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The combustion and emission characteristics of the diesel engine operated on a dual producer gas-diesel fuel mode

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Abstract

Producer gas is a low energy density gaseous fuel converted from carbonaceous materials through a thermochemical process. The gas can be exploited to operate a diesel engine on a dual-fuel mode to partially reduce diesel fuel use. The present study intends to investigate the impacts of gas flow rate and diesel injection timing (DIT) on a diesel engine operated on a dual-fuel mode at a high engine speed of 3,000 rpm. The findings highlight that the peak pressure occurred later and was lower at a higher gas flow rate. The peak pressure was higher and advanced when the DIT was advanced and the engine load was increased. Large increases in CO_2, HC, and CO concentrations were found in the dual-fuel mode, specifically at low loads. Unlike the findings using a medium engine speed, the specific NO_X emissions were higher for the dual-fuel mode operation. Based on these empirical results, a dual-fuel engine should be operated at a high engine load and a gas flow rate of 10 kg/h. A slightly advanced DIT is required – roughly 3 degrees of crank angle. Furthermore, a dual producer gas-diesel engine should not be operated at the maximum gas flow rate.

Keywords: Producer gas, Combustion characteristics, Diesel engine, Dual-fuel mode, Jatropha seed

Nomenclature

ABDC	After bottom dead center	g/kWeh	Grams per kilowatt hour of electrical power
ATDC	After top dead center	kWe	Kilowatts of electrical power
BBDC	Before bottom dead center	kWth	Kilowatts of thermal power
BTDC	Before top dead center	NHRR	Net heat release rate
CHR	Cumulative heat release	PG	Producer gas
deg	Degrees	rpm	Revolutions per minute
DIT	Diesel injection timing	SDC	Specific diesel consumption

1. Introduction

Exploration of bio-renewable energy is currently receiving increased interest owing to rising petroleum fuel costs and depleted fossil fuel reserves. Exploitation of carbonaceous material as feedstocks for gasifier-engine systems can partially or fully replace the petroleum fuels for stationary power generation. This technology is receiving more interest in remote districts that have poor accessibility to fossil fuels [1]. A gasifier converts biomass into a gaseous fuel, widely called producer gas, based on a thermo-chemical process. Gasification efficiency varies from 48% to 77%, and it is primarily influenced by the physicochemical properties of the biomass used, the design of the gasifier, gasification parameters, and gas cleaning cum cooling designs [2-3]. Producer gas is mainly composed of H_2, CO, CO_2, CH_4, and N_2, with a lower heating value of 4–6 MJ/kg and a density of 1.287 kg/m^3 at ambient conditions [3]. The gas can also be used to run spark ignition (SI) engines to fully replace gasoline, but a high power derating is observed in the range of 40–70% [4]. Dual fueling of producer gas to operate a diesel engine can partially replace petroleum diesel with a low power derating of 20–30% [5]. Not surprisingly, there are many previous studies of dual producer gas-diesel engines, e.g., on engine performance, exhaust gas emissions, and combustion characteristics.

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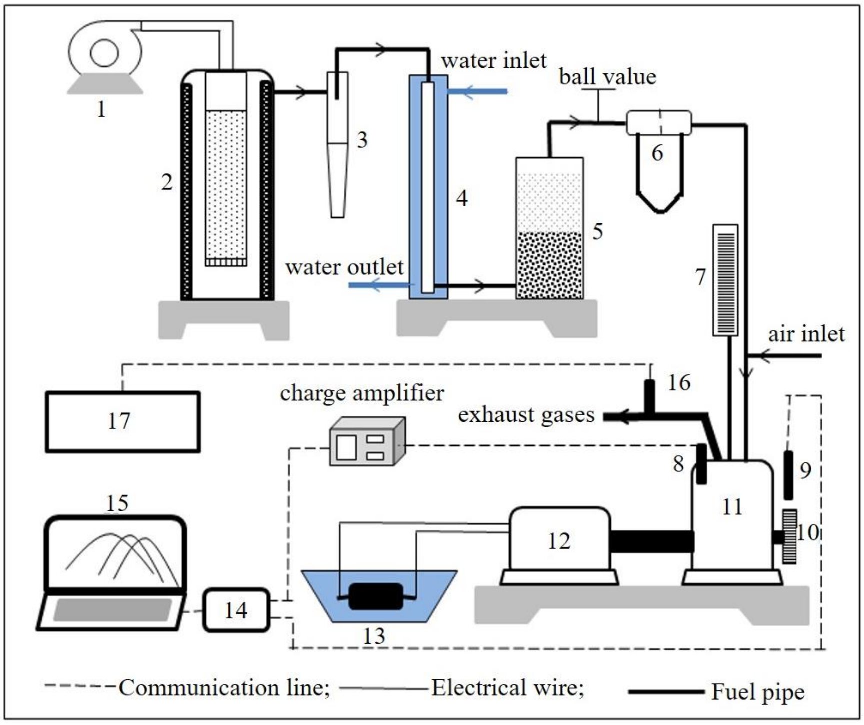

Figure 1 The diagram of the experimental set-up: 1. Air blower, 2. Gasifier, 3. Cyclone filter, 4. Heat exchanger, 5. Dried-bed filter, 6. Orifice and U-tube manometer, 7. Diesel glass burette, 8. Pressure transducer, 9. Inductive speed sensor, 10. Engine cooling fan, 11. Diesel engine, 12. Electrical alternator, 13. Water heaters, 14. Data logger, 15. Laptop, 16. Exhaust gas probe, 17. Exhaust gas analyzer

The current literature has explored the operation of dual-fuel engines at their maximum diesel replacement rates [6-9]. It was found that the diesel replacement rate can be increased up to 86% under an engine load of 60% [6], with lower NO_X emissions. However, their brake-thermal efficiency is lower and HC, CO and CO_2 emissions are higher. More recent studies have looked at the effect of gas mass flow [10-14]. An increase in gas flow rate raised the diesel replacement rate but exacerbated HC, CO and CO_2 emissions and smoke opacity [10, 12-14]. Advancing the DIT led to a reduction in unburnt fuel (i.e., HC and CO) [15]. A dual-fuel engine should not be operated at the maximum diesel replacement due to its less complete and inefficient combustion [16].

