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Combustion Modeling of SI Engine for Predicting Behavior of I.C. Engine: A Review

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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 analyzed. Combustion models in I.C Engines contains a set of equations governing the fluid mechanics, thermodynamic behavior of the engine working fluid as it passes through the cylinder of an operating engine. The objective of project work is to do combustion modeling of single cylinder spark ignition engine of bike by using computational fluid dynamics for predicting the turbulent flame speed at different equivalence ratio and engine speed. Premixed combustion model is to be considered in this study. The methane gas has been taken as fuel and single oxidation of methane with oxygen to form carbon dioxide and water vapor. Prediction of turbulent flame speed for premixed combustion is done in FLUENT software. Results show that turbulent flame speed increases for increment in equivalence ratio and engine speed.*

Keywords—*modeling of CI Engine, Physical data of SI Engine, Fluent Software for CFD, AUTO-CAD, Turbulent Flame speed, ANSYS, Equivalent Ratio, Engine Speed*

I. INTRODUCTION

Combustion model of engine is defined as a physically based description of the engine combustion process, which predicts the mass burning rate and the flame geometry as functions of engine design and operating variables. from experiments we can know that how much swirling and tumbling in-cylinder flows improve the combustion process and hence, it is necessary to understand (i) the actual processes by which these flows enhance the flame development and propagation processes, and also processes by which engine flames slow down and eventually are extinguished, The second major combustion modeling opportunity is in engine development and design. These activities need more complete analysis tools which predict the effect of the engine combustion process on engine performance more accurately, by including more of the relevant variables such as the fuel-air mixture preparation system characteristics. The geometric details of the combustion chamber include the major features of the inlet ports, valves, and the in-cylinder flow they produce, and the ignition system parameters. Especially, the intended use of the engine combustion model: the level of detail included, the accuracy, and the ease of use must all fit with the intended uses is important.

II-LITERATURE REVIEW

P. Boudier and S. Heniriot [5] developed a model for turbulent flame ignition and propagation in spark ignition engines. A model describing flame ignition in a premixed turbulent flow is coupled to a flamelet model for turbulent combustion to describe flame ignition and propagation in a spark ignition piston engine. During the first instants of ignition, a laminar ignition model (called LI) solves the one-dimensional spherical Navier-Stokes equations for a given chemical scheme to predict the radius of the first laminar flame kernel. In second phase, this kernel grows because of laminar effects but is also stretched by turbulent eddies. Later, a criterion based on the comparison of the laminar stretch (due to the laminar flame kernel growth) and of the turbulent stretch (generated by turbulent eddies) is used to initiate the computation of a fully turbulent combustion. During this turbulent propagation, an extended version of the coherent flame model (CFM) is used which includes a novel expression for turbulent flame stretch based on direct simulation results. The total model (LI-CFM) has been implemented in an improved version of KIVA including a k-e compressible model and a robust treatment for wall turbulence. The results are compared with experimental data obtained on a fully instrumented laboratory engine. The LI-CFM model correctly predicts flame ignition and growth for a set of typical engine cases with no parameter adjustments. Computed effects of changes in spark timing, total pressure or equivalence ratio agree well with experimental results.

K. A. Malik [6] developed a theoretical model for turbulent flame speed, based on turbulent transport process for spark ignition engine. The model was taken into account the effect of turbulence which 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.

Abu-Orf [7] developed a new reaction rate model and validated 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 behaves in a satisfactory manner in response to changes in fuel type, equivalence ratio, ignition timing, compression ratio and engine speed.

B. Fiorina and O. Gicquel [8] carried out premixed turbulent combustion modeling using tabulated detailed chemistry and probability density function. Tabulated chemistry and presumed probability density function (PDF) approaches are combined to perform RANS modeling of premixed turbulent combustion. The chemistry is tabulated from premixed flamelets with three independent parameters: the equivalence ratio of the mixture, the progress of reaction, and the specific enthalpy, to account for heat losses at walls. Mean quantities are estimated from presumed PDFs which is used to numerically predict a turbulent premixed flame diluted by hot burnt products at an equivalence ratio that differs from the main stream of reactants. The investigated flame, subjected to high velocity fluctuations, has a thickened-wrinkled structure. Comparisons of simulations with experimental measurements of mean velocity, temperature, and reactants are performed.

