Turbulent Flame Velocity Model for SI Engine

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Abstract - A new model for turbulent flame velocity of premixed flame in spark ignition engine for Iso-octane air mixture has been developed and validated for a wide range of engine operating parameters. The model developed is a zero dimensional thermodynamic model. The effects of engine speed (600-1160 rpm), equivalence ratio (0.7-1.1), unburnt mixture temperature (532-650 Rankine), compression ratio (5-8) and ignition timing (5-30 degree before top dead center) have been studied in detail. The comparison between theoretical and experimental burning velocity has been made for a wide range of engine operating parameters such as compression ratio, angle of spark, equivalence ratio, flame radius, engine speed and unburnt mixture temperature. The flame velocity obtained from the present model is in good agreement with the experimental and theoretical flame velocity of Cakir. The flame velocity computed by the present model is also in good agreement with the flame velocity calculated by Malik model.

Key words - *Flame velocity, Compression ratio, Equivalence ratio, Angle of spark.*

I. INTRODUCTION

The flame velocity is defined as the velocity of the unburned gases through the combustion wave in the direction normal to the wave surface. Practical flames are nearly all turbulent from that on a Bunsen burner to that in a spark ignition engine, there is no doubt that combustion in an internal combustion engine is always turbulent. Mathematical simulation of combustion in an engine cylinder can be of great assistance to the engine designer if they give a good representation of the engine system. 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. Babkin et al. [2] investigated experimentally and theoretically the effect of pressure on normal flame velocity by initial section method in a constant volume vessel. They found that the experimental and theoretical normal velocities were 62.2 and 65.6 cm/s for benzene, 55.4 and 58.2 cm/s for n-heptane, and 49.0 and 51.5 cm/s for iso-octane at 1 atmosphere pressure.

Cakir [3] theoretically formulated an expression for turbulent flame velocity. The flame velocity was determined experimentally using a closed vessel of a cylindrical shape with wall ignition. It was found that the highest flame speed measured for benzene was 23.5 m/s at 1160rpm. The change of compression ratio from 5 to 7 reduces the kernel development time for benzene and iso-octane by 50% and 28%. Blizard and Keck [4] developed a physical model of turbulent flame velocity for internal combustion engines and were tested experimentally. The model was based on mixing length theory. Experiments were conducted on simple cylinder research engine for the speed ranges (1000 to 3200 rpm), spark advances from 30 to 110 degrees and fuel air equivalence ratio from o.7 to 1.5 (Equivalence ratio is defined as the ratio of Actual Fuel Air ratio to Stoichiometric Fuel Air ratio, mathematically, $\phi = \frac{\left(F/A\right)_{Actual}}{\left(F/A\right)_{Stoichiometric}}$). Variation of cylinder pressure and position of the flame front as a function of crank angle were made simultaneously, for all operating conditions the experimental values were in good agreement with the predicted values of the model. Annand [5] developed a theoretical model of turbulent flame velocity for spark ignition engines based upon the analogy between molecular and turbulent transport processes. The turbulent flame velocity was related to the trapped conditions, residual turbulence at trapping and the engine mean position speed. Experimental measurements of flame speed obtained by ionization gap technique in a low cylinder engine were presented and compared with the model predictions.

Tabaczynski [6] formulated a turbulent flame propagation model that was dependent on the structure of the turbulent flow field and applied to combustion in a spark ignition engine for iso-octane air mixture. Comparison of the predicted versus experimental data shows good agreement for variation of equivalence ratio, dilution, speed, load and spark advance. Metaghalchi and Keck [7] have measured the laminar flame velocity of propane air mixture at high pressure and temperature in the range of 0.4 to 40 atmospheres and 298 to 750°K for equivalence ratio from 0.8 to 1.5. The measurements were made in a constant volume spherical combustion bomb at 500 °K. A thermodynamic analysis was used to calculate the laminar flame velocity. The measured values were correlated using both power and exponential expressions. Malik [8] 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 not only by the expanding flame front but also by the inlet valve geometry. The predicted turbulent flame speed values were compared with the experimental values in the speed ranges 600 to 1160 rpm and fuel air equivalence ratio from 0.8 to 1.25 and were found to be in good agreement. Abu-Orf [9] 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.

