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ORIGINAL ARTICLE

LPG diesel dual fuel engine – A critical review

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Performance

Abstract The engine, which uses both conventional diesel fuel and LPG fuel, is referred to as 'LPG–diesel dual fuel engines'. LPG dual fuel engines are modified diesel engines which use primary fuel as LPG and secondary fuel as diesel. LPG dual fuel engines have a good thermal efficiency at high output but the performance is less during part load conditions due to the poor utilization of charges. This problem can be overcome by varying factors such as pilot fuel quantity, injection timing, composition of the gaseous fuel and intake charge conditions, for improving the performance, combustion and emissions of dual fuel engines. This article reviews about the research work done by the researchers in order to improve the performance, combustion and emission parameters of a LPG–diesel dual fuel engines. From the studies it is shown that the use of LPG in diesel engine is one of the capable methods to reduce the PM and NOx emissions but at same time at part load condition there is a drop in efficiency and power output with respect to diesel operation.

1. Introduction

Environmental concerns and depletion in petroleum resources have forced researchers to concentrate on finding alternatives to conventional petroleum fuels. Excessive use of fossil based fuels exhausts the reserves and also increases the air pollution. These improve the awareness of the effective use of present reserves and slowly switches over to the alternative fuels, which are environment friendly [1–5,95]. One of the solutions to accomplish this is the use of gaseous fuels in addition to the liquid diesel in CI engine. The use of alternative gaseous fuels e.g. natural gas, liquefied petroleum gas (LPG), etc. is a promising approach for lowering the dependence on petroleum based liquid fuels and to reduce the emissions of CO_2 and other pollutants from diesel engine [6]. LPG is a viable alternative gaseous fuel (also known as ''Auto gas'') which is a gas product of petroleum refining primarily consisting of propane, propylene, butane and other light hydrocarbons [7–9]. It can be liquefied in a low pressure range of 0.7–0.8 Mpa at atmospheric temperature. So, storage and transportation of LPG is easier than other gaseous fuels. LPG has high calorific value compared to other gaseous fuels and also it has high octane number but a low cetane number. The high octane number of LPG makes it suitable for spark ignition engines. In contrast, the low cetane number of LPG makes it difficult to be used in large proportions in compression ignition engines, mainly due to high cyclic variation [10,11,101]. Hence it can be used in the CI engine in the dual fuel mode only and in this mode it has been extensively studied. It leads to better performance, low particulate and smoke emissions [12]. The engine, which uses conventional diesel fuel and LPG fuel, is referred to as 'LPG–Diesel dual fuel engine'. In this engine, LPG fuel is mixed with the air in the engine

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cylinders either through direct mixing in the intake manifold with air or through injection directly into the cylinder [13]. A dual fuel engine is basically a modified diesel engine in which a LPG fuel, called the primary fuel is inducted along with air. This fuel is the main source of energy input to the engine. The primary gaseous fuel is compressed with air, but does not auto ignite due to its high self-ignition temperature. A small amount of diesel, usually called the pilot, is injected as in a normal diesel engine near the end of compression of the primary fuel–air mixture. This pilot diesel fuel, auto ignites first and acts as a deliberate source of ignition for the combustion of the gaseous fuel–air mixture. The pilot diesel fuel, which is injected by the conventional diesel injection equipment normally, contributes only a small fraction of the engine power output. Thus the combustion process in a dual fuel engine is complex as it combines the features of SI and CI engines [15–19]. The dual fuel engines can also be reverted back to straight diesel operation easily [14]. Dual fuel operation has advantages compared to diesel counterparts and spark ignition (SI) engines, theoretically higher thermal efficiency resulted from faster burning, less toxic emissions, high power density, strong ignition sources providing more reliable [20]. By converting diesel engines to run on LPG we can significantly reduce the problem of diesel pollution while also improving emissions of greenhouse gases [21,22]. Such conversions are however not a simple matter of changing the fuel, many technical problems present particularly with availability of specific fuel supply system, fuel injection control and engine optimization to ensure that the engine performance is maintained and the exhaust emissions are minimized [23].

However, the dual fuel engine has some pitfalls such as the poor utilization of the LPG fuel at low and intermediate loads which results in poor engine performance (drop in engine efficiency), high HC, CO emissions and misfiring at higher gas inducted levels. Poor part load performance results from incomplete combustion of LPG. Due to this poor thermal efficiency high level of unburnt hydrocarbons in the exhaust is found [24–26,100,102]. The performance of a dual fuel engine at idling and low loads can be improved by optimizing some engine operating and design parameters, such as engine speed, load, pilot fuel quantity, injection timing, intake manifold condition and intake gaseous fuel compositions [27,28]. In this literature review, studies with wide range of diesel engine sizes and different types investigated at different operation conditions are reviewed. Similarly, different percentages of LPG were applied to optimize the engine output. Engine performance, combustion and emissions characteristics are discussed at different sections to get the clear scenario on the effects of using liquefied petroleum gas (LPG) in diesel engine in dual fuel mode.

2. LPG–diesel dual fuel operation

All internal combustion (IC) reciprocating engines operate by the same basic process. A combustible mixture is first compressed in a small volume between the head of a piston and its surrounding cylinder. The mixture is then ignited and the resulting high-pressure products of combustion push the piston through the cylinder. There are two ignition methods used in reciprocating IC engines, compression ignition (CI) and spark ignition (SI). The existing method of operation of diesel engine is by compression ignition method, where the intake air alone is compressed and at the end of the compression stroke the diesel fuel is directly injected at high pressure over the compressed air inside the combustion chamber which leads to ignite easily by virtue of its ignition temperature. But the LPG–diesel dual fuel engine utilizes the concept of both compression ignition and spark ignition principles to burn the mixture of primary gaseous (LPG) fuel and liquid pilot fuel [29–31].

In case of LPG–diesel dual fuel engine, the air-to-LPG mixture from the intake is drawn into the cylinder, just as it would be in a spark-ignited engine and this mixture is compressed in order to increase the temperature and pressure. At the end of the compression stroke the mixture is ignited by the injection of small quantity of pilot diesel fuel as shown in Fig. 1. This pilot injection acts as a source of ignition. The LPG gas-air mixture in the vicinity of the injected diesel spray ignites at number of places establishing a number of flame-fronts. Thus the combustion starts smoothly and rapidly. It is interesting to note that in a dual-fuel engine the combustion starts in a fashion similar to the CI engine but it propagates by flame fronts, i.e. in a manner similar to the SI engine. The power output of the engine is normally controlled by changing the amount of primary LPG gaseous fuel added to inlet manifold. The quantity of diesel fuel used will be varied depending upon the engine operating conditions and its design parameters, and generally the amount of pilot diesel required for the ignition is between 10% and 20% of operation on the diesel fuel alone at normal working loads [32,33]. The mass fraction of the LPG used in dual fuel mode is calculated by using the following expression, 'Z':

$$Z = \frac{m_{LPG}}{m_{Diesel} + m_{LPG}} * 100\%$$

where m_{Diesel} is mass flow rate of diesel and m_{LPG} is mass flow rate of LPG. And $z = 0\%$ represents diesel operation, $z = 10\%$, 20%, 30%, 40% represent the LPG mass fraction used in dual fuel mode [34–39,98].

3. Diesel engine modifications

Diesel engines can be readily configured to run on LPG–diesel dual fuel mode, where LPG is mixed into the air intake, while the normal diesel fuel injection system still supplies a certain amount of diesel fuel, but at a reduced rate [9,96]. The engine has to be modified to work in the dual fuel mode by attaching an LPG line to the intake manifold along with an evaporator

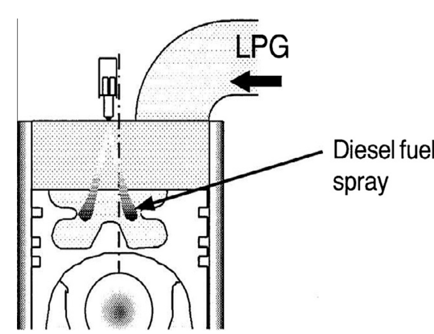

Figure 1 LPG diesel dual fuel engine [46].

[93]. Gaseous fuel flows through the regulating valve into the gas mixer assembled on the intake manifold [40–42]. The design of gaseous fuel supply system has a significant influence on HC emissions at light load where LPG is supplied to intake manifold by the small duct as discussed in the ref [43]. Supply of the LPG into the engine is accompanied by mechanical or electronic control for various loads and speed of the engine. Depending on the type of the engine (direct injection or port injection) and gaseous fuel supply system the combustion process and the engine output will be varied. The mass flow rate of LPG is in proportion to the pressure difference between the gas mixer and the evaporator where the pressure is maintained nearly same as atmospheric pressure. But controlling the flow rate of the primary LPG fuel and pilot diesel fuel at different engine operating conditions is very critical in the LPG dual-fuel engines. Lower LPG content may have no effect on soot emission reduction and performance improvement, and at the same time much higher LPG content probably makes in-cylinder pressure increased rapidly and damages the engine [45]. Various research activities have been carried out in the LPG–Diesel dual fuel engines by many researchers, such as modification of single cylinder [2,4,6,10,15,36,44,52,6] and multi cylinder [8,45,61] diesel engines in which the LPG is supplied in the dual fuel mode by manifold injection [7,20,22] and manifold induction [2,4,15,27,44,72]. It is found that the combustion, performance and emission characteristics of LPG–Diesel dual fuel engine depend on the type of the engine, LPG fuel supply system and engine operating conditions.

4. Combustion process of LPG–diesel dual fuel engine

The combustion process of pure diesel engine takes place in four stages, termed as Ignition delay, rapid combustion, controlled combustion and period after burning. Whereas the combustion process of the LPG dual fuel engine shares the characteristics of both SI and CI engine combustion characteristic and consists of five stages throughout the combustion process of Ricardo E6 single cylinder as shown in Fig. 2. The stages are pilot ignition delay (AB), pilot premixed combustion (BC), primary fuel (LPG) ignition delay (CD), rapid combustion of LPG fuel (DE), period after burning (EF).

Pilot ignition delay period (AB) is longer than the pure diesel fuel operation due to the reduction in concentration of intake oxygen, resulting from LPG inducted along with intake air and partly due to the change in the specific heat of the compressed mixture that resulted in lowering the compression temperature. The pilot premixed combustion (BC) leads to produce the small flame, which is first initiated by the small quantity of diesel and combustion increases the pressure smoothly. Primary fuel (LPG) occupying in compression process undergoes chemical pre-ignition reactions leading to primary ignition delay (CD) where the pressure starts to decrease in this period and its rate is very low. The phase of rapid combustion (DE) is very unstable, because it started with flame propagation that has been initiated by the spontaneous ignition of pilot diesel fuel. At first in this stage pressure is increased mainly due to the premixed burning of part or whole of the pilot diesel in addition to combustion of a small part of the LPG gas entrained. Then again the pressure is increased to its maximum level because of combustion of the major part of the LPG gaseous fuel and small amount of the remaining pilot

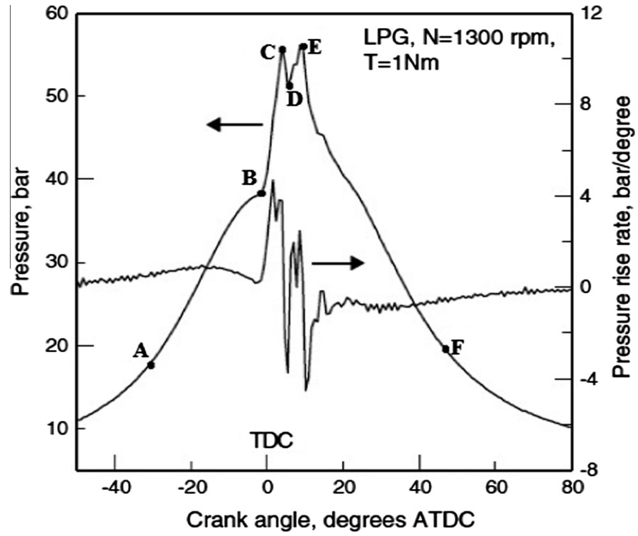

Figure 2 Typical combustion pressure & pressure raise rate vs crank angle of LPG diesel fuel engine [47].

diesel inside the cylinder. Period after burning stage (EF) starts at the end of rapid pressure raise and continues well into the expansion stroke. This is due to the slower burning rate of LPG fuel and the presence of diluents from the pilot fuel. The success of this phase primarily depends on the length of ignition delay.

However, the combustion process also generally has problems of knocking and misfiring when the percentage of inducted LPG fuel is increased [46,97,99]. So the mass of LPG used at higher loads has to be considered in order to overcome the knock and misfiring. Increasing the amount of diesel pilot fuel quantity reduces the engine knock at high output conditions. For LPG dual fuel engine, the maximum pressure is always higher than diesel fuel case, due to the combustion and extra heat released from gaseous fuel [47]. The higher LPG ratio in dual fuel modes leads to two effects. First, the premixed combustion and the speed of flame propagation increases but the mixing-controlled combustion for the liquid fuel reduces. Second, the reduced amount of pilot injection causes the smaller size of the ignition sources, therefore increases the path that the flame needs to propagate to consume all the premixed mixture in the chamber [48].

