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A review on natural gas/diesel dual fuel combustion, emissions and performance

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ABSTRACT

With the increasing concern regarding diesel engine emissions, including nitrogen oxides (NO_X) and particulate matter (PM), and the rising of energy demand as well, the utilization of alternative fuels in diesel engine has been found to be an attractive solution. Natural gas is a very promising and highly attractive fuel because of its domestic availability, widespread distribution infrastructure, low cost, and clean-burning qualities to be used as a transportation fuel. Natural gas/diesel dual fuel is an operation mode in which natural gas is introduced into the intake air upstream of the manifold and then ignited by the direct injected diesel in the cylinder. The aim of this paper is to identify the potential use of natural gas/diesel dual fuel on diesel engine. In this literature review, the combustion, emission and performance characteristics of natural gas/diesel dual fuel combustion-mode published mainly in scientific journals have been collected and critically analyzed. A wide range of natural gas mass ratio which represents the mass fraction of natural gas in the total fuel and different types of engines were involved. It has been found that dual fuel mode has a lower compression pressure and a longer ignition delay compared with normal diesel mode. The application of dual fuel mode significantly decreases the NO_X, carbon dioxide (CO_2) and PM emissions. However, the hydrocarbon (HC) and carbon monoxide (CO) emissions may increase by several times or even more than 100 times in comparison to normal diesel combustion. And there appears a trade-off relationship between NO_X and HC emissions with dual fuel mode. The engine power is decreased up to 2.1% at dual fuel mode, but the power loss can be reduced or recovered by changing some of the operating parameters. The brake thermal efficiency (BTE) of dual fuel mode is lower at low and intermediate loads, while under high engine load conditions it is similar or a little higher when compared with normal diesel mode, and the maximum increase is about 3%. The COV_{IMEP} seems to be generally higher than normal diesel mode and it decreases with the increasing engine load.

Contents

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1. Introduction

Diesel engines are widely used in the world due to their high combustion efficiency, reliability, adaptability and cost-effectiveness [1,2]. However, diesel engines are one of the major contributors to environmental pollutions [3,4]. The main harmful pollutants from diesel engines are NO_X and PM. NO_X emission is one of the major causes of photochemical smog. And it is also a cause of acid rain. Primary PM from diesel engines consists of various types of chemical components such as elemental carbon, organic carbon, inorganic ions, trace elements etc. [5–7]. These particles have extremely harmful effects on human health and environment. Numerous studies have proved that these particles cause respiratory and cardiovascular health problems [8–11] and neurodegenerative disorders [12,13]. Furthermore, the exhaust emissions of diesel engine have been identified as carcinogen by the World Health Organization in June 12, 2012 [14]. Therefore, emission regulations become increasingly stringent to reduce these harmful emissions. On the other hand, energy demand is increasing but the oil resources are diminishing. In order to ease the contradiction between the need for increased energy and the decreasing oil resources while at the same time reduce pollutant emissions, the utilization of alternative fuels has been found to be an attractive solution.

Among the various alternative fuels, natural gas is very promising and highly attractive in the transportation sector. Firstly, natural gas is available in several areas worldwide at encouraging prices. Beside the oil fields and natural gas fields, the natural gas industry is producing gas from increasingly more challenging resource types: sour gas, tight gas, shale gas, coal-bed methane, and methane gas hydrate [15]. Secondly, although the main component of natural gas, namely methane, is a greenhouse gas, natural gas still is an eco-friendly fuel. It can contribute to the reduction of CO_2 emission because it exhibits the lowest carbon-to-hydrogen ratio of all the fossil fuels. Natural gas can also substantially reduce the NO_X emission and at the same time produce almost zero smoke and PM [16–18]; which is extremely difficult to achieve in conventional diesel engines. But on the other hand, in order to avoid its own environmental pollution, we should try to reduce the leakage of natural gas. Thirdly, natural gas is not prone to knock due to its high methane number under normal circumstances. Therefore, it can be used in engines with relatively high compression ratio and obtain a higher thermal efficiency compared with that of normal gasoline engine.

Natural gas has been employed as a supplementary fuel widely in diesel engine for its economical and environmental benefits [19–23]. The main purpose of this study is to provide a comprehensive review of the literatures relate to the potential use of natural gas in diesel engine. In this literature review, a great variety of diesel engine sizes and types were researched at different operation conditions. Single cylinder direct injection research diesel engine was most frequently used and a wide range of natural gas mass ratio was involved. Combustion, emission and performance characteristics are discussed at different sections to get a clear understanding of the natural gas/diesel dual fuel engine.

2. Natural gas as an alternative fuel

The main component of natural gas is methane, which is the simplest hydrocarbon. The combustion of natural gas is clean and emits less CO_2 than almost all other petroleum-derivate fuels. Natural gas has been used to fuel vehicles since the 1930s [24].

2.1. Physicochemical properties of natural gas

Natural gas is a mixture of a variety of gases. It contains some kinds of lightweight alkanes, such as methane, ethane, propane, *n*-butane and isobutane, and pentanes. It may also contain carbon dioxide, nitrogen and trace amounts of water vapor. The composition and content of natural gas varies slightly depending on the source and the production process. The typical component and content of natural gas are listed in Table 1 [15]. Normally, methane accounts for 87–96% of natural gas. Therefore, the physicochemical properties of natural gas are very similar to methane. The properties of natural gas in comparison to diesel fuel and gasoline are given in Table 2 [25–28]. Natural gas is an environmentally friendly alternative fuel for transportation because it contains less carbon per unit of energy than any other fossil fuel and thus produces lower CO_2 emission per vehicle mile traveled. However, it is a little difficult for natural gas to be used in compression ignition engine for

Table 1
Typical component and content of natural gas [15].

Component	Typical analysis (vol.%)	Range (vol.%)
Methane	94.9	87.0–96.0
Ethane	2.5	1.8–5.1
Propane	0.2	0.1–1.5
Isobutane	0.03	0.01–0.3
n-Butane	0.03	0.01–0.3
Isopentane	0.01	Trace to 0.14
n-Pentane	0.01	Trace to 0.14
Hexane	0.01	Trace to 0.06
Nitrogen	1.6	1.3–5.6
Carbon dioxide	0.7	0.1–1.0
Oxygen	0.02	0.01–0.1
Hydrogen	Trace	Trace to 0.02

its high auto-ignition temperature. While natural gas is very suitable for spark ignition engine for its excellent anti-knock quality and it does not require any modification to the engine.

2.2. Natural gas used in diesel engine

The means of natural gas used in spark ignition engine are already well established, whereas its use in compression ignition engine is still under development. The usage of natural gas in diesel engine suffers from the poor ignition characteristics due to the high auto-ignition temperature and low cetane number compared with diesel fuel [29–31]. Therefore, the ignition source is always needed to ignite the natural gas in the cylinder. According to the way of natural gas into the cylinder and the ignition source, there are three main methods for applying natural gas in diesel engine. They are:

1. Dual fuel — in this mode, natural gas is inducted or injected in the intake manifold to mix uniformly with air and then is introduced to the cylinder and ignited by the direct injected fuels with high cetane number [32–39]. Fig. 1 shows the schematic diagram of dual fuel system.
2. High pressure direct injection (HPDI) — in this mode, a small amount of pilot diesel is firstly injected late in the compression stroke and then natural gas is directly injected. At some point during the time interval between the two injections or early in the natural gas injection, the diesel fuel auto-ignites, providing the ignition source to initiate the natural gas combustion [40–45]. Fig. 2 shows the schematic diagram of HPDI system.
3. Hot surface assisted compression ignition — in this mode, natural gas is injected directly into the cylinder close to a hot surface at the end of compression [46,47]. The hot surface is generally a glow plug with a temperature range of 1200–1400 K [48]. Fig. 3 shows the schematic diagram of this engine concept. The most important advantages of this concept are high specific power and thermal efficiency without the limitation of combustion knock. However, the critical element of the system is durability of the hot surface due to the high surface temperature required [47]. This mode is rarely used in recent years.

In HPDI mode, the directly injected natural gas burns in a predominantly non-premixed combustion [50]. This stratified combustion technique provides better fuel economy and more efficient combustion, and maintain the power output and the thermal efficiency of an equivalently-sized conventional diesel engine [51,52], especially at low and medium loads. However, a special concentric-needle, dual fuel injector [51] is needed for HPDI mode. The structure of the injector is more complicated and the cost is higher. In addition, the control difficulty increases. In contrast, only a low pressure gas induction or injection system is needed for dual fuel mode. The dual fuel mode is easier to implement, even in existing diesel engine without serious engine modification. Furthermore, more than 80% of diesel fuel can be reduced at dual fuel mode. The quantity of pilot diesel per cycle can usually be reduced to less than 10% of the total fuel amount to the engine [53]. In the studies of Krishnan et al. [54] and Srinivasan et al. [18] the pilot diesel only account for about 2–3% of the total injected energy and the natural gas substitution reached at 95%. For the above reasons, dual fuel mode has been widely researched all over the world [55–63].

