

An Examination of the Maximum Possible Natural Gas Substitution for Diesel Fuel in a Direct Injected Diesel Engine

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In recent years, much attention has been focused on the application of alternative gaseous fuels, particularly natural gas (NG), to diesel engines. This is due to the promising results obtained from the research on NG as an internal combustion engine fuel throughout the world which has shown that there is potential for minimizing air pollution and noise by replacing diesel fuel in existing engines by NG fuel. For partial substitution of diesel fuel with NG fuel in a diesel engine, a mixed combustion process called dual-fuelling approach has been adopted. Dual fuelling is the method whereby an alternative gaseous fuel such as NG is induced into the cylinder as a primary fuel, with air, and is subsequently ignited with a pilot injection of diesel fuel. Extensive tests on emissions, performance and different amounts of NG substitution in a direct injection (DI) diesel engine have been carried out for both diesel fuel only and dual fuel (DF) operations. The results show that 86 % NG could be substituted for diesel fuel at low speeds over the whole load range. Maximum NG operation showed higher carbon monoxide (CO) emissions than the diesel fuel only operation while smoke was reduced with DF operation. The maximum values of smoke emission were 30 Hatridge smoke numbers (HSN) with the maximum NG and 28.5 HSN in diesel fuel only operation.

NOMENCLATURE

Alphabetical symbols

$(F/A)_{stoich}$	stoichiometric fuel/air ratio
$(A/F)_d$	stoichiometric air/fuel ratio for diesel fuel
$(A/F)_{ng}$	stoichiometric air/fuel ratio for NG
m_d	mass of diesel fuel
m_{air}	mass of air
m_{ng}	mass of NG
Q_f	heating value of the fuel
$Q_{L,d}$	lower heating value of diesel fuel
$Q_{L,ng}$	lower heating value of NG
W	power

Greek symbols

ϕ	equivalence ratio
η	thermal efficiency

Subscripts

d	referring to diesel fuel
ng	referring to NG
$stoich$	referring to stoichiometric conditions

Introduction

With increasing restrictions of tailpipe emissions from vehicles powered by internal combustion engines and growing concern over the use of liquid hydrocarbon fuels, alternative fuels have gained popularity. Natural gas (NG) is a promising alternative fuel to diesel fuel for road transport vehicles. The use of NG as a fuel has been growing in recent years due to the considerable economic and environmental advantages. Its low cetane and high octane numbers makes it easily adaptable to spark ignition (SI) engines but not to compression ignition (CI) types. NG can be stored in a tank either in compressed form as compressed natural gas (CNG) or in a liquid form as liquefied natural gas (LNG) where it must be kept at -162°C , both restricting vehicle range. One scenario for the application of NG in heavy-duty vehicles is that for highway trucks to operate on diesel fuel in rural areas, switching to NG as they enter highly polluted city regions. This rules out the dedicated NG engine.

Diesel engines can be converted to run on gas either by conversion to spark-ignition (SI conversion), or by dual fuelling. The latter retains diesel injection either at a minimum (pilot) level or at a higher level if necessary to overcome combustion problems. There is some debate over which is the preferred option. The second option looks the best due to its ability for rapid reversion to full diesel mode when gas supplies are hard to obtain. If conversion of existing vehicles is considered, dual fuelling has a strong capital cost advantage but this is not so if the engines for gas operation are purchased directly from manufacturer in the form required.

Dual fuelling is a method of utilising gaseous fuels, whereby the primary fuel (gaseous fuel) is premixed with fresh air in the intake manifold (or occasionally injected into the cylinder) and inducted into the cylinder and ignited by a small quantity of diesel fuel (the "pilot") as the piston approaches the end of the compression stroke. During this stroke, the premixed gaseous fuel-air charge becomes subjected increasingly with time to higher temperatures and pressures as top dead centre is approached. Towards the end of the stroke, a small quantity of diesel fuel is injected into the cylinder under a high pressure from 20 to 150 MPa depending on the type of diesel fuel injection system. The finely atomised fuel particles mix with the air to form a combustible mixture which, following some physical and chemical changes, self-ignites due to the high temperature of the compressed gaseous fuel-air mixture. The flame subsequently consumes the gas within the spray in a direct manner or establishes a progressive flame front that moves away from the ignition source. In order for the ignition to be successful, the energy release rate in the early stage of the ignition must be greater than losses from the ignition flame kernel. If not, the flame extinguishes prematurely.

