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Effect of partial substitution of diesel fuel by natural gas on performance parameters of a four-cylinder diesel engine

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Abstract: This work presents the evaluation of some performance parameters with cost comparison for a four-cylinder diesel engine under a variety of operating conditions, before and after converting to dual fuel engine. The experiment used a converted dual fuel diesel engine which can run on a hybrid mixture of diesel and natural gas (NG). The focus of this study is on the engine thermal efficiency, exhaust gas temperature, lubricating oil temperature, air fuel ratio, specific fuel cost, and specific fuel consumption under different operating conditions. A comparison of the cost economy of pure diesel engine and gradually converted dual fuel engine is also performed. Performance parameters with engine power (kW) are compared as well. This study concludes that fuel cost can be reduced significantly by substituting diesel with NG. Few costly accessories are needed for converting stationary diesel engine into dual fuel engine without altering basic engine components. Other findings from this study include relation between brake thermal efficiency, brake specific fuel consumption, and other performance parameters of engine.

Keywords: diesel, natural gas, engine performance, dual fuel, hybrid fuel

1 INTRODUCTION

Fossil fuel has been the de-facto standard of energy consumption in the world. Emissions produced by the combustion of fossil fuel in the form of exhaust gas, dust, ash, and clinkers are penetrating the earth severely. These exhaust emissions have hazardous effect on the environment, which causes immediate local effect and creates global effect gradually [**1**]. Not only does the continued use of large amounts of fossil fuels pose a serious threat to the environment, but also the limited fuel reserves are creating continuous pressure to look for alternatives. Although there are debates amongst experts about the extractable amount of fossil fuels, general opinion indicates that almost half of the fossil fuel will be consumed at the beginning of the 21st century. The known present worldwide reserves of petroleum are 1000 billion barrels and these petroleum reserves are predicted to be consumed in about 40 years [**2**]. The other problem with petroleum as mentioned earlier is the emission of pollutants, such as CO_2, NO_X, CO, and hydrocarbons (HC). Much greater amounts of coal are known to exist but existing coal burning technologies are susceptible to much more pollution than most other fuels, particularly in terms of the green-house gas emissions per unit of useful energy released. The search for alternative fuel to cope with these two major problems was never over and researchers all over the world have tried several options till now. From several studies, the alternative fuels such as methane, hydrogen, and mixtures of hydrogen and methane found to be useful with respect to the environmental effects [**3**]. Natural gas (NG) can be considered as more

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environmental friendly in this regard and can prove to be useful if found abundantly. The goal of current research is to justify the possible use of NG as a second fuel in a hybrid dual fuel diesel engine and thus provide another way of dealing with the environmental problem associated with the conventional fuels.

2 NATURAL GAS IN A DUAL FUEL DIESEL ENGINE

NG is a mixture of different gases. Concentration of these gases may differ from one source to another. One component of NG is methane, which is typically up to 99 per cent of the total volume [**4**]. The composition of NG is never constant. Other constituents may include non-methane HC such as ethane, propane, and butane and in some cases, traces of higher HC. They may also include inert gases like nitrogen, helium with carbon dioxide, hydrogen sulphide, and sometimes water particles. NG also has some excellent properties as a fuel. It is the cleanest of all fossil fuels and present world reserve is 171 trillion-metre cubes, which is more than the same of crude oil [**5**]. Moreover, it is easy to transport and store, available on demand, and in many areas, it is cheaper than gasoline and diesel. It is available in huge quantities in many parts of the world, thus it is even cheaper in those areas. From an engineering perspective, a high-octane value of NG makes it suitable for engines with high compression ratio. As an added bonus, it mixes uniformly with air, resulting in efficient combustion with possible improvement on engine power output and efficiency. The most economical and environmental friendly choice where NG is easily available would be to replace conventional fuel by NG with modification to the existing engine. However, the required modification to the existing engine may not be economical and even impossible in some cases. The lucrative choice of NG has propelled the development of dual fuel engine, where engine can use hybrid mixture of fuel.

There have been many published works on the use of gaseous fuels in dual fuel engines. The use of NG in dual fuel engines has been studied in many literatures. The combustion duration and ignition delay [**6**, **7**], the performance and emissions [**8**, **9**], and combustion and knocking [**10**] are some of the parameters studied in earlier researches. Combustion and thermal loading and temperature distribution have also been studied for dual fuel engines [**11**]. Reference [**3**] studies pure methane in dual fuel engines and discusses the flame spread limits. Current study seeks for the possible use of NG in dual fuel engine, where the other conventional fuel is diesel.

Because of relatively high spontaneous ignition temperature of the compressed gas and air mixture, it does not self ignite. Therefore, either a spark plug is needed or a minimum amount (at least 10 per cent;) of fuel must be injected to ignite the mixture [**12**]. If a spark plug is installed instead of an injector, the engine is called a gas engine and if fuel is injected for auto ignition, the engine is called dual fuel engine. Dual fuel engines can operate on both NG and diesel fuel simultaneously. During normal operation, the majority of the burnt fuel is NG. This technology combines the benefits of a diesel engine, namely quick response to load change, reliability, high efficiency, and low cost, with the benefits of a NG engine, namely low emissions and typically lower specific fuel price, to provide a high performance, low emission, cost-effective, cleaner system that may also reduce fuel costs. Already millions of diesel engines are running in the world. It may not be economically suitable to replace all of the existing diesel engines by NG engine right now. However, it is possible to convert some of them into NG engine or dual fuel engine, which will reduce load on conventional fuel reserve.

