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Research Article

An Experimental Analysis of the Knock Response of Different Stoichiometric Mixtures of Gasoline-Natural Gas to Various Engine Speeds

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Abstract

Gasoline causes engine knock in higher compression ratios due to having lower spontaneous ignition temperature. Natural Gas (NG) has a higher octane number and is a proper fuel in terms of anti-knock properties; however, using it as the engine fuel results in a increases the emission of some exhaust gases due to lower burning velocity and gaseous nature and decline in the power of the engine. Using the mixture of gasoline and NG with gasoline as the predominant fuel can compensate the decline of the engine power and prevent the occurrence of engine knock. Bearing this in mind, in the present study, 4 different combinations of gasoline and NG consisting of 100%, 90%, 80%, and 70% gasoline and the rest NG (GA100, GA90, GA80, and GA70, respectively) were experimentally investigated using a single cylinder SI research engine at the equivalence ratio of 1.0, compression ratio of 11, and the engine speeds of 1500, 1800, and 2100 rpm.

Keywords : Dual fuel; Gasoline; Natural gas; Knock; Spark ignition engine.

1 Introduction

In the 21st century, the world faces numerous challenges in various areas. Some of the most important challenges are the insufficiency of easily-accessible and low-cost fuel resources, growing population, and growing demand for energy in all sectors of industry all over the world. Fossil fuels like coal, diesel fuel, and gasoline, which have caused environmental pollution, are rapidly decreasing. Therefore, they are not considered as permanent solutions to meet the energy needs of the world [1]. Furthermore, any shortage in this type of resources can bring about sharp and sudden changes in oil prices and pose a threat to the energy security and economy of the world [2].

The use of Natural Gas (NG) as an alternative fuel is one of the widely accepted solutions to cope with this problem. NG, whose main constituent is methane, has various advantages such as having

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lower levels of exhaust gas emission, better efficiency, desirable fuel economy, and higher availability. However, the use of this fuel in internal combustion engines is accompanied by a number of problems such as poor lean-burn capability and low flame speed [3]. Using Compressed Natural Gas (CNG) as engine fuel has many advantages some of which are lower tendency to knock, higher compression ratio, formation of homogeneous mixture, less emission of greenhouse gases, and more desirable fuel economy [4].

NG is a proper fuel regarding its octane number; however, it has lower density which causes a decline in the engine power. In addition, due to limited capacity of the NG tank, the need to refuel the vehicle is more frequent as compared to the case of gasoline. These problems are coupled with the inappropriate distribution of NG stations in the country which together have made it difficult to use this fuel.

Gasoline causes engine knock in higher compression ratios due to having lower spontaneous ignition temperature. That is why in SI engines which are fueled with gasoline, the compression ratio is limited to some specific extent. When NG is used in an engine which is designed to work with gasoline, its lower burning velocity leads to undesirable condition and negatively affects the engine power, engine temperature, spark advance, etc.

Nowadays, researchers and engine producers are mainly focused on increasing engine power and improving its efficiency. In recent years, due to limitations in fossil fuel resources and also the tightened regulations on exhaust gas emissions, researchers have shifted their attention towards developing more advanced combustion systems [5]. Therefore, power production with the use of the least possible amount of energy obtained from fossil fuel resources has become increasingly important.

One of the solutions to overcome such problems can be the use of the combined fuel of gasoline and NG. Gasoline has some special properties and advantages over NG in combustion; therefore, it can be used together with NG to improve the combustion characteristics of the air-fuel mixtures.

Raei Tabar et al. [6] investigated the use of gasoline-NG mixture in a 4-cylinder turbocharged engine and found that combining gasoline with 30% of NG at 3000 rpm improved knock resistance, reduced the levels of break specific fuel consumption (bSFC), CO emission, and HC emission by respectively 16%, 66%, and 50%, and increased NOx emission by 69% as compared to the case in which only gasoline is used.

In an experimental study, Bahattin Celik [7] investigated the use of the blend fuel of gasoline and ethanol in a high compression ratio using a small gasoline engine with low efficiency. First, he conducted some experiments on gasoline in the compression ratio of 6.1; then, he continued his experiments at the same compression ratio, engine speed, and pressure in E25 (25% ethanol and 75% gasoline), E50 (50% ethanol and 50% gasoline), E75 (75% ethanol and 25% gasoline), and E100 (100% ethanol) modes. Based on his findings, E50 was the most appropriate fuel in terms of engine performance and exhaust gas emission. He also investigated the performance of the engine in E0 mode at the compression ratio of 6.1 and in E50 mode at the compression ratio of 10.1 in full-load conditions and various engine speeds without any knock. The comparison of the experimental results indicated that using E50 as the engine fuel increased its power by 29% and decreased bSFC and the emission of CO_2, CO, HC, and NOx by approximately 3%, 10%, 53%, 12%, and 19%, respectively, as compared to E0 mode.

