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Full Paper

Modification and tuning of diesel bus engine for biogas electricity production

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Abstract: This study is to convert and tune a bus diesel engine for electricity production in a farm using biogas as fuel. The engine under study was a Hino K-13CTI 13,000cc 24 valve turbocharged engine coupled to a 3-phase 4-pole induction motor to produce electricity at 50 Hz. Modifications included an addition of biogas carburettor for air-fuel mixing, replacing the fuel injection system with spark ignition system, reduction of compression ratio from the original 16:1 to 8:1 using a cylinder head spacer, and modification of the turbocharger waste gate so the boost pressure can be adjusted. When the induction motor was synchronised to the power grid, the running speed of the engine was 1,500 rpm. Optimal engine efficiency was achieved at 28.6% by setting the lambda factor at 1.097, ignition timing at 54^{o} before top dead center, and the turbocharger boost at 56 kPa. With this setting, the generator power output was 134.20 kilowatt with emission of CO and NO_X being 1,154 and 896 ppm respectively.

Keywords: modification engine, engine for electricity, biogas engine, engine for biogas, gas engines

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Introduction

Biogas is a by-product of waste treatment in animal farms and can be used to replace fossil fuel to produce electrical energy. Biogas is formed by digestion of animal waste by anaerobic bacteria and the approximate composition is 60-80% methane, 20-40% carbon dioxide, and about 1% hydrogen sulfide and other trace gases. Biogas has a liquefying pressure of 200-300 bar and a heating value of about 23,400 kJ/m^3 [1,2]. The gas density is 1.2 kg/m^3 and has research octane number (RON) of about 130 [2,3]. From the above properties, it can be seen that it is difficult to liquefy biogas for storage or transport and it is quite suitable to be used as fuel in an internal combustion engine [4,5]. Statistics as of 2006 shows that in Thailand there are pig farms which have biogas waste treatment system that can produce biogas totalling 35,000,000 cubic metres per year. If all are used to produce electricity, the total energy of 35,000,000 kilowatt-hour can be produced per year [6]. With an increase in biogas production towards larger quantities, such technical utilisation as the transformation into mechanical energy becomes an issue to be researched on. While larger engines specifically designed for gas are on the market, smaller engines modified from standard Otto or diesel engines are seen to fill the gap for small-to-medium and decentralised applications. Indian [7,8], Chinese [9], and French [10] publications mainly dealt with the modification of small stationary diesel engines for dual fuel operation. Others went on to modify medium-sized diesel engines including their governors [11], or researched on the performance parameters of dual fuel biogas engines in more detail [12].

In Thailand, the bus is an expensive commodity compared to per capita income. As a result, buses are kept running long beyond its life expectancy set by the manufacturer. To keep these buses running, skilled mechanics as well as used engine markets and spare parts are well developed in Thailand. A used bus engine imported from Japan in a reasonably good condition can be had for about 5,500 US dollars. The cost for engine overhaul after about 1,000,000 kilometers of use is about 1,400 US dollars. With a used bus engine coupled to an electrical generator, a farm owner can make a 130 kilowatt power generator fueled by biogas for about 23,100 US dollars and the break-even of the investment can be as short as one year. These engines do not use a proper gas carburettor and are seldom tuned for best efficiency or low pollution [13]. This project is to investigate the proper adaptation of engine for durability and tune it for high efficiency and low emission.

Just about any engine can be converted to run on biogas and produce electricity in a farm. Smaller engines (under 5,000 cc capacity) are normally designed to perform best at higher engine speed and are normally not designed for full capacity output like those required in electrical power production [14]. On the other hand, larger engines like those found in heavy trucks or large buses are designed to carry a full load for most of its operating cycle and are also designed to run at lower engine speed. As a result, car engines will last about 150,000 kilometers between overhauls while buses and large trucks engines are expected to last more than 1,000,000 kilometers between overhauls. The long operating life of large engine makes it more suitable for electricity production in a farm and will probably cost less in the long run to operate [13].

Few variables need to be considered before the attempted adaptation of a diesel engine for electricity production. First consideration is given to the overall engine design itself. The engine will need to run at the generator synchronization speed to work as a generator. In this case, the generator requires the engine speed of 1,500 rpm for a four-pole generator and 3,000 rpm for a two-pole generator. The bus engine under consideration provides maximum torque at 1,500 rpm so it is suitable for the 1,500 rpm running speed. Another variable to be considered is the feasibility of reducing the compression ratio down to the spark ignition range. An overhead camshaft engine would be easy to adapt but the engine under consideration can also be adapted if the valve pushrod can be extended to accommodate the additional distance between the camshaft and the rocker arm, due to the thickness of the cylinder head spacer. The space issue when replacing the fuel injectors with spark plugs also needs to be considered before the adaptation.

