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Effect of engine parameters and gaseous fuel type on the cyclic variability of dual fuel engines

Mohamed Y.E. Selim[1]*

Mechanical Engineering Department, Faculty of Engineering, United Arab Emirates University, Jimi, P.O. Box 17555, Al-Ain, United Arab Emirates

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Abstract

This paper presents an analysis of the cycle-to-cycle combustion variation as reflected in the combustion pressure data of a single cylinder, naturally aspirated, four stroke, Ricardo E6 engine converted to run as dual fuel engine on diesel and gaseous fuel of LPG or methane. A measuring set-up consisting of a piezo-electric pressure transducer with charge amplifier and fast data acquisition card installed on an IBM microcomputer was used to gather the data of up to 1200 consecutive combustion cycles of the cylinder under various combination of engine operating and design parameters. These parameters included type of gaseous fuel, engine load, compression ratio, pilot fuel injection timing, pilot fuel mass, and engine speed. The data for each operating conditions were analyzed for the maximum pressure, the maximum rate of pressure rise—representing the combustion noise, and indicated mean effective pressure. The cycle-to-cycle variation is expressed as the mean value, standard deviation, and coefficient of variation of these three parameters. It was found that the type of gaseous fuel and engine operating and design parameters affected the combustion noise and its cyclic variation and these effects have been presented.

Keywords: Dual fuel engine; Cyclic variability; Combustion noise; LPG; Methane

1. Introduction

For the diesel engines, there has been usually a trade-off between several targets, e.g. reducing exhaust emissions, reducing engine noise, reducing fuel consumption and increasing specific outputs.

The noise emission of the diesel engines has been reduced in the past; however, further engine noise reductions are necessary and can be achieved by optimization of the engine structure and by reducing the combustion noise. The noise emission of the diesel engine is generally dominated by the combustion noise which leads to high sound pressure levels in the frequency range around 1600 to 2000 Hz. This is caused by pressure variations due to combustion chamber resonances of the main chamber/swirl chamber system and, in addition, low attenuation of the engine structure in the critical frequency range [1].

In contrast to spark-ignition engines the combustion of diesel engine is regarded as being very regular and stable. This is confirmed by observations of the cylinder pressure signal on an oscilloscope. However, it has been shown [1–3] that a detailed analysis of the diesel combustion process reveals significant variations from cycle to cycle resulting in combustion noise variations. Cycle-to-cycle variations in the sound pressure are typical of diesel engine noise. They are caused by cyclic fluctuations of the combustion excitation [4]. The variation in the combustion noise level of diesel engines can be as high as 12 dB which is one of the reasons for the annoying character of the diesel engine noise. It has been concluded [1] that there is a large potential for the improvement of noise emission if the significant cycle-to-cycle variations in combustion noise can be reduced. These cyclic variations can be reduced, e.g. by an optimized injection system which is part of required low emission, high efficiency diesel engines.

There have been many studies concerning the cycle variations and combustion stability of spark ignited gaseous engines [5–11]. The characteristics at ignition, the mixture

* Tel.: +971 3 7051566; fax: +971 3 7623158.
E-mail address: mohamed.selim@uaeu.ac.ae
[1] Permanent address: Mechanical Power Engineering Department, Faculty of Engineering at Matlaria, Helwan University, Cairo, Egypt.

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gas formation, and the combustion speed of the fuel that greatly influence the cycle variation are very different between CNG fuel and liquid fuel. It has been concluded that the combustion process becomes most unstable in lean combustion at part load due to non-uniform mixture and its cyclic variation.

However, in dual fuel engines that use the diesel as pilot fuel and the gaseous fuel as the main fuel, the ignition characteristics of the gaseous fuel are not well understood. In addition to the causes of cyclic variations exist in diesel engines, dual fuel engines appear to be more prone to cyclic variability in combustion noise due to the existence of gaseous fuel with the compressed air.

Therefore, it was considered necessary to research the cycle variation of the dual fuel engine that was not covered before. In the present study, the influence of the cycle variation in the dual fuel engine is considered in an attempt to improve the combustion stability and reduce the engine noise. The purpose is to focus on the attention of the combustion noise for the dual fuel engine, particularly on the phenomenon of combustion noise variations. The combustion pressure data are sampled for a number of consecutive cycles and analyzed for the purpose of presenting the cycle-to-cycle variation. Three parameters of each cycle represent the combustion variation [3]. These parameters are the maximum pressure, the maximum rate of pressure rise and the indicated mean effective pressure. The measures of the cycle-to-cycle variation are the standard deviation, mean value, and coefficient of variation of the three mentioned parameters. The effect of the dual fuel engine following variables on these parameters has been studied namely: type of gaseous fuel, engine load, compression ratio, pilot fuel injection timing, pilot fuel mass, and engine speed.

2. Engine test rig description

The test rig used in the present study is the Ricardo E6 single cylinder variable compression indirect injection diesel engine. The specifications of the engine are listed in Table 1. The engine cylinder head has a Ricardo Comet Mk V compression swirl combustion chamber. This type of combustion system consists of two parts. The swirl chamber in the head has a top half of spherical form and the lower half is a truncated cone which communicates with the cylinder by means of a narrow passage or throat. The second part consists of special cavities cut into the crown of the piston. The engine is capable to run on 100% diesel fuel or dual fuel. The engine is converted to run on dual fuel by introducing the gaseous fuel, pure methane, or LPG in the present work, in the intake manifold by a relevant nozzle. The gas is injected at a pressure slightly higher than atmospheric pressure.

Table 1
Engine characteristics

Model	Ricardo E6
Type	IDI with the pre-combustion chamber
Number of cylinder	1
Bore $\times$ stroke (mm)	76.2×111.1
Cycle	4-Stroke
Compression ratio	22
Maximum power (kW)	9, naturally aspirated
Maximum speed (rpm)	3000
Injection timing	Variable

The schematic diagram for the engine test rig is shown in Fig. 1. The engine is loaded by an electrical dynamometer rated at 22 kW and 420 V. The engine is fully equipped for measurements of all operating parameters and combustion noise data. The combustion pressure time history is measured by a water-cooled piezo-electric pressure transducer connected to the relevant amplifier. The liquid fuel flow rate is measured digitally by a multi-function micro processor-based fuel system, Compuflow System. The gaseous fuel flow rate is measured by using an orifice meter connected to electronic partial pressure transducer that is connected to a digital pressure meter. Two data acquisition systems are used to collect the important data and store it in two personal computers for offline analysis. The following parameters are fed into the first data acquisition system: liquid and gaseous fuel flow rate data, engine speed and torque, and air/oil/coolant/exhaust temperatures. A computer program in µMACBASIC language is written to collect the data and manage the system and a workstation operating system has been used to run the program.

