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Review of Compression Ignition Engine Powered by Biogas and Hydrogen

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Abstract: Unsustainable fossil fuels are mainly used to generate power in compression ignition (CI) engines in industry now. Due to fossil fuel depletion and potential environmental hazards, it is necessary for researchers to find alternative energy resources to adequately substitute hydrocarbon fossil fuels in current engines. A huge number of studies have focused on the use of renewable fuels in CI engines along with conventional petroleum fuels. Therefore, this paper aimed to analyze the effect of gaseous fuels added to CI engines as a supplement, such as H2, biogas and syngas, in dual fuel mode with diesel as an alternative fuel. This paper analyzed several important characteristics, on which engine evaluation of CI engines using gaseous fuel as an additive is based, such as combustion, performance and emissions, and compared them with those of CI engines operating in single-fuel mode. The findings of numerous empirical studies are shown in graphs of particular parameters, which were crucial for investigating and assessing the case. The main conclusions indicated that gaseous fuel enrichment caused slight decline of performance in CI dual-fuel engine but actually improved emissions. In addition, this paper thoroughly analyzed various methods to assess the performance of biogas in CI dual-fuel engines and investigated dangerous emission pollution.

Keywords: Biogas; Hydrogen; Brake thermal efficiency; CO_2; CO; NO_x

1 Introduction

CI engines are widely used for power production nowadays, particularly in industry and transport sectors. Power plants that burn petroleum diesel in CI engines also contribute to depleting fossil fuel reserves and degrading ecosystems, which proves these fuels have a disastrous impact on nature and the environment. In particular, the emitted pollutants in the atmosphere, such as nitrogen oxides (NO_x) and particulate matter (PM), deteriorate air quality while releasing carbon dioxide (CO_2), resulting in climate change [1]. Depletion of fossil fuel supplies and environmental pollution are undoubtedly the most critical problems that must be addressed today urgently. Fossil fuels account for about 80% of the total energy, and nearly 50% is related to the transportation sector, which primarily uses IC engines [2]. Considering these problems, it is necessary to switch to a more environmentally friendly fuel for CI engines. Use of renewable fuels in diesel engines is primarily driven by the need to reduce environmental damage from greenhouse gas emissions. Renewable energy accounts for roughly 9% of the total global energy, but is expected to grow up to 12% by 2040 [3]. The specific fuel consumption worldwide in the automotive sector is shown in Figure 1. It is a major research topic worldwide to eliminate or reduce harmful emissions from diesel engines in use today. It is possible to improve specific fuel consumption and reduce emissions using the most advanced electronic controller units in CI diesel engines. However, due to high cost, this technology has limited potential benefits and cannot be taken as a commercially viable solution. As additives of the baseline fossil fuel in diesel engine, several alternative fuels have been investigated and proposed to counteract the harmful effects of CI engine emissions over the past few decades, including various types of vegetable oil, compressed natural gas (CNG), liquefied petroleum gas (LPG), natural gas (NG), hydrogen (H_2) and biogas, etc. [4]. Due to superior air-mixing qualities, availability and being environmentally friendly, gaseous fuels, like natural gas, have been studied a lot to use them in CI engines. Hydrogen is another well-known renewable energy source, but the priority remains to develop a sustainable method to mass-produce the gas at a reasonable cost. Some studies have assessed additive blending with biodiesel, as well as other technologies and alternative fuels for CI engines [5]. Datta and Mandal [6] reviewed CI engines powered by biodiesel fuel, which was a watershed moment in the history of the subject. The potential of emulsion as an alternative fuel was discussed by Debnath et al. [7]. Srinivasan et al. [8] discussed the possibility of finding a new strategy to completely eliminate the dependence on diesel fuel. Therefore, this paper made a comprehensive literature review of the effects of incorporating gaseous fuels (e.g., H_2, biogas, syngas) into a CI dual-fuel engine in terms of its emission, performance, combustion and characteristics.

2 Biogas as an Alternative Fuel

The biogas fuel is considered as a mixture of diverse gases. Addition like CO_2 and N_2 almost absorbs heat released from combustion and lowers in-cylinder temperature. The Wobbe index determines the higher heating value which specifies quality of the fuel. Methane is the main constituent, which defines the biogas heating value. While CO_2 acts as diluent, which reduces the flame speed and higher heating value [9] as well as retards the combustion reaction rate and reduces in-cylinder temperature. CH_4 has a higher heating value which improves the biogas' performance and combustion process.

2.1 Performance Analyses

The performance of CI engines is evaluated by diverse parameters, such as break thermal efficiency (BTE), compression ratio (CR), biogas flow rate (BFR), ignition delay (ID) period, and brake-specific fuel consumption (BSFC). BTE was used to evaluate the ability of CI engines to convert thermoelectric energy into mechanical energy. The ability decreased under dual fuel (DF) mode. Because a large amount of gaseous fuel led to a decrease in combustion temperature and flame speed, the heat loss rate transmitted through the cylinder walls increased [10]. Furthermore, this decreased trend continued along with the increase of biogas share [11]. In addition, due to the slow flame propagation speed (FPS), an increase in BRF during the suction stroke badly affected the induced air volume and poor combustion process, which reduced the temperature inside the cylinder [12]. The maximum value of BFR and EGE essentially depended on the operation load of biogas fuel, the speed of engine, and the concentration of CH_4 in the blends. The local temperature and FPS improved well, while ignition delay (ID) duration lowered with increasing CRs. BTE improvement along with CRs was related to the enhanced in-cylinder temperature, which led to improved potential energy conversion. During higher load operation, CRs reported more dominance than low load because of shorter combustion duration (CD), higher flame speed and in-cylinder temperature [13]. In addition, injection timing (IT) significantly influenced the performance of CI engines, because the early injection of fuel contributed to enhancing the homogeneity of mixture and improving combustion process [14]. Figure 1 demonstrates the effect of biogas energy share (BGES) on the BTE. O_2 was used as a biogas supplement for diesel in DFM, which shortened the ID decline period and reduced the thermal combustion temperature, thus counteracting the carbon dioxide effect [15].

