Energy 53 (2013) 52–57

Contents lists available at SciVerse ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

Performance and emissions of a heavy-duty diesel engine fuelled with diesel and LNG (liquid natural gas)

Kraipat Cheenkachorn [a,c,*], Chedthawut Poompipatpong [b,c], Choi Gyeung Ho [d]

[a] Department of Chemical Engineering, Faculty of Engineering, Thailand
[b] Department of Automotive Engineering Technology, College of Industrial Technology, Thailand
[c] Automotive Eco-Energy Research Center, King Mongkut's University of Technology North Bangkok, Thailand
[d] School of Basic Science, College of Convergence, Daegu Gyeongbuk Institute of Science and Technology, Republic of Korea

A R T I C L E I N F O

Article history:
Received 8 August 2012
Received in revised form
8 February 2013
Accepted 9 February 2013
Available online 24 March 2013

Keywords:
Modified diesel engine
Natural gas
Engine performance
Emissions

A B S T R A C T

This paper presents the effects of liquid natural gas on a heavy-duty diesel engine. Natural gas was used as primary fuel, while a pilot amount of diesel was used as an ignition source. The amount of each fuel was adjusted to obtain comparable brake torque and power output from the dual engine operation, while no knocking was observed. The engine performance and emissions from the diesel and dual fuel engine tests were conducted over the engine speed range between 1100 and 1900 rpm. The engine performance included torque, power, specific fuel consumption (SFC), volume efficiency, and thermal efficiency. The emissions tested were total hydrocarbon (THC), nitrogen oxides (NO_x), carbon dioxide (CO_2) and carbon monoxide (CO) emissions. The results showed that the maximum portion of natural gas in the dual fuel engine operation was up to 77.90% at 1300 rpm. Compared to the diesel operation, the dual fuel operation showed less specific fuel consumption, thermal efficiency, and volumetric efficiency. The emissions of THC and CO from the dual fuel engine operation were higher, while the emissions of NO_x and CO_2 were lower.

1. Introduction

Global energy demand has recently been increasing and is projected to reach 12 MTOE by the end of 2012 [1]. Consequently, the high demand of energy consumption has led to the problem of global warming, which is mainly caused by the fossil fuel combustion that produces greenhouse gases. Diesel is one of the main fossil fuels currently used in the transportation and industrial sectors. Demand for diesel consumption is expected to grow continuously as a result of the dramatic increase in vehicles, especially in many Asian countries such as China, India, Korea, etc.

In seeking a compromise between the need for increased energy while at the same time reducing carbon dioxide (CO_2) emissions, alternative fuels have been found to be a viable alternative to conventional fossil fuels. Among the alternative fuels, natural gas has emerged as a preferred fuel in the transportation sector due to its advantages, such as greenhouse gas reduction, better combustion efficiency, attractive cost, renewability through the biomass production processes, etc.

Natural gas in the transportation sector can be used in the form of compressed natural gas (CNG) and liquid natural gas (LNG). Presently, CNG is gaining popularity in automobiles mainly due to its lower cost, while the use of LNG is mainly used for transportation and electricity production. Compared to CNG, LNG offers advantages, such as easier transportation, storage, and safety. Key challenges for LNG in terms of achieving a global market share, however, are the lack of ifrastructure, harmonized standards and regulations, adoption of the automobile manufacturers, and retrofit systems [2]. Previous studies have shown that LNG has proven to be the most viable option for long-range use due to its greater liquefied state compared to the CNG system in terms of economic potentials [3–5], environmental aspects [6], and technical performances [7,8].

Although natural gas finds its major uses in transportation due to its availability and environment benignity, it is still limited to small engines, especially spark-ignition (SI) engines, and is rarely found in large diesel engines. Therefore, the utilization of natural gas as the main fuel for large or heavy diesel engines seems to show more impact in terms of their operations because most transportation fleets are diesel operated. It is widely known that natural gas engines normally produce less power than diesel engines [9,10]. Moreover, another severe problem of the natural gas-diesel dual fuel engine is knocking phenomena [9,11–13]. Generally, knocking is still a main problem in small dual fuel engines, as reported in

* Corresponding author. Automotive Eco-Energy Research Center, King Mongkut's University of Technology North Bangkok, Thailand. Tel.: +66 2 913 2500; fax: +66 2 587 0024.
E-mail address: kpc@kmutnb.ac.th (K. Cheenkachorn).