O. Colin and A. Benkenida [9] have done the 3-zones extended coherent flame model (ECFM3Z) for computing premixed/diffusion combustion. The ECFM model is based on a flame surface density equation which takes into account the wrinkling of the flame front surface by turbulent eddies and a conditioning averaging technique which allows precise reconstruction of local properties in fresh and burned gases even in the case of high levels of local fuel stratification. In order to adapt the model to unmixed combustion for diesel application, a description of the mixing state has been added. It is represented by three mixing zones: a pure fuel zone, a pure air plus possible residual gases zone and a mixed zone in which the ECFM combustion model is applied. A mixing model is presented which allows progressive mixing of the initially unmixed fuel and air. This new combustion model, called ECFM3Z (3-zones extended coherent flame model), can therefore be seen as a simplified CMC (conditional moment closure) type model where the mixture fraction space would be discretized by only three points. The conditioning technique is extended to the three mixing zones and allows to reconstruct, like in the ECFM model, the gas properties in the unburned and burned gases of the mixed zone. Application of the model to internal combustion engine calculations implies the necessity of auto-ignition modeling coupled to premixed and diffusion flames description. The premixed turbulent flame description is given by the ECFM, the diffusion flame is now accounted for thanks to the three zones mixing structure which represents phenomenological the diffusion of fuel and air towards the reactive layer, that is the mixed zone. The model is presented in all its details and its behavior is analyzed when the relative duration of injection and auto-ignition delay are varied in a direct injection diesel engine. It is shown that the model is able to reproduce the relative importance of auto-ignition and diffusion flame on the total heat release, depending on the engine operating point considered.

III-THEORETICAL CALCULATION

An internal combustion engine works on Otto cycle. It uses methane as a fuel, then compression ratio of the engine is 9 and equivalence ratio at full load is 0.6. Atmospheric temperature and pressure are 1.01 bar and 25°C respectively. The specific heat at constant volume is 800 J/Kg-K.

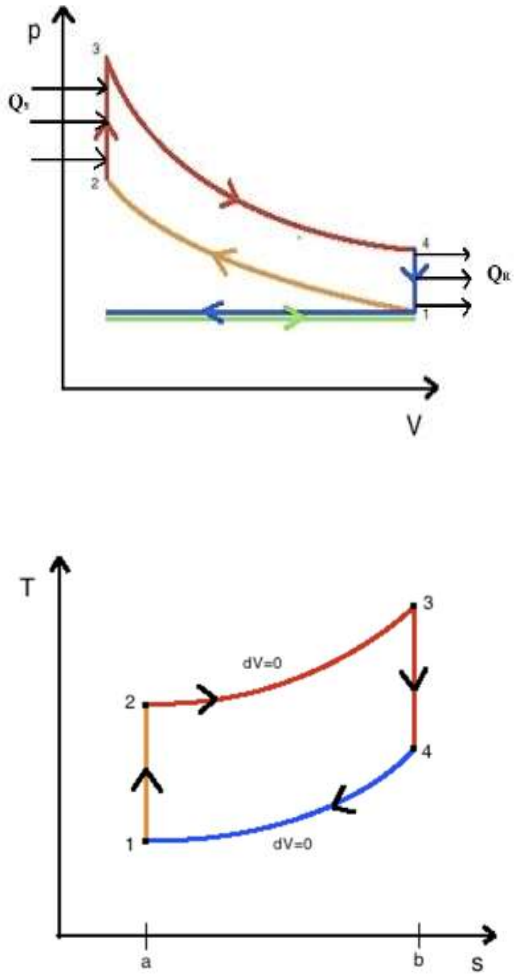

Figure:1- P-V diagram of ideal Otto cycle [1]

The cycle consists of two isentropic processes and two constant volume processes as shown in figure 5(a) and 5(b) on P-V and T-S diagrams. Process 1-2 is isentropic compression, the process 2-3 is heat addition at constant volume, process 3-4 is isentropic expansion, and the process 4-1 is heat rejection at constant volume.