The Rate of Heat Released (ROHR) of LPG diesel dual fuel shown in Fig. 3 for a single cylinder constant speed engine shows that there are three important burning phases of the combustion process, and they are premixed combustion of the pilot diesel fuel, premixed combustion of the primary LPG fuel and diffusion combustion of LPG fuel and the left over pilot diesel fuel. The first phase is mainly due to the pilot diesel fuel and small part of the LPG entrained by the spray of diesel is burning along with diesel. Heat energy released in this phase is mainly due to the diesel fuel burned. The second phase is due to the burning of a maximum part of the LPG fuel and a part of the rest of the pilot diesel fuel. The heat energy released during the premixed combustion phase of the LPG fuel depends on the increasing quantity of pilot diesel substitution. It means that the amount of heat energy released by the LPG is more when its quantity is increased. Hence, the gaseous fuel burns mostly during the second phase of the combustion.

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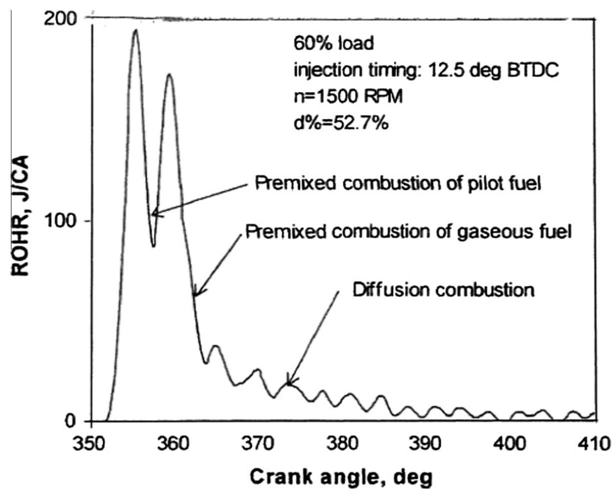

Figure 3 Rate of heat release rate vs crank angle of a dual fuel diesel engine combustion [49].

Finally, the third phase is due to the combustion of the rest of the two fuels which are not burned in the last phase. The diffusion combustion phase is due to the diesel droplets that did not burn during the premixed combustion or that are injected after the start of the premixed combustion. These droplets are burning slowly and gradually in a liquid phase inside the cylinder [49,9].

5. Performance, combustion and emission characteristic of an LPG–diesel dual fuel engine

The performance and emission characteristic of LPG–diesel dual fuel engine is influenced by engine operating and design parameters. The effects on parameters such as changes in the engine speed, load, pilot fuel quantity, injection timing, intake manifold condition and intake gaseous fuel compositions on the performance, combustion and emissions characteristics were investigated. Much research effort had been expended towards providing effective measures for the further improvement of LPG diesel dual fuel engine operating characteristics.

5.1. Effect of engine load

In LPG–diesel dual fuel engines, good diesel substitution levels are only obtained at mid-load range, at low load the pilot diesel injector still requires a substantial fuel delivery while at high load the prolonged ignition delay increases the tendency for diesel knock as well as end-gas knock [50,51]. The volumetric efficiency of the engine has been found to decrease with an increase in the LPG flow rate at all the loads. This is due to the fact that a part of the cylinder space is occupied by the LPG, providing reduced space available for the incoming air [52,53]. At light load, the dual fuel engine tends to exhibit inferior fuel utilization and power production efficiencies with higher unburnt gaseous fuel and carbon monoxide emissions, relative to the corresponding diesel performance. Operation at light load is also associated with a greater degree of cyclic variations in performance parameters, such as torque, peak cylinder pressure, and ignition delay, which tends to narrow the effective working range for dual fuel operations. This is due to poor flame propagation characteristics of the lean LPG air mixture and various ignition centers originating from the pilot diesel [54]. The dilute fuel air mixtures result in poor combustion of the gaseous fuel and cause the formation of high carbon monoxide and unburned hydrocarbon emissions at light load. This is mainly because of the flammability limit of the lean homogenous charge which leads to incomplete combustion or in the worst case absence of combustion [55]. Any measure that lowers the effective lean flammability limit of the charge and promotes flame propagation will improve the part load performance [56].

Selim has investigated in Ricardo E6 single engine about the combustion noise, knock and ignition limits of LPG fuel in dual fuel mode. He found that the output torque increases with increasing the amount of LPG fuel. The mass of gaseous fuel has been increased until the engine starts to knock as shown in Fig. 4 (a). He found that knock starts at lower torque itself. This early knocking onset of the LPG is due to the lower auto-ignition temperature of the LPG (about 400 °C). Value of maximum torque of a dual-fuel engine is limited by the occurrence of knock in the combustion [57,92]. As the mass of LPG fuel introduced increases with load the maximum combustion temperature increases, the gaseous fuel exists in the combustion chamber would be more susceptible to self ignite. Also the increase of LPG fuel may then cause an increase in the ignition delay period of pilot diesel which then auto-ignites and starts burning the gaseous fuel at higher rate of pressure raise. LPG, however in dual fuel mode, produces the higher pressure raise rate as shown in Fig. 4 (b) because of its high tendency to self ignite and produce knocking in the combustion [27,28,58,59].

In the LPG diesel dual fuel operation, physical/chemical properties of the compressed charge, characterized by the presence of the LPG fuel accompanied with air and residual gas at the time of pilot injection, result in change in ignition delay of the pilot [48]. It is observed from Fig. 5 that the ignition delay period of single cylinder engine in the dual fuel mode is always longer than in the diesel. This increase of delay period is due to partly the change in the specific heat of the compressed mixture that resulted in lowering the compression temperature (as compared to the diesel mode). The other reasons may be also due to the decreased oxygen concentration, and the direct influence of the gaseous fuel on the ignition of the pilot diesel spray. The gaseous fuel added can undergo significant reactions during the compression stroke leading to partial oxidation products. This can affect the pre-ignition processes of the pilot adversely. It may be noted that as the load increases, the temperature of the cylinder components and the mixture temperature will raise. This could lead to a reduction in the ignition delay [60].

In a four cylinder diesel engine Stewart et al. found that primary fuels exhibit in decreasing and then increasing peak rates of heat release (peak cylinder pressures) that occurs later in the cycle. This indicates that some change to the basic combustion processes has occurred as shown in Fig. 6 of heat release rate of propane. At low loads, a more reactive primary fuel will result in a delayed combustion process because of the pre-ignition reaction radicals. Considering the combustion processes to proceed in premixed and diffusion stages, it is revealed that, as the load increases the primary fuel shows the importance of the second stage of the combustion process for energy conversion. The diffusion section of the combustion process becomes more important, i.e. a larger proportion of the energy being released in the 'diffusion' stage. This shows that the premixed

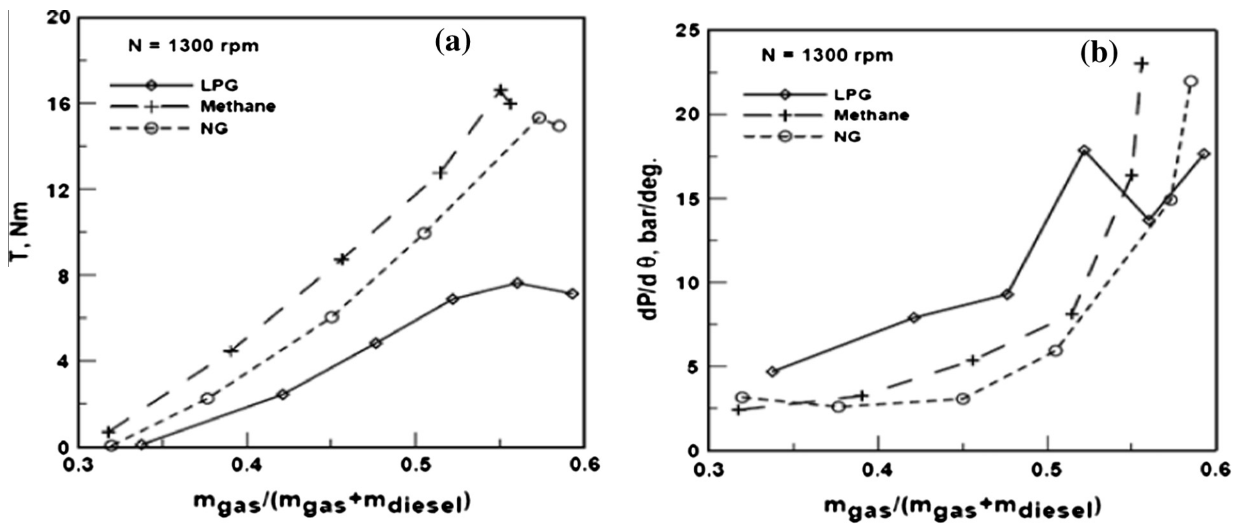

Figure 4 Effects of mass of gaseous fuel use on torque (a) and noise (b); N = 1300 rpm, IT = 35°BTDC, CR = 22, md = 0.37 kg/h [58].

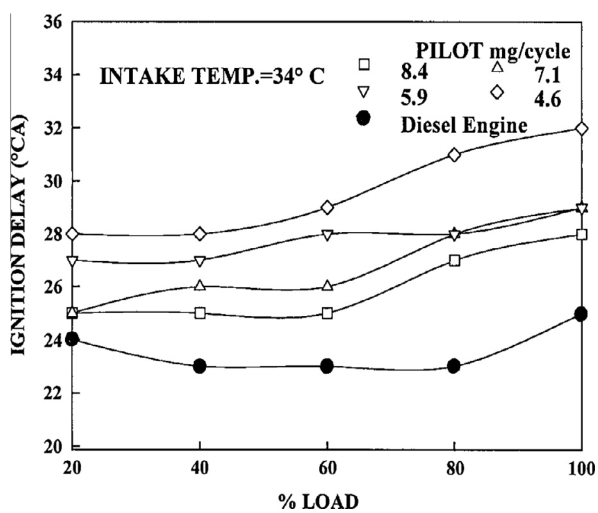

Figure 5 Effect of load on ignition delay [60].

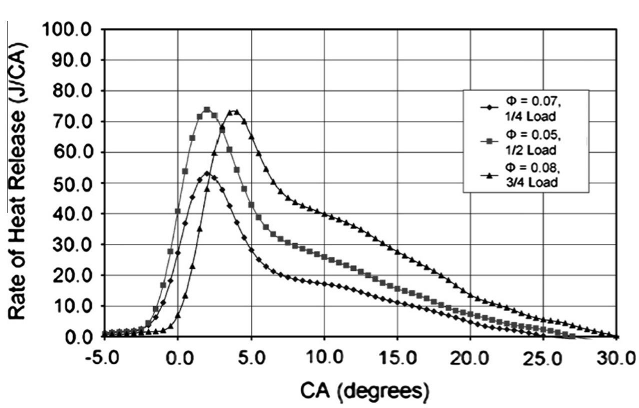

Figure 6 Heat release rate vs CA for propane–diesel at different loads and 1500 r/min [61].

Table 1 Modes (13) of the testing procedure [45].

Mode	Load	Speed (rpm)
1	Idle	600
2	10%	2000
3	25%	2000
4	50%	2000
5	75%	2000
6	100%	2000
7	Idle	600
8	100%	3000
9	75%	3000
10	50%	3000
11	25%	3000
12	10%	3000
13	Idle	600

charge is taking a more important role in the overall heat release of the fuel inside the engine. They found that for both half- and three-quarters-load cases a reduction of 20 percent energy consumption was recorded while using propane as primary fuel [61].

Emission measurements for the various engine loads were based on a 13 mode test procedure of the National Standards (GB 17691-1999 PR China) as specified in Table 1 which is investigated by Dong et al. under LPG dual fuel mode in a six cylinder diesel engine, in which specially designed CAM shaft is used to vary the LPG flow quantity in an automatic manner depending on the engine operating condition and they found that the NOx emissions were not obviously improved when diesel–LPG fuel was used in comparison with net diesel fuel, though the NOx emissions were slightly reduced on all modes except on mode 8, as shown in Fig. 7. The reason for this result is probably the prolonged ignition delay due to the use of diesel–LPG fuel. A sluggish ignition tends to be beneficial for diminishing the NOx emissions, but a higher in-cylinder pressure on the dual-fuel mode, tend to cause higher flame temperature, on which the rate of NOx formation

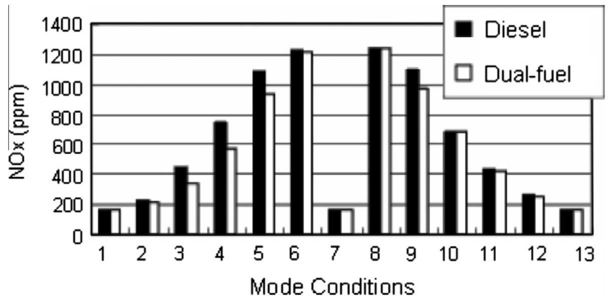

Figure 7 Effect of load on NOx emission [45].