From the above, it is clear that dual fuel mode is more practical compared with HPDI mode, so dual fuel mode is being considered to be one of the most promising ways to utilize natural gas in diesel engines.

Table 2
Physicochemical properties of natural gas, diesel and gasoline [25–28].

Fuel properties	Natural gas	Diesel	Gasoline
Low heating value (MJ/kg)	48.6	42.5	43.5
Heating value of stoichiometric mixture (MJ/kg)	2.67	2.79	2.78
Cetane number	–	52.1	13–17
Octane number	130	–	85–95
Auto-ignition temperature (°C)	650	180–220	310
Stoichiometric air–fuel ratio (kg/kg)	17.2	14.3	14.56
Carbon content (%)	75	87	85.5

3. Combustion characteristics

In-cylinder combustion of fuels is one of the most important processes which affect the formation of exhaust pollutants as well as the engine performance and durability [64]. Natural gas and diesel are two kinds of fuels with different physicochemical properties. The combustion of diesel in the cylinder is mainly the mixing controlled diffusion combustion [65], while natural gas is premixed combustion. The addition of natural gas into the cylinder affects the combustion characteristics due to the changes in the process of air/fuel mixture formation and combustion. The combustion characteristics with natural gas/diesel dual fuel were reviewed and summarized below.

3.1. Literature review on combustion characteristics

A few years ago, Selim et al. [66] experimentally investigated the effect of series of parameters on the in-cylinder pressure and pressure rise rate in a natural gas fuelled Ricardo E6 single cylinder indirect injection diesel engine. They observed that the maximum pressure rise rate decreased with the increase of engine speed but increased with the diesel injection timing advancing for both normal diesel and dual fuel mode. The reductions of maximum pressure rise rate for normal diesel and dual fuel mode were 2.57 bar/°CA and 3.65 bar/°CA respectively when engine speed increased by 900 rpm. At constant engine speed and with the increase of engine load, the maximum pressure rise rate with dual fuel operation increased obviously but for normal diesel mode it was nearly not affected. The peak in-cylinder pressure for the dual fuel mode was higher than that for normal diesel mode at all loads of a constant engine speed. They also found that as pilot diesel mass increase, the maximum pressure rise rate of the dual fuel mode first decreased and then started again to increase. In general, the dual fuel mode exhibited higher pressure rise rate compared to the normal diesel engine and the maximum increase could reach up to about 11.5 bar/°CA.

Papagiannakis et al. [67] analyzed the effect of natural gas mass ratio on combustion characteristics at three different engine loads and three engine speeds. Natural gas mass ratio in the tests was varied from 0 to over 80%. Results showed that the ignition delay and combustion duration were generally longer and peak in-cylinder pressure was lower under dual fuel operation compared to the ones under normal diesel operation for all test cases. As the natural gas mass ratio increasing, peak in-cylinder pressure decreased significantly, while the ignition delay and combustion duration increased.

Wannatong et al. [68] studied the combustion characteristics in a single cylinder natural gas/diesel dual fuel research engine. They found that the peak in-cylinder pressure was increased significantly and the ignition delay was shortened when the intake temperature

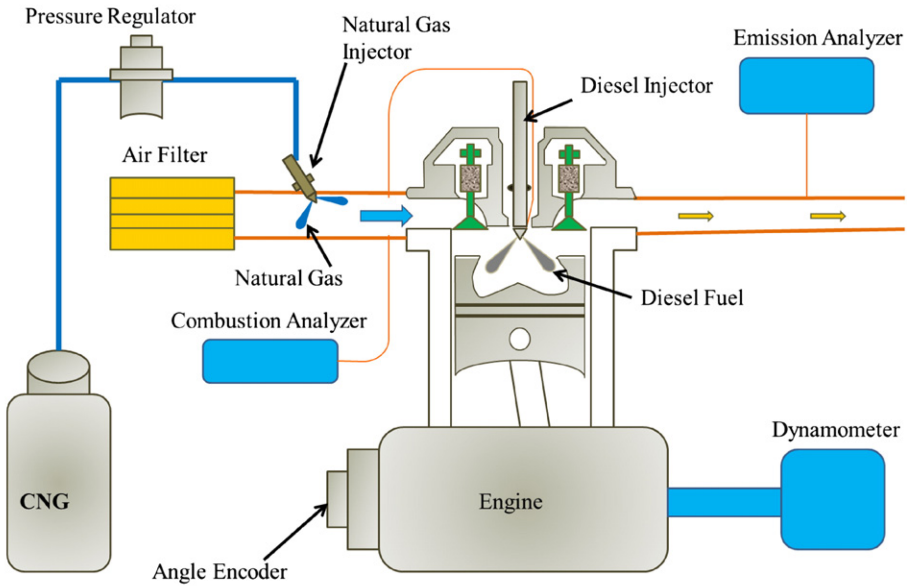

Fig. 1. The schematic diagram of dual fuel system. (CNG: compressed natural gas) [49].

was increased at constant engine speed, engine load and natural gas mixing ratio. And the heat release rate gradually transformed into a single peak form with the increase of intake temperature. However, the higher intake temperature might lead to the engine knock. When the pilot diesel quantity was kept constant and with increasing amount of natural gas supplied, the peak in-cylinder pressure and the second peak of heat release rate were increased significantly, while the ignition delay was slightly decreased.

Abdelaal et al. [17] compared the combustion characteristics of the conventional diesel mode and the natural gas/diesel dual fuel mode in a single cylinder diesel engine at 52% and 87% engine loads of 1600 rpm. The pilot diesel fuel was kept constant at 20% of the rated value and the rest power output was contributed by the intake sucked natural gas. They found that the in-cylinder pressure with dual fuel mode was 6.7 bar and 6.2 bar lower, the ignition delay was 2.8°CA and 5.5°CA longer, the maximum pressure rise rate was 0.39 bar/°CA and 1.14 bar/°CA lower at 52% and 87% engine loads respectively when compared with conventional diesel mode. In addition, the dual fuel mode exhibited lower peak values of heat release rate. They also reported the application of exhaust gas recirculation (EGR) to dual-fuel mode additionally decreased the in-cylinder pressure and increased the ignition delay, and with EGR percentage increasing the effect was enlarged. Lower in-cylinder pressure and lower maximum pressure rise rate with introduction of natural gas were also reported by Liu et al. [69]. There was a reduction of 10 bar for in-cylinder pressure and 2.3 bar/°CA for pressure rise rate, respectively.

The effects of natural gas dual fuel operation on in-cylinder pressure and heat release were studied by Lounici et al. [70] and Papagiannakis et al. [71]. They both found that the in-cylinder pressure during the compression stroke and the initial periods of combustion was slight lower for dual fuel mode. But as to the peak in-cylinder pressure their results were somewhat different. In Papagiannakis's study, the peak in-cylinder pressure of dual fuel operation was always less than the normal diesel mode, whether at low or high engine load. Lounici et al. also reported the lower peak in-cylinder pressure corresponding to dual fuel operation at low engine loads, but at high engine loads, the peak in-cylinder pressure with dual fuel mode became higher than that of normal diesel mode. It was the consequence of an improvement in the gaseous fuel combustion and hence a higher heat release rate under dual fuel mode.

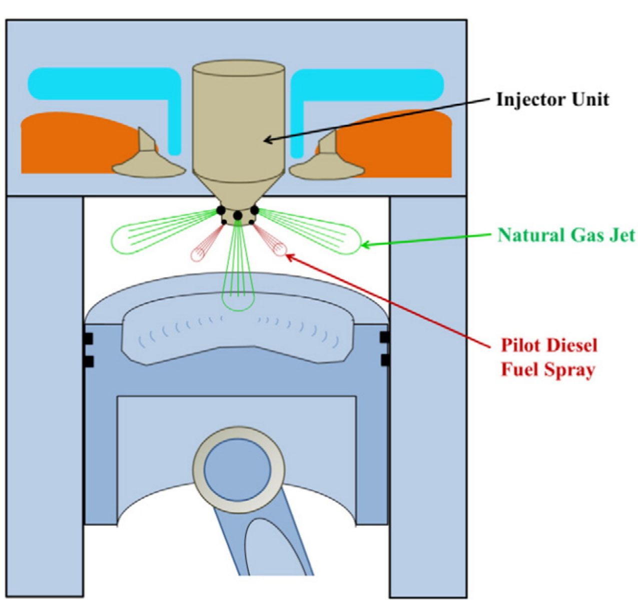

Fig. 2. The schematic diagram of high pressure direct injection system.