The pressure signal amplified by a charge amplifier is fed to the second data acquisition card linked to a personal computer. The acquisition card could collect data at a rate of up to 250 kHz. A Labview acquisition program is used to sample and store the pressure data fed to one channel of the data acquisition card. A Labview sampling program has been written to collect 900,000 pressure–time data points at a fixed sampling rate of 10,000 points per second. This can represent a high number of consecutive engine cycles required for such cyclic variation investigation. The number of cycles chosen forms an acceptable limit for generating statistically meaningful results for all parameters examined. It has been shown by [3] that a greater number of cycles than 400 cycles forms a safe limit for a statistical analysis. In their cyclic variability study a 650 successive cycles have been taken. The number of consecutive cycles stored depends on the engine speed (for constant sampling rate) which has been kept constant at most of the experiments at 1300 rpm. At this engine speed, the stored points represented about 987 consecutive engine cycles. For the engine speed test the engine speed was varied to 1000 and then to 1600 rpm. At these engine speeds, the numbers of consecutive cycles stored are 762 and 1217 cycles, respectively. A Visual Basic computer program has been written in MS Excel (macro) to find maximum pressure

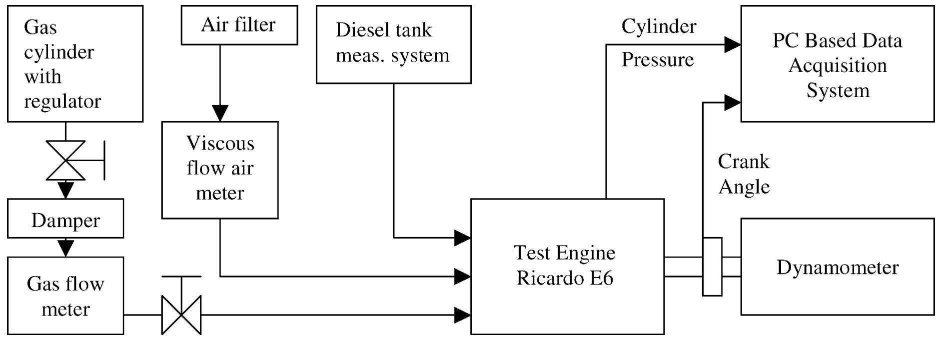

Fig. 1. Schematic diagram of the engine test rig.

(P_{max}), maximum pressure rise rate $(dP/d\theta)_{max}$ and mean value of all cycle pressure (which can represent the indicated mean effective pressure, imep) for each cycle. The program then writes these values in a separate worksheet and calculates the following for each for the number of cycles stored: the average value (average of P_{max}, average of $(dP/d\theta)_{max}$ and average of imep) the standard deviation (σ of P_{max}, $(dP/d\theta)_{max}$ and imep), and the coefficient of variation (COV of P_{max}, $(dP/d\theta)_{max}$ and imep) as follows:

Mean value of P_{max}:

$$\overline{P_{(max)}} = \frac{1}{n}\sum_{i=1}^{n} P_{(max)i}$$

Standard deviation of P_{max}:

$$\sigma_{P_{max}} = \sqrt{\frac{\sum_{i=1}^{n}(\overline{P_{(max)}} - P_{(max)i})^2}{n-1}}$$

Coefficient of variation for P_{max}:

$$\mathrm{COV}_{P_{max}} = \frac{\sigma_{P_{max}}}{\overline{P_{max}}}$$

where n is the number of cycle sampled. Similarly, these three quantities can be calculated for the maximum pressure rise rate, $(dP/d\theta)_{max}$, and imep.

Experiments have been carried out after running the engine for some time until it reaches steady state and oil temperature is at 60 °C ± 5, and cooling water temperature is at 70 °C ± 5.

Combustion pressure cyclic data (as average/standard deviation/COV of P_{max}, $(dP/d\theta)_{max}$ and imep) are presented for diesel and dual fuel (using LPG and methane as main fuel independently) against the tested engine operating design and operating parameters. The engine design and operating parameters have been varied at the following levels:

(1) Type of fuel included pure diesel fuel (base case as normal diesel engine), dual fuel of diesel + LPG, and dual fuel of diesel + methane.
(2) The engine load, and it is varied from 0.5 to 14 N m.
(3) The compression ratio, and it is varied at 20, 21, and 22.
(4) The pilot diesel fuel injection timing, and it is varied from 20 to 45° BTDC in steps of 5°.
(5) The pilot diesel fuel relative mass m_d (at constant mass of gaseous fuel admitted, m_g) from (m_d/m_d+m_g) of 18–42%.
(6) The engine speed, and it is varied at 1000, 1300, and 1600 rpm.

The experimental error is evaluated according to Ref. [12]. The maximum uncertainty in any quantity is calculated as the error divided by the average reading of the quantity. The maximum uncertainty in engine speed, torque, diesel proportion (m_d/m_d+m_g) are 2.5, 5 and 2.76%, respectively. The combustion pressure has been measured by the piezo-electric pressure transducer and then converted as pressure from the calibration data. It is then digitized by the A/D converter and fed into the computer for other calculations.

3. Results and discussion

3.1. Effect of fuel type

The effects of fuel type on combustion pressure data are illustrated in Fig. 2a–e. This is for the three fuels used: pure diesel fuel as single fuel case, dual fuel using LPG as main fuel, and dual fuel using methane as main fuel. These results are taken at common engine parameters of 1320 rpm, injection timing of 35° BTDC and compression ratio of 21. Fig. 2a shows the variation of average maximum pressure with the load for the three fuels. The load is found to increase the maximum pressure for the single fuel diesel engine as well as the dual fuel cases using either LPG or methane. However, it may be noticed that the maximum pressure is highest for LPG than that for methane then the pure diesel fuel gives the lowest value. This is confirmed by the maximum pressure histogram that is calculated over the number of consecutive cycled analyzed. It is clear that in most of the cycles LPG produces the highest maximum pressure followed by methane then diesel fuel.