0360-5442/$ – see front matter © 2013 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.energy.2013.02.027

previous studies [14–16]. This problem is even more pronounced for the utilization of gaseous fuel in large diesel engines. For this reason, natural gas is mostly used as a minor fuel in dual-fuel engine operation. Therefore, it is a key challenge to alleviate knocking problems when utilizing gaseous fuel in large dual-fuel engines while maintaining it as a major fuel [17,18].

The objective of the present work was to examine the effects of natural gas on the operation of a large dual diesel engine, including engine performance and emission characteristics. The engine performance tests included the following: brake torque, brake power, thermal efficiency, volume efficiency, and specific fuel consumption (SFC). The emissions testing included total hydrocarbons (THC), nitrogen oxides (NO_x), carbon monoxide (CO), and CO_2. The engine tests were conducted at full load condition and at a wide range of engine speeds without knocking problems. Natural gas was used as primary fuel, while a pilot amount of diesel fuel was used as an ignition source.

2. Experimental

As previously mentioned, the primary purpose of this experimental study was to examine the effects of LNG on a large diesel engine, where LNG was used as the main fuel. The basic properties of the fuels are given in Table 1. The engine used in the present study was the Hyundai D6CA (Direct Injection). The detailed specifications of this engine are shown in Table 2. The diesel-engine modification was done by installing LNG system. To avoid knocking, diesel injection timing and injection duration for each fuel were adjusted by ECU. Knocking was monitored by two knock sensors.

The engine was coupled to an AVL AC-dynamometer (up to 490 kW) rated at 2500 Nm and a maximum speed of 7500 rpm. The engine was dual-fuel operated at full load with speeds between 1000 and 2000 rpm. The Horiba (Model Mexa-7100 D) exhaust gas analyzer was equipped to measure the emissions. CO and CO_2 were analyzed using the non-dispersive infrared technique, while THC and NO_x were analyzed using a flame ionization detector and a chemi-luminescent detector, respectively. The experimental setup is shown in Fig. 1. The electronic fuel injection system was used to control the amount of each fuel. The experimental data presented represent the average of the duplicated tests.

Brake thermal efficiency and SFC were calculated using Equations (1) and (2).

$$\eta_{th} = \frac{\text{Power} \times 100\%}{(\dot{m}_D \times \text{Diesel Heating Value}) + (\dot{m}_{NG} \times \text{Gas Heating Value})} \quad (1)$$

$$\text{SFC} = \frac{(\dot{m}_D + \dot{m}_{NG})}{\text{Power}} \quad (2)$$

Table 1
Fuel specifications.

Fuel	Properties	Value
LNG	Composition	
	• CH_4 (vol.)	99.80%
	• C_2H_6 (vol.)	0.10%
	• N_2 (vol.)	0.10%
	Density (g/L)	415
	Tank temperature (°C)	−162
	Tank pressure (bar)	8–16
	Regulator and injector pressure (bar)	6 bar
	Lower heating value (kJ/kg)	49,244
Diesel	Density at 15 °C (kg/L)	0.8504
	Lower heating value (kJ/kg)	43,400

Table 2
Engine specifications.

Model	D6CA(L-ENG)
Displacement (cc)	12,920
Bore × stroke (mm)	133 × 155
Maximum power (PS/rpm)	440/1800
Maximum torque (kg m/rpm)	197/1400
Cooling type	Water cooling
Fuel supply system	Direct injection
Cylinders	6 in line
Compression ratio	17 : 1
Aspiration	Turbocharge intercooler

where η_{th} = thermal efficiency (%)

$\dot{m}_D$ = diesel flow rate (kg/hr)
$\dot{m}_{NG}$ = natural gas flow rate (kg/hr)

