Cycle-by-cycle variations in a spark ignition engine fueled with gasoline and natural gas

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Abstract. A comparative study of cycle-by-cycle variations (CCV) in a spark ignition (SI) engine fueled with gasoline and natural gas was conducted. The average peak cylinder pressure (PP), indicated mean effective pressure (IMEP) and maximum rate of pressure rise ((dp/dθ)max) are found to be low for engine fueled with natural gas in comparison to that of gasoline. However, the CCV of these parameters increases as engine operation is switched from gasoline to natural gas. The coefficient of variation (COV) of PP, IMEP and (dp/dθ)max significantly increases at very low load condition for both the fuels due to high residual gases. Natural gas engine shows more misfiring cycles in comparison to gasoline engine. Further, CCV increases with the increase of engine speed for both the fuels because of an increase in turbulence and instable combustion at high engine speed. As engine speed increases and engine load decreases the combustion based parameters become more independent of crank angle due to high CCV in these conditions.

Keywords: Cylinder pressure, gasoline, natural gas, CCV

1. Introduction

In recent years, considerable efforts have been made to search for alternative fuels that can burn efficiently in combustion engines satisfying emission norms. Natural gas is considered to be the most viable alternate option to gasoline for spark engine [1-4]. The engines fueled with natural gas have less tendency to knock due to its high ignition temperature and high octane number [5-7]. The higher octane number allows the engine to be operated with a higher compression ratio, which results in better thermal efficiency. Natural gas engines have lower brake specific fuel consumption (bsfc) and higher fuel combustion efficiency in comparison to gasoline engines. Further, natural gas has a low carbon/hydrogen (C/H) ratio, which results in a significant decrease of exhaust emissions like hydrocarbon (HC), carbon mono-oxide (CO), and carbon dioxide (CO2)[4,8-11].

Despite these advantages, the brake power delivered by natural gas engines is lesser by 8-16% in comparison to gasoline engines. It is due to low volumetric efficiency and low flame speed of natural gas. Natural gas also produces high oxides of nitrogen (NOX) emissions as compared to gasoline [6,12,13]. Further, low flame speed of natural gas results in incomplete combustion and high misfire ratio, specially under lean mixture operating conditions. As a consequence, higher CCV is present in engine operated with natural gas in comparison to gasoline. The presence of CCV reduces the power output of the engine by 10% and increases specific fuel consumption by 6% [14]. CCV also increases HC, CO and NOX emissions [14]. The misfire denotes an extreme case of CCV with high HC emissions and poor driveability [14-19]. Most of the work reported in literature discusses the influence of equivalence ratio on CCV of natural gas SI engine [20-22]. The effect on CCV with the change in
Figure 1. Schematic diagram of experimental setup along with the operating point.

other operating variables like engine load and speed is seldom discussed [23]. Further, research reported in the literature regarding the comparative analysis of CCV for SI engine fueled with natural gas and gasoline is extremely limited.

In this work, a detailed comparison of CCV between gasoline and natural gas fueled SI engine has been performed for different load and speed conditions using cylinder pressure based parameters like PP, IMEP and \((dp/d\theta)_{\text{max}}\). The cylinder pressure based variables are chosen as any variations in combustion process immediately get reflected in variation in cylinder pressure trace [20]. Further, measurement of cylinder pressure is comparatively easier in comparison to the measurement of other combustion related parameters like flame speed, burn angle, etc. The test rig used to perform the study of CCV is described in the next section.

2. Experimental Setup

The experiments to study CCV are performed on a single-cylinder, 4-stroke air-cooled engine. The engine is modified to operate on both gasoline and natural gas. A natural gas kit consisting of a high-pressure pipe, pressure reducer, natural gas and mixer, and the emulator is connected and installed with the engine. Eddy current dynamometer is used to load the engine. It can be operated at a constant torque and constant speed mode with the help of a PID controller. Cylinder pressure is measured at a resolution of one crank angle degree using piezoelectric combustion pressure. Other sensors installed on the set up are intake manifold pressure sensor, throttle angle position sensor, air flow sensor, fuel meter, universal exhaust gas oxygen sensor and crank angle encoder. Data acquisition system with suitable data acquisition card is used to acquire data. The schematic diagram of the engine test rig is shown in figure 1.
3. Processing of cylinder pressure data

The measured cylinder measure data is processed as explained below:

3.1 Number of cycles

The number of cycles for CCV study is chosen such that the 99.9% confidence limit of the population mean for normalized cylinder pressure at 10° after top dead centre (ATDC) do not differs from the sample mean by more than 3% \([24,25]\). The normalized pressure and its 99.9% confidence limits for reference operating condition for the engine operated with gasoline and natural gas are shown in figure 2. At high loads, relatively few cycles are needed to ensure the above condition. However, at low loads same confidence is achieved, with cylinder pressure measurement of at least 120 cycles. To maintain uniformity, 120 recorded cycles are used at all operating points for the CCV analysis.