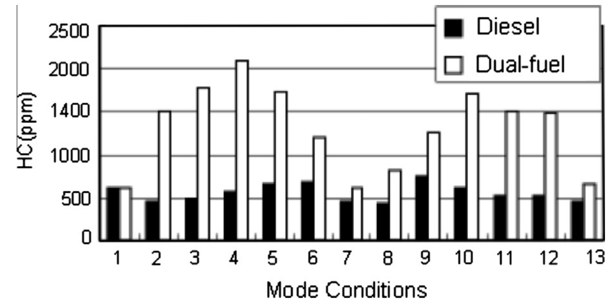

Figure 8 Effect of Load on HC emission [45].

increases. HC emissions are presented in Fig. 8 which were extremely high, especially when LPG content was high and the engine load was light. The probable reason for this result is a lower reaction temperature in the chambers at lower loads with diesel–LPG fuel. Moreover, the higher LPG content at lower loads tends to cause a prolonged ignition delay. At very light load conditions like mode 2 and mode 12, the HC emissions on the dual-fuel mode were much higher than on the diesel mode in spite of very low LPG content at this operating condition. The CO emission as shown in Fig. 9 was slightly reduced at full load but was significantly increased on the other mode conditions. Also at light load conditions the CO emissions on the dual-fuel mode were much higher than on the diesel mode in spite of very low LPG content at this operating condition [45].

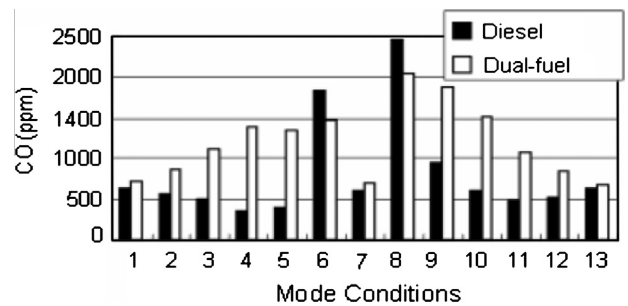

Figure 9 Effect of load on Co emission [45].

5.2. Effect of engine speed

Tuan and Truc examined in AVL 5402 single cylinder engine that at full load, higher LPG mass fraction can be reached at lower engine speeds due to knocking limit; however, the economic efficiency with dual fuel is better at higher engine speeds, as at these operating regimes the better diesel injection quality and better fuel evaporation can help to improve the combustion. Due to knocking limit the LPG mass fraction can reach 54% at engine speed lower or equal to 2000 rpm on full load curve as shown in Fig. 10. At higher engine speed, the LPG mass fraction can reach up to 40%. The total fuel consumption of dual fuel cases showed no improvement at 10% LPG mass fraction, but obvious improvement with 20%LPG (at 1400 rpm and higher), and with 30%LPG (at 1200 rpm and higher). These express that the economic efficiency with dual fuel is better at higher engine speeds as at these operating regimes the better injection quality and better fuel evaporation can help combustion improve [62].

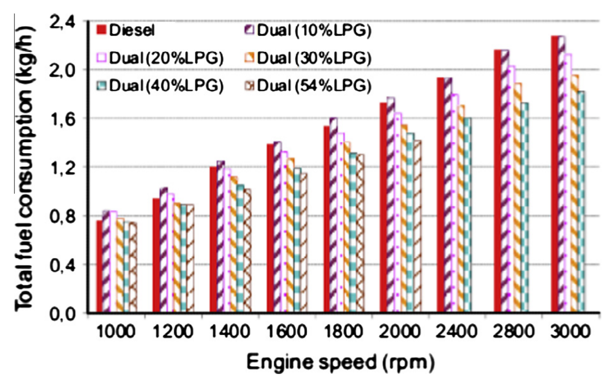

Figure 10 Total fuel consumption at full load and different engine speeds [62].

With increase in engine speed, both physical and chemical delays can be reduced since turbulence and compression temperature increase [63]. As the speed increases the torque output of the engine is increased marginally as shown in Fig. 11 (a). Higher the engine speed, lower the pressure raise rate generated in combustion which represents lower combustion noise as shown in Fig. 11 (b). This reduction in combustion noise may be explained by the improved mixing between air and gaseous fuel which enables the combustion to start smoothly and produces lower pressure raise rate. But it may be noted that increase in engine's overall noise with the increase in the engine speed may be postulated due to some factors. These factors include the injection pump, and injector's valves and gears produce more noise as they run at a higher speed. Also there is an increase in the intake and exhaust flow turbulence and noise as the engine runs at a faster speed. This may suggest that for higher engine speed a higher fuel and air flow rate is required which in turn increases the level of mixing and flow noise due to higher turbulence levels [58,59,64,65].

Vezir et al. studied the effect exhaust emission characteristics of a single cylinder diesel engine running with LPG at different engine speeds. They observed the minimum NOx at 5% LPG injection rate at the speed of 2000 rpm. Maximum decreases of NOx in the modes are obtained as 27.6% with 5% LPG, 24.7% with 10% LPG, 14.8% with 15% and 14.3% with 25% LPG respectively at 1600 rpm as shown in Fig. 12. The main reason for this decrease in NOx at this speed is that overall air–fuel ratio was very close to the rich mixture. Another factor contributing to decrease in NOx could be the

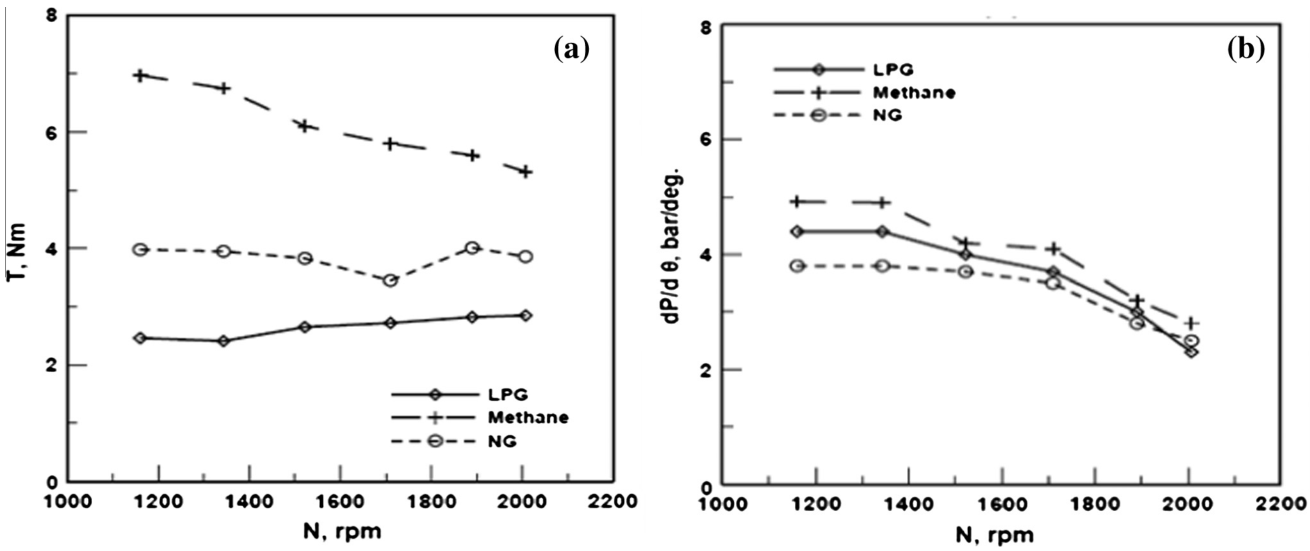

Figure 11 Effects of engine speed on torque (a) and noise (b); IT = 35°BTDC, CR = 22, md = (0.37–0.47) kg/h [58].

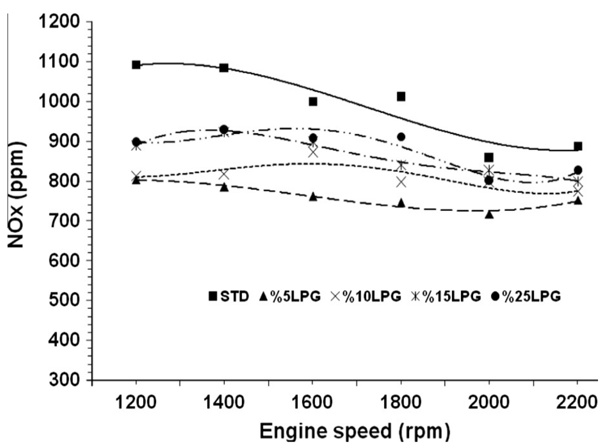

Figure 12 Variation of NOx emission for various rate of LPG at different speeds [66].

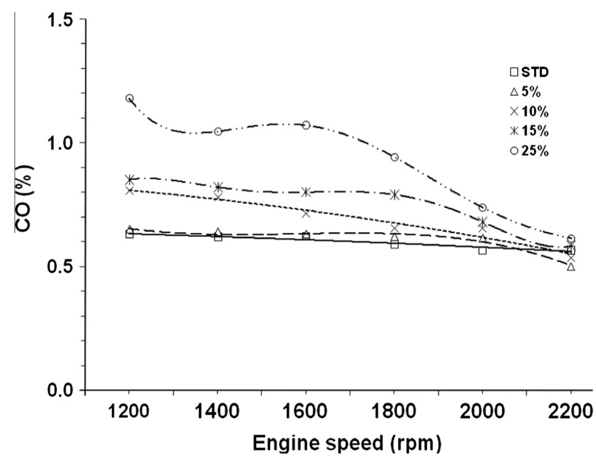

Figure 13 Variation of CO emission for various rate of LPG at different speeds [66].

fact that the heat release occurred later phase with addition compared to that of standard diesel engine. The effect of post flame gases, which dominates the NOx formation may decrease since the piston moved further toward bottom dead center (BDC) during expansion period leading to decrease in gas temperature and residence time for NOx formation. The results for CO emissions of the engine, which was operated under both Diesel fuel and Diesel fuel with different LPG compositions, are shown in Fig. 13 for all the speeds tested under full load condition. It is observed that CO emissions significantly increased between 1200 and 1800 rpm with 25% LPG rate. While CO emissions were measured as 0.6% with the diesel fuel operation at this speed, the emissions with the other LPG compositions at the same speed were 0.65% with 5% LPG, 0.8% with 10% LPG, 0.85% with 15% LPG and 1.2% with 25% LPG. This can be attributed to the volumetric efficiencies of the LPG mixtures which are lower than that of standard diesel engine. It is well known that the rate of CO formation is a function of unburned fuel and mixture temperature during combustion, since both factors control the rate of fuel decomposition and oxidation. Decreases in volumetric efficiency and richer mixture seem to significantly affect the rate of fuel decomposition and oxidation adversely. Similarly the HC emissions increased with increase of the LPG mass fraction at all the engine speeds tested. It is seen from Fig. 14 that HC emissions were higher for all the LPG ratios compared to the pure diesel fuel operation. The reason of increase in HC emissions at low speeds can be explained by lower volumetric efficiency. Richer mixture and lower volumetric efficiency lead to decrease combustion quality under lower cylinder temperature and hence, lead to increase the unburned HC emissions [66].

5.3. Effect of pilot fuel quantity

By increasing the amount of pilot diesel quantity which forms a strong ignition source, the poor performance and emission characteristic at low loads can be improved marginally, while increasing the amount of pilot fuel at high loads led to increase

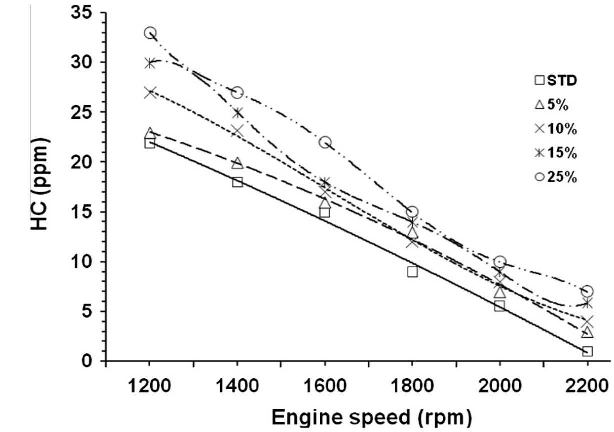

Figure 14 Variation of HC emission for various rate of LPG at different speeds [66].

in the combustion pressure and cause the early knocking. In dual fuel operation, the pilot fuel is normally introduced into the cylinder charge through the use of the conventional diesel injection system. The quantity of pilot fuel injected is often very small. It is known that most diesel fuel injection systems experience poor atomization and combustion when the amount of fuel injected per cycle is reduced below 5–10% of the maximum design level. Another important factor is to ensure that the injector does not get overheated when small pilot fuel quantities are injected, while significantly energy release through LPG in the combustion is proceeding within the cylinder. Increasing the quantity of pilot diesel fuel increases the torque output as shown in Fig. 15 (a) by means of larger size of pilot mixture envelope with greater entrainment of the LPG fuel, greater energy release on ignition and a larger number of ignition centers requiring shorter flame travels, higher rate of heat transfer to the unburned gaseous fuel–air mixture. These factors tend to increase the power output and thermal efficiency of the LPG–diesel dual fuel engine. The maximum pressure raise rate, as seen in Fig. 15 (b) is generally reduced when the pilot fuel quantity is increased. The decrease in the combustion noise when the pilot fuel mass is first increased may be postulated to the increase in flame volume resulted by the increase in pilot fuel mass which would burn the gaseous fuel smoothly and at lower rate of combustion. The increase in the initial pilot flame volume may have caused the air/gas mixture to ignite from more mini flames with smaller air/fuel mixture pockets ignited. However, when the pilot fuel mass increased beyond certain amount, the ignition delay period of pilot diesel may increase and may increase the pressure raise rate for the gas/air mixture [58,59,67].

