PREMIXED IGNITION CHARACTERISTICS OF BLENDS OF GASOLINE AND DIESEL-LIKE FUELS ON A RAPID COMPRESSION MACHINE

by

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Fuel ignition process is of importance in premixed diesel low-temperature combustion strategies because longer ignition delay could provide more fuel and air mixing time. Using blends of gasoline and diesel-like fuels might be a possible way for the ignition delay extension. In this study, a rapid compression machine is employed to investigate the characteristics of premixed ignition processes of blends of n-heptane and commercial gasoline. The proportion of gasoline in blended fuels and the compression ratio in this rapid compression machine are varied to investigate the effects of fuel component and compression ratio on ignition processes. It is found that blended test fuels have two-stage increases in their cylinder pressure traces, indicating that a low temperature heat release process exists before the main combustion stage. Increased gasoline proportion in test fuels reduces peak cylinder pressure and maximum pressure rise rates, while the 1st, 2nd, and overall ignition delay are extended. Increased compression ratio elevates the peak cylinder pressure and shortens the 1st stage, 2nd stage, and overall ignition delays. The maximum pressure rising rates are also increased with compression ratio, so when the low gasoline proportion test fuels are used, knock combustion tends to occur at high compression ratio conditions. However, as long as the gasoline proportion increases to a sufficient level, knock combustion is avoided.

Key words: *ignition, gasoline, n-heptane, rapid compression machine, premixed charge*

Introduction

Premixed low-temperature diesel combustion (LTC) mode in compression ignition engines possesses great advantages in simultaneous reduction of NO_x and soot emissions [1, 2]. In this advanced diesel combustion strategy, exhaust gas recirculation (EGR) is heavily used to reduce local combustion temperatures, leading the in-cylinder combustion occur outside the NO_x and soot formation regions in a equivalence ratio and combustion temperature plot [3, 4]. However, the soot formation peninsular is located at locally fuel-rich areas of lower combustion temperatures than the NO_x formation region. Therefore, in LTC, although the NO_x formation region has already been avoided, combustion temperature has to be further reduced to achieve ultralow soot formation regions by additionally increasing EGR rate. The utilization of large amounts of EGR probably causes problems such as deteriorated combustion quality and increased hydrocarbons (HC) and carbon monoxide (CO) emissions, thus reducing the engine thermal efficiency.

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Another way to achieve simultaneous reduction in NO_x and soot emissions in LTC strategies is to strengthen the fuel and air mixing prior to combustion. The improved charge mixing reduces local equivalence ratio, therefore the soot formation area could be avoided without significantly reducing local combustion temperatures. The low-cetane fuels have longer ignition delay prior to combustion, so the fuel and charge may have more mixing time and the locally rich equivalence ratio areas, where soot formation is favorable, could be substantially reduced due to the sufficient fuel and air mixing. To this end, some researchers tried to use fuels with lower cetane numbers in advanced LTC modes [5, 6]. Kusaka *et al*. found that the low-cetane fuels could improve NO_x and soot emissions as well as indicated thermal efficiency [7], but reduced fuel cetane number may change the composition of incomplete combustion products (HC and CO emissions) [8]. As cetane number decreases, CO emissions gradually become the major component of the incomplete combustion products instead of HC emissions. Han *et al.* tried to apply blends of diesel and gasoline in LTC mode, aiming to enhance the charge mixing prior to combustion by extending the ignition delay comparing to the diesel LTC [9, 10]. They found that increased gasoline proportion in blends of diesel and gasoline could dramatically extend ignition delay, and as gasoline proportion increases to a certain value, the traditional curve of soot emissions *vs*. EGR rate diminishes. The soot emissions do not first increase and then decrease with increased EGR rate, but are always held at an ultralow level regardless of the EGR changes.

As abovementioned, the fuel ignition process is of particular importance in premixed LTC because the length of ignition delay plays a primary role in soot emissions control. Therefore, the objective of this study is to experimentally investigate the premixed ignition characteristics of blends of gasoline and diesel-like fuels using a rapid compression machine. By adding commercial gasoline into n-heptane, which is used as the substitute of diesel, the auto-ignition tendency is gradually reduced. Also, by changing the compression ratio of this rapid compression machine, the end condition of compression process is varied and their effects on ignition behaviors are investigated.