3.2 Absolute pressure referencing

The measured pressure data must be scaled using absolute pressure reference as given by equation (1) \([25,28]\).

\[ p_m(\theta) = p(\theta) + p_0 \]  

(1)

where \(p\) is the actual combustion chamber pressure, \(p_m\) is the pressure reading obtained using sensor

![Figure 2](image_url). Normalized, averaged cylinder pressures vs.the number of cycles considered with 99.9 % confidence limit.

![Figure 3](image_url). LogP-logV diagram for (a) a fired cycle with arbitrary absolute pressure reference values. (b) a motored cycle arbitrary phasing.
and $p_0$ is the absolute cylinder pressure reference. The magnitude of shift ($p_0$) is determined by selecting any point on cylinder pressure curve during intake stroke. The pressure at this point is shifted so as to make it equal to the average reading of intake manifold pressure sensor, which is much easier to calibrate. After that, all the data is scaled using the same amount of shift as that of the difference in cylinder pressure and averaged manifold pressure at the selected point. Log-p-logv diagram is used to verify the correctness of pressure referencing. If pressure referencing is wrong, then as shown in figure 3(a) curvature is observed instead of a straight line as compression starts [25-28]. The pressure curve remain unaffected near TDC by absolute pressure reference.

### 3.3 Pressure Curve Phasing

Commonly correct pressure curve phasing is done using data from motored cycle obtained during cranking of engine. The pressure data for the nonfiring cycle is analyzed near TDC. It is understood that the peak pressure (PP) should occur around two degree before TDC due to the heat transfer through combustion chamber walls. If PP occurs more than two degrees before TDC or after TDC then pressure data should be shifted w.r.t. crank angle [25,26,28]. A quick check could be made to verify correct pressure curve phasing by using log-p-logv diagram for motored cycle which will show the expansion and compression curves to meet at a sharp point in case of correct phasing. Otherwise, in case of incorrect phasing a top loop will appear near TDC as shown in figure 3(b).

### Table 1. CCV measures for gasoline and CNG fueled engine.

| Factor | Gasoline | Gasoline | CNG | CNG |
|--------|----------|----------|-----|-----|
|        | N=2500 rpm, $T_L=5$ Nm | N=2500 rpm, $T_L=20$ Nm | N=2500 rpm, $T_L=5$ Nm | N=2500 rpm, $T_L=20$ Nm |
| $\Delta \left( \frac{dP}{d\theta} \right)_{\text{max}}$ (bar/deg) | 0.36 | 0.59 | 0.44 | 0.78 |
| $COV \left( \frac{dP}{d\theta} \right)$ (%) | 23.74 | 5.63 | 25.22 | 7.49 |
| $\Delta PP$ (bar) | 9.38 | 3.79 | 10.41 | 6.01 |
| $COV_{PP}$ (%) | 18.82 | 1.44 | 19.12 | 2.37 |
| $\Delta IMEP$ (bar) | 4.36 | 0.45 | 4.41 | 5.37 |
| $COV_{IMEP}$ (%) | 28.89 | 0.93 | 33.77 | 1.65 |

### 3.4 Data smoothing

The signal noise in the recorded p-θ diagram is removed by using a filter `filtfilt` available in the signal processing toolbox of Matlab®. It is a moving average method in which data is averaged in both forward and backward direction. By doing so, any phase shift is removed which is usually observed if data is processed in just one direction.

### 4. Measures of CCV

The CCV is measured using either absolute or relative measures of cyclic dispersion. The absolute measure usually involves the spread of a particular parameter over the number of cycles measured, whereas the coefficient of variation (COV) of a parameter is usually used as a relative measure to define cycle-by-cycle dispersion. The p-θ and p-V diagrams shown in figure 4 (a) and (b) for gasoline and natural gas, respectively, is used to decide whether to use absolute or relative measure for cyclic dispersion [14,27].