Some studies investigated the combustion characteristics of producer gas-diesel engines. The results typically show that the ignition delay is prolonged, the combustion peak pressure was lower and it occurred later in a producer gas-diesel fuel mode [17]. The peak pressure of a dual-fuel engine increases with the engine load yielding better combustion [11]. The ignition delay was shortened by adjusting the biodiesel blend [17]. More interestingly, HC, CO, and particulate emissions were reduced when biodiesel was used in lieu of fossil diesel [18]. The combustion pressure was possibly increased by advancing the injection timing (DIT) of the liquid fuel and increasing the H_2 level in the producer gas [15, 19]. Other approaches to improve combustion characteristics are to increase the compression ratio [20] and the injection pressure [21] as well as through pilot injection splitting [22]. The impact of these three approaches for the producer gas-diesel combustion characteristics has been studied [23]. The other most recent studies empirically investigated the effect of the geometry of the combustion chamber on the combustion characteristics of a dual-fuel engine. It was noteworthy that a Toroidal re-entrant combustion chamber (TrCC) increased the combustion pressure without any exhaust problems. This was due to better air entrainment and air-fuel mixing [24].

Much of the literature has focused on engine operation at medium speeds (roughly 1,500 rpm). The impact of the gas flow rate and DIT on the combustion characteristics of a dual producer gas-diesel engine are less studied at high engine speeds, e.g., 3,000 rpm [25-26]. The emission characteristics are largely unexplored at high engine speeds. Therefore, this study attempts to explore the effect of producer gas mass flow and DIT on the combustion and emission characteristics of a diesel engine operated on a dual-fuel mode at 3,000 rpm. A downdraft gasifier was used to convert the Jatropha seed into a gaseous fuel. The technical feasibility of high engine speed operation is expected to uncover a direction for future research and applications.

2. Methodology

This section briefly describes the experimental set-up and settings as well as the theoretical framework related to combustion used in the current study.

2.1 Experimental set-up

The schematic representation of the experimental set-up adopted in the current study is illustrated in Figure 1. The system typically consists of a gasifier, a gas cleaning unit and

Table 1 Basic technical specifications of the gasifier

Item	Description
Type	Closed top, throatless, downdraft
Gasifying agent	Air
The weight of gasifier (kg)	30
Critical dimension (mm)	D = 350 / h = 1800
Capacity (kWth)	130
Biomass consumption rate (kg/h)	5
Biomass type	Unpressed Jatropha seed
Efficiency (%)	~77

Table 2 Main characteristics of the test engine

Item	Description
Model	KM 186F
Engine type	Single cylinder, 4-stroke, air-cooled, direct injection, diesel engine
Bore×stroke (mm)	86×70
Connecting rod (mm)	117.5
Displacement (cm^3)	406
Rotational speed (rpm)	3,000
Compression ratio	19:1
Inlet valve	Open at an 8.5 degree BTDC, close at a 44.5 deg ABDC
Exhaust valve	Open at a 55.5 degree BBDC, close at a 8.5 deg ATDC
Rated output power (kW/rpm)	5.7/3,000

Table 3 Experimental settings

Case	Abbreviation used in figures and tables	DIT (BTDC)	Gas (kg/h)	Engine load (%)
1	9BTDC, 0PG, 53%	9 deg BTDC	0	53
2	9BTDC, 10PG, 53%	9 deg BTDC	10	53
3	9BTDC, 10PG, 35%	9 deg BTDC	10	35
4	9BTDC, 10PG, 70%	9 deg BTDC	10	70
5	12BTDC, 10PG, 53%	12 deg BTDC	10	53
6	12BTDC, 20PG, 70%	12 deg BTDC	20	70

a diesel generator set. The gasification unit of the current study was comprised of a closed top, throatless, downdraft gasifier and its cleaning system. Table 1 outlines the basic specifications of the gasifier.

A KM 186F engine was used in the experimental set-up. Its specifications are summarized in Table 2. The air intake system was modified to induct gas synchronized with the air at an inlet into the engine. Test instruments were integrated into the set-up to measure the engine speed, the producer gas mass flow rate, and the content of the exhaust gases. The engine speed was measured using a microprocessor tachometer with an accuracy of ±0.5 rpm. An MRU model DELTA 1600-L multi-exhaust gas emissions analyzer was employed to detect the concentration of flue gas components (CO_2, HC, CO, and NO_X). The measurement accuracy of this instrument is ±12 ppm for HC, ±0.06% for CO and ±5 ppm for NO_X. The producer gas flow rates were measured using an orifice and U-tube manometer with water as the manometric fluid. The gas flow rates were calculated based on Bernoulli's principle. The flow is assumed to be non-compressible because its streamline is inviscid and steady.

The in-cylinder pressures were measured using an AutoPSI pressure transducer mounted on the head of the combustion chamber. A pressure transducer was fastened to a charge amplifier. An inductive type speed sensor was mounted close to the 27-tooth cooling fan of the engine. This was done to detect the crank angle positions per revolution of the crankshaft irrespective of the engine speed. The data was recorded using a PicoLog 1000 Series Data logger as a medium-speed data acquisition system.