Process 1-2: Isentropic compression: (Isentropic compression)

Process 2-3: Heat addition at constant volume

Process 3-4: isentropic expansion: (Isentropic expansion)

Process 4-1: Heat rejection at constant volume

Valve Timing Diagram

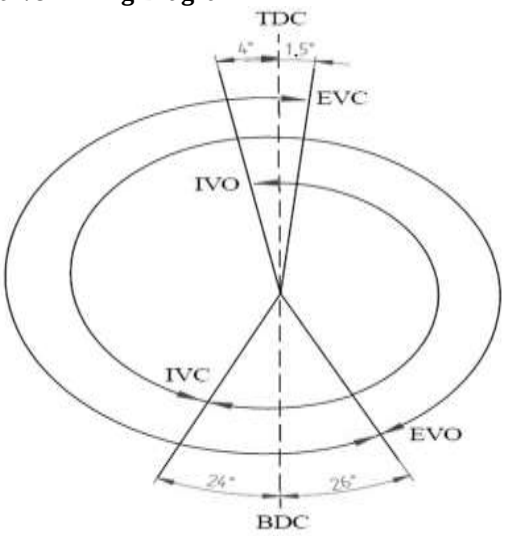

Figure:2- valve timing diagram [2]

The valve timing diagram shows the position of the crank angle when the various operations, i.e. suction, compression, expansion and exhaust begins and ends. The valve timing is the regulation of the position in the cycle at which the valves are set to open and close. Since the valves required a finite period of the time to open or close without abruptness, a slight lead time is necessary for proper operation. The design of valve operating cam provides for smooth transition from one position to the other.

Figure 2 shows valve timing diagram of splendor bike engine. The inlet valve opening occur 4o prior to the arrival of the position at TDC during the exhaust stroke. This is necessary to insure that the valve will be fully open and fresh charge starts to flow into the cylinder as soon as the piston starts to move down. If inlet valve is allowed to close at BDC, the cylinder would receive less charge than its capacity and the pressure of the charge at the end of suction stroke will be below atmosphere. To avoid this, the inlet valve is kept open for 24o rotation of the crank after the suction stroke. The kinetic energy of the charge packs more charge into the cylinder during this additional valve opening. Complete clearing of the exhaust is necessary to take in more charge. Earlier opening of the valve before reaching TDC facilitates the removal of the burnt gases. The kinetic energy of the fresh charge may also assist the removal of the burnt gases. It is obvious from the valve timing diagram that inlet and exhaust valves overlap for 5.5o of crank rotation; due to this overlap the burned gases to be sucked into intake manifold or fresh charge to escape through the exhaust valve. The ignition is takes place by producing spark at the end of compression, when piston is reaching 10o before TDC which burnt the charge instantaneously. However there is always time leg between the spark and ignition of the charge. The ignition starts some time after giving spark, therefore it is necessary to produce the spark before position reaches the TDC to obtain proper combustion without losses. The angle through which the spark is given is known as "ignition Advance" or "Angle of Advance"

For the motion of the valve a profile has to be prescribed. This profile can be written in *.txt format in any word processor. The extension of the document has to be renamed to *.prof and can then be loaded in FLUENT under the profile option in the define menu. An example of a profile is shown here.

(
(valve-movement)
(angle a1 a2 a3……..……an)
(lift x1 x2 x3…… xn)
)

The lift is defined in meters and the angle in degrees. The movement is linear interpolated between these n points.

Computational Modeling

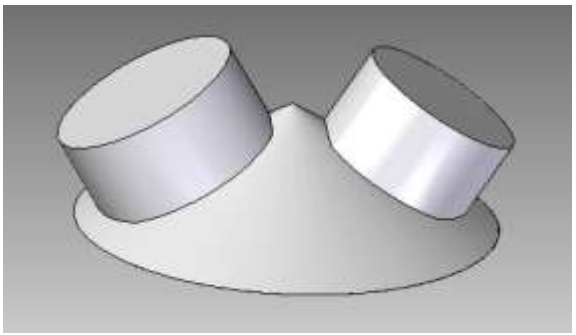

Figure:3(a)- Solid edge model created from real engine

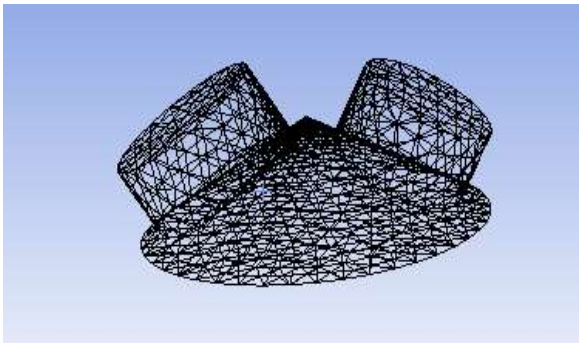

Figure:3(b)- Meshing of working fluid module

The three dimensional model is done in SOLID EDGE software using the original geometry of the engine of Splendor bike. The geometry formed includes the important details of the real engine. 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.