Jerzembeck and Peters [10] investigated experimentally the spherical expanding flames of n-heptane and iso-octane air mixtures using the constant volume bomb method. In order to investigate laminar flame velocities of combustible mixtures corresponding to mixtures in EGR (Exhaust Gas Recirculation) engines the fuel-air mixtures were diluted with nitrogen. The experiments were carried out in a preheated closed vessel. The expanded flames were tracked by a dark field Schlieren-Technique combined with a High-Speed-Camera-System. Experiments were carried out for fuel air mixtures with equivalence ratios from 0.7 to 1.2 and for fuel air mixtures diluted with nitrogen from 0.9 to 1.2. For all combustible mixtures the experiments were carried out under initial pressure conditions up to 20 bar and the initial temperature of 373 K. It was found that the flame velocity results were increases from the lean side to the stoichiometric conditions and they decrease when the mixture become rich. Laminar flame velocities of n-heptane air mixtures were larger than the corresponding iso-octane mixtures, but they were found smaller than the determined numerical results, within approx. an offset of 4.6 cm/s at 10 bar initial pressure and approx. 3-4 cm/s at 20 bar initial pressure. For the nitrogen diluted n-heptane air mixtures the experimental results were smaller than the numerical results within an offset of approx. 3 cm/s.

Jerzembeck [11] reported the laminar flame velocity results of n-heptane and isooctane air mixtures with two measurement techniques at 10 bar and 20 bar initial pressures for equivalence ratios from 0.7 to 1.2 and at 373 K. In the first approach, the spherical flame fronts were tracked using the captured pressure history of the fresh mixture during combustion. In the second approach, the flame was tracked by an optical Schlieren-Measurement-Technique using a Helium-Neon laser as a light source and a high speed camera to capture the flame photographs. The relative difference was approximately 2-5 cm/s between the results of these two measurement techniques. Optically determined results of laminar flame velocities at 10 bar initial pressure were within 5.5 cm/s underprediction, compared with the numerically determined ones, whereas the optically results at 20 bar initial pressure were within 4-5 cm/s agreement with the numerical predictions. The results determined from the pressure history data is within 6.5-7.5 cm/s underpredicted, compared to the numerical calculations for mixtures at 10 bar initial pressure. At 20 bar initial pressure, the “pressure”-results were in pretty good agreement within 4 cm/s to the numerical predictions. Abdelghaffar [12] used a zero-dimensional multi-zones phenomenological model to study the performance characteristics and NOx and CO emissions of a four-stroke spark ignition engine (SI) fueled by hydrogen, iso-octane, gasoline and methane. The study shows that the SI engine fueled by hydrogen produces lowers brake-power, brake-specific-fuel-consumption (bsfc) and brake thermal efficiency compared with iso-octane, gasoline, and methane fuels. Supercharging was found to be a more effective method of increasing the brake-power of the hydrogen engine rather than increasing the engine compression ratio of the engine.

The present study reported a new mathematical model of turbulent flame velocity for spark ignition engine. The various engine parameters that are considered for obtaining the flame velocity are engine speed (N) range 600 - 1160 rpm, compression ratio (CR) 5 - 8, angle of spark (θ_{ig}) 5 - 30 degree before top dead center, equivalence ratio 0.7 - 1.1 and unburned mixture temperature (T_{u0}) 532 - 650 Rankine.

II. MATHEMATICAL MODEL

Based upon the methodology of Lewis and Von Elbe [13, 14] a mathematical model of flame velocity for spherical flames in a spherical constant volume vessel with wall ignition has been developed. The model represents the actual flame propagation pattern rather than spherical shape. If during the travel of the flame through the vessel, the burned gas has not expanded, then an element dri would present the thickness of a shell at a temperature Ti and pressure Pi about to be traversed by the wave in the time element dt.

