THE PERFORMANCE OF A GAS ENGINE FUELED WITH CARBON DIOXIDE-NATURAL GAS (CO_2-NG) MIXTURES

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Abstract

Large percentage of natural gas fields in Malaysia have sour gas consisting of high percentage of CO_2 and possibly hydrogen sulfide (H_2S), thus rendering it uneconomically viable. Despite this, the little heat energy therein can be harnessed for energy sustainability. This paper, therefore, investigates the effect of CO_2 variation in a mixture of NG and CO_2 as a way of imitating the natural gas fields of different CO_2 composition, on the performance of a gas engine. The baseline fuel and different CO_2 percentages (10%, 20%, 30%, and 40%) in the mixture of NG-CO_2 were used to conduct tests on a single-cylinder spark ignition (SI) direct injection (DI) engine. The engine was run at 180°CA BTDC, stoichiometric, and at various engine speeds. At wide open throttle, the ignition timing was regulated to obtain the maximum brake torque (MBT). The results of the tests revealed that brake torque (BT), brake power (BP), and brake specific fuel consumption (BSFC) decreased as the CO_2 percentage was increased. While brake thermal efficiency (BTE) increased with an increase in CO_2 percentage. And for all the CO_2 percentages in the mixture, BT, BSFC, and BTE increased as the engine speed was increased up to peak value occurring at 3000 rpm and then decreased. While a direct relationship exists between brake power and the engine speeds. These outcomes suggest that at 3000 rpm, the NG-CO_2 mixture can be utilized up to 20% CO_2 percentage to achieve optimal engine performance. This finding, if applied, will help in energy sustainability development.

Keywords: *Natural gas; high carbon dioxide; direct injection; emissions; engine performance.*

Introduction

As the population of the world is fast increasing, our energy consumption continues to increase. Fossil fuel is the major source of energy for internal combustion engines and other applications. However, its depletion owing to its consumption by the increasing number of vehicles on roads, its resultant pollutant emissions threatens the survival of our ecosystem, and its increasing price have become a great concern. Therefore, the quest for alternative fuels to reduce dependency on fossil fuel, reduce toxic emissions, and improve engine performance becomes imperative [1]. Moreover, in order to meet the stringent emission legislation imposed from time to time and the future energy demands, alternative fuels are therefore sought and currently applied [24] to prevent the resultant negative effects of environmental pollutants produced by different sectors.

Many researcher studies have been conducted, on a wide range of alternative fuels ranging from liquid to gaseous chemicals, so as to ascertain suitable alternatives adaptable to spark-ignition (SI) and compression-ignition (CI) applications. Studies have shown that the most common alternative fuels for SI engines are acetylene and ethanol blended in gasoline, liquefied natural gas (LNG), compressed natural gas (CNG), hydrogen, and all proved to increase engine performance and lower vehicle emissions under suitable engine operating conditions [4]. Of the numerous research studies conducted on the effects of alternative fuels on SI engines, the work [5] studied the effects of acetylene-gasoline mixtures on the performance of a SI at stoichiometric conditions and different loads. The results of the experimental study show that gasoline consumption at constant output power decreased. While the HC and NOx emissions significantly reduced at all loads.

Alternative fuels such as biodiesel, biofuels [6, 7] operate in compression ignition engines just as diesel fuel. Studies have shown that similar performance and exhaust emissions are achieved when compared to petroleum diesel, most especially when it is blended with water or ethanol. A research study was conducted on a CI engine fueled with acetylene-ethanol mixture having 4.4 kW as the rated output power. The engine was adapted to use dual fuel at different gas flow rates. The outcome of the study shows reductions in the emissions of NOx, HC, CO, and CO_2, coupled with a gain in the diesel fuel consumption [8].

