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MODELLING OF COMBUSTION PROCESS IN A SPARK IGNITED HYDROGEN ENGINE

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Abstract—This paper reports the relative merits of the Eddy Entrainment model over the Reynolds Parameter model in describing the combustion process in a spark ignition engine using hydrogen as fuel. The relative performance of each model is analysed with reference to the experimental data obtained from a single-cylinder engine. From the present investigation and in comparison of the performance of these models with other fuels it is found that both the models are capable of satisfactorily predicting the performance of hydrogen fuel engines due to (a) higher laminar flame speeds and (b) small quench distance.

INTRODUCTION

With the imposition of stringent emission standards along with the decreasing availability of petroleum products, it is imperative that a search for low polluting alternative fuels is made, even though with current economics hydrogen would be a costly automotive fuel. Based on long-term considerations, hydrogen's relative cost standing may improve considerably [1]. In this context, alcohols could be viewed as immediate alternate fuels. When a cheaper method of hydrogen manufacture is available it could be used for aircraft, marine vessels, rail and guide-way systems and public service/commercial vehicles.

When one tries to use an alternative fuel it is essential that the combustion and emission characteristics of such fuels are studied in greater detail. However, a systematic experimental study will be time-consuming and costly. Hence, engine simulation using digital computers has become quite popular. But, turbulent combustion is a field which is quite complicated to understand and is not easily amenable to mathematical analysis. Hence modelling of such a process is necessary to understand the combustion and pollutant formation. Many researchers [2–4] have suggested different models. However relative performance of different models has not been compared. It will be worthwhile to make such a comparison since it will provide considerable information for the development and evaluation of new engine designs.

There are two basic approaches for the modelling of the combustion process. One approach is to assume the combustion to proceed with a laminar burning velocity and superimpose the turbulent component by a suitable model to determine the apparent burning velocity. This is called the Reynolds Parameter model developed at the Indian Institute of Technology, Madras [4]. The second approach is to assume that the turbulence is due to the jet nature of the flow which creates eddies and these eddies are responsible for flame propagation. This is called the Eddy Entrainment model developed at the Massachusetts Institute of Technology [5]. The details are given in the following paragraphs.

REYNOLDS PARAMETER MODEL

The basic concepts of this model are described in detail elsewhere [4], however, a brief description is given here. In this model a spherical flame front moving spherically outward from the spark plug is assumed and this flame is wrinkled due to turbulent fluctuation. These turbulent fluctuations occur all along the flame front and are a function of intake jet velocity, density and viscosity. The combustion in an engine is a closed vessel combustion and hence the flame propagation rate is further modified due to the expansion effect of the burning gases. To account for this a flame transportation coefficient, C_T, has been included. C_T is calculated from the pressure-burned volume history for each small interval of time during combustion and is used for the subsequent interval. From these, the instantaneous apparent velocity U_{app} can be written as

$$U_{app} = C_T[U_L^2 + C_1(Re)^{C}_{2}]^{1/2} \tag{1}$$

where (Re) is the Reynolds parameter given by

$$(Re) = U_i D\rho/T_{av}^{0.67} \tag{2}$$

U_L is the laminar burning velocity
U_j is the induction jet velocity
D is the cylinder bore diameter
ρ is the density
T_{av} is the average temperature.

C_1 and C_2 are constants dependent only on the engine geometry and independent of operating conditions and fuel. Hence C_1 and C_2 can be determined from known pressure crank angle histories for a given fuel and operating conditions and these could be used for predicting the performance with different fuels and operating conditions. Once the apparent burning velocity is determined, the burning rates can be calculated.

EDDY ENTRAINMENT MODEL

Before today no satisfactory equation has been developed to describe turbulent combustion in a spark ignition engine. However, empirical correlations [6] for the mass fraction of the gas burned have been formulated as a function of spark advance, induction angle and apparent burning angle. In this model it is assumed that the jet nature of the intake process creates eddies of small-scale turbulence that persist during the combustion phase [7, 8].