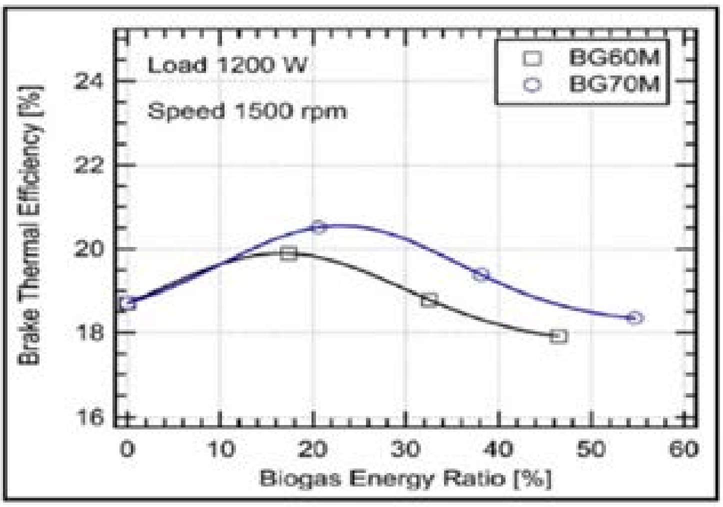

Figure 1. Effect of the biogas energy ratio on the BTE [16]

$$\text{Biogas energy share } (\%) = \frac{\dot{m} \times CV_{\text{biogas}}}{(\dot{m}_{\text{biogas}} \times CV_{\text{biogas}}) + (\dot{m}_{\text{diesel}} \times CV_{\text{diesel}})} \times 100 \tag{1}$$

where, $\dot{m}$ represents flow rate of fuel mass (kg/s); CV represents the fuel calorific value (kJ/kg).

The brake power (BP) is defined as the power achieved by the engine for the transmission shift after considering all losses caused by friction. In addition, BP showed an increasing trend along with the increase of BFR and energy share of CH_4, which was related to the increase of energy input to an engine combustion chamber of DFM from baseline diesel fuel [17]. Brake specific fuel consumption (BSFC) was generally represented by an integrated pilot and gaseous fuel flow and mainly specified by engine load change, which clearly depended on the diffusion of fuel particles on the combustion zone and charge mixture homogeneity under the injection pole. The BSFC increased with increased biogas because of the slowing biogas flame speed of combustion. In this way, an increase in the operating load improved the BSFC, because the atomization of fuel mixture and the combustion process enhanced [18]. The dilution nature of CO_2 lowered BSFC and reduced in-cylinder temperature, combustion efficiency, and flame speed. Oxygen-enriched air was used in DF diesel engine, which significantly reduced CO_2 emissions by decreasing the ignition delay and adiabatic flame temperature of fuel in combustion process [15]. In addition, BSFC enhanced when the load increased due to increased combustion rate and temperature, resulting in a huge air-to-fuel (A/F) ratio. This occurred when the ratio of air to fuel raised. Early start time of fuel injection also had a favorable influence on the BSFC as a result of higher combustion process rates and enhanced fuel formation [19]. Higher CR caused remarkable improvement in combustion efficiency and reduction in CD, contributing to reduction in BSFC [18]. As demonstrated in Figure 2, the BSFC varied against speed and load.

The exhaust gas temperature (EGT) of the diesel engine, associated with the combustion process, is an essential parameter for judging an engine's efficiency. For a DF engine using biogas test fuel, the combustion, which occurred through the power stroke and diffused to the ends of the stroke, lowered cylinder pressure and temperature, mainly because of longer ID and lower EGT [20]. The observed EGT reduction together with the increased BFR rates confirmed the increase of auto-ignition temperature and density equinoctial of the air-fuel mixture [11]. However, the slow-burning nature of biogas fuel resulted in higher exhaust temperatures. Bora et al. [21] observed an increase in EGT for engine working under DF mode at all operating load conditions compared with the single fuel (SF) mode.

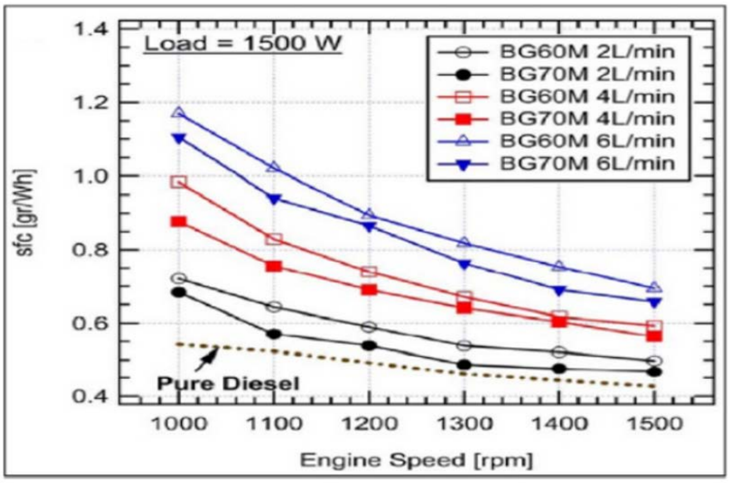

Figure 2. Effect of engine load and speed on the BSFC

In addition, the results showed that higher CR led to lower EGT due to faster flame propagation and shorter total CD. In contrast, Barik and Murugan [11] showed that the increase of CR led to an increase in the combustion temperature, therefore, the EGT increased at high and noticeable levels. Figure 3 shows the effect of the BRF and BP on the EGT.

2.2 Review of Biogas Combustion Mechanisms

When comparing fuels with significantly different heating values and intensity, the brake-specific energy consumption (BSEC) scale was used as a calibration scale, because it precisely described engine output power and the chemical heating energy [22]. Lower A/F rates and combustion cylinder temperatures at lower loads were considered as the causes of increased BSEC rates. However, due to drastically increasing combustion temperature rates, the discrepancy in BSEC between the diesel engine operating on SF mode and that on DF mode became less pronounced with increasing load [23]. In subgraphs (a) and (b) of Figure 4 show the effect of brake power and engine load on the BSFC.

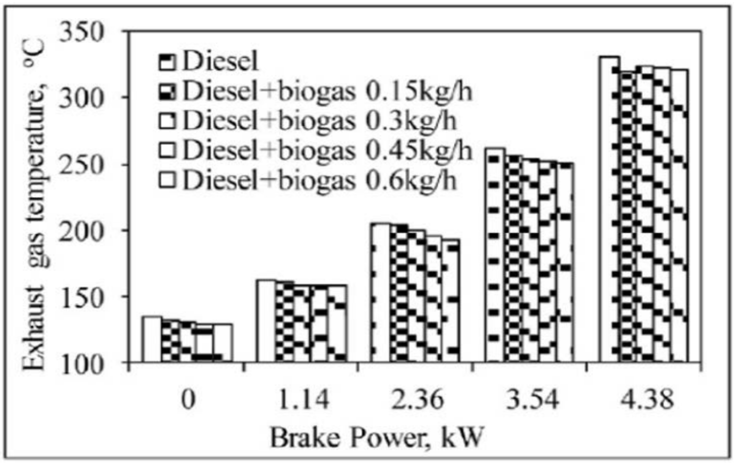

Figure 3. Effect of brake power on the EGT [11]