Given the availability and abundance, natural gas, therefore, qualifies to be one of the most leading alternative fuels, besides its adaptability to be used in SI and DI engines. Records have it that Malaysia is ranked 24th, considering the abundance of her natural gas reserves to the size of 2.4 trillion cubic meters [9]. However, of these reserves, 37 Tcf are undeveloped because it contains accumulated sour gases having significant proportion of CO_2 and possibly hydrogen sulfide, thus rendering it uneconomically viable [10].

However invaluable it may be, these reserves still have therein potential heat energy that can be harnessed to meet our energy need, particularly in the transportation section rather than abandoning it to lay fallow. Going by this, high carbon dioxide natural gas (CO_2-NG) is therefore employed to replicate or simulate the composition of the natural gas reservoirs/fields and used to conduct this research study. Biogas being the closest fuel that can be compared to the target fuel (high CO_2-NG) in terms of composition, is mostly quoted for the research study. Most previous research works conducted [11-14] have used simulated biogas (methane and carbon dioxide) of varying CO_2 proportions ranging from 0% to 40% in a diesel engine so as to study the impact of CO_2 variations in the mixture on engine performance. The tests were conducted at different air-fuel ratio, speeds, and compression ratios using a variable compression ratio Ricardo E6 research engine.

In another report, a SI adapted from a CI engine was used to examine the consequences of the methane concentration enhancement on the performance and emission of the engine [14]. Experimental results prove that CO_2 presence in the mixture improved NOx emissions with penalties of low cylinder pressures, engine power, and thermal efficiency as well as high unburnt HC. Also, at a lean mixture where the methane concentration in the mixture was increased, the engine performance increased with a reduction in HC emission. Brake thermal efficiency together with the power output considerably enhanced as the CO_2 percentage in the mixture was decreased. The amount of air intake and the increase in the methane concentration are the reasons for the improvement in the engine performance owing to faster burning velocity and higher temperatures. However, lower HC emission recorded at lean mixture was due to the decrease in the CO_2 percentage in the mixed fuel, which thus resulted to complete combustion.

In a similar study, [12] was reported to have examined the effect of carbon dioxide variation on engine performance using different mixtures of natural gas and CO_2. The outcome of the tests revealed that a high combustion flame temperature up to 1723°C of diesel was achieved and sufficiently high to instigate dissociation of CO_2 into CO and O_2. It was also noted that with 30% addition of CO_2 proportion, the engine performance was slightly improved when compared to the same running on natural gas, while the brake specific fuel consumption (BSFC) and the diesel flow rates reduced. The reason being that the dissociated CO caused the burning rate of the mixed fuel to accelerate, while the increase in O_2 content of the air intake improved the combustion of uHC and reduced ignition delay. But with a higher CO_2 proportion of more than 30%, the CO_2 became undissociated and behaved as an inert gas in the mixed fuel.

Because of the low engine output performance occasioned by the high presence of CO_2 in a NG-CO_2 mixture, several research studies have suggested effective means by which engine performance could improve. Optimization of operating conditions [15, 16], increasing of compression ratio [6], and advancing spark/ignition timing [15] are the suggestions offered to improve engine performance. Increasing compression ratio was reported to improve engine performance fueled with a high proportion of CO_2 in NG-CO_2 mixture though with penalties of increase NOx, CO_2, and HC emissions. But the NOx emissions are repressed with the presence of CO_2 in the fuel. According to [15] the low engine performance recorded due to the increase in diluent proportion in the fuel was improved by advancing the spark/ignition timing, particularly if the CO_2 is the diluent. Injection parameters (such as injection timing, pressure, and angle) are found to greatly improve engine output performance and emissions, most especially with a DI system [17]. Of the above parameters, an engine volumetric efficiency and output performance were reported to improve greatly with injection timing [18]. The outcome of a similar study of [19] established that with a precise injection timing, volumetric efficiency and brake power improved.

