CHAPTER 6

HEAT RELEASE ANALYSIS

6.1 INTRODUCTION

The very important and easily measurable parameter that is available to understand the combustion process of an engine is the measured pressure traces at every crank angle of engine rotation (Lancaster et al 1975). But this pressure history is influenced by many engine variables like changes in combustion chamber volume, heat transfer to walls, and mass leakage (Heywood 1989). Hence, to analyze the combustion phenomena critically, it is necessary to separate the effects of volume change, heat transfer and mass loss. This can be done by calculating heat release rate, which has the benefit of identifying the combustion indices like ignition delay, combustion duration, heat release rate and its crank angle position etc. The effect of heat transfer, gas leakage and crevice flow on combustion can be analyzed by this method.

Many literatures (Rassweller et al 1938, Blizard 1974, Lyn 1977) reveal the techniques for calculating the mass fraction burned and heat release rates for four-stroke SI engine.

Rassweller and Withrow (1938) developed a method for calculating the mass fraction of charge burned in a spark ignition engine. In their analysis, the difference between the measured and calculated pressures based on polytrophic compression was taken to be proportional to the mass fraction burned.
A two-zone model for the unburned air-fuel mixture and burned gas was developed by Krieger and Borman (1966). They also developed one-zone model for diesel engine assuming all the cylinder contents were uniform. In these models, the thermodynamic properties of the cylinder contents were accurately determined which involved substantial computation.

6.1.1 Heat Release Analysis of Two-Stroke Engine

Most of the research findings were focused on four-stroke SI engines. Combustion process is similar for two stroke and four stroke engines but not the parameters like heat transfer rate, trapped charge mass and purity of charge etc. Most of the two-stroke engines are air-cooled, which create difference in heat transfer rate. The usual assumption of initial charge trapped inside the cylinder for a four-stroke engine is based on the volumetric efficiency. But in a two-stroke engine, the large amount of exhaust residue and the unsteady nature of cylinder flow are the floating parameters. These intricate parameters were calculated with unsteady flow simulation model (Nedunchezhian 2000).

6.2 AIM AND SCOPE OF THE PRESENT WORK

The objectives of the present work are:

- To develop a heat release analysis code suitable for two-stroke SI engine combustion analysis.
- To fit empirical relations for total heat release value and mass fraction burned.
- To incorporate the effects of crevice flow and heat transfer.
• To calculate different crank angle positions of heat release values and other combustion parameters.

• To apply the heat release analysis procedure for comparing base and magnetically activated fuel on catalytic coated engine performance.

As the present work is with magnetically activated fuel on lean burn two-stroke SI engine, the problems like, cyclic variation, combustion nature of air-fuel mixture etc. need to be analyzed. The measured cylinder pressures are affected by crevice flow, mass leakage and changes in chamber volume. These effects are explicitly incorporated in the heat release analysis procedure based on the First Law of Thermodynamics. Another advantage of the above procedure is that the pressure changes can be directly related to the amount of fuel chemical energy released by combustion. The combustion parameters and calculated heat release data are essential for understanding the nature of combustion. Moreover, a comparison can be made between catalytic and non-catalytic combustion on the effect of high gauss magnetic activation.

6.3 METHODOLOGY

The methodologies involved in developing the code are:

• Calculation of instantaneous heat release rates and cumulative heat release values from the measured cylinder pressures.

• Calculation of mass fraction burned.

• Curve fitting with empirical relations for total heat released and mass fraction burned.

• Calculation of occurrence of crank angle of 5%, 10%, 50% and 90% heat release values.
- Determination of combustion parameters from heat release angles.
- Determination of flame burning speed from the burning rate for base and magnetically activated fuel on catalytic coated engine.
- Comparing the base and magnetically activated fuel on catalytic coated engine heat release rates.

The heat release analysis code is developed following the above methodology and its procedure is briefly described in the following sections. The code is written in C.

### 6.4 HEAT RELEASE MODEL

To calculate the heat release rate and mass fraction burned, a simple one dimensional heat release model is developed based on Gatowski et al (1984).

#### 6.4.1 Mathematical Model

Figure 6.1 schematically represents the thermodynamic system, control volume and the sign convention used in the model. For the piston movement through a small crank angle of dθ, the cylinder volume changes from V1 to V2. The instantaneous values of piston movement are known from the cylinder geometry. When the combustion process is in progress, a quantity of heat δQ_{ch} is released during this time interval. The cylinder pressure changes from p1 to p2 during the time step, and some quantity of heat δQ_{int} is lost through cylinder walls. The internal energy change is δu_s and the work done on the piston is δW during the time interval. A small amount of cylinder
gas dmi, flows into the crevice spaces. The First Law of Thermodynamics is applied for the system:

\[
\delta Q_{ch} = du_s + \delta W + \sum h_i dm_i + \delta Q_{ht}
\]  

(6.1)

During combustion, chemical energy is transformed into thermal energy of the charge through a change in composition from the unburned mixture to burned reaction products. The internal energies of both products and reactants are referred to the same datum, and combustion is treated within the \(du_s\) term of the above equation.