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Numerous works have been done over the last two decades. The performance of NG engines has been investigated with promising results and complex behaviours and underlying combustion mechanisms have been revealed¹⁻²¹. The emissions aspects are less well investigated^{1,3,4,6,10,22} but the preliminary results from investigations are encouraging for the particulate matter and NO_x emissions. As can be revealed from the literature survey many of the previous studies have primarily been concerned with performance and emissions and very little work has been done on the maximum amount of NG substitution for diesel fuel^{9,18}. More research on the maximum amount of NG substitution for diesel fuel in DF engine would be of great benefit for engine developers and operators. From this background, the objective of this work is to maximise the substitution of NG for diesel fuel while simultaneously maintaining the engine power specified by the manufacturer for the engine, operating on diesel fuel only and without major modification of the engine.

Experimental apparatus and Procedures

Experimental apparatus

The engine used in this study was a Cummins Model 5/9, a six cylinder, naturally aspirated, four stroke DI diesel engine. The engine specifications are given in Table 1.

Engine (four stroke cycle)	Model 5/9
Number of cylinders	6
Combustion system	Direct injection
Bore	102 mm
Stroke	120 mm
Displacement	6 liters
Compression Ratio	17
Maxim. Engine speed at full load	2500 RPM
Rated Brake Power	86 kW

Table 1: Test engine specifications

The engine was coupled to a Froude type eddy current dynamometer. The dynamometer torque was measured by using a strain gauge load cell and the speed was measured by using a magnetic inductive sensor and 60-tooth gear assembly. The air drawn in by the engine was first passed through a sharp-edged orifice plate, which was mounted on the side of an air-box. The pressure drop across the orifice plate was measured by a manometer. Meanwhile, a high-pressure cylinder containing NG was used to fuel the engine during DF operation. NG was introduced into the intake air stream at about 500mm from the inlet manifold of the engine as shown in Figure 1. A DF system composed of a regulating valve, two metering valves, solenoid operated safety valve and a Hastings mass flow meter with a flow range of 0.0 - 2.5 litre/sec and accuracy to ± 0.1 . The diesel fuel consumption rate was measured by a positive displacement flow meter, Fluidyne type (model 1213D). A Kistler (model 6001) quartz pressure transducer and a pressure sensor were used to record the cylinder pressure and the diesel fuel line pressure respectively. These electric signals, along with other electric signals such as the TDC signals, were logged by high-speed data acquisition system. During the experiments, a Tektronix Model TDS 360 digital oscilloscope was used to monitor cylin-

der pressure, crankshaft position and diesel fuel flow rate for selecting an appropriate time to record data. An infrared exhaust analyser Beckman type model 864-11-4 was used for the measurement of CO in the exhaust. NO_x and UHC emissions were impossible to measure due to the lack of funding for serving the NO_x/UHC equipment. Smoke density were obtained using a Hartridge Mk3-154 smoke meter.

