에너지공학, 제22권 제2호(2013)
*Journal of Energy Engineering*, Vol. 22, No. 2, pp. 197~204 (2013)

# Experimental Study on the Cycle–to–Cycle Combustion Variations in a Spark Ignition Engine

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(Received 9 May 2012, Revised 10 June 2013, Accepted 10 June 2013)

Abstract - A cyclic variability has long been recognized as limiting the range of operating conditions of spark ignition engines, in particular, under idling conditions. The coefficient of variation (COV) in indicated mean effective pressure (IMEP) defines the cyclic variability in indicated work per cycle, and it has been found that vehicle drivability problems usually result. For analysis of the cyclic variations in spark ignition engines at idling, the results show that cyclic variability by the COV, COV of IMEP, the lowest normalized value (LNV), and burn angles can help to design the spark ignition engine.

**Key words** : : Mass burn crank angle, Indicated mean effective pressure (IMEP), Coefficient of variation (COV), Lowest normalized value (LNV), standard deviation (STD)

## 1. Introduction

Modern vehicles need to have more and more attention paid to improving the fuel saving, the emission, and the safety performances. Many automakers or engineers have devoted effort to developing different electronic control systems to enhance vehicle performance. At the same time, drastic competition and pressure in energy and the environment are constantly compelling automotive companies to find better ways of developing new types of vehicles with now fuel consumption and low emissions using sophisticated electronic products, as well as reducing costs and time to market [1, 2].

With the strengthening requirements of automobile pollutant legislation and continuous improvement of the thermal efficiency of internal combustion engines, more comprehensive and detailed research work has to be conducted. Spark ignition engines have the lower thermal efficiency than the compression ignition engine. Traditional premixed charge combustion engines have lower thermal efficiency because of the avoidance of engine knock and the unavoidable throttling at the intake at partial load. Especially the emissions of HC and CO and cycle-to-cycle variation matter still remain as the main problem [3, 4].

Cycle-to-cycle combustion variability in spark-ignition engines limits the use of lean mixtures and lower idle speeds because of increased emissions and poor engine stability. The causes of the cycle-to-cycle variatios are summarized in some papers.

Although the causes of cycle-to-cycle variability are identifiable, there has been some difficulty in establishing quantitatively the contribution of each of these phenomena. Detailed investigation on this problem by experiment is very difficult because of the problem of controlling and measuring the changes of any of these influencing factors. Consequently it is not clear which factor is the

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most important in the combustion variations and hence how to reduce this variation. One way to examine this problem is through use of a computer simulation [5].

Much effort has been dedicated to extend the limit of lean bum operation－hereafter referred to as the lean stable operating limit－in order to improve fuel efficiency, as well as reducing exhaust gas emissions from spark ignition engines. The limit is imposed by increased cyclic variation of the combustion intensity which reduces the drivability and the effect is usually quantified through the coefficient of variation of the indicated mean effective pressure [6].

Some researchers found that there was a strong correlation between the cycle-by-cycle variations in turbulence intensity ahead of the flame and flame arrival time at the compressed natural gas (CNG) direct injection (DI) using a rapid compression machine, suggesting that fluctuation in the bulk turbulence intensity ahead of the flame cause variations in burn rate during the main combustion phase. The large cycle-by-cycle variation, however, restricts the lean operation limit of this type of homogeneous mixture engine [7, 8].

The objective of this paper is to clarify the most significant sources of cycle-by-cycle combustion variability in a spark ignition engine at idle. And the research could help to design the spark ignition engine.