Experimental set-up

Rapid compression machine

Rapid compression machine uses external forces (compressed gas or hydraulic drive force) to produce a high speed piston move. The piston does not move back after reaching top dead center, so the compression, ignition and combustion processes can be realized within a single compression stroke. Due to the fast piston movement in cylinder, the heat loss during the compression process is negligible and the compression process can be considered an adiabatic compression, with the in-cylinder pressure, temperature, and mixture uniformly distributed. Additionally, the thermodynamic parameters can be conveniently adjusted in a rapid compression machine, so rapid compression machine can serve as an ideal experimental platform for the engine ignition and combustion research.

The test bench used for premixed ignition research in this study includes three parts: a rapid compression machine, a mixture supply system, and a compressed gas supply system. The rapid compression machine consists of the main combustion system, the gas drive system, the brake system, the data acquisition system and the movement control system. The schematic of this test bench is shown in fig. 1 and detailed specifications of the rapid compression machine are listed in tab. 1.

Test method

Before the rapid compression test, fuel should be first injected into the mixing tank to mix with the air for an hour. Afterwards, the mixture is inducted into the cylinder of the rapid compression machine (RCM). A gas compressor provides compressed gas to the gas drive cylinder, the piston of which could push the RCM piston towards the top dead center (TDC). The cylinder pressure and piston displacement are measured using a data acquisition system. The cylinder pressure is taken by Kistler 6125b pressure transducer, whose linearity extent is below 0.5% of the full scale range, and the signal is amplified by a Kistler 5015A charge amplifier. Piston displacement is measured by a Shanghai Xinyue WDM300 displacement transducer with the repeatable accuracy of 0.015 mm.

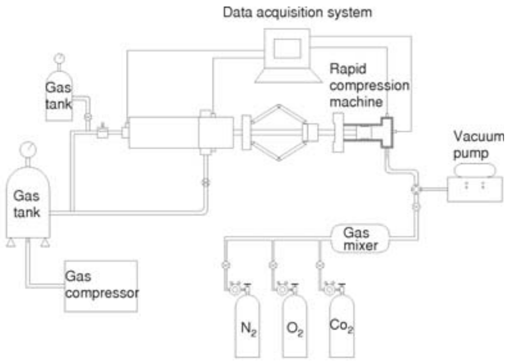

Figure 1. Schematic of test bench

Table 1. Specifications of rapid compression machine

Specifications	Data
Cylinder bore	50 mm
Cylinder thickness	13 mm
Piston shape	Flat type
Compression ratio	10.8-16.1

Test fuels

The test fuels in this study are prepared by blending n-heptane and commercial gasoline with research octane number (RON) of 93. The gasoline proportions in fuel blends are 0%, 20%, 40%, 70%, and 100%, respectively. The different blended test fuels are denoted as G0, G20, G40, G70, and G100, representing 0%, 20%, 40%, 70%, and 100% gasoline proportion in volume in the test fuels, and their RON are 0, 18.6, 37.2, 65.1, and 93, respectively. These fuels are premixed with air, and the initial charge pressure and temperature are held at 0.1 MPa and 298 ± 1 K, respectively.

Results and discussion

Definition of ignition parameters

Some parameters of ignition process are illustrated in fig. 2. As is seen, the in-cylinder pressure increases to a locally peak and then gradually decreases, meaning that the piston reaches the TDC and the heat loss causes a decrease in cylinder pressure. The temperature at TDC is calculated using the following equation, which considers that the compression stroke is an adiabatic process:

$$T(t_\mathrm{b}) = T(t_\mathrm{a})\left[\frac{P(t_\mathrm{a})}{P(t_\mathrm{b})}\right]^{\frac{k(t_a)-1}{k(t_a)}}$$

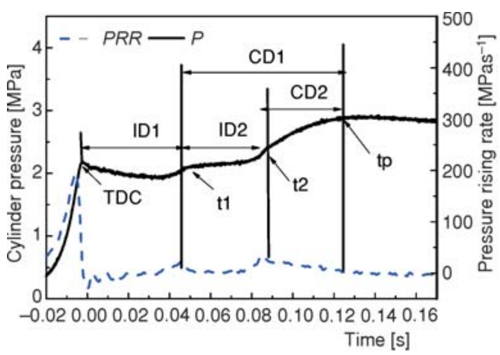

Figure 2. Definition of ignition and combustion parameters (G100, equivalence ratio Φ = 0.4, compression ratio = 12.5)

where T is the cylinder gas temperature, P – the cylinder pressure, and k – the specific heat coefficient.

