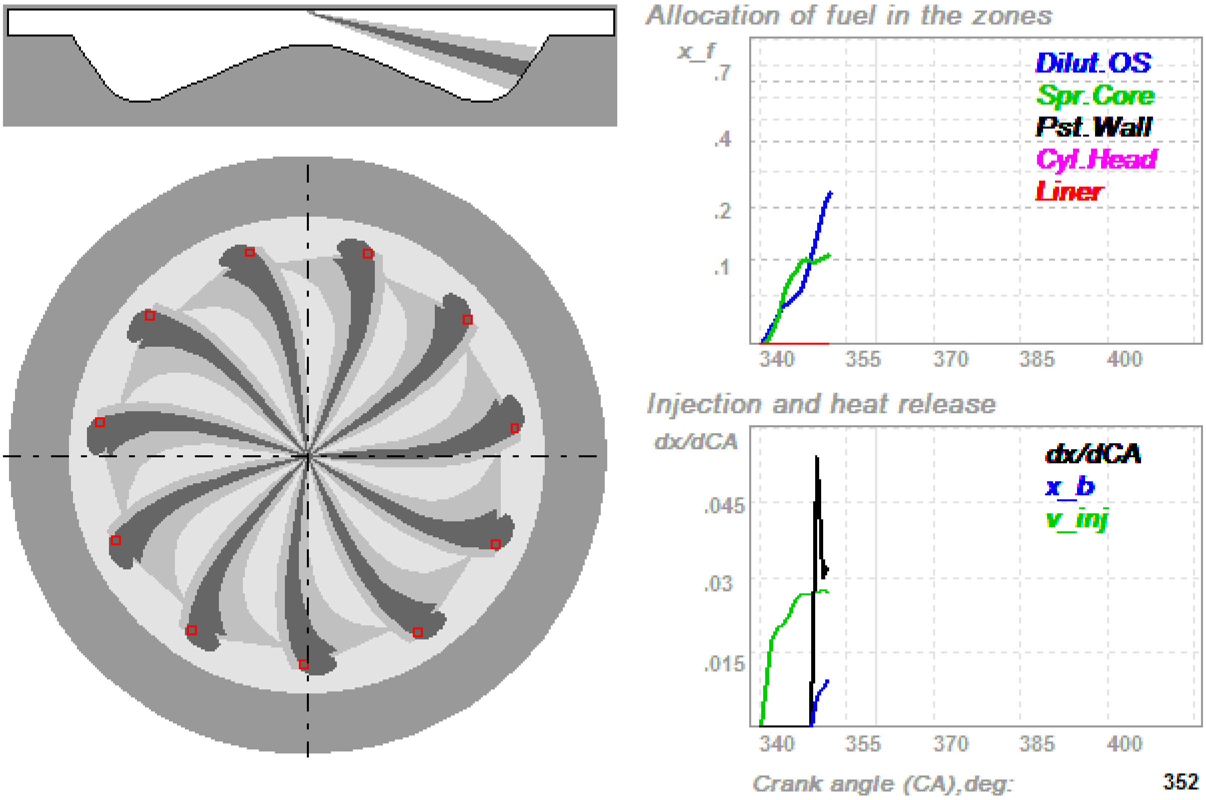

Fig. 12 Fuel spray visualization of Case – III at Crank angle 352°

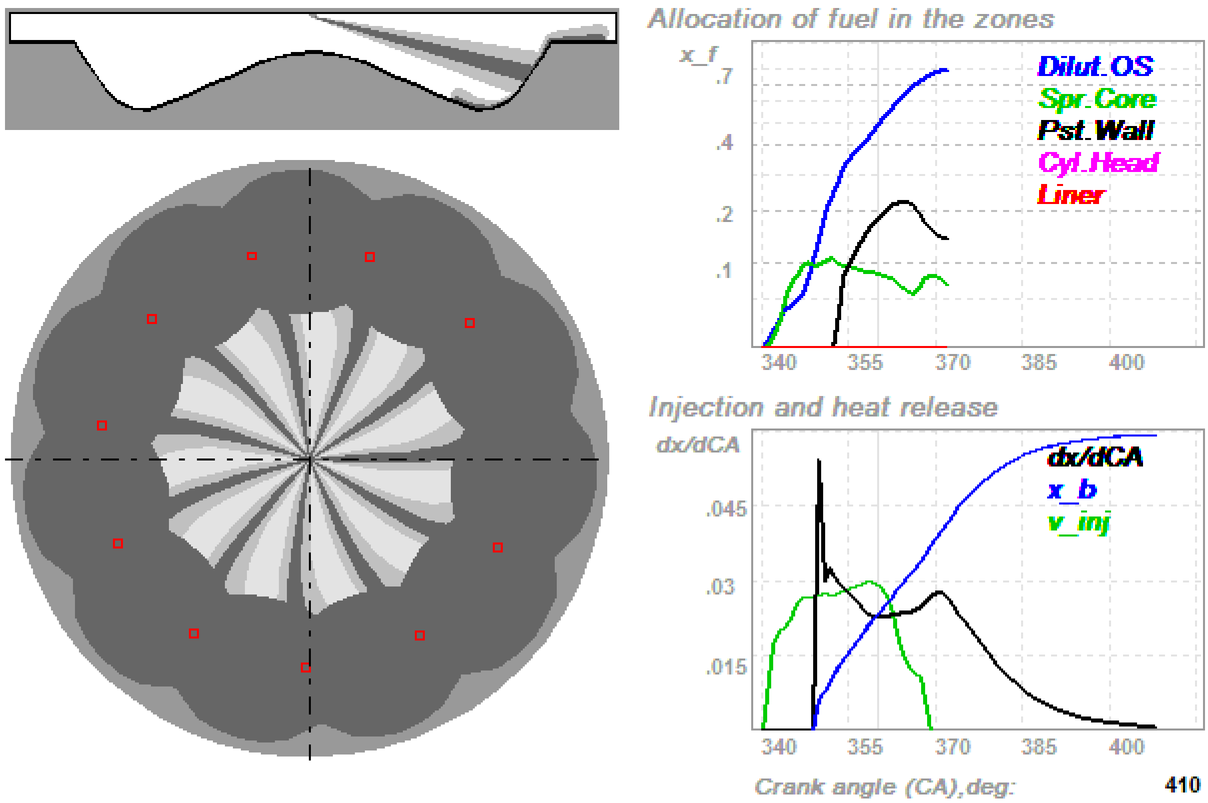

Fig. 13 Fuel spray visualization of Case – III at Crank angle 410°

The key parameters of the above engine simulation are tabulated in the table III.

Table III: Key parameters of CI engines simulated

	Case – I	Case – II	Case - III
Piston Engine Power (KW)	24.458	24.377	24.407
Mechanical Efficiency	0.76783	0.76713	0.76614
Air-fuel equivalence ratio	1.7500	1.7500	1.7500
Thermal Efficiency	0.35609	0.35507	0.35576
Brake Specific Fuel Consumption (Kg/KWh)	0.23787	0.23856	0.23810
Fraction of wet NOx in exh. Gas (ppm)	1354.9	1341.4	1504.4
Specif. NOx emis. reduc. to NO2 (g/KWh)	13.651	13.554	15.174
Specific SO2 emission (g/KWh)	0	0	0
Brake Mean Effective Pressure (bar)	6.5908	6.5688	6.5768
Indicated Mean Effective Pressure (bar)	8.5837	8.5628	8.5843
Volumetric Efficiency	0.93193	0.93143	0.93082
Maximum cylinder pressure (bar)	85.401	85.827	89.973

IV. Comparison of SI and CI Engines

The figures 14 to 19 show the comparison of different parameters of SI and CI engines. In spark ignited engines a high volatile fuel gasoline is used and it is mixed with air usually before getting entered into the chamber. Hence sufficient time is there for the fuel to mix with the air. This results in good combustion. But in compression ignited engines the fuel is diesel which has high viscosity and this viscosity tends to rise when temperature decreases. The fuel is sprayed on the compressed fuel and there is very little time for the fuel to get mingled with air. Also because the diesel has high viscosity the fuel will have less interaction with air and results in poor combustion. The compression ratio in spark ignited engines is in the range of 6 to 10; where as in compression ignited engines it is in the range of 16 to 20.