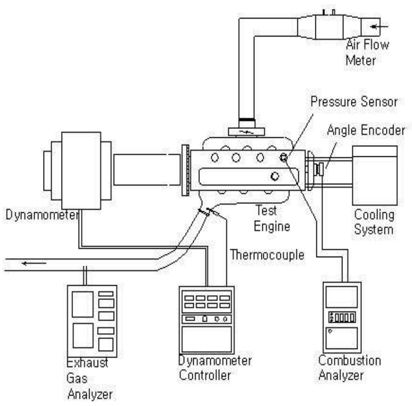


**Fig. 1.** Schematic diagram of experimental apparatus.

## 2. EXPERIMENTAL APPARATUS AND PROCEDURE

### 2.1 Engine Setup

Figure 1 shows the schematic diagram of the experimental apparatus. The engine was coupled to a dynamometer to control the engine speed and load. Oil temperature, coolant temperature, exhaust temperature, inlet pressure and exhaust pressure were measured with various sensors. The exhaust gas constituents (CO, $CO_2$, THC,$O_2$,$NO_x$)were measured in this research project by a gas analyzer (Mexa 9100DEGR, Horiba).

Signal from the crankshaft position sensor (CPS) installed on the crankshaft pulley and the hall sensor installed on the cam-shaft pulley is sent to the ignition control device, which then determines the amount of electric energy to be sent to the combustion chamber and controls the ignition timing. All experiments were conducted at 800RPM and a compression ratio of 9.5.

Table 1 shows the relevant characteristics of the engine used in our experiment. This engine had a pentroof head with a centrally-located spark plug and was modified to operate on a single cylinder to avoid multiple cylinder interactions. Thus, fuel is injected and a spark is supplied to only one cylinder, and the intake and exhaust runners of the

**Table 1.** . Specification of test engine.

| Component | Specification |
|---|---|
| Number of cylinder | 4 |
| Bore | 75.5 mm |
| Stroke | 82.0 mm |
| Total displacement | 1,468 cc |
| Compression ratio | 9.5 |
| Intake valve open | 18.5 BTDC |
| Intake valve close | 51.5 ABDC |
| Exhaust valve open | 51.5 BBDC |
| Exhaust valve close | 18.5 ATDC |

firing cylinder are isolated.

The engine was coupled to a dynamometer, which might be used to turn the engine while motoring or absorb when the engine was firing. The dynamometer was used to keep the engine at a constant speed of 800 rpm for all experiments. The engine coolant system was modified to include a water heater; this allowed the engine coolant temperature to be maintained at a temperature around 80℃ for all tests. Cylinder pressure was recorded with a Kistler 6051B piezoelectric pressure transducer. The transducer was connected to a Kistler model 5004 dual mode charge amplifier.

A surge tank was installed on the inlet side to minimize the experimental engine's intake pulsation, and an external pump was used for the coolant and the oil in order to minimize power loss.

## 2.2 Operating Condition

The spark timing and speed were used to be the values specified for the idle condition of the engine. The air/fuel equivalence ratio was kept at a value of 1.0 because the engine normally operates with a three-way catalyst, and so stoichiometric operation is a constraint of the system. The inlet manifold pressure was adjusted to give an average gross indicated mean effective pressure (IMEP), which is a value typical of an idle condition. All experiments were performed with the engine at fully warmed-up state.

In order to analyze the cyclic variation in the test engine, the fluctuations in burn parameters are used to determine the variations in the input parameter. The ideal set of burn parameters would characterize the combustion completely, from start to finish, as well as defining the total amount of energy released. Also, the burn parameters should be easy to be determined from the cylinder pressure data, since many cycles would be taken for statistical significance.

The injection timing was set to the optimum injection timing where the injection end timing is the same timing as the ignition start. As the data carried out 100 repeated combustion experiments under the same initial condition for each equivalence ratio; the amount of data is considered to be enough for investigating cycle-by-cycle variations using the engine.

## 2.3 Lowest Normalized Value and Coefficient of Variation

Two parameters are used to evaluate cycle-by-cycle variations in this paper: one is the lowest normalized value (LNV), which shows the departure of the variation from IMEP's average value, and the other is the coefficient of variation (COV) that has been widely used to evaluate the cycle-by-cycle variations in engine studies.