3. Results and discussions

As already mentioned, one of the main goals of the present work was to retain comparable engine performance after engine modification while maximizing the gaseous fuel. Fig. 2 shows the relations between brake torque and power at different engine speeds at full load condition. It can be clearly seen from the figure that the torque and power of the dual fuel operation were comparable to those of the diesel operation, less than 2.10%. Generally, for a small engine, it is expected that torque will increase as engine speed increases until it reaches the maximum point. After that, torque decreases at a higher speed because of the inability of the engine to ingest a full charge of air. As shown in Fig. 2, for the engine speed ranging from 1100 to 1600 rpm, torque remains constant and starts to drop after the engine speed of 1600 rpm. This trend can also be observed for both normal diesel and dual fuel engine operations. Since large heavy-duty diesel engines are normally designed for very high torque at a low speed range, torque starts to decrease when the engine speed increases. For this reason, the increase of torque at low engine speed was not observed. Also, as shown in Fig. 2, the power increases with engine speed and then considerably decreases at higher speeds of 1900 rpm. This is due to the fact that friction losses increase with speed and become a dominant factor at very high speeds.

Generally, for a small dual fuel engine, the pilot amount of diesel fuel is normally injected at a constant rate to cover the mechanical losses of the engine. The amount of injected gaseous fuel is then adjusted to increase the power output where a knocking problem is not observed, as reported in previous studies [19–22]. However, it is worth pointing out that knocking easily occurs in medium to large dual fuel engines because of the greater distance for the flame to travel. To avoid the knocking problem while maintaining other desired properties, the pilot amount of diesel fuel should be adjusted to improve the operating conditions of the dual fuel [11,12,23]. In the present research, the amount of both natural gas and the diesel pilot injection was, therefore, adjusted to meet three main purposes: Firstly, for economic benefits, the natural gas flow rate should be supplied as high as possible; secondly, the engine performance of dual fuel operation should be comparable to that of the diesel operation. Lastly, the knocking phenomena must be avoided, which is a key challenge for large diesel engines, as previously mentioned.

Fig. 3 compares the amount of each fuel consumed under normal diesel and dual fuel operations. Under normal diesel operation, the diesel flow rate seems to increase as the engine speed increases and then rapidly decreases at the speed of 2000 rpm, corresponding

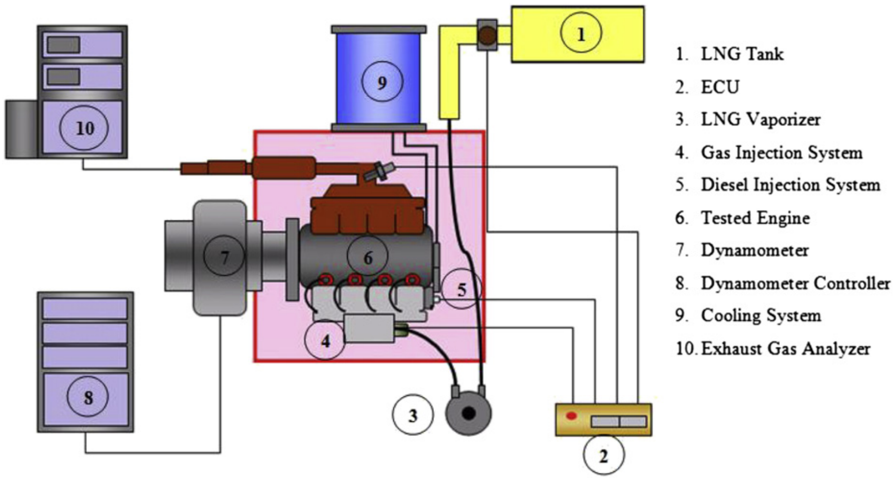

Fig. 1. The experimental setup.

to the sharp power drop, as shown in Fig. 2. For dual fuel operation, the consumptions of fuels were obviously divided into three regions based on the engine speed: low engine speed range (1100–1300 rpm), medium engine speed range (1300–1800 rpm), and high engine speed range (1800–2000 rpm).