![Figure 6.1 Representation of thermodynamic system](image)

Figure 6.1 Representation of thermodynamic system

The change in sensible internal energy is calculated from the mean charge temperature,

\[
u_s = m_c u(T)
\]

\[
du_s = m_c c_v(T) dT + u(T) dm_c
\]  

(6.2)

where \(T = \text{mean temperature} = (T_1 + T_2)/2\) and the temperatures, \(T_1\) and \(T_2\) are calculated from the equation of state as,

\[
T_1 = (p_1 V_1) / (m_c R) \quad \text{and} \quad T_2 = (p_2 V_2) / (m_c R)
\]
For premixed charge, the mass flow term consists of flow of gases into and out of crevice spaces. Hence, the above Equation (6.2) may be modified by substituting for:

\[ m_i = m_i = dm_{cr} \]

and the derivatives for \( du_s \) and \( \delta W \), as

\[ m_{ch} = d(mu) + h'dm_{cr} + pdV + dQ_{ht} \]
\[ m_{ch} = m_c c_v dT + (h' - u) dm_{cr} + pdV + \delta Q_{ht} \]  \hspace{1cm} (6.3)

where \( dm_{cr} \) is crevice mass flow, whose sign depends upon the flow direction

\( h' \) is evaluated at cylinder conditions when \( dm_{cr} \) is positive and at crevice conditions when \( dm_{cr} \) is negative.

The final result of the above equation is as in the following form:

\[ Q_{ch} = (pV / R)c_v \left\{ (dp/p + dV / V - RT / pV) dm_c \right\} + \]
\[ (h'-u) dm_{cr} + pdV + \delta Q_{ht} \]
\[ = (c_v / R) Vdp + \left\{ (c_v + R) / R \right\} pdV - \]
\[ (h'-u + c_v T) dm_{cr} + \delta Q_{ht} \]  \hspace{1cm} (6.4)

The most important thermodynamic property used for heat release calculations is the ratio of specific heat \( \gamma \), which is obtained by the linear approximation:

\[ \delta(T) = a + bT \]  \hspace{1cm} (6.5)
The final form of gross heat release takes the following form:

\[ \delta Q_{ch} = \gamma / (\gamma - 1) p dV + 1 / (\gamma - 1) V dp + \delta Q_{ht} + R \left\{ T' + T / (\gamma - 1) - 1 / b1n \left[ (\gamma - 1) / (\gamma' - 1) \right] \right\} dm_{cr} \]  
(6.6)

These equations can be solved by using the pressure traces, heat transfer model, and crevice mass flow.

### 6.4.2 Heat Transfer

The heat from the combustion chamber is transferred mainly by convective mode. The rate of heat conducted by convective heat transfer is given by,

\[ dQ_{ht} / dt = A h (T - T_w) \]  
(6.7)

Blair (1996) has suggested Annand’s (1962, 1963) method for heat transfer calculation during the closed cycle period of spark ignition engines. Annand (1963) developed a correlation for calculating the heat transfer coefficient which is of the form

\[ Nu = C Re^{0.7} \]

\[ Nu = hD/k = C Re^{0.7} \]

\[ = C \left( \rho \omega D / \mu \right)^{0.7} \]  
(6.8)

The equation is modified after substituting the ideal gas law for density and assumed temperature scaling for viscosity and conductivity.

\[ h \left[ w / m^2 K \right] = C_1 D \left[ m \right]^{-0.2} p \left[ \text{bar} \right]^{0.7} T \left[ K \right]^{-0.53} \omega \left[ m/s \right]^{0.7} \]  
(6.9)

The specific speed \( \omega \) is defined by Woschni (1967) as,
\[ \omega = C_2 \left( S_p \text{ [m / s]} + C_3 \left( \frac{V_{\text{disp}}}{t_{\text{pc}}} \right) \times \right) (pf - pm) \bigg/ Pt_{\text{pc}} T_{\text{tpci}} [K] \]  

(6.10)

where, \( S_p \) is the mean piston speed, \( pf \) is the firing pressure, \( p_m \) is the motoring pressure, \( V_{\text{disp}} \) is the displaced volume, \( V_{\text{tpc}} \) is the volume at transfer port closing and \( C_1, C_2, C_3 \) are constants.

### 6.4.3 Crevice Model

As the cylinder pressure rises during engine compression, unburned mixture is forced into the crevice regions (Heywood 1989). Since these volumes are thin, they have a large surface to volume ratio. Hence, the gas flowing into each crevice space cools by heat transfer to the adjoining walls. During combustion, while the pressure continues to rise, unburned mixture continues to flow into the crevice volumes. When the flame arrives at each crevice, it can either propagate into the crevice or partially burn the fuel within the crevice or it can quench at the crevice entrance. Flame quenching depends upon the crevice entrance geometry, the composition of the unburned mixture, and its thermodynamic state. After the flame arrival and quenching, burned gases will flow into each crevice until the cylinder pressure decreases. Once the crevice gas pressure is higher than the cylinder pressure, gas flows back from crevice into the cylinder.