3 EXPERIMENTAL SETUP, PROCEDURE AND SUBSTITUTION TECHNIQUES

3.1 Experimental setup

In this experiment, a four-cylinder four-stroke stationary diesel engine was used. Schematic layout of the test set up is shown in Fig. 1 and the technical specifications of the engine are given in Table 1. The air cleaner was removed from the intake manifold to facilitate the measurement of airflow. A flexible hose-pipe with 6.5 cm diameter and 180 cm length was attached to the intake manifold. The other end of the pipe is connected to an air drum via metal pipe. The metal airs drum of diameter 35 cm and length 70 cm had a parabolic nozzle on top of it. A water manometer was connected to the air drum. Air flow-rate was measured from the difference of the water height of the manometre. The length and diameter of gas inlet netting pipe to the engine intake port is 300 cm and 2 cm, respectively. The netting pipe was connected to a custom made T-shape-mixing chamber of metal. The mixing chamber was attached into intake manifold. The mixing chamber was used to mix NG and air before entering the combustion chamber. The other end of the netting pipe was connected to the commercial NG supply line. A gas control valve and gas flow meter (rotameter) was connected to the gas flow tube. NG with a pressure of about 101.4 kPa was supplied to the engine from the commercial NG supply line. A rotameter was

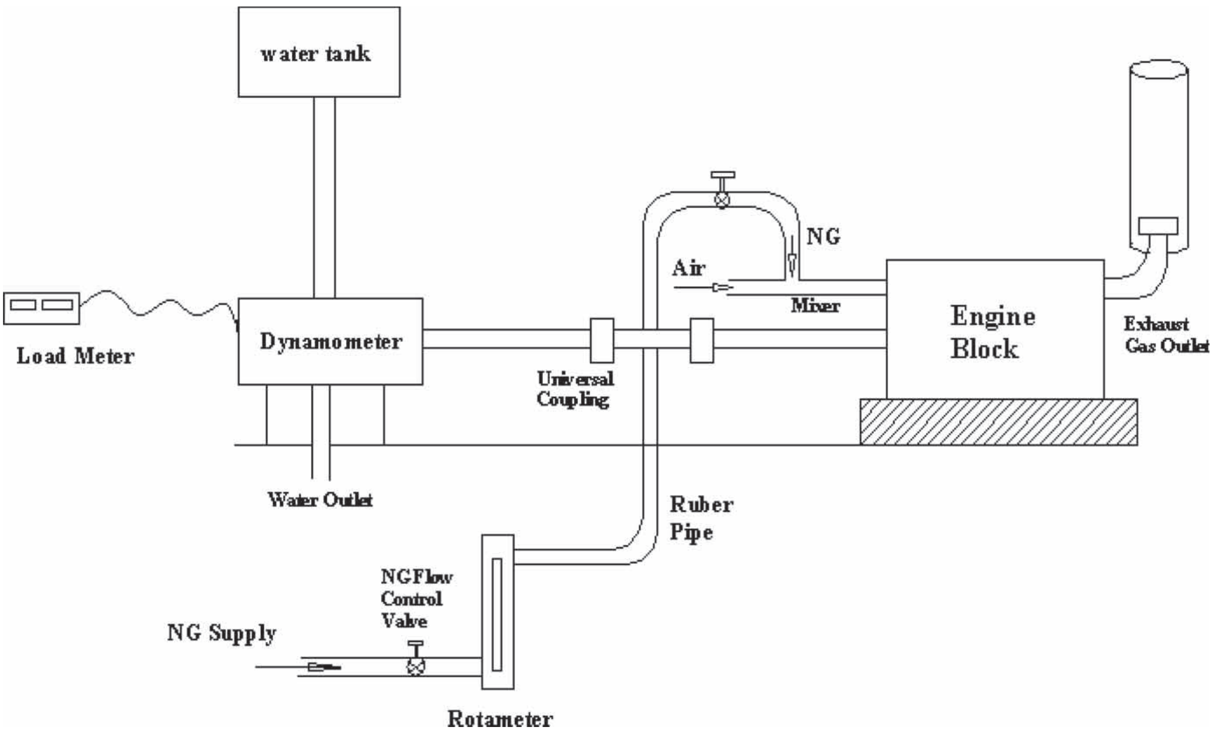

Fig. 1 Schematic diagram of experimental setup

connected between the NG supply line and the intake manifold of the engine. The rotameter gave the NG consumption in l/min. The output shaft of the engine was connected to a brake dynamometer by universal coupling. The dynamometer was connected with a load cell which gave readings of load and r/min. The exhaust gas and lubricating oil temperatures were measured by digital thermometer. The tip of the thermocouple of exhaust gas temperature was admitted through the hole in the exhaust manifold. The tip of another thermocouple was inserted through the plug of lubricating oil. The fuel tank of the engine was also removed. A long pipe was connected to the fuel inlet port in which diesel flew from a fuel tank fed by gravity. The tank was kept at a higher place to have a high pressure head. A measuring burette with a gate valve was also connected in the fuel pipe. This measuring burette gave fuel consumption rate with the help of a stopwatch. A sling psychometer was used for measuring the wet bulb and dry bulb temperatures. A hydrometer was used to measure the specific gravity of diesel.

Table 1 Specifications of the engine

Model	V1502
Manufacturer	KUBOTA diesel company Ltd, Japan
No. of cylinder	4 (four)
Rated power	17 kW
Speed	2250 r/min
Compression ratio	17.9: 1
Rotation	Anticlockwise
Lubricating system	Forced circulation
Fuel injection	Direct injection system

3.2 Substitution techniques

In conventional diesel engines, only air is admitted into the intake manifold. After the compression of air, diesel is injected into it and the heat of compressed air ignites the diesel. The self-ignition temperature of diesel is about 534 °C and the temperature produced after adiabatic compression of air in the engine cylinder is about 678 °C. As diesel is injected at the end of compression, there is no possibility of pre-ignition and the temperature of compressed air becomes sufficient to ignite the diesel. The self-ignition temperature of NG is about 700 °C. Therefore, NG can be admitted with air into the engine cylinder without any possibility of pre-ignition because the temperature produced at the end of compression is not sufficient to ignite the mixture. Hence, small amount of diesel is needed to ignite the gas–air mixture and the compression ratio will remain unchanged. Thus, by supplying gas (NG) through the intake manifold of the engine with air at the suction stroke and injecting some diesel, a diesel engine can be converted into dual fuel engine.

