Combustion characteristics and particulate emission in a natural-gas direct-injection engine: effects of the injection timing and the spark timing

Y-F Liu, B Liu, L Liu, K Zeng*, and Z-H Huang
State Key Laboratory of Multiphase Flow in Power Engineering, Xi’an Jiaotong University, Xi’an, People’s Republic of China

**Abstract:** In the present study, the effects of injection timing and spark timing on combustion characteristics and particle emission were investigated. The particulate number concentration and size distribution were measured using an electrical low-pressure impactor. The combustion parameters were calculated from the cylinder pressure data. The results indicate that the combustion versus the injection timing is primarily dependent on the mixing quality of the air–fuel mixture. The particulate number concentration increases as the fuel injection is advanced but the particle size distribution is not affected significantly with different injection timings. This is probably related to the in-cylinder combustion. The initial combustion duration increases as the spark timing is advanced and the fastest rapid combustion duration is obtained at the maximum brake torque spark timing. As the fuel injection timing and spark timing are advanced, the particle number levels are increased but the particle size distribution shows few variations. Most of the particulates are in the nanoparticle size range. The amounts of nitrogen oxide and hydrocarbon emissions increase with advanced fuel injection timing and spark timing, and the carbon monoxide concentration experiences small variations under all operation conditions.

**Keywords:** natural gas, direct injection, injection timing, spark timing, combustion, particulate, emissions

1 INTRODUCTION

With increasing concern about the limited reserve of crude oil and about atmosphere pollution, research on improving engine efficiency and emissions has become the focus of recent engine studies. Because there are relatively more reserves of compressed natural gas (CNG) and it has lower emissions, CNG is considered to be the most promising alternative fuel for engine applications. Moreover, because of its high octane number, CNG has a good anti-knock property and hence provides engines with the capability to operate under even a high compression ratio, leading to higher power output and higher combustion efficiency. Nowadays, homogeneous charge spark ignition (SI) CNG engines have already been widely applied in city buses and taxies. However, as natural gas occupies some air displacement in the intake manifold, it causes lower volumetric efficiency and decreases the engine power output. This effect is more obvious under high-load conditions [1]. In fact, more benefits could be achieved if the application of natural gas is combined with a lean-burn strategy. The lean-burn strategy can realize a higher thermal efficiency owing to lower pumping loss, lower heat transfer loss, and an increase in the specific heat ratio. Furthermore, nitrogen oxide (NOx) emission and the likelihood of knock can be reduced as a result of the decreased cylinder temperature which results from dilution. However, large cyclic variation and incomplete combustion limit the lean-burn operations [2–8].
To utilize fully the potential of natural gas in engine applications, the concept of CNG direct-injection (DI) combustion has been investigated in several approaches including engine studies, basic studies of rapid compression machines (RCM), or combustion bomb experiments [9–15]. Huang et al. [9] investigated the influence of injection timing on the combustion and emission characteristics using an RCM and revealed the importance of injection timing on stratified charge combustion. Huang et al. [10] studied the combustion behaviours of a direct-injection spark ignition (DISI) engine fuelled with CNG and hydrogen blends under various ignition timings and found that the brake mean effective pressure and the effective thermal efficiency increased when the combustion duration decreased, with decreasing interval times between the end of fuel injection and the start of ignition. Hassen et al. [11] modified a single-cylinder gasoline engine into a CNG DI engine and they reported that the optimized CNG DI engine can achieve a higher brake power and lower NOx emission with moderate increases in the hydrocarbons (HCs) and carbon monoxide (CO) compared with the original gasoline engine. However, the in-cylinder air–fuel mixing process could be easily affected by many parameters such as the compression ratio, combustion chamber geometry, injection pressure, nozzle-hole numbers and arrangement, and swirl intensity. More optimization work should be carried out to improve the combustion and emission performance of CNG DISI engines.