The volume corresponding to the initial pressure P_i and temperature T_i would be

$$V = 2\pi\, r_i\, h\, d r_i \qquad (1)$$

After expansion the radius of the shell increased from r_i to r_b, the pressure to P, and the temperature of the shell to T_u. The actual volume after expansion is given as,

$$V = 2\pi r_i h \left[\frac{T_u}{Ti} \times \frac{P_i}{P} \right] dr_i \qquad (2)$$

Since,

$$\frac{T_u}{T_i} = \left(\frac{P}{P_i} \right)^{\left(\gamma_u - 1 \middle/ \gamma_u \right)} \qquad (3)$$

From fig. 1

Volume 1 = h.A_1

$$V_1 = h\left[r_b^2.\alpha - lX \right]$$

Where the angle α is in radian,

$$V_1 = h\left[r_b^2 \alpha - X\sqrt{r_b^2 - X^2} \right] \qquad (4)$$

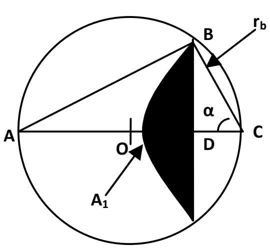

Fig. 1

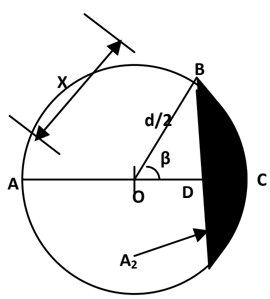

Fig. 2

Fig. 1 and Fig. 2 show areas and consequently volumes swept by flame front

Volume 2 = h.A_2

$$V_2 = h\left[\left(\tfrac{d}{2}\right)^2 \beta - 2\left(\tfrac{1}{2}\right)\left(\tfrac{d}{2} - X\right) l \right]$$

Where angle β is in radian,

$$V_2 = h.\left[\frac{d^2}{4}\beta - 2\sqrt{r_b^2 - X^2} + X\sqrt{r_b^2 - X^2} \right] \qquad (5)$$

Total volume

$$V = V_1 + V_2 \qquad (6)$$

Total burnt volume in terms of flame velocity is given by

$$V = h\left[r_b^2.\alpha + \frac{d^2}{4}\beta - \frac{d}{2}\sqrt{r_b^2 - X^2} \right] S_b.dt \qquad (7)$$

Also,

$$S_b.dt[r_b^2.a + \frac{d^2}{4}\beta - \frac{d}{2}\sqrt{r_b^2 - X^2}] =$$
$$2\pi r_i dr_i \left[\frac{T_u}{T_i} \times \frac{P_i}{P} \right] h$$

And

$$S_b = \frac{2\pi\, r_i\, h \times \left[\frac{T_u}{T_i} \times \frac{P_i}{P} \right] \left[\frac{dr_i}{dt} \right]}{\left[r_b^2\, .a + \frac{d^2}{4}\beta - \frac{d}{2}\sqrt{r_b^2 - X^2} \right]} \qquad (8)$$

Fraction of the charge burnt, from volume ratio is given by

$$n = \left[\frac{\dfrac{\pi}{4\, r_i^2\, h}}{\dfrac{\pi}{4\,(\frac{d}{2})^2\, .\, h}} \right] \qquad (9)$$

or

$$\frac{r_i}{d} = \frac{1}{2}.n^{\frac{1}{2}} \qquad (10)$$

And the flame velocity is given as,

$$S_b = \frac{2\pi r_i^2\, .h.\left(\frac{T_u}{T_i}\right)\left(\frac{P_i}{P}\right)\left(\frac{1}{8}\right)\left(\frac{d^2}{r_i}\right)\left(\frac{dn}{dt}\right)}{\left[r_b^2\, .\alpha + \frac{d^2}{2}\beta - \frac{d}{2}\sqrt{r_b^2 - X^2} \right]} \qquad (11)$$

$$S_b = \frac{\pi}{4}\left(\frac{P_i}{P}\right)^{-1/\gamma_u} .h\left[\frac{d^2}{r_b^2.\alpha + \left(\frac{d}{2}\right)^2 \beta - \frac{d}{2}\sqrt{r_b^2 - X^2}} \right] \frac{dn}{dt}$$