The combustion noise—as represented by the maximum pressure rise rate—over the analyzed cycled for the three

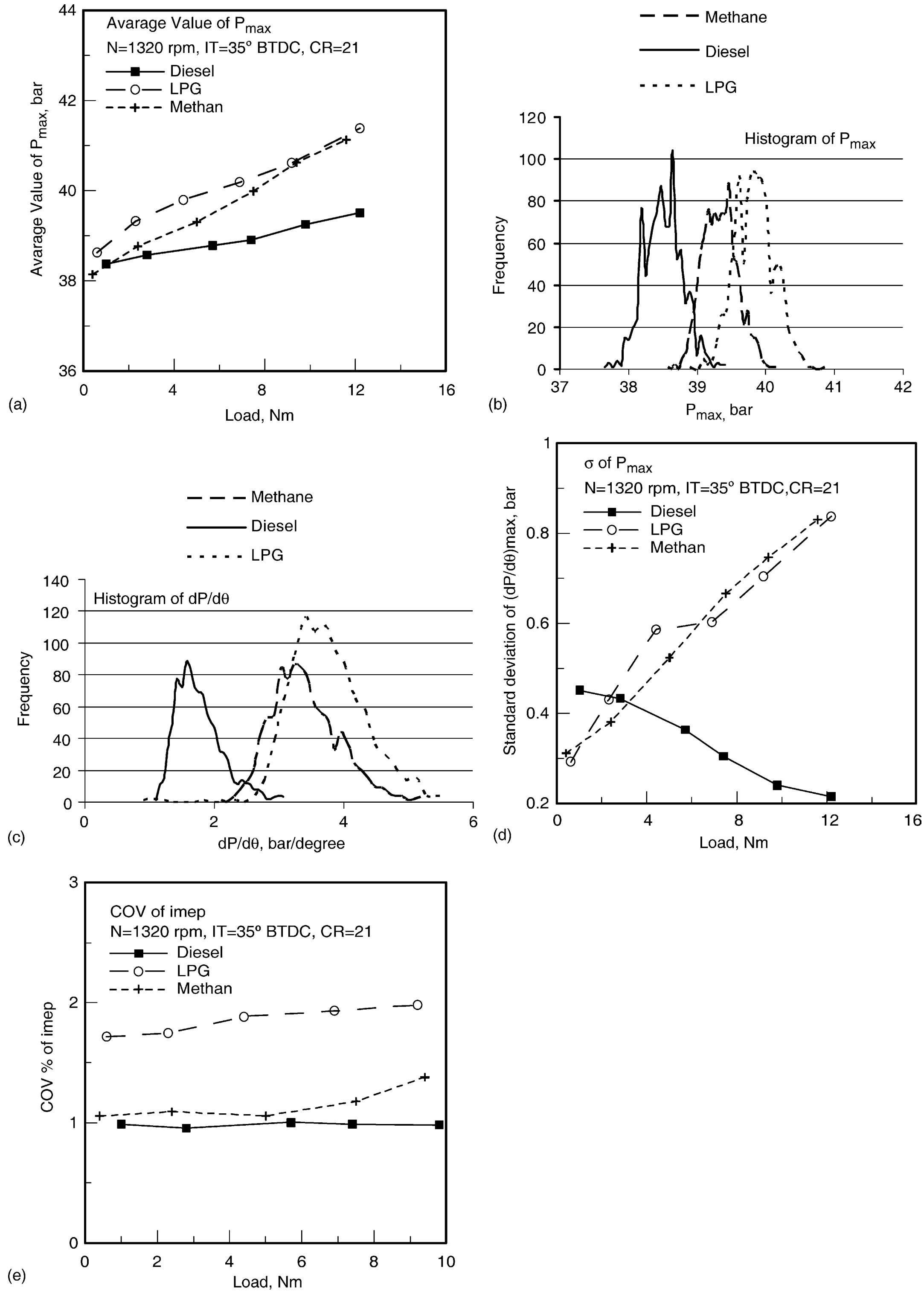

Fig. 2. Effect of fuel type on cyclic pressure data.

cases can be seen in Fig. 2c and d. Fig. 2c shows the histogram of maximum $(dP/d\theta)_{max}$ as occurred over the 987 cycles and it is evident that LPG and methane produced higher noise as compared to the single fuel case (pure diesel) for most of the cycles. It has been shown by [13] that the ignition delay period of propane–diesel–air mixture is longer than methane–diesel–air mixture and this is longer than the value for pure diesel–air mixture. When propane is used in dual fuel engine it exhibited longer delay than methane and diesel due to the differences in the associated changes in temperatures during compression, changes in preignition energy release, external heat transfer to the surroundings and the contribution of residual gases.

The standard deviation (σ) of $(dP/d\theta)_{max}$ is shown in Fig. 2d where it is shown that (σ) of $(dP/d\theta)_{max}$ for diesel engine running on diesel fuel is decreasing with load (or increasing the mass of fuel injected) while it increases with load for the gaseous fuels. This implies that for diesel engine, on increasing the load the $(dP/d\theta)_{max}$ becomes more repeated from cycle to cycle as the mass of fuel injected become bigger and the flame may become bigger and propagate smoother. While for dual fuel engine with gaseous fuels as main fuel, increasing the load means increasing the mass of gaseous fuel while keeping the mass of diesel pilot fuel constant. The increase in the mass of gaseous fuel caused the $(dP/d\theta)_{max}$ to vary from one cycle to another cycle. This is may be due to the variation in the gas–air mixture distribution especially at the start of ignition as indicated by [1]. It may be noticed also that (σ) of $(dP/d\theta)_{max}$ is following similar trend as $(dP/d\theta)_{max}$ for the individual cycles as well as the mean value, i.e. they are highest for gaseous fuels and lowest for diesel fuel. It was concluded [1] that if the (σ) of $(dP/d\theta)_{max}$ is decreased the average value of combustion noise can also be decreased.

Fig. 2e shows the COV of imep for the three cases where it is evident that the COV in imep for LPG is highest followed by methane and for diesel fuel is the lowest. It may be seen that all COV in imep are below 2% which means that the dual fuel engine will run stable with little fluctuation in output work, however it produces more noise. If the cycle-to-cycle variation of $(dP/d\theta)_{max}$ can be reduced the average value may also be reduced.