In the last two decades, particle emission from internal combustion engines has generated much interest. This is primarily due to the recent evidence which suggests that nanoparticles (particles smaller than 50 nm in diameter), which have the ability to penetrate deep into the interstitial tissue of the lungs, can have a negative effect on human health [16–18]. The new ambient particle standard adopted by the US Environmental Protection Agency (for particles smaller than 2.5 μm in diameter (PM2.5)) has led to more stringent future control for particle emissions. Currently, particle emissions from internal combustion engines have been regulated in terms of the total particle mass emitted per kilometre. This regulation is effective to control large-sized particle emissions because the finer particle emission will contribute little to the total particle mass. Unfortunately, these reductions in the total mass of particle emissions may have been accompanied by a dramatic increase in the particle number, especially in the ultrafine particle range (diameter below 0.1 μm). Diesel engines have been considered to be a major source of particle emissions produced by on-road vehicles. This is because mass-weighted particle emissions from SI engines are traditionally insignificant when compared with those from diesel engines. However, because a large fraction of particle emissions from SI engines are in the ultrafine size range, and because the number of miles travelled by gasoline-fuelled vehicles in urban areas is predominant, particle emissions from SI engines may still be significant [19, 20]. Direct in-cylinder fuel injection produced a lean and stratified fuel–air mixture (as opposed to port fuel injection (PFI)), and the local rich areas of combustion may result in significantly higher particle emissions. Marić et al. [21, 22] examined the particle number concentration and size distribution from both a single-fluid and an air-assisted gasoline DISI engine over a range of operating conditions. They found that the particle emissions exhibited a strong sensitivity to the injection timing and that the particle number concentration generally increased with retarded injection timing except over a narrow portion of the range where the trend reversed. Time-resolved measurement was also made on a late model vehicle equipped with a DISI engine. The experiment results showed that most of the particles emitted at medium and high vehicle speeds were in accumulation mode and only a small fraction of the total particles were in the nanoparticle size range [23]. The previous work concentrated mainly on gasoline DISI engines and not much has been reported in the literature on emission characteristics, and in particular on particle emissions from CNG DISI engines. CNG has a different HC ratio from gasoline and thus a different soot tendency. Furthermore, with gas fuel directly injected into the cylinder, the effect of liquid fuel impingement, which is likely to be an important mechanism for particulate matter (PM) formation of gasoline DISI combustion, can be isolated. Therefore, there may be some differences between particle emission in gasoline DISI engines and that in CNG DISI engines.

The present study aims to investigate the influences of injection timing and spark timing on the combustion and emission (particularly particle emission) characteristics of a modified CNG DISI engine. Sturgess et al. [24] investigated the relationship between the in-cylinder combustion and the particle number and distribution in a PFI gasoline engine. They revealed that there was a strong correlation between the geometric mean diameter (GMD) of particles and the burning rate during the
early stage of combustion. When the 50 per cent mass fraction burned (MFB50%) occurred far way from the top dead centre (TDC), the GMD of the particles was larger for similar load conditions. In this paper, the combustion parameters will also be calculated to study the relationship between in-cylinder combustion and particle emission.

2 EXPERIMENTAL SET-UP AND PROCEDURE

The experimental set-up is shown in Fig. 1. The test engine was converted from a single-cylinder DI diesel engine. The specifications of the engine are listed in Table 1. To maximize the engine power output, the compression ratio was selected to be 14.

Table 1 Engine specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of cylinders</td>
<td>1</td>
</tr>
<tr>
<td>Bore</td>
<td>100 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>115 mm</td>
</tr>
<tr>
<td>Length of the connecting rod</td>
<td>190 mm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>14</td>
</tr>
<tr>
<td>Displacement L</td>
<td>0.903</td>
</tr>
<tr>
<td>Ignition source</td>
<td>Spark plug</td>
</tr>
<tr>
<td>Injection pressure</td>
<td>8 MPa</td>
</tr>
<tr>
<td>Inlet valve opening</td>
<td>11° CA before top dead centre (BTDC)</td>
</tr>
<tr>
<td>Inlet valve closure</td>
<td>49° CA after bottom dead centre (ABDC)</td>
</tr>
<tr>
<td>Exhaust valve opening</td>
<td>52° CA before bottom dead centre (BBDC)</td>
</tr>
<tr>
<td>Exhaust valve closure</td>
<td>8° CA after top dead centre (ATDC)</td>
</tr>
</tbody>
</table>