3.3 Experimental procedure

The dynamometer was calibrated and the specific gravity of diesel was measured with it. The readings of wet bulb temperature (wbt) and dry bulb temperature (dbt) were also taken. The load metre was connected to the dynamometer and was allowed to warm up for about 10 min. The inlet valve of water supply line to the dynamometer was opened slightly. Then the engine was started at low load using 100 per cent diesel. The dynamometer and the diesel flow control valve were adjusted at 2250 r/min and with load about 12.5 kg. At this condition, the engine was kept running for about 20 min to attain steady-state condition. Engine speed, diesel consumption, time of diesel consumption, manometer reading for air flow, gas consumption, time of gas consumption, water cooling temperature, exhaust gas temperature, lubricating oil temperature, etc. were recorded at this condition. For 100 per cent diesel operation, obviously there was no reading of gas consumption. The same readings were taken for nearly same r/min 2250 and load 12.5, 13.9, 15.7, 17.9, 19.7, 22.2, 24.7 kg, respectively.

After taking all the readings at 100 per cent diesel, the NG was introduced to the engine cylinder by opening the gas control valve slowly. First, data of second experiment were taken by substituting 10 per cent diesel with gradual admission of NG flow. Then, the dynamometer load control handle, fuel flow, and gas control valves were adjusted to set the engine at load 12.5 kg with 90 per cent diesel and NG (equivalent to the remaining 10 per cent diesel). It was observed that at 12.5 kg load with 2250 r/min, 30 cc diesel flowed at 38 sec. The more NG was introduced, the more time was required for the flow of 30 cc diesel. For the reduction of 10 per cent diesel flow, NG gas flow was achieved by error and trial method at the point where 30 cc diesels flowed at 42 sec. The flowrate of NG was measured which was equivalent to 10 per cent diesel. After taking first data with 12.5 kg load, engine load was increased by increasing NG flow only and second set of data was taken at 13.9 kg load. Diesel and gas flowrate also has been measured at 13.9 kg dynamometer load. NG flows further increased by gas flow control bulb to reach dynamometer load close to 15.7, 17.7, 20.7 22.2, and 24.7 kg. Other data sets were obtained by taking first data with 20 per cent, 30, and 40 per cent substitution of diesel with NG by same procedure. Up to 64 per cent diesel could substitute with NG at the last experiment, which was started with 40 per cent substitution of diesel. The gas pressure of commercial supply line, which was used as source of NG in this study, was not enough to substitute more than 64 per cent diesel with NG. For the above conditions, the engine was tried to operate at constant speed (2250 r/min). However, small changes of engine speed were occuring with the increase of load which was considered in the calculation of present study. After taking complete set of readings for the above fuel supply, the gas supply was closed first, then the load was decreased, and diesel supply was gradually decreased to stop the engine. The final readings of dbt, wbt, and specific gravity of diesel were noted down.

4 RESULTS AND DISCUSSIONS

Five different sets of experiments were carried out with different proportions of diesel and NG in this study. Experiment on the engine with only diesel fuel was carried out in the very first experiment. Then experiments went on substituting 10 per cent diesel by NG and increasing the load gradually with increasing NG only. Same technique was applied for conducting experiments with 20, 30, and 40 per cent substituted diesel. The experimental data and results are shown in Tables 2 to 6 and the performance graphs obtained from the results are shown in Figs 2 to 7. On the legend of Figs 2 to 7, 0 per cent substituted diesel curve represents the performance of the engine while running only on diesel. On the other hand, curves corresponding to 10, 20, 30 and 40 per cent represent the sets of experiments, which were started with substituting 10, 20, 30, and 40 per cent diesel with NG, respectively. Maximum substitution of diesel achieved by NG was 64 per cent in this study. Engine performances curve did not show any downward trend up to 64 per cent NG and 36 per cent diesel mixture. More substitution of diesel by NG could not be achieved because of the lack of NG pressure on commercial supply line in the laboratory. It could be possible to go over 64 per cent NG substitution if this limitation could be overcome.

In this investigation, NG was admitted into the intake manifold of the air and was mixed with air using the cross shape mixer and it is expected that the gas concentration was more or less uniform across the cross-section. Then, the mixture passed through the pipe and it became more homogenous. Usually in a diesel engine, diesel is injected into the compressed air in the cylinder. Therefore, the diesel does not get much time to mix with air homogenously. As a result it can be considered that the proposed type of dual fuel (NG-diesel) engine gets more homogenous mixture (air/gas) than the conventional diesel engine. Hence, the combustion was expected to be better in dual fuel than in pure diesel operation.