Mesh Motion

The mesh motion which is the basic for all the simulations is achieved in ANSYS mesh module.we The mesh created is based on the crank angle specified and effort is made to match the dynamic mesh generation to match to the actual cylinder movement. The valve profiles help to exactly replicate the actual valve motion in the computational environment. the mesh motion with respect to crank angle information.In the current study, the first stroke starts at 0o TDC (0o CA) and continues up to 180o ATDC (180o CA). The second stroke starts from 180o ATDC (180o CA) and continues up to TDC (360o CA) (see figure 3.5) where the combustion occurs. The third stroke and fourth are similar to first and second strokes respectively. The third stroke starts at TDC (360o CA) and continues up to 180o ATDC (540o CA) which is end of expansion stroke. The last stroke or the exhaust stroke will continue from 180o ATDC (540o CA) to 360o ATDC (720o CA

A Simplified Two-Zone Model of Engine Combustion

Simplified models are often used to gain understanding of certain aspects of combustion in IC engines. The simplest model for a spark ignition (SI) engine consists of two zones, one for the burned gases and one for the unburned gases. Such a model may be used to assess overall heat release and perhaps predict the onset of knocking when an empirical model for the turbulent burning rate is properly tuned. The turbulent flame is modeled by a spherical flame front with its center located at the spark. In a more general model, the turbulent flame front can be modeled by a wrinkled front as shown in Fig.3.

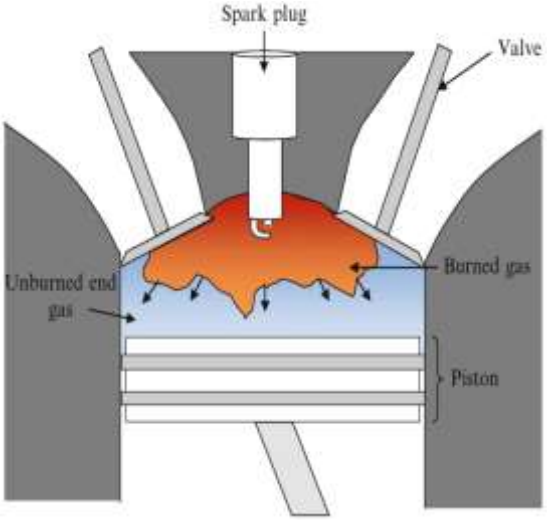

Figure:4- - A two-zone model for SI engine combustion with a turbulent flame front propagating from the burned zone into the unburned zone [3].

In most engines, experimental data indicate that the turbulent flame falls into the laminar flamelet regime. Under this regime, turbulent flame speed is reasonably correlated with laminar flame speed. For engineering purposes, the turbulent propagation flame velocity is represented by an empirical model that depends on several parameters [3]

$$\frac{U_t}{U_l} = f(u'/U_t\,, P/P_{m\,,}\theta_{ign})$$

Where U_t is the turbulent flame speed, U_l is the laminar flame speed, u' is the characteristic turbulent fluctuation velocity, P is cylinder pressure, Pm is the motoring pressure, θ_{ign} is the ignition timing in terms of CAD before TDC.

The governing equations for the two-zone model include those for energy conservation, mass conservation, and two ideal gas equations [3]

$$\frac{d(m_u u_u)}{dt} = h_u \frac{dm_u}{dt} - P\frac{dV_u}{dt} - \dot{q}_{u,L}$$

$$\frac{dm_{bu}u_{ub}}{dt} = h_b \frac{dm_b}{dt} - P\frac{dV_b}{dt} - \dot{q}_{b,L}$$

$$m_u + m_b = m$$

$$V_u + V_b = V$$

Where mu and mb denote the masses of unburned and burned mixtures respectively, hu and hb are the respective enthalpies, and Vu and Vb are the corresponding volumes. Heat transfer rates to engine walls, $\dot{q}_{u,L}$ and $\dot{q}_{b,L}$, are modeled by empirical correlations. The pressure is assumed to be uniform. Using the two ideal gas equations, we have

$$P = \frac{m_u R_{un} T_u}{V_u} = \frac{m_b R_{bn} T_b}{V_b}$$

The overall mass burning rate inside an IC engine is computed by

$$\frac{dm_b}{dt} = -\rho_u . A_f . U_t$$

Where ρu and Af are the unburned density and flame surface area respectively.