The crevice volumes in an engine are the volumes above the top ring between the cylinder and piston, threads around the spark plug and pressure transducer, space between cylinder block and head and the space around the spark plug center electrode. The crevice walls are cold, and hence the gas in the crevice is close to the wall temperature. Crevices may attain substantial amount of gas during the end of compression stroke (Namazian et al 1982). Some gas also leaks into the crankcase as blow by and causes loss
in heat release. The crevice volumes are modelled as a single aggregate volume of uniform pressure and varied temperature.

The amount of gas in crevice space at any time is,

\[ m_{cr} = \frac{pV_{cr}}{(RT_w)} \quad \text{and} \]

\[ dm_{cr} = \left[ \frac{(V_{cr}/RT_w)}{dp} \right] \quad (6.11) \]

The wall temperature and volume are taken to be constant. The gas flows in and out of crevices depending on the pressure differences. Mass flows into the crevice when the cylinder pressure is increasing. Mass returns to the chamber when the chamber pressure starts decreasing. These flows are modelled as one dimensional, compressible, frictionless and adiabatic.

### 6.5 ASSUMPTIONS FOR HEAT RELEASE ANALYSIS

To calculate the heat release values, certain constants are assumed and adjusted to suit the tested engine configuration. The crevice volume is assumed as a single volume of 1.5% of clearance volume. The constants for heat transfer calculations are taken from the previous works (Heywood 1989) and adjusted as:

\[ C_1 = 3.225 \text{ for base and 3.3 for catalytic engine} \]
\[ C_2 = 2.26 \text{ for compression, 2.28 for combustion and} \]
\[ C_3 = 0.0 \text{ for compression, } 3.25 \times 10^{-3} \text{ for combustion} \]

The cylinder wall temperature is taken as 500 K. Commercial petrol with 2% of SAE 40 oil is used as fuel and the lower calorific value is 43 MJ/kg.
6.6 MODEL VALIDATION

To validate the heat release model the gross heat energy input is calculated from the mass of charge trapped inside the cylinder and the lower calorific value. The mass of charge is calculated from the pressure and temperature at the time of exhaust port closing, the trapping efficiency and the residual mass. The values of trapping efficiency are used from the simulation model (Nedunchezhian 2000). The difference between the gross heat energy input and the calculated cumulative heat release is less than 3%. Since the engine is two-stroke air-cooled, this 3% unaccounted heat loss is within the limit and hence the model is taken as valid.

6.7 RESULTS AND DISCUSSIONS

The heat release analysis code is used to calculate the instantaneous heat release rates at each crank angle interval. The following sections describe the results of the heat release code.

6.7.1 Instantaneous Heat Release Rates

Figure 6.2 shows the cylinder pressure data, which is used as input for calculating the heat release data. The TDC, ignition and the crank angle occurrence of maximum pressure (CAP_max) are also indicated in the figure. The corresponding instantaneous heat release rate curve is given in Figure 6.3. Part of the cylinder pressure trace is also included in the figure for comparing the occurrence of peak pressure and maximum rate of heat release (CA(dQ/d\(\theta\))_{max}). It can be observed from the figure that the peak pressure occurs at 36° aTDC and the maximum heat release rate occurs at 32° aTDC. The heat release curve and the pressure curve are symmetrical about its peak.
Figure 6.2  Plot of measured cylinder pressure with crank angle

Figure 6.3  Plot of instantaneous heat release rates calculated from measured cylinder pressure
The general shape of the instantaneous heat release curve is of triangle and a simple relationship was suggested by Blair (1996) to calculate the area under this curve. The total area under the heat release curve will give the cumulative heat release value. The relation given by Blair is,

\[ QR = \left(\frac{14}{36}\right) \times \left[\frac{dQ}{d\theta}\right]_{\text{max}} \times \theta_{cd} \]  \hspace{1cm} (6.12)

The entire area of the instantaneous heat release curve is assumed to have different triangles and hence the above equation results. Figure 6.4 shows the assumed shape of heat release curve. However, at the lean mixture, the shape of the curve changes as the combustion rate changes. The normalized heat release curve for lean and rich mixtures is shown in Figure 6.5.

