COMBUSTION MODELING OF S.I. ENGINE FOR PREDICTION OF TURBULENT FLAME SPEED

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Abstract

Modeling is a process of developing and using the appropriate combination of assumptions and equations that permit the critical features of the process to be analysed. In case of spark ignition engine, fuel and oxidizer are mixed at the molecular level prior to ignition. Here, Combustion occurs as a flame front; propagating into the unburnt reactants called as premixed combustion. This paper presents the combustion modeling of single cylinder four stroke spark ignition engine having compression ratio of 9.0 and displacement of 97.2 cc using computation fluid dynamics for predicting turbulent flame speed by using premixed combustion model. The methane gas is considered as a fuel in this study. Prediction of turbulent flame speed at different equivalence ratio and engine speed is carried out using FLUENT software.

Keywords—**Combustion Modeling, Premixed combustion, Turbulent flame, Equivalence ratio.**

1. Introduction

Combustion process is a chemical phenomenon which involves exothermic chemical reaction between the fuel and the oxidizer (air) [3]. The term combustion is saved for those reactions that take place very rapidly with large conversion of chemical energy into sensible energy. When a combustible fuel-air mixture is ignited with a spark, a flame propagates with a velocity determined by the kind of fuel-air mixture and the external conditions. Accordingly; the velocity of flame propagation depends on whether the vessel is taken as a reference or the unburned gas is taken as reference. Usually, the former is referred to as the flame travel speed, while the latter is known as the flame propagation speed or the flame velocity. Turbulent flame speed for C.I and S.I engine can be predicted by using non- premixed combustion, premixed combustion and partially premixed combustion models. Garner and Ashforth [1] studied experimentally the effect of pressure on flame velocities of benzene-air and 2,2,4-trimethyl-air mixture below atmospheric pressure. They found that the flame velocity increases with the decrease in pressure. K. A. Malik [3] developed a theoretical model for turbulent flame speed, based on turbulent transport process for spark ignition engine. This model was then taken into account and the effect of turbulence was generated by (i) the expanding flame front and (ii) the inlet valve geometry. The predicted turbulent flame speed values were compared with the experimental values in the speed ranges 600 rpm to 1160 rpm and fuel air equivalence ratio from 0.8 to 1.25 which were found to be in good agreement with each other. Sunil U. S. Moda [15] did a computational investigation of heavy fuel feasibility in a gasoline direct injection spark ignition engine. He developed a computational model to explore the feasibility of heavy fuel in a gasoline direct injection spark ignition engine. A geometrical model identical to that of the Pontiac Solstice 2008 was developed using ANSYS and Gambit 2.4. Abu-Orf [4] developed a new reaction rate model and validated the results for premixed turbulent combustion in spark ignition engines. The governing equations were transformed into a moving coordinate system to take into account the piston motion. The model behaved in a satisfactory manner in response to changes in fuel type, equivalence ratio, ignition timing, compression ratio and engine speed. In accordance with the various literatures, the combustion modeling of single cylinder four stroke spark ignition engine of 97.2 cc was done and the turbulent flame speed was predicted using pre-mixed combustion.

2. The Role of CFD Analysis in Engine Design

I.C. engines involve complex fluid dynamic interactions between air flow, fuel injection, moving geometries, and combustion. Fluid dynamics phenomena like jet formation, wall impingement with swirl and tumble, and turbulence production are critical for high efficiency engine performance and for meeting emissions criteria. CFD has the potential for providing detailed and useful information and insights that can be fed back into the design process. This is because in CFD analysis, the fundamental equations that describe fluid flow are being solved directly on a mesh that describes the 3D geometry, with sub-models for turbulence, fuel injection, chemistry, and combustion. Using CFD results, the flow phenomena can be visualized on 3D geometry and analysed numerically, providing tremendous insight into the complex interactions that occur inside the engine. Hence CFD analysis is used extensively as part of the design process in automotive engineering, power generation, and transportation. With the rise of modern and inexpensive computing power and 3D CAD systems, it has become much easier for analysts to perform CFD analysis. In increasing order of complexity, the CFD analyses performed can be classified into:

Port Flow Analysis: Quantification of flow rate, swirl and tumble, with static engine geometry at different locations during the engine cycle.