The use of dual fuel mixtures has also been the focus of attention in compression ignition (CI) engines. Taghizadeh & Rezaei [8] investigated the addition of 2%, 4%, 6%, 8%, 10%, and 12% ethanol to diesel fuel in a 6-cylinder CI hengine at the speeds of 1600 to 2000 rpm. They found that in the use of dual fuel with 6% ethanol and the rest diesel fuel, the engine torque and power increased by 3.8% on average. They further observed that with the increase of the fraction of ethanol to 8% in the dual fuel, the ignition delay increased, which itself caused an increase in the in-cylinder pressure and thereby increased knocking tendency.

Another alternative which has been considered by researchers is the addition of a liquid fuel like kerosene to gasoline. Obodeh et al. [9] studied the engine-out emissions of a 4-cylinder engine fueled with the blend of kerosene and gasoline with different proportions of kerosene ranging from 0-50% by volume in step of 10%. They found that as the fraction of kerosene increased in the blend, engine-out emissions also increased.

In some studies, the use of dual fuels has been combined with the use of two common fuel injection techniques, that is, port fuel injection (PFI) and direct injection (DI). Wang et al. [10] investigated the use of the dual fuels of M-G (PFI-Methanol and DI-Gasoline), E-G (PFI-Ethanol and DI-Gasoline), E85W15-G (PFI-15% water + 85% ethanol +DI-Gasoline), and G-G (PFI- Gasoline +DI-Gasoline) in a 4-cylinder engine at stoichiometric conditions. Their findings indicated that combining gasoline and alcohols in stoichiometric conditions effectively decreased the occurrence of knocking and considerably improved fuel consumption economy.

Due to the paramount importance of the optimal blend of ethanol and gasoline, Tiwari et al. [11] evaluated the performance of a single cylinder SI engine powered by ethanol-gasoline dual fuel mixtures with the ratio of 10% (E10), 20% (E20), and 30% (E30) of ethanol and the rest gasoline. They found that the brake thermal efficiency increased for a specific blending percentage of alcohol and that this percentage was different for different alcohols. After this particular blending percentage, brake thermal efficiency decreased. They reported that in general, blending ethanol with gasoline improved combustion properties of the fuel. They also observed that increasing the alcohol percentage did not affect combustion after a particular level and lowered thermal efficiency. Based on their results, the performance of the engine was found to be optimal in E10 mode.

In an experimental study, Behrad et al. [12] investigated the knocking phenomenon in a single cylinder spark ignition engine with a high compression ratio (CR=11) at stoichiometric conditions using dual fuel combinations of GA100 (100% gasoline), GA90 (90% gasoline and 10% NG), GA80 (80% gasoline and 20% NG), and GA70 (70% gasoline and 30% NG). They found that at the optimum spark advance (OSA) of GA100, the knocking cycle percentage (KC%) was 5.3% and the distance between optimum spark advance and the impending knock limit advance ($\Delta\theta$) was less than 1°CA(Crank Angle). They also observed that at the related OSAs, with the increase in the fraction of NG in the dual fuel, the knocking cycle percentage decreased significantly and reached 0.5% in GA80; on the other hand, $\Delta\theta$ increased with the increase of natural gas fraction and reached 4.5°CA in GA70.

Sarabi & Abdi-Aghdam [13] evaluated four different gasoline and natural gas combinations including 100%, 90%, 75% and 60% gasoline and the rest natural gas (GA100, GA90, GA75 and GA60, respectively) in a single cylinder spark ignition engine at specific engine speeds, the CR=10, and the stoichiometric conditions. According to their findings, the increase of NG fraction up to G75 resulted in no significant changes in engine torque and power while the emission of all exhaust gases decreased.

Yekani et al. [14] investigated the performance response of a SI engine to the addition of NG to gasoline in different combinations including 100%, 87.5%, 75%, and 62.5% gasoline and the rest NG in lean-burn condition ($\phi = 0.9$) at the compression ratio of 11. Their results demonstrated that as the natural gas fraction in the dual fuel increased, CO emission decreased while NO_x emission increased. They found that the engine had a better performance in G75 as compared to G100 and other combinations.