Hence more air molecules are available in the case of CI engines. These air molecules help to reduce the NOx emissions. Because in CI engines more than enough air proportion is available for the nitrogen molecules to become nitrogen dioxide, but in SI engines due to the lack of air molecules the nitrogen may form NOx which is harm to the ecosystem. This can be verified from figure 17. From figure 19 it can be observed that the maximum pressure of the cylinder is high in CI engines than that of SI engines, correspondingly the weight and size of the CI engines will be high compared to that of SI engines. It also leads to the requirement of high work to drive the self-weight also. Due to less weight the speed attainable by the SI engines is high. From the figure 15 it can be observed that the thermal efficiency of SI engines is less compared to that of CI engines. This is because the compression ratio of SI engines is less.

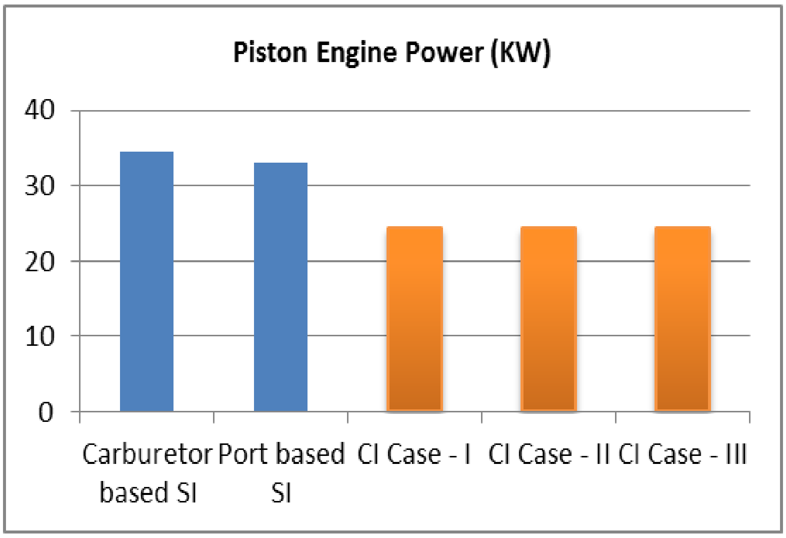

Fig. 14 Comparison of Piston engine power of SI and CI Engines

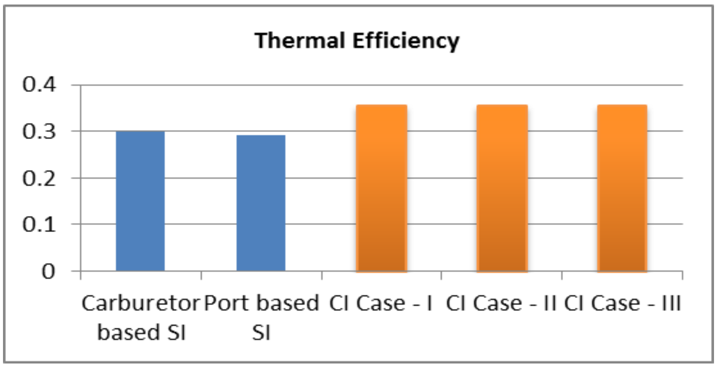

Fig. 15 Comparison of Thermal efficiency of SI and CI Engines

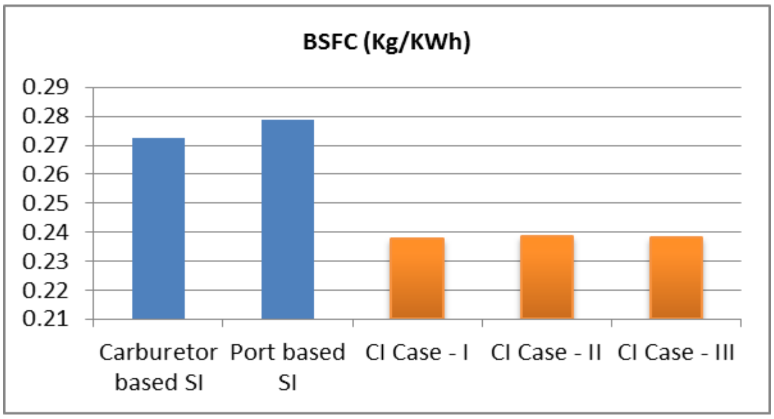

Fig. 16 Comparison of Brake specific fuel consumption of SI and CI Engines

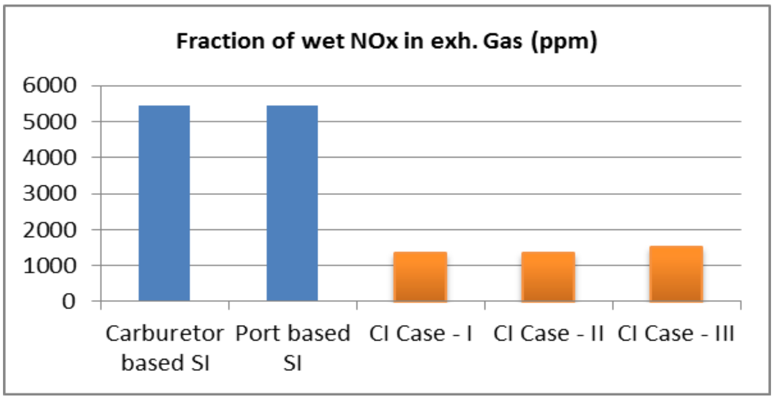

Fig. 17 Comparison of Fraction of wet NOx in exhaust gas of SI and CI Engines

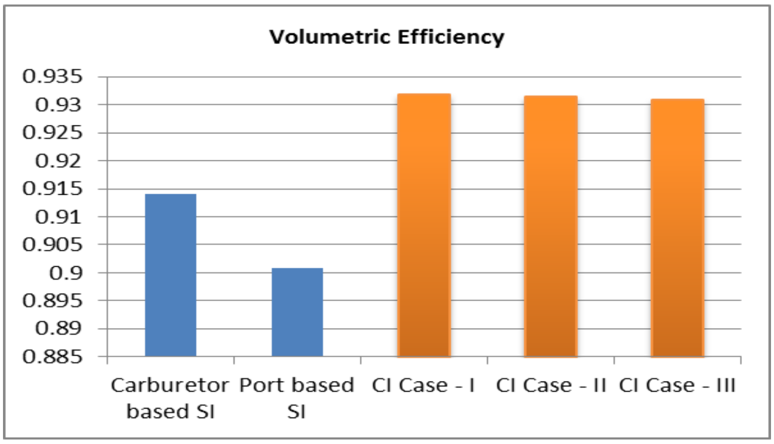