Each figure shows the fastest, the slowest and an average cycle for a low load ($T_L=10$ Nm) and a high load ($T_L=20$ Nm) condition. Table 1 shows the spread and COV for $(dp/d\theta)_{\text{max}}$, PP and IMEP for low
load and high load condition. The data are presented for both gasoline and natural gas fueled engine. It is seen from figure 4 (a) and (b) that cyclic dispersions are more at low load in comparison to high load for both gasoline and natural gas engine. However, it is seen in Table 1 that the spread of various parameters does not reflect the same. For an instance, the value of spread of $(dp/d\theta)_{\text{max}}$ at low load and high load are quite close to each other, despite high CCV at low load in comparison to high load.

Figure 4(a). P-\(\theta\) and P-V diagram for gasoline fueled engine at different loads.

Figure 4(b). P-\(\theta\) and P-V diagram for CNG fueled engine at different loads.
Figure 5. (a) COV_{PP}, (b) COV_{imep} and (c) COV_{(dP/dθ)max} w.r.t. load torque for gasoline and CNG.

Hence, the spread of cylinder pressure based parameters is not considered as a suitable measure for quantifying cycle-by-cycle dispersion. Further, it appears from Table 1 that CCV are better represented using COV of various parameters. For example, at low load (T_L=10 Nm) condition, COV_{(dP/dθ)max} is 9.38, which is approximately 1.65 times higher than COV_{(dP/dθ)max} at high load (T_L=20 Nm). Moreover, it is seen that COV_{(dP/dθ)max} is more sensitive to CCV in comparison to COV_{PP} and COV_{imep}. This is because the rate of pressure rise more closely represents the variation in the flame propagation process. In the present study comparison of CCV for gasoline and natural gas is done using COV of all the three parameters.

5. Engine measurement and results

This section discusses the results obtained from the experiments conducted on the test setup described in Section 2. The measurements were made for 120 consecutive combustion cycles at operating points shown in figure 1, which are obtained by changing torque and engine rpm around a nominal operating point of T_L = 15 Nm and N= 2500 rpm. The spark advance and equivalence ratio are fixed at constant values of 20° ATDC and 1, respectively. The results obtained are discussed below:

5.1 Effect of load torque on cycle-by-cycle variation

The comparative experimental results of CCV for varying load and at a fixed engine speed of 2500 rpm are presented in figure 5 for engine fueled with gasoline and natural gas. Figure 5(a) shows that the variation in PP over 120 cycles decreases with increase in engine load, for both gasoline and natural gas. The COV_{PP} (figure 5(a)) significantly decreases from 13.17% to 4.96% as engine load is changed from 5 Nm to 10 Nm and then steadily decreases from 4.96% to 1.62% as load torque is further increased from 10 Nm to 20 Nm. A similar trend is observed for natural gas fueled engine with a sudden change in COV_{PP} as load torque is decreased from 5 Nm to 10 Nm and a steady decline from 7.89% to 3% as engine load is further increased from 10 to 20 Nm. The high value of COV_{PP} at low loads is due to high residual gases at low load, which reduces the burning velocity of the mixture and may result in partial burns and misfires.
Further, it is seen in figure 5(b) that COV_{dp/dθ|max} at all operating points are higher in comparison to COV_{PP} for both gasoline and natural gas. This reflects a higher influence of burn velocity at a maximum rate of pressure rise in comparison to that of peak pressure. After that figure 5(c) shows the COV_{IMEP} as load torque is increased from 5 Nm to 10 Nm. The variation in COV_{IMEP} reflects the vehicle driveability which is considered to be driveable when COV_{IMEP}<10%. It is found that at high engine load IMEP spread is concentrated around its mean value. The engine fueled with natural gas shows more variation in IMEP in comparison to engine fueled with gasoline. At low engine load of 5 Nm couple of misfire cycles are seen for gasoline fueled engine, while for natural gas fueled engine misfired and partially burned cycles are more. This results in increase in CCV at engine load of 5 Nm with COV_{IMEP} to be 5.89 % and 9.39 % for gasoline and natural gas, respectively.