2.2 Experimental settings

Jatropha seed was used as the feedstock for the gasifier. The gas was cleaned and cooled to the ambient temperature prior to flow into the engine. The engine was operated at two different producer gas flow rates, 10 kg/h and 20 kg/h. Two different DITs at 9 deg before top dead center (deg BTDC) and 12 deg BTDC were used. Three different engine loads, low, medium and high loads at 35%, 53% and 70 of full load were employed. The attribute levels of each control factor were designed to minimize the experiment costs and duration. The engine speed was maintained at 3,000 rpm under all experimental conditions. Each setting is listed in Table 3. These settings allow us to elucidate the impact of the design factors on the combustion and emission characteristics of the dual producer gas-diesel fuel engine. While the gas flow rate was fixed, the diesel quantity was varied to reach the desired engine load. The engine was coupled to a 3 kWe electric generator loaded by five 0.5 kWe electric heaters connected in series. The engine load was varied using on-and-off switches. The properties of the gaseous and liquid fuels used in the current study are summarized in Table 4. The data reported for each experiment was the average of three replicated trials to ensure repeatability. The standard deviations of the output variables are listed in Table 5.

Table 4 The properties of test fuels

Properties	Diesel [27]	Producer gas [28]
Density (kg/m^3)	840	1.19
Caloric value (MJ/kg)	43	4.69
Viscosity (cSt)	2–5	–
Flashpoint (°C)	75	–
Cetan number	45–55	–
Cabon content	–	0.1266 (kg/kg Gas)
Gas compositions	–	H_2 = 7.9% CO = 8.8% CH_4 = 2.1% CO_2 = 18% N_2 = 60% O_2/Ar = 3.8%

Table 5 Standard deviations of the specific diesel consumption and emissions

Case	Experimental setting	Diesel (g/kWeh)	CO_2 (g/kWeh)	CO (g/kWeh)	HC (g/kWeh)	NO_X (g/kWeh)
1	9BTDC, 0PG, 53%	0.0025	1.5974	0.0439	0.0219	0.0258
2	9BTDC, 10PG, 53%	0.0061	4.9322	0.2257	0.1005	0.0199
3	9BTDC, 10PG, 35%	0.0046	2.4581	0.7132	0.2655	0.0909
4	9BTDC, 10PG, 70%	0.0058	4.4963	0.3734	0.0917	0.0550
5	12BTDC, 10PG, 53%	0.0079	9.3117	1.7769	0.0271	0.0396
6	12BTDC, 20PG, 70%	0.0048	6.9085	0.8085	0.0128	0.0234

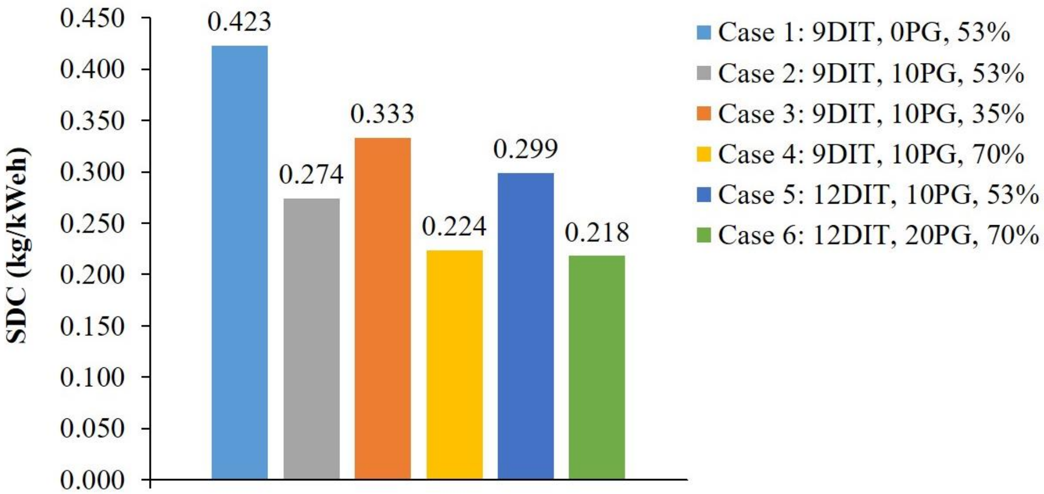

Figure 2 Specific diesel consumption (kg/kWeh)

2.3 Heat release analysis

The net heat release rate (NHRR) was calculated by applying the first law of thermodynamics when both the intake and exhaust valves are closed, and the crevice flow effects are ignored [29]. Thus, the cylinder is considered as the system boundary. The NHRR is a difference between the heat released by the combustion of fuel and the heat transfer from the system by convection. Additionally, the cylinder contents are assumed to be an ideal gas with uniform temperature at each instant in time during the combustion process. Equation 1 is used to plot the NHRR proposed by [29]. The specific heat ratio, γ, ranges from 1.30 to 1.35 for diesel heat release analysis, but the appropriate value is not well defined for the heat release information [29-30]. However, the effect of changes in γ on the NHRR is negligible [31]. γ = 1.35 is typically selected for study, and the cumulative heat release (CHR) can be calculated through integration of the NHHR [29-31].

$$\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} \qquad (1)$$

where p is the in-cylinder pressure, V is the cylinder volume, $\frac{dQ_n}{d\theta}$ is the NHRR (J/deg), γ is the specific heat ratio $\left(\frac{C_p}{C_v}\right)$, and θ is the crank angle (degrees).