![Figure 6.4 Possible model of heat release profile](image-url)
Figure 6.5 Variation of normalised heat release with normalised combustion duration

There is substantial variation in the shape of the curve and the above relation fails to predict the total heat release values. The relation given by Equation (6.12) is used to predict the total heat release value and the results are presented in Table 6.1. The predicted values are compared with the calculated value obtained from the instantaneous heat release rate curve. The error between the values obtained from the Equation (6.12) and the actual heat release values are also indicated in the table. It can be observed from the table that the errors are higher for both very rich and very lean mixtures.
Table 6.1  Comparison of calculated and predicted heat release values

<table>
<thead>
<tr>
<th>A/F Ratio</th>
<th>Coefficient Ce</th>
<th>θcd</th>
<th>(dQ/dθ) max</th>
<th>QR actual</th>
<th>QR predicted by Eqn (6.12)</th>
<th>% of Error</th>
<th>QR by Eqn. (6.13)</th>
<th>% of Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>11.1</td>
<td>0.32</td>
<td>92</td>
<td>6.01</td>
<td>167</td>
<td>215.02</td>
<td>28.75</td>
<td>176.93</td>
<td>5.61</td>
</tr>
<tr>
<td>13.9</td>
<td>0.35</td>
<td>94</td>
<td>5.15</td>
<td>162</td>
<td>188.26</td>
<td>16.21</td>
<td>169.43</td>
<td>4.38</td>
</tr>
<tr>
<td>15.3</td>
<td>0.42</td>
<td>104</td>
<td>3.22</td>
<td>138</td>
<td>130.23</td>
<td>-5.62</td>
<td>140.64</td>
<td>1.88</td>
</tr>
<tr>
<td>16.7</td>
<td>0.43</td>
<td>110</td>
<td>2.71</td>
<td>130</td>
<td>115.92</td>
<td>-10.8</td>
<td>128.18</td>
<td>-1.41</td>
</tr>
<tr>
<td>18.1</td>
<td>0.44</td>
<td>115</td>
<td>1.96</td>
<td>105</td>
<td>87.65</td>
<td>-16.5</td>
<td>99.17</td>
<td>-5.87</td>
</tr>
</tbody>
</table>

The relation given in Equation (6.12) contains both the maximum heat release rate and combustion duration, with the units of J/degree and degrees respectively, and hence the speed variation effect is included. The effect of variation in air-fuel ratio is not included in the above relation. The reason for this may be that a normal two-stroke engine runs with a narrow range of air-fuel ratio of 12:1 to 14:1

Whereas in the present study, the air-fuel ratio varies widely and hence the above relation is not sufficient. Hence, a new relation similar to that of Blair is suggested which predicts the total heat release values accurately even for lean mixtures.

The total heat release value is given by,

$$QR = Ce \left[ (\frac{dQ}{dθ})_{max} \theta_{cd} \right]$$  \hspace{1cm} (6.13)
where, Ce is a coefficient calculated from the measured data. The curve fit relation obtained for the coefficient is of the form,

\[ Ce = 0.0205053 \times A/F + 0.106542 \]  

(6.14)

Figure 6.6 illustrates the curve fit relation for the coefficient Ce. The measured data for both 3000 rpm and 4000 rpm speed are close to the curve fit line.

\[ Ce = 0.0205053 \times A/F + 0.106542 \]

**Figure 6.6  Curve fit for the coefficient Ce**

The relations given by Equation (6.13) is used to predict the total heat release for different air-fuel ratios and compared with the calculated actual values. The results are presented in Table 6.1. It can be observed from the table that the errors in this method are minimized maintaining the nature of variation with respect to lean and rich mixtures.
To further test the validity of the relation given by Equation (6.13), two sets of data from earlier works are compared. The first data set is taken from the experimental results of Blair (1996). The engine was 85 mm bore, 70 mm stroke, 400 cc loop scavenged, carbureted, water-cooled research engine. The combustion chamber was hemispherical type with a trapped compression ratio of 6.6:1.

The second data set is taken from the work of Gatowski et al. (1984), which is for a single cylinder, Ricordo Hydra MK III spark ignition engine, it has 85.7 mm bore, 86 mm stroke with a compression ratio of 8.4:1. The engine speed was 1500 rpm with an intake pressure of 0.7 atm. The results of total heat release value and the error involved are presented in Table 6.2.

<table>
<thead>
<tr>
<th>A/F Ratio</th>
<th>Coefficient Ce</th>
<th>ocd</th>
<th>(dQ/do)max</th>
<th>QR actual</th>
<th>QR by Eqn. (6.13)</th>
<th>% of Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>13</td>
<td>0.374</td>
<td>60</td>
<td>28.8</td>
<td>662.6</td>
<td>646.272</td>
<td>-2.5269</td>
</tr>
<tr>
<td>15.14</td>
<td>0.416</td>
<td>47</td>
<td>45</td>
<td>900</td>
<td>879.84</td>
<td>-2.2913</td>
</tr>
</tbody>
</table>

From the Table 6.2, it can be observed that the new relation given by Equation (6.13) is able to predict the total heat release value with good accuracy. Although, the two sets of data from different engines with different combustion rates, the error is within three percent. This confirms the general validity of the new relation given by Equation (6.13).
6.7.2 Cumulative Heat Release Values

The cumulative heat release values, which is the aggregate sum of instantaneous heat release rates is given in the Figure 6.7. This curve has the characteristic S shape. The rate of heat release is slow at the beginning and increases rapidly till reaching a peak value then again slows down.

