Numerical Simulation of Combustion Process in a DI Engine using Micro Chambers in Piston Head

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Abstract

Radical Ignition engine combustion is dependent on concentration on radicals, which have low autoignition temperatures at low compression ratios. The radicals are produced at low temperatures and pressures in an auxiliary chamber (Micro chamber) which are embedded in the piston head. Although, the micro-chamber combustion chamber engines are considered to be the future trend, too little information is available in the literature. The numerical and experimental studies to understand the working details of this mechanism are the need of the hour.

A qualitative numerical study of the combustion process in a DI engine using Micro chamber in piston head was taken up to predict the radicals generated in the micro chambers for the initial engine cranking phase. The numerical simulation was carried out for the flow parameters and chemistry interactions between the two chambers (micro and main chamber). The case was simplified by considering a two dimensional model for the simulation and a homogeneous mixture of methane and air in the chambers. The simulation was carried out for 600 and 1000 rpm. Standard two equation k-ε turbulence model was used with spark ignition to start the combustion process. The simulation involved detailed chemical kinetics using 26 species and 84 reactions.

The results showed larger amounts of radicals generated simultaneously in both the chambers. The radicals left over in the micro chamber at the start of exhaust valve open were predicated and found to be at lower levels than required. The radical levels may not be sufficient for the autoignition cycles. High jet velocities flowing out of the vents with high turbulence intensity were observed during the combustion process.

Keywords: Micro chambers, DI engine, combustion, Piston head

1. INTRODUCTION

The use of Direct Injection (DI) engines has many advantages like high power density, higher fuel efficiency and lower emissions. But this technology is limited to maximum injection pressures in diesel and gasoline engines. New emission norms are more stringent and demanding on the engine manufacturers. Of the several techniques of reducing the emissions from the engine, the in-cylinder technique is very challenging and effective. New combustion methods like the Gasoline Direct Engine (GDI), HCCI (Homogenously Charged Compression Ignition Engine) and Radical Ignition Engine (RI) are at the early stage of the developments and have advantages over the conventional Compression Ignition (CI) and Spark Ignition (SI) engines.

The combustion in RI engine is governed by the presence of the radicals. These radicals have lower autoignition temperatures as compared to the normal fuels. The radicals in the RI engine are produced in an auxiliary chamber called the Micro Chamber (MC) at low temperatures and pressures. This environment is created in the combustion chamber by providing small chambers in the piston head. Figure 1 shows the schematic representation of the piston head with the micro chambers.

Figure 2 shows the autoignition limits for a given fuel. The shaded region represents the cool flame region where slow oxidation of the fuel produces radicals. These radicals and intermediate species are highly active and enhance combustion process.

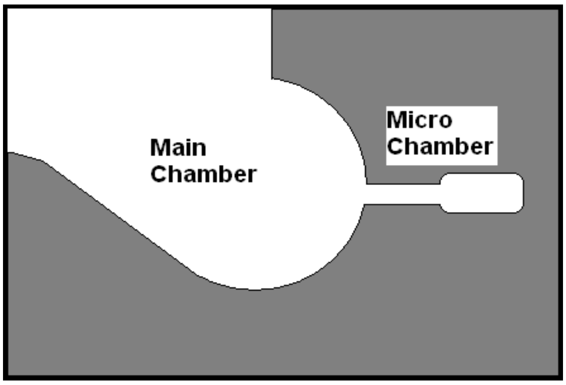

Fig. 2 Micro Chambers in Piston Head

Early work in this field was done by Gussak [1]. Many tests were taken up for a SI engine using a pre-chamber where a rich fuel was oxidized to generate the radicals. The main chamber of the engine was filled with lean mixture. The results showed that there was an increase in the combustion efficiency by 10%, overall combustion duration decreased by 3-4 times and also a decrease in ignition delay by 3-4 times. All these results were compared with SI engine characteristics. Further research was carried by Blank et al [2] for a DI engines using methanol. The work carried out was only numerical and involved multidimensional modeling. Two different codes were coupled to solve for flow parameters and the chemistry interactions (KIVA and

CHEMKIN). It was found that radicals like CH_2O and H_2O_2 were important for the combustion process involved.