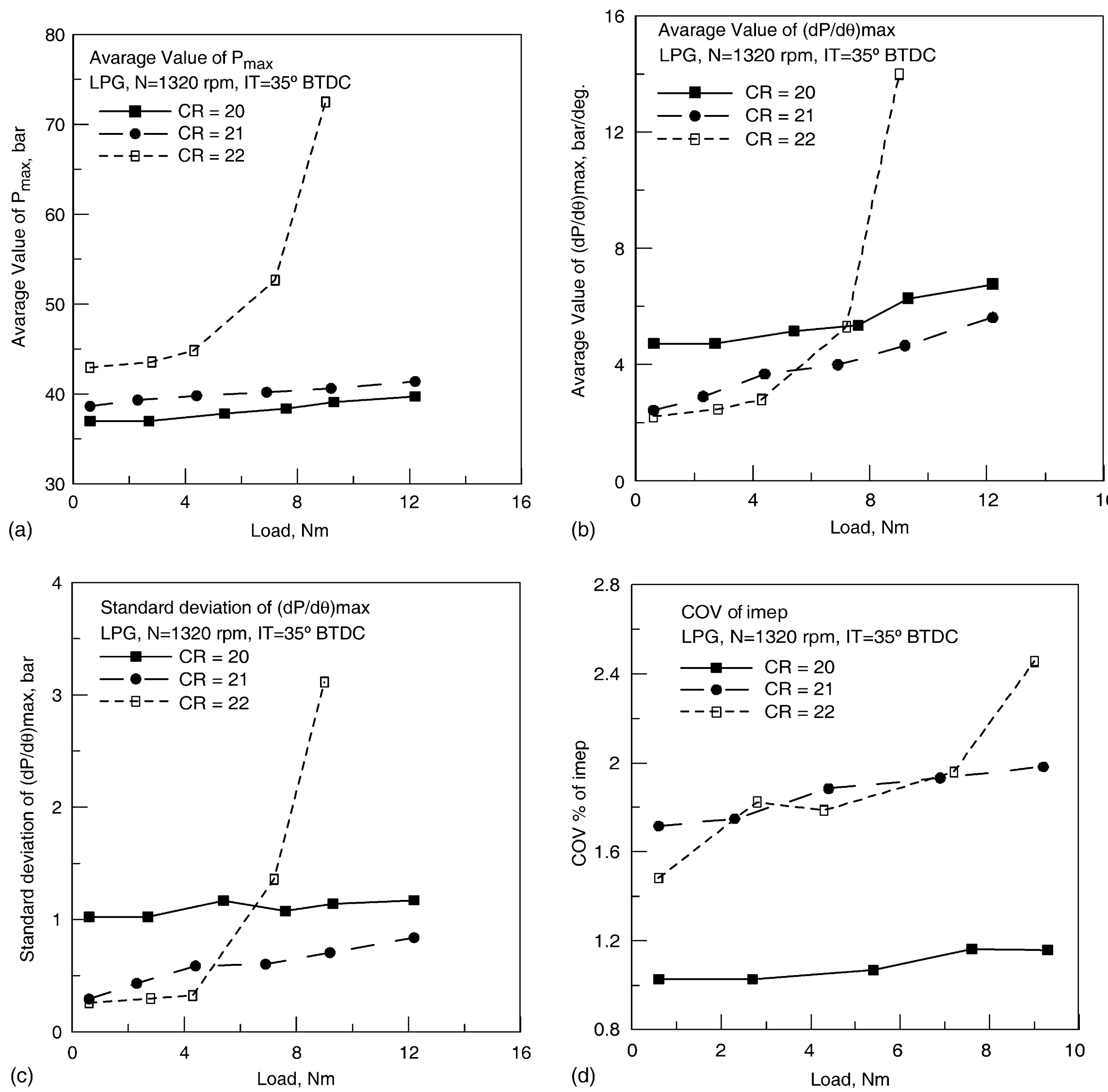

Fig. 3. Effect of compression ratio on cyclic pressure data.