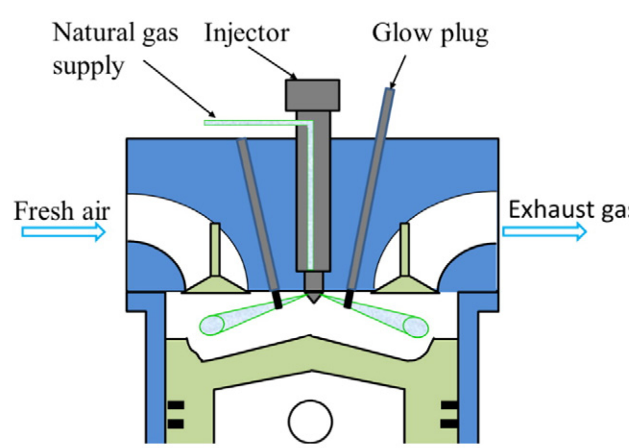

Fig. 3. The schematic diagram of hot surface assisted compression ignition.

Imran et al. [72] investigated the effect of natural gas addition and pilot diesel quantity on the combustion characteristics in a single-cylinder compression ignition engine. They found that the ignition delay was prolonged with the introduction of natural gas. And as the quantity of pilot diesel increasing, the ignition delay was generally shortened and the peak in-cylinder pressure increased slowly. The phase of heat release rate was also influenced by the quantity of pilot diesel. The highest peak of heat release rate was shifted in proportion to the ignition delay. In addition, the third peak in the heat release rate curve at lower pilot diesel condition indicated the after-burning of natural gas.

Recently, Sun. et al. [49] analyzed the effects of the pilot diesel quantity and injection timing on the combustion performance in a natural gas/diesel dual fuel engine. The results showed that the peak in-cylinder pressure and maximum pressure rise rate increased with the increase of pilot diesel quantity. And with earlier pilot diesel injection timing the peak in-cylinder pressure and maximum pressure rise rate were higher and appeared earlier. There were two peaks in the heat release rate curve. The first peak which corresponded to combustion of the pilot diesel increased and appeared earlier with increasing pilot diesel quantity, while the second peak which corresponded to combustion of the whole charge appeared at around 10 °CA after top dead center (ATDC). Zhou et al. [73] also investigated the pilot diesel quantity and injection timing on the combustion characteristics. They found that the ignition delay was prolonged with advance of the injection timing and was shortened by increasing the pilot diesel quantity.

Yang et al. [26,27] conducted parametric investigation of natural gas port injection and diesel pilot injection on the combustion characteristics in a turbocharged common rail dual-fuel engine. They found that under low and part engine loads retarding natural gas injection timing could enhance flame propagation and improve natural gas combustion efficiency. Moreover, better combustion performance, such as shorter ignition delay and combustion duration, higher brake thermal efficiency could be achieved under higher pilot injection pressure conditions at low engine loads. However, under high engine loads a negative impact on the combustion performance was observed by retarding natural gas injection timing.

3.2. Analysis of the trend of combustion characteristics

In-cylinder pressure, peak pressure, ignition delay and heat release rate are important parameters which can signify the combustion process. In the following sections these information are summarized and analyzed based on the above literature review.

3.2.1. In-cylinder pressure

In-cylinder pressure can be measured directly using piezoelectric transducer and many other combustion parameters such as maximum in-cylinder pressure, maximum pressure rise rate, *P-V* diagram, indicated mean effective pressure, heat release rate and so on can be calculated from the in-cylinder pressure [74]. Therefore, the in-cylinder pressure is a very valuable source of information during engine performance analyzing.

The in-cylinder pressure under dual fuel operation condition was obtained and compared with normal diesel mode by Abdelaal et al. [17], Papagiannakis et al. [67], Imran et al. [72], and many other authors mentioned above. The result showed that the in-cylinder pressure was slight lower for dual fuel mode during the compression stroke and the initial periods of combustion. Moreover, the start point of burning and the combustion phase were put off due to the longer ignition delay, as shown in Fig. 4 [67].

Under dual fuel operation conditions, natural gas is introduced at intake stroke and then is compressed together with fresh air at compression stroke. Because the specific heat capacity ratio of natural gas is much higher than that of air. The in-cylinder temperature of dual fuel mode is lower during the compression stroke. Therefore, the compression pressure is lower compared to normal diesel mode. While during the initial stages of combustion, the slower combustion rate of natural gas is responsible for the lower pressure under dual fuel mode.

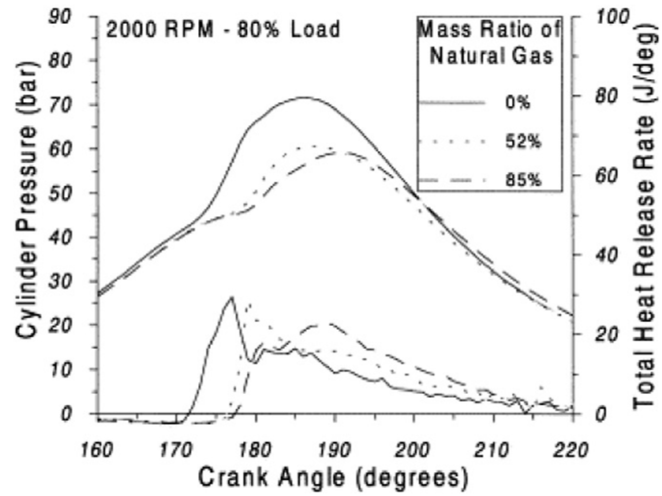

Fig. 4. Cylinder pressure and heat release rate under normal diesel and dual fuel operation [67].

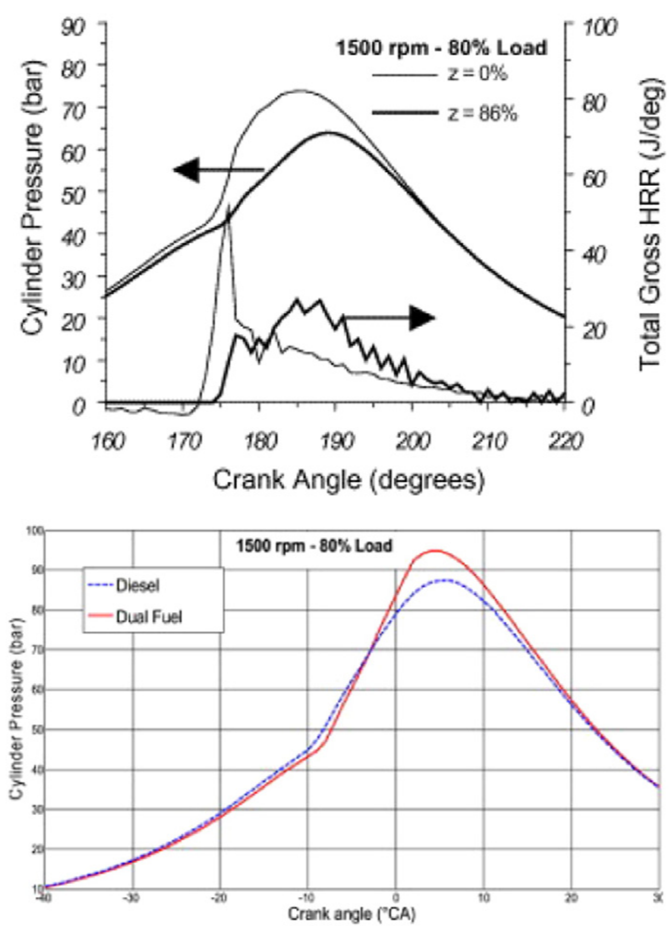

Fig. 5. Comparison of the in-cylinder pressure under two different operation modes. (a) Lower peak in-cylinder pressure under dual fuel mode [71]. (Z is the mass fraction of natural gas in the total fuel). (b) Higher peak in-cylinder pressure under dual fuel mode [70].

3.2.2. Peak in-cylinder pressure and pressure rise rate

The peak in-cylinder pressure and pressure rise rate are parameters closely related with the engine noise, vibration and service life. From the above literature review, it is clear that most authors found that peak in-cylinder pressure and pressure rise rate under dual fuel mode were lower in comparison with those of normal diesel mode, as shown in Fig. 5(a) [71]. The following reasons can be attributed for the decrease of peak in-cylinder pressure and pressure rise rate:

(1) The premixed combustion in dual-fuel mode suffers from very lean mixture and slower burning rate. These aspects have negative effect on combustion efficiency, and consequently, result in lower peak in-cylinder pressure and pressure rise rate.
(2) The longer ignition delay of dual fuel mode causes the whole combustion process to be shifted further into the expansion stroke. The increasing of the combustion chamber volume caused by downward moving of the piston in the expansion stroke moderates the pressure rising and hence results in lower peak in-cylinder pressure and pressure rise rate.

However, there are also some authors reported the higher peak in-cylinder pressure and pressure rise rate at dual fuel operation conditions, as shown in Fig. 5(b) [70]. The reason for the increase can be attributed to the rapid heat releasing of premixed mixture near the TDC, as shown in Fig. 6 [70].