At the time of ignition if it could be assumed that a spatially homogeneous volume of persisting eddies exist, then the combustion process can be viewed as a propagation of flame front with finite thickness through the combustible mixture at a speed determined by the rate at which eddies are entrained (U_e). This eddy entrainment speed is correlated with inlet gas speed as

$$U_e = \varepsilon[2D_V V_{LI}/(D/2)^2]\, NS_I \tag{3}$$

where

- ε is the volumetric efficiency
- D_V is the inlet valve diameter
- V_{LI} is the inlet valve lift
- D is the cylinder diameter
- N is the engine speed and
- S_I is the stroke length.

In the model it is assumed that the entrained eddies are immediately ignited due to diffusive transport of radicals such as H, OH and O between adjacent eddies and then burned at laminar flame speed U_L in a characteristic time

$$\tau = l_e/U_L \tag{4}$$

where l_e is the characteristic eddy radius given by 0.17 $(h_s/h_0)V_{LI}$ and h_s/h_0 is the reciprocal of the compression ratio at ignition. Significant engine parameters like speed, bore, stroke combustion chamber geometry, size and spark advance are incorporated in the characteristic eddy radius [5]. The laminar burning velocity is taken as a function of air fuel ratio, fuel type, residual fraction, inlet mixture density and is calculated from Semonov's equation [4].

Using the above concepts a set of three coupled integro-differential equations, which take into account mass fraction burned, temperature, area and volume of the flame are derived [5]. From the solution of the above equations the dimensionless flame radius, r_f, is approximated by the equation

$$r_f = [\theta - \theta_s - \theta_d(1 - e^{-(\theta - \theta_s/\theta_d)})]/\theta_b \tag{5}$$

where

- θ is any crank angle
- θ_b is the apparent burning angle
- θ_d is the induction angle
- θ_s is the spark angle.

During the combustion phase, when the change in the cylinder volume is relatively gradual, the mass fraction of the mixture burned is approximated by the following equations

$$X = \frac{2Ue}{\omega D}\frac{\theta_b \ln(y)}{(1 - 1/y_1)} \quad \text{when } \theta_\tau < \theta_d < \theta_t \tag{6}$$

where $\delta_\tau = \omega\tau$, $\theta_\tau = \theta - \theta_s$ and ω is angular speed and y_1 is the value of y for $X = 1$ and y_1, is given by

$$y_1 = [(b_0 + 1)/(\delta + 1)]^{1/\gamma} \tag{7}$$

where

$$b_0 = \frac{h_{fu} - h_{fb}}{C_{vu}T_0} \tag{8}$$

h_{fu} and h_{fb} are the specific enthalpies of burned and unburned mixtures.

$$\delta = \frac{\gamma_a - \gamma_b}{\gamma_b - 1} \tag{9}$$

$$\theta_b = \frac{\omega D}{2ue}\frac{1 - 1/y_1}{\ln(y_1)} \tag{10}$$

$$\theta_d = \frac{\omega 1e}{U_L}\frac{\gamma + \delta}{b_s - \delta}\ln(y_1). \tag{11}$$

Once the flame radius is evaluated the burning rate can

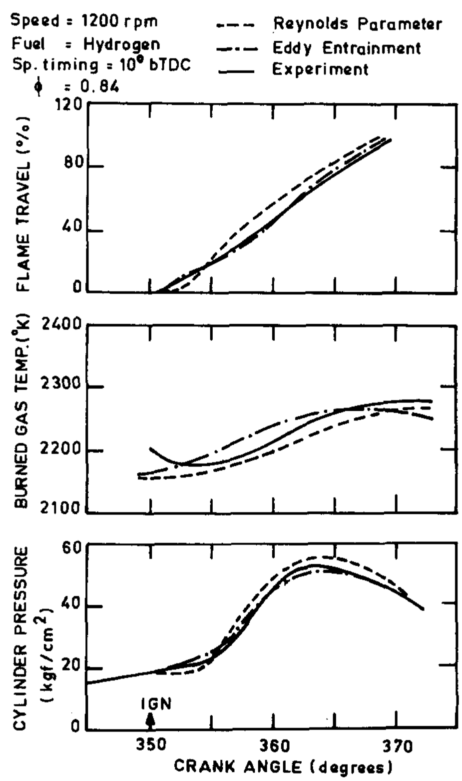

Fig. 1. Comparison of Reynolds Parameter and Eddy Entrainment models.