The use of gasoline and natural gas as single fuels for engines has some advantages and disadvantages. It seems that using a blend of these two fuels can decrease the engine tendency to knock and make it possible to increase the compression ratio(CR) with the predominant presence of gasoline in the mixture. In our previous study [12], the engine knock was investigated at a fixed engine speed using the same dual fuel combinations as in this study. However, due to the crucial effects of engine speed on the combustion characteristics, we believed that it would be valuable to investigate the response of the different dual fuel combinations to the variations in engine speed. Therefore, in the current experimental study, four different combinations of gasoline and natural gas dual fuel were investigated in a single cylinder spark ignition research engine at various engine speeds, the stoichiometric equivalence ratio, and a given compression ratio and their knocking characteristics at different engine speeds were compared.

2 Knocking

Several factors can affect the occurrence of engine knock. Some of these factors are compression ratio, spark advance, equivalence ratio, type of fuel, intake charge mass, the temperature of the inlet mixture, wall temperature, the shape of the combustion chamber, and spark position [15]. Knocking is generated as the result of the spontaneous ignition of a part of the unburned end-gas ahead of spark ignited flame front and is accompanied by a reasonable oscillating pressure wave [16]. The in-cylinder pressure oscillations and the speed of sound are identifiable in engine knocks. The pressure waves generated from engine knocks are usually in the frequency range of 4-20 kHz [17, 18].

Several methods have been proposed in the literature to detect engine knock. Some of the important ones are the analysis of the in-cylinder pressure oscillations [19, 20], radical species [21], engine block variations [22], emitted exhaust gases [23], and the released heat [24]. Bearing in mind that knocking directly affects in-cylinder pressure, in the current study, the variations of in cylinder pressure were monitored to detect the occurrence of engine knock. The analysis of in-cylinder pressure is one of the most common and fundamental methods used to detect the occurrence of engine knock and determine its intensity [25]. In this method, the in-cylinder pressure of the knocking cycle is first measured at an appropriate sampling rate. Then, using Fast Fourier Transform (FFT) algorithm, the time-domain data are converted to frequency-domain data. After that, the knocking and non-knocking signals are separated using a band-pass filter. The separated signals are converted to time-domain using Invers Fast Fourier Transform (IFFT). Finally, the pressure signal emitted by the engine knock is rectified and its MAPO is determined. The obtained MAPO value is taken as the indicator of engine knock.

Considering the above descriptions, MAPO is defined as follows:

$$MAPO_i = \max\left(|\bar{P}_i|_{q_0}^{q_0+w}\right). \tag{2.1}$$

where i represents the ith cycle, $\bar{P}_i$ is the pressure caused by knock in a cycle, q_0 is the ignition point, and W is the time period during which knocking is monitored. When the value of MAPO$_i$ in a cycle exceeds 1 bar, that cycle is considered a knocking cycle [26].

If a series of cycles are analyzed in an experiment, then the mean MAPO of those cycles and that of the knocking cycles can be expressed as the following:

$$\overline{MAPO}_{Tot} = \frac{1}{N_T}\sum_{i=1}^{N_T} MAPO_i \tag{2.2a}$$

$$\overline{MAPO}_{KC} = \frac{1}{N_{CK}}\sum_{i=1}^{N_{KC}} MAPO_i \tag{2.2b}$$

where N_T is the total number of cycles and N_{KC} is the number of knocking cycles in the experiment. Moreover, the subscripts of Tot and KC represent the mean $MAPO$ of total cycles and knocking cycles, respectively. Their standard deviation (σ) is used to evaluate the distribution of the $MAPO$ of different cycles and can be expressed as in Eq. 2.3 below.

$$\sigma = \left(\frac{1}{N_T}\sum\left(MAPO_i - \overline{MAPO}_{Tot}\right)^2\right)^{\frac{1}{2}} \tag{2.3}$$

3 Experimental equipments and methodology

In this study, a CT300 test stand produced by the German company of Gunt and a single cylinder spark ignition engine with adjustable speed coupled with an asynchronous dynamometer were employed. The compression ratio (CR) of the engine employed in this study was also adjustable. The characteristics of this research engine are represented in Table 1. Also, in Figure 1, schematic representation of the experimental setting, control set, and engine evaluation systems is provided.