In fact, the peak in-cylinder pressure and pressure rise rate under dual fuel mode both have a great relationship with the injection parameters of pilot diesel, intake temperature and so on. The authors all reported that with the increase of pilot quantity and the advance of pilot injection timing, the peak in-cylinder pressure and pressure rise rate increased. The reason is that, the higher pilot diesel quantity supplies a larger ignition source and greater ignition power which increases the heat release rate and hence results in the higher peak in-cylinder pressure. And with an advance in the pilot diesel injection timing, more energy is released during the compression stroke, which results in a higher temperature and hence a higher cylinder pressure. Fig. 7 [49] shows the increasing peak in-cylinder pressures with the advance of pilot injection timing.

3.2.3. Ignition delay

The ignition delay in a diesel engine is defined as the crank angle interval between the start of injection and the start of combustion. A series of physical and chemical processes occur during the ignition delay before the combustion. Based on the above literatures review, it is clear that authors all reported the longer ignition delay at dual fuel operation conditions compared with the normal diesel mode. The following reasons can be attributed for the longer ignition delay:

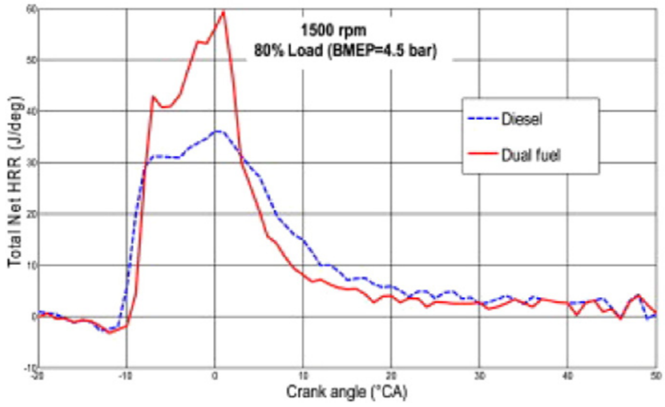

Fig. 6. Rapid heat releasing of the dual fuel mode [70].

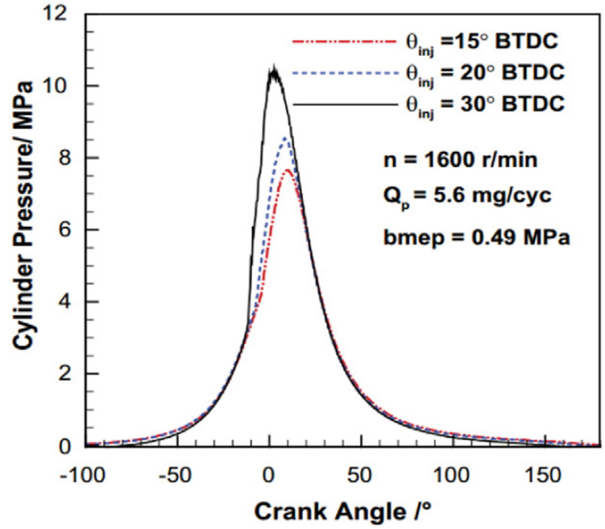

Fig. 7. In-cylinder pressures at different pilot diesel injection timing [49].

(1) The decrease in charge temperature, the decrease in partial pressure of oxygen and the absorption of some of the pre-ignition energy release which result from the higher specific heat capacity of natural gas lead to the extension of the ignition delay.
(2) Natural gas can suppress the auto-ignition of diesel because of the coupling effect amount the free radicals, resulting an increase in ignition delay. In fact, the major extension of the ignition delay is due to the chemical factors [75].

3.2.4. Heat release rate

The combustion of conventional diesel engines generally can be divided into four phases: (1) ignition delay, (2) premixed or rapid combustion phase, (3) mixing controlled combustion phase, (4) late combustion phase [65]. Most fuel is consumed at the mixing controlled combustion phase during the conventional diesel combustion. However, under dual fuel combustion mode, most of the diesel is replaced by natural gas and the ignition delay is longer, as a consequence, there is little or no mixing controlled combustion. Thus, the mixing controlled combustion phase is instead by the fame propagation combustion of natural gas at dual fuel combustion mode.

From the above literature review, it is clear that the combustion process of dual fuel mode is much different compared with the normal diesel mode, as shown in Fig. 4 [67]. At the second phase of dual fuel combustion, premixed pilot diesel starts to burn and ignites the natural gas. Because of the less quantity of pilot diesel and the low concentration of natural gas/air mixture, the heat release rate is lower than that of normal diesel mode. At the third phase of dual fuel combustion, due to the slower combustion rate of the gaseous fuel compared to that of diesel fuel and later ignition, the peak of heat release rate is usually not too high and the combustion duration is prolonged, as shown in Fig. 4 [67]. As a result, more fuel is burned in the fourth phase which may result in higher exhaust temperature. However, the premixed combustion of natural gas may be strengthened due to the increase of mixture concentration, the advance of pilot injection timing and so on, as shown in Fig. 6 [70].

4. Emission characteristics

Natural gas is considered to be a good alternative fuel for its good environmental effect. Numerous studies have been conducted by researchers around the world to explore the improvement of natural gas on the engine emissions. These investigations were carried out in different test engines and operation conditions. The emission characteristics with natural gas/diesel dual fuel were reviewed and summarized below.

4.1. Nitrogen oxides (NO_X)

NO_X is one of the most detrimental emissions from diesel engine and it is a grouped emission composed of nitrogen monoxide (NO) and nitrogen dioxide (NO_2). NO is the main component and usually accounts for more than 90% of NO_X emission inside the engine cylinder. The formation of NO in the combustion zone is complex chemically and two typical mechanisms are involved in, namely, thermal mechanism (Zeldovich mechanism) and prompt mechanism (Fenimore mechanism). According to thermal mechanism, the formation of thermal NO is greatly influenced by in-cylinder temperature and oxygen concentration. NO formation occurs when temperature is above about 1800 K and the formation rate increases exponentially with the increase of in-cylinder temperature [65,76]. According to prompt mechanism, the formation of prompt NO is led by the intermediate hydrocarbon fragments from fuel combustion – particularly CH and CH_2 – reacting with N_2 in the combustion chamber and the resulting C-N containing species then proceed through reaction pathways involving O_2 to produce NO [77]. The prompt mechanism is summarized as follows [78–80]:

$$CH + N_2 \rightarrow HCN + N$$

$$CH_2 + N_2 \rightarrow HCN + NH$$

$$HCN + OH \rightarrow CN + H_2O$$

$$CN + O_2 \rightarrow NCO + O$$

$$HCN + O \rightarrow NCO + H$$

$$NCO + O \rightarrow NO + CO$$

$$NCO + OH \rightarrow NO + CHO$$

The prompt NO is only prevalent under fuel rich conditions, where a reasonable amount of hydrocarbon fragments is available to react with N_2. The prompt NO possesses relatively weak temperature dependence in comparison to thermal NO [78,80]. Under most diesel engine combustion conditions, thermal mechanism is believe to be the predominant contributor to total NO_X [81–83].

4.1.1. Effect of natural gas/diesel dual fuel combustion on NO_X emission

Abd Alla et al. [84,85] analyzed the effects of the pilot diesel quantity and injection timing on the NO_X emission in a natural gas/diesel dual fuel engine. They found that the NO_X emission increased with increasing pilot diesel quantity at both low and higher engine loads. And with the advance of pilot diesel injection timing, NO_X emission also increased, which was consistent with the result of Singh et al. [86] since the pilot diesel injection timings were all after 30° CA BTDC (before top dead center).

Krishnan et al. [54] investigated the strategies for reducing NO_X emission in a single cylinder pilot-ignited natural gas engine at about ten years ago. Much lower NO_X emission was observed when small diesel pilots (2–3% on an energy basis) were used to ignite homogeneous natural gas–air mixtures compared to that of the baseline diesel. They further studied the effects of pilot injection timing, intake charge pressure and charge temperature on NO_X emission with natural gas fueling. They found full-load engine-out brake specific NO_X emission could be reduced to the range of 0.07–0.10 g/kwh from the baseline diesel value of 10.5 g/kW·h with appropriate control of the above variables. The same engine fueled with natural gas was also used by Singh et al. [86], and the effect of various operation conditions on the NO_X emission was studied in detail. The result showed that NO_X emission firstly increased and then decreased with pilot injection timing advanced from −15 to −60°CA ATDC。The injection timing corresponding to the maximum NO_X emission was nearby −35°CA ATDC. They found that operation of the engine at advanced timings could obtain low NO_X emission and high fuel conversion efficiency at the same time. However, increasing the equivalence ratio through an intake pressure adjustment and increasing the pilot quantity both resulted in the increase of NO_X emission.

The comparison of NO_X emission for normal diesel and dual fuel mode was also conducted by Papagiannakis et al. [71] at two engine speeds and four different engine loads varying from 20% to 80% of full load in a naturally aspirated single cylinder diesel engine. They reported that under all the operation loads, NO_X emission of dual fuel mode was lower than those of normal diesel mode. With the increase of engine loads the difference between the two modes was enlarged. At higher load, NO_X emission under dual fuel operation was considerably lower in comparison with those of normal diesel mode. And the further decrease of NO_X emission at dual fuel mode was observed with the increasing engine speed compared to normal diesel operation. Papagiannakis et al. [67,87] also investigated the effects of natural gas mass ratio on NO_X emission. They found that the increase of natural gas mass ratio resulted in a lower NO emission compared with the one of normal diesel mode.

