1. Introduction

The engine combustion process is exceedingly complex. Even in the conventional spark-ignition engine, where under many operating modes the fuel and air can be treated as premixed, the combustion process is initiated in a three-dimensional time-varying turbulent flow, concerns a fuel which is a blend of hundreds of different organic compounds whose combustion chemistry is poorly understood, and takes place in a space confined by the combustion chamber walls whose shape varies with time and whose wall directly influence aspects of the process.

A number of engine combustion model classifications have been proposed. The most useful classification follows from one proposed by Hyewood [1]. Its utility stems from the fact that different classes of combustion model, because of their formalism, are generally useful in examining different kinds of combustion related engine problems. These categories of combustion model are: 1) zero dimensional (sometimes called thermodynamics) model, 2) quasi dimensional (sometimes called entrainment) models, and 3) multidimensional (sometimes called detailed) models.

Zero dimensional and quasi dimensional models are structured around a thermodynamics analysis of the contents of the engine cylinder during the engine operation cycle [1].

Simulation of internal combustion engine processes involves, developing a combustion model using the combination of assumptions and equations for predicting the engine performance during the open period which involves suction and exhaust strokes and the closed period which comprises compression, combustion, and expansion stroke [2].

The objective of the present study is to develop a combustion model for a spark ignition engine run-
ning with iso-octane as a fuel and predicting its behavior.

2. Thermodynamic Engine Model

Thermodynamic model of the gasoline fueled engine is considered the closed system that both inlet and exhaust valves remain closed. The important events such as compression, combustion, and expansion take place during this period. The engine is reduced to a thermodynamic system, which consist of a homogeneous mixture of air, gasoline, and residual gas from the previous cycle. The boundaries of the system are the cylinder walls, cylinder head, and top of the piston head. Work is added to or taken from the system through the motion of the piston and heat is transferred to or from the system through the boundary surfaces [2].

Thermodynamic models can be classified into two groups; one-zone model and multi-zone model. In one-zone model, the cylinder charge is assumed to be uniform in both composition and temperature and the first law of thermodynamics is used to calculate the mixture energy. One-zone model represents the unburned mixture ahead of the flame and one the burned mixture behind the flame.

In two-zone model, the cylinder mixture is divided into burned and unburned zones, which are separate from each other by a surface of discontinuity. The composition and temperature of the burned and unburned gases are different and the pressure is uniform through out the combustion chamber. Two-zone model is used to calculate the mass fraction burned profile from measured cylinder pressure data [3].

In this model, the following assumptions have been used.

- The burnt and unburned gases are assumed to be ideal and non-reacting. The heat transfer from the burned to unburned gases is assumed to be negligible. The pressure is constant for both the zones. There is assumed to be no dissociation in the unburned gases prior to combustion.

Figure 1 shows the general structure of the thermodynamics model.

Thermodynamic engine model is based on the first law of thermodynamics expressed as [4, 5]:

\[ dU = \delta Q - \delta W + h_{in} dm_{in} - h_{out} dm_{out} \]

The governing equation for the burned and unburned gas zones can be written as:

\[ m = m_u + m_b \]

\[ \frac{dm}{d\theta} = \frac{dm_u}{d\theta} + \frac{dm_b}{d\theta} \]

\[ V = V_u + V_b \]

where the subscripts u and b denote the unburned...
and burned gases, respectively. The following system of equations:

\[
m_a C_v \frac{dT_u}{d\theta} = -p \frac{dV}{d\theta} - \frac{dQ_u}{d\theta} + h_u \frac{dm_u}{d\theta}
\]

\[
m_b C_v \frac{dT_b}{d\theta} = -p \frac{dV}{d\theta} - \frac{dQ_b}{d\theta} + h_b \frac{dm_b}{d\theta}
\]

\[Q_u = h A_u (T_w - T_u)\]

\[Q_b = h A_b (T_w - T_b)\]

The unburned temperature is determined using the isentropic relationship in following equations:

\[T_u = T_{uo}(P/P_o)^{(k-1)/k_u}\]

\[P_o = P_{in}(V/V_o)^{k_u}\]

In the model, heat from combustion is supplied using a Wiebe function:

\[x = 1 - \exp\left[-a\left((\theta - \theta_v)/\Delta\theta\right)^{m+1}\right]\]

\[dx = \left(\frac{m+1}{\theta_v^{m+1}}\right)\left(\theta - \theta_v\right)^m \exp\left(-a\left(\frac{\theta - \theta_v}{\theta_v}\right)^{m+1}\right)\]

With \(a=5\) and \(m=2\), \(\theta_v\) is the spark time at the beginning of combustion in crank angle and \(\Delta\theta\) is the total combustion duration \((\theta_v = 0 \text{ to } \theta_v = 1)\).