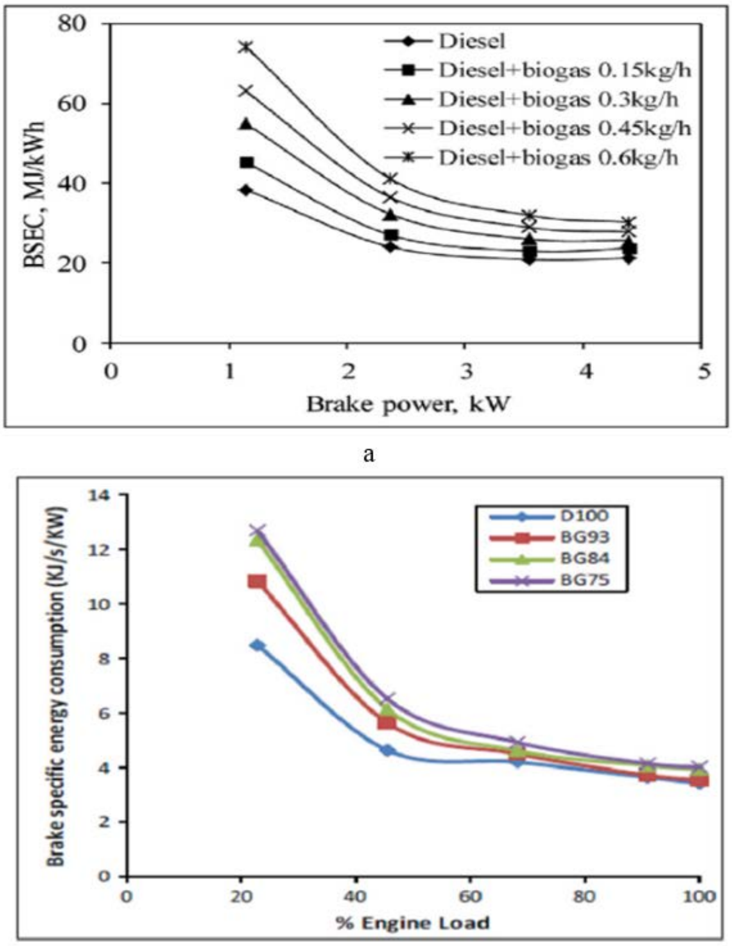

Figure 4. BSEC as a function of (a) BFR and BP, reproduced from the study of Barik and Sivalingam [23] (b) CH_4 content and load, reproduced from the study of Verma et al. [9]

2.2.1 Combustion analysis

Efficient fuel combustion is crucial for engines because this regulates the amount of gasoline used, the engine performance and the amount of pollution produced. Combustion in dual-fuel mode occurred after SI and CI combustion mechanisms were combined, and the two fuels had chemical and physical interaction. The in-cylinder pressure (ICP) directly affected the performance, emissions, and combustion characteristics of diesel engine, such as pressure rise rate, ignition delay (ID), and heat release rate (HRR), which made ICP a very important metric [24]. ICP depended on the premixed combustion phase, which was affected by the fuel mixture quality, operation engine load, local equivalency ratio, and an air-fuel mixture. Unlike the single fuel (SF) mode, DFM had a higher ICP. Furthermore, the DFM incidence of ICP extended to the later of the top dead center (TDC) in the expansion stroke [25]. In addition, increasing BFR and CH_4 concentration in biogas demonstrated the increase of ICP [9]. Although CO_2 concentration in biogas has not proved to significantly affect ICP, longer ID caused the pressure peak to shift a little away from TDC in the diffusion stroke later [11]. All the evidence suggested ICP levels increased with CR increase, advanced injection timing (IT), and O_2 enrichment, because the increased flame speed of combustion and adiabatic flame temperature following the compression stroke caused the strongest flame propagation throughout the combustion chamber and ICP rise. The addition of O_2 to the fuel mixture increased the ICT, ICP and the rest combustion characteristics, while coating with a thermal barrier led to reduction of heat losses [26]. Some investigators have also observed that ICP reduced for diesel engines operating under DF mode due to less reactive fuel mixture and higher specific heat of the gaseous charge mixture, which resulted in extending the combustion period and retarding the start of combustion (SOC) [25]. At a certain crank angle degree (CAD), the time interval between the start of fuel injection (SOI) and the first stages of combustion was represented graphically as the ignition delay (ID) period. ID may be either physical (due to factors like atomization, dispersion, mixing, or fuel characteristics) or chemical (due to factors like processes other than the combustion reaction, which are temperature and pressure dependent in the engine cylinder). Several important factors often affected ID fuel type, A/F ratio, CRs, fuel atomization rate, intake air temperature, ICP, and engine speed [27]. Since the introduction of gaseous fuel affected the physical and chemical process during the premixing processes and reduced the availability of O_2 content, ID was observed higher under DF mode [28]. When compared with air (1.4), biogas had a lower polytropic index (1.305), which reduced injection charge temperature and increased ID. In addition, CO_2 content within the air-fuel mixture lowered charge temperature, which caused ID increase and higher amount of fuel accumulation, leading to rapid burning rates during the premixed combustion phase. When considering the whole range of operation loads, ID also rose with increasing BFR [11]. The biogas induced by compression and intake stroke accumulated in a huge amount, which extended the ID and reduced ICT.

Several studies were examined to analyze the effect of CR on characteristics of diesel engine. The increase of CR and load led to the increase of flame speed, ICT, and pressure, while the ID and CD reduced to the lowest [14]. In addition, Verma et al. [29] found that the development of injection timing (IT) led to a rise in ID. As may be seen in Figure 5, the impact of IT is clear. However, the pre-ignition process was initiated by introducing more oxygen into the intake mixture port, leading to more efficient combustion and lower ID [15].