Figure 2 shows the variation of total break specific fuel consumption (bsfc) with engine power (kW) at different percentage of diesel substitution. As shown, at low load, total brake specific fuel consumption for dual fuel operation is found to be considerably higher compared to the one under normal

Table 2 Experimental data for engine running with diesel

No. of Obs	Dynamo-meter load W (Kg)	Engine speed (r/min)	kW at lab condition	kW at BS condition	Bsdc lab cond. (gm/ kW h)	Bsdc BS cond (gm/ kW h)	Bstfc lab (gm/ kW h)	Bstfc BS (gm/ kW h)	Amount of diesel (cc)	Time of Collection (s)	Gas flowrate (l/min)
1	12.50	2253	7.00	7.28	342.81	341.32	342.81	341.32	30.00	38.00	0.00
2	15.30	2252	9.13	9.49	293.95	292.66	293.95	292.66	30.00	34.00	0.00
3	17.60	2252	9.85	10.24	308.53	307.19	308.53	307.19	30.00	30.00	0.00
4	20.20	2252	11.30	11.75	304.33	303.00	304.33	303.00	30.00	26.50	0.00
5	22.50	2254	12.61	13.10	307.82	306.48	307.82	306.48	30.00	23.50	0.00
6	25.10	2252	14.05	14.60	309.07	307.71	309.07	307.71	30.00	21.00	0.00
7	27.60	2250	15.44	16.04	328.20	326.78	328.20	326.78	30.00	18.00	0.00

No. of Obs	Mass flowrate of diesel (gm/h)	Eq. mass flowrate of diesel	Total mass flowrate of fuel(gm/h)	Manometric Deflection (mm)	Air flowrate (Kg/h)	Air fuel ratio	Brake thermal Efficiency	Lub oil temp (°C)	Exhaust gas temp (°C)	Cost of diesel (Tk/ kW h)
1	2399.74	0.00	2399.74	41.50	90.83	37.85	24.71	78	337	8.89
2	2682.06	0.00	2682.06	41.00	90.28	33.66	28.82	92	384	7.62
3	3039.67	0.00	3039.67	40.00	89.18	29.34	27.45	101	428	7.99
4	3441.14	0.00	3441.14	40.00	89.18	25.91	27.83	105	476	7.89
5	3880.43	0.00	3880.43	59.00	88.05	22.69	27.52	110	532	7.98
6	4342.39	0.00	4342.39	38.80	87.83	20.23	27.41	115	579	8.01
7	5066.12	0.00	5066.12	37.50	86.34	17.04	25.81	119	620	8.50

Table 3 Data of experiment stared with 10 per cent diesel substitute by NG

No. of Obs	Dynamo-meter load W (Kg)	Engine speed (r/min)	kW at lab condition	kW at BS condition	Bsdc lab cond. (gm/ kW h)	Bsdc BS cond (gm/ kW h)	Bstfc Lab (gm/kW h)	Bstfc BS (gm/kW h)	Amount of diesel (cc)	Time of collection (s)	Gas flow rate (l/min)
1	12.10	2248	6.76	7.02	313.65	312.30	350.77	349.24	30.00	43.00	5.00
2	13.90	2248	7.77	8.07	266.84	265.67	331.45	330.01	30.00	44.00	10.00
3	15.70	2247	8.77	9.11	231.10	230.09	316.95	315.57	30.00	45.00	15.00
4	17.90	2248	10.01	10.40	198.20	197.33	298.55	297.25	30.00	46.00	20.00
5	17.70	2248	9.89	10.28	196.18	195.32	323.04	321.63	30.00	47.00	25.00
6	22.20	2248	12.41	12.89	153.14	152.49	274.52	273.33	30.00	48.00	30.00
7	24.70	2248	13.80	14.34	136.23	135.63	263.46	262.36	30.00	48.50	35.00

No. of Obs	Mass flowrate of diesel (gm/h)	Eq.mass flowrate of diesel	Total mass flowrate of fuel (gm/h)	Manometric deflection (mm)	Air flowrate (Kg/h)	Air fuel ratio	Brake thermal Efficiency	Lub oil temp (°C)	Exhaust gas temp (°C)	Total cost of fuel (Tk/ kW h)
1	2120.70	250.94	2371.64	40.80	90.06	37.98	24.15	102	312	8.34
2	2072.50	501.88	2574.39	40.00	89.18	34.64	25.56	107	342	7.28
3	2026.45	752.82	2779.27	39.50	88.62	31.89	26.73	109	373	6.48
4	1982.40	1003.76	2986.16	38.70	87.72	29.37	28.37	111	414	5.70
5	1940.22	1254.71	3194.92	38.20	87.15	27.28	26.22	114	447	5.81
6	1899.80	1505.65	3405.44	37.50	86.34	25.35	30.86	117	491	4.65
7	1880.21	1756.59	3636.80	36.30	84.95	23.36	32.15	122	537	4.25

Table 4 Data of experiment stared with 20 per cent diesel substitute by NG

No. of Obs	Dynamo-meter load W (Kg)	Engine Speed (r/min)	kW at lab condition	kW at BS condition	Bsdc lab cond. (gm/ kW h)	Bsdc BS cond (gm/kW h)	Bstfc Lab (gm/kW h)	Bstfc BS (gm/kW h)	Amount of diesel (cc)	Time of collection (s)	Gas flowrate (l/min)
1	12.50	2250	6.99	7.26	289.86	288.51	361.66	360.09	30.00	45.00	10.00
2	14.20	2251	7.94	8.25	239.12	238.07	327.54	326.11	30.00	48.00	14.00
3	17.20	2250	9.62	10.00	201.69	200.82	306.05	304.71	30.00	47.00	20.00
4	19.30	2250	10.79	11.22	177.86	177.08	294.10	292.81	30.00	47.50	25.00
5	21.10	2250	11.80	12.27	158.63	158.63	282.66	281.44	30.00	48.50	29.00
6	22.50	2251	12.59	13.08	144.24	144.24	272.44	271.25	30.00	50.00	32.00
7	24.20	2249	13.53	14.06	134.22	135.34	264.65	263.51	30.00	50.00	35.00
No. of Obs	Mass flowrate of diesel (gm/h)	Eq.mass Flowrate of diesel	Total mass flowrate of fuel (gm/h)	Manometric deflection (mm)	Air flowrate (Kg/h)	Air fuel ratio	Brake thermal efficiency	Lub oil temp (°C)	Exhaust gas temp (°C)	Total cost of fuel (Tk/ kW-hr)	
1	2026.45	501.88	2528.33	41.50	90.83	35.93	24.34	112	337	7.93	
2	1899.80	702.63	2602.43	40.50	89.73	34.48	26.88	113	362	6.71	
3	1940.22	1003.76	2943.98	39.50	88.62	30.10	28.76	119	415	5.82	
4	1919.79	1254.71	3174.49	38.50	87.48	27.56	29.93	117	446	5.27	
5	1880.21	1455.46	3335.67	38.00	86.92	26.05	31.14	119	479	4.83	
6	1823.80	1606.02	3429.83	38.20	87.15	25.41	32.31	120	506	4.48	
7	1823.80	1756.58	3580.39	37.00	85.77	23.95	33.26	122	534	4.24	