The output torque of the engine was measured using a lever arm equipped with a strain gauge. Since the asynchronous machine is coupled with the engine, the speed of the engine is controlled by adjusting the speed of the machine. In the current study, a Kistler shaft encoder, type 2613B was used to identify TDC and crank angle. The use of this tool made it possible to adjust spark advance and the time of the injection of each fuel with 1°CA. This shaft encoder was attached to the free end of the crankshaft. A 4-channel data logger (DAQ-2005, ADLINK) with the maximum data acquisition frequency of 500kHz and the voltage domain of ±10V was used to record analog data. The 4 channels of the data logger were respectively used for recording in-cylinder pressure, intake manifold absolute pressure, TDC pulses, and the crank angle of the shaft encoder.

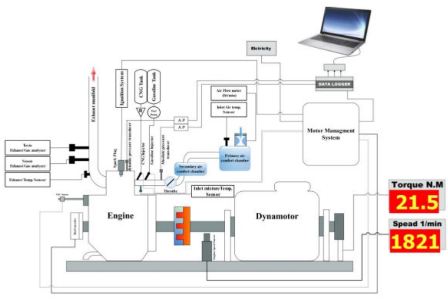

Figure 1: Schematic representation of the equipment used in the experiment [12]

The inlet air temperature was measured with a PT100 thermocouple and the exhaust gas temperature was measured using a NiCr-Ni thermocouple with the temperature range of 0-900C. In order to analyze exhaust gases, Saxon Infralyt CL gas analyzer was used. This analyzer could indicate the concentration of CO, CO_2, HC, O_2 (in the form of volume/volume percentages or ppm), and λ. Using the λ obtained from the analyzer, the compression ratio was adjusted to the desired level via changing the amount of the fuel injection.

Since the experiment was designed to be conducted in various engine speeds, the compression ratio of the engine was fixed at the value of 11 (CR=11). The inlet flow throttle valve was widely opened to make the engine operate at full load. After adjusting the water flow rate to 80 lit/min (as proposed by the test stand manufacturer), the engine was started in motoring mode. By activating the management system, the engine was run using gasoline as the fuel. After warming up (when the oil temperature reached 65°C, pressure attained 5 bars, and the outlet water temperature was around 60-80°C), the engine got to a steady state and the situation became ready for conducting the experiments. Based on the previous studies conducted on the simultaneous injection of two fuels [12], the experiments were conducted at the engine speeds of 1500, 1800, and 2100rpm using different combinations of gasoline-natural gas, that is, mixtures including 100%, 90%, 80% and 70% gasoline and the rest natural gas (labelled as GA100, GA90, GA80 and GA70, respectively) at the equivalence ratio of 1.0 (ER=1.0) and various spark advances. The data were acquired at the sampling rate of 120 kHz and were recorded using the AD-LOGGER software. The acquired raw data were analyzed by a program written with FORTRAN programming language and the Indicated Mean Effective Pressure (IMEP), peak pressure value, and its angle were determined, the transformation of the pressure data via FFT and IFFT was conducted, the knocking signal in each cycle was extracted, and other required analyses were performed.

4 Results and Discussion

In the middle mode of the bi-fuel combinations (GA80), a series of experiments were conducted in various spark advances. As expected, more knocking cycles were observed in higher spark advances. In order to clarify the knocking phenomenon, cyclic variations, and the filtering process of knocking signals, one knocking cycle and

Table 1: Characteristics of the research engine [11].

Parameter	Value
Cylinder bore/mm	90
Piston stroke/mm	74
displaced volume/cm3	470
Compression ratio	11
Ignition system	Electrical with 1 _CA adjustable ignition timing
Fueling system	GasolineNG adjustable injector
Lubrication system	Pressure-Splash
Cooling system	Water (single flow)
Number of valves and their arrangement	2 OHV

one non-knocking cycle were selected from among the cycles recorded in the spark advance of 25°CA. Figure 2 represents the $P-\theta$ diagram of these two cycles in GA80.

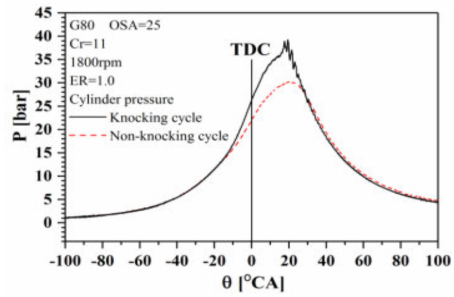

Figure 2: The $P-\theta$ diagram of the representative knocking and non-knocking cycles in the spark advance of 25°CAD in GA80 mode.