The effect of variation of pilot fuel quantity with different intake charge temperatures is investigated by Poonia et al. in a single cylinder constant speed diesel engine, which indicates the effect of pilot fuel quantity on brake thermal efficiency at an intake temperature of 34 °C, at different loads as shown in Fig. 16. It is observed that up to 60% load, brake thermal efficiency increases with increase in pilot fuel quantity. However, at 80% and 100% loads, with large pilots, it decreases. At high outputs, large pilot quantity leads to the high rates of combustion. This is the reason for decrease in efficiency when the pilot quantity is increased beyond optimum values. As load increases, brake thermal efficiency also increases. It is observed from Fig. 17 that the ignition delay period in the dual fuel mode is always longer than in the diesel mode. The gaseous fuel added can undergo significant reactions during the compression stroke leading to partial oxidation products. This can affect the pre-ignition processes of the pilot adversely [60].

Fig. 18 shows the rate of heat release at 80% of full load at the optimum intake temperature of 50 °C. It is seen that in pilot fuel quantity of 10.7 mg/cycle, three phases of combustion are clear. The first stage of heat release is strongly dependent on the pilot fuel quantity. The peak heat release rate in the second stage decreases with a reduction in the pilot quantity, i.e., with an increase in the LPG air ratio. At a high pilot quantity of 10.7 mg/cycle the rate of heat release in the first and second stages rises to very high values. Hence it can be concluded that with higher pilot fuel quantities knocking occurs mainly

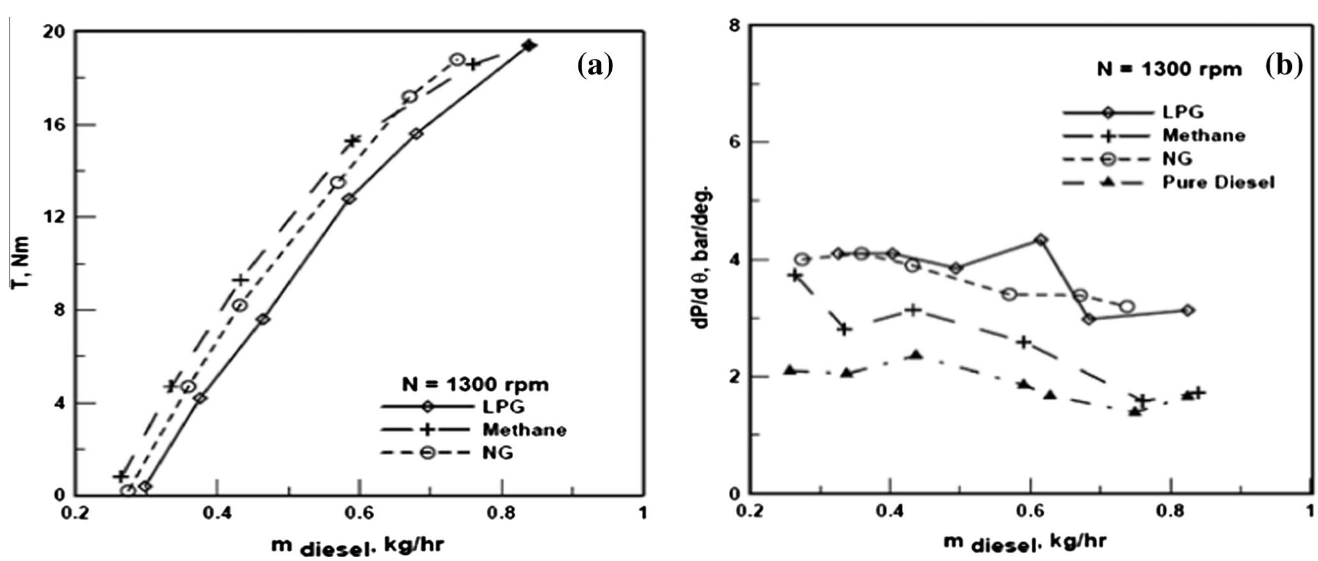

Figure 15 Effects of pilot fuel mass on torque (a) and noise (b); N = 1300 rpm, IT = 35°BTDC, CR = 22 [58].

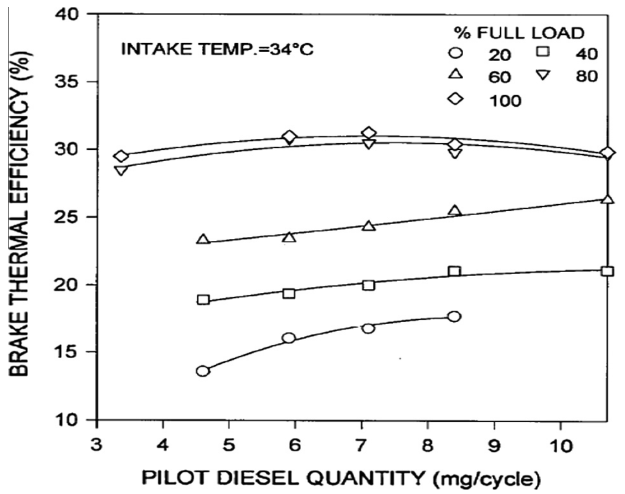

Figure 16 Effect of brake thermal efficiency vs pilot fuel quantity [44].

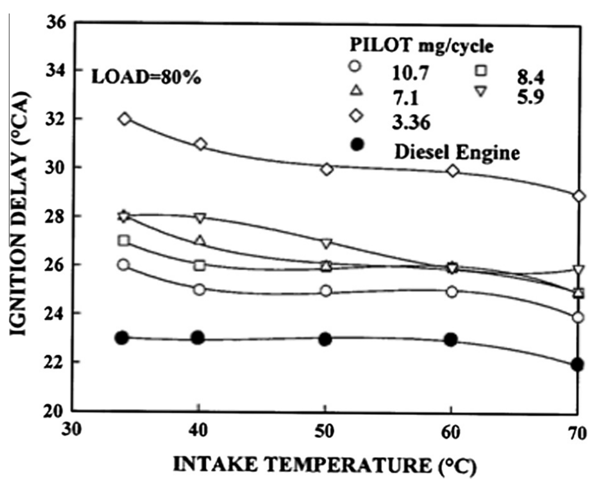

Figure 17 Effect of ignition delay vs intake temperature [60].

because of rapid combustion in the first and second stages. An increase in the pilot fuel quantity leads to a decrease in the ignition delay and also affects the amount of diesel burning in the premixed mode. It also increases the amount of gas entrained with the pilot. All these factors influence the heat released in the first phase of combustion of dual fuel engine [44,31,70].

Effect of pilot fuel quantity on performance of dual fuel engine is investigated by Abd et al. in a Ricardo E6 single cylinder engine. With larger pilot fuel the combustion of the LPG fuel is better leading to higher mass fraction burnt and higher brake thermal efficiency in dual fuel mode as compared to the corresponding diesel operation. To avoid knock, which occurs at high outputs in dual fuel engines, low pilot fuel quantities and low intake temperatures are better. At higher loads the combustion duration was longer with the high pilot as compared to lower pilot fuels, due to relatively slow diffusive burning of the pilot diesel in the later part of the combustion process [56]. With increasing pilot fuel quantity, the volume of the charge that is affected by the combustion of the pilot fuel envelope will increase thus increasing the burned fraction of the gaseous fuel and accordingly, decreasing the pollutants emitted in the exhaust. A larger pilot fuel quantity provides, in principle, a large pilot fuel envelope and a greater multitude of ignition centers with larger reaction zones within the overall, very lean, gaseous fuel air mixture. Moreover, the flame propagation path from each ignition center within the charge becomes relatively shorter and thus, combustion is better. Furthermore, for low load involving very lean gaseous fuel and air mixtures, the employment of a large pilot fuel quantity improves the injection characteristic which leads to stable combustion of the pilot fuel that contributes to combustion of the gaseous fuel without hunting. At higher loads, when the gaseous fuel concentration in the air charge is below the lean combustion limit, the flame is able to propagate through most of the combustion chamber unaided and during that period varying the pilot fuel quantity has little effect. The change in oxidation reactions from unsuccessful to successful flame propagation reduces the hydrocarbons and carbon monoxide emissions slightly as shown in Figs. 19 and 20.

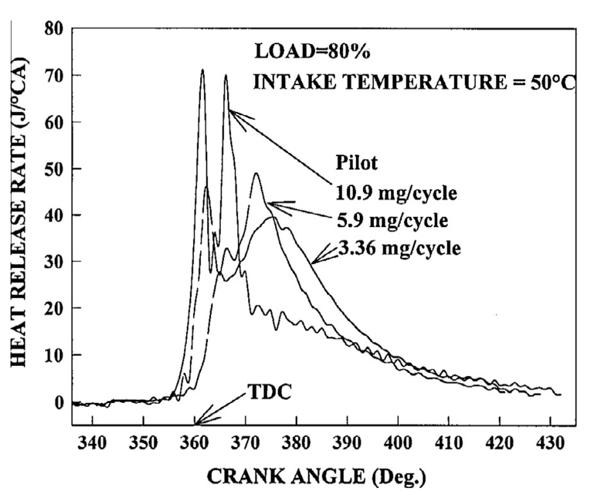

Figure 18 Effect of heat release rate for the different pilot quantity [60].

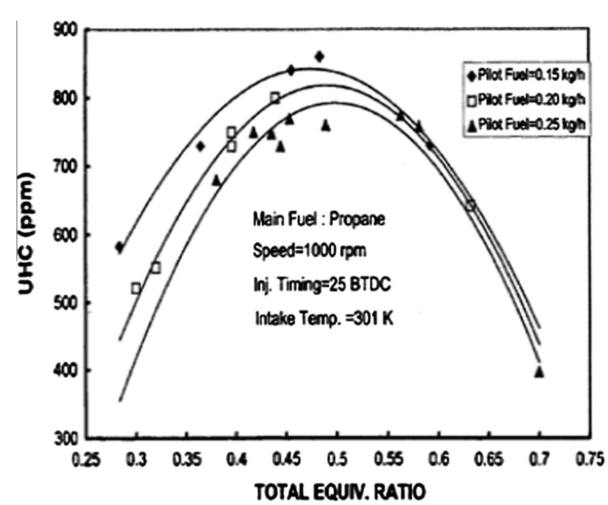

Figure 19 Variations of unburned hydrocarbons with total equivalence ratio and pilot fuel quantities [68].

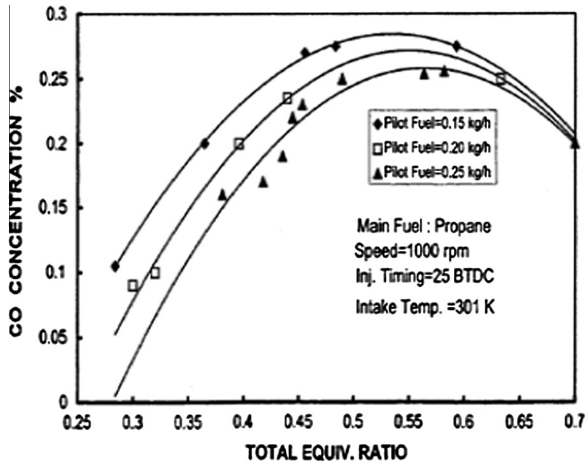

Figure 20 Variations of carbon monoxide with total equivalence ratio and pilot fuel quantities [68].

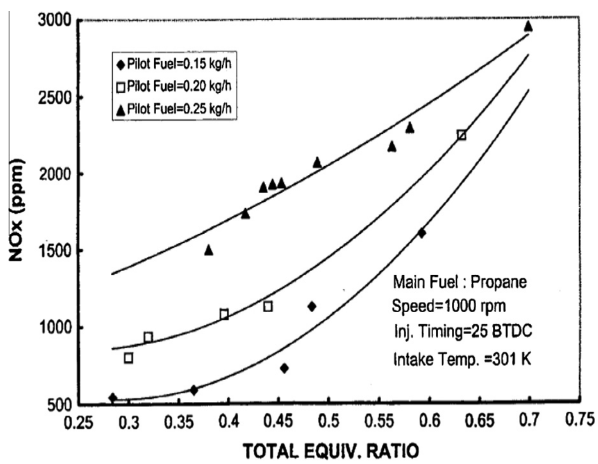

Figure 21 Variations of NOx with total equivalence ratio and pilot fuel quantities [68].