Table 5 Data of experiment stared with 30 per cent diesel substitute by NG

No of Obs	Dynamo-meter load W (Kg)	Engine Speed (r/min)	kW at lab condition	kW at BS condition	Bsdc lab cond. (gm/ kW h)	Bsdc BS cond (gm/ kW h)	Bstfc lab (gm/ kW h)	Bstfc BS (gm/ kW h)	Amount of diesel (cc)	Time of Collection (s)	Gas flowrate (l/min)
1	12.90	2240	7.18	7.33	230.83	230.19	335.64	334.71	30.00	55.00	15.00
2	13.50	2240	7.52	7.67	209.17	208.58	329.35	328.43	30.00	58.00	18.00
3	14.90	2241	8.30	8.47	180.11	179.60	313.14	312.27	30.00	61.00	22.00
4	16.20	2241	9.02	9.21	160.40	159.96	299.44	298.61	30.00	63.00	25.00
5	18.00	2240	10.02	10.22	142.18	141.77	287.39	286.59	30.00	64.00	29.00
6	19.70	2240	10.97	11.19	127.91	127.54	274.32	273.56	30.00	65.00	32.00
7	21.40	2240	11.92	12.15	115.96	115.64	263.38	262.64	30.00	66.00	35.00
No. of Obs	Mass flowrate of diesel (gm/h)	Eq.mass flowrate of diesel	Total mass flowrate of fuel (gm/h)	Manometric deflection (mm)	Air flowrate (Kg/h)	Air fuel ratio	Brake thermal efficiency	Lub oil temp (°C)	Exhaust gas temp (°C)	Total cost of fuel (Tk/kW h)	
1	1658.00	752.82	2410.83	41.00	90.28	37.45	25.24	93	326	6.60	
2	1572.24	903.39	2475.63	40.50	89.73	36.25	25.72	101	338	6.13	
3	1494.92	1104.14	2599.06	40.00	89.18	34.31	27.05	105	363	5.44	
4	1447.46	1254.71	2702.17	40.00	89.18	33.00	28.29	107	387	4.96	
5	1424.85	1455.46	2880.31	39.00	88.05	30.57	29.47	110	413	4.53	
6	1402.93	1606.02	3008.95	38.50	87.49	29.08	30.88	112	443	4.17	
7	1381.67	1756.59	3138.26	38.00	86.92	27.70	32.16	115	468	3.86	

Table 6 Data of experiment stared with 40 per cent diesel substitute by NG

No. of Obs	Dynamo-meter load W (Kg)	Engine speed (r/min)	kW at lab condition	kW at BS condition	Bsdc lab cond. (gm/kW h)	Bsdc BS condition (gm/kW h)	Bstfc lab (gm/kW h)	Bstfc BS (gm/kW h)	Amount of diesel (cc)	Time of collection (s)	Gas flowrate (l/min)
1	12.50	2244	6.97	7.04	198.16	197.89	338.53	338.06	30.00	66.00	19.50
2	13.50	2243	7.52	7.60	165.96	165.74	332.67	332.20	30.00	73.00	25.20
3	14.60	2243	8.14	8.22	141.81	141.61	320.61	320.16	30.00	79.00	29.00
4	16.60	2244	9.26	9.35	121.59	121.42	295.04	294.62	30.00	81.00	32.00
5	18.60	2244	10.37	10.48	104.64	104.49	273.96	273.57	30.00	84.00	35.00
6	20.90	2243	11.66	11.77	92.07	91.94	255.73	255.38	30.00	85.00	38.00

No. of Obs	Mass flowrate of diesel (gm/h)	Eq. mass flowrate of diesel	Total mass flowrate of fuel (gm/h)	Manometric deflection (mm)	Air flowrate (Kg/h)	Air fuel ratio	Brake thermal efficiency	Lub oil temp (°C)	Exhaust gas temp (°C)	Total cost of fuel (Tk/kW h)
1	1381.67	978.67	2360.34	42.00	90.28	38.25	25.02	93	316	5.97
2	1249.18	1254.71	2503.89	40.00	89.18	35.62	25.46	102	335	5.28
3	1154.31	1455.56	2609.77	39.80	88.95	34.08	26.42	103	354	4.73
4	1125.80	1606.02	2731.83	39.50	88.62	32.44	28.71	111	390	4.17
5	1085.60	1756.59	2842.19	39.00	88.05	30.57	29.47	112	420	3.70
6	1072.83	1907.15	2979.98	38.50	87.49	29.08	30.88	113	450	3.35