The injector used in this experiment is a modified version from a gasoline DI engine made by the manufacturer Hitachi [25]. Figure 2(a) is a close-up photograph of the injector used in the experiment. To ensure the penetration capability of gas fuel injection, the swirler near the tip of the nozzle was removed. Figure 2(b) shows the conventional swirl-type fuel injector. The injection pressure was regulated at 8 MPa, since the gas velocity from the injector nozzle is kept at a constant value of the sonic velocity due to the choke condition during fuel injection; thus, the amount of injected fuel is determined by the injection duration. The in-cylinder pressure data were obtained using a Kistle...
6117B piezoelectric pressure transducer which is integrated with the spark plug, and 100 continuous cycles were recorded for analysis of the combustion parameters. A free-programmed engine control system was used for engine operation control. The injection timing, injection duration, and spark timing were controlled by the electronic control unit and can be adjusted in the experiment. The composition of CNG is given in Table 2. CNG was supplied to the fuel injector through a pressure regulator. A Horiba 7100 exhaust analyser was used to measure the exhaust HC, CO, carbon dioxide (CO₂), and NOₓ concentrations, and the measurement has a resolution of 1 p/min for HC, 0.01 per cent for CO, 0.01 per cent for CO₂, and 1 p/min for NOₓ.

The particle number and size distribution was measured using the electrical low-pressure impactor (ELPI) with an extra filter stage. The specifications of the ELPI are listed in Table 3. The ELPI is designed for real-time monitoring of the aerosol particle size distribution. The operating principle of the ELPI is based on the particulate charge, inertial classification in a cascade impactor, and electrical detection of aerosol particles. The detailed process and conditions as well as restrictions of particle measurement using an ELPI can be found in references [26] to [28]. In order to avoid overloading the ELPI’s electrometer, the exhaust gas was diluted with clean air using a two-stage diluter (DI-2003) manufactured by Dekati. To eliminate the unwanted particle condensation and nucleation during dilution, the first stage of the diluter is heated and the second stage of the diluter operates at ambient temperature. Although the ELPI with a filter stage can measure the particle size distribution in the size range from 7 nm to 10 μm, the present study focuses on particles in only the PM₂.₅ range. Therefore, data from stage 1 to stage 11 were recorded. By weighting the distributions by $D_{50%}^3$, where $D_{50%}$ is the particle diameter, and by assuming a particle density of unity, the distribution of the particle mass versus the size was obtained [21].

In the present work, the test consists of two parts. The first part is to examine the effect of the injection timing on the particle emission and combustion characteristics. Injection timings were varied in increments of 10° crank angle (CA) from 170° CA before top dead centre (BTDC) to 210° CA BTDC, while the spark advance was fixed at 32° CA BTDC instead of maximum brake torque (MBT) [21, 22] to isolate the effect of the spark timing. In the second part, to investigate the particle emission and combustion characteristics under different spark advances, the injection timing was fixed (180° CA BTDC) and the ignition timing was varied in increments of 4° CA from 24° CA BTDC to 40° CA BTDC. The engine was warmed up until the cooling-water temperature rose to 85°C and the lubricating oil temperature rose to 70°C. The throttle was set to maintain a relatively constant engine load. Figure 3

---

**Table 2** Composition of natural gas

<table>
<thead>
<tr>
<th>Gas</th>
<th>Amount (vol. %)</th>
</tr>
</thead>
<tbody>
<tr>
<td>CH₄</td>
<td>96.16</td>
</tr>
<tr>
<td>i-C₅H₁₀</td>
<td>0.021</td>
</tr>
<tr>
<td>n-C₅H₁₀</td>
<td>0.005</td>
</tr>
<tr>
<td>H₂S</td>
<td>0.0002</td>
</tr>
<tr>
<td>C₃H₈</td>
<td>1.996</td>
</tr>
<tr>
<td>i-C₅H₁₀</td>
<td>0.021</td>
</tr>
<tr>
<td>N₂</td>
<td>0.001</td>
</tr>
<tr>
<td>H₂O</td>
<td>0.006</td>
</tr>
<tr>
<td>C₃H₈</td>
<td>0.136</td>
</tr>
<tr>
<td>i-C₅H₁₂</td>
<td>0.006</td>
</tr>
<tr>
<td>CO₂</td>
<td>2.54</td>
</tr>
</tbody>
</table>