The impacts of natural gas blended with different percentages of nitrogen gas were examined on the combustion and emissions characteristics of a CI DI engine. A singlecylinder CI engine was modified and adapted to operate on DI of a natural gas fueling system. 20% and 40% of N_2 (by volume) mixed in natural gas were used to operate the engine at 1200 rpm. The timing for the start of injection of gaseous fuel was set at

1.0 ms after the end of the diesel injection. Different combustion timings of 0, 5, 10, 15° ATDC were used from the start of the expansion stroke – TDC to 15° ATDC. The experimental outcomes revealed that the initial combustion intensity and the maximum heat release rate (HRR) reduced because of the lower chemical energy that resulted after ignition in the partially premixed charge.

At all the injection timings and with 20% N_2 dilution, the NOx emissions slightly increased but reduced with a 40% N_2 addition. HC emissions and N_2 dilution were observed to be inversely related as a N_2 addition resulted to a decrease in HC emissions. Besides, a given mass of fuel was noted to produce a constant amount of the emissions reduction. At most combustion timings, CO was equally noticed to reduce significantly with the N_2 addition to the mixture while the total particulate matter (PM) reduced significantly.

Overall, the previous studies presented so far mainly focused on CI engines with port injection system using biogas and imitated biogas. While some studies reported to use carbureted engines. Besides, some of the engines employed were modified CI engines and adapted to operate a DI system, although with pilot fuel. Moreover, the different variables examined in the past literature were conducted using engines fueled with premixed mixtures. However, in this study, the two gaseous fuels used to conduct this test employed a DI system where the fuels were introduced simultaneously into the engine with no premix. With this, a study on the performance of a gas engine with various mixtures of CO_2-NG was carried out to gain understanding of the effects of CO_2 percentage in the mixed fuel. A SI DI-CNG engine operating at 180°CA BTDC injection timing and various engine speeds was used.

Experimental Setup and Procedures

The research study was carried out using a single-cylinder SI DI research engine. Table 1 provides the detailed specifications of the engine and figure 1 presents the experimental setup diagrams. For the fuels used, CO_2 had 99% purity, while CNG has its detailed composition displayed in table 2 [20] for the tests. The power of the engine was measured with a direct current dynamometer coupled to the engine. All tests were performed at the CAREM stationed in the Department of Mechanical Engineering, Universiti Teknologi PETRONAS (UTP).

Table 1: SI DI Engine Specifications

Displacement volume	**399.25 cm^3**
Cylinder Bore	76 mm
Cylinder Stroke	88 mm
Compression Ratio	14:1
Exhaust Valve Closed	350º BTDC
Exhaust Valve Open	225º ATDC
Inlet Valve Open	372º BTDC
Inlet Valve Closed	132º BTDC
Dynamometer	Direct Current with maximum reading is 50 Nm
ECU	No of Inputs: 21 analogs and 4 digital inputs No of Outputs: 14 multi-purpose outputs Battery Voltage: 8 to 16 V Power Supply dropout: 0.1ms

