Cycle-to-Cycle Variations Under Cylinder-Pressure-Based Combustion Analysis in Spark Ignition Engines

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Combustion analysis based on cylinder-pressure provides a mechanism through which a combustion researcher can understand the combustion process. The objective of this paper was to identify the most significant sources of cycle-to-cycle combustion variability in a spark ignition engine at idle. To analyse the cyclic variation in a test engine, the burn parameters are determined on a cycle-to-cycle basis through the analysis of the engine pressure data. The burn rate analysis program was used here and the burn parameters were used to determine the variations in the input parameter—i. e., fuel, air, and residual mass. In this study, we investigated the relationship of indicated mean effective pressure (IMEP), coefficient of variation (COV) of IMEP, burn angles, and lowest normalized value (LNV) in a spark ignition engine in a view of cyclic variations.

Key Words : Cycle-to-Cycle Variation, Indicated Mean Effective Pressure, Idle, Coefficient of Variation, Lowest Normalized Value.

1. Introduction

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 variations are summarized in some papers (Ozdor et al., 1994; Brehob and Newman, 1992). Although the causes of cycle-to-cycle variability are identifiable, there has been some difficulty in establishing the contribution of each of these phenomena quantitatively. 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 most important in the combustion variations and hence how to reduce this variation (Han and Chung, 1998). One way to examine this problem is through use of a computer simulation (Shen et al., 1994; 1996).

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Cycle-to-cycle variations in the growth rate and location of the flame kernel at very early times were found to be the major cause of cycle-to-cycle pressure variations in spark ignition engines (Gatowski et al, 1984; Ho and Santavicca, 1987; Keck, 1987). They concluded that the most important parameters controlling the initial flame growth were the laminar flame speed at the spark plug and the size of the first eddy burnt. They also found no increase in flame speed due to energy input from the spark but that the position of the so-called 'wall contact flame center' was the major cause of variations in the burning rate during the 'fast' burning stage. Their experiment was designed to eliminate fluctuations in equivalence ratio and the observed variations in the initial burning speed were within the expected fluctuations in the residual mass fraction.

Much effort has been dedicated to extend the limit of lean burn operation—hereafter referred to as the lean stable operating limit—in order to improve fuel efficiency, as well as reducing exhaust gas emission from the spark ignition engine. The limit is imposed by increased cyclic variation of the combustion intensity which reduces the driveability and the effect is usually quantified through the coefficient of variation of the indicated mean effective pressure (Kiyoshi et al., 1997).

Pischinger and Heywood (1990) found that the cyclic variation of flame propagation near the spark plug influenced the amount of heat release at the spark plug gap and this greatly influenced the so-called rapid burn rate. However, Robinet et al. (1997) did not find a good relationship between the indicated mean effective pressure (IMEP) and the direction and expansion speed of the initial flame kernel. Brown et al. (1996) investigated the relationship between the initial speed of combustion and the IMEP variation, in an experiment which minimized the effect of residual gases.

Combustion analysis based on cylinder-pressure provides a mechanism for combustion process (Heywood, 1988; Sztenderowicz and Heywood, 1990; Randolph, 1994; Brunt 1996). Although the lean limit of air-fuel ratio has indeed been extended as a result of the development, the stable operating limit still remains unknown and there is yet no definitive agreement on either the spatial and temporal origin of cyclic variation and the reason for its increase at the limiting A/F ratio. Some of the current ideas on this subject are reviewed below.

Recently, numerical calculation has allowed prediction of the flow inside cylinders and several studies have reported variations in the initial stage of combustion phase due to variations in heat transfer at the spark plug (Kerstein, 1990; Meyer et at., 1993; Hinze, 1993). Through simultaneous laser doppler velocimeter and ionization probe measurements in an engine, some researchers found that there was a strong correlation between the cycle-to-cycle variations in turbulence intensity ahead of the flame and flame arrival time at the LDV probe volume, suggesting that fluctuation in the bulk turbulence intensity ahead of the flame caused variations in burn rate during the main combustion phase (Keck et al., 1987; Hinze, 1997).

The objective of this paper is to clarify the significant sources of cycle-to-cycle combustion variability in a spark ignition engine at idle.

2. Experimental Procedure

2.1 Engine setup

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 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. The heater also allowed preheating of the engine block and head, greatly reducing the amount of warm-up time required before data could be taken. The spark plug ground electrode orientation was kept con-

Table 1 Specification of engine used

Model	Nissan SR20DE
4-cylinder, Engine Type	4 valve/cylinder, dual ovehead cam
Displacement/cylinder	499.6cc
Bore×Stroke	8.6×8.6cm
Number of Cylinder	4
Compression Ratio	9.5
Intake Valve	Open 13°BTDC Close 55°ABDC
Exhaust Valve	Open 57°BTDC Close 3°ATDC
Valve overlap	16°

tant for all experiments. The plug used was a NGK BKR7E, with a plug gap of 0.9mm, as pecified by the manufacturer. The spark energy leposited was 50 mJ.

