Effect of Engine Parameters on Cyclic Variations in Spark Ignition Engines

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Abstract—Elimination or reduction of cyclic variations in combustion and in the subsequent pressure development in the cylinder can help, to some extent, move toward the required fuel economy and emissions level. From the work that has been done it appears that higher engine emissions would result from cyclic variations in combustion. Another very important benefit stemming from control of cyclic variation is the reduction in engine surge and improved steady state vehicle derivability. Noise due to engine roughness might be reduced by controlling cyclic pressure variations.

In this study, flow characteristics were changed by using different types of piston and intake valve and effects of these parameters in cycle-to-cycle pressure and combustion variations inside a spark ignition engine were observed.

Keywords—Spark ignition engines, pressure, combustion, cyclic variations, turbulence.

I. INTRODUCTION

The phenomenon of cycle-to-cycle variation is widely known in the case of spark ignition engines. Even under constant conditions, consecutive cycles are not exactly the same; combustion process does not progress in the same way, resulting a different in-cylinder pressure curve [1]. Combustion in spark ignition engines varies appreciably from cycle to cycle in maximum pressure (and also in flame speed and combustion duration) [2].

Cycle-to-cycle variation is most important problem for spark ignition engine. Cyclic combustion variability (CV) in spark ignited (SI) engines has been observed and debated for over 100 years. Proposed explanations for the causes have ranged from turbulent, in-cylinder mixing fluctuations to deterministic effects of residual gas (the so-called prior cycle effect). The CV issue continues to be important because these combustion instabilities are responsible for higher emissions and limit the practical levels of lean-fueling and EGR which can be achieved [3].

Concerns about cyclic variability (CV) are highly relevant today because economic and regulatory pressures are pushing engine manufacturers to design engines that are particularly prone to this problem. For example, there is a trend to operate automotive engines with lean fueling and exhaust-gas recirculation (EGR) to increase fuel economy and minimize NOx emissions. CV occurs more frequently with lean fueling and EGR and actually limits the potential benefits, which can be derived from these operating modes [4].

Recent studies have demonstrated that cyclic combustion variations in spark-ignition engines under lean fueling exhibit patterns that can be explained as the result of noisy nonlinear combustion instabilities. These instabilities are dominated by the effects of residual cylinder gas and noisy perturbations of engine parameters. Because this dynamical noise obscures the underlying deterministic patterns, it is difficult to observe changes in these patterns as nominal engine parameter values are changed [5].

Studies are done in order to eliminate or reduce cycle-to-cycle variations as possible. By controlling cycle-to-cycle variations, fuel consumption, emission level, and noise due to engine roughness may be reduced, and also engine drivability may be improved.

The quantity of most interest is often the cylinder pressure time history, since this provides a direct and practical measure of combustion, as well as representing the primary motive force. Even in nominally steady state conditions however, real cylinder pressure data exhibits considerable cycle-to-cycle variability [6]. If the cyclic variability were eliminated, there would be even 10% increase in the power output of the engine [7].

II. CYCLIC VARIATIONS

The flame development and subsequent propagation obviously vary, cycle by cycle, since the shape of the pressure, volume fraction enflamed, and mass fraction burned curves for each cycle differ significantly. This is because flame growth depends on local mixture motion and composition. These quantities vary in successive cycles in any given cylinder and may vary cylinder-to-cylinder. Especially significant are mixture motion and composition in the vicinity of the spark plug at the time of spark discharge since these govern the early stages of flame development. Cycle-to-cycle and cylinder-to-cylinder variations in combustion are important because the extreme cycles limit the operating regime of the engine [8].