3. Results and Discussion

3.1 Specific diesel consumption (kg/kWe)

Figure 2 illustrates the specific diesel consumption (SDC). The SDC of the dual-fuel mode was lower than that of the 100% diesel mode (see cases 1 and 2). The SDC for all the gas flow rates decreased when the engine load was increased in light of better combustion (see cases 2, 3, and 4). The SDC increased slightly when the DIT was advanced from 9 to 12 deg BTDC (see cases 2 and 5). Advancing the

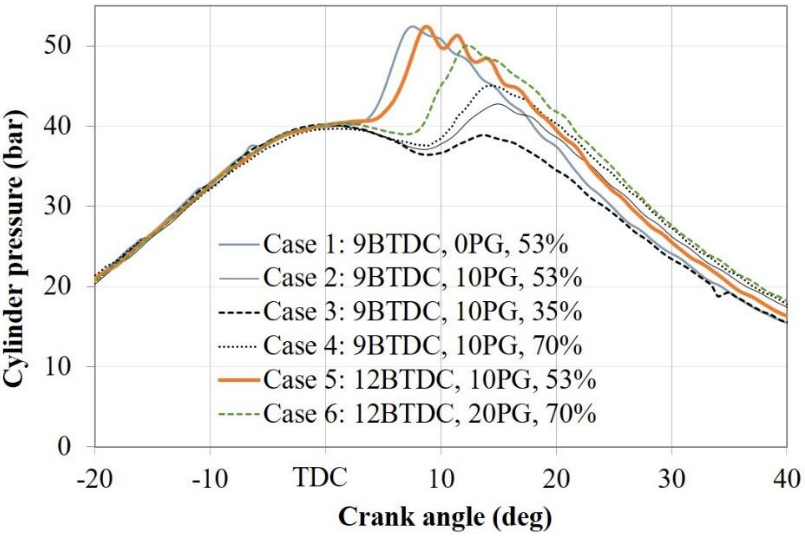

Figure 3 Combustion pressure (bars)

diesel injection timing results in better spray development and a more rapid burning during the premixed combustion phase [3, 32]. However, a further advance of pre-injection timing provokes a decrease in in-cylinder temperature and spray pressure [33]. A suboptimal position of peak pressure could decrease the peak value. Increased use of fossil diesel is probably required to address this issue. The SDC was marginally decreased as the gas flow was increased from 10 to 20 kg/h when the DIT was advanced by 3 deg BTDC at a high engine load (see cases 4 and 6). A further increase in the gas flow rate in excess of 10 kg/h results in less efficiency, requiring substitution of more fossil diesel fuel [28], especially at high loads and advanced diesel injection timing.

3.2 In-cylinder pressure and rate of pressure rise

The pressure curves are the average of 20 consecutive combustion cycles. Figure 3 shows the profiles of combustion pressures in terms of crank angle (degrees). The peak pressure occurred later and was lower for the dual-fuel mode compared with the 100% diesel fuel mode (cases 1 and 2). This is generally due to an increased ignition delay period and mixture preparation during the ignition delay [3]. The ignition delay results from the slow-burning nature of the producer gas and the reduced diesel spray used as an ignition source [3, 32]. Producer gas is generally composed of a high percentage of inert gases (i.e., N2 and CO2) that degrade the quality of a gaseous fuel. Fumigating producer gas into a CI engine, therefore, reduces the diesel quantity, as can be seen from the reduced cetane number of the fuel-air mixture. A decrease in pilot diesel weakens the potential ignition of the fuel-air mixture and provokes an increased ignition delay [32].

As apparent from cases 2 through 4, the peak pressure of the dual-fuel mode was found higher with an increase in the engine load. The gas flow rate was fixed and the diesel quantity was increased. This produces more ignition centers that lead to faster combustion during the premixed combustion phase and higher temperatures inside the combustion cylinder. However, the location of peak pressure was not noticeably changed. It was likely that the peak pressure was influenced by the diesel quantity, while its position was affected by the producer gas flow rate. Similar findings were reported in other studies [11, 21].

The peak pressure was higher and increased with an advanced DIT (cases 2 and 5). Advancing the injection timing resulted in better spray development and a more rapid burning during the premixed combustion phase [3, 33], while the peak pressure moved closer to top dead center [23]. However, advancing the injection timing at a medium engine load led to a pinging at peak pressure in the dual-fuel mode (detonation or knocking), as illustrated in case 5. Advancing the injection timing above the limited value could cause engine damage owing to knocking [15] and lead to a decrease in in-cylinder temperature and pressure in the spray [34].

Knocking during combustion in the dual-fuel mode was possibly alleviated by increasing the engine load and gas mass flow. This is evident from case 6 relative to case 5. An engine operated at a high load is associated with high combustion temperature, thereby accelerating the chemical reaction of the fuel-air mixture and improving the thermodynamic properties of the intake charge. Operation of an engine at high load promotes fuel-air mixing and increases the fraction of premixed combustion. This improves the combustion intensity of fuel to enhance combustion pressure and pressure rise rate [34].

The rate of pressure rise is the derivative of the in-cylinder pressure, and the former influences the peak pressure. Figure 4 shows the rates of pressure rise. The bottom and the peak of pressure rise rate were lower and occurred later in the dual-fuel mode relative to the 100% diesel fuel mode (cases 1 and 2). The peak of the pressure rise rate increased with an increased engine load (cases 2 through 4). The bottom pressure rise rate occurred earlier at an advanced DIT. The peak of pressure rise rate was noticeably higher and earlier with advancing the DIT (cases 2 and 5). The peak and the bottom of the pressure rise rate was lower and occurred later, respectively, with further gas increases even if the engine load was increased (cases 5 and 6). All the peaks of pressure rise rates were less than 4 bar/deg (Figure 4). The pressure rise rate below 8 bar/deg is acceptable for the engine's durability and prevention of knocking-related problems [35].