The ignition timing is defined as the location where the regional peak pressure rising rate (PRR) appears. As is seen in fig. 2, the rising process of cylinder pressure after TDC can be divided into two different stages, indicating a two-stage heat release process. Therefore, in the following discussion, two ignition timings will be denoted, which are the first-stage ignition timing (t1) and the second-stage ignition timing (t2), respectively. The periods between TDC and the first-stage ignition timing is defined as the first-stage ignition delay (ID1), while the second-stage ignition delay (ID2) is the interval between the first-stage and second-stage ignition timing. As the heat release process proceeds, cylinder pressure gradually increases until reaching the peak. Two combustion durations are also indicated in fig. 2, which are the first combustion duration (CD1), the period between the first stage ignition timing and peak pressure location, and the second combustion duration (CD2), the period between the second stage ignition timing and the peak pressure location.

Effect of fuel composition on ignition process

Table 2. Pressures and temperatures of different test fuels at TDC in rapid compression machine

Fuels	Pressure [MPa]	Temperature [K]
G0	2.37	723.8
G20	2.38	725.1
G40	2.38	724.6
G70	2.42	728.5
G100	2.45	730.8

In this section, the effects of fuel composition on the ignition characteristics of premixed charge are investigated. Five test fuels, G0, G20, G40, G70, and G100 are selected and the equivalence ratio (Φ) is maintained at 0.4. The RCM compression ratio is 12.5. Pressures and calculated temperatures of different test fuels at TDC using are listed in tab. 2.

The pressure traces *vs.* time of different test fuels in this RCM are illustrated in fig. 3. TDC is defined as 0 ms, and the heat release after TDC causes pressure increase. With increased gasoline proportion, the peak cylinder pressure gradually decreases and its phasing is retarded. The pressure trace of high-gasoline-proportion fuel shows an obvious two-stage increase after TDC, indicating that the ignition and heat release can also be divided to two stages. The two-stage ignition characteristics are mainly due to the internal isomerization reaction of peroxy radicals, which could easily occur in molecules containing long carbon chains allowing the formation of C-C-C-O-O-H structure. Although the pure gasoline contains relatively small amount of long straight molecules (n-heptane), its pressure trace still features a two-stage increase, which agrees with previous researches [11]. Machrafi and Cavadias [12]. found that the homogenous charge compression ignition combustion has three heat release stages: they are the cool flame reactions by long straight molecules such as n-heptane, the obstructed preignition by the formation of benzyl radical from aromatics such as toluene, and the main heat release stage [12]. The two heat release

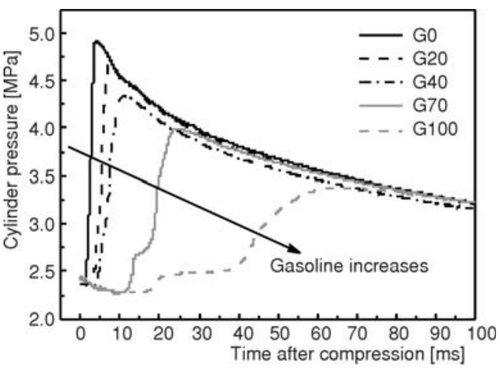

Figure 3. Pressure traces of different test fuels in RCM

stages before the main ignition might lead to the first pressure increase in this study. As the proportion of n-heptane increases, the two ignition phasing are gradually close with each other.

In fig. 4, the maximum pressure rising rate of two stages of ignition processes are illustrated, as different test fuels are used. From fig. 4, it is found that the maximum pressure rising rates of the first stage ignition are basically lower than the second stage ignition, indicating that the heat release in the first stage ignition is relatively weak. Additionally, as the gasoline proportion in test fuels increases, the maximum pressure rising rate of two ignition stages are reduced, which means the addition of gasoline to n-heptane could reduce the knock intensity of the premixed combustion.

