BRIEF NOTE ON DEVELOPMENT OF THE MODEL FOR FLAME PROPAGATION IN THE S.I. ENGINE

The proposed model in this work is an extension of Samaga's flame propagation model [23] which is suitably modified for application to stratified combustion with in-cylinder fuel injection. A brief note on Samaga's model is given below:

Basic considerations in development of Samaga's model:

The model chosen is based on the following considerations:

1. Combustion in a S.I. engine is due to a spherical (but wrinkled) flame front originating from the spark plug and propagating in the unburned mixture through a continuously varying pressure and temperature conditions associated with a progressively varying closed chamber volume.

2. The apparent flame velocity observed at any instant of flame travel during the combustion period is the laminar burning velocity corresponding to the instantaneous condition of the unburned mixture, augmented by the engine gas turbulence and further modified due to flame transportation by the expansion of the burning gases behind the flame front.

The condition of the unburned mixture progressively changes due to piston movement as well as the compression effect arising from the expansion of burning gases. As a result, its temperature and enthalpy increase, the instantaneous values of which determine the adiabatic flame temperature and the laminar burning velocity. The turbulent flame velocity is assumed to be related to the laminar burning velocity.
through a wrinkled flame model as proposed by Shelkin [70]. The final observed flame velocity is further increased due to the enmass displacement of the flame front into the end gas region on account of the expansion of burning gases just behind the flame front. The magnitude of this effect depends on the instantaneous position of the flame front in relation to spark point, gradually decreasing to zero at the end of the flame travel.

3. The flame accelerates to full velocity during the first 10% of its travel and again decreases from its full value to near zero during the last 5% of its travel towards the end wall.

These assumptions have been based on experimental observations on engine flame development [10] and correspond to preliminary and final phases of combustion, wherein, increased heat transfer and reduced turbulence play a prominent role in combustion retardation. The preliminary phase includes the initial delay period too and it is thought that the effect of engine variables on the duration of this phase may be better represented by specifying a pre-fixed distance from the spark point, rather than a pre-fixed mass of burned mixture, owing to the above reasons of wall effects.

The following assumptions have been made in the development of the model.

1. The unburned and burned gases are separated into two energy systems (Fig 3-a) by a thin spherical and wrinkled flame front originating from the spark point.

2. Temperature in the reaction zone of the flame front remains at the equilibrium adiabatic flame temperature.
3. The fundamental reaction velocity remains at the laminar burning velocity, but the increase in flame speed resulting from the macroscopic gas turbulence is due to the effect of increased flame surface with wrinkling.

4. The magnitude of this turbulent velocity fluctuation may be functionally related to an engine Reynold’s number.

5. All the reactants and products are in the gaseous state and the products are in chemical equilibrium during and after combustion.

6. Variation in the transport properties of the gaseous mixture with respect to temperature are similar to those of air.

7. No fuel remains unburned after the passage of the flame except for any indirect heat release due to adjustments in chemical equilibrium.

8. Effect of heat transfer and end gas pre-combustion energy release on flame propagation is by way of changing the temperature and total enthalpy of unburned mixture.

3.2. Calculation of laminar burning velocity:
Dugger et al [71,72] compared the various theories of laminar flame propagation and showed that Semenov’s thermal model [17] could successfully predict the temperature dependence of flame velocity for hydrocarbon-O₂-N₂ mixtures, with an assumed overall activation energy of 40 kCal/gmol. In the present investigation, therefore, the same model is applied to estimate the instantaneous laminar burning velocities during combustion. With the assumption no 6 mentioned above, the equation simplifies to the following form.
\[ V_L = \left[ k \cdot \left( \frac{n_1}{n_2} \right)^2 \cdot T_U^2 \cdot T_f \cdot \frac{1}{E^2} \cdot \frac{T_f}{1 + \frac{T_U}{T_f}} \cdot e^{-\frac{E}{R T_f} \cdot (1 - A \cdot Y)} \cdot \frac{375 \cdot (1 - \frac{T_U}{T_f})^2}{(1 - \frac{T_U}{T_f})^2} \right]^{1/2} \tag{3-1} \]

where, \( Y = 1 - \frac{R \cdot T_f}{E \cdot (1 - \frac{T_U}{T_f})} \) and \( A = 1/\phi \) for \( \phi > 1 \) and \( \phi = \phi \) for \( \phi <= 1 \)