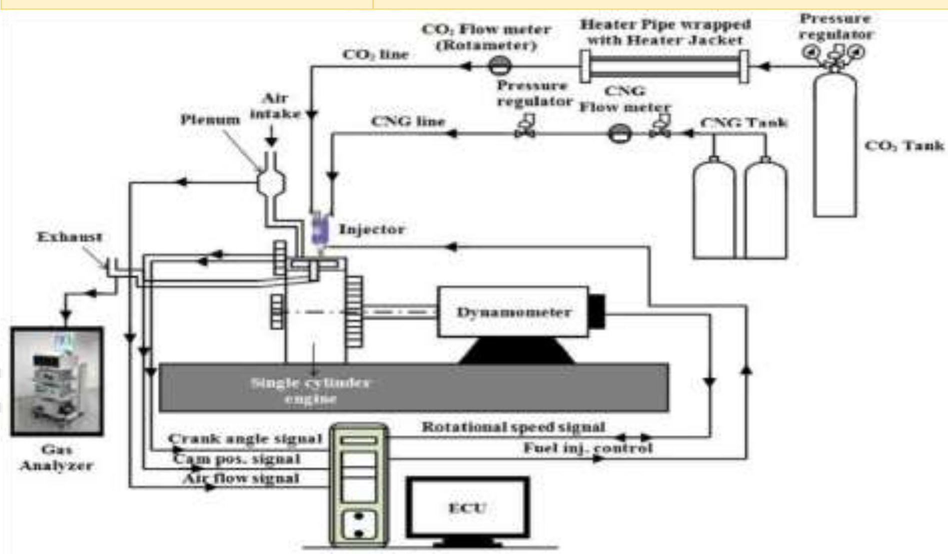


Figure 1: A schematic diagram of the experimental setup

Table 2: Typical CNG Composition in Malaysia [20]

Component	Symbol	Typical Vol. (%)	Max. (%)	Min. (%)
Methane	CH_4	87.3	92.8	79.0
Ethane	C2H6	7.1	10.3	3.8
Propane	C3H8	1.8	3.3	0.4
Butane	C4H10	0.7	1.2	0.1
Nitrogen	N_2	2.2	8.7	0.5
Carbon dioxide	CO_2	0.9	2.5	0.2

For the experimental work, the engine operation was controlled with a computer system, installed with an electronic control unit (ECU) remote interface. With the interface, the engine operation was controlled, and the data were acquired. After the operation of the engine for less an hour, the data were captured and recorded. The incylinder pressure was measured using a water-cooled Kistler piezoelectric pressure transducer together with a charge amplifier. The crank angle encoder was equally connected to the data acquisition system to establish the position crank angle. At the top center of the combustion chamber was the fuel injector with a spark plug placed
6 mm offset of the injector. For this experiment, a narrow-angle injector (NAI) with 30^o cone angle was selected to ensure that the spray penetration of the mixed fuel is maximized.

A baseline fuel (CNG) and four CO_2 proportions by volume (10, 20, 30, and 40%) in the mixture of CNG-CO_2 were used to conduct the study. With the fuel injection system, two fuels (CNG and CO_2) can be operated at the same at high injection pressure. From figure 2, inlet gas 1 and 2 are the intake ports that supplied the CNG and CO_2 fuels at 18 bar injection pressure to the engine via the injector respectively. At mixing points, the mixture of the fuels briefly mixed before being injected into the engine cylinder. A calibrated CNG and CO_2 mass flow meters and digital mass flow were used to control and measure the flow rate of CNG and CO_2, respectively.

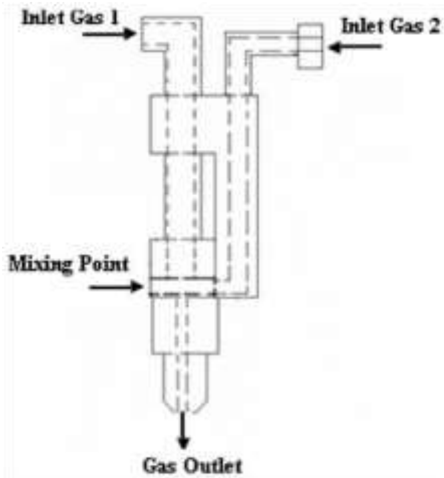


Figure 2: A typical dual fuel injector

At a stoichiometric ratio and 180^{o}CA BTDC injection timing, the ignition timing was adjusted to obtain the maximum brake torque for all the tests carried out. Various engine speeds ranging from 1500 rpm and 4000 rpm were used to run the engine.

Under the set experimental conditions, the engine was run at wide open throttle (WOT) with the AFR controlled by the ECU remote interface.

Experimental Results and Discussion

At 180^{o}CA BTDC injection timing, the effects of CO_2 proportion in the mixture of CNG-CO_2 are illustrated in figures 3-6.