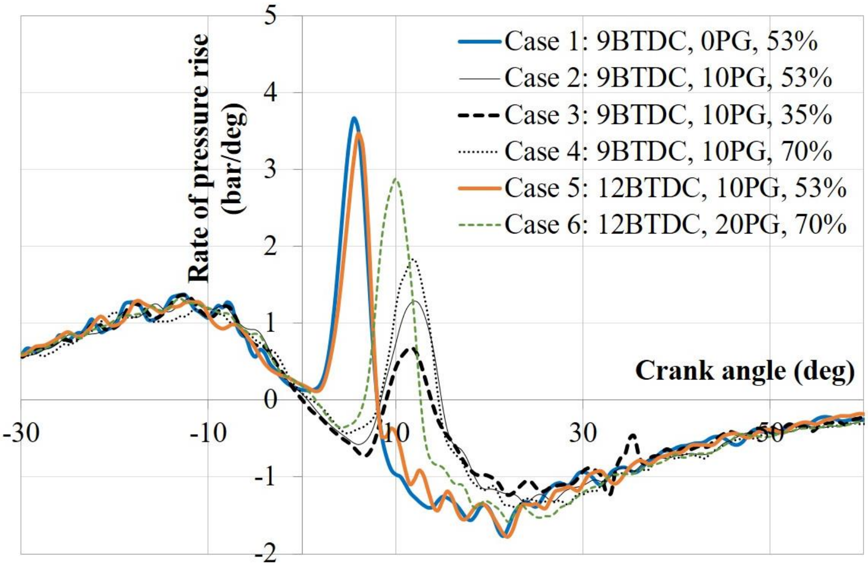

Figure 4 The rate of pressure rise (bar/deg)

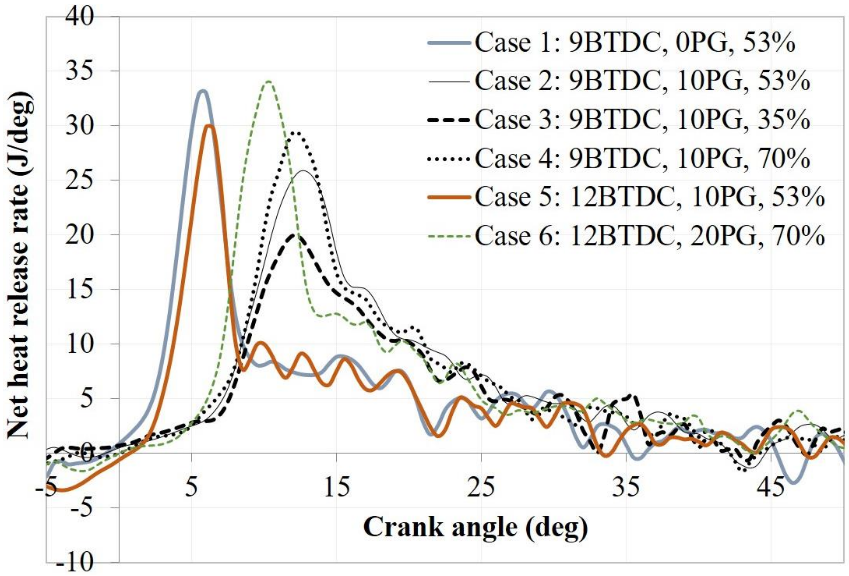

Figure 5 Net heat release rate (J/deg)

3.3 Net heat release rate (NHRR)

Figure 5 illustrates the NHRR. The peak NHRR was delayed at about 7 deg and was lower when the producer gas flow was increased from zero to 10 kg/h (cases 1 and 2). A similar result was reported for a dual LPG-diesel fuel mode [32]. The NHRR peak of the dual-fuel mode was increased when the engine was operated under a high load, but the peak position was not noticeably changed (cases 2, 3 and 4). The NHRR peak of the dual-fuel mode was higher and advanced if the DIT was advanced, but the fluctuation of NHRR was determined after the peak value (see case 5). The fluctuation was considerably mitigated when the gas flow was doubled and the engine load was increased to 70% of full load.

Dual gas-diesel combustion consists of three stages [32]. In the first, heat is mainly released by the premixed combustion of the entire pilot diesel fuel and a small amount of the gas entrained in the spray. In the next stage, that portion of the gas-air mixture in close proximity to the pilot spray is auto-ignited, and the diffusive combustion phase of the remaining diesel fuel occurs. In the last stage, the heat release is contributed to flame propagation of the gas-air mixture initiated from the spray zone. As a consequence, the NHRR was higher for the dual-fuel mode after the NHRR peak, compared with that of the 100% diesel mode (cases 1 and 2).