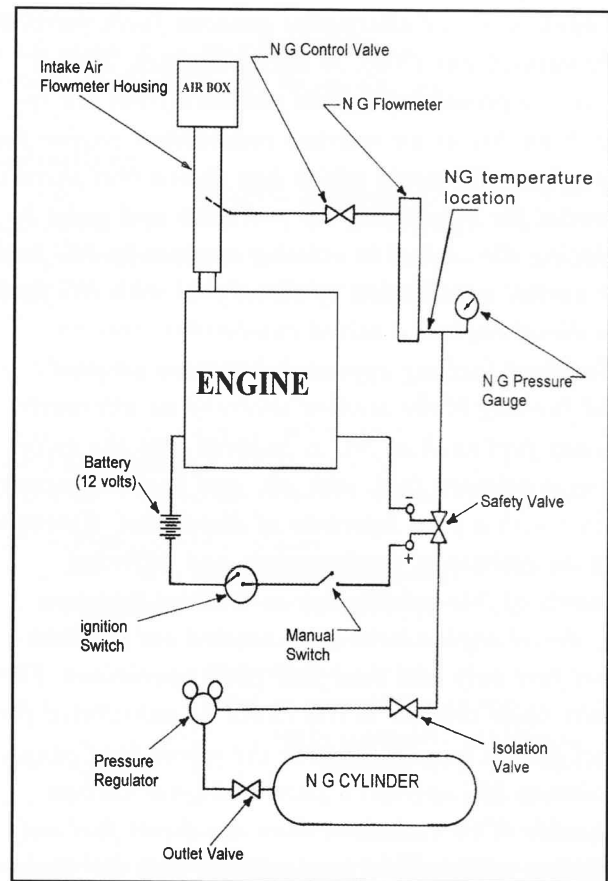


Figure 1: Schematic of the Experimental-setup

The properties of the fuels used in this study are shown in Table 2.

Experimental Procedure

The following experimental procedure was carried out during the study:

- Tests were conducted with diesel fuel only (baseline tests);
- Tests were conducted with the maximum replacement of diesel fuel by NG.
- Criteria for this maximum gas substitution were:
 - Severe diesel "knock" or "end gas auto-ignition" being evident. While pressure traces were helpful in this evaluation, a more subjective audible technique proved to be very consistent;
 - Combustion loss, noticeable as misfire and inability to sustain a steady torque output, occurred.

Baseline Experiments

Before the main investigation of DF operation was started, a number of tests with the engine running on diesel fuel only were carried out. The results from these tests produced the basis for the comparison with the performance of the DF operational mode.

In order to ensure steady and normal conditions, the engine

NG	
Methane (CH ₄)	94.1 % vol.
Ethane (C ₂ H ₆)	2.8 % vol.
Propane (C ₃ H ₈)	0.1 % vol.
Carbon Dioxide (CO ₂)	1.5 % vol.
Nitrogen (N ₂)	1.5 % vol.
Molecular mass	17.8
Gas density (relative to air)	0.62
Liquid relative density	0.44
Stoichiometric air/fuel by volume	9.85
Stoichiometric air/fuel by mass	16.0
Flammability limits (%volume) lean	5.0
Flammability limits (%volume) rich	15.0
Flammability limits at lean limit	0.52
Laminar flame speeds (cm /s at 1 atm and 300 K)	app. 35
Boiling point at 1atm	-162
Ignition temperature in air	50-750 °C
Research octane number (approx.)	130
Lower heating value (MJ/kg)	46.5
Diesel fuel	
Lower heating value (MJ/kg)	44
Density (kg/m ³)	836
Cetane Number	62.2

Table 2: Fuels properties

was first run for approximately 15 minutes, without any load, until the following operating conditions were reached:

- speed of 1200 rpm;
- coolant inlet temperature of 65 °C;
- exhaust temperature of 175 °C.

Once a steady state condition was obtained, testing took place with engine speed varied from 1000 rpm to 2400 rpm. The dynamometer load was set for engine loads of quarter, half, three-quarter, and full load, for each particular engine speed (i.e. 1200, 1400, 1600, 1800, 2000, 2200 and 2400 rpm). The full load was assumed to be maximum torque, which is close to the rated torque. After the engine had been stabilised, readings such as engine speed, torque, pilot diesel fuel flow-rate, NG flow-rate, temperatures and CO emissions were recorded. The Hartridge smoke number was measured throughout the study.

DF Investigation

Before any NG was introduced, the engine was first run for 15 minutes with diesel fuel only at approximately half load. This load was maintained until the engine reached normal operating conditions. For the DF mode of operation, the engine was brought to a required operating point on diesel fuel only. By reducing pilot diesel fuel quantity using the rack control lever and simultaneously opening the NG control valve, the engine could be

held to the required speed and load. After the engine has been stabilised, the measurements were recorded. The proportions of NG and pilot diesel fuel were varied until the maximum NG flow rate was established.