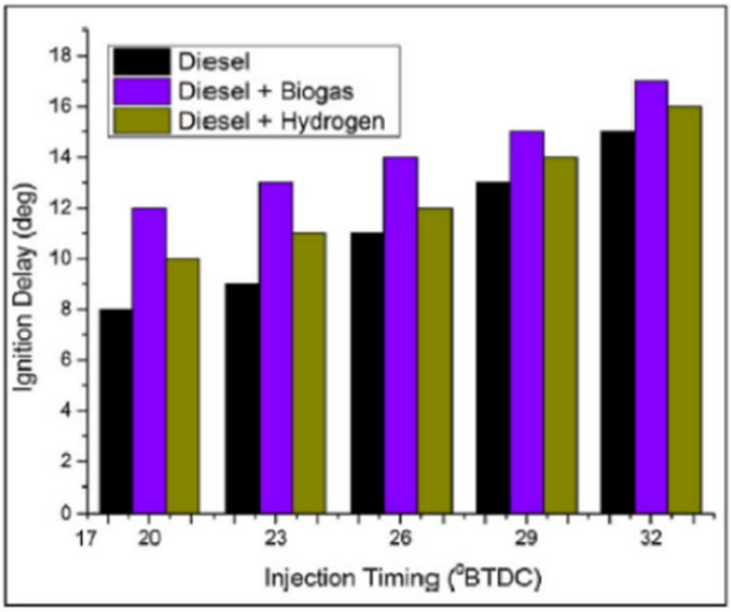

Figure 5. Relationship between ID and IT, reproduced from the study of Verma et al. [29]

Because DFM burned two different fuels in a single combustion chamber, it was somewhat complicated to calculate the heat release rate (HRR), which often influenced the ID, ICP, and premixed combustion phase. HRR was affected by the input charge mixture quality, mass flow rate of the engine medium gas, and fuel heating value [30]. Fuel accumulation during the premixed phase was caused by higher ID and the cooling influence of liquid fuel in the dual-fuel diesel engine (pilot diesel and gaseous fuel), which caused significantly higher HRR observed under DF mode [26]. Similarly, the delay of maximum heat release rate gain was caused by higher BFR due to increased carbon dioxide content, which had a slower combustion rate in the biogas. Increasing CR and advancing the injection timing (IT) of pilot fuel might raise the maximum HRR. One of the consequences of an increased CR was that the temperatures and pressures reached a high level after the compression stroke due to combustion efficiency improvement [31]. In addition, the lower energy content of biogas contributed to reducing the HRR and flame combustion temperature [32].

The time interval between 10% to 90% total injected fuel is consumed is known as the combustion duration (CD) in terms of the crank angle. The combustion period increased with the increase of BFR, because greater specific heat and lower flame velocity rate, which are the most important characteristics of biogas, contributed to a longer ignition delay [12]. The total combustion period increased as the load increased, which may be attributed to higher gaseous fuel consumption at full load and slow burning characteristic of biogas. However, as burning rate increased under operation condition that was somewhat close to stoichiometric, CD reduced and the equivalence ratio increased [33]. In addition, the reduced CD with DF mode biogas may be attributed to rapid burning rate and high energy release, which was linked to faster combustion process in pre-mixing stage [34].

2.2.2 Analysis of emission characteristics

CI engines cause a huge amount of pollution to the environment because they produce a great quantity of toxic gaseous pollutants. Nitrogen decomposition and burning under non-stoichiometric conditions are considered the primary causes of pollution. Nitrogen oxide emissions, including nitrogen monoxide (NO) and nitrogen dioxide (NO_2), are the most harmful emissions in nature, and must be reduced. They are products of general equation due to the N_2 and O_2 reaction in the combustion chamber at high temperature and may be extracted into three types: fuel, prompt, and thermal. Nitrogen dioxide oxidation created thermal NO_x emissions when the temperature reached a high level of 1800 K, and the oxidation rate increased drastically as the temperature rose [35]. The nitrogen oxides emitted as pollutants were typically produced from the combustion process due to oxidation of the N_2 component at a higher-level range. The reaction between the hydrocarbon molecules in the fuel and the nitrous oxide in the air was the primary source of the emitted nitrogen oxides, which represented a very small amount of the total NOx emissions formed during the process. Utilization of biogas in the combustion reaction process resulted in eliminating the oxygen content in the air-fuel mixture, which finally contributed to a considerable reduction in the emitted nitrogen oxide level. Increased carbon dioxide content and higher BGES lead to a decrease in NO_x emissions, due to higher biogas specific heat and lower combustion temperature that came from higher CO_2 content, which affected the global burning reaction and cooling nature of CO_2 emissions [33]. A few expert researchers showed that increasing the CR resulted in increased NO_x emissions because this was related to the increasing ICT under certain biogas DF mode [31]. In addition, NO_x emission rates increased significantly with the advanced start of injection (SOI) strategy due to homogeneity formed in air-fuel mixture [36]. Since the improvement caused by H_2 used as an additive for biogas fuel increased adiabatic combustion temperature and the speed of flame propagation, which led to an increase in NO_x emissions [37]. Variation of the NO_x with methane fraction, operation load and biogas flow rate (BFR) are demonstrated in subgraphs (a) and (b) of Figure 6.

Smoke emissions and soot generation reduced when biogas was introduced, because the biogas had higher diffusivity speed, a more homogeneous charge formed, and a huge amount of carbon fuels were replaced by biogas [38]. Improper rich mixture of air-fuel resulted in higher content of the PM and smoke emission attributed to incomplete combustion process due to insufficient oxygen available. The fuel had a low hydrogen-carbon (H/C) ratio and emitted more smoke. In comparison to biogas, diesel fuel had low hydrogen-to-carbon ratio and generated a significant amount of smoke and particulate matter, which was in stark contrast to the biogas DF engine system. The absence of aromatic compounds in biogas contributed to the reduction of smoke opacity from biogas under DF mode. However, there was a noticeable rise in smoke due to lower combustion temperature at higher BFR [39]. The effect of BFR on the smoke emission is shown in Figure 7. Use of biodiesel as a supplement fuel potentially further reduced smoke emissions due to the increased oxygen concentration in biodiesel, thus enhancing combustion [31]. The early start of injection (SOI) strategy also led to improvements in the first combustion phase and oxidation reaction rate inside the combustion chamber, as well as a reduction in the smoke emissions contents. In addition, higher CRs helped to reduce smoke emissions through a combination of two effects, namely, shorter ignition delay and higher combustion temperature, both of which resulted in enhanced combustion and oxidation reaction.

The lack of oxygen (O_2) required in the combustion chamber led to an increase in carbon monoxide (CO) because of the termination of completion of the combustion series. The dilution effect associated with increasing BFR resulted in spiked CO emissions [39] (Figure 8). The reduction in CO emissions was correlated with various parameters, such as operation load increasing, early SOI and elevated CR, due to the improved air-fuel mixture homogeneity, lower ID, and higher in-cylinder temperature [40]. Equivalence ratio increase resulted in richer mixture and higher speed flame propagation, leading to increased combustion rate, which in turn caused a decrease in carbon dioxide emissions. However, some experimental studies have shown that using EGR led to increased