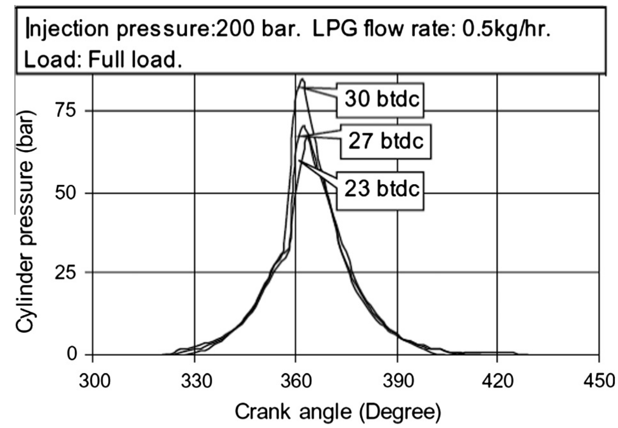

Figure 23 Pressure vs crank angle for different injection timings in dual fuel mode [72].

Gaseous fuel concentration in the air charge is above the lean combustion limit, and the flame is able to propagate through most of the combustion chamber unaided and varying the pilot fuel. The experimental results of Fig. 21 confirm such explanation by showing that the production of nitrogen oxides is influenced markedly by both the quantity of the pilot fuel employed and the overall equivalence ratio. The use of large pilot fuel quantities and high charge equivalence ratios results in a significant increase in the production of nitrogen oxides. In dual fuel the effective size of the combustion zone, which relates to the size of the pilot fuel zone, is another important factor that determines the quantity of nitrogen oxides produced. On this basis and for the same total equivalence ratio increasing the pilot fuel quantity increases the charge temperature which tends to increase the formation of NOx [68,69,103].

5.4. Effect on injection timing of pilot diesel

Injection timing of pilot fuel plays a vital role in the combustion process of LPG–diesel dual fuel engine. Because of the

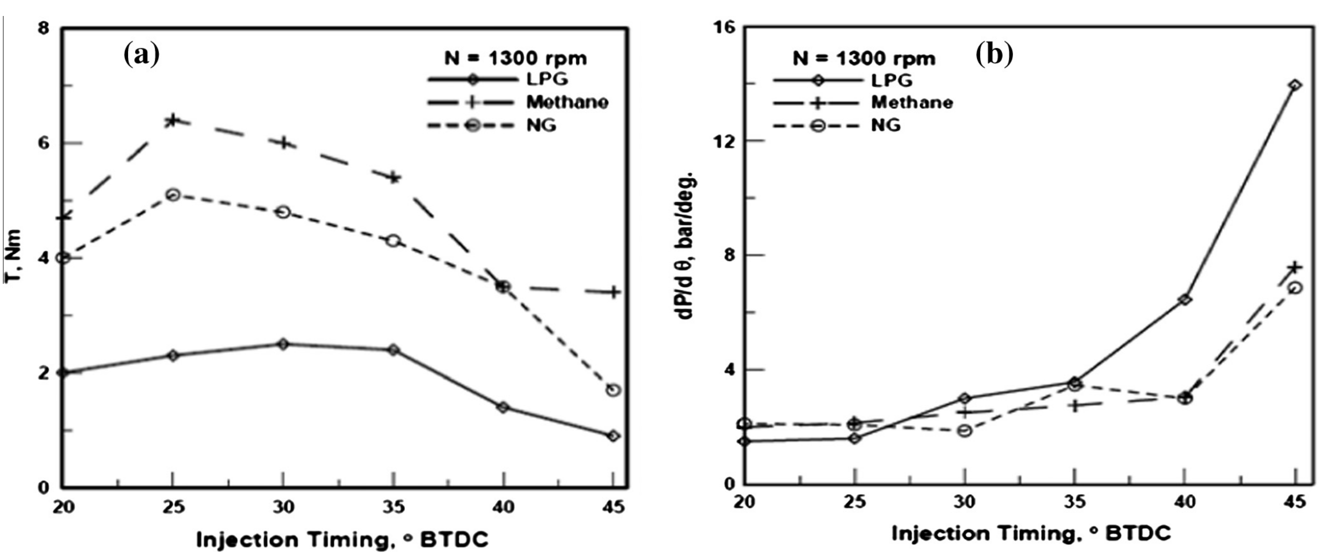

Figure 22 Effects of pilot fuel injection timing on torque (a) and noise (b); N = 1300 rpm, CR = 22, md = 0.37 kg/h [58].

presence of the second fuel (LPG) the ignition delay for dual fuel is higher than that in full diesel operation. Advancing the injection timing for likewise caused an earlier start of combustion relative to TDC. The advance in pilot fuel injection results in an increase in ignition delay period of the fuel which in turn will lead to the combustion of large diesel fuel mass or bigger flame to propagate at higher speed. This may have caused the combustion to start earlier and the maximum pressure to increase. Because of this cylinder charge being compressed as the piston moves to TDC, had relatively higher temperature and leads to reduce the emission especially the HC. Retarding the injection timing decreases the peak cylinder pressure because more of the fuel burns after TDC. This is because, the pilot fuel combustion is delayed and thus, the temperature of the mixture is not enough to propagate the flame in the whole gaseous fuel–air mixture and consequently, incomplete combustion of the gaseous fuel mixture takes place. The charge temperature increases with advancing the injection timing of the pilot fuel and the associated higher energy release rates of the mixture. Similarly, the rates of pressure raise during the combustion of the gaseous fuel increase with advancing the injection timing of the pilot fuel [71].

It may be seen from Fig. 22 (a) that the torque output vs pilot fuel injection timing, shows the highest torque at certain timing and it decreases at earlier or later timing. Too much earlier injection of pilot causes the maximum pressure to increase and to occur before top dead center in compression stroke which in turn reduces the maximum pressure during expansion stroke and then the torque output reduces. The pressure raise rate, as may be seen in Fig. 22 (b) generally increases as the pilot diesel injection advance increases. This may be attributed to the increase in ignition delay of the diesel fuel, since the liquid fuel is injected earlier in lower air pressure and temperature. The longer delay period would result in higher pressure raise rate. With the presence of gaseous fuel in the mixture, any advance in pilot injection would result in longer ignition delay period and the pressure raise rate is expected to increase. For the late injection of pilot, 20–25°BTDC the combustion noise is low. However, as the injection advance increases, 25–40°BTDC, the dual fuel engine produces higher rate of pressure raise [58,59].

Sudir et al. studied the effects of injection timing on single cylinder constant speed LPG–diesel dual fuel engine by retarding and advancing it. They found that advancing the injection timing beyond 30° btdc resulted in severe knocking even at part load operation of dual fuel engine. When injection timing was retarded bellow 23° btdc engine tended to suffer from poor efficiency and was not able to take up full load at the rated speed. Due to above reasons the experiments were restricted only for 23°, 27° and 30° btdc static injection timings. It is observed from Fig. 23 that advancing the injection timing (30° btdc) resulted in increases of cylinder pressure in dual fuel operation. This is due to early ignition of diesel which results in higher peak pressure and this occurs near to top dead center. Advancing the injection timing of the pilot fuel to 30° btdc increases the residence time and the activity of the partial oxidation reactions of the injected fuel. This will result in better and complete combustion of primary and secondary fuel–air mixtures.

For the injection timing of 30° btdc decrease in Smoke density is higher when compared to other injection timings as shown in Fig. 24. This may be due to burning away of particulate in the course of LPG–air premixed mixture. This is likely because of higher pressure and temperature conditions, presence of flame and also availability of air for oxidation. Atmospheric nitrogen exists as a stable diatomic molecule at low temperatures and only very small traces of oxides of nitrogen are found. However, at high temperature that occurs in the combustion chamber of an engine, some diatomic nitrogen brakes down to monatomic nitrogen which is reactive. In addition to temperature, the formation of NOx depends on pressure and combustion time within the cylinder. Hence it becomes obvious that on advancing the injection timing to 30° btdc, NOx emission increases. From Fig. 25 it is seen that for injection timing of 30° btdc, NOx emission is higher compared to other injection timings. At full load operation peak pressure and the combustion temperature increase leads to increase in NOx emission to a considerably higher value than that in full diesel operation [72,94].

Advancing the injection timing leads to reduce the unburned hydrocarbons due longer ignition delay period as shown in Fig. 26 of Ricardo E6 single cylinder engine in a dual fuel mode. The longer ignition delay could have allowed a fuller spray penetration and development, creating a larger amount of the pilot fuel–air–gaseous fuel mixture (or flame propagation region) prior to ignition. The higher combustion

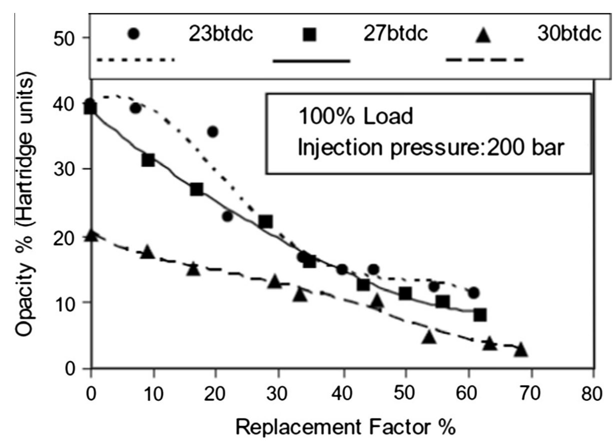

Figure 24 Effect of different Injection timings on smoke density [72].

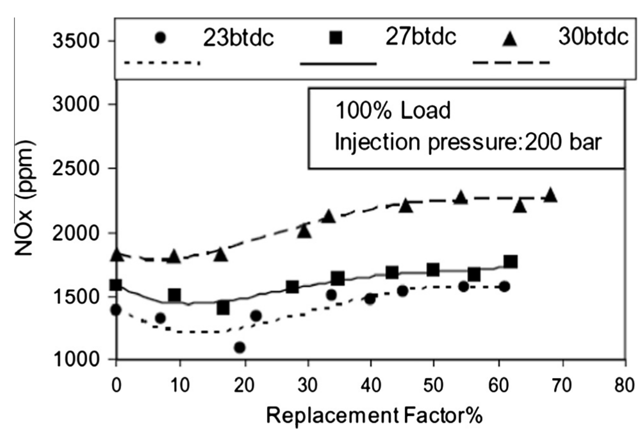

Figure 25 Effect of different injection timings on NOx [72].

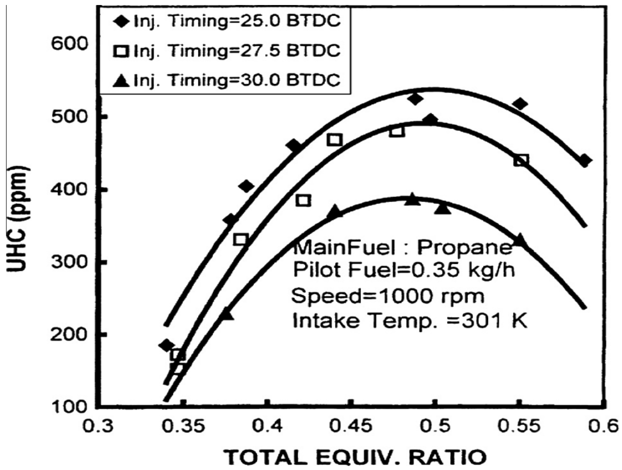

Figure 26 Effect of different injection timings on unburned hydrocarbons [71].

rates of the larger premixed regions yielded higher combustion temperatures thus lowering the unburned hydrocarbons. Advancing the injection timing likewise caused an earlier start of combustion relative to TDC, which had relatively higher temperature and thus lower unburned hydrocarbons. Better overall combustion may also have been due to the longer period of high temperatures within the cylinder and the activity of the partial oxidation reaction with bigger injection advance, which leads to reduce the carbon monoxide emissions as shown in Fig. 27 effectively, and widens the lower combustion limit boundary of the overall lean mixtures [71].

5.5. Effect of intake manifold conditions

Dual fuel operation with LPG fuel can yield a high thermal efficiency, almost comparable to the same engine operating on diesel fuel at higher loads. However, engine performance and emissions suffer at low loads when operating in the dual fuel mode. The main reason for this poor low load performance is very lean LPG–air mixtures are inducted in the case of dual fuel engine. These lean mixtures at low loads are hard to ignite and slow to burn. In this mixture the flame propagation is not complete and most of the fresh air–gas mixture remains unburnt resulting in lower brake thermal efficiency and high HC emission [73,74]. There are lot of studies carried out of modifying the intake charge conditions by, using glow plug, preheating the intake charges, throttling the intake charges and EGR to improve the low load performances of LPG–diesel dual fuel engine [2,36,60,75–81].