The constant \( K \) has been evaluated based on the Ref 72 which has come out to be equal to 59.0 for an activation energy of 40 kCal/gmol. When applied to engine conditions, \( T_u \) and \( T_f \) correspond to the instantaneous values of unburned mixture temperature and the adiabatic flame temperature during the flame progress.

3.3. Combustion acceleration due to engine gas turbulence:

The laminar burning velocity calculated from equations 3-1 was augmented for engine gas turbulence by expressing the turbulent velocity fluctuations as a function of an engine Reynold's parameter.

Concept of "Engine Reynold's parameter" - The criterion for turbulent flame propagation.

As suggested by the observed variations of engine flame speeds with respect to engine variables a dimensionless parameter was sought, which could effectively represent the influence of various physical factors on the turbulent component of flame propagation. On this basis a Reynolds parameter was defined as -

\[ R_p = V_f D \rho_{av} / T_{av}^{0.67} \]
and the apparent flame speed was calculated from the formulation as -

\[ V_{\text{app}} = \left( V_L^2 + C_1 R_p C_2 \right)^{1/2} . C_T \quad \rightarrow \quad (3-2) \]

Equation 3-2 gives the instantaneous rate of growth of the flame radius from the ignition point, as can be observed optically or photographically. However, as mentioned earlier, the initial phase of combustion involving the development of flame kernel takes place rather slowly, due to reasons of wall effects. It is approximated by an assumption that the flame front accelerates linearly with distance to its full value (given by equation 3-2) at 10% of its total travel.

When the flame approaches the end wall it is again assumed that during the last 5% of the travel, the flame velocity decreases linearly with distance tending to zero at the end of travel.

From 10 to 95% of the travel, the rate of flame progress is completely defined by equation (3-2). The flame profile is taken to be spherical, neglecting any effects of flame front distortion due to heat transfer at the contacting wall surfaces.

It can be seen from equation (3-2) that the flame velocity is made up of a laminar and turbulent component. The effect of chemical factors such as fuel characteristics (activation energy and hydrogen/carbon ratio) and equivalence ratio will be taken care of mainly by the laminar component but also indirectly to a smaller extent, by the turbulent terms, in as much as they affect the temperature and pressure of the gases. The effect of physical factors such as engine speed, size and configuration will be taken care of by the turbulent term. For high engine speeds, the flame velocity is mainly determined by the turbulent component. In the extreme case of
zero engine speed, the flame speed corresponds to laminar burning velocity as in a closed combustion vessel containing a quiescent mixture.

This model was validated in respect of large number of experiments conducted on Variable Compression Ratio (V.C R) engine with large combination of operating conditions. The constants $C_1$ and $C_2$ have been evaluated to be $C_1=2785$, $C_2=20$, with appropriate units.

3.4. Application of model for stratified charge combustion in the present two-stroke engine:

The stratification scheme is assumed as a linear variation in the fuel-air equivalence ratio from the spark plug point to the end wall. The above combustion model is applied through the mixture with continuously varying mixture strength. The intake jet velocity $V_j$ in Samaga's model is substituted by the effective velocity of charge entering through the transfer port. This is calculated as

$$V_j = \frac{PSM \cdot AP}{ATP} \times \frac{180}{27.5}$$

where 27.5 is taken as the effective crank angle duration of the transfer port opening during which the charge from the crank-case gushes into the cylinder. The scavenging efficiency is implicitly taken into account by calculating the exhaust fraction of the residual gases remaining in the cylinder under various throttle openings, assuming that at the exhaust port closure, the trapped mass will be at atmospheric pressure at the mixed temperature of fresh charge and residual gases.
By a few trial computations, the stratified scheme is optimized for best matching of the predicted $P-\phi$ diagrams with the measured average $P-\phi$ traces for various operating conditions.