Cold Flow Analysis: Engine cycle with moving geometry, air flow and no fuel injection or reactions.

In-Cylinder Combustion Simulation: Power and exhaust strokes with fuel injection, ignition, reactions, and pollutant prediction on moving geometry.

Full Cycle Simulation: Simulation of the entire engine cycle with air flow, fuel injection, combustion, and reactions.

3. Premixed Combustion

In premixed combustion, fuel and oxidizer are mixed at the molecular level prior to ignition. Combustion occurs as a flame front propagating into the unburnt reactants. The turbulent premixed combustion model, involves the solution of a transport equation for the reaction progress variable. The closure of this equation is based on the definition of the turbulent flame speed. Examples of premixed combustion include aspirated internal combustion engines, lean premixed gas turbine combustors, and gas-leak explosions. Premixed combustion is much more difficult to model than non-premixed combustion. But, the essence of premixed combustion modeling lies in capturing the turbulent flame speed, which is influenced by both the laminar flame speed and the turbulence. The model is based on the assumption of equilibrium small-scale turbulence inside the laminar flame, resulting in a turbulent flame speed expression that is purely in terms of the large-scale turbulent parameters.

3.1 Propagation of the Flame Front

In many industrial premixed systems, combustion takes place in a thin flame sheet. As the flame front moves, combustion of unburnt reactants occurs, converting unburnt premixed reactants to burnt products. The premixed combustion model thus considers the reacting flow field to be divided into regions of burnt and unburnt species, separated by the flame sheet. For computation of perfectly premixed turbulent combustion, it is common practice to characterize the progress variable c(for unburned gas c = 0 and for the product gas c = 1). The transport equation is as following [22].

$$\frac{\partial \bar{\rho}\,\tilde{c}}{\partial t} + \frac{\partial}{\partial x_j}\left(\bar{\rho}\tilde{u}_j\tilde{c}\right) = -\frac{\partial y}{\partial x}\left(\overline{\rho u_j'' c''}\right) + \bar{\rho}\tilde{W}$$

Where,

t is time.

x_j and u_j the coordinate and flow velocity component respectively.

ρ is the gas density.

$\tilde{W}$ is the mean rate of product creation.

3.2 Turbulent Flame Speed

The turbulent flame speed is influenced by the following factors:

- ✓ Laminar flame speed, which is, in turn, determined by the fuel concentration, temperature, and molecular diffusion properties, as well as the detailed chemical kinetics

- ✓ Flame front wrinkling - stretching and flame thickening. The former is influenced by large eddies and the latter is influenced by small eddies. The turbulent flame speed is computed [22]

$$U_t = Au'\left(\frac{\tau_t}{\tau_c}\right)^{1/4}$$

Where,

A = model constant.

u' = RMS (root mean square) velocity (m/s)

$\tau_t = l_t / u^1$ turbulence time scale (s)

$\tau_c = \alpha / U^2{}_1$ = chemical time scale (s)

4. Modeling and Meshing

The geometry of the engine of 97.2 cc is modelled in SOLID EDGE software. To simplify the meshing, the geometry cleanup is done in ANSYS meshing module software. A SOLID EDGE model is imported into FLUENT and the geometry clean-up is performed. The simple geometry is meshed and specific zone names and types are assigned.

Table 1 Specification of the engine [19].

Input Parameter	Value of Parameter
Type	Air cooled, 4-stroke, O.H.C engine
Cylinder arrangement	Single cylinder 80° inclined from vertical.
Bore	50.0 mm
Stroke	49.5 mm
Displacement	97.2 cc
Compression ratio	9.0:1
Inlet Valve Opening (IVO)	4° BTDC
Inlet Valve closing (IVC)	24° ABDC
Exhaust Valve Opening (EVO)	26° BBDC
Exhaust Valve closing (EVC)	1.5° TDC

4.1 Geometric Model

The fluid model that was designed in SOLID EDGE is shown below. It was designed according to the specification obtained from Workshop manual of a 97.2cc bike engine.