The effect of intake charge preheating is studied by Poonia et al. in a single cylinder constant speed engine. They found that increase in intake temperature from 34 °C to 70 °C marginally affects the peak heat release rate in the first stage of combustion as shown in Fig. 28. Total heat release during the first stage of combustion is approximately 15–20% of the total energy supplied, which is around 50% of the total pilot fuel energy. Thus it is clear that the entire pilot does not burn in the first stage itself and some of the pilot fuel along with the inducted gaseous fuel burns in the second stage of heat release. It is also seen from the figure that as the intake temperature raises the second stage of combustion dominates at 80% load. Thus at higher temperatures the second stage becomes significant at high outputs due to rapid combustion of the gaseous fuel. It appears that the LPG–air mixture near the pilot zone could even ‘auto-ignite’ at higher temperatures and outputs. The magnitude of the peak of the heat release rate during this stage at high outputs seems to be a strong function of the intake temperature. At lower intake temperatures as shown in figure the second and third stages of heat release are not distinctive. The heat release rate and mass fraction burned in the first stage of combustion do not change much with intake temperature.

At 20% load, as seen in Fig. 29 (a) the peak cycle pressure decreases with a decrease in the pilot fuel quantity and increases in intake temperature. The peak cycle pressure at light load is mainly controlled by the first phase of combustion namely premixed burning of part or whole of the pilot fuel along with a small part of gas entrained in the spray. As the intake temperature increases at any given pilot quantity the peak pressure decreases probably due to decrease in the delay period. The peak pressures at 80% of full load Fig. 29 (b) in

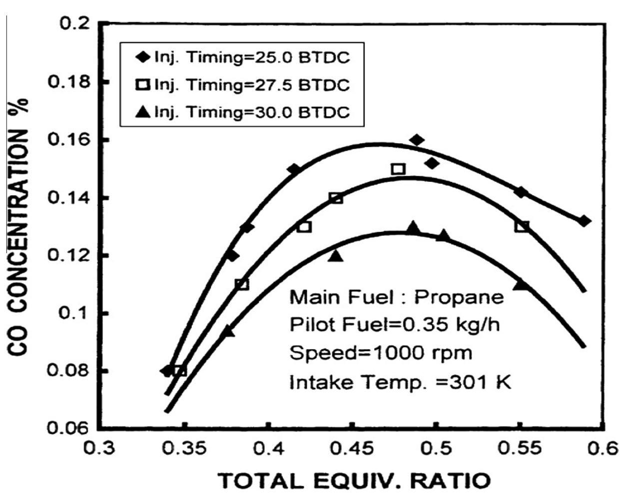

Figure 27 Effect of different injection timings on CO concentration [71].

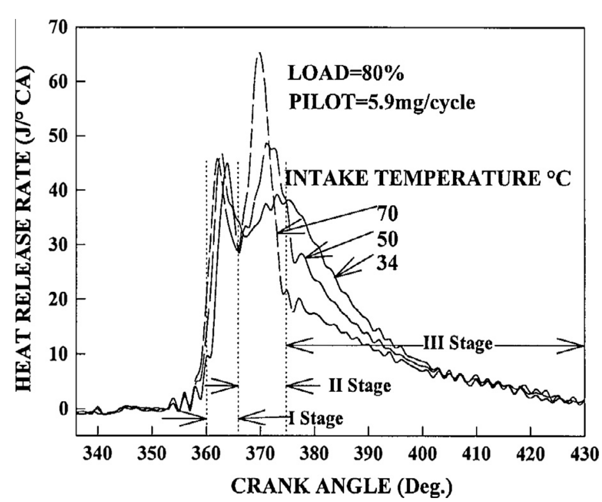

Figure 28 Effect of intake temperature on heat release [60].

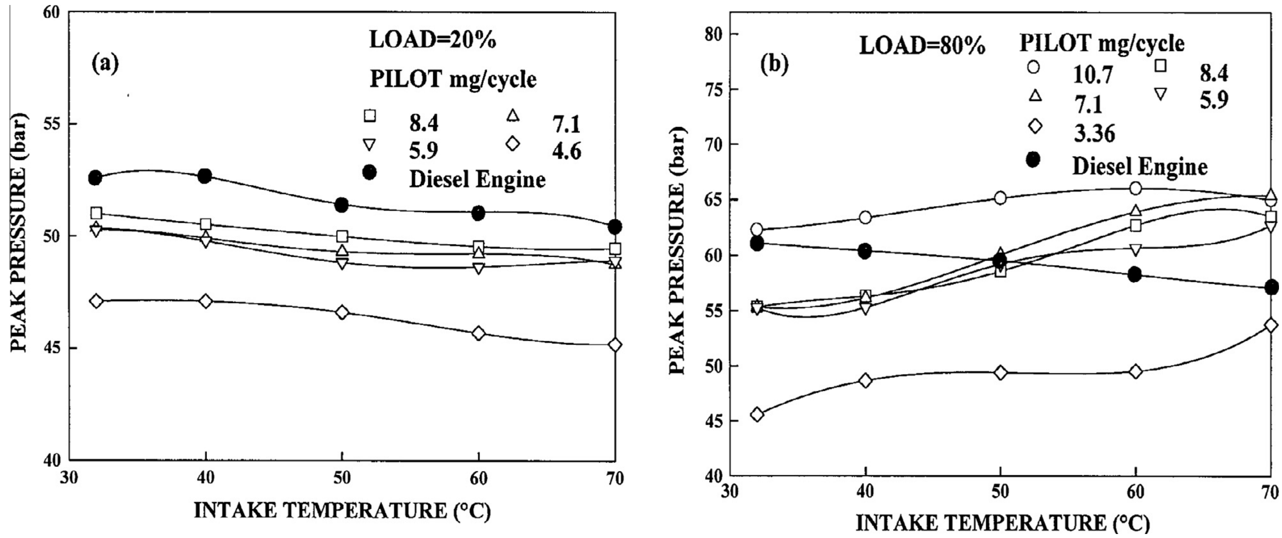

Figure 29 Effect of intake temperature on peak pressure for 20% and 80% load [60].

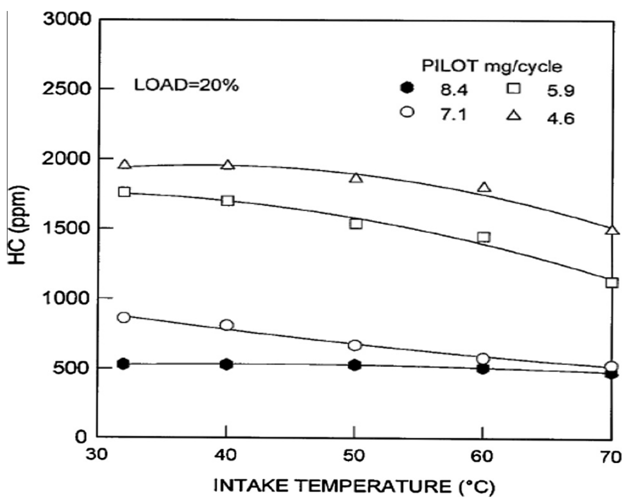

Figure 30 Effect of intake temperature on HC emissions [44].

the dual fuel mode with the largest pilot (10.7 mg/cycle) are higher than straight diesel operation at every intake temperature whereas with the smallest pilot (3.36 mg/cycle) the peak pressures are observed to be lower than straight diesel operation. At low pilots heat is mainly released by flame propagation. As the intake temperature increases above 50 °C, the second stage of combustion starts dominating resulting in higher peak pressure as compared to the straight diesel mode. The peak pressure at optimum pilot (5.9 mg/cycle) and 60 °C intake temperature reached as high as 63 bar as compared to 57 bar in the straight diesel mode at 80% of full load [60,81].

As seen in Fig. 30 at a low output of 20% of full load, HC in the exhaust increases as the pilot fuel quantity is decreased. When a small pilot quantity is used, a significant portion of the gaseous fuel does not burn completely as the ignition source is weak. HC emission level decreases with increase in intake temperature. This decrease is more pronounced at lower pilots

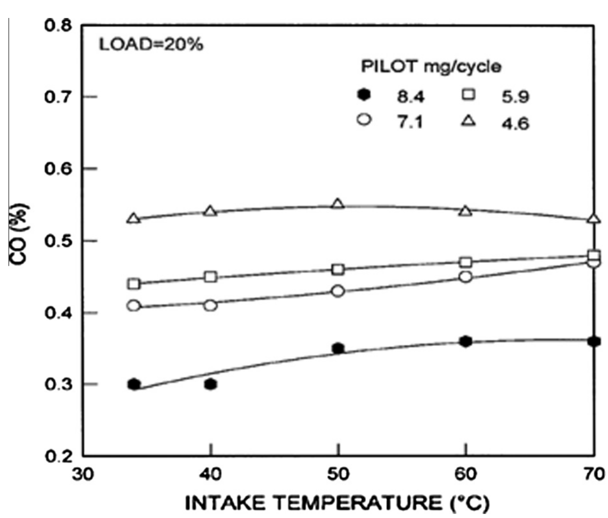

Figure 31 Effect of intake temperature on CO emissions [44].

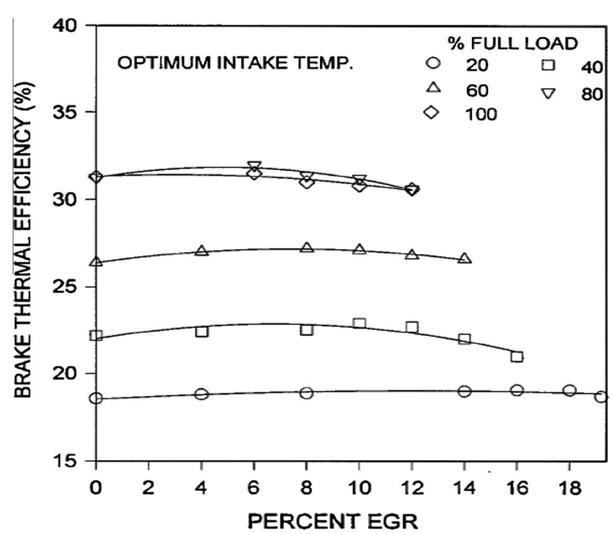

Figure 32 Effect of intake temperature on thermal efficiency [44].

Table 2 Values of EGR (%) and intake temperatures for different loads [44].

Load	Intake temperature (°C)	Pilot (mg/cycle)	EGR (%)
20	70	8.4	18
40	70	10.7	10
60	60	10.7	8
80	60	5.9	6
100	40	7.1	4

where the combustion is poor. At 20% load, the maximum intake temperature (70 °C) resulted in the minimum HC level. Raise in the intake temperature leads to increased pre-flame reactions in the homogeneous gas air mixture and elevates the combustion rate leading to more complete combustion and low HC levels. Carbon monoxide level generally increases with increase in the intake temperature at 20% load as shown in Fig. 31. At light loads only the LPG–air mixture near the pilot is burned. Thus some partial oxidation products (which include carbon monoxide) may appear in the exhaust. At higher intake temperatures, the concentration of these partial oxidation products could increase. Also at these conditions the inducted charge will be rich due to drop in volumetric efficiency. This is thought to be the reason for the raise in the CO emission level with intake temperature at 20% load [44].

Effect of EGR on brake thermal efficiency at different loads for a single cylinder constant speed engine is illustrated in Fig. 32. At every load with the addition of EGR, brake thermal efficiency first increases and then decreases. EGR improves the combustion by recirculation of active radicals which will enhance pre-flame reactions that take place during the compression stroke. However, too large a percentage of EGR affects flame propagation adversely and probably this is the reason for decrease in efficiency at high outputs when EGR quantity is increased beyond optimum values. There is an improvement in brake thermal efficiency up to 80% load whereas at 100% load, the improvement is not significant. The optimum percentage of EGR as mentioned in Table 2 decreases as the load increases. At optimum EGR, HC emissions reduce at low and medium outputs as seen in Fig. 33. At 20% load HC emissions reduced from 450 ppm to 390 ppm. At high EGR rates slow flame propagation and flame quenching lead to high HC levels. EGR displaces fresh air and leads to higher percentage of CO emissions at every load as shown in Fig. 34. Thus LPG–diesel dual fuel mode EGR favorably alters brake thermal efficiency and HC emissions but affects CO emission adversely [44].

5.6. Effect of compression ratio

For LPG–diesel dual fuel operation occurrence of knock onset and ignition failure for the three compression ratios of 18, 20 and 22 for a Ricardo E6 single cylinder engine is shown in Fig. 35 (a). It may be seen that for higher compression ratio of 22, knock starts very early, at an engine torque of about 8.1 N m, while any increase in gaseous fuel quantity increases the knocking intensity and starts to reduce the output torque until ignition failure occurs at about 7.85 N m and output then drops sharply. As the compression ratio is reduced to 20, the torque at which knocking starts is shifted to a higher value of about 17.6 N m, and ignition failure occurs at about 17 N m. For lower compression ratio of 18, the knocking limit is shifted to about 20 N m, and the ignition failure is shifted to 18.5 N m. It may be concluded that reducing the compression ratio has resulted in retarding the occurrence of knock onset in the dual fuel engine (from 8.1 N m to 17.6 N m to 20 N m) and also extended the ignition limits greatly (from 7.85 N m to 17 N m to 18.5 N m). This may be postulated to the early knocking at high compression ratios associated with higher pressure and temperatures and lower self-ignition temperatures of LPG. For extended ignition limits and knock-free operation of the dual fuel, the compression ratio has to be then reduced to lower values. Fig. 35 (b) illustrates the maximum pressure raise rate at different compression ratios for the LPG–diesel dual fuel mode. It may be seen that increasing the compression ratio generally increases the combustion noise due to the higher self-ignition possibility of the gaseous fuels at higher pressures and temperatures. As the compression ratio is reduced the combustion noise is also reduced and ignition limits are extended [58,59,65].