It is worth mentioning here that during the experiment, the engine timing was not altered from the standard 20° BTDC.

Experimental Results

The relative contribution of NG to the total fuel consumption was calculated on an energy basis, lower heating values (LHV) for both diesel fuel and NG being used. For the diesel fuel, a heating value of 44 MJ/ kg was used while NG was calculated from the proportion of gases in the mixture of NG as 46.5 MJ/kg. The NG percentage of the total fuel was calculated as follows:

$$\% NG = \frac{m_{ng} \times Q_{L_{ng}}}{m_{ng} Q_{L_{ng}} + m_d \times Q_{L_d}} \times \frac{100}{1} \quad (1)$$

Equivalence ratio, ϕ , defined as actual fuel/air ratio to stoichiometric fuel/air was calculated from:

$$\phi_{tot} = \frac{\left(\frac{A}{F}\right)_{stoich} \times m_{ng} + \left(\frac{A}{F}\right)_{stoich} \times m_d}{m_{air}} \quad (2)$$

This equation assumes that the excess air is proportional to each fuel according to its mass with an identical equivalence ratio for each.

The basic full throttle curve for diesel fuel only is shown on Figure 2. These are very close to the manufacturer's values. To evaluate more fully, a range of engine maps based on engine torque and speed as ordinates were constructed for the maximum NG substitution based on an energy percentage, efficiency, equivalence ratio and exhaust temperature. These are shown on Figures 4, 5 and 6. These figures compare the diesel operation with the maximum NG substitution DF mode. It should be noted that there is no simple way of drawing such contour plots (engine maps) and some smoothing of the curves has been carried out.

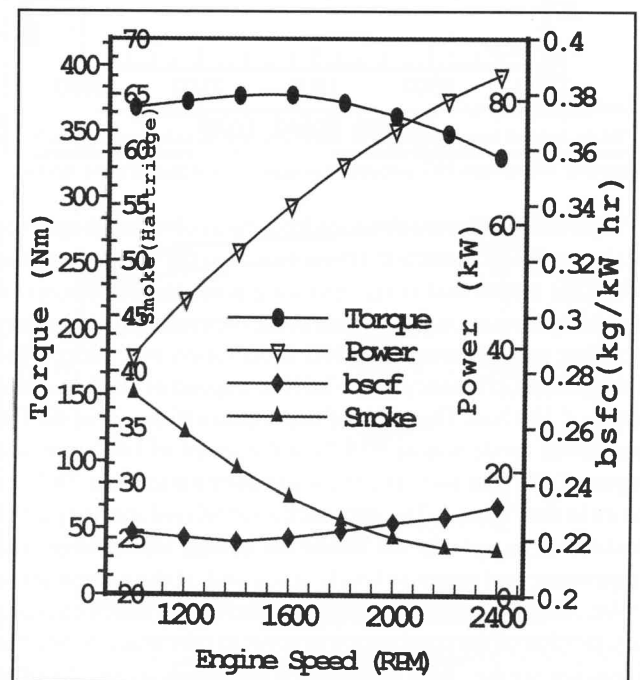


Figure 2: Basic full throttle curves for fuel diesel fuel only operation

From these results, basic information about the maximum NG substitution based on an energy percentage, efficiency, equivalence ratio and exhaust temperature of the respective concepts can be determined immediately.