![Graph showing Cumulative Heat Release Values with Crank Angle](image)

**Figure 6.7 Variation of cumulative heat release values with crank angle**

This is reflected in the Figure 6.7, where the cumulative release value is low in the beginning, increases exponentially and slows down at the end. The maximum cylinder pressure occurs at 46° aTDC, nearly 10° after the occurrence of maximum heat release rate. At the time of \( P_{\text{max}} \), nearly 80% of the heat is released as seen in the Figure 6.7.
6.7.3 Crevice Effect

The crevice flow is shown in the Figure 6.8, where the instantaneous mass flow rate into the crevice is plotted as a function of crank angle. The positive flow rate represents the gas flowing into the crevice and a negative flow rate indicates the flow back of gases into the chamber.

![Figure 6.8 Variation of crevice mass flow rate with crank angle](image)

As the pressure rises during compression and early combustion process, the gases flow from the cylinder into the crevice regions. Most of the gases in the crevice will flow into the cylinder when the cylinder pressure decreases, and a small amount of gas will peak through the rings and will end up in crankcase as blowby.

The accumulated mass as a percentage of total mass trapped in the crevice region is shown in Figure 6.9. The volume of service region is only 1.5% of total clearance volume and the percentage of energy content of mass
accumulated in the crevice is as much as 15%. This is due to the higher density of crevice mass compared to the cylinder gases. The peak of the curve shown in Figure 6.9 coincides with the \( \text{CAP}_{\text{max}} \), where 80% of heat release is over.

![Graph showing variation of mass and energy of cylinder and crevice with crank angle.](image)

**Figure 6.9** Variation of mass and energy of cylinder and crevice with crank angle

The crevice contains as much as 15% of total cylinder mass and this mass crevice back into the cylinder after most of the heat release is over. Hence the charge trapped in crevice burns in the latter part of expansion stroke and does not do much useful work. It affects the heat release rate and contributes UBHC emissions (Heywood 1989). This is reflected in Figure 6.10, where the effect of crevice mass on heat release is shown. Due to the crevice flow effect the net energy available for doing useful work is reduced. Although, most of the crevice mass returns to cylinder, it does not contribute to useful work due to its late burning. The figure indicates that a substantial amount of energy is lost due to the crevice gases.
6.7.4 Heat Transfer Effect

In Figure 6.10, the lower curve indicates the heat release for useful work. The upper curve is for the chemical heat release from the trapped charge. The difference between these two is the energy lost in heat transfer and the crevice. The heat from the combustion product is transferred to the walls in both convection and radiation. The important mode of heat transfer is convection. The heat transfer rate increases during flame quenching at the walls. The flame quenching at the end of combustion is minimum where most of the cylinder contents are already burned. Hence, the maximum heat transfer occurs at the end of heat release as shown in the figure.

![Figure 6.10  Heat transfer and crevice flow effect on heat release values](image-url)
6.7.5 **Mass Fraction Burned**

The cumulative heat release values divided by the total energy content of trapped charge is called mass fraction burned. A typical mass fraction burned curve is shown in Figure 6.11. It is convenient to use the mass fraction burned values to compare the engine performances at different operating conditions. As in the cumulative heat release value curve, the rate at which the air-fuel mixture burns increases in the early stage of combustion and reaches maximum in the middle of burning process and then decreases close to zero at the end of combustion process. The value of maximum mass fraction burned is less than one. The combustion inefficiency is indicated by this difference.

![Figure 6.11 Experimental mass fraction burned with crank angle](image)

6.7.6 **Functional Form of Mass Fraction Burned**

The mass fraction burned curve can be represented by a functional form called Viebe function (Viebe 1970). The mathematical expression is
fitted from the experimental data. The expression contains exponential term with numerical coefficients and relates the mass fraction burned with crank angles and duration of burning. It is expressed as,

\[ x_{\theta b} = 1.0 - \exp \left[ -a \left( \frac{\theta_s - \theta}{\theta_{cd}} \right)^m + i \right] \]  

(6.15)

where \( \theta \) is the crank angle at which the mass fraction is required, \( \theta_s \) is the start of combustion, \( \theta_{cd} \) is the total combustion duration, and ‘a’ and ‘m’ are adjustable parameters. Varying ‘a’ and ‘m’ changes the shape of the curve significantly.

The experimentally obtained mass fraction curve and the calculated mass fraction curve using the relation given by Equation (6.15) are shown in Figures 6.12.to 6.14. The values of the parameters ‘a’ and ‘m’ are also indicated in the figures. Both ‘a’ and ‘m’ values are changed with air-fuel ratio. The value of ‘a’ decreases linearly as the mixture becomes leaner.