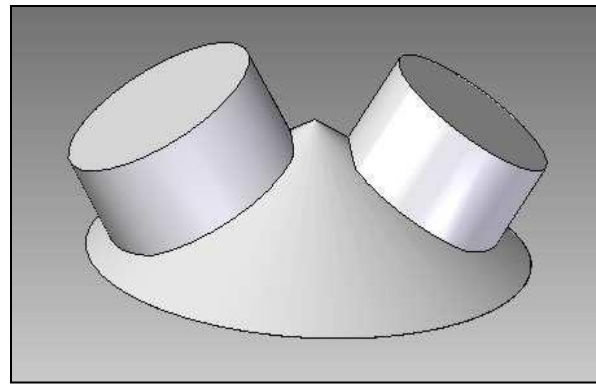

Fig.1 Solid edge model created from real Engine.

4.2 Mesh Motion

The mesh motion which is the basic for all the simulations is achieved in ANSYS mesh module. The mesh created is based on the crank angle specified and the results obtained from the dynamic mesh generation agreed with the actual cylinder movement to a considerable extent. The valve profiles help to exactly replicate the actual valve motion in the computational environment. Figure 2 and 3 show the mesh motion with respect to the crank angle.

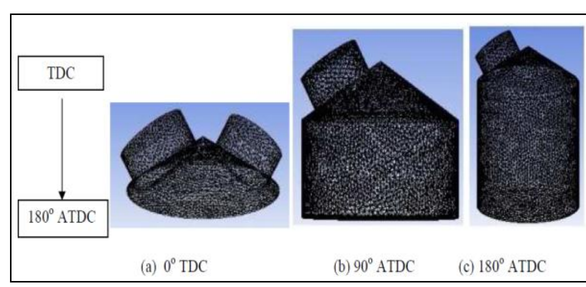

Fig. 2 Mesh motion for first stroke

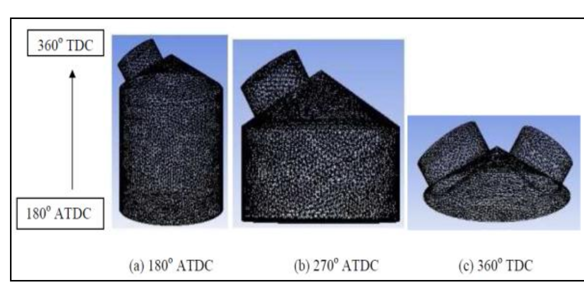

Fig. 3 Mesh motion for second stroke

5. Hypothesis

The combustion analysis includes hypothesis of the model (flame structure, species diffusion, chemical species involved, burnt gases chemical reactions, and chemical equilibrium) and analysis of the working cycle for calculation of pressure, temperature and volume for suction, compression, combustion and expansion process.

5.1 Flame Structure

The main assumption made for this work is that the combustion of fuel occurs in a premixed regime, even for very lean or very rich combustion. The region of fuel consumption is supposed to be very thin and we assume that it separates unburnt from burnt gases and also that no fuel remains in the burnt gases. This assumption can be easily justified because the high temperature existing in the burnt gases leads to fuel molecules decomposition [24].

5.2 Species Diffusion

In the work presented here, all the species are supposed to have the same diffusivity. As the main contribution to species diffusion is "turbulent diffusion", which is a convection term, it is a fair assumption. The "turbulent" Schmidt number does not differ from one specie to another. To be consistent with the finite volume element approach, all the properties of the fluid are supposed to be locally homogeneous and isotropically distributed [24].

5.3 Chemical Species Involved

It is assumed that the unburnt gases are only composed of fuel, molecular oxygen and nitrogen, carbon dioxide and water. Also it is assumed that the existence of other components in the fresh gases will add one transport equation per added component, which is not cost effective. The burnt gases are supposed to be composed of molecular and atomic oxygen, nitrogen, and hydrogen, carbon monoxide and dioxide, OH and nitrogen monoxide [24].

5.4 Burnt Gases Chemical Reactions

All the reactions computed in the burnt gases are supposed to be bulk reactions. That means that no local structure of the reaction zone is taken into account and that these reactions are only function of the mean local quantities computed in the burnt gases. The reactions are solved using conditioned burnt gases properties [24].

5.5 Chemical Equilibrium

A single step oxidation of methane with oxygen to form carbon dioxide and water vapor is considered. The following species has been used for methane combustion fuel O, O2, H_2, OH, CO, CO_2 [25].