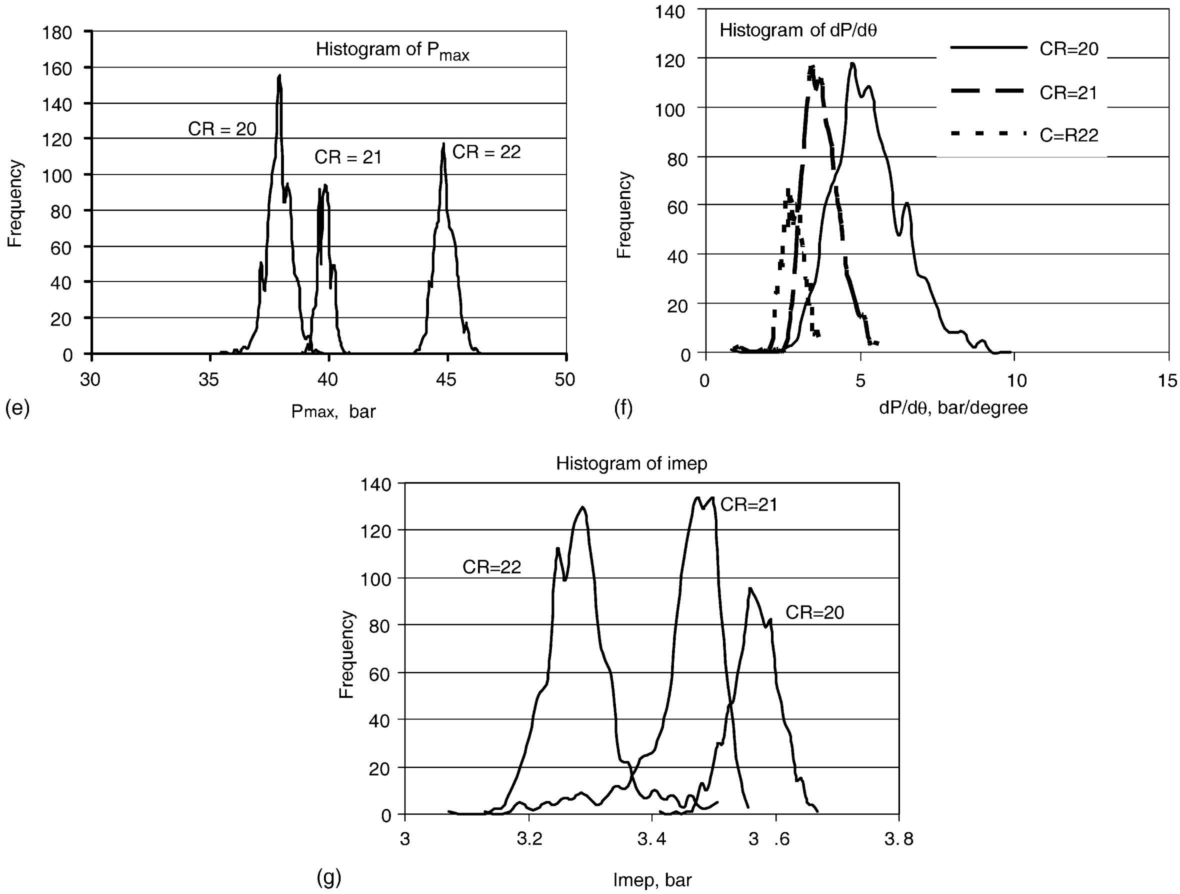

Fig. 3 (*continued*)

3.2. Effect of compression ratio

The effects of the engine compression ratio is shown in Fig. 3a–g. Fig. 3a shows the variation of the average value of maximum pressure at three compression ratios of 18, 20 and 22. It may be shown that increasing the compression ratio increased the value of maximum combustion pressure at all loads. This is also confirmed by the maximum pressure histogram shown in Fig. 3e. This increase in $(P)_{max}$ may be postulated to the increased compression pressure and temperature of the mixture which in turn causes the LPG gas to be more susceptible to self-ignite earlier in compression stroke and increase the maximum pressure of the cycle.

The maximum pressure rise rate for the three compression ratios is shown in Fig. 3b (average value of $(dP/d\theta)_{max}$), Fig. 3c (standard deviation of $(dP/d\theta)_{max}$) and Fig. 3f (histogram of $(dP/d\theta)_{max}$). At low loads, the combustion noise decreases generally with increasing the compression ratio (Fig. 3b and f). When the load was increased above 7.5 N m, the combustion noise is increased at the high compression ratio of 22. This is also shown in the standard deviation of $(dP/d\theta)_{max}$ (Fig. 3c). At the low loads, the mass of gaseous fuel admitted was low while this mass is increased at higher outputs. The increase in gaseous fuel mass seems to increase the charge temperature and pressure (especially at higher compression ratios) and cause earlier self ignition to the fuel air mixture (less ignition delay [13]). It is also clear from Fig. 3c that at high compression ratio and high loads, the cycle-to-cycle variation in the $(dP/d\theta)_{max}$ is increased which may be due to the cyclic variation in the charge composition at the start of ignition [1]. The high loads with high amount of gaseous fuel and high temperatures due to high compression ratio seem to have caused the propane to have less ignition delay and auto-ignites early in compression stroke and increase the pressure rise rate and noise [13].

The increase in compression ratio has also resulted in an increase in the cycle-to-cycle variation in imep as seen in Fig. 3d. However, the COV of imep is slightly higher than 2% over the range of loads examined which can be acceptable for the engine to produce a steady output power. Fig. 3g shows the histogram of imep where it may be noticed that the indicated mean effective pressure (imep) decreased as a result of increasing the maximum pressure rise rate and maximum pressure.