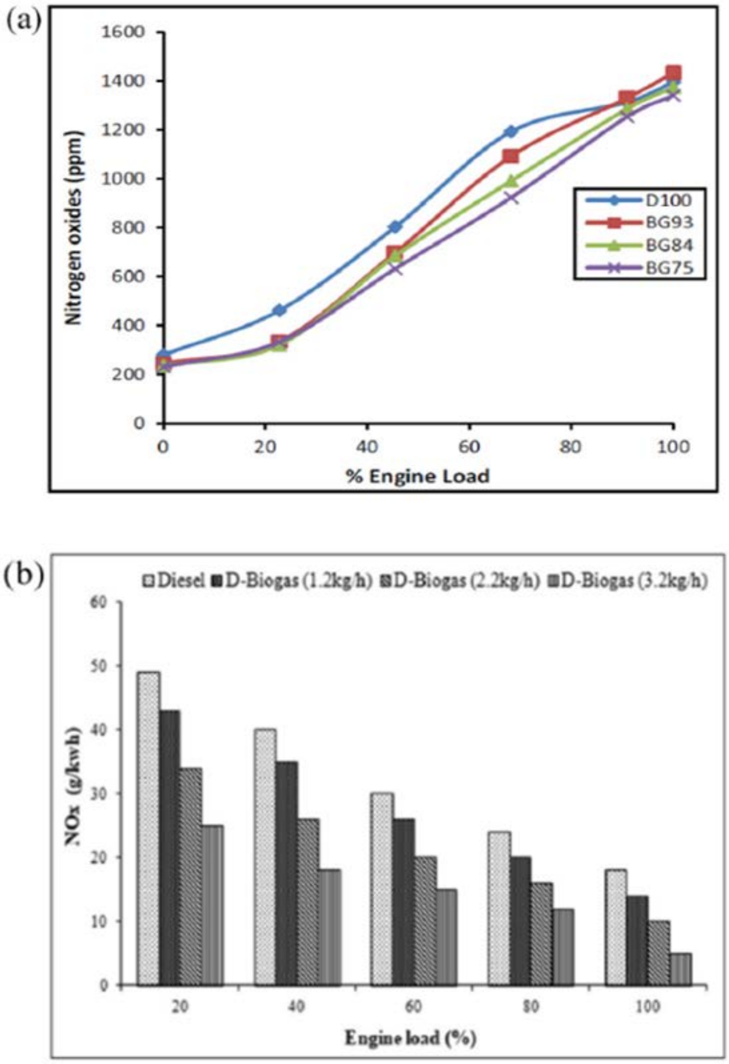

Figure 6. NO_x emissions with (a) CH_4 content and load, reproduced from the study of Verma et al. [9] with permission from Elsevier; (b) BFR and load, reproduced from the study of Mahla et al. [38]

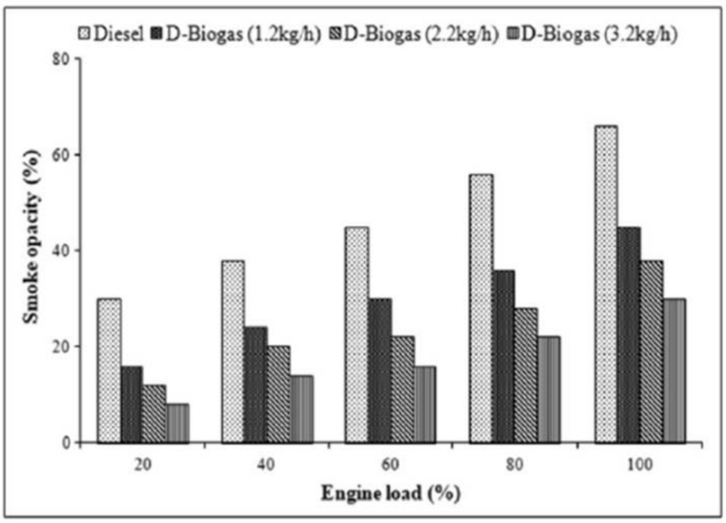

Figure 7. Variation of smoke emission with engine load, reproduced from the study of Mahla et al. [38]

carbon dioxide emissions because the O_2 existing in the air was replaced with the ratio of reaction-produced gases [41].

Unburned hydrocarbon (UHC) formation is relevant to the lack of combustion reaction completion. Some studies have shown that it is also associated with increasing biogas proportion in the baseline fuel. Combustion engine under DF mode produced higher HC emission, due to deficiency of O_2 induced through port charge intake, which was obvious sufficient with BFR increasing. Combustion engine under DF mode produced more HC emissions due to deficiency of O_2 induced through intake port charge and BFR increase also obviously played a significant role in the mitigating oxygen content [39]. In addition, the combustion process was poor in low load operation conditions due to the low-speed flame propagation through the cylinder engine, which led to higher HC emissions observed. Elevated load enhanced the combustion efficiency and reduced HC emissions [42]. The combined influence of increasing CRs and equivalence ratio under DF mode biogas resulted in reducing HC emissions due to shorter ignition delay (ID) and higher ICT in combustion process enhancement [40]. In addition, EGR contributed to reducing the combustion temperature, which led to increased emissions due to combustion efficiency [43]. Figure 9 shows the effect of the BP and BFR on the HC emissions.

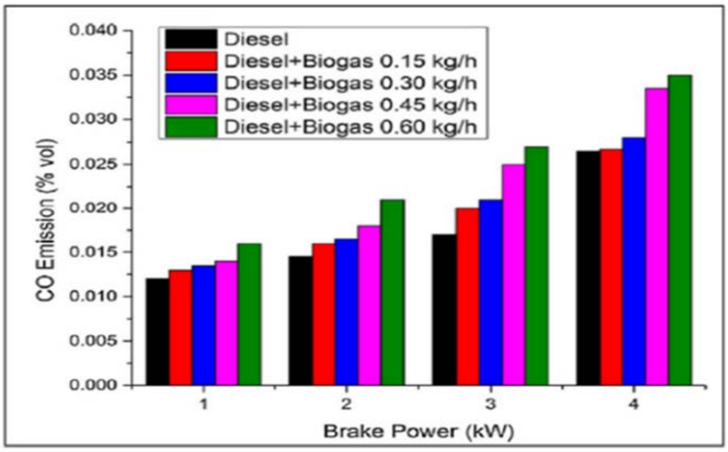

Figure 8. CO emissions as a function of BP and BFR, reproduced from the study of Barik and Sivalingam [39]