$CH_4 + 2O_2 = CO_2 + 2H_2O$

6. Theoretical Analysis

A S.I. engine works on the principle of Otto Cycle. The theoretical analysis was done on the basis of equations obtained from Otto cycle and also from various other literatures.

For instance:

At site condition of engine:

Atmospheric pressure $P_1 = P_a = 1.01 \text{ x } 105$ bar

Suction temperature $T_1 = 25^{\circ} C = 298K$

$$P_1 V_1^{\gamma} = M_{ch}.\ R_{ch}.\ T_1$$

Where $M_{ch} = 1.00$ Kg.

$V_1 = 0.8736\ m^3$

Similarly, the other values that were obtained are shown in the table below:

Table 2 Result of Theoretical Analysis.

P_1 :	1.01 bar
T_1 :	298 K
V_1 :	0.8736 m^3
P_2 :	20.49430 bar
T_2 :	671.3996 K
V_2 :	0.0970 m^3
P_3 :	84.9205 bar
T_3 :	2782.0246 K
V_3 :	0. 0970 m^3
P_4 :	4.1850 bar
T_4 :	1233.91 K
V_4 :	0.8730 m^3
γ compression	1.37
γ expansion	1.37

7. Computational Analysis

The computational analysis was performed in FLUENT. The values that were obtained using the theoretical analysis were considered as boundary conditions.

7.1 Input Parameters

Following parameters are taken as input to fluent:

1. Valve opening and closing position, which includes Inlet valve opening position, Inlet valve closing position, Outlet valve opening position and Outlet valve closing position.
2. Pressure and Temperature at inlet and outlet.

Table 3 Properties of Methane Gas

Property	Value
Density	0.6679 kg/ m^3
C_P	2222 J/Kg-k
Thermal conductivity	0.0332 w/m-k
Viscosity	1.087e-05 Kg/m-s
Molecular weight	16.04303 Kg/Kg mol

Engine Parameter:

a. Crank shaft speed = 1500 rpm to 3000 rpm
b. Crank period = 720 deg
c. Crank angle step size = 0.5 deg
d. Piston stroke = 0.0495 m
e. Connecting rod length = 0.122 m

7.2 Output Obtained

Turbulent kinetic energy, turbulent intensity, turbulent flame speed, density and temperature at different region of cylinder are obtained from FLUENT which are presented in the results and discussion.

8. Results and Discussion

Simulation of single cylinder four stroke spark ignition engine for premixed combustion model has been done in FLUENT software by using methane gas as fuel. Following figure shows the different results obtained for equivalence ratio of 0.6 and engine speed of 2500 rpm at 419.5 degree of crank angle.

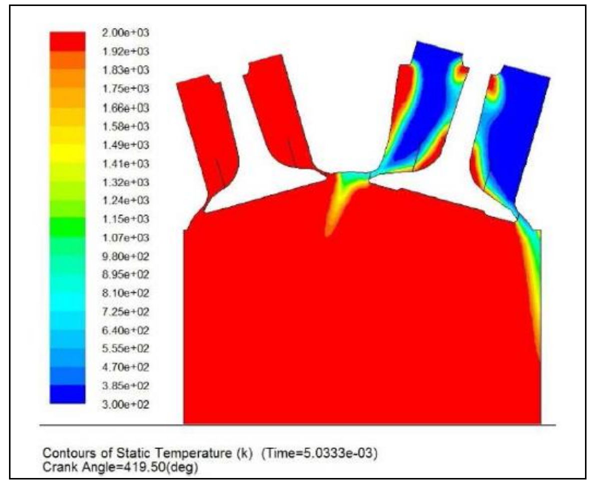

Fig.4 Contour of temperature (K) at 419.50° of crank Rotation.

As the spark ignites at 350° of crank rotation, the flame kernel starts developing which separate unburnt reactants and burnt reactants. As the combustion takes place, the temperature inside cylinder is increases from 300 K to 2000 K as shown in figure 4. Increase in temperature decrease the density of the mixture inside the cylinder. The density of the mixture decreases from 1.23 kg/m^3 to 0.184 kg/m^3 which is observed from figure 5.