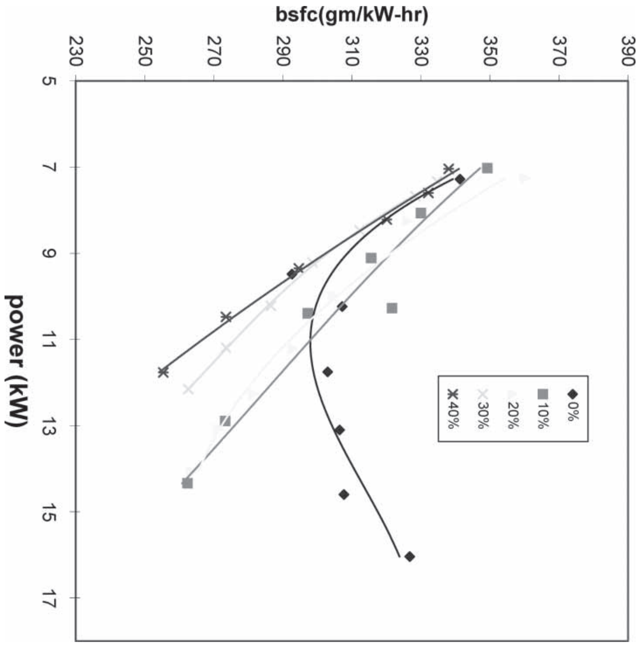

Fig. 2 Brake specific fuel consumption (bsfc) versus power (kW)

diesel operation. This reveals a poor utilization of the gaseous fuel due mainly to the lower temperature and air fuel ratio inside the combustion chamber of the engine, resulting in a slower combustion rate as observed from the results of the heat release rate analysis [**13, 14**]. On the other hand, at high load, the improvement of gaseous fuel utilization leads to a relevant improvement of the total brake specific

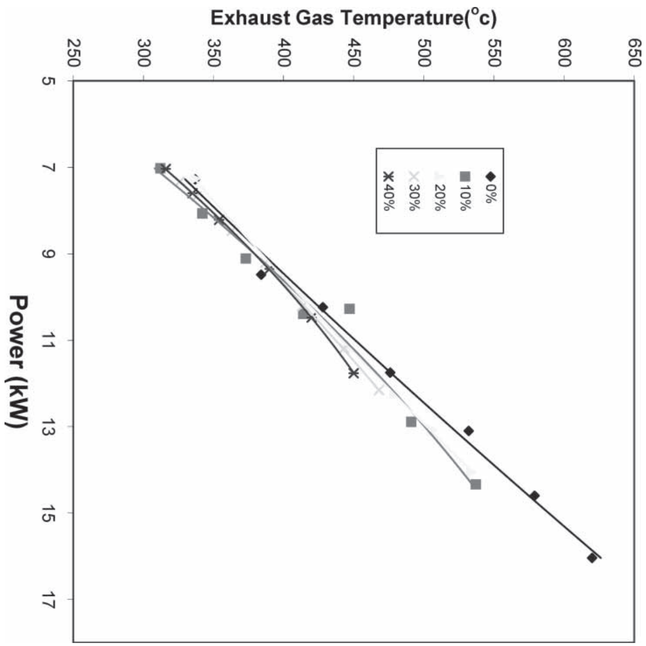

Fig. 3 Exhaust gas temperature (°C) versus power (kW)

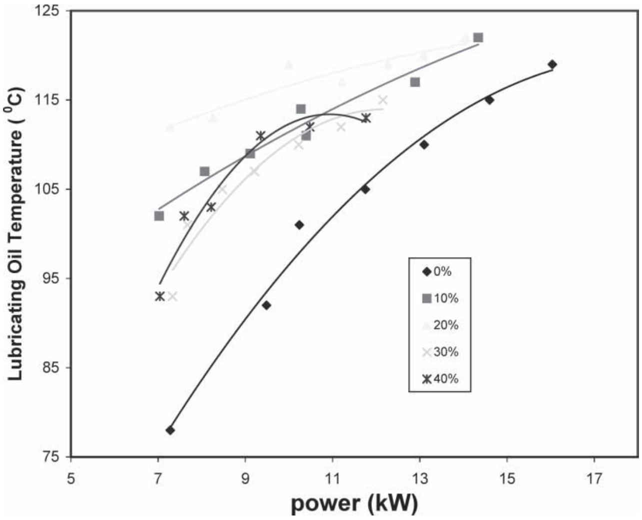

Fig. 4 Lubricating oil temperature (°C) versus power (kW)

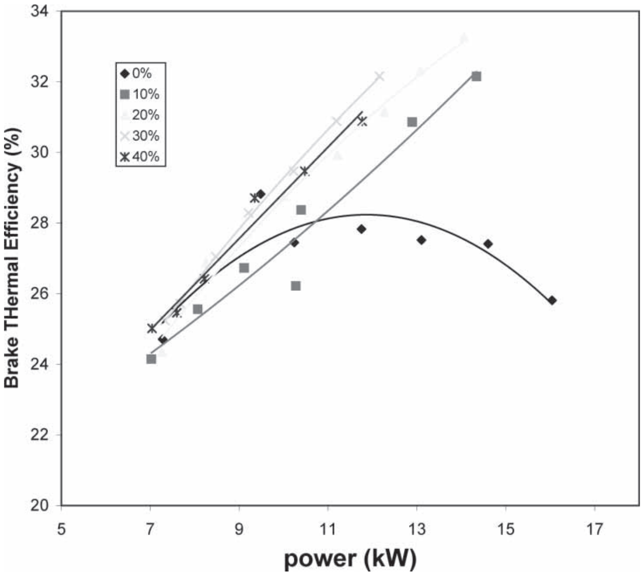

Fig. 6 Brake thermal efficiency versus power (kW)

fuel consumption under dual fuel operation, which tends to converge to the one under normal diesel operation, but its value continues to get higher compared to the one under normal diesel operation. It must be stated here that the heating value of NG is higher compared to the diesel fuel used, and thus, the total bsfc under dual fuel operation would be even higher if it were corrected to the heating value of the diesel fuel.