Observation of cylinder pressure versus time measurements from a spark ignition engine, for successive operating cycles, shows that substantial variations on a cycle-to-cycle basis exist. Since the pressure development is uniquely related to the combustion process, substantial variations in the combustion process on a cycle-to-cycle basis are occurring. In addition to these variations in each individual cylinder, there can be significant differences in the combustion process and pressure development between the cylinders in a multicylinder engine. Cyclic variations in the combustion process are caused by variations in mixture motion within the cylinder at the time of spark. Cycle by cycle variations in the amounts of air and fuel fed to the cylinder.
A cycle-to-cycle variation in the combustion process is important for two reasons. First, since the optimum spark timing is set for the “average cycle”, faster than average cycles have effectively over advanced spark timing and slower than average cycles have retarded timing, so losses in power and efficiency result. Second, it is the extremes of the cyclic variations that limit engine operation. The fastest burning cycles with their over advanced spark timing are most likely to knock. Thus, the fastest burning cycles determine the engine’s fuel octane requirement and limit its compression ratio. The slowest burning cycles, which are retarded relative to optimum timing, are most likely to burn incompletely. Thus these cycles set the practical lean operating limit of the engine or limit the amount of exhaust gas recycle (used for NO emissions control) which the engine will tolerate. Due to cycle-to-cycle variations, the spark timing and the average air/fuel ratio must always be compromises, which are not necessarily the optimum for the average cylinder combustion process. Variations in cylinder pressure have been shown to correlate with variations in brake torque, which directly relate to vehicle drivability [8]. Elimination or reduction of cyclic variations in combustion and in the subsequent pressure development in the cylinder can help, to some extent, move toward the required fuel economy and emissions level. If cyclic combustion variations could be controlled so that all cycles were made to burn as well as the best cycle, some fuel economy improvements might be realized. Another very important benefit stemming from control of cyclic variation is the reduction in engine surge and improved steady observe vehicle drivability, particularly with lockup torque converters and manual transmissions, which do little to damp out engine torque variations. Noise due to engine roughness might be reduced by controlling cyclic pressure variations.

Fuel lean operation is desired in spark ignition engines to reduce nitrogen oxides and hydrocarbon emissions as well as improve fuel efficiency. One of the major constraints to practical lean operation has been the large number of misfires and partial burns. A single misfire is capable of destroying modern catalytic converters. Misfires and partial burns are caused when cyclic variations are large enough to push the local in-cylinder equivalence ratio for cycle very near to or less than the lean limit. Therefore, minimization of cyclic variation is a key requirement for operating near to or extending the effective limit [9].

Swirl is a form of rotating bulk flow inside the engine cylinders. The axis of this type of flow is parallel to the axis of the cylinder. In general the purpose of introducing swirl flow into the cylinder of spark ignition engines is to increase turbulence intensity. This in turn increases combustion rate and extends the flammability limit which may lead to improve thermal efficiency. Along with improving efficiency, fast burning may reduce hydrocarbon (HC) and carbon monoxide (CO) emission because of reduction in cyclic variations. It is only during the last three decades that the laser velocimetry techniques have been available to measure instantaneous in-cylinder velocity and scalar properties under motored and firing conditions [10].

### III. Material and Methods

To understand the cycle-to-cycle variations in this study, experimental data are used for plotting the required graph. These data are taken during an experiment, which is done by Aydin in Liverpool in England [11].

In this study, the main element of the experiment is a Ricardo E6/T variable compression engine used in the standard petrol engine configuration which is driven by electric motor for starting the engine and converted to a generator can than measure the brake torque. It is used with a petrol engine head in this study. The combustion chamber is cylindrical in shape, the ends being formed by the flat surfaces of the cylinder head and piston. This gives a compact combustion chamber of good anti knock quality. Later on in the research, a bowl in diesel piston is also used to create different amount of mixture motion in the cylinder to compare the results with the flat piston. Two types of inlet valves, standard and shrouded, are used to investigate the effectiveness of the inlet swirl on the combustion. Tests were performed on the Ricardo E6/T engine at the full throttle and with the configurations given in table 1.

<table>
<thead>
<tr>
<th>Engine configuration</th>
<th>Bore (mm)</th>
<th>Stroke (mm)</th>
<th>Engine Speed (rpm)</th>
<th>Compression ratio</th>
<th>Connecting rod length (mm)</th>
<th>Swept volume (ccs)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>76.2</td>
<td>111.125</td>
<td>1000-3000</td>
<td>4.5 - 20</td>
<td>241.9782</td>
<td>506</td>
</tr>
</tbody>
</table>

Data has been collected during the periods of 364 cycles.