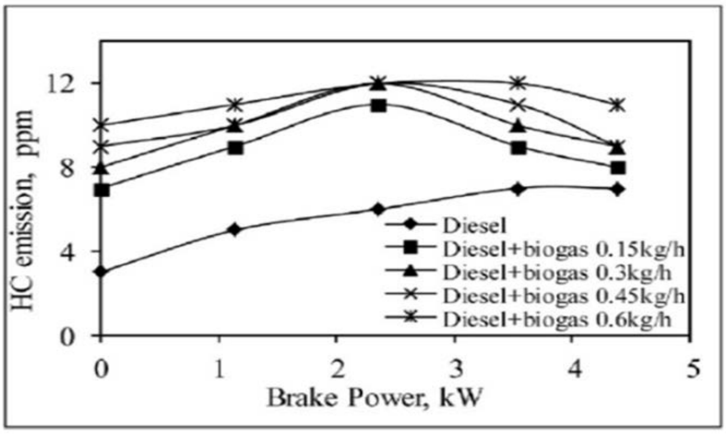

Figure 9. The effect of BP and BFR on the HC emissions [23]

3 Hydrogen as Fuel Supplement

As evidence that the industrial process is based on renewable energy, hydrogen (H_2) is considered to have the lowest carbon footprint and the highest efficiency production energy. The combustion of H_2 produced water vapour, and did not emit PM, CO or HC, because the carbon content was zero within the structure of the H_2 composition [44]. The auto ignition for H_2 did not occur easily in modern internal combustion engines throughout compression stroke, because H_2 required higher temperature to initiate the preignition and lower cetane number in comparison of diesel fuel. H_2 was utilized in CI engines with various injection strategy (manifold port, carburation and in-cylinder direct) and provided ignition source [45]. The diesel pilot fuel functioned as series of sequential ignition sources for the H_2 blends in DF mode, which had a potential over the step-edge of biogas and natural gas. Compared with the hydrocarbons fossil fuel, H_2 had higher flame propagation speed (FPS) and lower ignition energy even at lean conditions of air-fuel mixture. H_2 enhanced the performance of CI engines by increasing the absorption of fuel thermal energy of the CO_2 inert, which existed in the biogas throughout the combustion process. H_2 was used as an additive to the blends of biogas fuel to improve combustion and flame characteristics due to numerous physics parameters, such as high speed of flame propagation, and flammability range higher by ($4-74\%$) in comparison to biogas fuels [46]. Compared to methane, H_2 improved the range of flammability due to the lower Lewis number and higher speed of flame propagation [32]. Decreased irreversibly of the combustion reaction and enhanced CI engine performance attributed to increasing H_2 energy share percentage in Eq. (2) in the fuel blends of DF mode [47]. Verma et al. [9] investigated the influences of the HES variation ($5-20\%$) on baseline biogas fuel DF mode engine. In addition, the study showed that the H_2 energy share ($> 5\%$) helped to improve the combustion quality, which finally led to enhancing the characteristics of combustion and emissions. Some studies showed that increased H_2 energy share contributed to spiked NO_x emissions compared with baseline diesel fuel, due to increase of ICT caused by H_2 addition [48].

$$H_2\, energy\, share = \frac{\dot{m}_{\text{hydrogen}} \times LCV_{\text{hydrogen}}}{(\dot{m}_{\text{hydrogen}} \times LCV_{\text{hydrogen}}) \times (\dot{m}_{\text{diesel}} \times LCV_{\text{diesel}})} \times 100 \tag{2}$$

where, $\dot{m}$ represents the fuel mass flow rate (kg/s), $LCV_{\text{hydrogen}} = 120MJ/kg$, and $LCV_{\text{diesel}} = 43MJ/kg$.

3.1 Performance Analysis

The energy required for power generation in kilowatt is defined as brake thermal efficiency (BTE). The introduced energy was specified through the consumed and calorific value of the used fuel in the diesel engine Eq. (3).

$$BTE = \frac{BP}{(\dot{m}_{\text{hydrogen}} \times LCV_{\text{hydrogen}}) \times (\dot{m}_{\text{diesel}} \times LCV_{\text{diesel}})} \tag{3}$$

where, BP represents brake power (kw), and LCV represents the lower calorific value (KJ/kg).

Increase of H_2 energy share and the fuel flow rate led to the increase of BTE efficiency, by expanding the range of flammability limits, high speed diffusivity, the local fuel dilution, and enhancing the air-fuel mixing rate [49] (Figure 10). Saravanan and Nagarajan [50] investigated the effected of H_2 on the BTE, and showed that increased H_2 proportion had a positive impact on the BTE at the higher and medium load operation conditions, mainly because the local fuel equivalent ratio was high under high load operating conditions, which led to higher FPS due to the improved combustion characteristics [26].

Increased H_2 volume ratio at the expense of oxygen content in the air-fuel mixture led to improper combustion reaction and unevaluated effect on performance characteristics [51]. However, the addition of H_2 to the DF engine natural gas-diesel improved the BTE, because combustion reaction enhanced in the first steps of the pr-mixed stage [52]. The unique properties of H_2, such as higher FSP and heating capacity, and lower heating value, contributed to the significant BTE improvement in primary biogas fuel in DF mode [53]. For both diesel and biodiesel types of engine, the H_2 supplements sufficiently improved engine characteristics under diesel mode compared with biodiesel [54]. Some studies showed that lower BSFC was consistent with increased hydrogen flow rate (HFR), because higher H_2 diffusion led to more complete fuel combustion and fewer pollutants [10]. BSFC significantly increased at higher H_2 concentrations, because the reaction between fuel combustion and insufficient amount of oxygen required resulted in exhaust unburnt fuel [51]. In addition, the H_2 supply to the blends of NG-diesel in DF modes improved the BSFC due to enhancement of diesel engine characteristics and conversions of the potential energy [52]. The engine's breathing capacity was related to volumetric efficiency (VE), which decreased as the H_2 energy share increased, because higher diffusion displaced the majority of air through the intake stroke [55]. Moreover, higher temperature of the intake port system reduced the air density and caused a drastic drop in VE at the high load operation condition. H_2 injection through main port induction significantly affacted the VE of the diesel engine. The port injection technique improved VE, because discontinued H_2 supply into the core of air-fuel port induction provided better baseline diesel fuel substitution as well as air quantity [13]. The low heating value (LHV) of the fuel played a vital role in spiked in-cylinder gas temperature. The LHV to the H_2 (120 MJ/g) was higher than the diesel (42.5 MJ/g), contributing to increasing exhaust gas temperature (EGT) for diesel engine under DF mode [56]. Increase of H_2 energy share improved the combustion rate, which drived the reaction to complete, because the traffic air-fuel mixing process resulted in increased EGT [51, 57] (Figure 11). The in-cylinder gas temperature decreased with the exhaust gas recirculation (EGR) because increased specific heating value of the air-fuel mixture reduced the O_2 concentration in the combustion chamber.