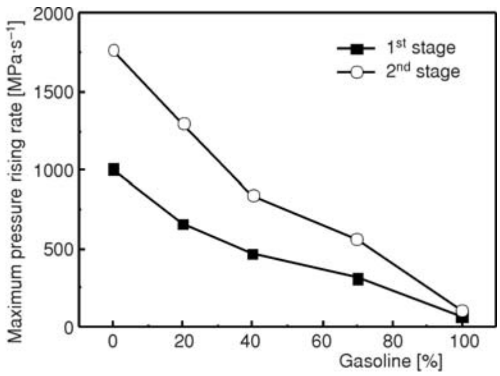

Figure 4. Maximum pressure rising rates of two ignition stages with different test fuels in RCM

The ignition delays of different test fuels are shown in fig. 5. Three ignition delays, the first stage ignition delay, the second stage ignition delay and overall ignition delay, increase with the increase of gasoline proportion in test fuels. The extension of ignition delay by gasoline

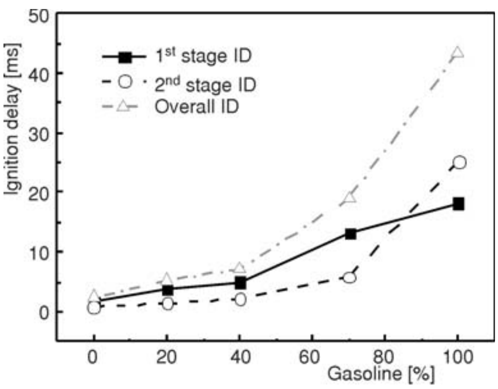

Figure 5. Ignition delays of different test fuels in RCM

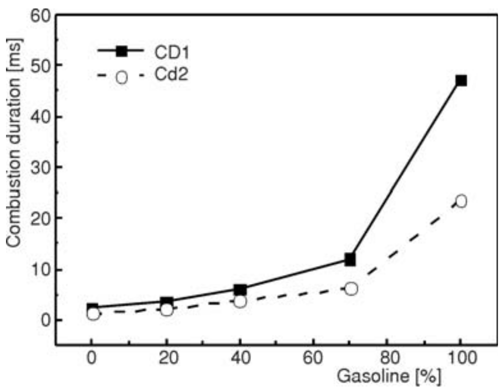

Figure 6. Combustion durations of different test fuels in RCM

addition might provide sufficient time for the fuel and air mixing. Also, the length of the first stage ignition delay is longer than the second stage ignition delay for most test fuels. Two combustion duration parameters, denoting the time difference between the two-stage ignition timings and the peak pressure location are shown in fig. 6. It can be seen that the increased gasoline percentage could dramatically extend the combustion durations. As pure gasoline is used, the CD1 and CD2 can be extended to approximately 50 ms and 25 ms, respectively. The extended combustion duration slows down the combustion rate, reducing the peak pressure rising rate and therefore the knocking intensity. Also, it is noted that CD1 is always longer than CD2, due to the addition of the second ignition delay.

Effect of compression ratio on ignition characteristics

The effect of compression ratio on ignition processes of different test fuels are studied in this section. Four compression ratios (10.8, 12.5, 14.0, and 16.1) are selected, and the test fuels are G0, G40, G70, and G100, respectively. The fuel and air equivalence ratio is held at 0.4. As compression ratio increases, test fuels with less gasoline proportion tend to cause knock phenomenon. As shown in fig. 7, at knock conditions, cylinder pressure oscillates apparently, causing deteriorated repeatability of test results. Therefore, the compression ratio selection in this section should avoid knock combustion of all test fuels. The "un-knock" compression ratio range of each given fuel is illustrated in fig. 8. With increased gasoline proportion, the compression ratio selection range is broadened. For example, as n-heptane is utilized, only two compression ratios can avoid knock combustion, but as the gasoline proportion reaches 70%, knock combustion never occurs for all the four compression ratios.