Discussion of the results

The maximum amount of NG fuel that can be substituted depends upon the criterion used. Using a knock criterion, the maximum NG fuel substituted on an energy basis was 86 % as shown in Figure 3. Thus maximum replacement of NG fuel (86 %) is possible over the whole load range at 1200 rpm, yielding the same torque levels equivalent to those for full load (diesel fuel only) operation. Following this contour, as the engine speed increases, the amount of NG (86 % by energy) substituted can only occur at a torque lower than the maximum torque value for diesel fuel only operation. Above that 86 % substitution contour level, knock occurs. At 2400 rpm, the 86 % substitution can only be used below about 20 % of the rated (diesel fuel only) torque value. However, increased engine speed increases the piston speed which in turn increases the turbulence and accelerates the combustion process in the pre-mixed NG/air mixture. Thus, efficient burning of the gas is likely to occur at high speed, although the quantity of gas that can be used falls dramatically as the torque rises.

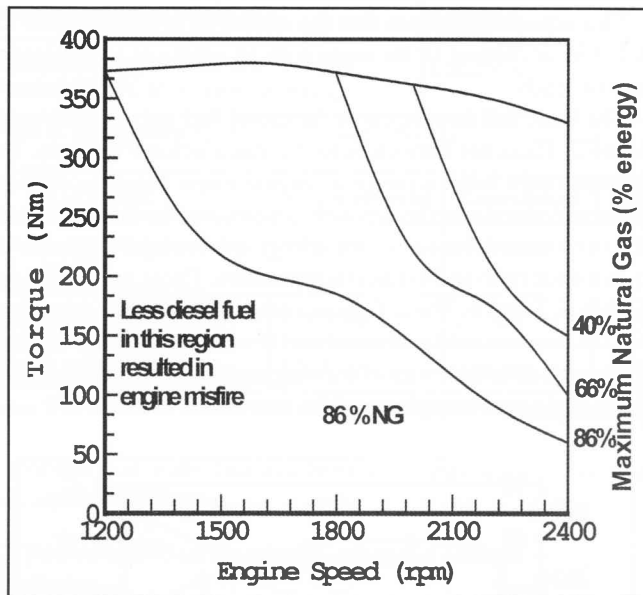


Figure 3: Maximum NG substitution against Torque/Engine speed

Figures 4 and 5 were obtained from data corresponding to the maximum NG substitution levels based on the knock criterion only. This means that every operating point has its own maximum NG substitution. The engine brake thermal efficiency at any operating point decreased as NG substitution was added. The highest diesel efficiency was 37.5% at a speed of 1600 rpm and torque of 380 Nm. The point of maximum efficiency in the DF operational mode was at 30.9 % at the speed of 1400 rpm and torque of 377 Nm and maximum gas substitution was 76 % as shown in the Figure 4. The introduction of NG reduces the partial pressure of oxygen in the intake air charge and changes the temperature and pressure levels at the end of the compression stroke. As a result, the ignition delay increases, which causes a large portion of the combustion process to take place during the expansion stroke. This results in a reduction of the thermal efficiency during DF operation as shown in the figure 4. As can

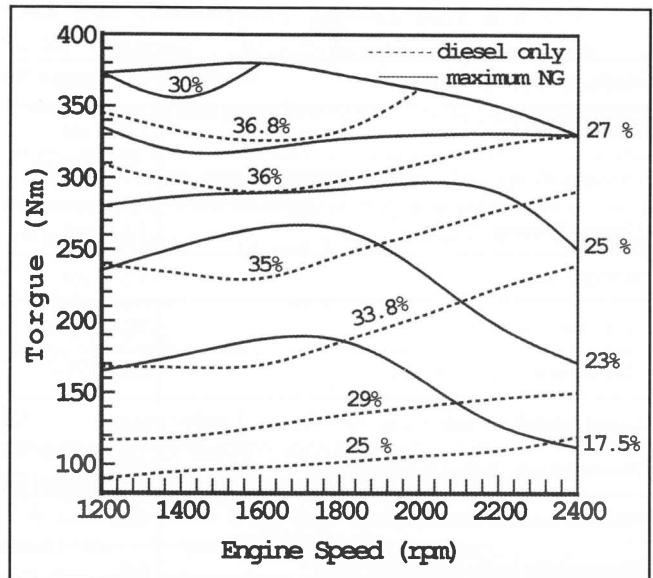


Figure 4: Efficiency against Torque/engine speed for both DF and diesel fuel only