3.4 Cumulative heat release (CHR)

The CHR traces in terms of crank angle are illustrated in Figure 6. Compared with the CHR of the 100% diesel fuel mode, the CHR of the dual-fuel mode was found marginally lower before 19 deg BTDC as a result of late combustion during the premixed combustion phase. However, the CHR was much higher from this crank angle onwards due to a longer combustion of the producer gas in the diffusive combustion phase (cases 1 and 2). It is noteworthy that the CHRs of the dual-fuel modes during the diffusive combustion phase were comparable, between 53% and 70% of the full engine load (cases 2 and 4). This indicates that

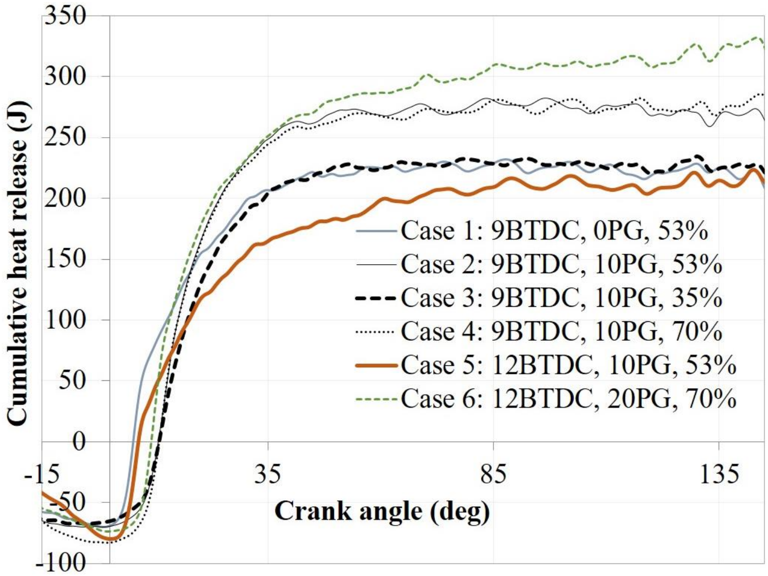

Figure 6 Cumulative heat release (J)

Table 6 Ignition delay and combustion duration periods (deg of crank angle)

Case	Abbreviation	Start of combustion (deg ATDC)	End of combustion (deg ATDC)	Ignition delay (deg)	Combustion duration (deg)
1	9BTDC, 0PG, 53%	1	55	10	54
2	9BTDC, 10PG, 53%	6	80	15	74
3	9BTDC, 10PG, 35 %	7	65	16	58
4	9BTDC, 10PG, 70 %	5	85	14	80
5	12BTDC, 10PG, 53%	2	90	14	90
6	12BTDC, 20PG, 70%	4	145	16	144

combustion of the dual-fuel mode was more efficient when the engine was operated at higher loads with increased levels of pilot diesel (cases 2, 3, and 4 of Figure 2). When the DIT was advanced from 9 to 12 deg BTDC, the CHR was higher during the premixed combustion phase (from 3 to 13 deg ATDC), but it was considerably lower during the diffusion combustion phase (after 13 deg ATDC), as can be seen in cases 2 and 5. Injection timing has a significant effect on the combustion pressure and the stability of engine operation (e.g., engine knocking or pinging). The noticeably lower CHR of case 5 relative to case 2 could be due to a fluctuation of the NHRR ascertained after the peak value. The CHR during the diffusion combustion phase considerably increased when the gas flow was augmented from 10 to 20 kg/h, even if the engine load was increased from 53% to 70% of the full load (cases 5 and 6).

3.5 Ignition delay and combustion duration periods

Ignition delay is a time interval between the injection and the start of combustion, while combustion duration is the time between the start and end of combustion. The injection time is pre-determined, whereas the initiation of ignition is variable depending on the pressure rise rate close to top dead center [36]. It can be determined from Figure 4. The end of combustion may be considered at the maximum point of the CHR curve [30-31], and it can be identified in Figure 6. The ignition delay and the combustion duration values are listed in Table 6.

The ignition delay was prolonged with an increase in gaseous fuel in the charge (cases 1 and 2). It has been reported for LPG, biogas, and mixed producer gas-H_2 [19, 30, 32, 37]. Mixture preparation before the pre-ignition phase [30, 37], reduction of pilot diesel quantity and oxygen concentration [19], lower temperature and pressure during the start-up of the ignition [19, 37] were thought to be the main interacting causes of ignition delay. Additionally, the presence of CO_2 in producer gas increases the ignition delay as CO_2 has a high specific heat capacity [30] that is associated with a reduction of in-cylinder temperature during the compression stroke. Producer gas cannot be auto-ignited at the same pressure as diesel fuel and the gas combustion, therefore, totally depends on an ignition source, i.e., the pilot diesel. Fumigation of producer gas to partially reduce the diesel used is associated with a reduction in the ignition source. This causes a heat release during the diffusive combustion phase rather than the premixed combustion phase. Correspondingly, a sharp drop of heat release rate during the premixed combustion phase leads to an increase in the ignition delay. For the dual-fuel mode, the ignition delay was lower at a higher engine load due to an increased diesel fuel quantity (i.e., the ignition centers) that leads to an increased cetane number and flame propagation rate (cases 2, 3, and 4). Longer ignition delays at low engine loads are due to the presence of more fuel accumulated in the premixed combustion phase [38]. When the injection timing was

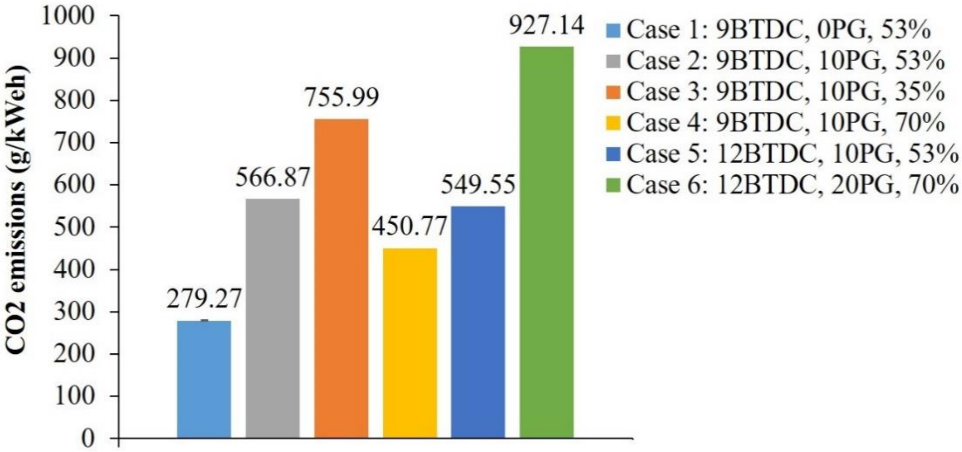

Figure 7 Specific CO_2 emissions (g/kWeh)