Cylinder pressure was recorded with a Kistler 6051B piezoelectric pressure transducer. The transducer was connected to a Kistler model 5004 lual mode charge amplifier. The voltage output of the amplifier was sampled by a PC-based ligital data acquisition system using a Data Translation A/D converter DT2828. Pressure lata were taken at a 1°interval using the pulse from a 360 pulse/revolution optical shaft encoder as an external trigger. The shaft encoder also provided a reference pulse at the bottom center.

2.2 Operating condition

Table 2 shows the operating condition selected. The spark timing and speed were used to be the values specified by SR20DE 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 IMEP of 1.55bar, which is a value typical of an idle condition. All experiments were performed with the engine at fully warmed-up state.

2.3 Analysis procedure

In order to analyse the cyclic variation in the test engine, the fluctuations in burn parameters are used to determine the variations in the input parameter—i. e., fuel, air, and residual mass. These burn parameters are determined on a cycle-to-cycle basis through the analysis of the engine pressure data. The burn rate analysis program that was used here was described in detail by Cheng and Heywood (1993). An 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 analysis.

3. Results and Analysis

Figure 1 shows the spark map at the same speed and inlet pressure condition as idle. The only occurrence of misfire was in the 35° BTDC advanced case; there were partial burns in the 35° case and the 5° case (Fig. 2 and 3). The misfired cycles and partial burns were eliminated for calculation of average. Figure 1 shows that optimum timing—provided there are no misfired cycles—is well advanced from the idle timing. From this figure, MBT would appear to be at 35° BTDC or earlier; however, the presence of misfire prevents this spark timing from being practical. This is a common problem at low load: the low in-cylinder temperature hinder ignition. Since in-cylinder pressure, and therefore temperature, is lower for more advanced timing, the occurrence of misfire increases as spark is advanced. This figure shows that IMEP is quite sensitive to change of phase at idle timing, whereas close to MBT the curve

Table 2 The operating condition selected

Engine speed	800 RPM
Inlet manifold pressure	0.32bar
Inlet air temperature	299K
Air/fuel equivalence ratio	1.0
Air mass flow rate	0.60g/s
Gross IMEP	1.55bar
Spark timing	5, 10, 15, 20, 25, 30, 35° BTDC

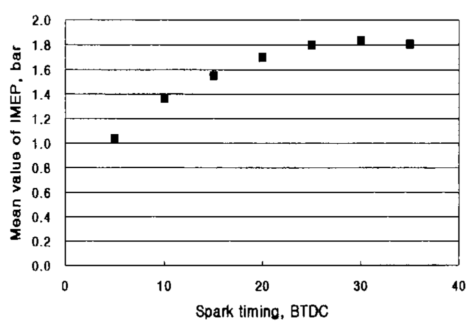

Fig. 1 Mean value of IMEP vs. spark timing 5, 10, 15, 20, 25, 30, and 35°BTDCs

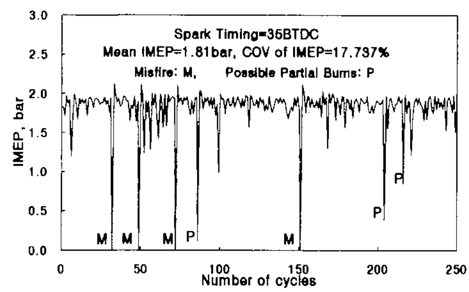

Fig. 2 Individual-cycle IMEP for 250 consecutive engine cycles in a production engine at idle

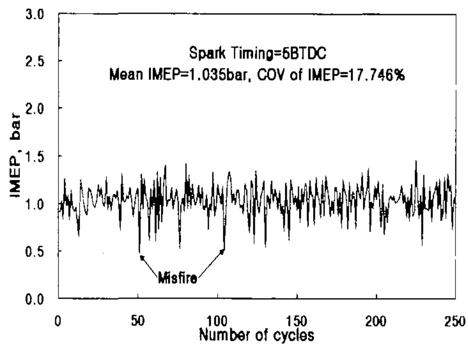

Fig. 3 Individual-cycle IMEP for 250 consecutive engine cycles in a production engine at idle