In the low engine speed range, as the engine speed increases, the portion of natural gas increases from 69.33% at 1100 rpm to 77.90% at 1300 rpm. The lower engine speed resulted in less natural gas. This was due to the fact that natural gas has longer ignition delay and slower burning rates compared to diesel. Less natural gas must be supplied, which leads to an acceptable rate of pressure rise and prevents diesel knock [9–12]. For the medium engine speed range, higher engine speed results in higher pilot diesel and less natural gas portions. This is because of the increase in mechanical losses, which are dominant at high engine speed [18,19]. More input energy from the combustion of both fuels was required for the dual fuel engine. Since diesel fuel shows higher energy density and volumetric efficiency compared to that of natural gas, a greater portion of diesel must be supplied to provide enough power, which is comparable to single diesel operation. The results in this study agreed with the previous investigation by Mohamed [15], that for dual fuel engine operation, an increase in diesel fuel injection leads to greater energy release on ignition, improved pilot injection characteristics, a larger-size envelope of gaseous-entrained mixture, and a larger number of ignition centers, resulting in larger output power and higher thermal efficiency [14,15]. Fig. 3 shows that the portion of natural gas in the dual fuel engine operation was up to 77.90% at 1300 rpm, which represents a key success in a large diesel engine, as previously mentioned. In the high engine speed range, the portion of natural gas in the dual fuel engine operation decreased significantly, while the opposite trend was observed for that of diesel fuel. This is because a higher portion of natural gas always leads to knocking problems when the engine is operating at a very high speed. The portion of natural gas must be supplied as little as possible in order to avoid knocking.

Thermal efficiency and specific fuel consumption are shown in Fig. 4. As can be seen in the figure, the thermal efficiency of the dual fuel operation was less than that of the single one, averaging less than 3.50%. However, in the practical speed range (less than 1700 rpm), the specific fuel consumption of the dual engine operation was slightly lower than that of the single operation. As presented in Equations (1) and (2), the brake thermal efficiency indicates how efficiently the input energy is converted to useful output energy while brake specific fuel consumption shows the rate of fuel consumed per output power [24]. Therefore, the dual fuel operation was more economically attractive because of the much cheaper price of natural gas compared to that of diesel.

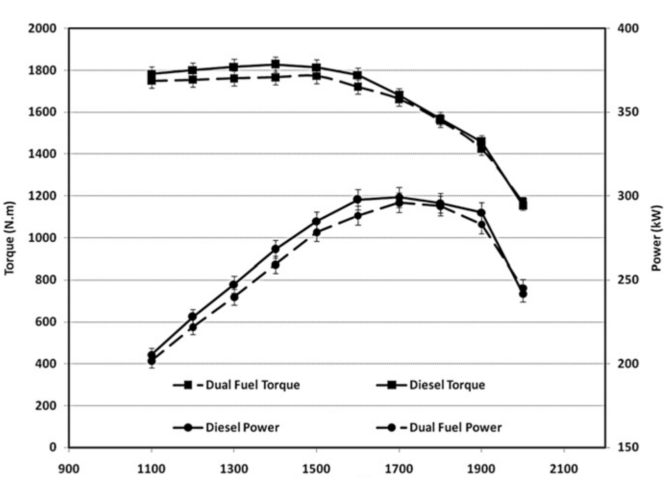

Fig. 2. Brake torque and power at different engine speeds.

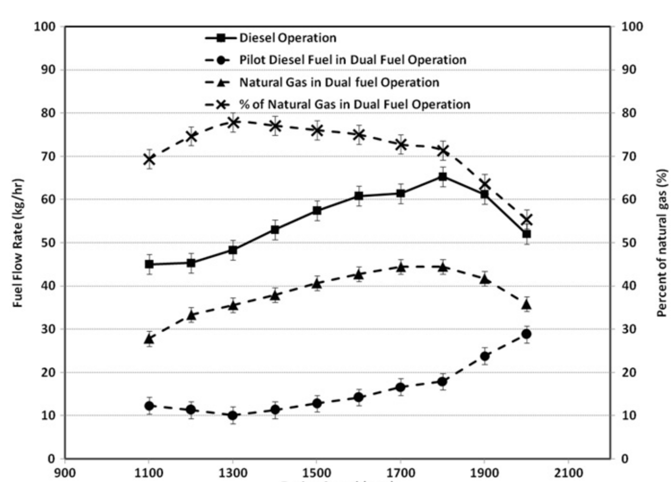

Fig. 3. The amount of each fuel consumed under normal diesel and dual fuel operations.