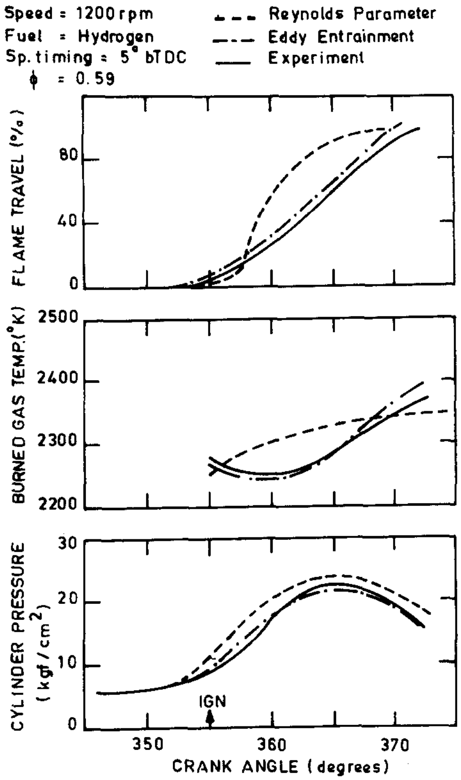

Fig. 2. Comparison of Reynolds Parameter and Eddy Entrainment models.

be determined. Even though theoretically the accuracy could be improved by successive iterations, the first-order solutions are found to be sufficiently accurate.

To evaluate and compare the relative performance of the two models the experimental results on a CFR engine converted to hydrogen Otto cycle operation with pressure crank angle diagrams [1] are used. This engine has a cylindrical combustion chamber with spark plug located at the periphery. Hydrogen is injected under pressure during the compression stroke at 105° b TDC to avoid the problem of incomplete combustion due to inadequate mixing of hydrogen with air.

RESULTS AND DISCUSSION

The engine selected for hydrogen operation is similar to the engine used by Samaga and Murthy [4] and the values of C_1 and C_2 as reported are used for Reynolds Parameter model analysis.

Using the two models, the flame travel—an indication of the burning rate—has been computed for hydrogen fuel and the relevant parameters are plotted for the combustion phase. Figure 1 shows the relative performance of the two models along with the experimental values of flame travel. From the figure it is evident that both the models are capable of predicting the performance with reasonable accuracy. This is due to the fact that laminar burning velocities of hydrogen air mixtures is of the order of 160 m/s compared to 50 m/s in gasoline air mixture. Due to this the errors in turbulent fluctuations are not reflected in the apparent burning velocities. To reduce the effect of high laminar burning velocities, the models are compared at an equivalence ratio of $\theta = 0.59$ where the laminar burning velocity is only 100 m/s.

Figure 2 shows the variation of different parameters at $\theta = 0.59$, 1200 rev/min at 8:1 compression ratio. From this it is evident that the Eddy Entrainment model is better than the Reynolds Parameter model.

It is generally difficult to predict pressure and flame radii when the flame is very close to the wall [9]. This is due to the flame quenching effect. The quenching distance in the case of hydrogen is very small compared to gasoline (0.062 cm for H_2 compared to 0.25 cm for CH_4 at $\theta = 0.84$) [1]. Hence the deviation in the predicted and measured values when the flame is close to the wall is less at $\theta = 0.84$ than at $\theta = 0.59$. The Reynolds Parameter model tends to predict higher flame travel lengths.

CONCLUSION

From the present investigation it is seen that the Eddy Entrainment model has an advantage over the Reynolds Parameter model for the following reasons.

1. Independence of fuel and engine parameters making it universally applicable for new engine development without evaluating constants.
2. Predicted flame travel distances are in agreement with experimental data up to about 90% of the travel for all equivalence ratios with little modification needed for end gas quenching.

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