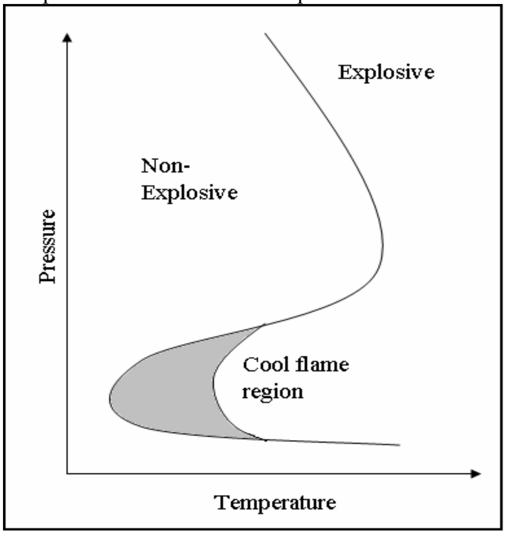

Fig. 3 Auto-ignition Limits [5]

Further Blank and Pouring [3] studied the feasibility of the RI engine using Hydrogen (H_2) as the fuel along with EGR. The effect of EGR was prominent and still further reduced the autoignition limits of the radicals. In another work by Blank and Pouring [4] for the same fuel, the effect of the injection timing on autoignition limits was studied in detail. The results indicated that the injection timing can be effectively used to control the autoignition limits of the fuel. The work also involved the study of mixing of the radicals in the intake stroke. A 15° delay in opening of the inlet valve lead to optimized mixing of the radicals left over from the previous combustion cycle. Blank [5] also explored the use of methane as the fuel for a RI engine. This work was also mainly studied numerically. Very highly active radicals like OH, H_2O_2, CH_2O and HO_2 governed the combustion process.

Numerous literatures have showed the importance of CFD, which have proved that numerical methods for combustion simulation are economical and provide a tool to study the parameters independently. The two equation *k-ε* turbulence model has been widely used in for turbulent flows [6, 7& 8]. The combustion models like Ricardo Two Zone Flamelet (RTZF) and Eddy Dissipation Concept (EDC) have been effectively used for the combustion models.

The literatures reviewed show that radical ignition engines can run on different fuels. The effect of the injection timing, dimension and position of the vents are vital for the radical generation process. But no work has been published for the initial radical generation process. Thus a qualitative attempt has been made to study the radical build up during the engine cranking phase.

2. MODEL CONSTRUCTION

2.1 Engine Specifications

The engine specifications for the current work have been taken up from [5] and are given as in Table 1.

Table 1 Engine Specifications

Parameter	Specification
Bore	104.8 mm
Stroke	130 mm
Compression ratio	15.5:1
Bowl volume	55.2 cc
Fuel	Methane (CH_4)
Number of micro chamber	6
Micro chamber volume	1.92 cc
Injection timing	58° BTDC
Injection duration	18° CA
Exhaust Valve Open	500° CA

2.2 Geometric Model and CFD model

For the current study the dimensions are drawn from [5]. A 2D simulation was carried out to simplify the problem. The wire frame combustion model was generated in CATIA.

The wire frame model was imported to Gambit 2.3.16 and grid was generated. A quadrilateral cell with 4 node was used so as to obtain better results with minimum computational requirements. The grid consisted of 34818 cells at 180° CA. The

2.3 Numerical Model

A commercial CFD code Fluent 6.3.26 was used for the simulation. This tool also can solve the chemical reactions specified in the CHEMKIN format along with the governing equation of CFD. A pressure based implicit solver was used where the continuity, momentum and energy equations are solved sequentially. Since the fluid domain changes with respect to time, an unsteady time based solver was adopted. Standard *k-ε* turbulence model was used for the turbulence model. The computational time requirements for a coupled analysis are high, thus first order discretization schemes were used. The initial boundary conditions for the simulations were pressure as 0.8 bar at 180° CA and 300 °K as the temperature. Also a convective heat transfer coefficient of 1000 W/ m^2 °K was assumed. A top layering dynamic mesh was used. This method is particularly important since the cells collapsed and generated are near the cylinder head. Figure 3 shows the computational model with the applicable boundary conditions at start of simulation at 320° CA.