As can be observed, the pressure oscillations as the result of combustion in the knocking cycle were considerable such that the peak pressure value reached 39.2 bars while that of the non-knocking cycle was found to be about 30.25 bars. In the current study, in order to distinguish the pressure oscillations resulting from engine knocks, whose frequencies are usually in the range of 4-20 kHz, FFT filter was used. In so doing, first the P-t data of the two cycles, which were in the time domain, were converted to the frequency domain via the use of FFT. The pressure oscillations of these two cycles in various frequencies are depicted in Figure 3.

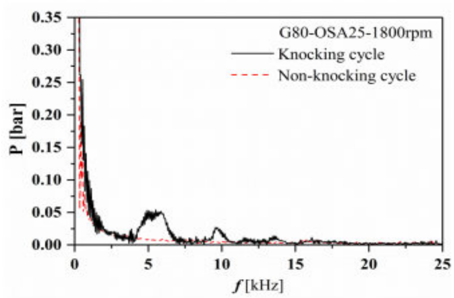

Figure 3: In-cylinder pressure oscillations during combustion converted to frequency domain using FFT.

It can be seen that in the knocking cycle, waves with noticeable amplitudes appear in the domain of 4-20 kHz while in the non-knocking cycle, there are no such indications. After filtering the knocking signals, the related special process was followed to distinguish knocking and non-knocking signals. Then, the data were converted to time domain using Invers FFT (IFFT). Figure4 shows the filtered $P-\theta$ diagram of these two cycles together with the variations resulting from engine knock. Based on the pressure oscillations arising from the knocking cycle, the MAPO was found to be 1.69 bars. It can also be observed that no pressure oscillation were generated as the result of the filtering process in the non-knoking cycle.

The filtering process was conducted on each of the 400 successive cycles of each experiment and the MAPO for each of them was determined. As mentioned earlier in this paper, cycles with the MAPO value of higher than 1 bar are considered as knocking cycles [18].

Figure 5 shows the variations of $\overline{MAPO}_{Tot}$ and σ at various spark advances and the engine speed of 1800rpm in GA80 mode. As can be seen, both of them increase with the increase of the spark advance. The increase of the spark advance causes

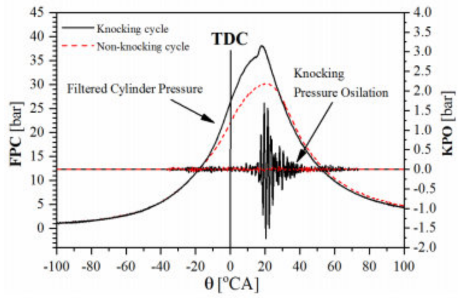

Figure 4: Pressure oscillations and knocking variations at various crank angle degrees in the two representative knocking and non-knocking cycles in GA80 mode.

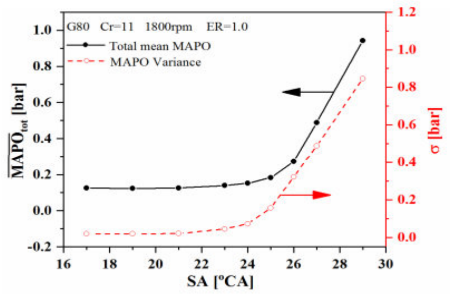

Figure 5: Variance and mean in GA80 mode.

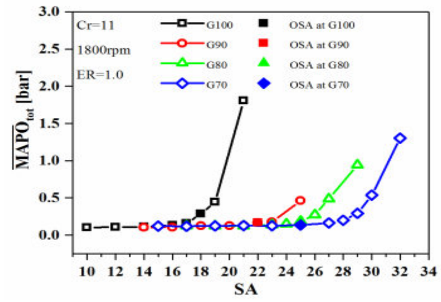

Figure 6: Mean $\overline{MAPO}_{Tot}$ in various dual furel mixtures

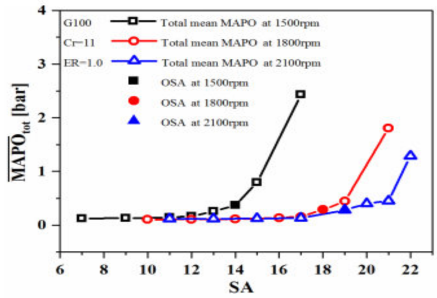

Figure 7: $\overline{MAPO}_{Tot}$ versus spark advance at three engine speeds

an increase in peak pressure value as well as in the temperature of the unburned mixture and thereby paves the way for the occurrence of engine knock. In the GA80 mode, KC% at each SA was calculated from the obtained results.