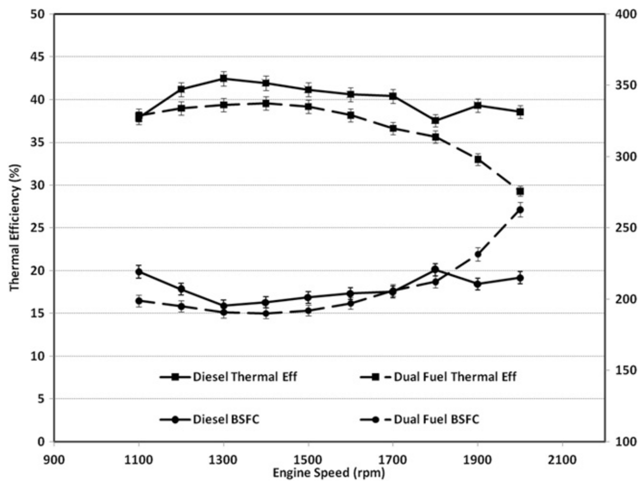

Fig. 4. Thermal efficiency and specific fuel consumption at different engine speeds.

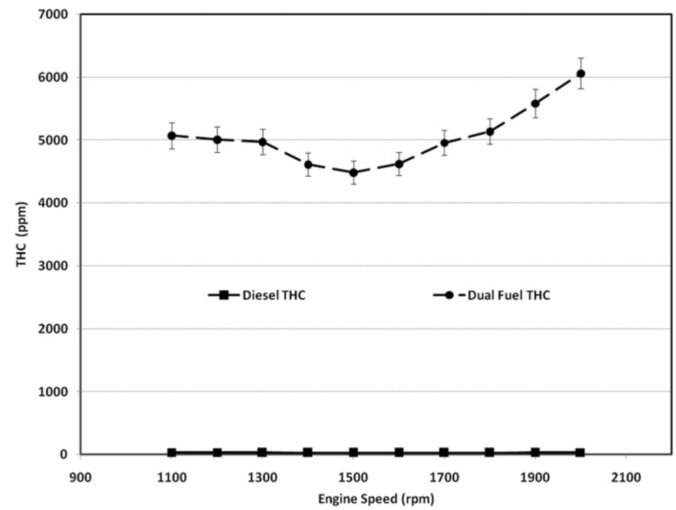

Fig. 6. THC emissions at various engine speeds.

For the engine speed higher than 1700 rpm, the fuel consumption of the natural gas increases rapidly while the thermal efficiency significantly drops. Nevertheless, this condition is rarely used in a normal operation.

Fig. 5 shows the volumetric efficiency at various engine speeds. For both normal diesel and dual fuel operations, the volumetric efficiency decreases as the engine speed increases. Since-large or heavy-duty engines are designed for low-speed and high-torque operation, high volumetric efficiency was observed at lower speed. Compared to the dual fuel operation, the diesel operation showed higher volumetric efficiency. It is worth noting that, in Fig. 5, the volumetric efficiency of the normal diesel operation was 102.7%. This is because the tested engine was installed with a turbocharger. After natural gas is supplied into the dual fuel operation mode, natural gas substitutes the fresh-air charge and leads to lower volumetric efficiency [23]. Approximately 2.73% reduction in the volume efficiency was observed.

The results from the emission measurements, including THC, NO_x, CO, and CO_2, are shown in Figs. 6–9, respectively. THC emissions, which mainly consist of unburned hydrocarbons, represent combustion efficiency. Normally, THC emissions are very low for a diesel engine. As shown in Fig. 6, the THC emissions from the diesel fuel operation remained constant (less than 35 ppm) for the tested speed range. For the dual fuel operation, it showed significantly higher THC emissions (approximately 5,085 ppm) than the diesel operation. This was caused by the higher proportion of the escape of gases due to several factors, such as slower burning rate, lean combustion, and valve timing [9,12,19]. The variation of the THC emissions showed no clear trend over the engine speeds tested. This was due to the fact that not only does the engine speed affect THC emissions, but the amount of each fuel injected also has an effect on these emissions [18]. At a speed range between 1200 rpm and 1500 rpm, the increase of the engine speed results in a decrease of THC emissions. This resulted from the increase of air-fuel mixing, leading to better combustion. Above the engine speed of 1500 rpm, the increase of the engine speed results in a significant increase in THC emissions. This can be explained by the fact that, in this engine speed range, the engine was run with less gaseous fuel, which resulted in a decrease in the burnt gas temperature. This consequently leads to less oxidation of unburned hydrocarbons and higher THC emissions [18].