(12)

$$\cos\ \alpha = \frac{r_b}{d},\ \alpha = \cos^{-1} \varepsilon$$

Where

$$\varepsilon = \frac{r_b}{d},\ \sin\beta = 2\varepsilon.\sin\alpha,\ X = r_b \cos\ \alpha$$

$$S_b = \frac{\pi}{4}\left[\frac{P_i}{P}\right]^{-1/\gamma_u} *$$

$$h\left[\frac{d^2}{r_b^2\cos^{-1}(\varepsilon)+\frac{d}{2}\cos^{-1}\left(\pm\left(1-2\varepsilon^2\right)\right)-\frac{d}{2}r_b\sqrt{1-\varepsilon^2}}\right]\frac{dn}{dt} \tag{13}$$

Equation (13) is numerically solved to determine flame velocity for various parameters such as P, r_b, n, and h.

Where

$$r_b = f(\theta)$$

$$\alpha = f(r_b)$$

$$\beta = f(r_b)$$

$$X = f(r_b)$$

The various equations used to determine the burning velocity are as follows,

Cylinder volume at each crank angle is written as,

$$V_\theta = V_t\left\{\frac{CR-1}{CR} * \left[\frac{1}{2}(1-\cos\theta)+\frac{1}{2}\frac{L}{r}\left(1-\sqrt{1-\left(\frac{r}{L}\right)^2\sin^2\theta}\right)\right]+\frac{1}{CR}\right\} \tag{14}$$

Where $L_c = L/r$

Change in volume of cylinder with change in crank angle is given by,

$$DV_\theta = \frac{dV_\theta}{d\theta} = \frac{V_t}{2}\left[\frac{CR-1}{CR.\sin\theta}\right] * \left[1+\left(\frac{1}{L_c}\right)\left(1-\frac{1}{L_c}\right)\times\left(\sin^2\theta\right)^{\frac{-1}{2}}.\cos\theta\right] \tag{15}$$

Fraction of the charge burnt at given angle is given as

$$n = \frac{1}{2}\left[1-\cos\left(\pi\frac{\theta-\theta_{ig}}{\Delta\theta_c}\right)\right] \tag{16}$$

Equation for calculating the fraction of the charge burnt with change in crank angle is written as,

$$Dn = \frac{dn}{d\theta} = \frac{1}{2}\frac{\pi}{\Delta\theta_c}.\sin\pi\left[\frac{\theta-\theta_{ig}}{\Delta\theta_c}\right] \tag{17}$$

Equation for calculating the initial clearance height,

$$H_o = \frac{S}{CR-1} \tag{18}$$

Total volume of cylinder is given as,

$$V_t = \frac{\pi}{4}d^2\left(H_o - S\right) \tag{19}$$

Total mass of unburnt gas is written as,

$$m_u = P_m V_t\left(R_u T_i\right) \tag{20}$$

Equation for calculating the unburnt compression temperature of air fuel mixture,

$$T_{uc} = T_i\left[\frac{P_c}{P_m}\right]^{\frac{\gamma_u-1}{\gamma_u}} \tag{21}$$

The equation is used to calculate the compression pressure,

$$P_c = P_m\left[\frac{V_t}{V_\theta}\right]^{\gamma_u} \tag{22}$$

For calculating the initial pressure inside the cylinder,

$$P_i = P_m\left[\frac{V_t}{V_{\theta s}}\right]^{\gamma_u} \tag{23}$$

This equation is used to calculate the change in pressure with change in crank angle,

$$DP = \frac{dP}{d\theta} = \frac{C_1 - C_2 + C_3}{C_4}\times\left(\Delta\theta\right) \tag{24}$$