To initiate the combustion process, spark ignition model was used and the combustion model was the finite rate chemistry. This model uses the Arrhenius equations to calculate the reaction rates. Since the spray modeling was not considered due to the complexity of the problem, a stoichiometric (ϕ = 1) and rich (ϕ = 1.2) homogenous air fuel ratio was assumed in main and micro chambers respectively at the start of the simulation. The reactions for the combustion process were imported in CHEMKIN format [5].The simulation was run for 600 and 1000 rpm with 350° CA and 345° CA spark timings respectively.

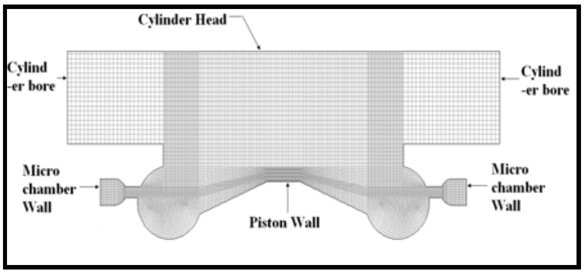

Fig. 4 Computational Domain

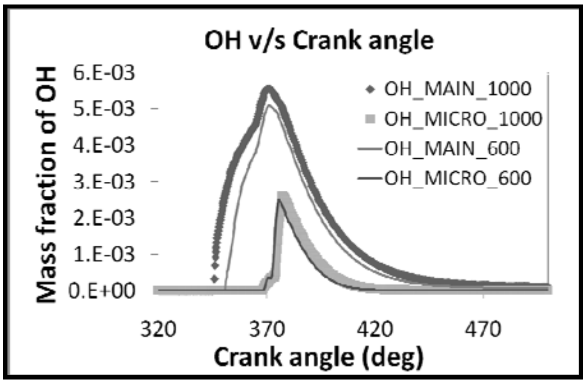

Fig. 5 OH v/s Crank Angle

3. RESULTS AND DISCUSSION

3.1 Motoring cycle, Combustion temperature and pressures

A motoring cycle was run to calculate the pressure and temperature with the initial conditions mentioned above. The peak pressure and temperature obtained from the simulation were 37.11 bar and 858°K.

The combustion pressures and temperatures obtained for the 600 and 1000 rpm cycles have been summarized in table 2 and table 3 respectively.

3.2 Radicals

The radicals generated in the process in main and micro chamber has been shown in Fig.4-6. The important results were the radicals being generated in main and micro chambers during the combustion process. Radicals like OH, H_2O_2, CH_2O and HO_2 were found to be the most dominant of all other radicals taking part in the combustion process.

Table 2 Combustion Pressures

Rpm	Pressure (bar)
1000	83.01
600	81.26

Table 3 Combustion Temperatures

	Main chamber		Micro chamber	
Rpm	Temp (°K)	CA (deg)	Temp (°K)	CA (deg)
1000	2883.52	373.25	2669.95	378
600	2803.88	373	2561.02	381.25

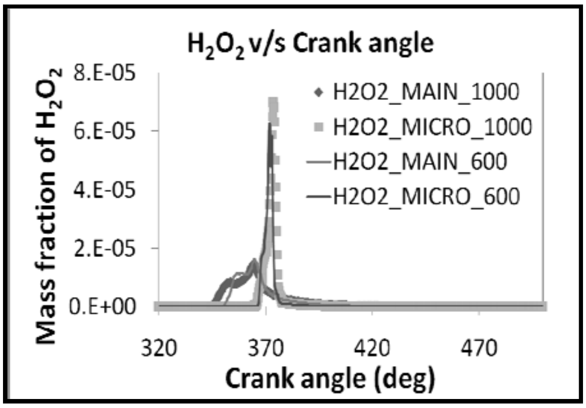

Fig. 6 H_2O_2 v/s Crank Angle

The mass fractions of the radicals are seen soon after the spark ignition is given in the cycle. The OH radical is the most dominant during the combustion reactions. The other radicals also react with the fuel but also increase the overall mass fraction of OH radical in the process.