Engine Brake Torque

The graph of the engine brake torque at constant injection duration against the engine speeds for several percentages of CO_2 is presented in figure 3. Generally, it can be observed that the torque curves for all the cases increased to the maximum value at about 3000 rpm and then began to decrease as the engine speeds were increased. It is clearly shown from the graph that as the CO_2 percentage in the mixed fuel was increased, the torque curves continued to decrease at all the operating speeds with the maximum torque achieved occurred at 3000 rpm. Moreover, the graph indicates that 10% and 20% CO_2 proportions in the mixture can be accommodated at all the operating engine speeds, while 30% and 40% CO_2 proportions can accommodate up to 3000 rpm operating engine speeds before the engine became unstable. At all the engine operating speeds and injection timing of 180°CA BTDC, the torque decreased with an increase in the CO_2 percentages. The reason was that the heating content value of the mixed fuel reduced as the CO_2 percentage was increased. The consequence of this was a reduction in the burning velocity and, of course, in the engine output performance. But, the reduction in torque from baseline fuel for range of 10% - 20% CO_2 proportion in the mixed fuel seems insignificant compared torque values of higher CO_2 percentages. This corroborates the work of [12] that the added CO_2 proportion within the range of 10% - 20% dissociated into CO and O atom. When this occurred, the dissociated gases aided combustion and hastened mixture reaction.

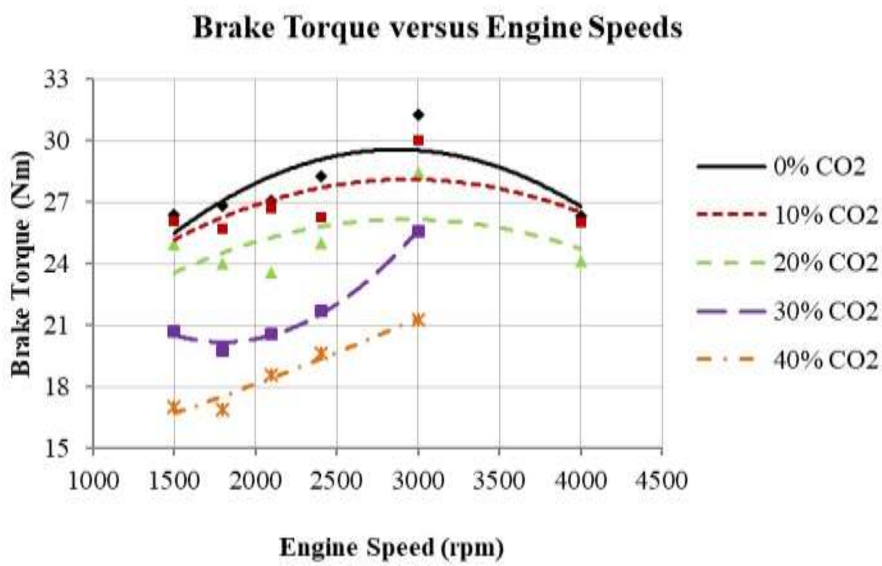


Figure 3: Brake torque against CO_2 percentages at 180°CA BTDC

At 1500 rpm and 180°CA BTDC injection timing, slight reductions were recorded for the BT at 10% and 20% CO_2 percentages in the mixed fuel, while the reductions were significant for 30% and 40% CO_2 percentages in the mixture. However, the torque reduction observed at higher engine speed was due to the decrease in volumetric efficiency and friction. At 4000 rpm the drop in the BT from the pure CNG was noted to be less than 10% at 10% and 20% CO_2 proportion, respectively. While for 30% and 40% CO2 proportion, the drop in their BTs were significant. The maximum torque was observed to occur at 3000 rpm for all the CO_2 percentages in the mixed fuel.

Engine Brake Power

The effect of CO_2 percentage in the mixed fuel of CNG and CO_2 on brake power (BP) is illustrated in figure 4. For all the CO_2 percentages, the BP curves increased as the engine speed was increased until it reached its peak point at 4000 rpm engine speed.

Moreover, with an increase in the CO_2 percentage in the mixed, the BP decreased at all engine speeds. For 10% and 20% CO_2 percentage in the mixed fuel, slight decrease in BP from the pure CNG was noticed at 1500 rpm; while a significant decrease in BP was noticed for 30% and 40% CO_2 percentages in the mixed fuel. The BP peak point occurred at 4000 rpm, and the drop in performance was observed at 10% and 20% CO_2 percentage in the mixed fuel.