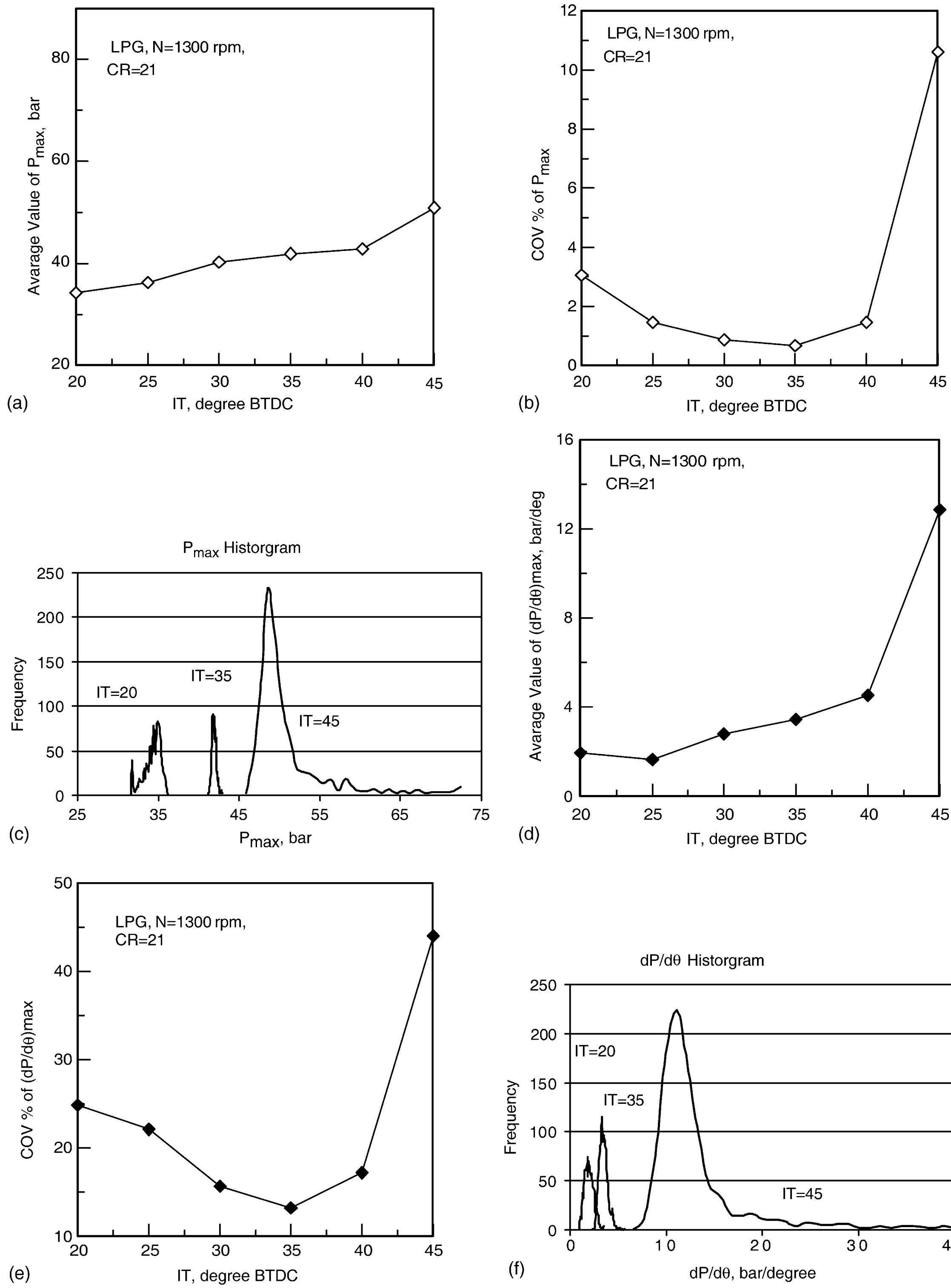

Fig. 4. Effect of pilot fuel injection timing on cyclic pressure data.

3.3. *Effect of pilot fuel injection timing*

Fig. 4a–i illustrates the effect of injection timing of pilot diesel fuel on the pressure data of the dual fuel engine running on LPG as main fuel at 1300 rpm, compression ratio of 21 and middle load.

The effect of injection timing on the average of $(P)_{max}$ is shown in Fig. 4a, its cyclic variation (COV) is shown in Fig. 4b while its histogram is shown in Fig. 4c. The advance in pilot fuel injection results in an increase in ignition delay period of the fuel which in turn will lead to the combustion of large diesel fuel mass or bigger flame to propagate at higher

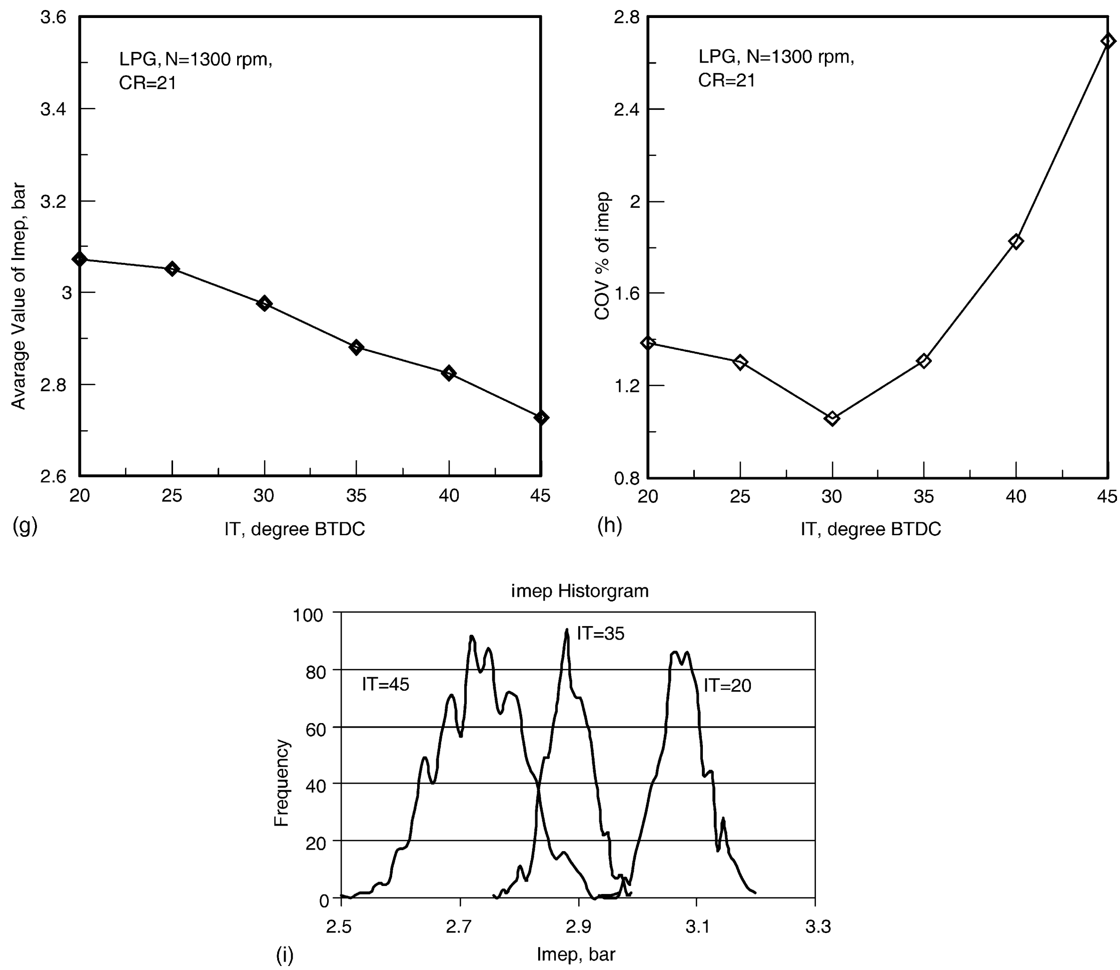

Fig. 4 (*continued*)

speed. This may have caused the combustion to start earlier and the maximum pressure to increase. The COV of $(P)_{max}$, can be seen in Fig. 4b to be much affected by the injection timing as it is minimum at injection timing of 30–35° BTDC. Earlier or later timing causes $(P)_{max}$ to exhibit large cycle-to-cycle variation. This may also be seen from the histogram in Fig. 4c. For injection timing of 35° the maximum pressure values are centered around 42 bar, while for 20°, $(P)_{max}$ is centered around 34 bar and it is slightly distributed over a range of 31–37 bar. For injection timing of 45°, however, it is more distributed over a wider range of 46 to 70 bar.

The average pressure rise rate $(dP/d\theta)_{max}$ (combustion noise) variation with injection timing can be seen in Fig. 4d. It may be seen that, generally, as the pilot diesel injection advance increases the combustion noise $(dP/d\theta)_{max}$ increases for the dual fuel engine. This may be attributed to the increase in ignition delay of the diesel fuel, since the liquid fuel is injected earlier in lower air pressure and temperature. The longer delay period would result in higher pressure rise rate $(dP/d\theta)_{max}$. This has been also shown by Abdalla et al. [14] as they presented the pressure rise rate in the pre-combustion chamber for 100% diesel and all diesel–methanol blends to increase as the injection advance increased.