Where $C_1, C_2, C_3 and C_4$ are given as,

$$C_1 = T_m.\frac{dn}{d\theta}.\left[C_{\gamma_u}.T_i\left(\frac{P}{P_i}\right)^{\gamma_u-1/\gamma_u}\right] * \left(\gamma_b-\gamma_u\right).\left(A_b-A_u\right)\left(\gamma_b-1\right) \tag{25}$$

$$C_2 = \gamma_b.P.\frac{dV_\theta}{d\theta} \tag{26}$$

$$C_3 = \left(\gamma_b - 1\right).Q \tag{27}$$

$$C_4 = V_\theta + V_{\theta s}\left[\frac{\gamma_b-\gamma_u}{\gamma_u}\left(\frac{P}{P_i}\right)^{-1/\gamma_u}\left(1-A\right)\right] \tag{28}$$

Equation used to calculate the variable pressure,

$$P = P + DP \qquad (29)$$

These are the equations needed for calculating the burning velocity for a single cylinder spark ignition engine using iso-octane as a fuel with wall ignition.

III. RESULTS AND DISCUSSION

The flame velocity obtained from the present model with various engine parameters such as engine speed, compression ratio, equivalence ratio and angle of spark etc. are presented and compared with the data available in the literature.

The comparison between calculated and experimental flame velocity with respect to compression ratio for engine speed of 600 rpm and 1160 rpm is shown in Fig. 3. At the engine speed of 600 rpm the theoretical flame velocity obtained from present model are in good agreement with the experimental and theoretical flame velocity of Cakir [3] and Malik [8] for a range of compression ratio. As the engine speed increases up to 1160 rpm the present flame velocity deviates at higher compression ratios, although the present model develops the results for burning velocity up to a compression ratio of 8.

Fig. 4 shows the comparison between calculated and experimental burning velocities with respect to compression ratio for engine speed of 1160 rpm, ignition timing 26° BTDC and compression ratio of 5. It is depicted from Fig. 5 that the present model has lower burning velocity from Cakir experimental [3] at low equivalence ratios. As the equivalence ratio increases up to a certain value the flame velocity reaches to its maximum value, and then decreases for higher equivalence ratios. The present flame velocity is higher than the Cakir experimental [3] with in the equivalence ratio range of 0.875-1.0.

Fig. 5 shows the effect of engine speed for different equivalence ratios on flame velocity. It is seen from Fig. 5 that the flame velocity increases as the engine speed increases. In case of equivalence ratio of 1.1 the trend of flame velocity is different, the flame velocity continuously decreases with an increase in engine speed. It may be due to the fact that the combustion is not complete at higher equivalence ratio because the mixture is too rich.

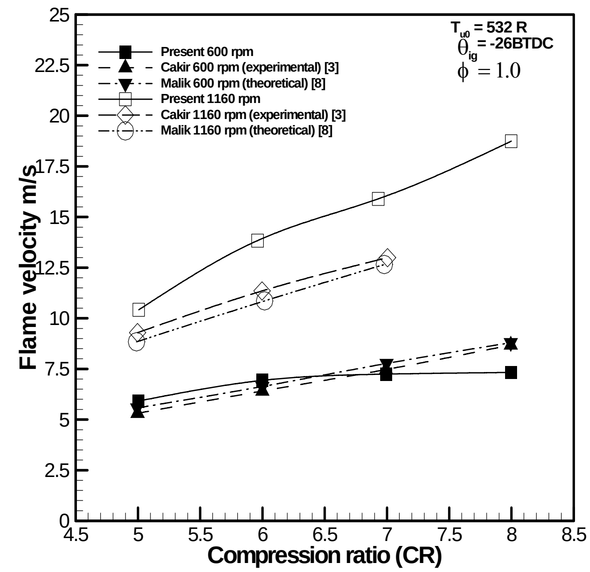

Fig. 3 shows the comparison between calculated and experimental flame velocities with respect to compression ratio at unburned mixture temperature 532 R, angle of spark -26 BTDC and equivalence ratio1.0