**Figure 1. Pistons and valves those are used in experiments**
dimensional in that they offer no spatial resolution. The combustion process can be considered to be modeled as either one or two zones. In a single zone procedure, no differentiation is made between burnt (product) and unburnt (reactant) gas properties; a mean temperature and pressure define the system state [11].

The MFB is determined by time marching through from spark time to exhaust valve opening time (EVO), using the iteration scheme. The two zone analysis allows tracking of concentrations of particular chemical species. An initial estimate of the MFB during a crank angle interval is made, and the resultant product gas temperature. The correct MFB can then be found by iterating the procedure until closure of the first law occurs to within a certain accuracy [11].

The simulation of the combustion process is based upon a homogenous gas-air combustible mixture through which the flame propagates from the spark. This automatically introduces the concept of two very distinct zones separated by the infinitely thin flame front. The pressure in reactant and product zones is assumed the same but all other properties are different, however assumed homogenous within the zones. Heat transfer to the cylinder walls and product dissociation are considered. The properties of the reactants and the products are determined by balancing the first law of thermodynamics for each zone. Heat transfer is calculated using Annand’s equations [12].

B. Calculation of Wall Temperature

The wall temperature is calculated using an existing procedure of the compression stroke (between inlet valve closure (IVC) and spark timing) in which the gas temperature is made equal to the wall temperature when the heat transfer is zero and there is no combustion. Assuming for a step:

\[ P \cdot V^n = \text{Constant} \]  

then

\[ \ln(P) = -n \cdot \ln(V) + \ln(\text{Constant}) \]  

is found. A second order curve is fitted to pressure data between IVC and spark timing and ‘n’ is calculated from the slope of the curve. Specific heat \((C_p)\) is a function of temperature \((T)\) for a step then

\[ \frac{\partial Q}{\partial t} = \gamma \cdot n \cdot p \cdot dV = h \cdot A_w \left( T_w - T_e \right) \]  

can be written. When ‘\((\gamma-n)\)’, then the reactants temperature should be equal to the wall temperature and the wall temperature is found.

The same procedure is applied on the late part of expansion assuming combustion is complete and there reactant in the cylinder to calculate coefficient ‘A’, however during the misfire and late burning cycles incorrect coefficient ‘A’ is obtained. Therefore, coefficient ‘A’ is adjusted to meet 100% MFB.

C. Computer Program Algorithm

After reading the pressure value \((P_i)\), calculating the cylinder volume \((V_i)\), calculating reactant temperature \((T_i)\) and setting the fraction burnt to zero, the end of the step is designated by 2. After values become the beginning values of the next step and this is done until the whole cycle is solved or MFB exceeds a predetermined value. If there are a total of ‘N’ points, there will be up to ‘N-1’ intervals to be solved.

D. Statistical Analysis

A two zone model of reactants and products separated by a thin flame front is used to determine the mass fraction burnt (MFB) from an indicator diagram. The model incorporates dissociation and uses the Annand method for heat transfer. Having calculated the MFB, it is necessary to quantify the results. The data is here used for two proposes, one to calculate an equivalent flame speed and surface area from the idealized geometry of the product zone, the other to study the initial ignition period 0-5% MFB (delay time), and 5-95% MFB (main flame travel time or combustion time) [12].

The delay and flame travel time results can be processed to produce the distribution cycle by cycle as a histogram using Sturgess’s rule as follows [13]:

\[ k = 1 + 3.3 \log_{10} N \]  

where \(k\) = number of histogram in the group, \(N\) = number of data in the group.

