NUMERICAL COMBUSTION ANALYSIS AND IGNITION TIMING OPTIMIZATION OF 4 STROKE SI ENGINE

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ABSTRACT

The design and manufacture of IC Engines is under significant pressure for improvement. Engine designing is a herculean task, as each stroke needs to be carefully studied before manufacturing. Computational Fluid Dynamics (CFD) is a powerful tool used for predicting the behaviour of difficult problems in a cost effective way and the solutions converging are approximately close to the original. Optimal ignition timing is necessary for producing maximum gas pressure and low brake specific fuel consumption (BSFC) for a particular speed and it is obtained by finding maximum brake torque (MBT). In this work, the numerical combustion analysis of a single cylinder 4 stroke SI engine at different ignition timings for a particular speed (4000 rpm) is conducted and the optimum spark timing is identified using genetic algorithm (GA).

Key words: CFD, MBT, SI engine and GA optimization


1. INTRODUCTION

IC engines are part of human world when Otto developed the first spark ignition engine (SI) and Diesel developed compression ignition (CI) during 19th century. Today, all engine manufactures are given priority to the improvement of engine rather than developing a newer one. Advancement of computational fluid dynamics (CFD) gives an opportunity to all aspiring peoples in the field of IC engine to predict the
behaviour and optimization of same before manufacturing. By aiming optimization of different aspects of engine means it doesn’t affect the regular working of engine and also gives better results. Optimization of spark timing can result in flame front to travel the least distance and consumes the mixture as fast as possible; thereby gas pressure and torque get there optimal range.

The ignition timing can be used as an alternative way for predicting the performance of internal combustion engines and also volumetric efficiency, BMEP have increased with rising ignition timing [1]. The optimization of the spark timing and the air-fuel ratio, and timing the opening of inlet and exhaust valves in an SI engine equipped with VVT system will lead to a lower BSFC and higher torque at all engine speeds [2]. Both numerical simulations and theoretical tools are used to optimize the performance of a spark ignition engine by analyzing the influence of some key parameters spark advance angle, fuel ratio, and cylinder internal wall temperature in the operation of the engine [3]. The load carrying capacity of the engine reduces as the ignition timing is advanced and retarded from 320° CA due to increase of flame speed and decrease in its volumetric efficiency [4].

In this numerical analysis, the premixed combustion inside the combustion chamber (2D) of a 4 stroke SI engine at various spark advance angles (spark timing) for a particular speed (4000 rpm) are studied. And from the results of combustion analysis the optimization of spark timing are done using Genetic algorithm.

2. METHODOLOGY
2.1. Modelling
The 2D combustion chamber of the 4 stroke SI engine is modelled and meshed using Ansys workbench 15.0.7 and the geometric specification of 4 stroke SI engine is summarized in the Table 1.

| Bore, B in mm | 63 |
| Stroke, S in mm | 56 |
| Connecting rod length, L<sub>cr</sub> in mm | 98.24 |
| Compression ratio, ε | 8.5 |
| Type | 175 cc, Single cylinder 4 stroke SI engine [7] |

![Combustion chamber 2D model prepared in Ansys workbench](http://www.iaeme.com/IJMET/index.asp)
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The combustion chamber is designed by considering the cylinder, cylinder head and piston surfaces which is shown in the Figure 1 and the mesh on the designed surface is shown in Figure 2. After meshing, the file is transferred to the Ansys fluent 15.0.7 for numerical analysis.

2.2. Numerical Analysis

In a 4 stroke engine, the cycle of operation is completed in 4 strokes of piston or two revolutions (720°) of the crank shaft and the starting and ending of one cycle of operation is taken as 360° and 1080°. Out of the four strokes only compression (starts at 600° CA) and expansion strokes (ends at 840° CA) are take part in the combustion process. So, only these strokes are considered while doing the analysis.

Throughout the numerical analysis engine speed is taken as 4000 rpm. It is because, during a normal ride period, most of the time engine runs in the range of 4000 rpm; so, at this particular speed engine needs to be optimized.

Heat analysis (Analytical method) is conducted for obtaining the different operating and boundary conditions of engine and it is shown below along with other several assumptions.