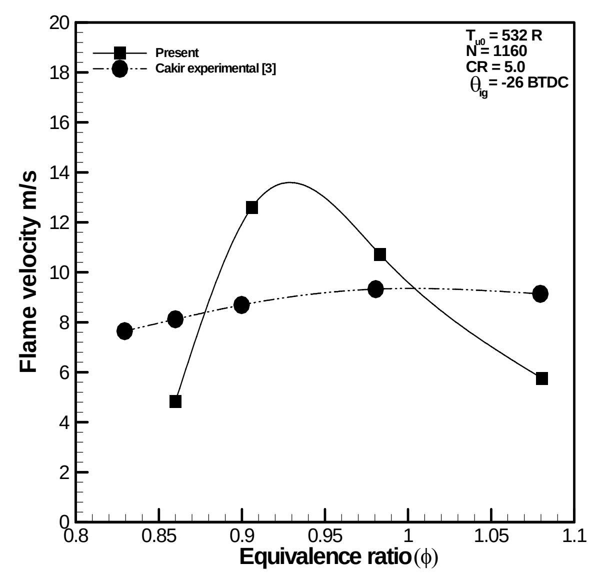

Fig. 4 shows the comparison between calculated and experimental flame velocities with respect to equivalence ratio at engine speed 1160 rpm, unburned mixture temperature 532 R, angle of spark -26 BTDC and compression ratio 5.0

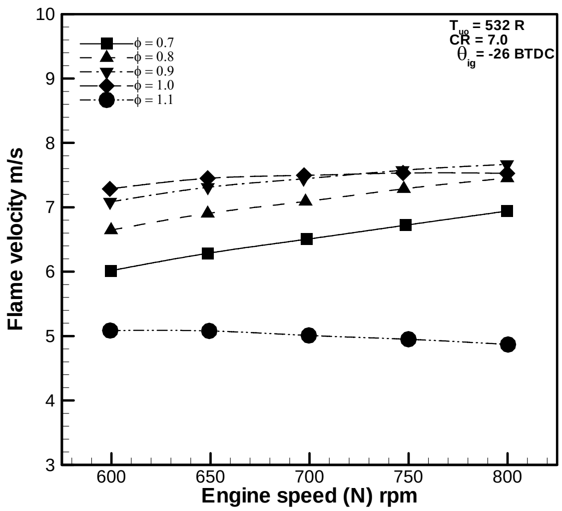

Fig. 5 shows the effect of engine speed and equivalence ratio on flame velocity at unburned mixture temperature 532 R, angle of spark -26 BTDC and compression ratio 7.0

Fig. 6 and Fig. 7 show the effect of engine speed for different equivalence ratio on flame velocity for unburned mixture temperature of 600 and 650 R. It is depicted from figure that the flame velocity increases with an increase in engine speed. For equivalence ratio of 1.1 the flame velocity decreases with an increase in engine speed, it is due to the fact that incomplete combustion occurs at higher equivalence ratios.

Fig. 8 shows the effect of equivalence ratio and compression ratio on flame velocity. It is seen from Fig. 8 that the flame velocity increases as the equivalence ratio increases up to a certain maximum value and decreases thereafter. At a fixed equivalence ratio the flame velocity increases as the compression ratio increases.

Fig. 9 shows the effect of compression ratio and engine speed on flame velocity. It is depicted from Fig. 9 that the flame velocity increases as the compression ratio increases. It is seen from Fig. 9 that at lower compression ratios the flame velocity is minimum for higher engine speed but at high compression ratio more than 6.5 the flame velocity reaches maximum for higher engine speed.

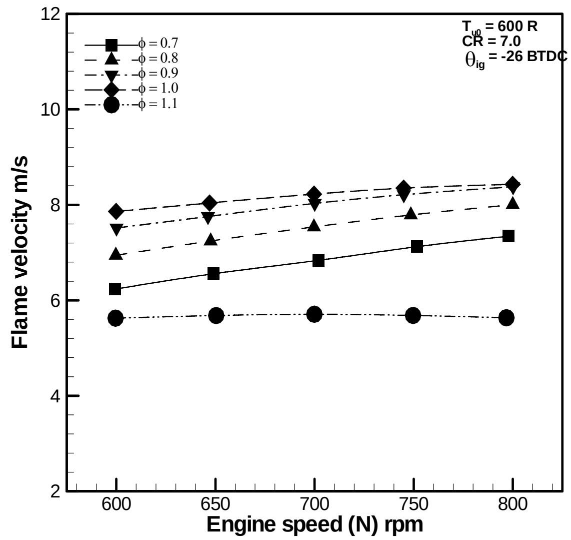

Fig. 6 shows the effect of engine speed and equivalence ratio on flame velocity at unburned mixture temperature 600 R, angle of spark -26 BTDC and compression ratio 7.0