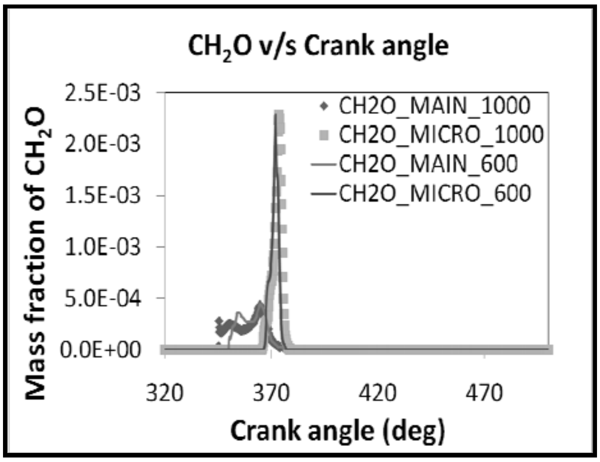

Fig. 7 CH_2O v/s Crank Angle

3.3 Mass flow rate

The mass flow rate between the two domains (micro and main chambers) is shown in Fig 7.

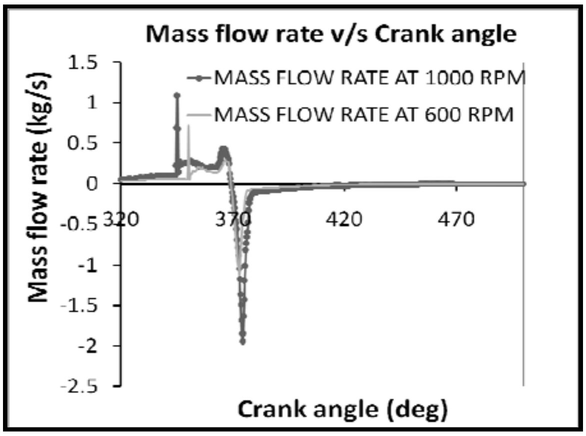

Fig. 8 Mass Flow Rate v/s Crank Angle

The mass flow into the micro chamber is considered as positive. Until the start of the expansion process, the mass flows from the main chamber to micro chamber due to two reasons.

- Compression stroke
- Pressure created due to combustion

As soon as the pressure in main chamber is less than that of the micro chamber, the difference in the pressures drives the mass from the micro chamber to main chamber with great velocities. Since the connecting vents are very small compared to overall dimensions of the combustion chamber, the mass flowing out of the micro chamber gives a stratifying effect to the combusted material in main chamber. A steep fall in the mass flow rate in the Fig 7 represents this phenomenon.

3.4 Jet Velocities and Turbulence Intensity

Figure 8 and 9 show the peak velocities of mass flowing out from the vents for 600 and 1000 rpm respectively. The peak velocity for 600 and 1000 rpm at respective CA are given in table 4.

Table 4 Jet velocity

RPM	Velocity (m/s)	Crank angle (deg)
1000	39.4	375.25
600	23.2	373.75

The presence of high velocity jets in the process increases the turbulence intensity in the main chambers. Figures 10 and 11 show the peak turbulence intensity (%).