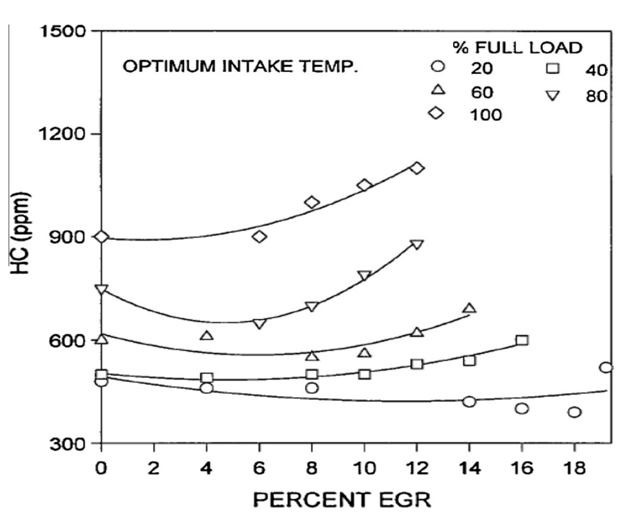

Figure 33 Effect of EGR on HC emissions [44].

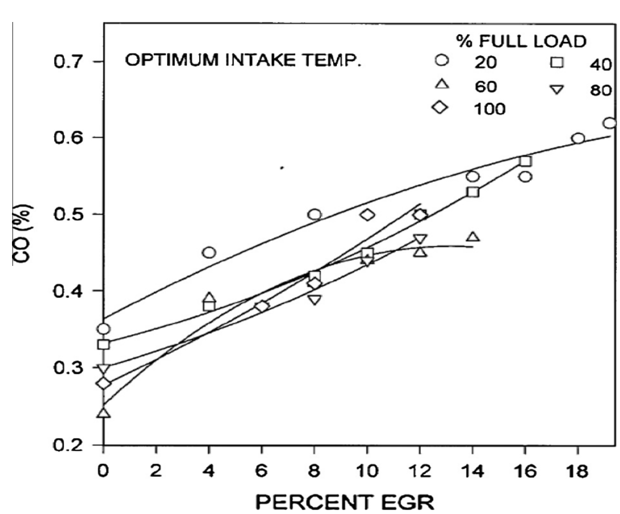

Figure 34 Effect of EGR on CO emissions [44].

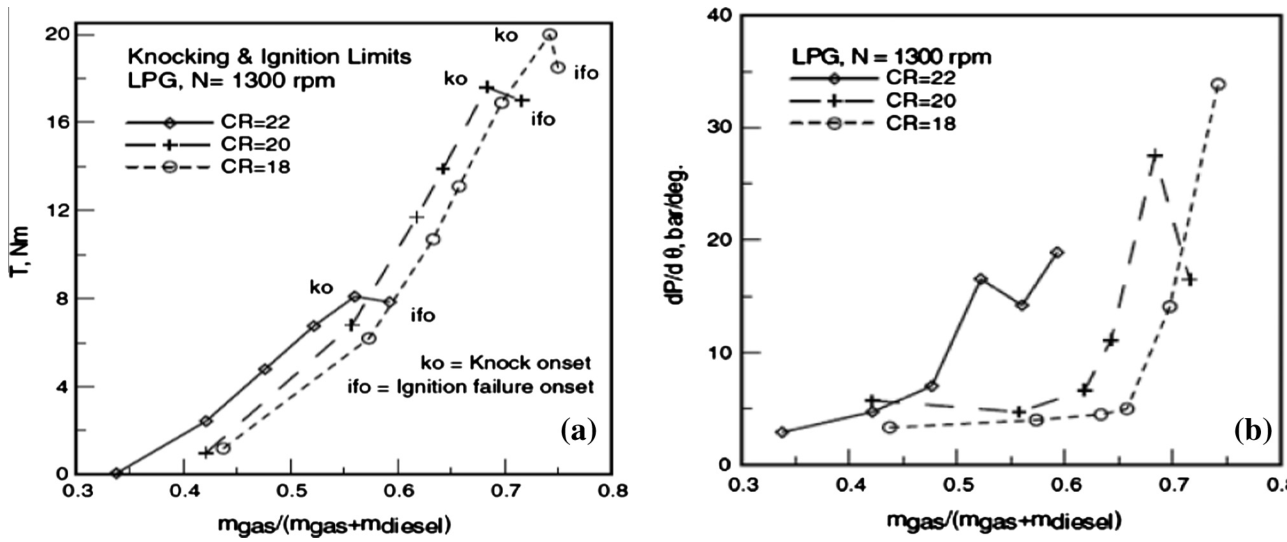

Figure 35 Effect of compression ratio on torque (a) and combustion noise (b) for LPG $N = 1300$ rpm, IT = 35°BTDC, md = 0.37 kg/h [58].

Table 3 Tested LPG fuels composition (% vol.) [82].

Tested fuel	Propane	Butane
Fuel #1	100	–
Fuel #2	90	10
Fuel #3	70	30
Fuel #4	50	50
Fuel #5	30	70

5.7. Effect of varying the compositions of the gaseous fuel

Some theoretical and experimental investigations were carried out to examine the performances of a LPG–diesel dual fuel engine at various LPG compositions, such as propane to butane ratio [8,82–86] and inducted mixture of hydrogen and LPG [1,87–91]. To understand how LPG fuel composition affects the exhaust emissions and engine performance in a twin cylinder dual fuel engine, is investigated by Saleh for the five different compositions of LPG fuel as listed in Table 3. The effect of various engine loads on the thermal conversion efficiency of various LPG blends comparable to the pure propane (fuel #1) in dual fuel operation is shown in Fig. 36. It will be seen that the efficiency increases with load for all various LPG blends. As shown in the figure, three LPG blends behaved very similarly, and there is no appreciable difference between the thermal conversion efficiency with fuel #2 of 10% butane, fuel #3 of 30% butane and pure propane (fuel #1) at full load but at 25% load the thermal conversion efficiency with fuel #3 was lower than that with pure propane about 3%. The thermal conversion efficiency with fuel #4 of 50% butane, and fuel #5 of 70% butane was lower than that with fuel #1 (pure propane) by about 2%, 3.6% at full load.

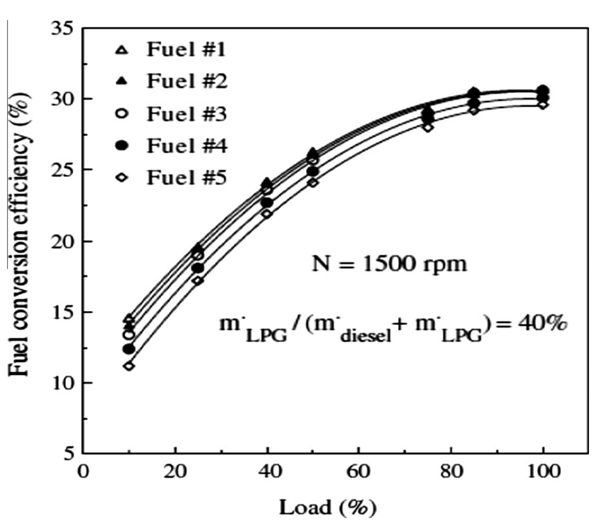

Figure 36 Variation of fuel conversion efficiency vs engine load for LPG blends [82].

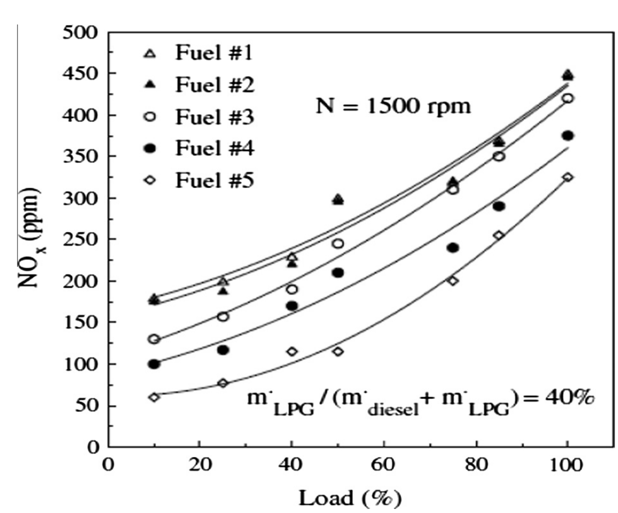

Figure 37 Variation of NOx concentration with engine load for LPG blends [82].

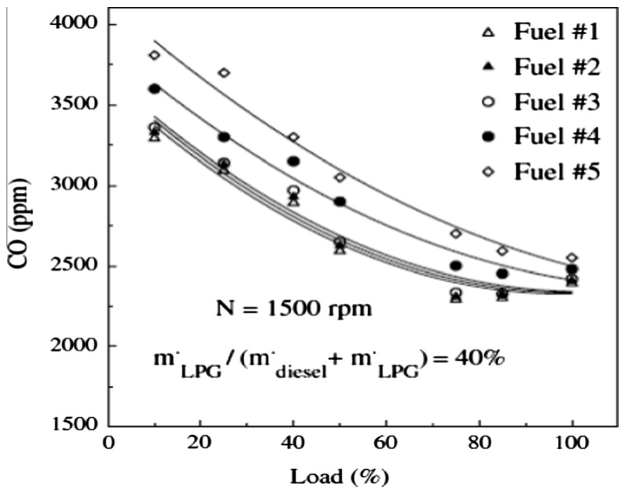

Figure 38 Variation of CO concentration with engine load for LPG blends [82].

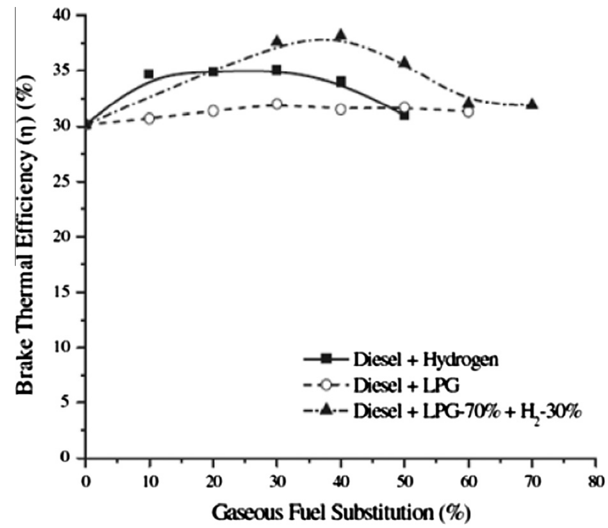

Figure 39 Comparison of brake thermal efficiency for the cases II, III and IV at 80% load condition [88].

Table 4 Test matrix of addition H_2 with LPG [88].

Case no	Primary fuel	Secondary fuel	Load percentage (%)
I	Diesel	–	10, 20, 30, 40, 60, 80
II	Diesel	Hydrogen	10, 20, 30, 40, 60, 80
III	Diesel	LPG	10, 20, 30, 40, 60, 80
IV	Diesel	LPG + Hydrogen	10, 20, 30, 40, 60, 80

The main reason for the lower thermal conversion efficiency may be explained that a change in ignition delay, nature of the combustion process and heat release are caused by changes in physical and chemical properties due to different mixing ratios of propane and butane.

Influence of the various LPG blends on NOx emissions comparable to the pure propane (fuel #1) in dual fuel operation is shown in Fig. 37. For all LPG blends, NOx emissions increased as the load on the engine was increased. It is shown that, the increase of butane percent produced a corresponding decrease in NOx emissions. NOx with fuel #2 of 10% butane, fuel #3 of 30% butane, fuel #4 of 50% butane and fuel #5 of 70% butane was lower than that with tested fuel #1 (pure propane) by approximately, 1%, 6.7%, 16.7% and 27.8%, respectively at full load. In general, high amounts of oxygen and high temperatures will result in high levels of NOx formation; these conditions are not present in the combustion chamber with increasing the butane percent. As mentioned above, the increase in butane percent tend to increase in the ignition delay which in turn is linked with the decrease of the maximum pressure reached during the combustion. This results in temperature reduction inside the cylinder. The effect of various LPG blends on the CO emissions under the different loadings comparable to the pure propane (fuel #1) in dual fuel operation is presented in Fig. 38. As expected, increasing the butane percent by 30%, 50%, and 70% in the LPG composition produced the highest carbon monoxide emissions especially at part loads, with percentage of 1.3% corresponding to CO of 3140 ppm, 6.5% corresponding to CO of 3300 ppm and 19% corresponding to CO of 3700 ppm at 25% load, respectively. At full load, it is observed that test fuel #3 with 30% butane was with percentage of 1.2% corresponding to CO of 2430 ppm. There is no appreciable difference between CO emissions of fuel #2 with 10% of butane and pure propane (fuel #1). Increasing the butane percent was expected to produce more CO emissions than pure propane gas due to its higher carbon to hydrogen ratio, longer ignition delays, lower flame temperatures and flame speeds and this was reflected in the results [82].