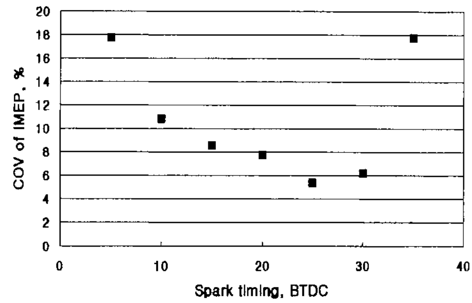

Fig. 4 Cofficient of variation of IMEP vs. spark timing 5, 10, 15, 20, 25, 30, and 35°BTDCs

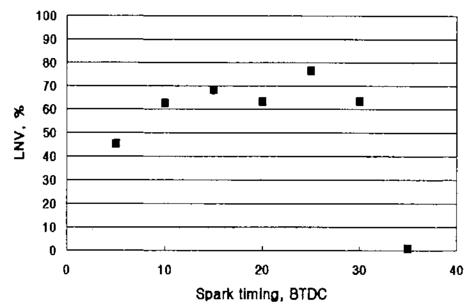

Fig. 5 Lowest normalized value for spark sweep

becomes flat. Thus, relative changes in combustion phase have a larger influence on IMEP at idle because of this high sensitivity.

One important measure of cyclic variability, derived from pressure data, is the coefficient of variation in indicated mean effective pressure. It is the standard deviation in IMEP divided by the mean IMEP. It defines the cyclic variability in indicated work per cycle, and it is found that the vehicle driveability problems usually result when COV of IMEP exceeds about 10%.

Figure 4 shows the influence of combustion phase on the COV of IMEP. As the spark is advanced, the COV goes down because relative changes in combustion phase have a smaller effect on the IMEP, as the spark timing indicated. However, for 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.

Hoard (1997) proposed another 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 correlated well with driver's subjective rating of engine smoothness. The LNV is defined as

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

where $IMEP_{min}$ is the minimum IMEP value and $\overline{IMEP}$ is the mean IMEP value. This value is plotted for the spark sweep in Fig. 5. Also, Hoard (1997) suggested that an appropriate acceptable value for the lower limit of LNV would be 75%. As the figure shows, the experimental engine only matches that criterion at 25° BTDC spark timing.

Figure 6 shows the mean value of burn angle versus spark timing for 0-2%, 0-10%, 0-50%, 0-90%, and 10-90% burn angles.

The coefficient of variation of burn angle versus spark timing for 0-2%, 0-10%, 0-50%, 0-

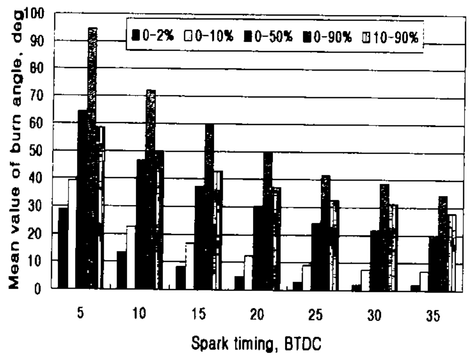

Fig. 6 Mean value of burn angle vs. spark timing for 0-2%, 0-10%, 0-50%, 0-90%, and 10-90% burn angles

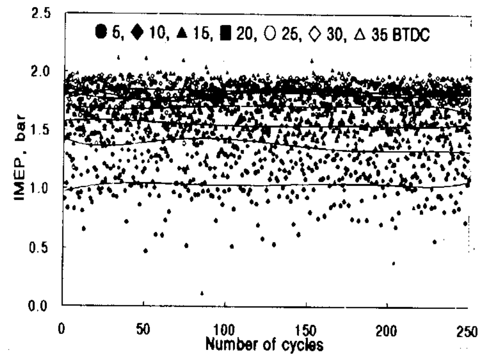

Fig. 8 IMEP vs. number of cycles for spark timing 5, 10, 15, 20, 25, 30, and 35°BTDCs

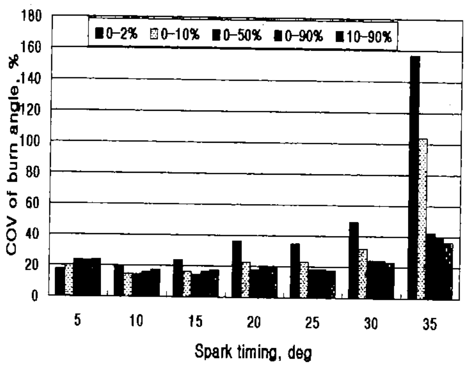

Fig. 7 COV of burn angle vs. spark timing for 0-2%, 0-10%, 0-50%, 0-90%, and 10-90% burn angles

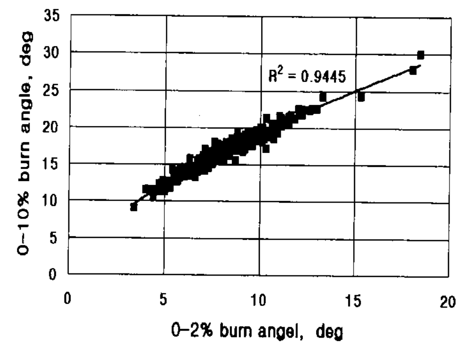

Fig. 9 0-10% burn angle vs. 0-2% burn angle for 250 cycles at 15° BTDC spark timing

90%, and 10-90% burn angles was shown in Fig. 7.