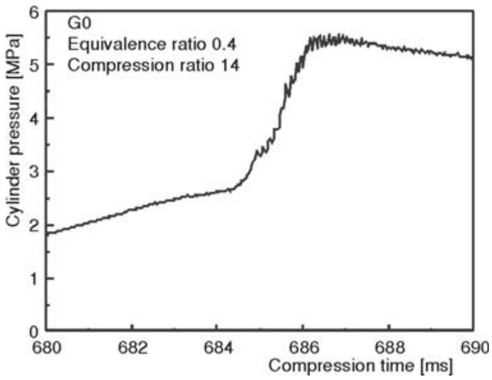

Figure 7. Pressure characteristics at knocking conditions: G0, compression ratio 14, Φ = 0.4

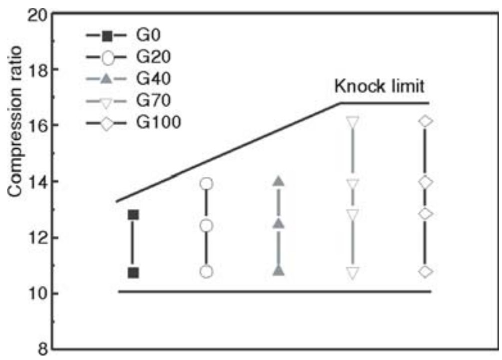

Figure 8. Compression ratio range of different fuels (Φ = 0.4)

The effects of fuel composition and compression ratio on peak cylinder pressure are shown in fig. 9. As can be seen, increased compression ratio can increase peak cylinder pressure, because the pressure and temperature by the end of compression stroke are dramatically enhanced with increased compression ratio. Besides, increased gasoline proportion in test fuels, the same as discussed in the previous section, reduces the peak cylinder pressure.

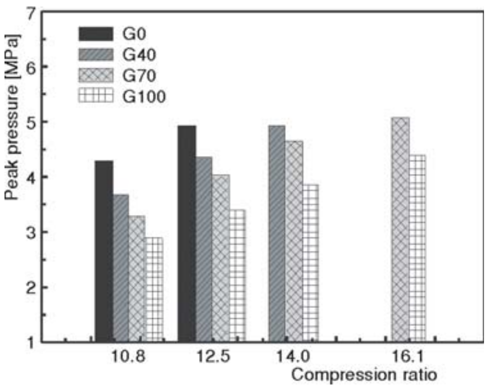

Figure 9. Peak pressure of different fuels *vs.* compression ratio (Φ = 0.4)

The relationship of two-stage maximum pressure rise rates with compression ratio is shown in fig. 10. Like fig. 4, increased gasoline proportion could effectively reduce the maximum pressure rise rates of the first and second heat release processes, but increased compression ratio dramatically enhance the maximum pressure rising rates. It is also noted that at high compression ratio conditions, the maximum pressure rise rates of the second stage heat release processes are much higher than those of the first stage.

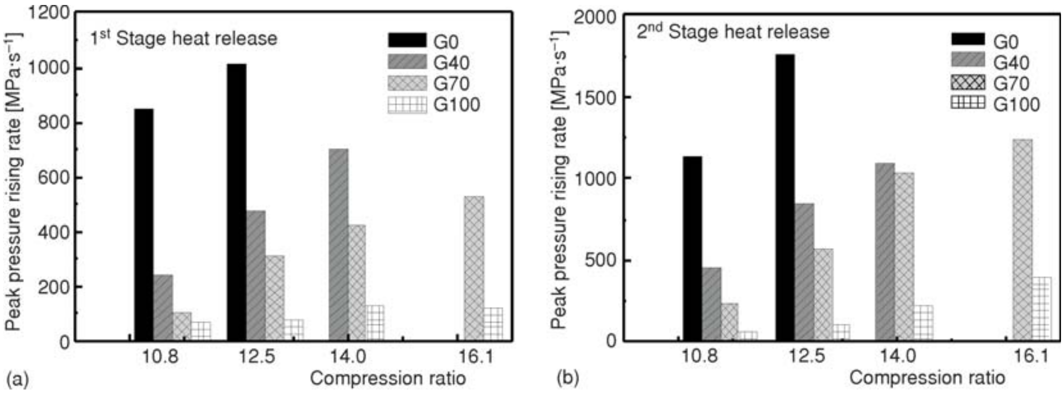

Figure 10. Maximum pressure rising rates of different fuels *vs.* compression ratio (Φ = 0.4); (a) 1st stage heat release, (b) 2nd stage heat release