Probability density function of normal variable can than be calculated as follow;

\[ f(Z) = \frac{1}{\sqrt{2\pi} \cdot \sigma} e^{-\frac{(Z-\mu)^2}{2\sigma^2}} \]  

where \(f(Z)\) = probability density function  
\(\mu = \text{mean} \)  
\(\sigma = \text{variance} \)

The histograms usually, but not always, have a non normal distribution with more long period events than short period. The distribution is therefore converted to a log-normal distribution with horizontal axis converted to

\[ Z = \log(Z - Z_{\text{min}}) \]  

where ‘\(Z_{\text{min}}\)’ is the minimum value possible, which in the case of a time axis could physically mean the minimum possible delay, or flame travel time.

In obtaining a fit of a normal distribution to the log-normal histogram, ‘\(Z_{\text{min}}\)’ is adjusted to produce a minimum error, assessed by a chi-squared test on the result [13] Chi-squared significance test is applied such that:

\[ \chi^2 = \sum_{i=1}^{n} \frac{(O_i - E_i)^2}{E_i} \]  

where

\(O_i\) (i=1,2,...,N) = observed frequencies  
\(E_i\) (i=1,2,...,N) = expected frequencies

Should a normal fit produce a lower error than the log-normal then the normal is accepted as the best fit. From the distribution a most probable, and a maximum and a minimum value can be obtained taken here as 99.9% probability (±3.29 standard deviation) [14].

IV. RESULTS AND DISCUSSION

The gained results are plotted by SURFER computer program in order to recognize the cycle-to-cycle variations. As explained in the previous section, during the experiment the
engine operating parameters and engine components are changed. The aim is to understand the effect of piston type; valve; compression ratio (CR), engine speed; air fuel ratio (AFR) and ignition advance on cycle-to-cycle variations. To observe the variations clearly we compare the total combustion time vs. delay time; flame travel time vs. delay time according to cycle number.

Using flat piston, shrouded valve, CR=7:1, N=2000 rpm, AFR=15:1, IA=35ºCA cycles average delay time 1.92 msec and flame travel time 1.70 msec (figure 2). By keeping the entire variables constant and changing piston type to bowl-in; average delay time increased 2.060 msec and flame travel time 1.725 msec (figure 3). Using flat piston, shrouded valve, CR=7:1, N=2000 rpm, AFR=15:1, IA=35ºCA cycles average delay time 1.92 msec and total combustion time 3.650 msec (figure 4). By keeping all the variables constant and changing piston type to bowl-in; average delay time increased to 2.060 msec and total combustion time 3.775 msec (figure 5).

By keeping all the variables constant (figure 2) and changing inlet valve type from shrouded valve to standard valve; average delay time decreased to 1.69 msec and flame travel time to 1.725 msec (figure 6). By keeping all the variables constant (figure 4) and changing valve type to standard valve; total combustion time decreased to 3.10 msec (figure 7).

Figure 2. Change of flame travel time and delay time with number of cycles (flat piston, shrouded valve, CR=7:1, N=2000 rpm, IA=35ºCA, AFR=15:1)

Figure 3. Change of flame travel time and delay time with number of cycles (bowl-in piston, shrouded valve, CR=7:1, N=2000 rpm, IA=35ºCA, AFR=15:1)

Figure 4. Change of total combustion time and delay time with number of cycles (flat piston, shrouded valve, CR=7:1, N=2000 rpm, IA=35ºCA, AFR=15:1)

Figure 5. Change of total combustion time and delay time with number of cycles (bowl-in piston, shrouded valve, CR=7:1, N=2000 rpm, IA=35ºCA, AFR=15:1)

Figure 6. Change of flame travel time and delay time with number of cycles (flat piston, standard valve, CR=7:1, N=2000 rpm, IA=35ºCA, AFR=15:1)
and flame travel time decreased to 1.50 msec. By keeping all the variables constant (figure 4) and increasing ignition advance to 45°CA (figure 12) average delay time increased to 2.0 msec and total combustion time decreased to 3.625 msec and increasing ignition advance to 55°CA (figure 13) average delay time increased to 2.120 msec and total combustion time decreased to 3.625 msec.

By keeping all the variables constant (figure 2) and reducing the speed from 2000 rpm to 1500 rpm; average delay time increased to 2.41 msec and flame travel time 2.15 msec (figure 8). By keeping all the variables constant (figure 4) total combustion time increased to 4.575 msec (figure 9).