be noticed, the shape of the maximum NG efficiency lines (solid lines) at low torque is opposite to that at higher torque regions, the latter being smaller than all the diesel efficiency contours. At the low torque regions, the maximum NG efficiency contour lines first increase to a peak value and then decrease. A likely explanation of this is that, at low speed operation, utilisation of the NG is poor due to low turbulence and hence slows burning of the lean NG. As the engine torque increases, more NG fuel is introduced into the cylinder and as a result, a more effective rich mixture is produced which improves the low speed performance. For low load at higher engine speeds, the increased turbulence improves the burning rate of the lean gas mixture. The maximum power output for DF operation is reached at about 1750 rpm and a torque of about 370 Nm. At the high-speed regions, there is a limitation of the substitution of NG and the very high diesel proportions at high torque cause the maximum NG efficiency value to slightly approach those of diesel fuel only operation. It is, however, not entirely clear why the maximum NG efficiencies fall off at high speed/medium to low load. It is possible that the NG burning is too slow with the energy being released too late in the expansion stroke at high speed to be used effectively.

Figure 5, shows a comparison of the overall equivalence ratios of the DF and diesel fuel only operation. As can be seen, the overall equivalence ratio points on the engine map for the DF operational mode were always at a considerably higher level than the diesel fuel only operation. At a given torque and speed (i.e. power) level, equivalence ratio is inversely proportional to the product of thermal efficiency and airflow to the engine as shown in the Equation (3)

$$W = \eta \phi (F/A)_{stoich} m_{air} Q_f \tag{3}$$

Substitution of gas displaces the air in the intake and as a result the volumetric efficiency is low in the DF mode. That is, the air mass flow rate (m) becomes lower and power (W) decreases if equivalence ratio ϕ remains the same. This may be the major reason for the increased equivalence ratio. In addition, the thermal efficiency reduces as already discussed and the associated fall in power (W) must again be compensated for by increasing ϕ . It can be seen from the figure that overall equiva-

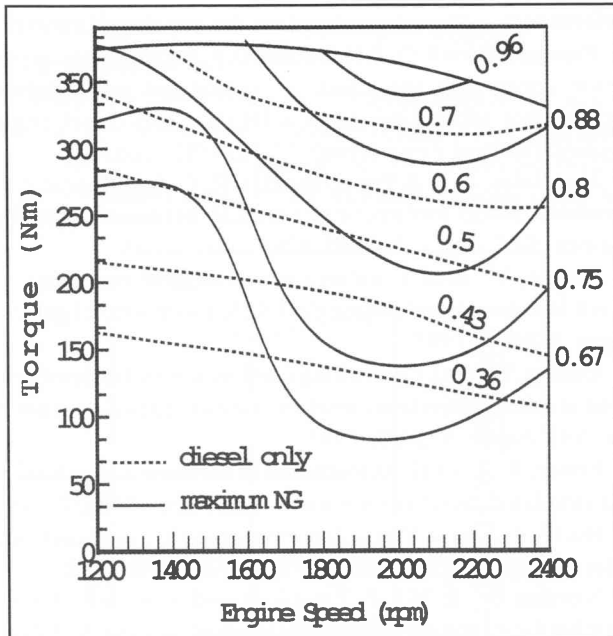


Figure 5: Equivalence ratio against torque/engine speed for both DF and diesel fuel only

lence ratio values in high speed, high load region, approach the diesel fuel only values. This is because the tendency towards end-gas autoignition as the stoichiometric value is approached limits the gas substitution. At low speed (1400 rpm) and high torque, the maximum NG equivalence ratios approach the diesel fuel only equivalence ratio values [$\phi = 0.8$ for DF (solid line), $\phi = 0.75$ for diesel fuel only (dotted line)]. As the engine torque decreases further, a combination of poor mixing of the air/fuel mixture and a lean gaseous mixtures occurs. The turbulent flame ignited by the diesel spray may then not propagate through the charge mixture. As a result, some of the NG fuel will survive to the exhaust. Even if not, the late burning will lower the efficiency. This will decrease the output power. At about 1400 rpm, maximum torque is achieved for contour lines of $\phi = 0.67$ and 0.75 . Below this speed, turbulent flame propagation becomes very poor. Above 1400 rpm, as the engine speed increases with the same value of ϕ , the torque falls rapidly. This is due to the low efficiency experienced in the region.