Figure 11 shows the influences of compression ratio on the 1st stage, 2nd stage, and the overall ignition delays of different test fuels. For a given test fuel, the ignition delays of two heat release stages are both shortened with increased compression ratio. As can be seen, the increase in compression ratio from 10.8 to 16.1 can reduce the 1st stage ignition delay of G100 by around 40 ms, and also cause a reduction of around 25 ms in the 2nd stage ignition delay. Therefore, the overall ignition delay of G100 can be shortened by over 60 ms. Similar phenomenon can be observed in other test fuels as compression ratio increases, and the higher gasoline proportion fuels

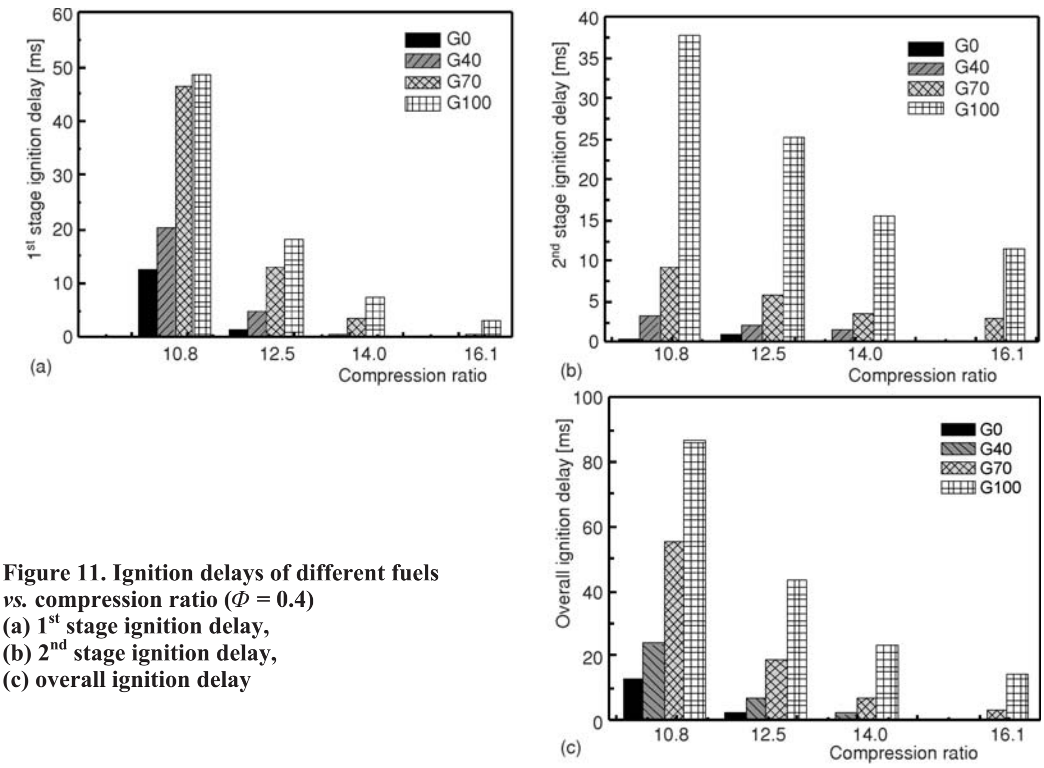

Figure 11. Ignition delays of different fuels *vs.* compression ratio (Φ = 0.4)
(a) 1st stage ignition delay,
(b) 2nd stage ignition delay,
(c) overall ignition delay

possess a more apparent reduction in ignition delay with increased compression ratio. Given a compression ratio, the increased gasoline proportion obviously increases the 1st and 2nd stage ignition delays, as well as the overall ignition delay. The effects of increased gasoline proportion on the extension of ignition delay are more apparent in the 2nd stage ignition process. From fig. 11(b), G100 test fuel could dramatically extend the 2nd stage ignition delay at each compression ratio, with the most obvious extension observed at compression ratio of 10.8. As the gasoline proportion increases from 70% to 100%, the 2nd stage ignition delay at compression ratio of 10.8 is elevated over 25 ms.