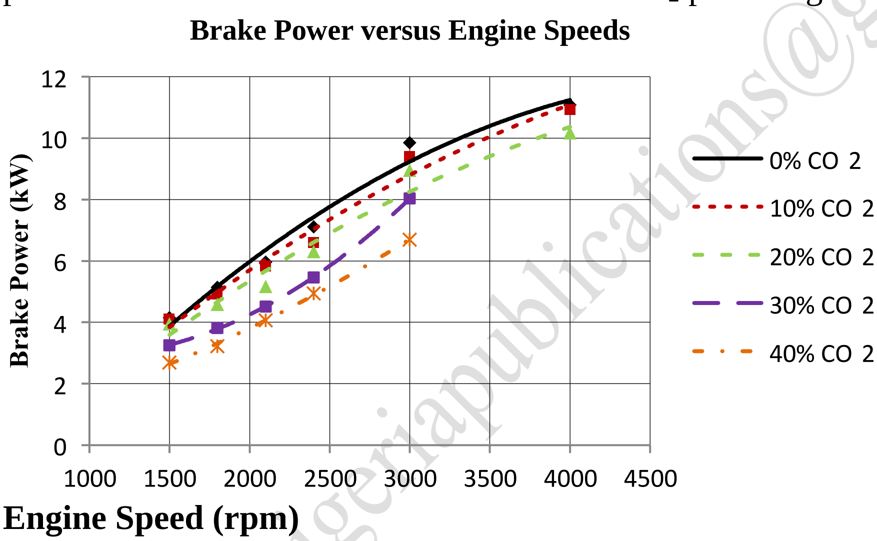
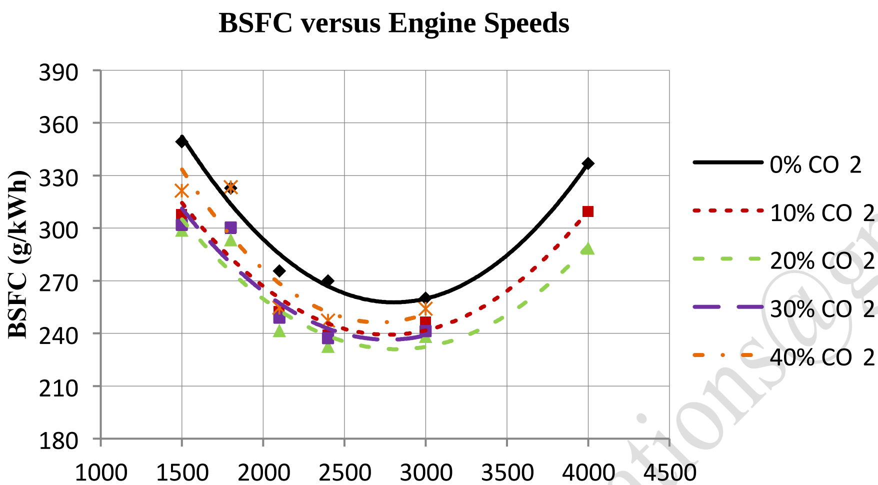


Figure 4: BP against CO_2 percentages at 180°CA BTDC

Brake Specific Fuel Consumption

Figure 5 presents the BSFC against the engine speeds for different CO_2 percentages in the mixed fuel. For the various CO_2 percentage in the mixed fuel, the BSFC decreased with an increase in the engine speed till around 3000 rpm and later increased as the engine speeds were further increased up to 4000 rpm. Moreover, when compared with the baseline fuel, the BSFC decreased as the CO_2 percentage was increased. For all the engine speeds, the BSFC dropped to the lowest value when the engine speed was about 3000 rpm, which is the speed at which the peak engine torque was achieved. It is worthy of note from figure

5 that with the addition of 20% CO_2 proportion in the mixture at 3000 rpm, which is the speed at which peak torque was achieved, the BSFC value reduced when compared to the pure CNG. However, a further increase in the CO_2 percentage to 30% and 40%, BSFC was slightly lowered when compared with the baseline fuel.