![Graph showing experimental vs fitted mass fraction burned at A/F ratio of 11.1:1](image)

**Figure 6.12** Comparison of experimental and fitted mass fraction burned at an A/F of 11.1:1
Figure 6.13 Comparison of experimental and fitted mass fraction burned at an A/F of 16.7:1

Figure 6.14 Comparison of experimental and fitted mass fraction burned at an A/F of 18.1:1
The ‘a’ and ‘m’ parameters values obtained from the experimental data are curve fitted for various air-fuel ratios. These parameters are correlated with the air-fuel ratio and the results are shown in Figures 6.15 and 6.16. The best fit for the parameter ‘a’ is given by,

\[
a = (A/F)_{\text{Stoic}} - 0.6345 \times (A/F) \quad (6.16)
\]

and the parameter 'm' is represented by

\[
m = 2.0 \times (A/F)_{\text{Stoic}} - 6.0438 \times (A/F) \\
+ 0.416937 \times (A/F)^2 - 0.0094 \times (A/F)^3 \quad (6.17)
\]

The curve fit relations are useful in calculating the mass fraction values for different air-fuel ratios.

Figure 6.15  Curve fit of coefficient ‘a’ from experimental mass fraction burned
The start and end of combustion in diesel engines can be identified from the dP/d\(\theta\) curve, which shows definite mark (Varaprasada et al 1992). But in spark ignition engines, these definite marks are absent, and hence it is difficult to identify the start and end of combustion. It is a usual practice to define the combustion duration by the heat release angles (Heywood 1989). The crank angle occurrence of heat release values is convenient to use and characterize the different stages of combustion process. The flame development process is one such stage. It is influenced by the mixture state, composition, and air motion etc. (Turns 1996). The variation in this duration greatly influences the cyclic variation of combustion (Keck et al 1987).

The heat release rates reflect the combustion history of trapped charge. Crank angle of heat release values such as 5\%, 10\%, 50\% and 90\%
indicate different phases of combustion (Heywood 1989). Some important combustion parameters can be determined from the heat release angles. The crank angle position of heat release values and the related combustion parameters are indicated in the Figure 6.17. The following combustion parameters can be defined from the crank angle position of heat release.

![Instantaneous Heat Release Curve]

**Figure 6.17 Combustion parameters**

$\theta_{\text{delay}}$ – Ignition Delay. The crank angle interval between the occurrence of spark and 5% of heat release.

$\theta_{\text{fd}}$ – Flame Development Angle. The crank angle interval between 5% heat release and the time of a small amount of charge burned i.e. 10% heat release angle.

$\theta_{\text{rb}}$ – Rapid Burning Angle. The crank angle interval between 10% heat release angle and 50% heat release angle.
\( \theta_{fb} \) – Flame Burning Angle. The duration for burning 90\% of fuel air charge: i.e. the interval between 10\% and 90\% heat release.

\( \theta_{cd} \) – Combustion Duration. The total duration of combustion from the time of spark to end of combustion i.e. the crank angle interval between spark and 90\% heat release, i.e. \( \theta_{fb} + \theta_{id} + \theta_{delay} \).

6.7.8 Heat Release Analysis of Magnetically Activated Fuel on Catalytic Coated Engines

The above procedure is applied to the base, magnetically activated base and magnetically activated fuel on catalytic coated engines to calculate the heat release rate, crank angle position of heat release values, mass fraction burned and combustion parameters. The following sections compare the base, magnetically activated fuel on base and magnetically activated fuel on catalytic coated engines based on these parameters.

6.7.8.1 Flame burning speed

Mass burning rates are used to calculate the flame burning speed (Heywood 1989). It is defined by,

\[
S_{fb} = \frac{(dm_b / dt)}{\left( \rho_u A_b \right)} \text{ m / s} \quad (6.18)
\]

where, \( dm_b \) is the burning rate, \( \rho_u \) is the unburned mass density and \( A_b \) is the spherical burning area. Unburned mass density is calculated from the closed cycle simulation (Nedunchezhan 2000). The flame is assumed to burn in a spherical shape in which all the burned gases are contained. From the geometry of combustion chamber, the spherical burning area is calculated.
The calculated flame burning speed for base, magnetically activated fuel on base and magnetically activated fuel on catalytic coated engines are plotted in Figure 6.18 as a function of air-fuel ratio.

![Figure 6.18 Variation of burning speed with air-fuel ratio](image)

It can be observed that the flame burning speed increases linearly with the air-fuel ratio, indicating the strong dependence of burning speed with air-fuel ratio. The flame burning speed increases, when the mixture is made fuel rich. BASEMG1 shows 15% improvement over base engine but BASEMGE performs 22% better than base engine. This indicates that the magnetically activated fuel burns better than the base engine at rich and lean side. In the case of catalytic coated engines, ZIRMGE shows 40% improvement over base engine and 20% over BASEMGE, indicating faster combustion. It is observed that ZIRMGE performs better in all aspects other than richer side. This is due to fuel flow characteristics which can be solved by introducing fuel recirculation unit before carburetor. COPPMGE also performs better in leaner side compared to other categories of engine.
6.7.8.2 Crank angle of heat release values

The crank angle position of different heat release values are presented in Figures 6.19 to 6.22. The variation of crank angle position of 5% heat release is plotted in Figure 6.19. The CAQ5 is a measure of ignition delay, which is less for the magnetically activated fuel on catalytic coated engines as seen in the Figure 6.19. The base engine shows a lower value of CAQ5 near the stoichiometric. Whereas in the magnetically activated fuel on catalytic coated engines, CAQ5 is lower at richer air-fuel ratios. At the reach of lean limit, CAQ5 increases, indicating the increased delay in igniting the mixtures. All the categories show similar trend that lower in ignition delay at stoichiometric air-fuel ratio and then increased towards leaner operation. ZIRMGE proves that there is a negative range of delay period showing no delay in ignition but just combustion.