Variation of exhaust gas temperature with engine power (kW) for different diesel consumption is shown in Fig. 3. This figure shows that for any given fuel consumption, the exhaust gas temperature increases as kW increases. The curves of low percentage of diesel are not so steep like 100 per cent diesel. The increase of exhaust gas temperature is due to higher fuel consumption at higher engine power. At lower percentage of diesel and at lower engine power (kW), the small quantity of diesel supply at this time is not sufficient to produce the right environment to burn all the gases. Therefore, presumably burning continued till the end of expansion stroke and that caused high temperature of the exhaust gas. As engine power (kW) increases and more diesel is introduced, it produces the right environment to burn all the gases properly in due time and the exhaust gas temperature do not rise so high. That is why exhaust gas temperature curves at

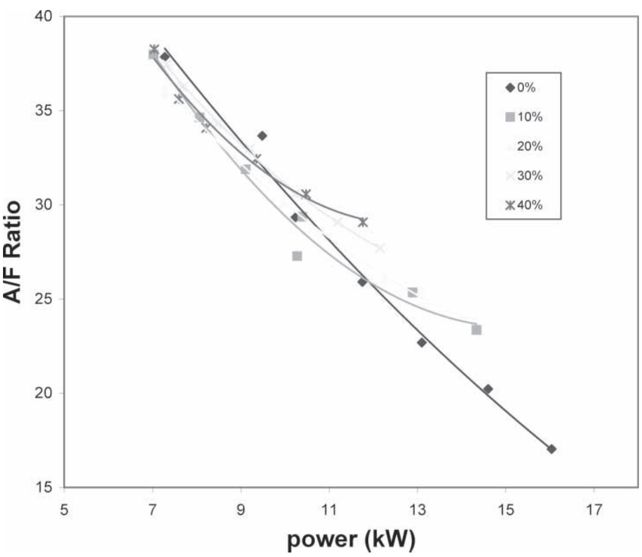

Fig. 5 A/F ratio versus power (kW)

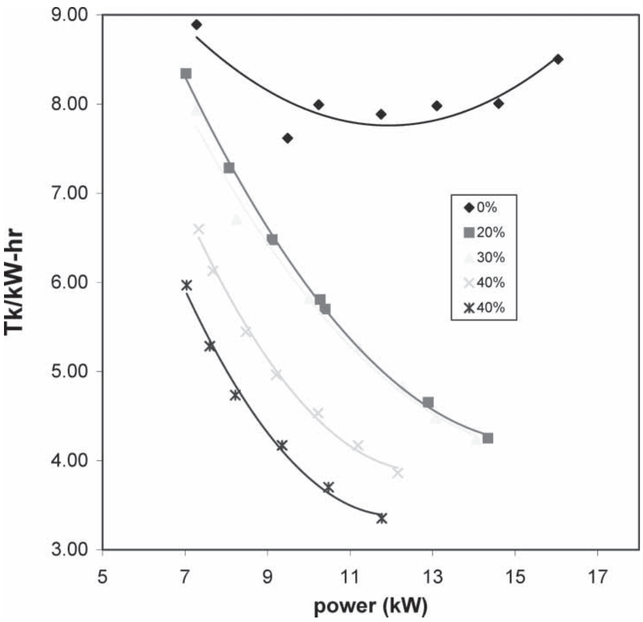

Fig. 7 Specific cost (Tk/ kW h) versus power (kW)

lower percentage of diesel are not as steep as the curve of 100 per cent diesel.

Figure 4 shows the variation of lubricating oil temperature with engine power (kW) for different percentage of diesel consumption. From this Figure it can be seen that for a given fuel consumption, lubricating oil temperature increases as engine power (kW) increases. Curves for diesel consumption less then 100 per cent is above the curve of 100 per cent diesel consumption. It is found from the experimental data (Tables 2 to 6) air flowrate decreases as engine power (kW) increases. The probable cause of less air flowrate at higher engine power (kW) is that the engine draws the same quantity of air in each suction stroke, but the increase in engine power (kW) is changed by increased fuel supply, thus producing more heat in the engine cylinder. As lubricating oil takes away heat from some parts of engine cylinder, lubricating oil temperature increases with increasing engine power (kW).

Figure 5 shows the variation of engine power (kW) with air fuel ratio. It is seen from this figure that engine power (kW) decreases as air fuel ratio increases. However, the rate of decrease of engine power (kW) with AFR is not same for all cases. The engine draws same quantity of air in each suction stroke, but the engine power (kW) increase is caused by increased fuel supply. At lower AFR, air–fuel mixture is rich that facilitates more burning and therefore engine power (kW) is higher at lower AFR. Engine power (kW) gradually decreases with increasing AFR, because at higher AFR lean mixture causes reduced burning.

Figure 6 shows the variation of brake thermal efficiency with engine power (kW). It is seen from the graph of 100 per cent diesel consumption that the efficiency increases with the increase in engine power (kW) up to a certain limit. After that the efficiency falls. With partial substitution of diesel by NG, the efficiency curve shows no downward drift within the range of operation. Comparing the cases as explained above, it can be concluded that the engine running with lower percentage of diesel in the fuel can produce higher outputs. After entering in the line, NG mixes with air and poses some path before entering into engine cylinder. Therefore, NG mixes with air homogenously. Moreover, NG is lighter fuel than diesel. Almost 100 per cent of NG is combustible, whereas diesel contains some components which do not burn easily. Because of the fact of homogeneity effect and clear burning nature of NG cause more energy production in dual fuel operation.