**Table 3** ELPI specifications

<table>
<thead>
<tr>
<th>Stage</th>
<th>$D_{50%}$ (μm)</th>
<th>$D_i$ (μm)</th>
<th>Minimum</th>
<th>Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.007</td>
<td>0.0208</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>0.029</td>
<td>0.0392</td>
<td>90</td>
<td>9 x 10⁵</td>
</tr>
<tr>
<td>3</td>
<td>0.057</td>
<td>0.0715</td>
<td>50</td>
<td>5 x 10⁶</td>
</tr>
<tr>
<td>4</td>
<td>0.101</td>
<td>0.1208</td>
<td>26</td>
<td>3 x 10⁶</td>
</tr>
<tr>
<td>5</td>
<td>0.165</td>
<td>0.2036</td>
<td>15</td>
<td>2 x 10⁶</td>
</tr>
<tr>
<td>6</td>
<td>0.255</td>
<td>0.3188</td>
<td>9</td>
<td>9 x 10⁵</td>
</tr>
<tr>
<td>7</td>
<td>0.393</td>
<td>0.4874</td>
<td>5</td>
<td>5 x 10⁵</td>
</tr>
<tr>
<td>8</td>
<td>0.637</td>
<td>0.7672</td>
<td>3</td>
<td>3 x 10⁵</td>
</tr>
<tr>
<td>9</td>
<td>0.99</td>
<td>1.2393</td>
<td>1.6</td>
<td>2 x 10⁵</td>
</tr>
<tr>
<td>10</td>
<td>1.61</td>
<td>1.9657</td>
<td>0.8</td>
<td>8 x 10⁴</td>
</tr>
<tr>
<td>11</td>
<td>2.46</td>
<td>3.1061</td>
<td>0.36</td>
<td>4 x 10⁴</td>
</tr>
</tbody>
</table>

*D₅₀% represents the particle size with 50 per cent collection efficiency.

$D_i$ represents the geometric mean of the $D_{50%}$ values.

---

**Fig. 3** IMEP versus operation conditions
illustrates the indicated mean effective pressure (IMEP) versus the operation conditions. The engine speed was fixed at 1200 r/min and the fuel injection duration was kept constant (11.96 ms). A thermodynamic model is used to calculate the combustion process parameters. The model neglects the leakage through the piston rings \cite{29} and thus the energy conservation in the cylinder can be written as

\[
\frac{dQ_B}{d\theta} = mT \frac{dC_V}{d\theta} + p \frac{C_p}{R} \frac{dV}{d\theta} + \frac{C_V}{R} \frac{dp}{d\theta} + \frac{dQ_W}{d\theta}
\]

where the heat transfer rate is given by

\[
\frac{dQ_W}{d\theta} = h_C A(T - T_W)
\]

The heat transfer coefficient \( h_C \) uses the correction formula given by Woschni \cite{30}.

Properties such as the constant-pressure heat capacity and the constant-pressure heat capacity for species are obtained from the National Aeronautics and Space Administration database \cite{31, 32}. Because of the unknown mass fraction burned at every time step, a simple iterative method is introduced to calculate the thermodynamic parameters of the mixture, which includes the burned and unburned parts.

The flame development duration is defined as the interval of CA from the start of ignition to the angle at which 10 per cent accumulated heat is released. The rapid combustion duration is defined as the interval of CA from the 10 per cent accumulated heat release to the angle of 90 per cent accumulated heat release. The total combustion duration is the duration of the overall burning process and it is the sum of the flame development duration and the rapid combustion duration.

The CA of the heat release curve’s centre is determined by the formula

\[
\theta_c = \frac{\int_{0}^{\theta_s} (dQ_B/d\theta) d\theta}{\int_{0}^{\theta_s} (dQ_B/d\theta) d\theta}
\]

in which \( \theta_s \) is the CA at the beginning of heat release and \( \theta_c \) is the CA at the end of heat release.

3 RESULTS AND DISCUSSION

3.1 Effect of the injection timing

The combustion durations versus the fuel injection timing are illustrated in Fig. 4. The initial combustion duration decreases linearly as fuel injection is advanced. When fuel injection is retarded, the initial combustion duration is expected to be shortened owing to the increased degree of charge stratification and the intensity of turbulence \cite{9, 10}. However, a contradictory trend is observed in the test results. The initial combustion duration increases with advancing fuel injection. The contribution of this contradiction is probably related to the fuel mixing quality under a high compression ratio. With increased compression ratio, the fuel jet penetration is decreased owing to the increased back pressure of fuel injection. The local mixture near the spark plug may be over-rich when fuel injection is retarded. That may slow down the formation of the flame kernel and thus prolong the initial combustion duration. The rapid combustion duration reached a minimum value when fuel was injected at 190° CA BTDC. This suggests that the fastest flame propagation is at this fuel injection timing. The total combustion duration is decreased when fuel injection is advanced. This is mainly due to the decreased initial combustion duration.