The variation of CAQ10 with air-fuel ratio is plotted in Figure 6.20. It can be observed that CAQ10 is lower at richer side and increased up to the leaner limit. ZIRMGE shows good improvement over base and catalytic engine. This is due to catalytic activation of charge with support of magnetic field strength.

The corresponding CAQ50 variation is shown in Figure 6.21 for different categories of the engine. The CAQ50 position corresponds to the flame propagation period. The flame propagation depends upon the flame speed (Keck et al 1987). Higher the flame speed earlier will be the occurrence of 50% heat release. This is reflected in Figure 6.21, whereas the CAQ50 of ZIRMGE engine is 25% lower than the base engine and 10% lower than the BASEMGE category.
Figure 6.19 Variation of crank angle of 5% heat release with air-fuel ratio

Figure 6.20 Variation of crank angle of 10% heat release with air-fuel ratio
For every category, the BASEMG1 and BASEMG2 show around 20.2% poor performance than the BASEMGE and ZIRMGE. This is due to the low gauss magnetic activation and their poor support for combustion. BASEMG1 is with 3000 gauss magnet fitted in the fuel line close to the carburetor causing no significant improvement in all the properties charted in Figures 6.18-6.23. In the graph it is clearly shown that only high gauss magnet suits for the improvement but care must be taken to safeguard the instruments or else this will permanently damage the system or will prepare for malfunction.

The variation of CAQ90 is plotted in Figure 6.22 for different air-fuel ratios. The CAQ90 represents the end of combustion. From the Figure 6.22, it can be observed that the combustion ends much earlier for the magnetically activated lean burn engine at all air-fuel ratios. This indicates shorter combustion duration compared to the base engine.
The earlier occurrence of CAQ90 for magnetically activated fuel on catalytic coated engines may be due to the increased flame speed. De-clustering the fuel molecules with the help of high gauss magnetic flux causes the flame speed. CAQ90’s improvement for ZIRMGE is around 25% than the base engine and 14.5% than BASEMGE. In the chart, the COPPMGE engine shows that there is about 10% improvement over base engine but lesser than ZIRMGE category. The BASEMG1 is having only 2% improvement over the base engine. The heat is released faster in the case of catalysts due to catalytic activity of charge.

Figure 6.22 Variation of crank angle of 90% heat release with air-fuel ratio

Among the categories of the engine with different gauss values of magnet fitted on the fuel line close to the carburetor, ZIRMGE shows a faster combustion followed by COPPMGE. BASEMGE also touches the good part of performance parameters in non catalytic range.
6.7.8.3 Combustion parameters

The variation of combustion parameters such as ignition delay, flame development period, rapid burning period, flame burning period and combustion duration are plotted for different air-fuel ratios in Figures 6.23 to 6.27. These combustion parameters are defined in Section 6.7.7.

Figure 6.23 shows the variation of ignition delay with air-fuel ratio for base, magnetically activated fuel on base and magnetically activated fuel on catalytic coated engines. The ignition delay of base engine is higher than that of catalytic engines at all air-fuel ratios. ZIRMGE category engine shows 12.2% lower ignition delay among the catalytic engines followed by COPPMGE engine of 8.5% lower than the base engine. It is apparently visible from the graph that the ignition delay is lower at the leaner operation for all the categories of the engine.

![Figure 6.23 Variation of ignition delay with air-fuel ratio](image)
The flame burning duration, which is the duration for heat release between 5% and 10%, is plotted in Figure 6.24. It can be observed that catalytic engines show lower $\theta_{fd}$ compared to the base engine in the lean range. Whereas, in the rich range the $\theta_{fd}$ of base engine is lower than the catalytic engines. Almost all the catalytic engines show similar values of $\theta_{fd}$. The graph shows that ZIRMGE engine performs 14.3% improvement in flame burning duration compared to the base engine. COPPMGE improves 9.7% and BASEMGE shows 7.2% progress compared to the base engine. This happening is due to the breaking of hydrocarbon molecules into a number of smaller sizes by high gauss magnetic field. These smaller size molecules have more contact surface area to bond with oxygen molecules and hence the burning duration is reduced.