Two combustion durations of different test fuels at different compression ratios are illustrated in fig. 12, in which the length of CD1 is almost two times as that of CD2. Similar with fig. 6, increased gasoline proportion apparently extends both CD1 and CD2. Even with compression ratio of 16.1, G100 could achieve the long combustion durations of 20.8 ms for CD1 and 9.3 ms for CD2, whereas the combustion durations of G70 are 5.3 ms and 2.4 ms for CD1 and CD2, respectively. In contrast, the increase in compression ratio could enhance the temperature and pressure at the end of compression stroke, thus reducing the length of both combustion durations. Take G100 as an example, as compression ratio increases from 10.8 to 16.1, the CD1 and CD2 are both reduced, from 86.8 ms to 20.8 ms for CD1 and from 46.4 ms to 9.3 ms for CD2.

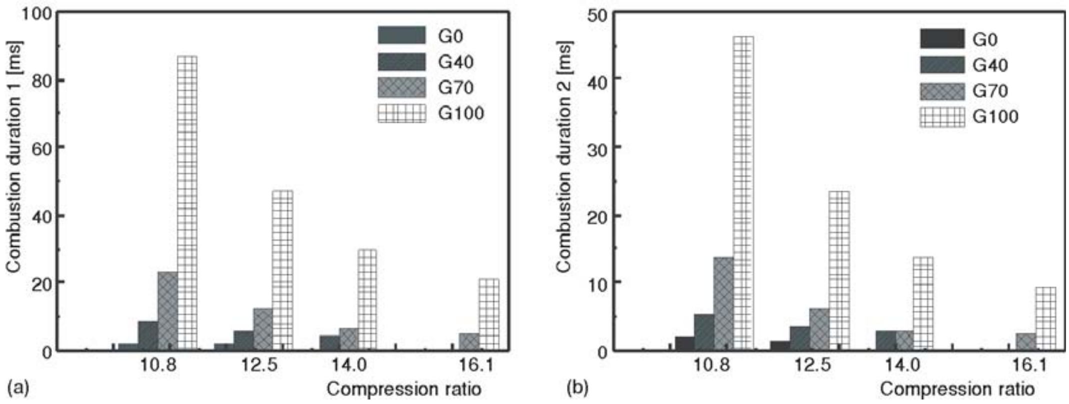

Figure 12. Combustion durations of different fuels at different compression ratios; (a) CD1, (b) CD2

Conclusions

In this study, a RCM is employed to investigate the ignition processes of blends of n-heptane and commercial gasoline. The proportion of gasoline in blended fuels is varied to investigate the effects of fuel component on the ignition processes. Five test fuels, G0, G20, G40, G70, and G100, indicating that the gasoline proportion in fuel blends is 0%, 20%, 40%, 70%, and 100%, are employed for testing. Besides, the compression ratio of the RCM is also adjusted to change the temperature and pressure conditions at the top dead center. Therefore, effects of compression ratio on ignition processes are also investigated.

The conclusions of this study are:

- For the blended test fuels with different proportions of n-heptane and commercial gasoline, two-stage increases are found in their cylinder pressure traces, indicating that a low temperature heat release process exists before the main combustion stage. These two combustion stages are closed to each other when high n-heptane proportion fuels are used, while high gasoline proportion fuels could result in more separate combustion stages.

- The increased gasoline proportion leads to a reduction in the peak cylinder pressure and a retard in the phasing. Besides, increased gasoline proportion also reduces the maximum pressure rise rate of two combustion stages, and extends the ignition delays of the 1st and 2nd stage ignition delay as well as the overall ignition delay. The combustion duration, as gasoline proportion increases, is dramatically extended, indicating that a more moderate combustion might be achieved using high gasoline proportion fuels.
- Increased compression ratio elevates the peak cylinder pressure, and shortens the 1st stage, 2nd stage and overall ignition delays, as well as the combustion durations. The maximum pressure rising rates of two combustion stages are also increased with compression ratio, so when the low gasoline proportion test fuels are used, knock combustion tends to occur at high compression ratio conditions. However, as long as the gasoline proportion increases to a sufficient level, knock combustion is avoided. Therefore, higher compression ratio can be selected for the compression ignition of high gasoline proportion test fuels.

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Acronyms

EGR – exhaust gas recirculation
LTC – low temperature combustion
TDC – top dead center
RCM – rapid compression machine
RON – research octane number
PRR – pressure rising rate

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