Modeling Premixed Combustion [4]

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. 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. The reason for this is that premixed combustion usually occurs as a thin, propagating flame that is stretched and contorted by turbulence. For subsonic flows, the overall rate of

propagation of the flame is determined by both the laminar flame speed and the turbulent eddies. The laminar flame speed is determined by the rate that species and heat diffuse upstream into the reactants and burn. To capture the laminar flame speed, the internal flame structure would need to be resolved, as well as the detailed chemical kinetics and molecular diffusion processes. Because practical laminar flame thicknesses are of the order of millimeters or smaller, resolution requirements are usually unaffordable.

The effect of turbulence is to wrinkle and stretch the propagating laminar flame sheet, increasing the sheet area and, in turn, the effective flame speed. The large turbulent eddies tend to wrinkle and corrugate the flame sheet, while the small turbulent eddies, if they are smaller than the laminar flame thickness, may penetrate the flame sheet and modify the laminar flame structure.

Non-premixed combustion, in comparison, can be greatly simplified to a mixing problem. 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.

In premixed flames, the fuel and oxidizer are intimately mixed before they enter the combustion device. Reaction then takes place in a combustion zone that separates unburnt reactants and burnt combustion products. Partially premixed flames exhibit the properties of both premixed and diffusion flames. They occur when an additional oxidizer or fuel stream enters a premixed system, or when a diffusion flame becomes lifted off the burner so that some premixing takes place prior to combustion.

Premixed Combustion Theory

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.

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 [4].

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

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

Reynolds averages denoted by overbears as well as the farve average such as $\bar{\rho}\,\tilde{c} = \overline{\rho c}$ are used where $c'' = c - \tilde{c}$ and $c' = c = \bar{c}$.

Farve averaging [5]

Let ∅ be any dependent variable. This variable can be decomposed into a mean part $\tilde{\emptyset}$ and a fluctuating part $\emptyset''$ using a density weighted average in the following way:

$$\emptyset = \tilde{\emptyset} + \emptyset''$$

$$\tilde{\emptyset} = \frac{\int_T^t \rho(t)\,\emptyset(t)dt}{\int_T^t \rho(t)dt} = \frac{\overline{\rho\emptyset}}{\bar{\rho}}$$

Where overbars (e.g $\overline{\rho\emptyset}$) denote averages using the Reynolds decomposition.

Auxiliary relation include

$$\overline{\rho\emptyset''} = 0$$

$$\overline{\rho\emptyset''} = \bar{\rho}\tilde{\emptyset} = \overline{\rho\emptyset}$$

Favre averaging is used in compressible flow to separate turbulent fluctuations from the mean-flow. In most cases it is not necessary to use Favre averaging though, since turbulent fluctuations most often do not lead to any significant fluctuations in density. In that case Reynolds averaging method can be used. Only in highly compressible flows and hypersonic flows it is necessary to perform the more complex Favre averaging.

Turbulent Flame Speed

The key to the premixed combustion model is the prediction of Ut, the turbulent fame speed normal to the mean surface of the flame. The turbulent flame speed is influenced by the following:

- 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 and stretching by large eddies, and flame thickening by small eddies.

The turbulent flame speed is computed by [4]

$$U_t = A(u')^{3/4}\, U_l^{1/2} \alpha^{-1/4} l_t^{1/4}$$

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

Where

A = model constant

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

Ul = laminar flame speed (m/s)

α = k/cp = molecular heat transfer coefficient of unburnt mixture (thermal difusivity) (m2/s)

lt = turbulence length scale (m)

Ƭt = lt / u1 turbulence time scale (s)

Ƭc = α / U2l = chemical time scale (s)

The turbulence length scale is computed from [4]

$$l_t = C_D \frac{(u')^3}{\epsilon}$$

Where ∈ is the turbulence dissipation rate.

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. The default value of 0.52 for A is recommended, and is suitable for most premixed flames. The default value of 0.37 for CD should also be suitable for most premixed flames.

Laminar Flame Speed

Laminar flame speed is a property of a combustible mixture. It is the speed at which an un-stretched laminar flame will propagate through a quiescent mixture of unburned reactants. The laminar flame speed (Ul in Equation 4.7) can be specified as constant, or as a user defined function. A third option appears for non-adiabatic premixed and partially remixed flames and is based on the correlation proposed by Meghalchi and Keck

$$U_l = U_{l,ref} \left(\frac{T_u}{T_{u,ref}}\right)^{\gamma} \left(\frac{\rho_u}{\rho_{u,ref}}\right)^{\beta}$$

In Equation, Tu and pu are the unburnt reactant temperature and pressure ahead of the flame, Tu,ref = 298K and pu,ref = 1atm.