By keeping all the variables constant (figure 2) and increasing ignition advance to 45°CA (figure 10) average delay time increased to 2.0 msec and flame travel time decreased to 1.69 msec and increasing ignition advance to 55°CA (figure 11) average delay time increased to 2.120 msec and total combustion time decreased to 3.625 msec.

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Figure 7. Change of total combustion time and delay time with number of cycles (flat piston, standard valve, CR=7:1, N=2000 rpm, IA=35°BTDC, AFR=15:1)

Figure 8. Change of flame travel time and delay time with number of cycles (flat piston, shrouded valve, CR=7:1, N=1500 rpm, IA=35°CA, AFR=15:1)

Figure 9. Change of total combustion time and delay time with number of cycles (flat piston, shrouded valve, CR=7:1, N=1500 rpm, IA=35°CA, AFR=15:1)

Figure 10. Change of flame travel time and delay time with number of cycles (flat piston, shrouded valve, CR=7:1, N=2000 rpm, IA=45°CA, AFR=15:1)

Figure 11. Change of flame travel time and delay time with number of cycles (flat piston, shrouded valve, CR=7:1, N=2000 rpm, IA=55°CA, AFR=15:1)

Figure 12. Change of total combustion time and delay time with number of cycles (flat piston, shrouded valve, CR=7:1, N=2000 rpm, IA=45°CA, AFR=15:1)
By keeping all the variables constant (figure 2) and changing AFR from stoichiometric mixture (15:1) to lean mixture (18:1); average delay time increased to 2.33 msec and flame travel time to 2.31 msec (figure 14). By keeping all the variables constant (figure 4) total combustion time increased to 4.65 msec (figure 15).

By keeping all the variables constant (figure 2) and changing compression ratio from 7:1 to 8:1; average delay time increased to 2.290 msec and flame travel time to 2.450 msec (figure 16). By keeping all the variables constant (figure 4) and changing compression ratio from 7:1 to 8:1; average total combustion time increased to 4.625 msec (figure 17).

V. CONCLUSION

Gaining results and effects of performance parameters are given below in detail.

A. Effect of Piston Type

Using bowl-in piston instead of flat piston, however fresh charge causes an increase in the turbulence kinetic energy the distance from the spark plug to the centre of bowl-in is longer than the flat piston which raised delays time 7.29%, flame travel time 1.47%, total combustion time 3.42%.

B. Effect of Valve Type

When using shrouded valve; fresh charge will enter into the cylinder with a swirl which caused homogenous mixture inside the cylinder. Therefore the turbulence kinetic energy is increased and that caused a faster burning thus cyclic variation reduced. Using normal standart valve instead of shrouded valve; delay time 19.27%, flame travel time 44.12%, total combustion time 27.26% increased.

C. Effect of Engine Speed

A 500 rpm reduction in the engine speed causes a reduction in the turbulence kinetic energy caused delay time 25.52%, flame travel time 26.47% and total combustion time 25.34% increased. Finally a decrease in the turbulence kinetic energy caused a reduction in the mixing quality and flame speed.
which causes the increase in the delay time, total combustion time and flame travel time.

D. Effect of Ignition Advance

As the ignition advance increased cycle to cycle variation increased this cause an increase in the delay time.

E. Effect of Air/Fuel Ratio

An increase in the air/fuel ratio from 15:1 to 18:1 caused an increase in the distance between fuel molecules so delay time 20.31%, flame travel time 37.06%, total combustion time 27.40% were increased.

F. Effect of Compression Ratio

Increasing compression ratio from 7:1 to 8:1 thermal efficiency increased thus average delay time 12.5%, flame travel time 15.29% and total combustion time 15.07% decreased.

The experimental results showed that an increase in turbulence kinetic energy is caused an increase in the quality of mixture and flame speed on the other hand total combustion time, delay time, flame travel time and cyclic variations are reduced.

REFERENCES