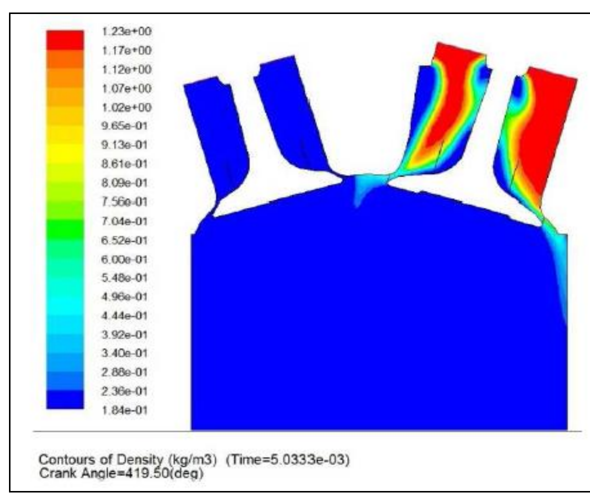

Fig.5 Contour of Density (Kg/m^3) at 419.50° of crank rotation

Turbulence is characterized in terms of turbulent kinetic energy and turbulent intensity. Figure 6 and 7 shows the result obtained of turbulent kinetic energy and turbulent intensity from the simulation.

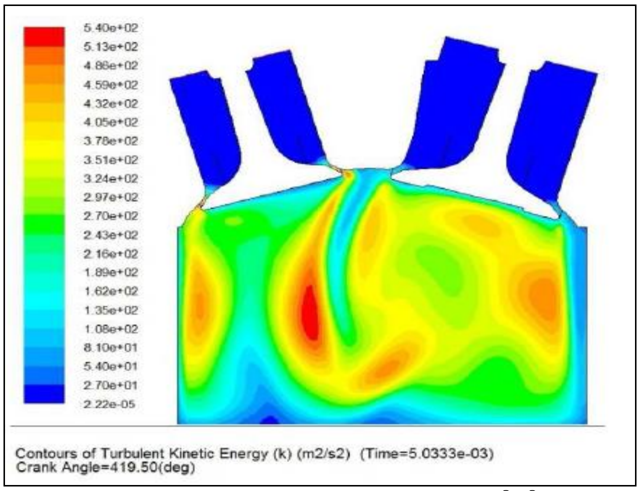

Fig.6 Contour of Turbulent Kinetic Energy (m^2/s^2) at 419.50° of crank rotation

The turbulent kinetic energy does not remain constant during the compression and power stroke. The maximum turbulent kinetic energy obtained is 540.268 m^2/s^2 ,that can be observed from contour of turbulent kinetic energy.

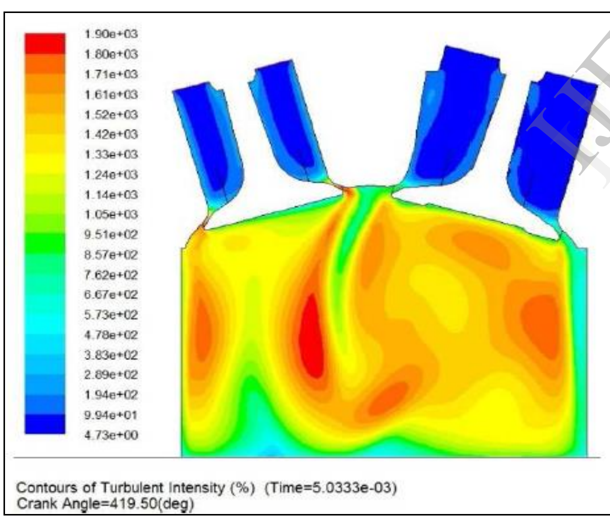

Fig.7 Contour Turbulent Intensity (%) at 419.50° of crank rotation

Turbulent intensity shows the turbulence level in the combustion chamber. Turbulent intensity increases with increase in turbulent kinetic energy. Turbulent intensity is varying in the range from 4.73% to 1897.83% as shown if figure 7.