Figure 5 gives the location of MFB50% and the centre \( \theta_c \) of heat release for various fuel injection timings. As the fuel injection timing is advanced, the location of MFB50% and \( \theta_c \) become closer to the TDC. This indicates that the compactness of heat release is increased with advancing fuel injection since the spark timing is fixed. Figures 6 and 7 give the particle number and the mass distribution respectively for various fuel injection timings. The peak particle number concentration occurs at a diameter of 0.039 μm and reaches a magnitude of \( 10^7 \) cm\(^{-3} \). This observation is con-
sistent with the results obtained by Ristovski et al. [33], who reported that the count median diameter of particles emitted by CNG engines was in the range between 0.02 μm and 0.06 μm. As fuel injection is retarded, the particle number and mass over all the measured size range decreased. The tendency is different from the test results which were obtained on DISI engines fuelled with liquid gasoline fuel [21, 22, 34, 35]. This difference is probably due to the effect of liquid fuel impingement. Liquid fuel impingement on the piston may be different for different fuel injection timings; the wall-wetting condition and degree of fuel evaporation may also be different in the liquid fuel injection case. Concerning the gaseous CNG DI case, the effects of fuel impingement and evaporation on particle emission can be isolated. Therefore, the local air–fuel ratio distribution in the cylinder may be the dominant parameter creating PM emission in CNG DISI combustion. It was expected that retarded fuel injection would create more PM emission because the mixture inhomogeneity increased. The results obtained do not comply with this expectation yet. This discrepancy can be explained by a combination mechanism of PM behaviour with respect to fuel injection timing. With retarded fuel injection, particle formation may be aggravated by the increasing inhomogeneity of the mixture. However, particle oxidation affected by the temperature and oxygen availability is also promoted. When fuel injection is retarded, more particles may be formed because of the richer local mixture in the cylinder, but the accompanying particle oxidation may also be aggregated because of post-combustion. It seems that, over all the injection timings, particle oxidation was dominant during the competition with particulate formation. For the CNG DI case, there are no liquid fuel droplets distributed in the fuel–air mixture; it may be that the particle formation in terms of mixture inhomogeneity is weaker than in the liquid fuel injection case.

Figure 8 gives the exhaust concentrations versus the fuel injection timing. The exhaust HC emission increased with advancing fuel injection timing. The peak cylinder gas pressure increases as fuel injection is advanced. The higher peak cylinder gas pressure will increase the injected fuel entering into the top-land crevice region and thus produce more unburned HCs. Meanwhile, the lower exhaust temperature at the expansion and exhaust strokes when fuel injection is advanced will also slow down the HC oxidation. These two factors result in an increase in the HC emission. The NOx emission largely increases with advancing fuel injection. A relatively high
burning rate and high in-cylinder temperature are the main contributing factors to the increase in the NO\(_x\) emission. The CO concentration shows small variations and remains low over all injection timings. This is probably due to the relatively high oxygen concentration in the lean-burn mixture.

3.2 Effect of the spark timing

Figure 9 gives the combustion durations versus the spark timing. The initial combustion duration increases when the spark timing is advanced. As the spark timing is retarded, the initial in-cylinder temperature and pressure increase because of more compression work. This effect promotes the formation of the flame kernel and thus shortens the initial combustion duration. The rapid combustion duration first decreases with advancing ignition timing and then slightly increases after reaching a minimum value, whose changing trend is similar to that of PFI engines [8]. When the ignition timing is retarded to the MBT, the higher in-cylinder temperature and pressure reduce the flame propagation duration. When the ignition timing is retarded too far away from the MBT, part combustion may take place in the expansion stroke, where the in-cylinder temperature and pressure are relatively low. This leads to prolonged flame propagation duration. The total combustion duration reaches a minimum value where the MBT spark timing is obtained.

Figure 10 gives the location of MFB50% and the centre \(\theta_c\) of heat release versus the spark timing. Like the case of sweeping injection timing, the location of MFB50% and \(\theta_c\) move close to TDC as the spark timing is advanced.