The reference laminar flame speed, Ul,ref , is calculated from

$$U_{l,ref} = C_1 + C_2(\emptyset - C_3)^2$$

Where ∅ is the equivalence ratio ahead of the flame front, and C1, C2 and C3 are fuel specified constants. The exponents α and β are calculated from,

$$\gamma = 2.18 - 0.8(\emptyset - 1)$$

$$\beta = -0.16 + 0.22\,(\emptyset - 1)$$

Limitations of the Premixed Combustion Model [4]

The following limitations apply to the premixed combustion model:

- The premixed combustion model is not available with the density-based solver. So pressure-based solver should be taken in to use.
- The premixed combustion model is valid only for turbulent, subsonic flows. These types of flames are called deflagrations. Explosions, also called detonations, where the combustible mixture is ignited by the heat behind a shock wave, can be modeled with the finite-rate model using the density-based solver.
- The premixed combustion model cannot be used in conjunction with the pollutant (that is, soot and NOx) models. However, a perfectly premixed system can be modeled with the partially premixed model, which can be used with the pollutant models.
- It is not possible to use the premixed combustion model to simulate reacting discrete-phase particles, because these would result in a partially premixed system. Only inert particles can be used with the premixed combustion model.

CONCLUSION

Figure 5a shows graph of turbulent flame speed vs. equivalence ratio and is observed that for lean mixture at equivalence ratio 0.6 and 0.8, the turbulent flame speeds are 7.1053 m/sec and 7.2429 m/sec respectively which is less as compare to equivalence ration 1.0 of 7.3269, because for lean mixture less thermal energy produce results in decrease in flame temperature, which in turn decreases the flame speed. Maximum flame speed 8.03865 m/sec is obtained when the mixture is 20% richer than stoichiometric. Hence it is concluded that turbulent flame speed is increases with the increase in equivalence ratio.

Figure 5b shows the graph of turbulent flame speed vs. engine speed. It is observed that increase in turbulent flame speed increases the engine speed or vice versa. At 1500 rpm, the turbulent flame speed is 7.1053 m/sec, which is increase to 7.2429 m/sec and 7.3269 m/sec as the engine speed increases to 2000 rpm and 2500 rpm respectively. The maximum turbulent flame speed obtained is 8.03865 m/sec at 3000 rpm. Hence it is concluded that higher the engine speed, the greater the turbulence inside the cylinder due to which turbulent flame speed increase

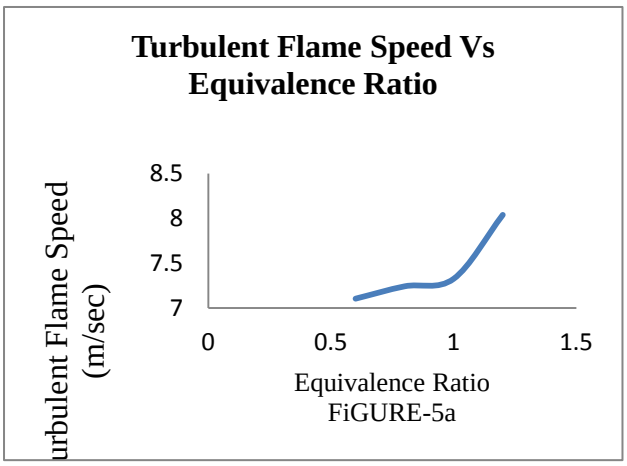

FiGURE-5a

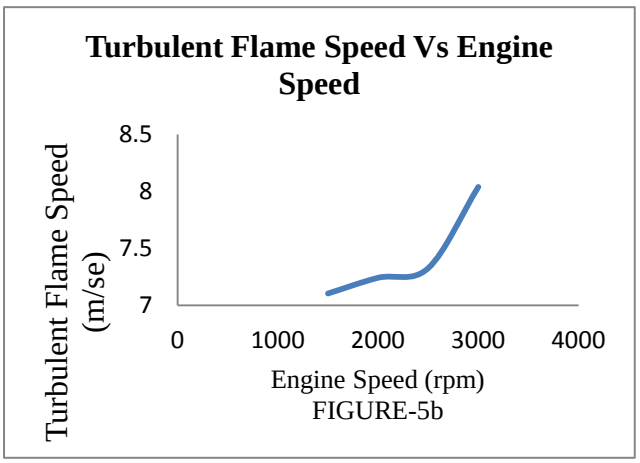

FIGURE-5b

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