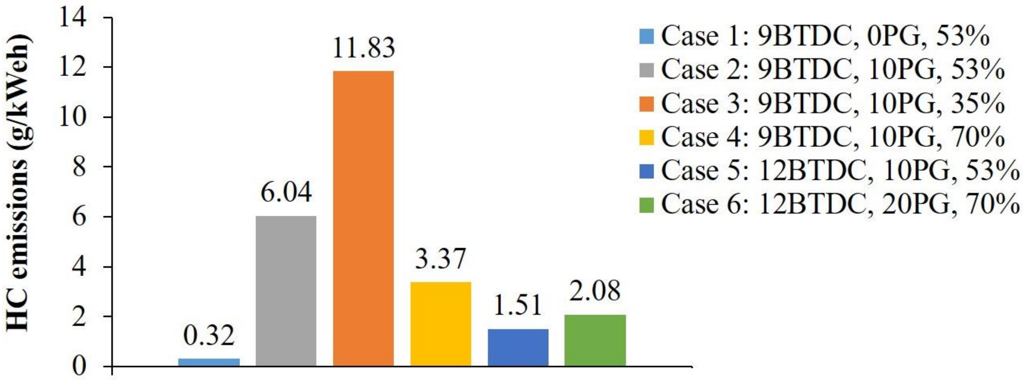

Figure 8 Specific HC emissions (g/kWeh)

advanced from 9 to 12 deg BTDC, the ignition delay was slightly decreased from 15 to 14 deg of crank angle (cases 2 and 5). The same finding also confirmed that the ignition delay was reduced by advancing the DIT [23, 39]. Advancing the injection timing could improve fuel-air mixture reaction activities, thereby increasing the heat release of the fuel combustion during the premixed combustion phase. When the gas flow was further increased to 20 kg/h, the ignition delay was increased to 16 deg of crank angle due to a lower oxygen concentration (displaced by the gas) to support the fuel combustion, even if the engine load increased (cases 5 and 6).

An increase in the gas flow rate and engine load resulted in a higher combustion duration, as can be seen in Table 6. This has been previously reported [19]. An increase in combustion duration of the dual-fuel combustion may result from weakened ignition centers [19]. Therefore, a greater degree of dual-fuel combustion takes place in the diffusive combustion phase than in the premixed combustion phase. The combustion duration increased with greater engine loading (cases 2 through 4). This has also been reported [30]. It requires a longer time for heat release with mixed fuel-air combustion. For case 5, fluctuation of the NHRR for the dual-fueling with advanced injection timing caused the remaining fuel be burned in the late combustion phase. This prolongation of the end of combustion, for durations of up to 90 deg of crank angle (case 5). It is noteworthy that the end of combustion in case 6 was not reached until the exhaust valve was opened. This is due to excessive use of producer gas (i.e., 20 kg/h) that restricts the amount of air inlet and consequently slows the combustion rate [3].

3.6 Emission characteristics

Figure 7 shows the specific CO_2 emissions at various operating settings. The specific CO_2 emissions were higher for the dual-fuel mode (cases 1 and 2) due to the initial presence of CO_2 in the gaseous fuel. However, the specific CO_2 emissions were lower at a higher engine loads by virtue of better combustion (cases 2, 3, and 4). It is noteworthy that the specific CO_2 emissions of the dual-fuel mode were slightly reduced when the diesel fuel injection was advanced from 9 to 12 deg BTDC (cases 2 and 5).

The specific HC emissions are illustrated in Figure 8. The specific HC concentration sharply increased from 0.32 to 6.04 g/kWeh when the gas flow rate was increased from zero to 10 kg/h. This could have been due to a lower air-fuel ratio and subsequently slower combustion with the escape of fuel from the combustion process [3]. The HC emissions decreased when the engine load was increased (cases 2, 3, and 4). It was noteworthy that the HC emissions were reduced by advancing the injection timing of the liquid fuel (cases 2 and 5), as has been reported [23]. Based on case 6, the dual-fuel engine operation at a high gas flow rate should be operated with a high load and advanced DIT.