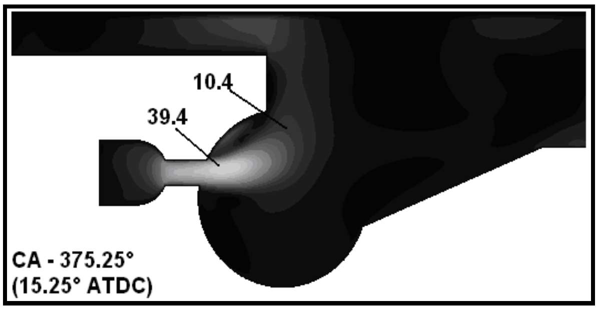

Fig. 9 Jet Velocity at 1000 rpm

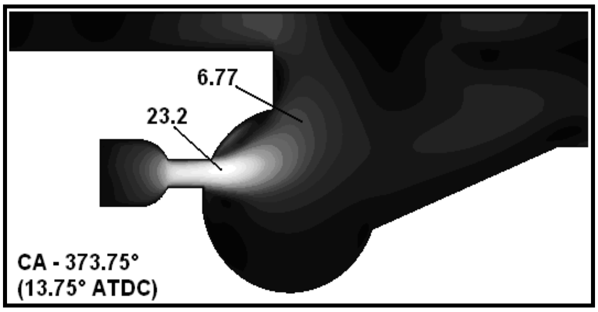

Fig. 10 Jet Velocity at 600 rpm

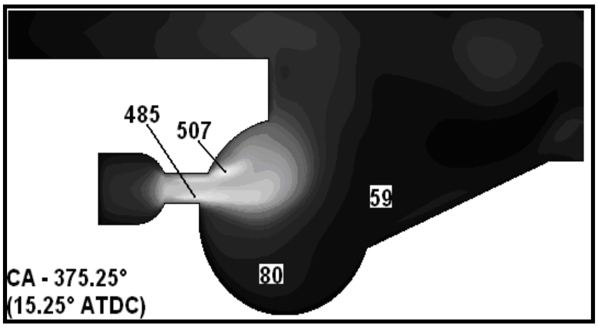

Fig. 11 Turbulence Intensity at 1000 rpm

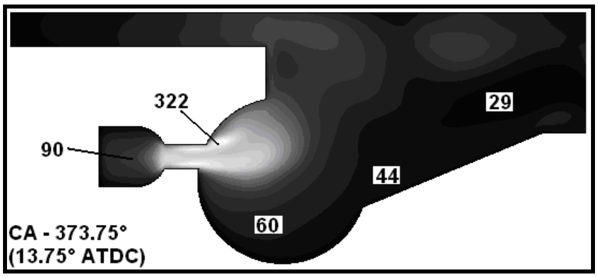

Fig. 12 Turbulence Intensity at 600 rpm

4. CONCLUSIONS

A qualitative numerical simulation was carried out to understand the combustion process in RI engines and also to predict the initial radical build up for the autoignition cycles.

The conclusions that can be drawn from the current work are:

- The results found out were based on a 2D simulation and could not be compared to any experimental works. The results also cannot be compared to any works, since the flow characteristics differ from 2D to 3D.
- The simulation provided a deep insight to the combustion process of a RI engine.
- Radical are produced in large quantities simultaneously in main and micro chambers respectively.
- All radicals produced in micro chamber were greater in terms of percentage by mass than in main chamber.
- Of all the radicals, OH dominated the reactions.
- High jet velocities upto 39.4 m/s were observed in vents.
- A gradual build up radicals in micro chamber was observed as the engine speed was increased from 600 rpm to 1000 rpm.
- Increase in turbulence intensity of 39% between 600 rpm to 1000 rpm.
- The current level of radials in micro chamber may not be sufficient enough for the next cycle autoignition.

REFERENCES

[1] Gussak L. A (1975), "High Chemical Activity of Incomplete Combustion Products and a Method of Prechamber Torch Ignition for Avalanche Activation of Combustion in Internal Combustion Engines", SAE Paper # 750890.

[2] David A. Blank, Andrew A. Pouring and Jiang Lu (2001),"Methanol Combustion in Low Compression Ratio D.I. Engines Enabled by Sonex Piston Design", SAE Paper # 2001-01-1197.

[3] David A. Blank and Andrew A. Pouring (2003), "Frozen Equilibrium and EGR Effects on Radical-Initiated HD2 Combustion Kinetics in Low-Compression D.I. Engines Using Pistons With Micro-Chambers", SAE Paper # 2003-01-1788.

[4] David A. Blank and Andrew A. Pouring (2004), "Radical-Controlled Autoignition At Reduced Compression Ratios in a Hydrogen D.I. Diesel Engine With Piston Micro-Chambers", SAE Paper # 2004-01-1846.

[5] David A. Blank (2004), "CNG/Methane Combustion Kinetics and Frozen Equilibrium in the Radical Ignition Reduced Compression Ratio DI Diesel Engines using Piston with the Micro Chamber", SAE Paper # 2004-01-1677.

[6] Hou Z. X., and Abraham J. (1995), "Modelling of Diesel Spray in a Very High Pressure Chamber, Part II: Effects of Combustion", SAE Paper # 950603.

[7] Yasuo Murakami, Hitoshi Kurosaka and Hiroshi Kamiya (2004), "Practical Application of Combustion Simulation using CFD for Small Engine of Two – Wheeled Vehicles", SAE Paper # 2004-32-0006

[8] Ming Jia and Maozhao Xie (2007), "Numerical Simulation of Homogeneous Charge Compression Ignition Combustion using Multi-dimensional Model", Journal of Automobile Engineering, April 2007