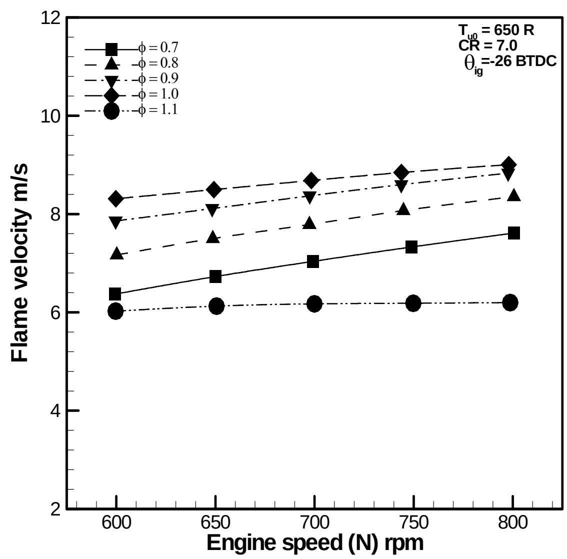

Fig. 7 shows the effect of engine speed and equivalence ratio on flame velocity at unburned mixture temperature 650 R, angle of spark -26 BTDC and compression ratio 7.0

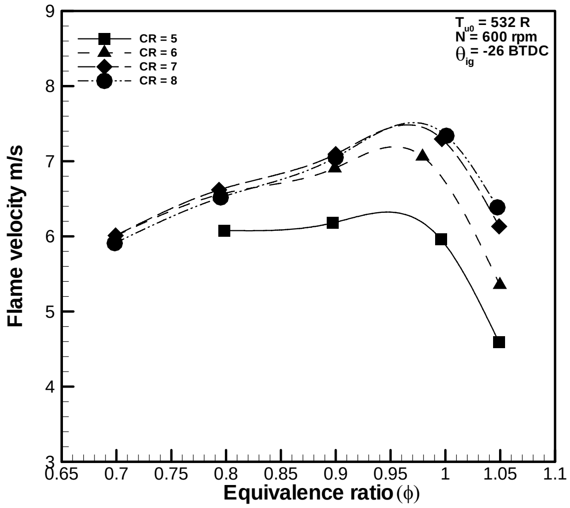

Fig. 8 shows the effect of equivalence ratio and compression ratio on flame velocity at unburned mixture temperature 532 R, angle of spark -26 BTDC and engine speed 600 rpm

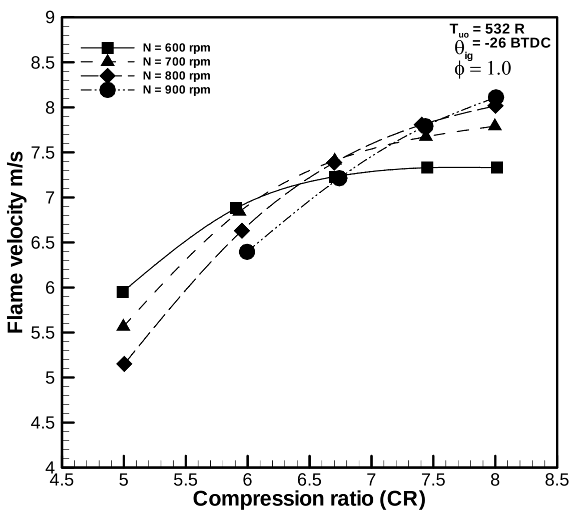

Fig. 9 shows the effect of compression ratio and engine speed on flame velocity at unburned mixture temperature 532 R, angle of spark -26 BTDC and equivalence ratio 1.0

Fig. 10 shows the effect of angle of spark and equivalence ratio on flame velocity. Fig. 10 shows that as the angle of spark increases the flame velocity increases up to a certain value and then decreases, it is also seen from Fig. 10 that higher equivalence ratio has higher flame velocities. In case of equivalence ratio 1.1the flame velocity increases and then decreases as the angle of spark increases, it has lower value because of incomplete combustion.