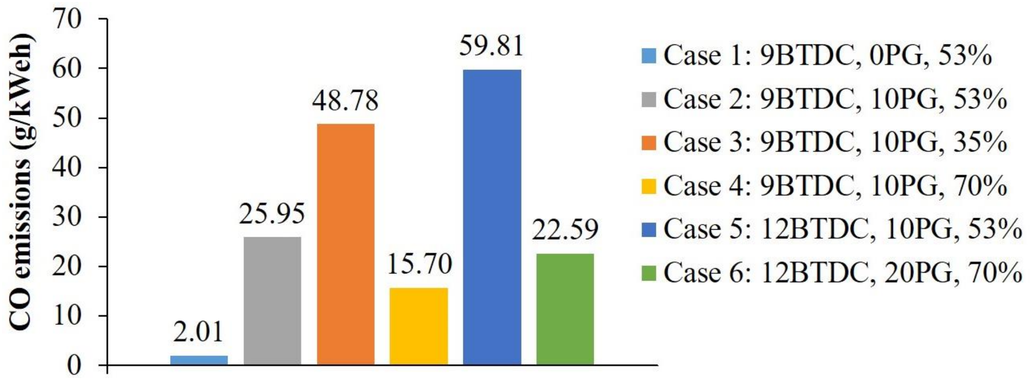

Figure 9 Specific CO emissions (g/kWeh)

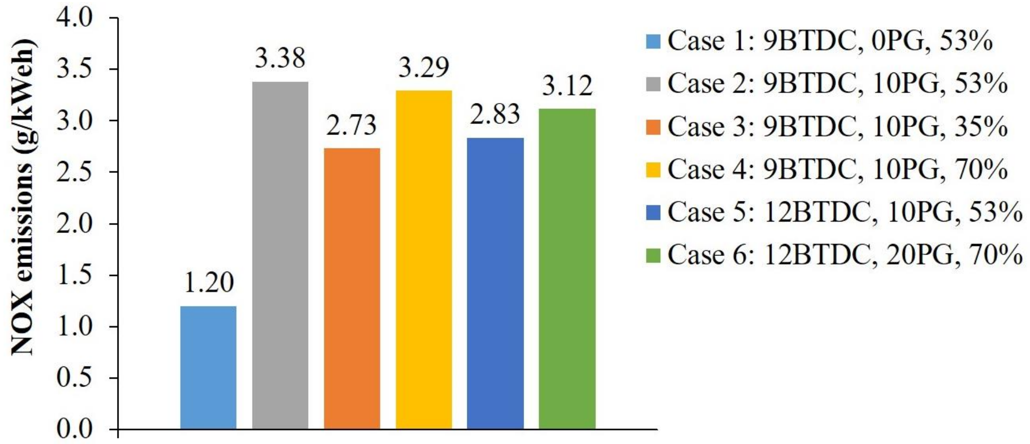

Figure 10 Specific NO_X emissions (g/kWeh)

The specific CO emissions are illustrated in Figure 9. The CO concentration considerably increased when the gas flow was increased from zero to 10 kg/h (cases 1 and 2). This is because of the initial presence of CO in the gaseous fuel as well as less complete combustion [3]. The CO emissions can be decreased by operating the engine at higher loads (cases 2, 3, and 4). Unlike the HC emissions, the CO emissions sharply increased when the DIT was advanced from 9 to 12 deg BTDC. This was caused by a higher CHR during premixed combustion and a lower CHR during diffusive combustion (case 5 of Figure 3), which implies that more HC than CO was burnt. However, the CO emissions were reduced when the gas flow rate and engine load were increased. Beyond that, knocking during combustion was not observed (see Figure 3).

Contrary to engine operation at medium speeds, the specific NO_X emissions were increased from 1.20 to 3.38 g/kWeh (see Figure 10) when the gas flow was increased from zero to 10 kg/h (cases 1 and 2). One explanation is that the CHR of the dual-fuel mode was much higher than that of the 100% diesel fuel mode, which is likely the main cause of the higher combustion temperature (see Figure 6). The exhaust temperature is relatively higher for the dual-fuel mode [10-12], which is an indication of a higher combustion temperature. The availability of oxygen and nitrogen at a high temperature are the two major reasons for NO_X emissions. Nitrogen is typically an inert gas at a low temperature, but it reacts with oxygen to produce NO_X at temperatures in excess of 1,100 ℃ [40]. Similarly, higher NO_X emissions were associated with an increase in the engine load in cases 2 and 3. Nonetheless, the NO_X emissions did not significantly change when the engine load was increased from 53% to 70% (cases 3 and 4). This was probably due to the comparable CHRs of these two cases (see Figure 6). The NO_X concentration decreased from 3.38 to 2.80 g/kWeh when the injection timing was advanced from 9 to 12 deg BTDC (cases 2 and 5). It is likely that the CHR of case 5 was much lower than that of case 2 during the diffusive and late combustion phases, as illustrated in Figure 6.

4. Conclusions

The empirical findings from a dual producer gas-diesel engine operated at a high speed are as follows:

- The peak pressure and NHRR were lower and occurred later for the dual-fuel mode relative to a 100% diesel fuel mode. The peak pressure and NHRR increased with advancement of DIT.

- The ignition delay was noticeably prolonged with an increase in the gas flow. However, it was marginally shortened by increasing the engine load due to a better spray development that improved the fuel-air mixture and burning rate during the premixed combustion phase. A higher combustion duration was seen with increased gas flow rate and engine load as well as an advance in DIT.

- The specific CO_2, HC, and CO emissions were extremely high for the dual-fuel mode. However, the HC and CO emissions were mitigated when engine loading was increased. Unlike previous studies of engines operated at medium engine speeds, the specific NO_X emissions of the dual-fuel mode were found higher than that of the 100% diesel fuel mode.

Based on the experimental findings, the dual-fuel engine should not be operated at its maximum gas flow rate. About 10 kg/h is suggested for this engine. Furthermore, a dual-fuel engine should be operated at a high load, and the DIT should be marginally advanced to increase the peak pressure, by roughly 3 deg of crank angle. Further research should develop mathematical models of specific diesel fuel consumption and emissions in terms of the control factors. Then, models can be used to optimize the operating settings to simultaneously minimize diesel fuel consumption, operating costs, and flue gas emissions. Further research effort is required to compare the operation of a dual-fuel engine at high speed with medium speed.

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