Fig. 18 Comparison of volumetric efficiency of SI and CI Engines

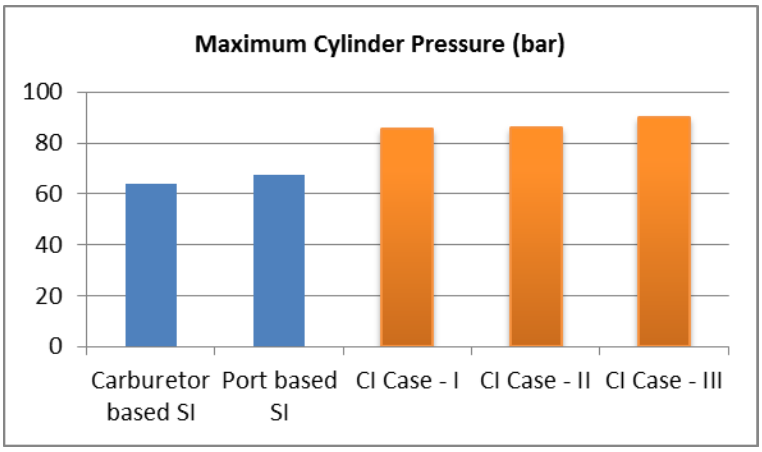

Fig. 19 Comparison of maximum cylinder pressure of SI and CI Engines

V. Conclusions

In this paper a comprehensive modeling of both SI and CI engines was presented. The carburetor based and port based engines are considered in SI category. Three cases of CI engines are considered. In each- case different spraying characteristics, different injection velocity patterns and different bowl characteristics are taken. The SI and CI engines are compared based on performance metrics like piston engine power, specific fuel consumption, NOx levels, and cylinder pressure, thermal and volumetric efficiency. The advantages and limitations of SI and CI engines are elaborated. It might be questioned to devise an engine with the advantages of both SI and CI engines. There are three alternative for this. One by adding diesel fuel with air ahead of time or indirectly, sufficient time must be provided for the diesel fuel to mix with air. Second, the gasoline fuel may be sprayed on the compressed air. The last, the gasoline may be mixed with air before the time and then by compressing it as in the case of CI engine. The possibilities and implementations of the above three alternatives are the current state of art in the IC engine development.

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