NO_x emissions from the internal combustion engine are the product of the oxidation of atmospheric nitrogen within the cylinder, which are directly proportional to the combustion temperature. The effect of engine speed on the NO_x emission is shown in

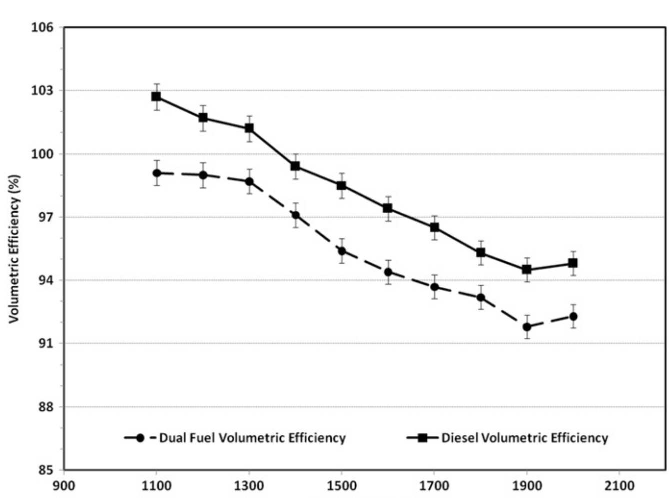

Fig. 5. Volumetric efficiency at various engine speeds.

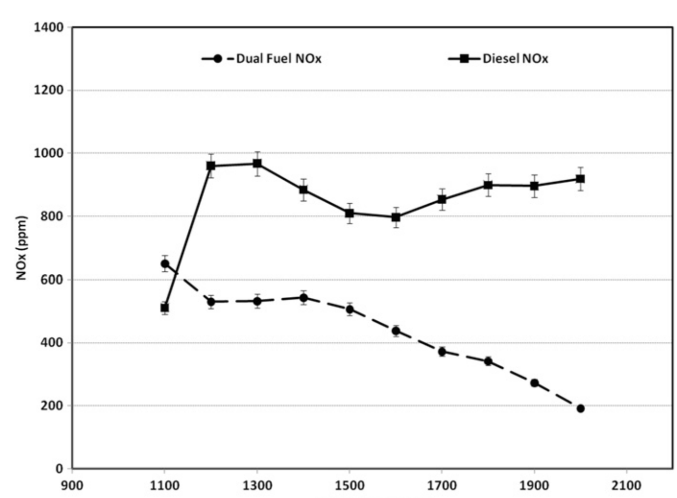

Fig. 7. NO_x emissions at various engine speeds.

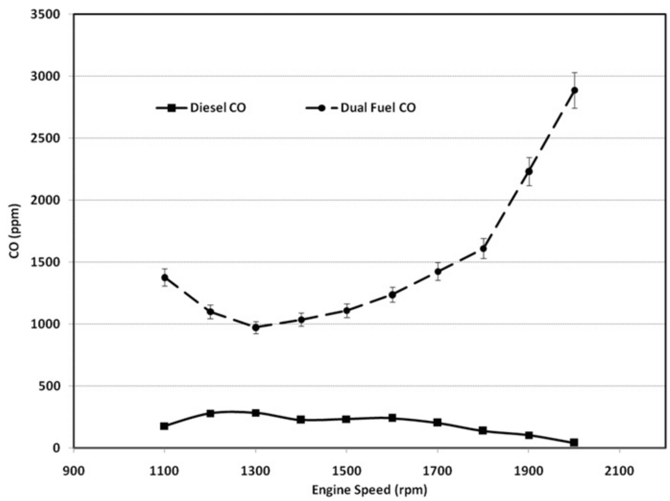

Fig. 8. CO emissions at various engine speeds.

Fig. 7. For both diesel and dual fuel operations, NO_x emissions slightly decreased with engine speed. As expected, the dual fuel operation significantly showed fewer NO_x emissions than with the diesel operation. This resulted from the lower combustion temperature. The results of this study are consistent with previous research [14,19,25].

Fig. 8 shows the variation of CO emissions with respect to engine speed. Normally, CO emissions in a diesel engine are low compared to those of a gasoline engine. Therefore, compared to diesel fuel operation, the dual fuel operation showed considerably high CO emissions for all of the engine speed ranges tested. As expected, the trend of the CO emissions was similar to that of the THC emissions. At an engine speed range between 1100 rpm and 1500 rpm, an increase in engine speed results in a decrease in CO emissions. This can be explained by the increase of the air-fuel mixing rate, leading to better combustion, as discuss previously [23]. Above the engine speed of 1500 rpm, CO emissions increased with engine speed. This can possibly be explained by the less time available for gaseous combustion, which possesses a low combustion rate and results in higher CO emissions [18,19].