Lata et al. have conducted the experiments on four cylinder diesel engine in dual fuel mode by the addition of hydrogen along with LPG as a primary gaseous fuel and diesel as pilot fuel as shown in the test matrix Table 4 for the determination of performance, combustion and emission characteristics [1,87–90]. They found that mixture combination in Case IV of LPG:hydrogen ratio (70:30) has the raise in brake thermal efficiency of 25%, 9% and 22% as compared to Cases I, II and III, respectively at 80% load conditions as shown in Fig. 39. LPG flame tends to become unstable, while, hydrogen–air flames tend to be stable. Therefore hydrogen fraction leads to stabilization of flame. This nature may also result from the opposite diffusion behavior of propane (LPG) and hydrogen. Diffusivity of hydrogen (0.61 cm^2/s) is more than LPG (0.12 cm^2/s) in air. Hydrogen has strong buoyancy and high diffusivity while LPG has lower flame speed and narrower flammability limits as compared to hydrogen [1].

Variation of unburned hydrocarbon at 10% and 80% load conditions for the Cases II, III and IV is shown in Fig. 40 (a) and (b). At 10% load condition, Cases II, III and IV show HC emissions of 6.86 g/kW h, 5.9 g/kW h and 6.94 g/kW h, respectively, as compared to 1.72 g/kW h of Case I. This may be due to reduction in pilot quantity, which causes poor ignition of gaseous fuel and inducted mixture is too lean to burn. At 80% load condition, Cases II, III and IV show maximum HC emission of 5.64 g/kW h, 4.57 g/kW h and 1.07 g/kW h, respectively, as compared to 1.8 g/kW h of Case I operation. At higher load condition Case IV shows lower HC than Case I due to better and fast combustion rate leading to more

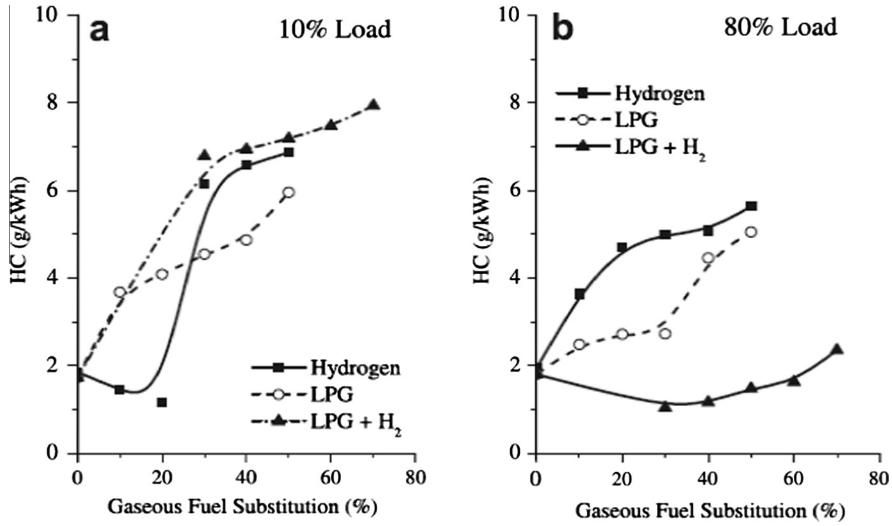

Figure 40 Un-burnt HC (g/kW h) vs diesel + gaseous fuels substitution (%) [88].

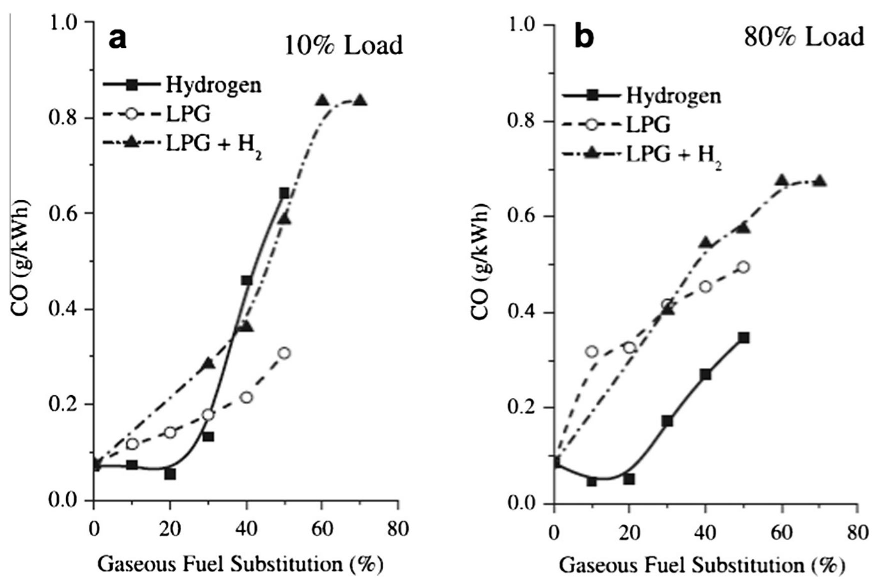

Figure 41 CO vs diesel + gaseous fuels substitution (%) [88].

complete combustion and hence low HC emissions. While, in Cases II and III, at a low pilot quantity (average 4.15 mg/cycle/cylinder) HC emission is high at gaseous fuel substitution. High load condition results in increased ignition delay and cylinder gas temperature which may lead to dispersion of the pilot fuel prior to ignition. This will lead to poor combustion of the gaseous fuel–air mixture. The variation in CO is shown in Fig. 41 (a) and (b). At 10% load condition Cases II, III and IV show 89%, 77% and 75% raise in CO emission, respectively, as compared to Case I operation. At 10% load condition gaseous fuel–air mixture near the pilot is burned due to less turbulence. Thus some partial oxidation product such as carbon monoxide may come out in the exhaust. At higher concentration of gaseous fuel, the concentration of the partial oxidation product could increase. Moreover, inducted mixture becomes rich due to more displacement of air. This is thought to be the reason for the raise in CO emissions. At 80% load condition, maximum raise in CO emission for the Cases II, III and IV is 76%, 84% and 80%, respectively, as compared to Case I, due to rich mixture and prominent temperature at this condition. Fig. 42 (a) and (b) shows variation of NOx for the Cases II, III and IV at 10% and 80% load conditions. It is observed that dual fuel operation produces less NOx at all load conditions than Case I operation. At 10% load condition 60%, 33% and 93% drops in NOx emission were observed for the Cases II, III and IV, respectively, as compared to Case I. Similarly, at 80% load condition 20%, 41% and 84% reductions in NOx for the Cases II, III and IV were observed,

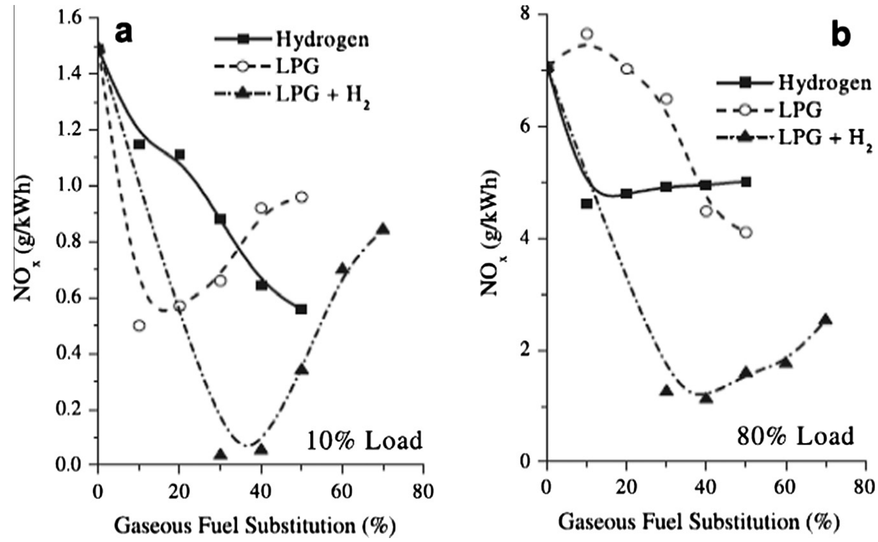

Figure 42 NOx (g/kW h) vs diesel + gaseous fuels substitution (%) [88].

respectively, as compared to Case I. This may be due to more uniform temperature distribution obtained with the gaseous fuel–air mixture. This causes reduction in high temperature region around the diesel flame [88].

6. Conclusions

Studies carried out by the various scientists showed that use of LPG in the diesel engine as dual fuel operation is one of the prominent and effective measures to overcome the fossil fuel scarcity and exhaust emissions. The performance, combustion and emission characteristics of the LPG diesel dual fuel engine have been reviewed from the various experimental studies and indicate that the part load characteristic can be improved by optimizing the engine operating parameters and design factors such as engine speed, load, pilot fuel quantity, injection timing, intake manifold condition and intake gaseous fuel compositions. The following conclusions are drawn based on the present work on an LPG–diesel dual fuel engine.

6.1. Effect of engine load

- LPG–diesel dual fuel engines suffer from the problems such as poor brake thermal efficiency and high HC emissions, particularly at low load conditions and improved brake thermal efficiency at higher loads.
- Poor part load performance and emissions can be improved by modifying the intake charge conditions by various methods.
- However, increase in diesel replacement NOx emission decreases at part loads and PM drastically decreases for all operating loads.
- Dual fuels experience higher noise levels compared to single diesel fuel as the load is increased by increasing the LPG fuel with fixed diesel flow rate.
- The combustion duration in the dual fuel mode is higher than diesel values at low outputs because ignition delay is high. However it is lower than diesel values at high outputs.

6.2. Effect of engine speed

- Increasing the engine speed reduces the combustion noise for the dual fuel engine.
- Increasing the engine speed resulted in a thermal efficiency improvement and its cyclic variation.
- Pressure raise rate decreases with increase in engine speed and is higher than that for diesel case.

6.3. Effect of pilot fuel quantity

- Increasing the pilot fuel quantity increases the torque output, thermal efficiency and maximum pressure. However the combustion noise generated is greater than pure diesel mono-fuel case.
- Use of high pilot quantities is very effective in reducing HC and CO levels at all outputs.
- The increase in NOx emissions with increasing amount of pilot fuel was attributed to increases in the maximum temperature of the charge.
- High pilot diesel quantities have to be used at low outputs to ensure proper combustion of the gaseous fuel. As the power output increases the pilot quantity has to be reduced to control rapid combustion and knock.

6.4. Effect of injection timing

- An improvement in brake thermal efficiency is achieved by advancing the injection timing and smoke opacity, HC and CO emissions decreased.
- For the advanced injection timing, peak temperature increases; hence, NOx emission will be higher and also the reduction of HC and CO emissions was observed due to increase in temperature.
- Advancing the injection timing of pilot fuel increases the overall engine's noise level and the combustion noise level. Dual fuel cases are showing higher levels of noise compared to the diesel fuel especially at early injection timing.

6.5. Effect of intake manifold conditions

- As the intake temperature increases at any given output the ignition delay reduces with both diesel and dual fuel operation.
- Increase in the intake charge temperature results in improved brake thermal efficiency and reduces the HC and CO emission.
- Optimum EGR flow improves brake thermal efficiency and reduces HC levels particularly at low and medium outputs.
- The rate of pressure raise is marginal for all EGR percentages at part loads; however the rate of pressure raise reduces significantly at higher loads.

6.6. Effect of compression ratio

- Higher compression ratio below the level of knocking, improves the engine overall efficiency with higher levels of combustion noise generation.
- Increasing the compression ratio while using LPG in dual fuel mode leads to excessive combustion noise and cyclic variation at high loads in addition to loss in indicated mean effective pressure.

6.7. Effect of varying the compositions of the gaseous fuel

- Variation in LPG composition causes a variation in the exhaust emissions, the exhaust gas temperatures and the fuel conversion efficiency in dual fuel operation.
- LPG with butane content 30% was the best LPG blends in a dual fuel operation since the overall engine performance was equivalent to the conventional diesel engine except that at part loads.
- Use of hydrogen and LPG as secondary fuel enhances the brake thermal efficiency at high load conditions while it produces reverse effect at low load conditions.
- A mixture of hydrogen and LPG as secondary fuel reduces the un-burnt hydrocarbon, NOx and smoke at higher load conditions.

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