The heat release rate is given by:

\[\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} P \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} - \frac{C_p(T)}{R} \frac{dV}{d\theta}\]

\[\frac{dQ}{d\theta} = P \frac{dV}{d\theta} \left(1 + \frac{C_v}{R_b} + \frac{C_v}{R_b} \frac{dP}{d\theta}\right)\]

The specific heat ratio can then be calculated

\[\gamma = \frac{C_p}{C_v - R}\]

Pressure is calculated from the equation:

\[\frac{dp}{d\theta} = \frac{T \sum R_i \frac{dm_i}{d\theta} + \sum R_i m \frac{dT}{d\theta} - p \frac{dV}{d\theta}}{V}\]

The total volume of the gases inside the cylinder at a particular crank angle is given by the expression:

\[\frac{dV}{d\theta} = \frac{\pi}{4} D^2 \left(\sin\theta + \frac{1}{2\lambda} \sin 2\theta\right)\]

3. Results

Figure 2 shows the cylinder pressure in a spark ignition engine as a function of crank angle. The pressure from this thermodynamics model reveals that it is shown the peak pressure 42 bar at around 18 degree after TDC (top dead center).

The calculation condition for compression, combustion, and expansion predictions is shown in Table 1. The model assumes that crank angle is simulated. This implies that the diagnostic version of the heat release equation can be used to calculate the quantities of heat released at each crank angle [7, 8].
Figure 3. Variation of heat release with crank angle.

Under simulated condition, the heat release is about 16 cal/cycle.

During the compression, combustion and expansion, the combustion period is assumed that the spark ignited flame front divides the in-cylinder gas mixture into two zones: the burned and unburned zones. To simplify the two-zone combustion model, the shape of the burned zone is assumed to be a circle centered at the cylinder [9]. For the spark ignition combustion the temperature of the unburned zone is quite different from that of the burned zone as seen in Figure 4 and Figure 5.

At any given flame radius, the geometry of the combustion chamber and the spark plug location govern the flame front surface area, the area of the approximately spherical surface corresponding to the leading edge of the flame contained by the piston, cylinder head, and cylinder wall. The larger this surface area, the greater the mass of fresh charge that can cross this surface and enter the flame zone [3].

The flame is assumed to propagate spherically from the one end of the cylinder. As the flame proceeds from one end to other the traveled portion of cylinder volume was assumed completely burned, while the other as unburned [2].

Figure 6, 7, 8, and 9 show the variation of volume area, flame radius, flame area, and side area with crank angle, respectively.

An important inherent property of a combustible fuel, air, burned gas mixture is its laminar burning velocity. This burning velocity is defined as the velocity, relative to and normal to the flame front, with which unburned gas moves into the front and is transformed to products under laminar flow conditions [3]. Figure 10, and 11 show the variation of laminar burning velocity, and turbulence burning ve-
4. Conclusions

The simulation has become a powerful tool as it saves time and also economical when compared to experimental study. The following conclusions can be made from above research;

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spark ignited flame front divides the in-cylinder gas mixture into two zones: the burned and unburned zones.

The effects of various cylinder pressure, heat release, flame temperature, unburned gas temperature, flame properties, laminar burning velocity, turbulence burning velocity, turbulence flame velocity, etc. were simulated. The simulation and analysis show several meaningful results.

Nomenclature

| Symbol | Description |
|--------|-------------|
| A      | flame area  |
| B      | cylinder bore |
| D      | diameter of piston |
| h      | heat transfer coefficient |
| k      | specific heat ratio |
| m      | mass of mixture |
| P      | gas pressure |
| Q      | heat of the gases |
| R      | universal gas constant |
| r      | crank radius |
| S      | stroke |
| SL     | laminar flame speed |
| T      | temperature |
| θ      | crank angle |
| ∀      | specific heat ratio |
| ρ      | density of mixture |

subscripts

| Subscript | Description |
|-----------|-------------|
| b         | burned |
| u         | unburned |

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