Figure 6 depicts the exhaust temperature contours for DF operation. Also shown is the temperature deviation between the DF operational mode and that of diesel fuel only, as calculated from the following formula.

$$T_{Deviation} = T_{maximum\ NG} - T_{diesel\ fuel\ only} \quad (4)$$

At high load, higher exhaust temperatures were observed for the diesel fuel only mode than for DF operation. This is the area of the map above the 0°C contour. This is due to the introduction of NG in DF mode, which in turn reduces the temperature and pressure at the end of the compression stroke because of its different isentropic index and heat transfer coefficient. As a result, the mixture of NG/air burns at a lower temperature than does the diesel fuel only. The deviation in temperature is more pronounced in the middle speed range (1600 – 1800 rpm) and higher load region. In this region, the DF engine is operating on mixtures approaching stoichiometric ratio and, as a result exhaust temperatures increase. As shown in Figure 6, the deviation of temperature is higher at low load and low speed due to the poor

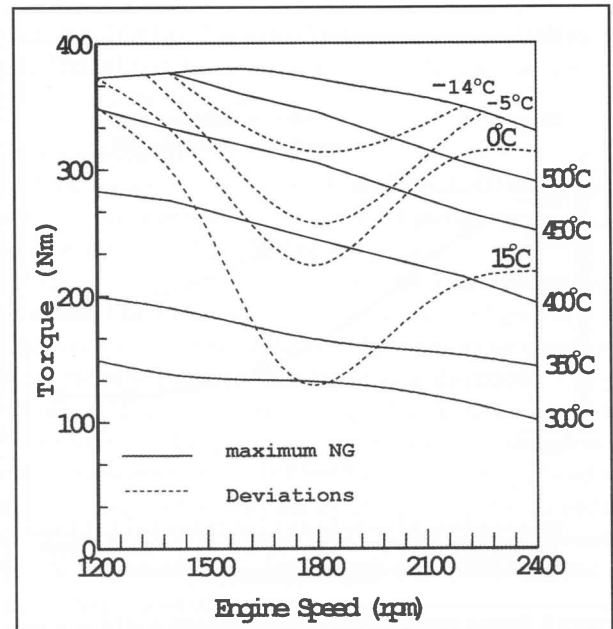


Figure 6: Exhaust temperatures against torque/engine speeds for DF operation

combustion of the NG, which is a result of the reduction of the oxygen fraction by the addition of the gaseous fuel. Some of this NG gas remains unreacted, surviving to the exhaust stage¹²⁻¹⁴. On the other hand, increases of the deviation of temperature are also noticed at high speed with low load due to the high overall equivalence ratio.

The results for CO emissions under diesel fuel only and DF operation are given in Figure 7 for full load at the maximum gas substitution level. It should be noted that every operating point has its own maximum gas substitution. As known, the rate of CO formation is a function of the oxygen concentration, flame temperature, gas residence time at high temperature and combustion chamber turbulence. All of these variables control the rate of fuel decomposition and oxidation^{22,23,24}. As observed, CO emissions under DF operations are higher compared to the ones under diesel fuel only operation. This is mainly due to the slow combustion rate of NG fuel, which maintains the charge tempera-