- **Solver**: pressure based solver
- **Viscous model**: standard k-ε, standard wall fn
- **Species**: premixed combustion.
- **Material**: air-fuel mixture
- **Boundary conditions**: piston (T = 873 K), cylinder and cylinder head (T = 463 K)
- **Pressure velocity coupling scheme**: PISO
- **Pressure discretization**: PRESTO
- **Initial pressure**: 0.125 MPa (at 600° CA)
- **Initial temperature**: 361.47 K

The same heat analysis is used for validating the numerical analysis and it is done by comparing in cylinder combustion peak pressure values and temperature. The experiment is conducted for various spark advance angles (spark timing) ranges from 670° to 705° having an interval of 5° and the peak pressure values and the corresponding peak pressure crank angles are taken for optimization.

Figure 2 Combustion chamber after meshing
2.3. Optimization of spark timing using GA

The optimum timing which gives the maximum brake torque called maximum brake torque, or timing (MBT), and timing which is advanced or retarded from this optimum gives lower torque [6]. The turning moment (torque) is calculated by using the peak pressures and the various peak pressure crank angles and it is done using the following equation.

\[ \text{Tangential pressure, } P_T = \frac{[(p_g + p_i) \sin (\phi + \beta)]}{\cos \beta} \text{ in MPa} \]  

\[ \text{Turning moment, } M = P_T \times F_p \times R \text{ in Nm} \]  

Where, \( p_g \) = in cylinder combustion peak pressure in MPa, \( p_i \) = pressure due to inertial effect of reciprocating masses in MPa, \( \phi \) = peak pressure crank angle in degrees, \( \beta \) = angle between connecting rod and cylinder axis in degree.

The torque and the corresponding spark advance angles are plotted and the resulting 6 degree polynomial curve equation are used to predict the optimum spark timing which is done with the help of Matlab genetic algorithm (GA) toolbox.

3. RESULTS AND DISCUSSION

3.1. Model validation

Numerical analysis results needed to be validated and here, it is done using analytical method results. Table 2 shows the peak pressure values of both numerical and analytical method at spark advance angle of 35° before TDC (685° CA). From the result obtained, it can say that numerical results are in good match with analytical results having error percentage of about 0.003%.

**Table 2** Peak pressure values of both numerical and analytical method at spark advance angle of 35° before TDC

<table>
<thead>
<tr>
<th>Spark timing</th>
<th>Peak pressure, MPa</th>
<th>Absolute Error</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Numerical method</td>
<td>Analytical method</td>
</tr>
<tr>
<td>685 (35° BTDC)</td>
<td>6.4319</td>
<td>6.4077</td>
</tr>
</tbody>
</table>

3.2. Numerical Analysis and optimization using GA

The spark advance angle, peak pressure, peak pressure angle and torque are shown in Table 3. At spark timing 40° before TDC, the torque shows maximum value and it can say that MBT may be at 40° before TDC or angle close to this angle.

**Table 3** peak pressure, peak pressure angle and torque at different spark advance angle

<table>
<thead>
<tr>
<th>Spark timing</th>
<th>Peak pressure, MPa</th>
<th>Pressure Peak Angle, ( \phi ) (ATDC)</th>
<th>Torque (M), Nm</th>
</tr>
</thead>
<tbody>
<tr>
<td>670 (50° BTDC)</td>
<td>7.965411</td>
<td>12</td>
<td>184.90</td>
</tr>
<tr>
<td>675 (45° BTDC)</td>
<td>7.623077</td>
<td>13.5</td>
<td>198.45</td>
</tr>
<tr>
<td>680 (40° BTDC)</td>
<td>6.823003</td>
<td>16</td>
<td>209.18</td>
</tr>
<tr>
<td>685 (35° BTDC)</td>
<td>5.620062</td>
<td>19</td>
<td>202.85</td>
</tr>
<tr>
<td>690 (30° BTDC)</td>
<td>4.887985</td>
<td>20.5</td>
<td>189.42</td>
</tr>
<tr>
<td>695 (25° BTDC)</td>
<td>3.894717</td>
<td>21</td>
<td>154.39</td>
</tr>
<tr>
<td>700 (20° BTDC)</td>
<td>3.055303</td>
<td>16</td>
<td>93.67</td>
</tr>
<tr>
<td>705 (15° BTDC)</td>
<td>2.563677</td>
<td>6</td>
<td>29.97</td>
</tr>
</tbody>
</table>
The torque vs. spark advance angle is shown in Figure 3. A trend line is plotted for obtaining curve equation of the torque vs. spark advance angle for doing optimization using GA algorithm. The polynomial equation having order 6 of the trend line is shown below.

\[ y = 0.000002891x^6 - 0.0119088x^5 + 20.4333799x^4 - 1697.50899x^3 + 9623331.6841x^2 - \\
2641445283.6626x + 302080267421.165 \]

Figure 3 Torque vs. Spark advance angle

The torque from numerical analysis and torque from GA optimization (by doing it on the GA toolbox of Matlab) are compared in Table 4. From the results, Torque is at its peak (MBT), when spark advance angle is at 681° CA (39° BTDC) and it should be the optimal spark timing to produce maximum brake torque.