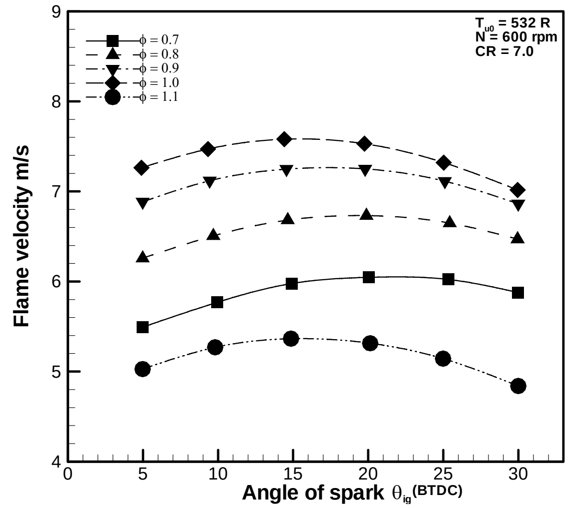

Fig. 10 shows the effect of angle of spark and equivalence ratio on flame velocity at unburned mixture temperature 532 R, engine speed 600 rpm and compression ratio 7.0

IV. CONCLUSION

In the present work a mathematical model of flame velocity for iso-octane air mixture has been developed in terms of fraction of the charge burnt, flame radius, equivalence ratio, clearance height and pressure ratio. The effects of engine speed, equivalence ratio, unburnt mixture temperature, compression ratio and ignition timing have been studied. The present model for turbulent flame velocity is in good agreement with the experimental and theoretical results of results of Cakir [3]. The present model requires further validation for a wide range of engine operating speeds, compression ratios, and angle of spark, so as to give actual results for flame velocity inside the engine cylinder.

V. NOMENCLATURE

d	Diameter of cylindrical vessel
h	Clearance height of the vessel
V_t	Total volume of engine cylinder
r_b	Burnt radius
CR	Compression ratio
θ	Angle in radian
θ_{ig}	Angle of spark
$\Delta\theta_c$	Combustion duration
L_c	Length of connecting rod

S Stroke length

P_m Manifold pressure

T_i Initial temperature

P_c Compression pressure

R_u Unburnt gas constant

$V_{\theta s}$ Volume at spark

Q Heat transfer rate

$\gamma_u \,\&\, \gamma_b = \dfrac{C_p}{C_v}$ Specific heat ratio (unburnt & burnt gas)

m_u Mass of unburnt mixture

A_b, A_u, A Are constants

ϕ Equivalence ratio

T_u & T_i Unburnt and initial temperature

n Fraction of charge burnt

P_i & P Initial and variable pressure

r_i Radius at the time of ignition

H_o Initial clearance height

ρ / ρ_0 Density ratio

T_{uc} Unburnt compression temperature

VI. ENGINE SPECIFICATION

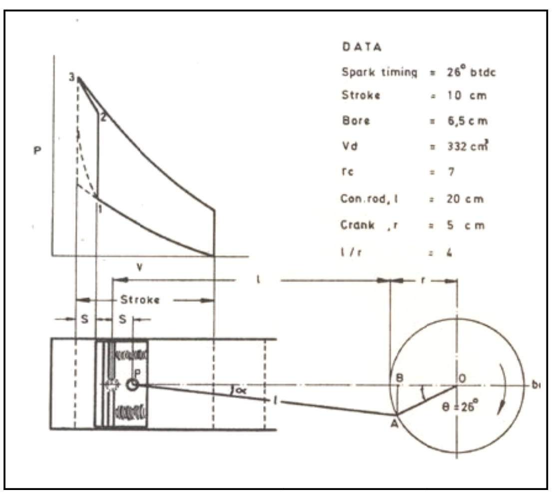

Fig. 11 : shows the volumes at different points on the cycle.

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