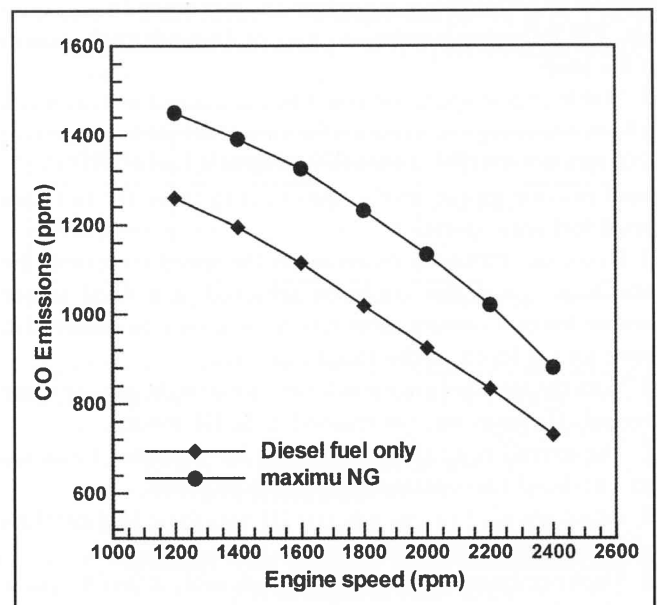


Figure 7: Carbon monoxide concentrations under diesel fuel only and DF operation vs. engine speeds at full load

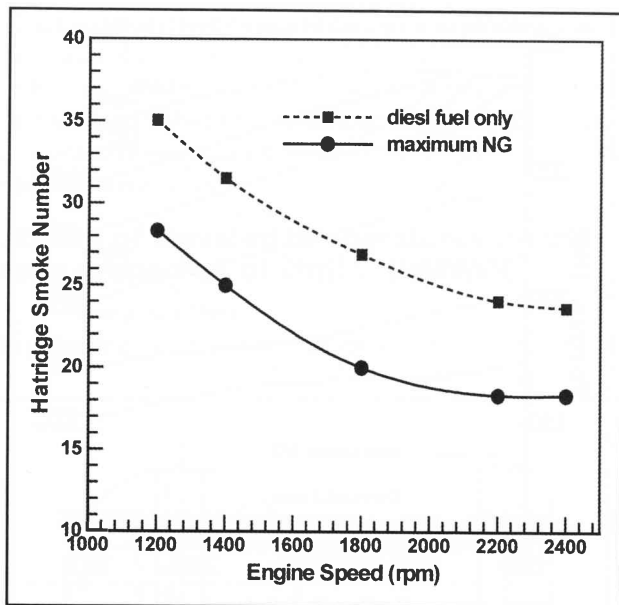


Figure 8: Smoke emissions under diesel fuel only and DF operation vs. engine speeds at full load

ture at low levels resulting in a reduction of the oxidation process of CO. With the increase of engine speed, there is a sharp decrease of CO under both diesel fuel and DF operations. This is due to the increase of gas temperature, faster combustion rate and higher combustion chamber turbulence that help oxidise efficiently the CO emissions.

Smoke emissions were observed by comparing the HSN measurements for full load. The DF operational mode yielded lower smoke emissions than that of the diesel fuel as shown in Figure 8. The reduction in smoke formation is expected since the homogeneity of the air-fuel mixture improves for DF operational mode cases. This minimises the occurrence of locally rich zones, which can produce smoke emissions.

Conclusion

The main thrust of this study was to evaluate the maximum possible NG fuel substitution for diesel fuel in a DI diesel engine. No major modification was carried before testing the engine to improve its performance, emissions and maximum NG substitution. The following conclusions may be drawn from the results of the study:

- Stable engine operation could be maintained with as much as 86% NG energy substitution for diesel fuel. At low speeds of 1200 rpm to 1300 rpm, the maximum substitution of NG to 86% could provide torque levels equivalent to those for full load diesel fuel only operation;
- Based on the knock criterion, as the speed increased, the substitution level that could be achieved at a fixed torque became lower. Constant substitution could only be achieved at lower torque levels as the speed increased;
- With the standard engine settings, the straight diesel engine thermal efficiency was not reached in the DF mode;
- The overall equivalence ratios for DF exceeded those for straight diesel fuel operational mode at all loads;
- Emissions of CO were higher for DF than those for diesel fuel only operation and,
- Smoke emissions were decreased noticeably in the DF operational mode.

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