Figure 8 shows the IMEP versus number of cycles for spark timing 5, 10, 15, 20, 25, 30, and 35° BTDCs.

The early burn period is characterized by the 0-10% burn angle, also known as the flame development angle; it represents the crank angle interval between spark and the time when 10% percent of the charge mass has been burned. This is often taken as a measure of the time to achieve a fully developed turbulent flame in the cycle. By the time when 10% of the charge mass is burned, the flame may be as large as 30% of the total combustion chamber volume (Gatowski 1984).

Thus, the 0-2% burn angle may be preferable when the combustion period of interest is the very early flame development; however, it is found that the resolution of the burn rate analysis is insufficient for detecting changes in 0-2% burn angle between the perturbed and non-perturbed cycles.

Figure 9 shows a plot of the 0-10% burn angle versus the 0-2% burn angle for 250 cycles at idle. The R^2 value is 0.9445, demonstrating a very strong correlation between the two burn parameters; this suggests that the 0-10% burn angle may be considered an adequate measure of the early stages of flame development in an engine.

The 0-50% burn angle may be considered as an approximate measure of the time which 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 (Ho 1987). 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 phase 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.

In production engines and race engines, IMEP variability becomes highest at low speed conditions because of backflow of combustion products into the intake system. Misfires (negative IMEP) and very poor burning cycles (low IMEP) can be quickly identified by investigating IMEP on an individual-cycle basis (250 consecutive engine cycles from a production engine at idle are plotted in Fig. 2 and 3). Engine design features which may impact IMEP variability include in-cylinder charge motion, ignition system design, excessive valve-event overlap (in-cylinder dilution caused by backflow of exhaust into the intake tract during valve overlap), and poor combustion phasing. Examining IMEP as a function of the location of peak pressure on an individual-cycle basis can provide an indication of three important parameters: phasing of the combustion event relative to phasing for peak torque, overall burn variability of the midpoint of the burn, and the torque toss generated by this variability.

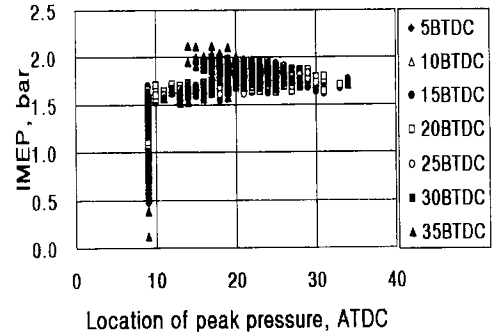

Fig. 10 Individual-cycle IMEP as a function of the location of peak pressure at several different spark timing in a production engine at idle

Figure 10 shows IMEP as a function of location of peak pressure for several spark timings at a very stable operating condition (250 individual cycles plotted at each spark timing). The crank angle for peak cylinder pressure can be used as a measure of bulk combustion phasing because it generally occurs near the midpoint of burn. Moreover, the location of peak pressure is one of the most sensitive combustion parameters because it usually occurs near the crank angle of peak combustion rate. Optimal phasing generally corresponds to a location of peak pressure of between 15 and 25 degrees ATDC. At a given operating condition, the minimum spark advance at which an average engine cycle is optimally phased defines MBT spark timing. IMEP is maximized when combustion is optimally phased. Spark timing either advanced or retarded from MBT lowers IMEP. Similarly, combustion phasing which is either advanced or retarded from optimum phase lowers IMEP. On an individual-cycle basis, cyclic phase variability causes a distribution in individual-cycle IMEP about the mean. If IMEP continually increases as the phasing retards, the spark is advanced from MBT. Conversely, if IMEP continually decreases as the phasing retards, the spark is retarded from MBT.

4. Conclusions

The results of this study suggest that a profit-ıble method of investigation for reducing cycle-o-cycle combustion variations. To analyse the :yclic variation in spark ignition engine at idle, a nethodology was developed whereby the input 'ariations in air mass, fuel mass, and residual nass could be identified through analysis of the 'ariations in the output burn parameters, i. e., the)-2%, 0-10%, 10-50%, 0-90%, and 10-90% ›urn angles, IMEP, COV of IMEP, LNV, and so ›n.

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