The lowest normalized value (LNV) on the spark timing, LNV is defined as

$$LNV\,(\%) = \frac{IMEP_{MIN} \times 100}{\overline{\overline{IMEP}}}$$

where

$IMEP_{MIN}$ is the minimum IMEP value in the data set, and $\overline{IMEP}$ is the mean IMEP of the data set. It is proposed one other means of characterizing cycle-to-cycle variations, the lowest normalized value (LNV). The purpose of this parameter is to assess the misfire tendency of an engine; test has shown that LNV correlates well with driver's subjective rating of engine smoothness.

The influence of combustion phasing on the COV of IMEP is the standard deviation in IMEP divided by the mean IMEP.

$$COVx = \frac{STD}{average\,(x)}$$

where

$$average\,(x) = \frac{1}{N}\sum_{i=1}^{N} x_i$$

and the standard deviation (STD) is

$$STD = \sqrt{\frac{\sum_{i=1}^{N}\left[x_i - average\,(x)\right]^2}{N}}$$

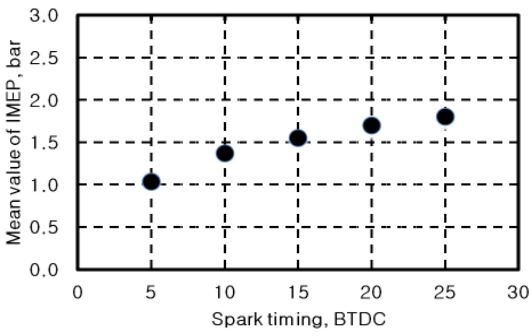


**Fig. 2.** Mean value of IMEP versus spark timings.

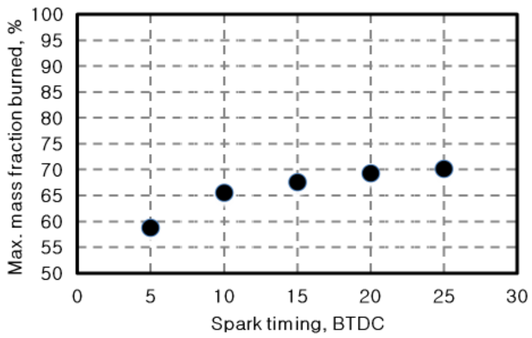


**Fig. 3.** Maximum mass fraction burned versus spark timings.

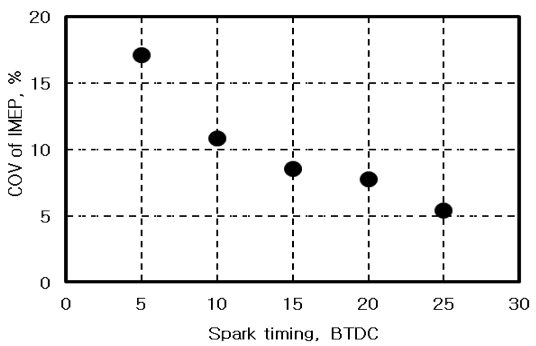


**Fig. 4.** COV of IMEP versus spark timings.

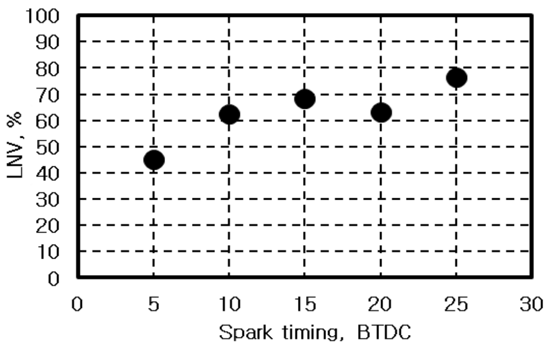


**Fig. 5.** LNV versus spark timings..

The combustion durations used to analyze combustion characteristics in this engine study are the flame initial development duration, 0-2% mass burn crank angles, the rapid combustion duration 10-90% mass burn crank angles, the late combustion duration 90-100% mass burn crank angles and total combustion duration 0-100% mass burn crank angles. The 0-10% mass burn crank angles are defined as the duration from the ignition start to 10% of the pressure rise due to combustion, the duration of 10-90% of the pressure rise due to combustion, the duration of 90-100% and the duration from the ignition start to 100% of the pressure rise due to combustion.