From Fig. 7 it is possible to make an assumption about the variation of cost at different percentage consumption of diesel. It can be seen from Fig. 7 that for 100 per cent diesel consumption, the specific cost (Tk/kW h, 1 USD $\sim$ 63 BDT) is higher at lower engine power (kW). Cost reduces slightly up to a certain engine power (11.75 kW) and then increases with engine power (kW). The most economic operating point for 100 per cent diesel consumption is at 11.75 kW that costs 8.89 Tk/kW h. From the observation of the curves for NG consumption, it is prominent that cost per kW h is less than 100 per cent diesel consumption at lower kW and the cost falls rapidly as the kW increases. Further consumption of NG will reduce the cost further. Figure 7 shows that reduction of fuel cost per kW hr was continued with increasing the substitution of diesel by NG. Fuel cost was 3.35 Tk/kW h when engine was running with 64 per cent NG and 11.77 kW power.

Thus, from the results of present study it can be concluded that two-third of fuel cost reduction is possible by running diesel engines with diesel–NG together.

5 CONCLUSION

The efficient running of the engine largely depends on the property of fuel used. Generally, the main component of NG is methane. NG used in the present study was obtained from Bangladesh. The density of NG was 0.79 Kg/m^3 and contains 96–99 per cent methane. The property of NG depends mostly on property of methane. The octane rating of methane is about 120 and it has been found that the flame speed of methane is less than that of diesel fuel. When the engine ran with less NG, the homogeneity effect was prominent than the flame speed. However, when the engine ran with more NG, the time required for combustion got increased due to less flame speed of NG than diesel. Thus, the flame speed was prominent than homogeneity effect and inefficient combustion was obtained. Moreover, diesel engine runs inefficiently at lower load and efficiently at higher load. Therefore the combustion of dual fuel engine depends mainly on three factors. These are homogeneity effect, flame speed, and load.

At lower engine power (kW), the bsfc of dual fuel operation with respect to 100 per cent diesel operation is higher and at higher engine power (kW) it is lower. For any given fuel consumption, the exhaust gas temperature increases as engine power (kW) increases. For a given AFR, engine power (kW) increases with partial substitution of diesel. Brake thermal efficiency is higher at higher NG consumption. At market price in January 2004 in Bangladesh, the specific fuel cost (Tk/kW h) to operate the engine is much less in case of higher substitution of diesel.

Experiments could be conducted up to 64 per cent diesel substitution with NG in this study. From the results, it can be concluded that more substitution is possible if the limitation in the NG supply was overcome. Emissions data could not be taken for the lack of proper equipment in the laboratory. The emission study of dual fuel diesel engine can be performed in future. The substitution technique used in the present study is suitable only for stationary diesel engines especially those used in power plants and irrigation.

REFERENCES

1 **Weaver, C. S.** Natural gas vehicles – a part of the state of art sierra research inc. SAE paper no. 892133, Sacramento, CA, USA, 1989.
2 **DeLuchi, M. A., Johnston R. A.,** and **Sperling D.** Methanol vs. natural gas vehicles: a comparison of resource supply, performance, emissions, fuel storage, safety, cost and transitions. SAE paper no. 881656, University of California, Davis, CA, USA, 1988.
3 **Douvile, B.** *Performance, emissions and combustion characteristics of natural gas fueling of diesel engines*. MASc Thesis, University of British Columbia, Canada, 1994.
4 **Akansua, O. S., Dulger, Z., Kahraman, N.,** and **Veziroglut, N.** Internal combustion engines fueled by natural gas–hydrogen mixtures. *Int. J. Hydrog. Energy*, 2004, **29**, 1527–1539.
5 **International Energy Outlook 2005.** Worldwide look at reserves and production. *Oil Gas J.*, 2005, **102**(47), 22–23.
6 **Daisho, Y., Yaeo, T., Koseki, T., Saito, T., Kihara, R.,** and **Quiros, E. N.** Combustion and exhaust emissions in a directinjection diesel engine dual-fuelled with natural gas. SAE paper no. 950465, 1995.
7 **Selim, M. Y. E.** Sensitivity of dual fuel engine combustion and knocking limits to gaseous fuel composition. *Energy Convers. Manage.*, 2004, **45**, 411–425.
8 **Selim, M. Y. E.** Thermal loading and temperature distribution of a pre-combustion chamber diesel enginerunning on gas oil/natural gas. ICE Division ASME International Spring Conference, Florida, April 1998.
9 **Ristovski, Z. D., Morawska, L., Hitchins, J., Thomas, S., Greenaway, C.,** and **Gilbert, D.** Particle emmision from CNG engines. *J. Aerosol. Sci.*, 2000, **31**(4), 403–413.
10 **Thomas, S.** and **Dawe, R. A.** Review of ways to transport natural gas energy from countries which do not need the gas for domestic use. *Energy*, 2003, **28**, 1461–1477.
11 **Ristovski, Z., Morawska, L., Yoko, G. A. A., Johnson, G., Gilbert, D.,** and **Greenaway, C.** Emissions from a vehicle fitted to operate on either petrol or compressed natural gas. *Sci. Total Environ.*, 2004, **323**, 179–194.
12 **Matsuoka, H., Kishishita, K., Nakashima, K., Sasaki, H.,** and **Kawamura, H.** Structure and performance of heat insulated natural gas engine, *JSAE Rev.*, 1997, **18**, 377–384.
13 **Hountalas, D. T.** The effect of operating parameters on the net and gross heat release rates of a direct injection diesel engine. Proceeding of the Second Biennial ASME–ESDA International Conference on *Design of energy systems*, New York, 1994, **3**, p. 63.
14 **Heywood, J. B.** *Internal combustion engine fundamentals*, 1988 (McGraw-Hill Book Co., New York).