The optimized stratification scheme thus obtained is as follows: The mixture strength near the spark plug which is close to the wall will be the richest due to the evaporation of the wall-deposited fuel and is taken to be $\phi = 1.4$, progressively decreasing down to $\phi = 0.8$ at the end wall opposite to the spark plug for an overall $\phi_{av} = 1.1$. At lower loads the injected fuel being lesser in quantity is assumed to be distributed on a smaller volume, but maintaining the same stratification scheme commencing from the spark plug point. With higher quantities of fuel injected the scheme of stratification remains the same but mixture strength at each point is enhanced from wall to wall, commensurate with the quantum of total fuel injected.

3.5. Validation of the model:

Plate 3.a shows different views of the experimental test set-up. Plate 3.b shows a typical view of the combustion chamber configuration. Plate 3.c shows sample $P-\phi$ traces obtained with the engine indicating system along with the motored diagrams. Fig. 3.1 to 3.17 shows a comparison of simulated $P-\phi$ diagrams with the experimentally obtained averaged $P-\phi$ traces for various engine operating conditions. The comparison is found to be fairly good which shows the validity of the model for combustion prediction in such an engine.
Flame Front
Heat release (pre-comb reaction)
Heat loss
Heat Release (combustion)
Heat loss
Work (compr) —^ Work (compr)
J
Piston work

Fig. 3-a: The energy system
Plate No 3.a.
Different views of the experimental test setup
Plate No 3.b.
Combustion chamber configuration under study
Plate No 3.c.
Sample P-6 traces obtained with the engine indicating system (Motored and Fired)
FIG 31 COMPARISON OF MEASURED AND COMPUTED P-Θ DIAGRAMS
FIG 3.2 COMPARISON OF MEASURED AND COMPUTED P-Φ DIAGRAMS

- EXPERIMENTAL
- COMPUTED FROM THE MODEL

SPEED 1500 RPM
\( \phi_{av} = 1.13 \)
MASS FLOW RATE OF AIR 94.62% \( W_{air} \)
SADV 21° CA BTDC
SPEED: 1500 RPM

\[ \phi \approx 1.08 \]

MASS FLOW RATE OF AIR: 79.49% \( \text{Wair} \)

SADV: 21° CA BTDC

EXPERIMENTAL

COMPUTED FROM THE MODEL

FIG 3.3 COMPARISON OF MEASURED AND COMPUTED P-0 DIAGRAMS
SPEED 1500 RPM
\( \phi_{av} \) 1.24
MASS FLOW RATE OF AIR: 62 25\% \( W_{air} \)
SADV 21° CA BTDC

FIG 3 4 COMPARISON OF MEASURED AND COMPUTED P-\( \theta \) DIAGRAMS
SPEED: 1500 RPM
\[ \phi_{av} = 1.28 \]
MASS FLOW RATE OF AIR: 60.7 kg/s
SADV: 21° CA BTDC

FIG 35 COMPARISON OF MEASURED AND COMPUTED P-Ø DIAGRAMS
FIG 3.6 COMPARISON OF MEASURED AND COMPUTED P-θ DIAGRAMS

- EXPERIMENTAL
- COMPUTED FROM THE MODEL

SPEED 2000 RPM
φav .1 085
MASS FLOW RATE OF AIR 86.05% W_air
SADV 21° CA BTDC

PRESSURE [K.PASCAL x (10^-2)]

TDC

*CRAN ANGLE

0 20 40 60 80 100 120
FIG 3.7 COMPARISON OF MEASURED AND COMPUTED P-V DIAGRAMS

PRESSURE [PASCAL x (10^-2)]

CRANK ANGLE (° BTDC)