CO_2 emissions are an indicator of combustion efficiency. Fig. 9 shows the effect of engine speed on the emissions of CO_2.

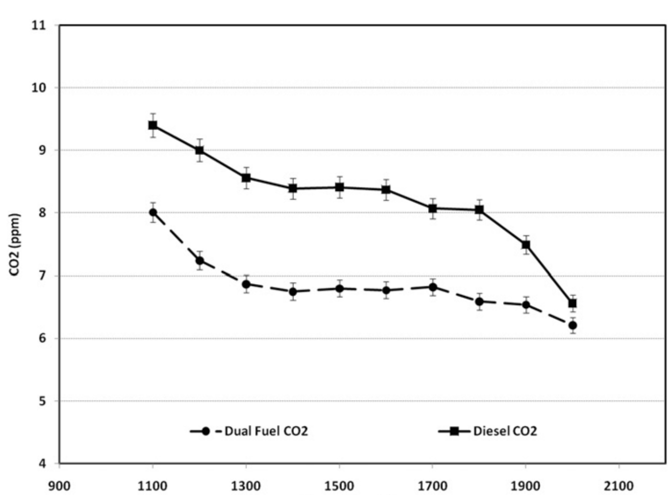

Fig. 9. CO_2 emissions at various engine speeds.

For both diesel and dual fuel operations, CO_2 emissions decreased as engine speeds increased because of the decrease in combustion efficiency. At higher engine speeds, the results were consistent with the increase of THC and CO emissions. However, at a low engine speed range, no correlation among CO_2, THC, and CO emissions was observed. This is due to the fact that, at this low engine speed range, the air-fuel ratios are lower than those at higher speed ranges, which yield higher CO_2 emission [12]. Compared to diesel operation, the dual fuel operation showed significantly lower CO_2 emission for all engine speeds due to the higher hydrogen-to-carbon ratio.

4. Conclusions

In this study, the effects of natural gas on a large diesel engine were investigated. Natural gas was used as the primary fuel, while a pilot amount of diesel was used as an ignition source. The amount of each fuel was adjusted to avoid knocking, and the brake torque and power output from the dual engine operation were comparable to those of diesel operation. The engine performance and emission characteristics were conducted at full load condition, and at an engine speed range of between 1000 rpm and 2000 rpm. The following conclusions can be drawn:

1. The portion of natural gas depends on the engine speed. For a speed range of between 1300 rpm and 1800 rpm, the portion of natural decreases with engine speed. The maximum portion of natural gas in the dual fuel engine operation was 77.90% at 1300 rpm.
2. The thermal efficiency of the dual fuel operation was less than that of the single fuel operation, averaging less than 3.50%. In terms of the practical speed range of a diesel engine, which is less than 1700 rpm, the specific fuel consumption of the dual fuel engine operation was slightly lower than that of the single operation.
3. For both diesel and dual fuel operations, volumetric efficiency increased with engine speed. The diesel engine operation showed higher volumetric efficiency than the dual fuel operation.
4. Compared to diesel operation, the dual fuel engine operation showed higher THC and CO emissions, while the emissions of NO_x and CO_2 were lower.