## 3. Results and Analysis

Figure 2 shows spark map at the same speed and inlet pressure condition at 800 RPM that is idling speed. This figure shows that IMEP is quite sensitive to phasing changes at idle timing. The relative changes in combustion phasing have a larger influence on IMEP at idle because of this high sensitivity. The maximum value of IMEP at the spark timing of 25°BTDC recorded the largest value.

Figure 3 shows the maximum mass fraction burned in each case. The maximum mass fraction burned is the normalized value with respect to the value for 100 cycles. The maximum value of the mass fraction burned at the spark timing of 25°BTDC recorded the largest value. The minimum value of the mass fraction burned was about 58% at the spark timing of 5°BTDC. The figure shows the general characteristics that, as the IMEP value increased, the speed of initial combustion became shorter, mass fraction burned became bigger. Here, we adopt the definition that the speed of initial

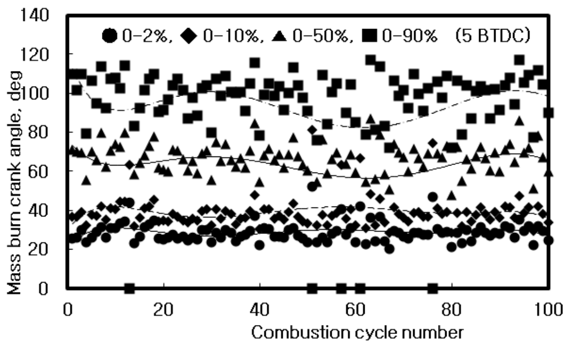


**Fig. 6.** Mass burn crank angle versus combustion cycle number for 5°BTDC.

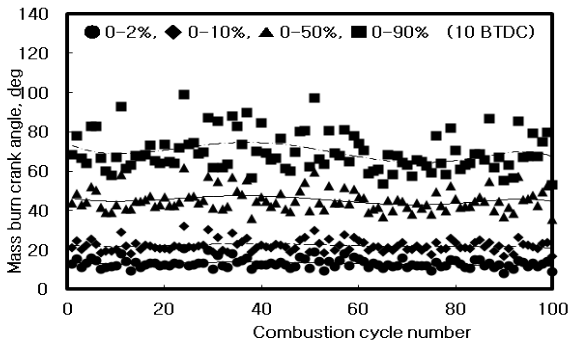


**Fig. 7.** Mass burn crank angle versus combustion cycle number for 10°BTDC.

combustion is the crank angle interval between spark discharge and the time at which 2% of the mass burned.

Figure 4 shows the influence of combustion phasing on the COV of IMEP. One important measure of cyclic variability, derived from pressure data, is the coefficient of variation in IMEP. It is the standard deviation (STD) in IMEP divided by the mean IMEP. It defines the cyclic variability in indicated work per cycle, and it has been found that vehicle drivability problems usually result when COV of IMEP exceeds about 10%. In this figure, as the spark is advanced, the COV goes down because relative changes in combustion phasing have a smaller effect on the IMEP, as the spark timing indicated. However, for the two most advanced cases, the COV begins to increase again, probably because the lower temperature at the time of spark is adversely affecting the ignition. Note that the COV of IMEP at the idle spark timing is slightly under 10%, which is typically considered the limiting point above which combustion variability is unacceptably high.

Figure 5 shows the LNV on the spark timing. This value is plotted for the spark sweep in Figure 5. In general, it is suggest that an appropriate acceptable value for the lower limit of LNV would be 70 percent. As the figure shows, the experimental engine only matches that criterion at the 15°BTDC spark timing.