With the presence in gaseous fuel in the mixture, any advance in pilot injection would result in longer ignition delay period and the pressure rise rate is expected to increase [15]. This is shown, generally, in Fig. 4a as the pressure rise rate $(dP/d\theta)_{max}$ for dual fuel engine is increasing as the injection advance increases, 20 to 45° BTDC.

The pressure rise rate, however, exhibits a large cyclic variability at very early or very late injection, Fig. 4e and f. The lowest COV of $(dP/d\theta)_{max}$ occurs at a middle range of about 30–35° BTDC where the combustion noise is also in the middle range. The high COV of $(dP/d\theta)_{max}$ may also be explained by its histogram in Fig. 4f. At 20° BTDC, the pressure rise rate is lowest and it is more distributed than that for 35° BTDC. For 45° injection timing $(dP/d\theta)_{max}$ is higher and at more distributed over a wider range of values from 7 to 40 bar/deg.

Fig. 4g illustrates the variation of average imep with the injection timing. It may seen that more advance in the pilot fuel injection timing caused a decrease in the imep as the maximum pressure and $(dP/d\theta)_{max}$ are increased and this caused maximum pressure to occur earlier in the compression stroke. This can reduce the output work or imep. Middle injection timing of about 30° BTDC caused lowest cycle-to-cycle variation in the imep as indicated by the COV of imep as seen in Fig. 4h and the histogram of imep in Fig. 4i. Early or late advance of pilot injection caused the imep to increase; however, the highest is around 2.7% which can be acceptable for steady output from the dual fuel engine.

3.4. Effect of pilot fuel mass

The effects of pilot fuel mass on the combustion pressure data may be found in Fig. 5a–f. The average of maximum pressure with its COV are shown in Fig. 5a and b. It may be seen that as the mass of pilot fuel increased

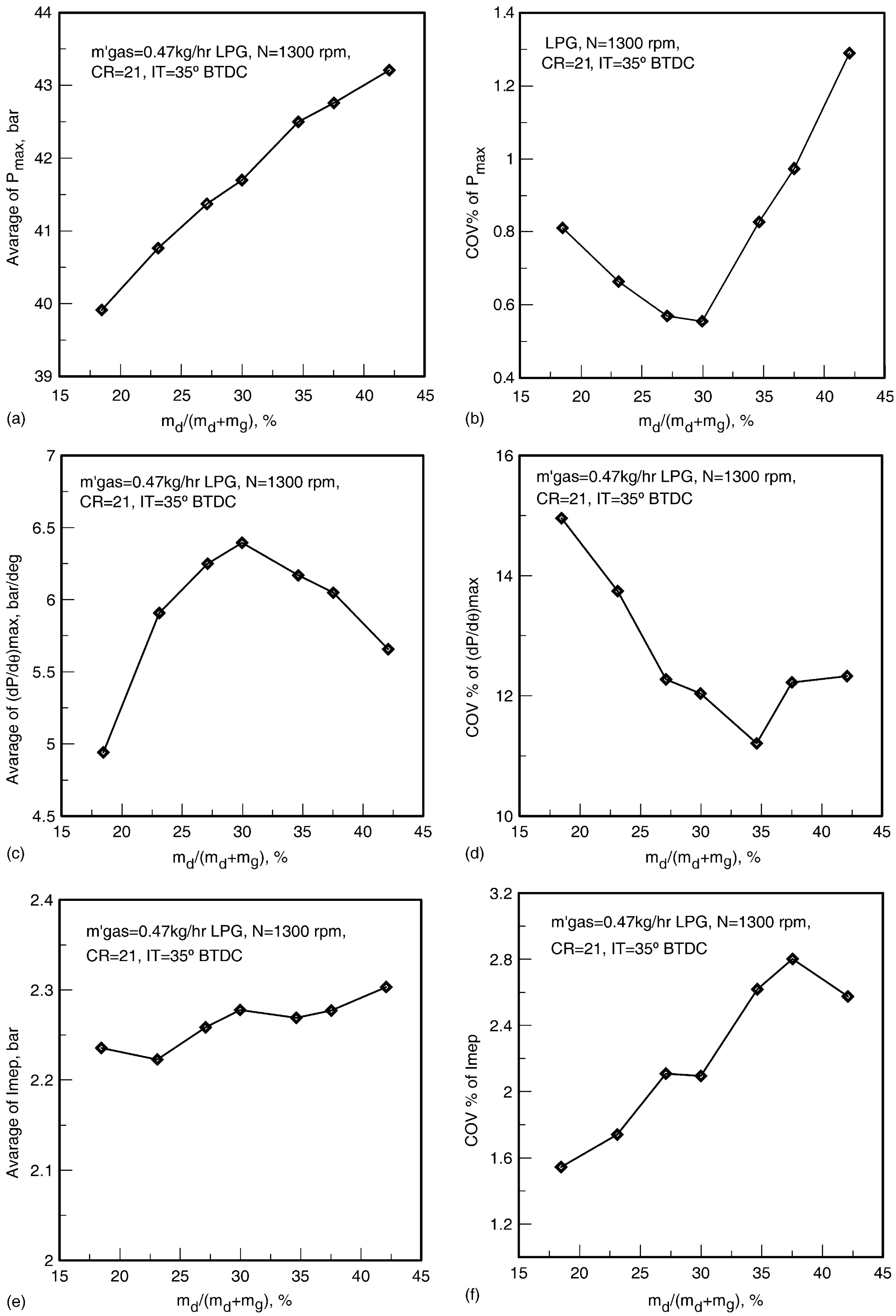

Fig. 5. Effect of pilot fuel mass on cyclic pressure data.

(or $m_d/(m_d+m_g)$) the average of (P_{max}), average $(dP/d\theta)_{max}$ (Fig. 5c), and average imep (Fig. 5e) increased. This is expected to be due to a number of contributing factors [16]. These include a greater energy release on ignition, correspondingly improved pilot injection characteristics, a larger size of pilot mixture envelope with a greater entrainment of the gaseous fuel, a larger number of ignition centers requiring shorter fame travels, higher rates of heat transfer to the unburned gaseous fuel–air mixture and an increased contribution of hot residual gases.