![Figure 6.24 Variation of flame development angle with air-fuel ratio](image)

The corresponding rapid burning periods are plotted in Figure 6.25. The $\theta_{rb}$ is the duration for heat release between 10% and 50%. During this period, the heat release occurs in a rapid manner (Tagalingan 1986). It can be
observed, that the ZIRMGE and COPPMGE show 3.5% lower than the base engine. The $\theta_{rb}$ of BASEMGE engine is 1.4% higher than the magnetically activated fuel on catalytic coated engines. The $\theta_{rb}$ is initially higher at richer side then reaching the lowest at the stoichiometric and then improves at the leaner side more than the richer side. The same trend is maintained by almost all the categories of engines tested.

![Figure 6.25 Variation of rapid burning angle with air-fuel ratio](image)

Figure 6.25 illustrates the variation of $\theta_{rb}$, which is the duration for 10% and 90% heat release. A clear trend can be seen, where the $\theta_{rb}$ is higher in the lean range, decreases near the stoichiometric value and again increases in the rich range. The trend is common for all the category of the engines tested. In addition, $\theta_{rb}$ of the base engine is 26% higher compared to the remaining categories of the engines over the air-fuel ratio tested. COPPMGE shows 24.3% lowest among the other categories of the engines considered.
Similar trend is reflected in Figure 6.27, where the variation of combustion duration is plotted over the air-fuel ratios. It can be observed that ZIRMGE category engine shows 17.4% shorter $\theta_{cd}$ compared to the base engine. Also, the base engine has a higher $\theta_{cd}$ compared to the other catalytic and non-catalytic engines. It is observed that all categories of engines show almost 12.3% reduction in combustion duration compared with the base engine. One possible reason for this could be the acceleration of combustion by active radicals due to magnetic and catalytic activation of charge prior to combustion.
The instantaneous heat release rate and cumulative heat release values are recorded for all the categories of engine with the help of AVL Indimeter 617 and the same is shown in Figures 6.28 - 6.33. In Figures 6.28, 6.29 and 6.30, the variation of heat release pattern with respect to air-fuel ratio for different categories of the engine is apparently shown. ZIRMGE shows good effect on rich mixture and significantly proves that the variation is remarkable compared to the base engine. The high gauss magnetic effect on these categories of engines invariably proves better than the base engine.
Figure 6.28 Variation of instantaneous heat release rate and cumulative heat release value of BASE engine at 3000 rpm with crank angle for different air-fuel ratios.
Figure 6.29  Variation of instantaneous heat release rate and cumulative heat release value of BASEMGE engine at 3000 rpm with crank angle for different air-fuel ratios
Figure 6.30  Variation of instantaneous heat release rate and cumulative heat release value of ZIRMGE engine at 3000 rpm with crank angle for different air-fuel ratios
The same data are plotted in different phases so as to understand the magnetic effect on BASE, BASEMG1, BASEMG2, BASEMGE, COPPMGE and ZIRMGE and are shown in Figures 6.31, 6.32 and 6.33. The graphs show that the rate of heat release at lean side of the magnetically activated fuel on catalytic coated engine is better than the rich side of the non catalytic engine. It can be noticed that heat release value of magnetically activated non catalytic engine is higher than the catalytically activated engine for the same air-fuel ratio. The variation in rate of heat release is higher at lean side and lower at rich side.

6.8 SUMMARY

Following are some of the important points arrived at from the above discussions:

- The heat release code developed is able to account for 97% of heat release during combustion.

- The results of crevice flow model show that as much as 15% of the cylinder charge is trapped by the crevice volume at the time of peak cylinder pressure.

- Although most of the crevice mass return back during the expansion process, they escape the main combustion process and hence do not contribute to useful work.

- The modified correlation proposed for predicting the total heat release is more accurate.
Figure 6.31  Variation of Instantaneous heat release rate and cumulative heat release value with crank angle for different stages of engine at 3000 rpm and an air-fuel ratio of 11.1:1
Figure 6.32  Variation of instantaneous heat release rate and cumulative heat release value with crank angle for different stages of engine at 3000 rpm and an air-fuel ratio of 15.3:1
Figure 6.33  Variation of instantaneous heat release rate and cumulative heat release value with crank angle for different stages of engine at 3000 rpm and an air-fuel ratio of 16.7:1
• The heat transfer effect plays an important role and contributes to substantial loss of heat release during combustion.

• The mass fraction burned profile is represented by the Viebe form. The parameters ‘a’ and ‘m’ are fitted from the experimental values as a function of air-fuel ratios.

• From the crank angle of heat release values, various combustion parameters are defined to represent different combustion phases.

• The flame burning speed is calculated for base, magnetically activated fuel on base and magnetically activated fuel on catalytic coated engines. This parameter indicates that faster combustion is taking place in magnetically activated fuel on catalytic coated engines.

• The crank angle of heat release values and combustion parameters indicate that a faster heat release occurs for ZIRMGE engine followed by COPPMGE. The combustion duration of base engine is higher than that of the other categories of engines tested.

The heat release analysis is carried out for base, magnetically activated fuel on base and magnetically activated fuel on catalytic coated engines. These results prove that magnetically activated fuel on catalytic coated engine have reduced ignition delay and faster combustion.