Table 4 Torque from Numerical analysis and GA optimization

<table>
<thead>
<tr>
<th>Spark timing</th>
<th>Torque from numerical analysis in Nm</th>
<th>Torque using GA algorithm in Nm</th>
<th>Absolute Error %</th>
</tr>
</thead>
<tbody>
<tr>
<td>670 (50° BTDC)</td>
<td>184.90</td>
<td>183.583</td>
<td>0.71</td>
</tr>
<tr>
<td>675 (45° BTDC)</td>
<td>198.45</td>
<td>200.955</td>
<td>1.26</td>
</tr>
<tr>
<td>680 (40° BTDC)</td>
<td>209.18</td>
<td>208.555</td>
<td>0.30</td>
</tr>
<tr>
<td>681 (39° BTDC)</td>
<td>210.643</td>
<td>208.72</td>
<td>0.91</td>
</tr>
<tr>
<td>685 (35° BTDC)</td>
<td>208.85</td>
<td>204.313</td>
<td>0.71</td>
</tr>
<tr>
<td>690 (30° BTDC)</td>
<td>189.42</td>
<td>186.155</td>
<td>1.72</td>
</tr>
<tr>
<td>695 (25° BTDC)</td>
<td>154.39</td>
<td>152.012</td>
<td>1.54</td>
</tr>
<tr>
<td>700 (20° BTDC)</td>
<td>93.67</td>
<td>99.81</td>
<td>6.55</td>
</tr>
<tr>
<td>705 (15° BTDC)</td>
<td>29.97</td>
<td>27.478</td>
<td>8.32</td>
</tr>
</tbody>
</table>

The peak pressure vs. Crank angle (CA) for three different spark timings are shown in the Figure 4. It is clear from the figure that when spark timing is advanced (at CA 670), high rate of combustion occurs which result in high rate of pressure rise producing higher peak pressure at a point closer to TDC (12° ATDC). It increases force as well as power output of the engine but, it may result in knocking and rough running of the engine because of vibrations produced in the crankshaft rotation. If spark timing is retarded (CA 695), longer time is required for combustion and also reduces peak pressure. So, there is a compromise between these opposing factors is needed for obtaining peak pressure close to the beginning of power stroke, yet maintaining smooth engine operation and optimum spark timing (CA 681) fills the
The peak pressure and its corresponding angle at optimal spark timing is 7.3095 MPa and 737° CA (17° ATDC).

Figure 4 Pressure vs. Crank angle

Figure 5a shows the temperature in the cylinder at 677° CA, and the temperature was about 600 K throughout the chamber. The air–fuel mixture is ignited at 681° CA. At 737°CA (Figure 5b), the cylinder temperature has risen to about 2490 K due to the combustion of fuel and thermal energy in the cylinder is converted into the mechanical energy of the piston. The temperature of the gas in the cylinder therefore decreases. Figure 5c shows the temperature contour at 845° CA; the mean temperature is about 1400 K. Then the temperature further decreases due to heat loss to the cylinder walls and the loss of combustion gas to the exhaust.

Figure 6 shows the mass fraction burned (MFB) profile, when engine runs according to the optimized spark timing (39° BTDC). The MFB profile provides a convenient basis for combustion characterisation, which divides the combustion process in its significant intervals, flame development (first 10%), rapid burning (between 10 and 90%) and combustion termination (remaining 10%), in the CA domain. It can also see that half of the charge (about 53%) is burned 10° ATDC, which is one of the requirement of optimal spark timing.
4. CONCLUSIONS
The following are the conclusions obtained by conducting this study:

- Numerical analysis combined with genetic algorithm promises a valid methodology for obtaining the optimal spark timing.
The optimized spark timing for a single cylinder engine (175 cc) running at a speed of 4000 rpm is 681° CA (39° BTDC) and it can say that the maximum brake torque will be at this spark timing.

The peak pressure (7.3095 MPa) obtained at the optimal sparking condition enables smooth engine operation and the corresponding peak pressure angle (17° ATDC) is close to beginning of power stroke.

At 10° ATDC, the burned mass fraction is about 53 %. It can say that half of the air fuel mixture is burned close to TDC and may also considered as one of the empirical rule for spark timing optimization.

REFERENCES