The combustion noise (average of $(dP/d\theta)_{max}$) first increased with increasing the mass of pilot fuel. The employment of a large pilot fuel quantity can lead to successful flame propagation and, consequently, leads to increasing the pressure rise rate which leads also to the knocking early [17]. This indicates that using a greater pilot fuel quantity to enhance the combustion process at low loads, will lead to increasing the tendency to knock at high loads. It has been shown also that the thermal efficiency for dual fuel engine is improved with using more pilot fuel. However, more increase of pilot fuel mass (higher than $m_d/(m_d+m_g)$ of 30% leads to a decrease in the pressure rise rate. Increasing the mass of pilot fuel in the mixture means using less mass of the gaseous fuel which in turn may lead a decrease in the in the ignition delay period of the gaseous fuel—less gaseous fuel in the mixture [12,17,18]. Also using more pilot fuel produces bigger and bigger initial flame which propagate easier with less pressure rise rate.

The cycle-to-cycle variation in $(P)_{max}$ is minimum at $m_d/(m_d+m_g)$ of 30%, Fig. 5b, however it is generally lower than 1.5% for all pilot fuel masses used. The COV of $(dP/d\theta)_{max}$ (Fig. 5d), however, exhibits high cycle-to-cycle variation in the pressure rise rate which tends to decrease with using more diesel pilot fuel (or less gaseous fuel). This is shown above that using more gaseous fuel to increase the load output tends to increase the cycle-to-cycle variation in the pressure rise rate (Fig. 2d). The COV of imep shown in Fig. 5f illustrates the slight increase in cycle-to-cycle variation in the indicated mean effective pressure with a maximum value of about 3%. The increase in diesel pilot fuel seems to increase the cycle-to-cycle variation in output work which may be postulated to the increased in fluctuation in spray formation, droplet size, mixture formation, and ignition delay of diesel fuel [1].

3.5. Effect of engine speed

The effect of engine speed on combustion noise (average of $(dP/d\theta)_{max}$) is depicted in Fig. 6a and its COV in Fig. 6b for dual fuel engine running at three different speeds of 1000, 1300 and 1600 rpm. Generally, as the engine speed increases, the pressure rise rate $(dP/d\theta)_{max}$ and its COV tend to decrease for most of the output loads. It has been concluded by [12] that increasing the engine speed has resulted in a decrease in the ignition delay period of the dual fuel mixture which results in a decrease in the pressure rise rate. It has also been shown by Imoto et al. [19] that the combustion noise $(dP/d\theta)$ decreased when the engine speed increases. The authors measured the pressure rise rate in the combustion chamber of IDI diesel engine and related it to the sound pressure level (SPL) in decibel (dB) and also measured the SPL in the intake and exhaust manifold. They have shown a reduction in $(dP/d\theta)$ as the engine speed increased. As the same time the SPL has increased in intake and exhaust manifold as engine speed increases. They have also shown a decrease in the heat release rate $(dQ/d\theta)$ with the increase in engine speed. The reduction in combustion noise was postulated to the reduction in the maximum rate of heat release and the COV in pressure rise rate may be caused by the reduction in heat release fluctuation.

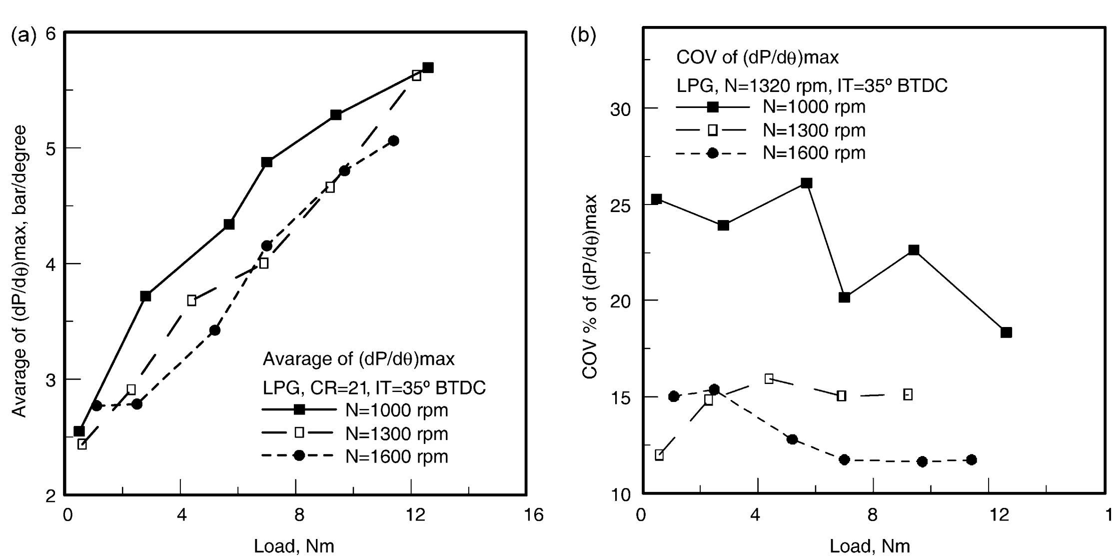

Fig. 6. Effect of engine speed on cyclic pressure data.

The effects of the above mentioned variables on thermal efficiency and exhaust emissions for the same engine may be also found in [17,20,21].

4. Conclusions

This work presents a statistical analysis of the cycle-to-cycle variation of measured values obtained from a single cylinder Ricardo E6 engine working on dual fuel of diesel and LPG and compared to diesel–methane and pure diesel fuel. From the study carried out the following conclusions may be drawn:

- The observed values of the combustion noise and their cyclic variability in dual fuel engine are strongly dependent on the type of gaseous fuels used and their concentrations in the cylinder charge.
- Dual fuel engine using LPG as main fuel exhibits higher combustion noise than that using methane.
- Increasing the dual fuel compression ratio while using LPG as main fuel leads to excessive combustion noise and cyclic variation at high loads in addition to loss in imep.
- Advancing the injection timing of the pilot diesel fuel for dual fuel engine using LPG as main fuel resulted in an increase in the combustion noise, cyclic variation, and loss in imep. Injection timing of about 30–35° BTDC resulted in the least cyclic variations and moderate combustion noise.
- Increasing the mass of pilot diesel fuel resulted in an increase in the imep, however, it increased the combustion noise.
- Increasing the engine speed resulted in a decrease in the combustion noise and its cyclic variation.

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