References

[1] Annual energy outlook 2011with projections to 2035 [Internet]. Washington DC: US energy information administration, c2011. Available from: electricdrive.org/index.php?ht=a/GetDocumentAction/id/27843, [last accessed 31.01.12].
[2] Kumar S, Kwon HT, Choi KH, Lim W, Jae HC, Tak K, et al. LNG: an eco-friendly cryogenic fuel for sustainable development. Applied Energy 2011;88:4264–73.
[3] LNG in Europe- an overview of European import terminals [Internet]. London: King & Spalding. Available from: www.kslaw.com/library/pdf/LNG_in_Europe.pdf; 2006c [last accessed 30.01.12].
[4] Powars C, Pope G. California LNG transportation fuel supply and demand assessment. Sacramento (CA): California Energy Commission; 2002 Jan. Report No.: P600-02-002F. Contract No.: 500-00-002.
[5] Verbeek R, Kadijk G, van Mensch P, Wulffers C, van den Beemt B, Fraga F. Environmental and economic aspects of using LNG as a fuel for shipping in the Netherlands. Wageningen: The Dutch Maritime Innovation Programme; 2011 Mar. Report No.:TNO-RPT-2011–00166 March 2011.
[6] Arteconi A, Brandoni C, Evangelista D, Polonara F. Life-cycle greenhouse gas analysis of LNG as a heavy vehicle fuel in Europe. Applied Energy 2010;87: 2005–13.
[7] Chandler K, Proc K. Norcal prototype LNG truck fleet: final results [Internet]. Golden (CO): National Renewable Energy Laboratory (US). Available from: www.afdc.energy.gov/pdfs/35427.pdf; 2004c [last accessed 03.05.12].
[8] Frailey M. Development of LNG-powered heavy-duty trucks in commercial hauling. Golden (CO): National Renewable Energy Laboratory (US); 1998 Dec. Report No.: NREL/SR-540–25154. Contract No.: Contract No. DE-AC36-98-GO10337.
[9] Nwafor OMI. Effect of advanced injection timing on the performance of natural gas in diesel engines. Sadhana 2000;25:11–20.

[10] Cheikh M, Abdelhamid B, Abdelkader A, Francoise G. Gas–diesel (dual-fuel) modeling in diesel engine environment. International Journal of Thermal Sciences 2001;40:409–24.
[11] Nwafor OMI. Knock characteristics of dual-fuel combustion in diesel engines using natural gas as primary fuel. Sadhana 2002;27:375–82.
[12] Nwafor OMI. Effect of advanced injection timing on emission characteristics of diesel engine running on natural gas. Renewable Energy 2007;32:2361–8.
[13] Karim GA. Combustion in gas fueled compression: ignition engines of the dual fuel type. Journal of Engineering for Gas Turbines and Power 2003;125:827–36.
[14] Abd Alla GH, Soliman HA, Badr OA, Abd Rabbo MF. Effect of injection timing on the performance of a dual fuel engine. Energy Conversion and Management 2002;43:269–77.
[15] Mohamed YE. Sensitivity of dual fuel engine combustion and knocking limits to gaseous fuel composition. Energy Conversion and Management 2004;45:411–25.
[16] Mohamed YE. Pressure–time characteristics in diesel engine fueled with natural gas. Renewable Energy 2001;22:473–89.
[17] Liu Z, Karim GA. An examination of the ignition delay period in gas-fueled diesel engines. Journal of Engineering for Gas Turbines and Power 1998;120:225–31.
[18] Papagiannakis RG, Hountalas DT. Experimental investigation concerning the effect of natural gas percentage on performance and emissions of a DI dual fuel diesel engine. Applied Thermal Engineering 2003;23:353–65.
[19] Papagiannakis RG, Hountalas DT. Combustion and exhaust emission characteristics of a dual fuel compression ignition engine operated with pilot diesel fuel and natural gas. Energy Conversion and Management 2004;45: 2971–87.
[20] Papagiannakis RG, Kotsiopoulos PN, Zannis TC, Yfantis EA, Hountalas DT, Rakopoulos CD. Theoretical study of the effects of engine parameters on performance and emissions of a pilot ignited natural gas diesel engine. Energy 2010;35:1129–38.
[21] Tippayawong N, Promwungkwa A, Rerkkriangkrai P. Long-term operation of a small biogas/diesel dual-fuel engine for on-farm electricity generation. Biosystems Engineering 2007;98:26–32.
[22] Henham A, Makkar MK. Combustion of simulated biogas in a dual-fuel diesel engine. Energy Conversion and Management 1998;39:2001–9.
[23] Abdelghaffar WA. Performance and emissions of a diesel engine converted to dual diesel-CNG fuelling. European Journal of Scientific Research 2011;56: 279–93.
[24] Willard WP. Engineering fundamentals of the internal combustion engine. New Jersey: USA: Prentice Hall; 2004.
[25] Korakianitis T, Namasivayam AM, Crookes RJ. Natural-gas fueled spark-ignition (SI) and compression-ignition (CI) engine performance and emissions. Progress in Energy and Combustion Science 2011;37:89–112.