Egusquiza et al. [88] investigated the effect of natural gas addition on NO_X emission in a 4-cylinder direct injection diesel engine. Five different engine loads from BMEP = 1.2 bar to 12.7 bar at the speeds of 1600 rpm and 2600 rpm were selected, and the substitution ratios of natural gas to diesel fuel were varied from 0 to the maximum value of the each selected operation conditions. Their results showed that NO_X emission was strongly dependent on the engine load and substitution ratio. NO_X emission reduced with the increase of substitution ratio over a wide range of engine loads at the both operation speeds. While at higher engine loads, NO_X emission showed a contrast tendency. For both diesel and dual fuel operation conditions, NO_X emission increased with the increase of engine load and showed a slight decline as engine speed increasing.

Cheenkachorn et al. [89] compared the NO_X emission of normal diesel operation and natural gas-diesel dual fuel operation in a heavy-duty diesel engine operating with the maximum portion of natural gas at full load of a wide engine speeds range. The effect of engine speed on the NO_X emission was also investigated. They reported that the NO_X emission reduced significantly compared to the single diesel operation due to the lower combustion temperature. The results also showed the NO_X emission slightly decreased with the engine speed for both diesel and dual fuel operations.

Liu et al. [90] conducted experiment in a CNG-diesel dual fuel engine and investigated the effect of pilot diesel fuel quantity on NO_X emission under three different engine speeds. They reported that NO_X emission of dual fuel mode was reduced by 30% averagely in comparison to those of diesel mode. That was because most of the fuel was burned under lean premixed conditions which resulted in lower local temperature. With the increase of engine speed, the NO_X emission reduced due to less residence time in high temperature.

Imran et al. [24] conducted experiment over a wide range of engine speed and load and obtained the maps of NO_X emission at both normal diesel mode and natural gas dual fuel mode in a single-cylinder compression ignition engine. They reported significant reduction in NO_X emission with dual fuel mode compared to normal diesel mode which might be contributed to the higher specific heat capacity of natural gas reducing the in-cylinder temperature. Since the in-cylinder temperature at lower power outputs was relatively lower and the addition of natural gas might further reduce the temperature. Therefore, results showed NO_X reduction was more significant and ranged between 40% and 53% at lower power outputs.

NO_X emission of dual fuel operation with natural gas was investigated by Lounici et al. [70]. The tests were conducted at four different engine speeds and engine loads from 20% to 80% of full load in a modified single cylinder diesel engine. Results showed that NO_X emission of dual fuel mode was lower at low and medium loads for all the four engine speeds compared with normal diesel operation. However, when the engine loads increased, NO_X emission with dual fuel operation increased rapidly and became higher than those of normal diesel mode. The authors attributed the higher NO_X emission to higher charge temperature and higher oxygen concentration of the dual fuel mode.

4.1.2. Summary

Based on the literatures review above, variation in results has been found since most authors reported lower NO_X emission with natural gas/diesel dual fuel combustion compared to normal diesel operation and some authors reported higher NO_X emission at high engine load. They also reported that NO_X emission increased with increasing pilot diesel quantity and engine load and decreased with the increase of engine speed at dual fuel mode. The reduction of NO_X emission can be explained by attributing the following points:

(1) The specific heat capacity ratio of natural gas is significantly higher than that of air. The addition of natural gas increases the overall heat capacity of the in-cylinder mixture which results in a reduction of the mean temperature at the end of compression stroke and during the overall combustion process. The lower combustion temperature leads to the reduction of NO_X formation. The effect is more evident at lower engine loads since the combustion temperature is already very low at low loads.
(2) The longer ignition delay of natural gas/diesel dual fuel combustion and the poor quality of natural gas combustion caused by lean premixed condition will reduce the combustion temperature, resulting reduction in NO_X emission.
(3) The introduction of natural gas reduces the amount of air and concentration of oxygen in the cylinder charge which might have a negative effect on the oxygen available for NO_X formation, and hence reduces the NO_X emission.
(4) With the increase of engine speed, there is less residence time for NO_X formation, and hence NO_X emission reduces.

The following reasons can be attributed for the increase of NO_X emission:

(1) Greater intensity of heat release in the premixed combustion stage caused by improvement of the natural gas combustion increases the maximum combustion temperature which might increase the NO_X emission.
(2) A better combustion of fuels forms less CO emission and consumes less oxygen. So more oxygen might be available for NO_X formation and increase the NO_X emission.

4.2. Carbon monoxide (CO)

CO is another harmful emission from the engine and its formation is a function of in-complete burned fuel availability and in-cylinder combustion temperature, both of which control the rate of fuel decomposition and oxidation [65,91]. Higher CO is usually generated in the over rich region due to the lack of oxygen. However, large amount of CO can be also produced in the fuel lean region when the combustion temperature is less than 1450 K [92].

4.2.1. Effect of natural gas/diesel duel fuel combustion on CO emission

Abd Alla et al. [84,85] analyzed the effect of pilot diesel quantity and injection timing on the CO emission of a special single cylinder compression ignition research engine fueled with natural gas. They reported that the CO emission reduced both with the increase of pilot diesel quantity and the advance of injection timing. In their opinion, the reductions of CO emission were both due to general improvement in the combustion process.

Significantly higher CO emission was also observed by Papagiannakis et al. [87] in a high speed diesel engine operating in natural gas/diesel dual fuel. They reported that the increase of natural gas mass ratio, which was accompanied with a reduction of the total relative air–fuel ratio, favored the CO formation mechanism. At low and intermediate loads, the effect became more evident in comparison to the one at high load. With the increase of natural gas mass ratio, the increase of CO emission at low and intermediate loads under dual fuel operation was varied from up to 190% to up to 600%. While at high engine loads, a slight decrease of CO emission was observed with the increasing natural gas mass ratio.

Egusquiza et al. [88] analyzed the CO emission in a 4-cylinder direct injection diesel engine under various substitution ratios of natural gas to diesel and engine loads at the speeds of 1600 rpm and 2600 rpm. They reported that CO emission was remarkably higher under dual fuel operation than that of diesel operation. With the increase of substitution ratio, CO emission firstly increased and then followed by a sudden decrease as the substitution ratio between 60% and 70%. In their study, the effect of engine speed was also evaluated, and it seemed to have no significant effect on CO emission.

Gatts et al. [93] investigated the effect of natural gas addition and engine load on the CO emission at different engine loads of a constant speed of 1200 rpm in a heavy-duty diesel engine. They found that CO emission increased almost linearly with the increasing addition of natural gas until the maximum CO emission was observed at the condition of about 2.5% natural gas added in the inlet gas. Then CO emission started to reduce with the further increase of natural gas. From their results, it was clear that engine load also had significant effect on CO emission. It seemed to emit less CO emission at higher engine loads.

Cheenkachorn et al. [89] investigated the CO emission at a varied engine speed of 1100 rpm to 2000 rpm in a heavy-duty diesel engine with natural gas/diesel dual fuel. They found that the dual fuel operation showed significantly high CO emission for all of the engine speed ranges tested compared to normal diesel fuel operation. With the engine speed increasing from 1100 rpm to 1500 rpm, the CO emission decreased, while above the engine speed of 1500 rpm, the CO emission increased with engine speed.

Liu et al. [90] experimentally studied the CO emission of a CNG-diesel dual fuel engine with different pilot diesel fuel quantity and optimized pilot injection timing. The results showed that the CO emission under dual fuel mode was considerably higher than that under normal diesel mode even at high load. They attributed this phenomenon to the existence of some flam extinction regions which indicated that most of the CO emission was from the incomplete oxidation of the premixed CNG.

4.2.2. Summary

Based on the above literature review, it is clear that all the authors reported obviously increase of CO emission under natural gas/diesel dual fuel combustion mode in comparison with normal diesel mode. They also reported that CO emission increased with the increasing natural gas mass ratio in a certain extent and with the further increase of natural gas CO emission started to decrease. The following reasons can be attributed to increase the CO emission in natural gas/diesel dual fuel combustion:

(1) Natural gas air mixture gets trapped in the crevices, deposits and quench layer due to the long time stay in the cylinder. The trapped natural gas air mixture is released and cannot be completely oxidized because of the low temperature during the expansion stroke. And with the increase of natural gas, the effect is enhanced. So CO emission increases remarkably with the increasing natural gas mass ratio in a certain extent.
(2) Bulk quenching because of mixing of hotter post-combustion gases with cooler surroundings may lead to CO formation.
(3) Natural gas air mixture is ignited by the pilot diesel under dual fuel operation and the flame has to propagate through the charge. The mixture in some region is too lean to sustain the flame propagation. For this, the local temperature falls and the reactions of CO oxidation freezes. Therefore, CO emission increases.
(4) The competition between HC and CO oxidation reactions may result in high CO emission since HC oxidation rate is much faster than CO [94–96]. It is possible that HC is oxidized early in the combustion process but late-cycle CO oxidation is perhaps limited by lower bulk gas temperature and less time available for the reactions to completed [94].