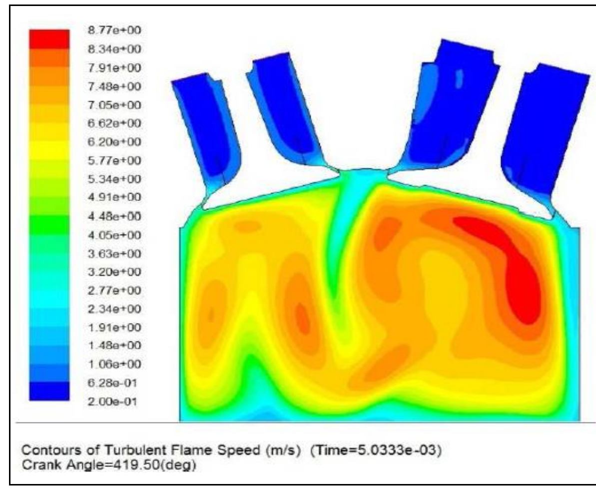

Fig.8 Contour of Turbulent flame speed (m/sec) at 419.50° of crank rotation.

The maximum turbulent flame speed obtained in this study is 8.76 m/sec. From the figure 8, it is observed that the turbulent flame speed is higher in the middle region than that at the wall. The turbulent flame speed is varying in the range from 0.2 m/sec to 8.76 m/sec.

9. Turbulent Flame Speed

A simulation is conducted on the engine for speed ranges from 1500 to 3000 rpm and fuel-air equivalence ratio 0.6 to 1.2 to find the effect of equivalence ratio and engine speed on propagation of turbulent flame speed. Table 4 shows the obtained values of turbulent flame speed at different equivalence ratio and engine speed using FLUENT software.

Table 4 Values of Turbulent Flame Speed

Equivalence Ratio	**Engine Speed (rpm)**	**Turbulent Flame Speed (m/sec)**
0.6	1500	7.1053
0.8	2000	7.2429
1.0	2500	7.3269
1.2	3000	8.03865

Figure 9 and 10 shows clearly the relation between the three parameters namely Equivalence Ratio, Engine Speed and Turbulent Flame Speed.

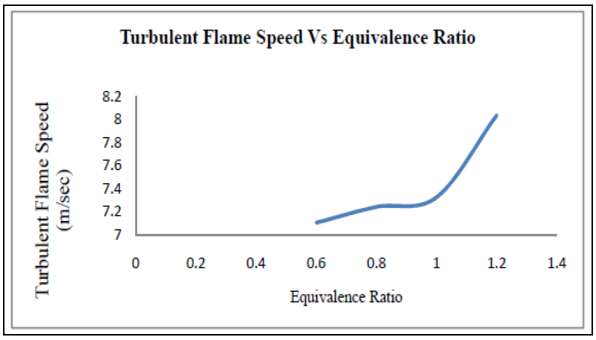

Fig.9 Turbulent flame speed (m/sec) Vs Φ

From the above graph it can be seen that the turbulent flame speed increases with the increase in equivalence ratio.

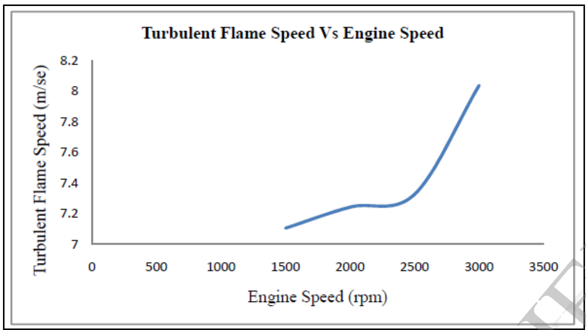

Fig.10 Turbulent flame speed (m/sec) Vs Engine speed (rpm)

From the above graph it can be seen that higher the engine speed, the greater the turbulence inside the cylinder due to which turbulent flame speed increase.

10. Conclusion

The following can be concluded from the results:

- ✓ Turbulent flame speed has increased from 7.1053 m/sec to 8.0386 m/sec for equivalence ratio of 0.6 to 1.2. Hence it is concluded that turbulent flame speed increases with increase in equivalence ratio.
- ✓ Turbulent flame speed has increased from 7.1053 m/sec to 8.0386 m/sec for engine speed of 1500 rpm to 3000 rpm. Hence it is concluded that turbulent flame speed increases with increase in engine speed.

The above results were validated with the work of K. A. Malik [3] and various other literatures.

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