Extensive empirical experience with combustion parameters and vehicle development has confirmed the results of the earlier experiment.

For the idling operation in the spark ignition engine, 5 to 10°BTDC is generally used for the minimum spark advance for the best torque (MBT). Figure 6 shows the mass burn crank angle versus combustion cycle number for 0-2%, 0-10%, 0-50%, and 0-90% burn angles at the 5°BTDC spark timing. And Figure 7 shows the mass burn crank angle versus combustion cycle number for 0-2%, 0-10%, 0-50%, and 0-90% burn angles at the 10°BTDC spark timing.

Figure 8, 9, 10, 11, and Figure 12 show 0-2%, 0-10%, 0-50%, 0-90%, and 10-90% mass burn crank angles versus combustion cycle number at 5°BTDC and 10°BTDC, respectively.

The early burn period is characterized by the 0-2% burn angle, also known as the flame development angle; it represents the crank angle interval between spark and the time when 2% percent of the charge mass has been burned. This is often taken as a measure of the time it takes to achieve a fully developed turbulent flame in the cycle. Thus, the 0-2% burn angle may be preferable when the combustion period of interest is the very early flame development; however, it was found that the resolution of the burn rate analysis was

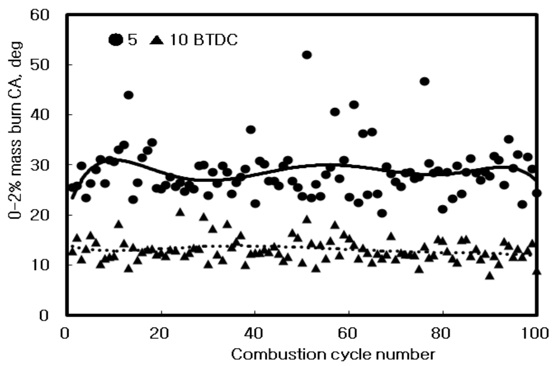


**Fig. 8.** 0-2% mass burn crank angle versus combustion cycle number for 5 and 10°BTDC.

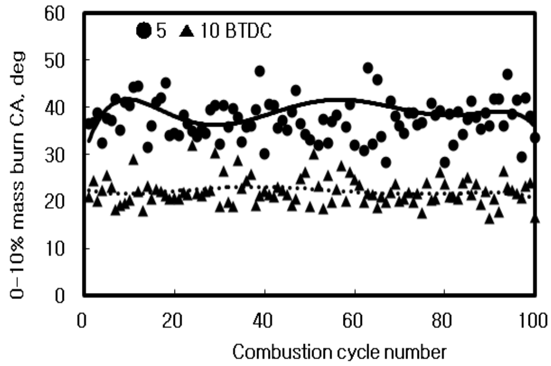


**Fig. 9.** 0-10% mass burn crank angle versus combustion cycle number for 5 and 10°BTDC.

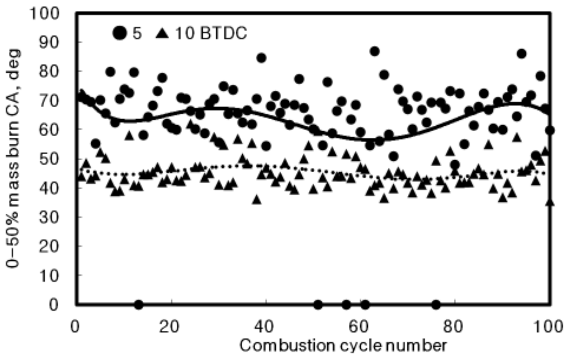


**Fig. 10.** 0-50% mass burn crank angle versus combustion cycle number for 5 and 10°BTDC.

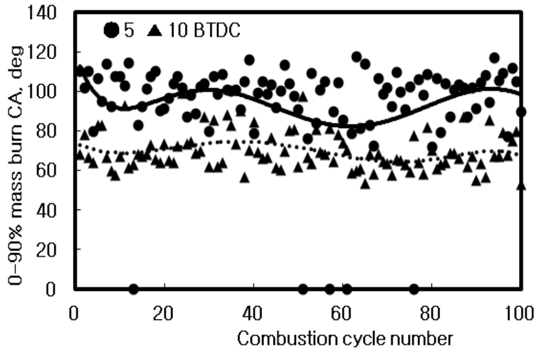


**Fig. 11.** 0-90% mass burn crank angle versus combustion cycle number for 5 and 10°BTDC.

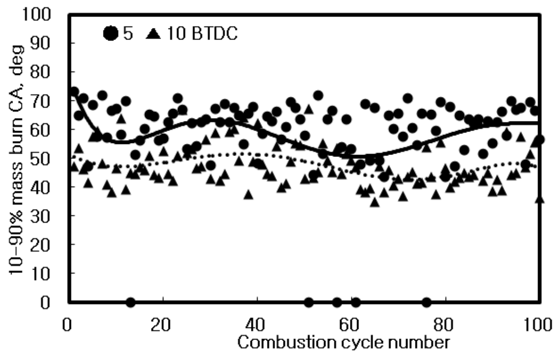


**Fig. 12.** 10-90% mass burn crank angle versus combustion cycle number for 5 and 10°BTDC.

insufficient for detecting changes in 0-2% burn angle between perturbed and non-perturbed cycles.

By the time 10% of the charge mass is burned, the flame may be as large as 30% of the total combustion chamber volume.