SPEED 2000 RPM

MASS FLOW RATE OF AIR 75.03 kg/s

EXPERIMENTAL

COMPUTED FROM THE MODEL
FIG 3-8 COMPARISON OF MEASURED AND COMPUTED P-Θ DIAGRAMS

- EXPERIMENTAL
- COMPUTED FROM THE MODEL

SPEED 2000 RPM
φ av . 1.08
Mass Flow Rate of Air 78.88 W Air
SADV 21° CA BTDC
FIG. 3-9 COMPARISON OF MEASURED AND COMPUTED P-Θ DIAGRAMS
FIG 3.10 COMPARISON OF MEASURED AND COMPUTED P-Θ DIAGRAMS

SPEED 2000 RPM

φ av 1 23

MASS FLOW RATE OF AIR 5163% W

SADV 21° CA BTDC

EXPERIMENTAL COMPUTED FROM THE MODEL

O O O

PRESSURE [KPa] x (10\(^{-2}\)]
FIG 3.11 COMPARISON OF MEASURED AND COMPUTED P-Θ DIAGRAMS

- EXPERIMENTAL
- COMPUTED FROM THE MODEL

SPEED = 2000 RPM
\( \phi_{av} = 1.096 \)
MASS FLOW RATE OF AIR = 62.05 % \( \dot{m}_{air} \)
SADV = 21° CA BTDC
FIG 3.12 COMPARISON OF MEASURED AND COMPUTED P-θ DIAGRAMS

- SPEED: 2500 RPM
- \( \phi_{av} \): 104
- MASS FLOW RATE OF AIR: 91.28% Wair
- SADV: 21°CA BTDC

- EXPERIMENTAL
- COMPUTED FROM THE MODEL

Pressure [Kpascal x (10^-2)]

- TDC CRANK ANGLE

\[ -120 \quad -100 \quad -80 \quad -60 \quad -40 \quad -20 \quad 0 \quad 20 \quad 40 \quad 60 \quad 80 \quad 100 \]

\[ \text{TDC} \]
FIGURE 3.13 COMPARISON OF MEASURED AND COMPUTED P-θ DIAGRAMS

- **EXPERIMENTAL**
- **COMPUTED FROM THE MODEL**

**Specifications:**
- **SPEED:** 2500 RPM
- **Φ AV:** 0.978
- **MASS FLOW RATE OF AIR:** 81.43 % \( W_{air} \)
- **SADV:** 21° CA BTDC

**Graph Details:**
- **Y-axis:** Pressure (kPa)
- **X-axis:** Crank angle (°)
- **Range:**
  - Pressure: 0 to 24 kPa
  - Crank angle: -120° to 120°
FIG 3.14 COMPARISON OF MEASURED AND COMPUTED P-θ DIAGRAMS

SPEED : 2500 RPM
Φ av : 0.969

MASS FLOW RATE OF AIR : 72.85% W_air
SADV : 21° CA BTDC
FIG 3.15 COMPARISON OF MEASURED AND COMPUTED P-Θ DIAGRAMS

- EXPERIMENTAL
- COMPUTED FROM THE MODEL

SPEED : 2500 RPM
ϕ av = 0.958
MASS FLOW RATE OF AIR : 67.5 % w_air
SADV : 21° CA BTDC

CRANK ANGLE

PRESSURE [k PASCAL x10^{-2}]

TDC

* CRANK ANGLE
FIG 3.16 COMPARISON OF MEASURED AND COMPUTED P-Θ DIAGRAMS

- EXPERIMENTAL
- COMPUTED FROM THE MODEL

SPEED : 2500 RPM
Φ av : 0.990
MASS FLOW RATE OF AIR : 58.4% \( \dot{m}_{air} \)
SADV : 21 CA BTDC

PRESSURE [k PASCAL x (10^-2)]

CRANK ANGLE
FIG 3.17 COMPARISON OF MEASURED AND COMPUTED P-Θ DIAGRAMS

- **SPEED**: 2500 RPM
- **Φ av**: 1.09
- **MASS FLOW RATE OF AIR**: 47.7% W_air
- **SADV**: 21° CA BTDC

**Explanations**
- **Experimental Points**: ○
- **Computed from the Model**: ●