However, when natural gas increased to a very high level, the mixture becomes rich and the flame propagation is promoted which may result in the more complete combustion. This is the reason for CO emission decreasing with the further increase of natural gas.

4.3. Hydrocarbon (HC)

HC is also the consequence of incomplete combustion of hydrocarbon fuel. While the temperature for complete oxidation of HC is lower. It has been found to be around 1200 K with independence of the original fuel type [90].

4.3.1. Effect of natural gas/diesel duel fuel combustion on HC emission

Shioji et al. [97] investigated the effect of some operational parameters on the HC emission in a single cylinder dual fuel engine. They found that in order to suppress HC emission in the middle and low output range, the pilot diesel quantity should be increased and its injection timing should be advanced. In addition, HC emission was also improved by avoiding too lean natural gas mixture by restricting intake charge air.

Papagiannakis et al. [67,71] and Abdelaal et al. [17] analyzed the HC emission of the natural gas-fueled diesel engine. They all found the HC emission with conventional diesel combustion was very low, almost close to zero. While the maximum HC emissions of natural gas/diesel dual fuel were increased up to near 150 g/kg diesel [17] or 6000 ppm [67,71]. There were hundreds of times increase compared with conventional diesel combustion, particularly at part load conditions. Although HC emission of dual fuel combustion reduced with the increase of engine loads, the value continued to be much higher than that of conventional diesel combustion. In addition, in their studies HC emission was found to have an opposite change trend with the NO_X emission. It is obvious that there was a trade-off relationship between NO_X and HC emissions with dual fuel mode.

Egusquiza et al. [88] investigated the effect of natural gas on HC emission under various engine loads at the speeds of 1600 rpm and 2600 rpm in a 4-cylinder direct injection diesel engine. They observed considerably higher HC emission under dual fuel operation compared to diesel operation. Especially at low engine loads, HC emission rose rapidly with the increasing substitution ratio of natural gas to diesel fuel. For the BMEP (brake mean effective pressure) of 1.3 bar at 1600 rpm the HC emission rose from 160 ppm to 20,360 ppm with an increase of about 127 times when the substitution ratio increased from 0 to 69%. While at high engine load, HC emission increased more slowly or even showed a little decrease under high substitution ratios. Their results also showed that for a constant substitution ratio, HC emission decreased with the increase of engine load but increased with the increasing engine speed. Moreover, NO_X emission was also investigated in the literature and it can be found that the trade-off relationship between NO_X and HC emissions of dual fuel mode was very obvious.

Cheenkachorn et al. [89] observed the significantly higher HC emission in a heavy-duty diesel engine operating at full load of engine speeds ranging from 1100 rpm to 2000 rpm with natural gas/diesel dual fuel. The HC emission of dual fuel operation reached above 4000 ppm which was at least 110 times higher than that of the single diesel operation.

Liu et al. [90] investigated the effect of pilot diesel fuel quantity on HC emission in a CNG-diesel dual fuel engine. HC concentrations of over than 10,000 ppm under dual fuel mode were observed at low and medium loads which were much higher compared with significantly less than 100 ppm in normal diesel mode. At high load operation conditions with the increase of pilot diesel quantity, HC emission reduced significantly, but there was still higher than 2000 ppm of HC emission at full load conditions. Their results also showed that around 90% of the HC emission was composed by unburned methane. While as for the low load operation conditions with the increase of pilot diesel quantity, HC emission was also decreased [98].

The increase of HC emission due to natural gas addition was also reported by Imran et al. [24]. They reported a significant increase in HC emission at lower and medium power outputs. There was an increase of about 800% compared to normal diesel mode when natural gas contributed approximately 45% of the total energy. The difference between the two modes was gradually narrowing with the increase of engine load. The HC emission at maximum load conditions in which natural gas took off more than 60% of the total energy was 2.5 times higher than that of normal diesel mode.

The effects of the pilot diesel quantity and injection timing on the exhaust emission in a natural-gas/diesel dual fuel engine were investigated by Sun. et al. [49]. They found that the HC emission decreased linearly with the increase of pilot diesel quantity at both low and higher engine loads, but NO_X emission increased. And with the advance of pilot diesel injection timing, HC emission also decreased and NO_X increased. There was a trade-off relationship between NO_X and HC emissions for dual fuel mode. The trade-off relationship with methane/diesel dual fuel combustion was also confirmed by Gibson et al. [99].

4.3.2. Summary

From the above literature review, it is clear that natural gas/diesel dual fuel combustion produces much more HC emission compared to normal diesel combustion and the increase may exceed 100 times. And there was a trade-off relationship between NO_X and HC emissions. The authors also reported that HC emission under dual fuel mode decreases with the increase of engine load and pilot diesel quantity and the advance of pilot diesel injection timing. The following reasons can be attributed for the increase of HC emission:

(1) Due to the presence of valve overlap period, small part of natural gas air mixture is directly discharged into the exhaust during the scavenging process which leads to increase in HC emission.
(2) Similar with the formation mechanism of CO emission, trapped in the crevices and flame quenching make the unburned fuel hard to be ignite in the latter part of the combustion process, resulting increase in HC emission.
(3) The mixture is very fuel-lean and the in-cylinder temperature is low especially at low engine load condition. It is difficult for combustion to propagate throughout the charge and the unburned mixture may result in higher HC emission.

With the increasing engine load and the advance of pilot diesel injection timing, the combustion is improved and the in-cylinder temperature becomes higher. Thus parts of the above effects might be weakened and HC emission decreases. On the other hand with the increase of pilot diesel quantity, the pilot diesel spray atomization characteristic is improved and the envelope of the pilot diesel spray is also extended, which strengthens the combustion of natural gas air mixture, resulting a decrease in HC emission.

4.4. Carbon dioxide (CO_2)

Carbon dioxide (CO_2) is a product of hydrocarbon fuels completely burning. Hydro carbon fuel is first oxidized to CO during the combustion process. And then if the in-cylinder temperature is high enough and with the presence of oxygen, CO is oxidized to form CO_2 sequentially. Thus, the formation of CO_2 strongly depends on in-cylinder temperature and oxygen concentration.

4.4.1. Effect of natural gas/diesel duel fuel combustion on CO_2 emission

Nwafor et al. [100] compared the CO_2 emission under three different operation mode of normal diesel, natural gas dual fuel and dual fuel with advanced pilot injection timing in a single cylinder diesel engine. The results showed that a net reduction in CO_2 emission was observed under natural gas dual fuel operation compared to the results obtained under normal diesel mode. And the lowest CO_2 concentrations in the exhaust were recorded at dual fuel mode with advanced pilot injection timing. The effect of advanced injection timing was evidence for reducing CO_2 emission.

The effects of natural gas on CO_2 emission using a heavy-duty diesel engine at different engine speed conditions was experimentally investigated by Cheenkachorn et al. [89]. They operated the engine with the maximum portion of natural gas at full load of engine speeds ranging from 1100 rpm to 2000 rpm. They found that natural gas/diesel dual fuel operation showed significantly lower CO_2 emission for all the tested engine speeds compared to single diesel operation. The results also showed that CO_2 emission decreased with the increase of engine speed for both diesel and dual fuel operations.

Lounici et al. [70] analyzed the CO_2 emission of dual fuel operation with natural gas at different engine speeds and engine loads from 20% to 80% of full load in a modified single cylinder diesel engine. Their results showed that CO_2 emission of dual fuel mode was lower than those of normal diesel mode at all the tested engine loads. At low loads, the difference between CO_2 emission for normal diesel and dual fuel mode was not significant. However, the difference was enlarged at higher loads due to the increasing amount of natural gas.

Imran et al. [24] investigated the effect of natural gas addition on CO_2 emission in a single-cylinder compression ignition engine. They found that dual fuel operation produced less CO_2 emission due to lower carbon to hydrogen ratio of natural gas. There was a reduction between 23% and 30% when the engine was operated with dual fuel mode.

4.4.2. Summary

Based on the above literature review, it is clear that all the authors reported significant decrease in CO_2 emission with natural gas/diesel dual fuel combustion. The following reasons can be attributed to the reduction of CO_2 emission.

(1) Natural gas which is mainly composed by methane has one of the lowest carbon contents among hydrocarbons. The combustion of natural gas has a potential of producing lower CO_2 emission than that of neat diesel.
(2) Under dual fuel mode, incomplete combustion is more serious. Some of the fuel is incomplete oxidized to CO and discharged into the exhaust pipe, which may decrease the CO_2 emission.