The 0-50% burn angle may be considered an approximate measure of the time it takes the flame to develop from the spark to the peak mass burning rate. The 10-50% burn angle is the earlier －and faster－part of the turbulent flame propagation, representing a significant part of the total rapid burning angle. This portion of the combustion process is very important from a phasing standpoint; the location of the 50% burn angle may be used as an indicator of combustion phasing with respect to optimum. Typically, the peak mass burning rate occurs within a few crank angle degrees of the 50% burn angle; thus, from this time onward combustion is slowing down. The burn speed after this point has a smaller influence on IMEP since it is so retarded with respect to the expansion stroke. Because the burned gas is substantially less dense than the unburned mixture, by the time 90% of the mass has been burned, almost the entire combustion chamber volume has been engulfed by the burned gas.

The rationale for dividing the 10-90% burn angle in half for the purposes of this analysis has to do with the phenomena that govern the flame growth in this period. During the early part of turbulent flame propagation, flame growth is, in large part, influenced by the expansion of the burned gases behind the flame; by the time 50% of the charge mass has been burned, 80% of the combustion chamber volume has been engulfed by the flame. The later turbulent flame growth, on the other hand, is characterized by a slower flame front growth that has significant interaction with the cylinder walls. Admittedly, the division at 50% mass fraction burned is somewhat arbitrary; however, it is a convenient point to separate the early and the late turbulent flame phenomena, and it is easily defined with the analysis tools available.

## 4. Concluding Remarks

For experimental study on the cycle-to-cycle combustion variations in a spark ignition engine for the hybrid electric vehicle, the main results are summarized as follows:

(1) To analyze the cyclic variation in spark ignition engine at idle, a methodology was developed whereby the input variations in air mass, fuel mass, and residual mass could be identified through analysis of the variations in the output burn parameters, i.e., the 0-2%, 0-10%, 10-50%, 0-90%, and 10-90% burn angles, IMEP, COV of IMEP, LNV, and so on.

(2) Experimental data suggest that COV of IMEP and burn angles and LNV levels for acceptable idle quality are fairly similar for a wide range of test engine.

(3) The COV of burn angle shows minimum value at the spark timing 5 to 10°BTDC that maximum COV of IMEP appears, and the COV of burn angle increases according as spark timing is advanced.

(4) As the spark is advanced, the COV of IMEP goes down because relative changes in combustion phasing have a smaller effect on the IMEP.

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