Figure 6 shows the interdependency of PP w.r.t. its location for different engine loads. It is seen that in comparison to natural gas, PP data band for gasoline is centered at a crank angle nearer to TDC. This reflects fast burning in case of gasoline in comparison to natural gas. As load torque increases the center of data band moves closer to TDC for both gasoline and natural gas. The early arriving of PP is due to fast combustion and increase of burning velocity. With the increase in load torque the PP spread

Figure 6. Interdependency between PP and LPP of gasoline and CNG for different load torque.
becomes more concentrated about its average value which also reflects decreased combustion variations. It may be attributed to less dependency of combustion behaviour on turbulence at high engine load. It is seen in figure 6 that for all the loads PP and LPP spread increases as combustion is retarded (A to B) in comparison to an average cycle. The spread at low loads is found to be less in case of gasoline in comparison to natural gas. At very low load, burning rate variations are more and for some of the cycles, combustion is so slow that the increase in pressure due to combustion gets offsets due to decrease in pressure due to increase in volume. LPP is found to be closer to TDC in these cycles, resulting in a classical hook-back effect [20,29,30].

Figure 7 shows the interdependency of IMEP w.r.t. crank angle. Average IMEP is found to be more in case of gasoline in comparison to natural gas. IMEP data band for natural gas is found to be centered closer to TDC in comparison to gasoline. The data bands at higher load for gasoline are found to be relatively flat. However, for natural gas, it shows some scattering due to the presence of partially burned cycles. The spread in IMEP at a fixed value of LPP is largely attributed to variations in the air-fuel ratio. However, for low loads, the variations are also due to partially burned and misfired cycles.
5.2 Effect of engine speed on cycle-by-cycle variation

The variation of averaged PP and IMEP over 120 cycles w.r.t engine speed is shown in figure 8(a). It is seen that PP and IMEP first increase and then decreases with engine speed for both the fuels. This can very well be related to the similar nature observed for variation in volumetric efficiency. At low engine speed, volumetric efficiency is low because of back flow and charge heating effect [16]. After that, it increases with increasing engine speed and later at high engine speed it further decreases due to increasing effects of flow friction and choking at high engine speed [16].

Figure 8(b) shows COVPP and COVimep w.r.t. engine speed, respectively for both the fuels. It is seen that COVPP and COVimep increases with increasing engine speed. This is due to the instability of combustion at high engine speed with an increase in turbulence. This instability in combustion is found to be more in the gaseous fuel like natural gas in comparison to a liquid fuel like gasoline. The residual gas fraction also increases with increasing engine speed, which results in dilution of the mixture. The dilution results in less heat release rate, the effect of which is more prominently seen in the interdependency plot of PP and IMEP w.r.t. LPP as shown in figure 9 and 10, respectively. It is seen that the spread in PP and IMEP increases with increasing engine speed. The center of the data band for both PP and IMEP shifts away from TDC as engine speed is increased. This is due to increasing burn duration and phasing of pressure curve w.r.t. crank angle. At very high engine speed, hook back effect is seen for a gasoline engine, with some partial burn cycles.

Figure 8 (a). Variation of averaged PP and IMEP over 120 cycles for different engine speed.

Figure 8 (b). COVPP and COVimep w.r.t. engine speed for gasoline and CNG.
Figure 9. Interdependency between PP and LPP of gasoline and CNG for different engine speed.
6. Conclusions
Cycle-by-cycle variations of spark ignition engine fueled with gasoline and natural gas fuel are studied. The main results are summarized as follows:

(i) The average value of PP and IMEP decreases when the engine is operated with natural gas in comparison to gasoline. The reduction in PP is about 7% and 13% for extreme engine loads of 20 Nm and 5 Nm, respectively. On the same line, IMEP reduces by 5.3% and 30% for engine loads of 20 Nm and 5 Nm, respectively.

(ii) The cycle-by-cycle variations (COV_{PP}, COV_{imep} and COV_{(dp/dθ)max}) increase as engine operation is switched from gasoline to natural gas.

(iii) Cycle-by-cycle variations decrease with the increase of load torque for both fuels.

(iv) Interdependency plot between PP and LPP shows data band to be closer to TDC in gasoline as compared to natural gas. This is due to fast combustion and increase of burning velocity. Further, data band also shifts towards TDC with increases in engine load. A couple of misfire cycle is observed in gasoline at low load condition but in case of natural gas, many more misfire and partial burn cycles are observed which leads to high cycle-by-cycle variations.

(v) COV_{PP}, COV_{imep} and COV_{(dp/dθ)max} decreases with the increase of load torque. The high value of COV_{PP} at low loads is attributed to high residual gasses.

(vi) PP and IMEP first increase with engine speed and then decrease for both the fuels. This is due to variation in volumetric efficiency w.r.t. engine speed.

(vii) COV_{PP}, COV_{imep} and COV_{(dp/dθ)max} increases with the increase of engine speed for both fuels. This is due to the instability of combustion at high engine speed with an increase in turbulence.
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