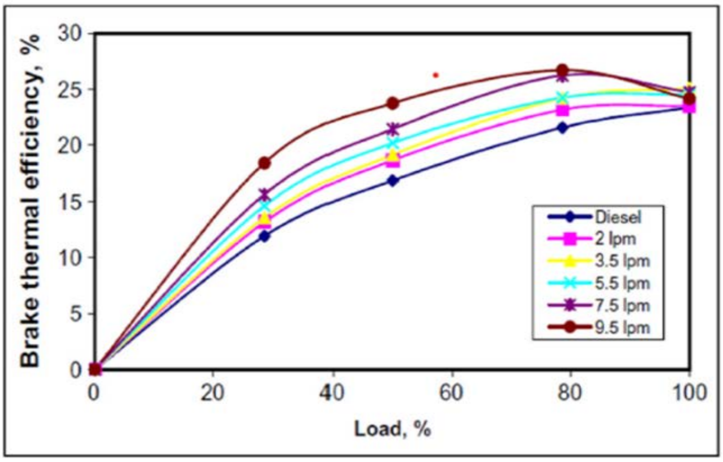

Figure 10. Effect of the load and H_2 fraction ratio on the BTE [49]

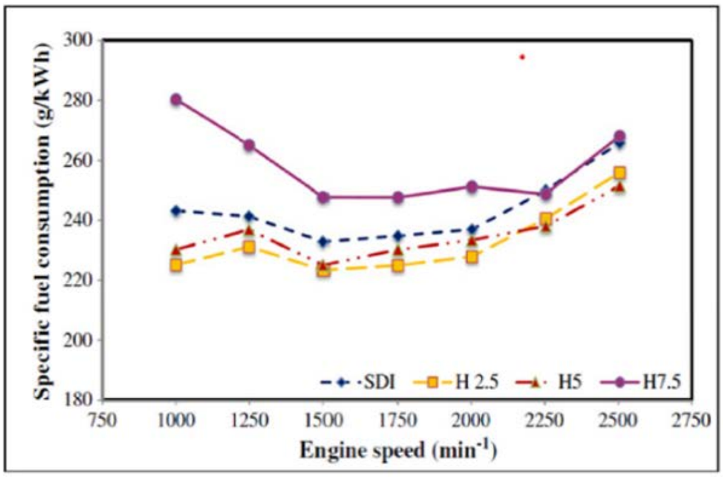

Figure 11. Variation of BSFC HES and engine speed, reproduced from the study of Köse and Ciniviz [51]

3.2 Analysis of Combustion Characteristics

Combustion in the diesel engine under the DF mode was very complex, because the fuels in the gas and liquid phases faced each other in different physical and chemical compositions. The H_2 injected through intake main port, mixed with induced air through section stroke till end of the compression stroke, which provided opportunity to ensure the mixture homogeneity. The main fuel of gaseous fuel needed to be used as ignition source to inject pilot diesel fuel at the end of compression stroke, thus igniting the compressed mixture of gaseous fuel. The current section discussed the combustion characteristics of diesel, such as combustion in-cylinder pressure (ICP), peak cylinder pressure (PCP), ignition delay (ID), heat release rate (HRR), and combustion duration (CD). All studies focused on PCP, which reflected the in-cylinder gas behaviors and combustion responding. Some studies showed that the PCP increased when H_2 was added to the blends of fuels for diesel engine under high and medium operation condition, while decreased at low load [58] (Figure 12). However, a slight of pilot diesel fuel was injected into the in-cylinder engine in the low load operations in order to initiate the primary ignition series required for the compressed air-fuel mixture. The centers of combustion reduced in this status and in-cylinder pressure attained lower level. Furthermore, feeding higher percentage of H_2 at the expense of less pilot diesel fuel to an engine running at low load resulted in further drop in in-cylinder pressure, because the lack of diesel-gaseous mixtures delayed combustion starts and extended ignition delay (ID) [58].

The benefit of adding H_2 to blends was higher flame speed propagation (FSP), which through combustion resulted in an immediate increase in PCP and HRR corresponding to higher fuel rate (HFR). Addition of various proportions of H_2 to the blends of biogas increased the PCP due to highly combustion ability. Rosha et al. [44] showed that addition of H to natural gas (NG) blends at higher loading operations increased the PCP.

Ignition Delay (ID) refers to the suspension in ignition of the air- fuel formulation. This period ends when the injection starts to initiate fuel combustion in crank angle (CA) degree. ID depends on various physical and

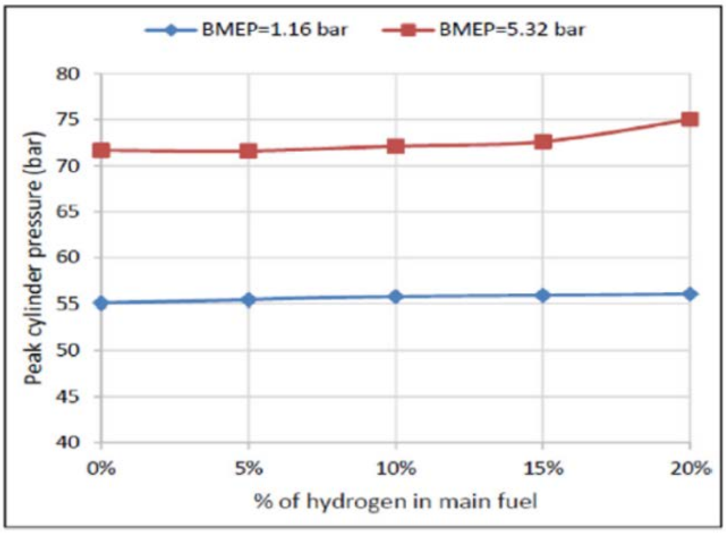

Figure 12. Relation between PCP and HES, reproduced from the study of Santoso et al. [58]

chemical parameters, such as fuel chemical formula composition, in-cylinder temperature, in-cylinder pressure as well as the fuel properties. When the diesel engine operated under high load conditions, the high heat energy supply (HES) contributed to shorter ID of the reverse with low and medium load [59]. The ID was prolonged at high load because the low O_2 during the section air stroke in the inlet manifold reduced the fuel reaction rate and delayed the auto-ignition event. Besides, as absorbed in Figure 13, the ID reduction in the diesel engine operating at high load, due to the high gas temperature and improved air-fuel mixture homogeneity of the pilot diesel fuel and blend of gaseous [60]. Moreover, increasing the share of H_2 energy under H_2-Diesel DF mode led to ID prolongation, because faster diffusion of hydrogen displaced fresh air at the inlet manifold port, causing the combustion reaction to suffer a deficiency of required O_2 [44].