Engine Speed (rpm)

Figure 5: BSFC against CO_2 percentages at 180°CA BTDC

The possible reason for this is that the amount of CNG in the mixed fuel continually displaced as the CO_2 percentage added into the mixed fuel was increased, thus, resulted to a decrease in the BSFC of the mixture of CNG-CO_2. Owing to this, the BSFC of the mixed fuel continued to decrease when compared to baseline fuel.

Brake Thermal Efficiency

The brake thermal efficiency (BTE) for the mixed fuel at all the engine speeds and constant injection duration is illustrated in figure 6. Expectedly, the resulting plots that produced the BTE characteristics was the inverse of the BSFC plot. The BTE increased with an increase in the engine speed till around 3000 rpm, and after that began to decline as the engine speed was further increased. With 20% CO_2 percentage in the mixed fuel, the peak BTE occurred at 3000 rpm. A possible explanation for this might be that with not more than 20% CO_2 percentage in the mixed fuel; there was more oxygen availability coming from oxygen components of intake air and dissociated carbon dioxide.

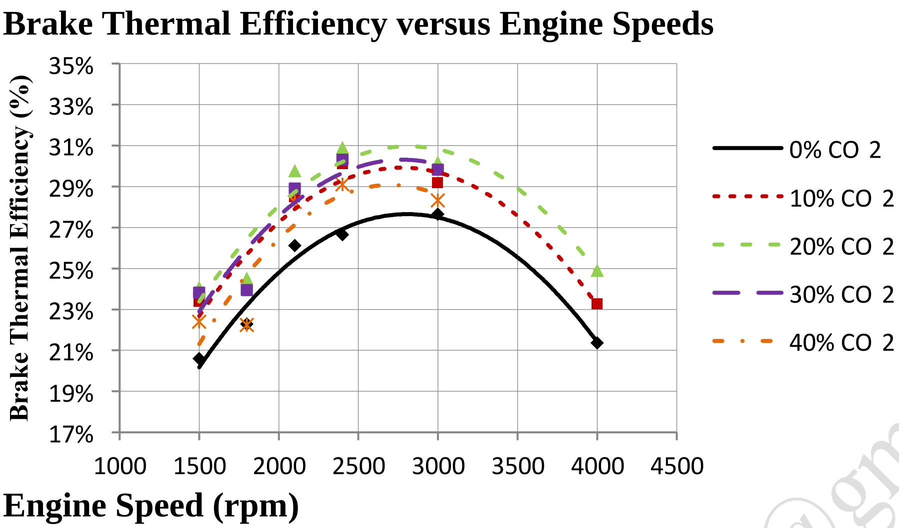


Figure 6: BTE against CO_2 percentages at 180°CA BTDC

Conclusions

This paper examines the impact of CO_2 percentages in the mixed fuel of CNG-CO_2 at various engine speeds on the performance of the DI engine. The engine was set to run at 180°CA BTDC and stoichiometric air-fuel ratio. The experimental results show that with the presence of CO_2 into the baseline fuel, there were reductions in brake torque and brake power when compared to the baseline fuel (pure CNG). However, at 3000 rpm engine speed and 20% CO_2 addition, brake specific fuel consumption, and brake thermal efficiency seem better than CNG. For the utilization of high carbon dioxidenatural gas content and for a DI CNG engine to achieve optimal performance, the CO_2 proportion and engine speed must be set at 20% and 3000 rpm, respectively. However, further research should be undertaken to study the exhaust emissions and combustion visualisation characteristics of high carbon dioxide content natural gas. The outcomes of this research, if implemented, will help to reduce fossil fuel consumption and encourage sustainable development through the use of alternative fuels.

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