Figure 6 represents the variations of $\overline{MAPO}_{Tot}$ at various spark advances in GA100, GA90, GA80, and GA70 modes at the engine speed of 1800rpm. It can be observed that as the fraction of NG in the mixture increased, the OSA increased too. Furthermore, the value of $\overline{MAPO}_{Tot}$ at the OSA of the GA100 mode was found to be higher than that of the other combinations.

Considering the importance of the knocking phenomenon in this study and its serious occurrence in the GA100 mode, the $\overline{MAPO}_{Tot}$ values in terms of spark advance for the three engine speeds of 1500, 1800 and 2100rpm in this mode are illustrated in Figure 7. To highlight the $\overline{MAPO}_{Tot}$ value at the OSA, their statuses are represented using filled in symbols at the three engine speeds.

In order to detect knocking conditions, usually 10% is used as the knocking onset [27]. For each combination, KC% was obtained at each spark advance through the analysis of the data. Figure 8 shows the variations of KC% at various spark advances for the GA80 mode together with its knocking condition limit (KCL). From the intersection of the diagram with the knocking condition limit (KCL) line, the knocking condition onset and its distance from the optimum spark advance ($\Delta\theta$) can be determined. As this distance increases, the working condition of the engine at the OSA gets farther from knocking condition. Such a condition provides a stable state for the engine to work without any problem. With an appropriate value for Δ ∴, an engine does not enter the knocking limit as the result of minor variations in its working condition.

Figure 9 indicates the KC% at various spark advances for each of the four gasoline-NG mixtures. Considering the OSAs in various combinations and the knocking onset of 10%, it can be observed that the $\Delta\theta$ is the lowest in GA100 and the highest in GA70.

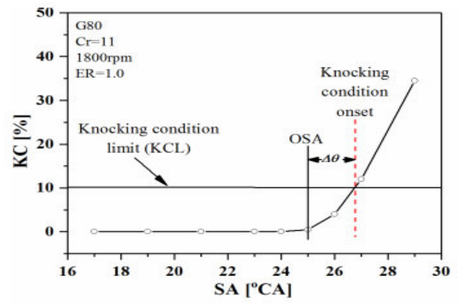

Figure 8: KC% versus spark advance and the knocking condition onset in GA80 mode

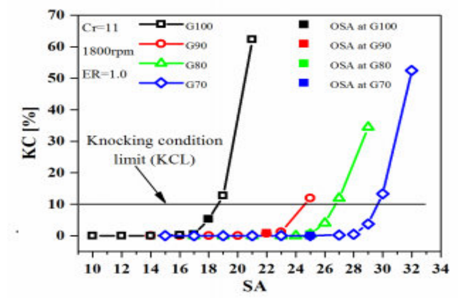

Figure 9: KC% versus spark advance and the knocking condition onset in various gasoline-NG combinations

5 Conclusion

In the current experimental study, four different stoichiometric mixtures of gasoline and natural gas were experimentally investigated in a single cylinder research spark ignition engine at CR=11 and the engine speeds of 1500, 1800, and 2100rpm. The data obtained from 400 successive cycles in each experiment were analyzed. Then, the knocking performance of the four dual fuel mixtures were investigated at different engine speeds and the following results were obtained:

In GA100 mode, at the engine speed of 1800rpm and the OSA, $\overline{MAPO}_{Tot}$ and knocking cycle percentage (KC%) were found to be 0.289 bar and 5.25%, respectively. In addition, the distance between OSA and the impending knock limit advance ($\Delta\theta$) was observed to be less than 1°CA.

As the fraction of NG in the mixture increased, $\overline{MAPO}_{Tot}$ and knocking cycle percentage (KC%) significantly decreased at the OSA such that their value reduced respectively to 0.184 bar and 0.5% at the OSA of GA80; $\Delta\theta$, on the other hand, increased to 1.77°CA in this mode.

With the increase in the engine speed, $\overline{MAPO}_{Tot}$ increased in a way that its value in the OSA of GA100 at the engine speeds of 1500, 1800, and 2100 rpm reached 0.371, 0.289, and 0.277 bar, respectively.

As the fraction of natural gas in the mixture increased, $\Delta\theta$ also increased; its value in GA80 and at the engine speeds of 1500, 1800, and 2100 rpm reached 0.74, 1.78, and 3.31 CA, respectively.

As the fraction of gasoline in the mixture decreased, the values of $\overline{MAPO}_{Tot}$ and σ at the OSAs decreased.

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