Figures 11 and 12 illustrate the particle number concentration and mass distribution respectively for various spark timings. When the spark advance was retarded from 40° CA BTDC to 24° CA BTDC, both the particle number and the particle mass concentration decreased. The PM number concentrations at 40° CA BTDC are over one order of magnitude higher than at 24° CA BTDC. However, the shape of the size distribution curve is approximately constant. Both the peak temperature and the peak pressure in the cylinder decrease as the spark timing is retarded. The lower peak temperature results in more serious post-combustion which can post-oxidize the PM in the expansion stroke and exhaust pipe. The lower peak cylinder pressure decreases the HC mass in the crevice volume and so decreases the precursor of...
particulate formation. When the spark timing is retarded, the interval between the end of fuel injection and the start of ignition increased and the time available for mixture preparation was greater; thus PM as well as HCs are reduced owing to the decrease in the inhomogeneity of the mixture.

The exhaust gas concentrations versus the spark timing are given in Fig. 13. The exhaust HC emission increases linearly with advancing spark timing. The result is similar to the HC emission with respect to the injection timing but the increase in the HC concentration is more obvious. When the spark timing is advanced, the unburned HCs in the crevice volume increase owing to the higher peak cylinder pressure. Furthermore, the retarded spark timing promotes post-combustion and thus more unburned HCs can be oxidized in the expansion stroke and exhaust pipe. The NOx concentration increases because of the higher combustion temperature with advanced spark timing. The CO emission also shows few variations when tuning the spark timing.

4 CONCLUSIONS

The influences of the injection timing and the spark timing on combustion and emission, and in particular the particle emission characteristics, were investigated. More advanced fuel injection seems to prepare a better-stratified air–fuel mixture and to achieve the fastest flame development. The combustion characteristics versus the injection timing are primarily dependent on the mixing quality of the air–fuel mixture. Adequate mixture stratification can promote the formation of a flame kernel and thus reduce the initial combustion duration but the rapid combustion duration is prolonged slightly. However, no clear evidence is given for the stratification process and further studies are needed in future work. The combustion durations versus the spark timing are more like the PFI CNG combustion case. The initial combustion duration increases as the spark timing is advanced and the fastest rapid combustion duration is obtained at MBT spark timing. The NOx and HC concentrations increase with advancing injection timing and spark timing. The CO concentration shows little variation during all operation conditions.

The particle emissions of CNG DISI engines show an unexpected trend with various injection timings when the spark timing is fixed. The particle number concentration increases as fuel injection is advanced. This is related to the fuel mixing quality.
which also causes some differences in the combustion characteristics compared with the results obtained before. The particle number concentration versus the spark timing is consistent with the results obtained by other researchers. As the spark timing is retarded, the number concentration decreased. The oxidation effect of post-combustion is responsible for this tendency. The particle size distribution shows little change when tuning the injection timing and the spark timing. Most particles are in the nanoparticle size range. Evidence shows that there is a correlation between MFB50% and the particle number concentration. This can be used as a control variable for controlling directly the particle emission. However, the observation was obtained from a limited number of test results and more comprehensive studies should be carried out for particle control applications.

ACKNOWLEDGEMENTS

This study was supported by the National Natural Science Foundation of China (Grant 50636040). The authors also wish to acknowledge colleagues in their research group for their helpful comments and advice during manuscript preparation.

© Authors 2010

REFERENCES


30 Woschni, G. Universally applicable equation of the instantaneous heat transfer coefficient in the internal combustion engine. SAE paper 670931, 1967.


APPENDIX

Notation

- \( A \) wall area (m²)
- \( BTDC \) before top dead centre
- \( C_p \) constant pressure specific heat (kJ/kg K)
- \( C_v \) constant volume specific heat (kJ/kg K)
- \( dp/d\theta \) rate of pressure rise with the crank angle
- \( dQ_h/d\theta \) heat release rate with respect to the crank angle
- \( dQ_w/d\theta \) heat transfer rate with respect to the crank angle
- \( h_c \) heat transfer coefficient (J/m² s K)
- \( m \) mass of in-cylinder gases (kg)
- \( MFB50\% \) 50 per cent mass fraction burned
- \( R \) gas constant (J/kg K)
- \( TDC \) top dead centre
- \( T \) mean gas temperature (K)
- \( T_w \) wall temperature (K)
- \( V \) cylinder volume (m³)
- \( \theta_c \) centre of heat release (deg crank angle after top dead centre)
- \( \theta_{ld} \) flame development duration (deg crank angle)
- \( \theta_{rd} \) rapid combustion duration (deg crank angle)
- \( \theta_{td} \) total combustion duration (deg crank angle)