4.5. Particulate matter (PM)

The emission of PM is one of the primary concerns of diesel engines. Diesel PM is a complex mixture of elemental carbon, a variety of hydrocarbons, sulfur compound and other species [101]. It mainly consists of carbonaceous material known as soot, some absorbed organic compound and sulfates [102–104]. The formation and oxidation of soot particles have a great relationship with the local temperature and oxygen concentration. Soot particles are formed very early in the diffusion combustion process due to the dissociation of fuels under high temperature and lack of oxygen condition. Then most of them are oxidized at very high temperature with the present of oxygen. Although the combustion of diesel engine is generally under conditions of excess oxygen, forming homogeneous mixture is very important to reduce soot particles.

4.5.1. Effect of natural gas/diesel duel fuel combustion on PM emission

Liu et al. [98] investigated the smoke emission in a natural gas and diesel dual-fuelled compression ignition engine. They found that the dual fuel engine was always smokeless under low-speed and low-load conditions. And under high-load conditions, the addition of natural gas also lowered the smoke emission obviously in comparison with the normal diesel mode.

Papagiannakis et al. [71] compared the soot emission of normal diesel mode and dual fuel mode at different engine speeds and loads in a naturally aspirated single cylinder diesel engine. Under dual fuel operation, the amount of pilot diesel was kept constant, while the engine load was adjusted by increasing or decreasing the amount of natural gas. The results showed that with the increase of engine load, soot emission of normal diesel mode increased obviously. However, under dual fuel mode, a reduction of soot emission was observed with increasing engine load. For all conditions examined, soot emission under dual fuel mode was significantly lower compared to those under normal diesel mode. Papagiannakis et al. [67] also investigated the effect of natural gas mass ratio on soot emission. They reported that soot emission decreased sharply at part load as the natural gas mass ratio increased. While at high loads, dual fuel operation with a high mass ratio of natural gas was an efficient way to reduce soot emission.

Liu et al. [90] analyzed the effect of pilot diesel fuel quantity on the PM emission in a CNG-diesel dual fuel engine under various engine loads and speeds. They reported significantly lower PM emission of dual fuel mode compared to that of normal diesel mode. With the increase of pilot diesel quantity, more pilot diesel participated in the diffusion combustion process, which resulted in an obviously increase of the PM emission.

The effect of dual fuel operation with natural gas on soot emission was experimentally investigated in a modified single cylinder diesel engine by Lounici et al. [70]. They found that dual fuel operation was a very efficient technique in reducing soot emission especially at high loads conditions in which soot emission was largely produced. Soot emission of dual fuel operation was considerably lower in comparison with those corresponding to normal diesel mode for all the tested conditions. Natural gas had very small tendency to produce soot emission.

4.5.2. Summary

Based on the above literature review, it is clear that there is a significant decrease in PM emission under dual fuel mode. The following reasons can be attributed for the reduction in PM emission:

(1) Under dual fuel combustion, most of diesel fuel has been replaced by natural gas and the amount of pilot diesel is very small. So less diesel fuel is burned in the diffusion mode and most fuel is burned in the premixed combustion. Therefore, soot formation is less and results in a reduction in PM emission.
(2) Natural gas addition prolongs the ignition delay which may increase the lift-off length of pilot diesel and provide more time for better fuel/air mixing. Therefore, fuel rich areas decreases and initial soot formation is prevented [105].
(3) Natural gas does not contain C-C bond and is free of aromatics and sulfur. Thus, natural gas has very small tendency to produce

soot. In addition, the combustion of homogenous natural gas air mixture contributes to the oxidation of the soot formed from the combustion of pilot diesel. Due to these reasons, PM emission reduces.

5. Engine performance characteristics

Performance characteristics are crucial to the engines. Engine power and brake thermal efficiency (BTE) are important performance indicators. In addition, the cyclic variability which quantifies the stability performance is also very important to dual fuel engine. These factors with natural gas/diesel dual fuel were reviewed and summarized below.

5.1. Engine power

Engine power is one of the most important performance parameters of diesel engines. It characterizes the work capacity of the engine.

5.1.1. Effect of natural gas/diesel duel fuel combustion on engine power

Liu et al. [69] investigated the torque and power in a natural gas diesel dual-fuel turbocharged heavy-duty diesel engine under full-load conditions. They found that the maximum torque was lower than that of normal diesel operation and the corresponding speed increased. However, the maximum power could be tuned the same at rated operation condition.

Gharehghani et al. [106] numerically and experimentally conducted a detailed analysis of performance characteristic of a heavy duty diesel engine in dual fuel mode. They found that increasing the percentage of natural gas in dual fuel mode lead to a decrease in engine output power. But the amount of power loss in dual fuel mode was reduced when increasing intake temperature and pressure. However, these increases may lead to initiation of engine knock.

The engine torque and power of normal diesel operation and natural gas dual fuel operation were compared by Cheenkachorn et al. [89] in a heavy-duty diesel engine. The test conditions were chosen at full load of engine speeds ranging from 1100 rpm to 2000 rpm. They found that the torque and power of the dual fuel operation were slightly lower compared to those of the diesel operation. The maximum decline was 2.1%.

5.1.2. Summary

From the above literature review, the following reasons can be attributed to decrease in engine power with dual fuel combustion:

(1) The heat value of natural gas/air mixture is lower than that of diesel/air mixture which may lead to decline in engine power.
(2) The addition of natural gas in the intake stroke reduces the amount of intake air and lowers the volumetric efficiency at the same time, resulting in a decrease of engine power, especially under high load conditions.

The power loss can be reduced or recovered by changing some of the operating parameters under dual fuel combustion mode, such as increasing the pilot diesel quantity [84,89], increasing intake temperature and pressure [106] and so on. But these adjustments may cause the engine knock.

5.2. Brake thermal efficiency (BTE)

BTE is the percentage of the brake power and the fuel energy consumed by the engine. It indicates how efficiently the input energy is converted to useful output energy [107]. BTE is calculated by the following formula:

$$BTE = \frac{3600 \times P_b}{q_{m,g} \times Q_{LHV,g} + q_{m,d} \times Q_{LHV,d}} \times 100\%$$

where $q_{m,g}$ and $q_{m,d}$ are the mass consumption of natural gas and pilot diesel respectively, in kg/h; $Q_{LHV,g}$ and $Q_{LHV,d}$ are the low heating value of natural gas and pilot diesel respectively, in kJ/kg; P_b is the brake power in kW.

5.2.1. Effect of natural gas/diesel duel fuel combustion on BTE

Abd Alla et al. [84,85] conducted tests and investigated the effect of pilot diesel quantity and injection timing on the thermal efficiency of a special single cylinder compression ignition research engine fueled with natural gas. Their results showed that both increasing the amount of pilot diesel and advancing the injection timing could improve the thermal efficiency of the engine at light loads, because of the corresponding high pressure and temperature. However, both of the two measures led to early knocking at high engine loads.

Krishnan et al. [54] investigated the effect of pilot diesel injection timing on fuel conversion efficiency at full load of 1700 rpm in a single-cylinder natural gas fueled diesel engine. They found that fuel conversion efficiency increased from 38% to about 43% when pilot diesel injection timing was advanced from 15 to 45°CA BTDC. With further advance of the injection timing, the efficiency began to decrease and reduced to about 40% at 60°CA BTDC.

Papagiannakis et al. [87] investigated the BTE of a high speed dual fuel diesel engine operating in a wide range of natural gas/diesel fuel proportions. The results showed that with the increase of natural gas mass ratio, the total relative air–fuel ratio decreased, which resulted in a lower BTE compared to the one under normal diesel operation. The deterioration of the BTE under dual fuel operating mode was more evident at low and intermediate loads. They reported that the BTE of dual fuel operation at low loads was almost half as compared to that under normal diesel mode. And at intermediate loads the decrease of BTE was varied from up to 40% to up to 75%. While at high loads and high natural gas mass ratios the BTE was slightly improved.

Abdelaal et al. [17] analyzed the BTE of a natural gas-fueled dual-fuel diesel engine at the operating speed of 1600 rpm from 43% up to 95% of the engine full load. They reported that the BTE of the dual-fuel mode was lower at part loads and a little higher at high load compared with conventional diesel mode at the same conditions. There was a maximum increase of about 3% at 95% of the engine full load.

Cheenkachorn et al. [89] compared the BTE of single diesel mode and natural gas/diesel dual fuel mode in a heavy-duty diesel engine at full load of engine speeds ranging from 1100 rpm to 2000 rpm. The portion of natural gas was fixed at the maximum value of each operating condition. Their results showed that the BTE of dual fuel operation was a little less than that of single diesel operation during the entire operation speeds. Especially for the engine speed higher than 1700 rpm, the BTE dropped significantly compared with the single diesel operation. The measured average reduction was 3.50% for the tested speed range.