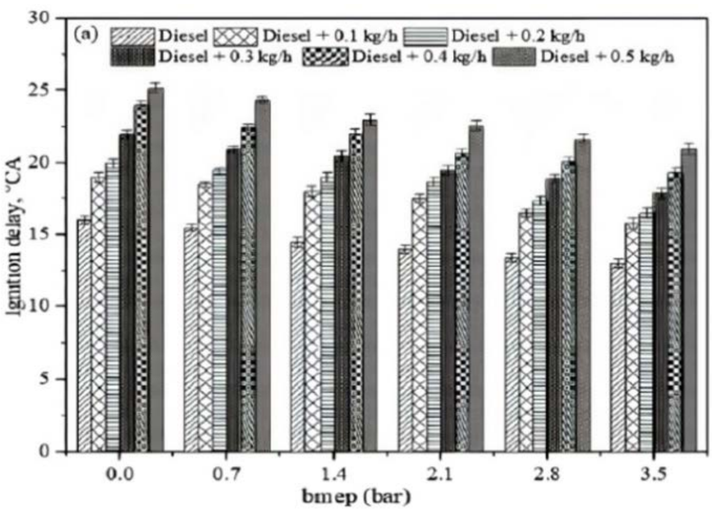

Figure 13. Variation of ID with HEB flow rate, reproduced from the study of Rosha et al. [44]

Heat release rate (HRR) may be divided to three stages. The first was precombustion stage, which released the main heat caused by burning the liquid pilot fuel and the main gaseous fuel (Figure 14). The second was combustion stage, where the pilot liquid of diesel fuel burned a large amount of gaseous fuel in the surrounding area of the ignition source. The third stage involved burning the remaining charge of the total fuel and diffusing the combustion until the end of the power cycle [61]. The addition of H_2 to the blends of gaseous fuel increased the combustion and released more HRR precisely at the premixed combustion phase, because higher flame speed propagation (FSP) enhanced the mixing rate [62]. Many studies showed that natural gas (NG) blends enriched with different energy shares of H_2 increased HRR, due to three parameters, such as higher FSP, improved homogeneity of gaseous fuel, and auto-ignition ability of the mixture [52].

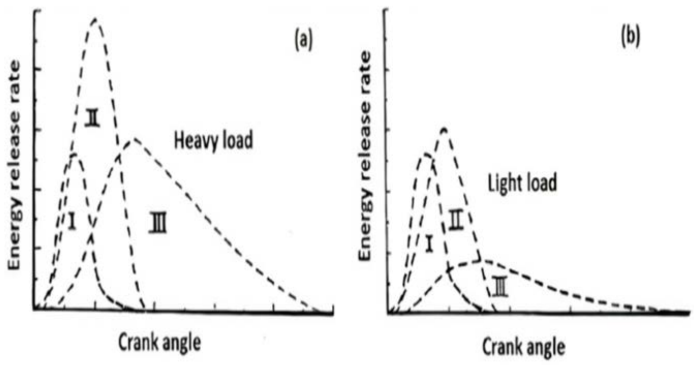

Figure 14. Various components of HRR in a DF engine under (a) high loading (b) low loading, reproduced from the study of Chintala and Subramanian [56]

The duration between the start and end of the combustion process per rotating crank angle (CA) is defined as combustion duration (CD). The CD decreased along with the heat energy supply (HES) due to higher amount of free radicals available, such as H, OH and O, which was one of the most important requirements for completion of combustion reaction [63].

3.3 Analysis of Emission Characteristics

Incomplete combustion reaction (partial oxidation) caused carbon monoxide (CO) emissions, because the lack of oxygen required for all the carbon-containing materials led to carbon dioxide ($(\mathrm{CO_2})$) emissions. The local air-fuel mixture and in-cylinder temperature played a vital role in the CO emissions. Introduction of $\mathrm{H_2}$ in intake main port under DF mode improved air-fuel mixture because H/C increase reduced emission-based carbon concentration [57]. Due to the reduction of carbon content in the fuel blend enriched by the $\mathrm{H_2}$, the CO, $(\mathrm{CO_2})$ and HC in the combustion chamber dropped [64].

Engines operating with dual fuel mode tended to have high NO_x level (oxide of nitrogen, such as NO and NO_2). NO_x formation could be propagated through combustion zone due to high in-cylinder temperature, oxygen concentration and available time for oxidation reaction. Although NO_x formation definitely had higher percentage within the bounded zone in an almost fuel-rich condition [65], hydrogen-based fuel operated within high flame adiabatic temperature, mainly because of the significant increase in NO_x emission level. Moreover, Liu and Karim [66] showed that NO_x increased along with in-cylinder temperature, which in return increased to the peak of NO_x emission at the combustion peak temperature. In addition, the study showed that major NO_x formation accrued at 20° crank angle degree (CDA) after initial combustion started. In addition, NO_2 emission calculation depended on kinetic chemical energy, which was closely related to flame front zone [67]. Under DF mode, $\mathrm{H_2}$ was used with very small modification of induction or direct injection with baseline diesel fuel in a CI engine. At the low and medium engine operation condition, the NO_x reduced while brake thermal efficiency increased.

4 Conclusions and Recommendations

Hydrogen and biogas are respected as an alternative fuel in power generation, because they meet the high energy requirements and sustainable development goals for future generations due to their renewable nature and do not contain environmental pollutants. Replacing fossil fuels with the proposed renewable fuels is very beneficial, which liberates the dependence IC engines on traditional fuels. The industrial use of these renewable fuels helped reduce overall pollution emissions and really contributes to creating a clean environment like the boiler. After reviewing the literature about the use of biogas and $\mathrm{H_2}$ as additive fuels in CI engines, the following conclusions were made and summarized as follows:

1) The energy required depended on the concentration of methane in the biogas, which was a mixture of combustible and non-combustible compositions.

2) Biogas fuel ratio and biogas fuel energy played a vital role in increasing BTE under biogas DF mode. Some strategies were developed in the operation system to enhance the BTE, such as advancing SOI, increasing CRs, and increasing the load.

3) All studies showed that combustion engine under DF mode increased PCP and prolonged ID due to higher $\mathrm{CO_2}$ concentration.

4) For the combustion engines working with biogas-DF mode, NO_x and some emissions tended to reduce due to the lower in-cylinder temperature and combustion efficiency.

5) H_2 was considered as effective alternative fuel for the traditional fossil fuel due to higher LHV and FSP.

6) Introduction of H_2 into various proportions of biogas fuel blend improved BTE, and reduced the combustion duration, because high FSP and higher diffusivity speed increased mixture homogeneity in high and medium load. While volumetric efficiency decreased due to lower H_2 density, which displaced the air in the intake port during section stroke.

7) When the H_2 energy share in the blends of biogas for the engine working under DF mode increased, carbon contents in the fuel mixture and in-cylinder temperature decreased. In addition, increase of higher low calorific value resulted in more NO_x emissions.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

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