Thermal efficiency at a wide range of engine speed and load of both normal diesel mode and natural gas dual fuel mode was investigated by Imran et al. [24] in a single-cylinder compression ignition engine. They found that the thermal efficiency of dual fuel mode was similar or a little higher as compared to normal diesel mode at higher power outputs, whereas at relatively lower power outputs, it exhibited a slightly lower. They explained that the fuel/air ratio might have a great relationship for the difference. At the highest power outputs the fuel/air ratio of dual fuel mode was 3.73% higher than that of normal diesel mode resulting in an approximately 3.1% increase in thermal efficiency.

5.2.2. Summary

From the above literature review, it is clear that the BTE under dual fuel combustion is a little lower than that of normal diesel mode, especially at low and intermediate loads. While under high engine load conditions the BTE is similar or a little higher as compared to normal diesel mode. The maximum increase is about 3% at near the engine full load conditions. The reduction of BTE can be attributed to the following reasons:

(1) The natural gas/air mixture under dual fuel combustion is very lean at part engine loads. It is difficult for pilot fuel to ignite and sustain adequate combustion of the mixture. Thus, the very lean mixture cannot be burned and goes with the exhaust, which result in poor fuel utilization efficiency and lower BTE.
(2) The slower burning rate due to the slower flame propagation speed increases the heat loss during the combustion process, resulting decrease in BTE.

The improvement of BTE at high engine load conditions can be explained by attributing the following reasons:

(1) At high loads, more natural gas is being introduced to the cylinder. The natural gas/air mixture becomes rich and hence becomes able to form a sustainable flame. Consequently, the fuel utilization is improved which may result in the improvement of BTE.
(2) The temperature of intake and combustion chamber wall are higher, and the mixing process of fuels and air is better at high loads. At the same time, the in-cylinder temperature is higher. These factors are beneficial for the completely combustion of the homogenous mixture which tends to increase the BTE.

5.3. Cyclic variability

The cyclic variability which is defined as the criterion for combustion stability is often expressed by the coefficient of variation (COV) in the indicated mean effective pressure (IMEP), the peak in-cylinder pressure or other combustion parameters. Among them the coefficient of variation in indicated mean effective pressure (COV_{IMEP}) is the most common used and it is defined as follows:

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{\overline{IMEP}} \times 100\%$$

where σ_{IMEP} is the standard deviation of net IMEP and $\overline{IMEP}$ is the average of the IMEP of a specific combustion cycles.

5.3.1. Effect of natural gas/diesel duel fuel combustion on cyclic variability

The cyclic variability of a natural gas/diesel dual fuel engine were investigated by Sun. et al. [49]. The results showed that with the increase of engine load the COV_{IMEP} decreased. They also reported a decline of COV_{IMEP} with the advance of pilot diesel injection timing. The decreasing COV_{IMEP} with an increase in engine load was also observed by Tarabet et al. [63] in a natural gas/biodiesel engine.

Srinivasan et al. [18] examined the cyclic variability of natural gas/diesel dual fuel combustion on a single cylinder research engine. Two dual fuel combustion modes were involved. At conventional dual fuel combustion mode, cyclic variations increased with increasing natural gas addition. The maximum COV_{IMEP} reached about 12% at 95% natural gas substitution. While at dual fuel partially premixed combustion mode, the highest COV_{IMEP} was about 25% which is much higher than conventional dual fuel mode.

The effects of pilot injection timing were investigated in a biodiesel natural gas dual fuel engine by Ryu et al. [38]. The results showed that the COV_{IMEP} with biodiesel natural gas dual fuel combustion was minimized at 17 °CA BTDC and increased with advance/delay in injection timing. But the changes were small and the COV_{IMEP} was below 1.3% under all conditions. Krishnana et al. [54] and srinivasan et al. [19] also investigated the influence of pilot injection timing on cyclic variability in a single-cylinder pilot-ignited natural gas engine. Their results were similar with that of Ryu et al. [38]. The COV_{IMEP} varied little at the middle pilot injection timings and increased obviously with both advance and delay in injection timing. From the study of Krishnana et al. [54], it can also be found that the COV_{IMEP} decreased with an increase in engine load. The highest COV_{IMEP} was 7.8% at full load when the pilot injection timing was advanced to 65°CA BTDC.

Some measures were taken by many researchers to reduce the cyclic variability of natural gas/diesel dual fuel combustion. Srinivasan et al. [56] studied the influence of intake charge temperature and pilot injection quantity on cyclic variability at low engine load and increasing the intake charge temperature was proved to be an effective strategy to reduce variation. The COV_{IMEP} was decreased from 10.6% to 3.8% at half engine load and from 26.1% to 6.6% at quarter engine load when the intake temperature was increased from 75 °C to 115 °C. However, NO_X emissions increased a lot with the increasing intake temperature. Singh et al. [86] also confirmed the improvement of increasing intake temperature on reducing cyclic variability. As intake temperature increased from 75 °C to 105 °C, the COV_{IMEP} reduced from 23.5% to 11.5%. Srinivasan et al. [57] also investigated the effect of adding hot EGR on the combustion stability at low engine load. Their results showed that COV_{IMEP} generally decreased with the increasing hot EGR addition. When the injection timing was at 60°CA BTDC, COV_{IMEP} was reduced from about 30% at 0% EGR to about 5% at 21% EGR. However, the engine performance needs to be optimized with the addition of hot EGR.

5.3.2. Summary

Based on the above literature review, it is clear that the COV_{IMEP} with natural gas/diesel dual fuel seems to be generally higher than normal diesel combustion. This can be attributed to the increasing variations of ignition and flame propagation for natural gas. However, the combustion stability can be enhanced by increasing the intake charge temperature or introducing hot EGR at low engine load.

It can also be found that with the engine load increasing the COV_{IMEP} decreases. At low engine load, the air/fuel mixture is very lean which results in high cyclic variations. While with an increase in engine load, the excess air coefficient decreases and the in-cylinder temperature increases at the same time, so the flame propagate becomes faster and the combustion duration gets shorter and hence the COV_{IMEP} decreases.

6. Conclusions

Because of the growing environment and energy issues, natural gas has been considered as a promising alternative fuel. Natural gas/diesel dual fuel mode is a more practical and low cost mean of using natural gas in diesel engine. Therefore, this method has been given a lot of attention by many researchers in order to improve the engine performance and reduce the diesel consumption. After consulting and summarizing plenty of related literatures in dual fuel mode the following general conclusion could be drawn:

(1) In-cylinder pressure during the compression and the initial periods of combustion is slight lower and the ignition delay is longer under dual fuel mode. The peak in-cylinder pressure and pressure rise rate and the value of heat release rate under dual fuel mode all have a great relationship with the injection parameters of pilot diesel. Under normal situation, they are lower and the combustion duration is longer compared with normal diesel mode.
(2) The dual fuel mode can significantly decrease the NO_X, CO_2 and PM emissions, but the HC and CO emissions may increase by several times or even more than 100 times. NO_X emission under dual fuel mode is affected by engine loads and pilot diesel quantity; it may increase at high engine load. With the increase of engine load and pilot diesel quantity and the advance of pilot diesel injection timing, HC emission under dual fuel mode all decreases but NO_X emission increases. There is a trade-off relationship

between NO_X and HC emissions during dual fuel combustion. While with the increase of natural gas, CO emission shows a decreasing trend after the first increase.

(3) The engine power of dual fuel mode is lower than normal diesel mode, the maximum decline has been found to be 2.1%. While the power loss can be reduced or recovered by changing some of the operating parameters such as increasing the pilot diesel quantity, increasing intake temperature and pressure and so on.

(4) The dual fuel mode suffers from low BTE especially at low and intermediate loads, while under high engine load conditions the BTE is similar or a little higher as compared to normal diesel mode. The maximum increase is about 3% at near the engine full load conditions.

(5) The COV_{IMEP} of dual fuel mode seems to be generally higher than normal diesel mode and it decreases with an increase in engine load. The combustion stability with dual fuel can be enhanced by increasing the intake charge temperature or introducing hot EGR at low engine load.

Nomenclature

$q_{m,g}$	mass consumption of natural gas, kg/h
$Q_{LHV,g}$	low heating value of natural gas, kJ/kg
P_b	brake power, kW
σ_{IMEP}	standard deviation of net IMEP
$q_{m,d}$	mass consumption of pilot diesel, kg/h
$Q_{LHV,d}$	low heating value of pilot diesel, kJ/kg
Z	mass fraction of natural gas in the total fuel, %

Abbreviations

NO_X	Nitrogen Oxides
CO_2	carbon dioxide
CO	carbon monoxide
NO_2	nitrogen dioxide
BMEP	brake mean effective pressure
COV	coefficient of variation
BTE	Brake thermal efficiency
CA	crank angle
ppm	parts per million (in volume)
r/min	rotation per minute
PM	particulate matter
HC	hydrocarbon
NO	nitrogen monoxide
CNG	compressed natural gas
BTDC	before top dead center
IMEP	indicated mean effective pressure
HPDI	High pressure direct injection
EGR	exhaust gas recirculation
Net HRR	net heat release rate
mg/cyc	milligram per cycle

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