This study aims to adapt a large diesel engine to run on biogas only and the adaptation involves an addition of biogas carburettor for air-fuel mixing, replacing the fuel injection system with spark ignition system, reducing the compression ratio to suit biogas fuel using a cylinder head spacer, and modification of the turbocharger waste gate so the boost pressure can be adjusted. The test rig used a Hino K-13CTI 13,000cc 24 valve turbocharged engine coupled to a 3-phase 4-pole induction motor to produce electricity at 50 Hz. The engine was then tuned by changing air/fuel ratio, ignition timing, and turbocharger boost pressure to obtain the optimal running condition.

Materials and Methods

The Hino K-13CTI engine being studied was a turbocharged bus diesel engine that came with a fuel injection system as well as a high compression ratio of 16:1 and a fixed waste gate boost control. For biogas fuel adaptation, a biogas carburettor was designed, manufactured, and installed. The fuel injection system was replaced with a spark ignition system, the compression ratio was reduced to 8:1, and the waste gate was modified so the boost pressure could be adjusted. Each of these modifications is discussed in turn in the following section of the paper.

Carburettor design

The biogas carburettor designed and installed in this study is shown in Figure 1. A literature survey shows that a suitable carburettor for a biogas engine should be a venturi with the accelerator cone being tapered as a curve of 40 mm radius and the diffuser cone angle of 10^{o}. The biogas is fed into the venturi through multiple circular ports around the throat area and the throat air velocity should be between 100 to 150 meters/second [2]. With this information, a carburettor designed for the 13,00cc engine operating at 1,500 rpm should have the throat diameter of 7.5 mm. The metering needle for the gas inlet was fabricated with a square root profile to provide some linearity between the needle position and the gas flow rate. The venturi was machined from aluminum stock and the carburettor body was fabricated from PVC pipe parts.

Spark ignition

The distributor and ignition coil was adapted from the one used in 6-cylinder Toyota 5-ME engine. The vacuum and centrifugal advance was disabled because the engine would run at a constant speed and a full load when used to drive the generator. The distributor was driven by the original fuel injection distributor mechanism. The fuel injection nozzle in the cylinder head was removed and replaced with a spark plug and an appropriate guide tube. The spark plug modification detail is shown in Figure 2.

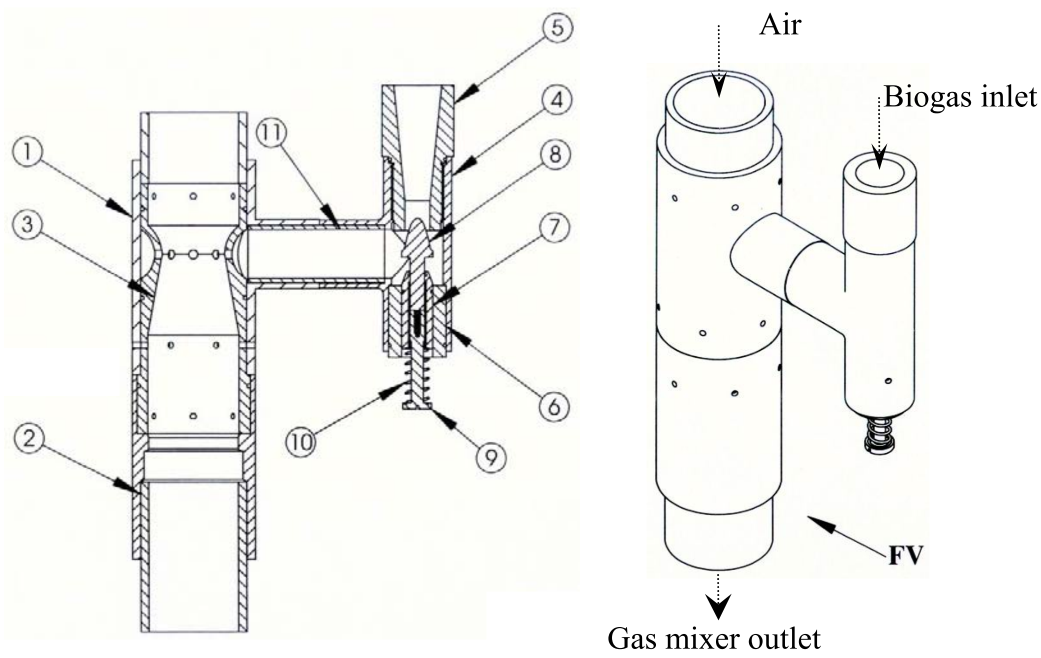

Figure 1. Biogas carburettor design: (1) Venturi housing, (2) Venturi base, (3) Venturi mixer, (4) Metering housing, (5) Main jet, (6) Metering adjusting nut spacer, (7) Metering adjusting nut, (8) Metering needle, (9) Metering adjusting screw, (10) Return spring, and (11) Pipe junction

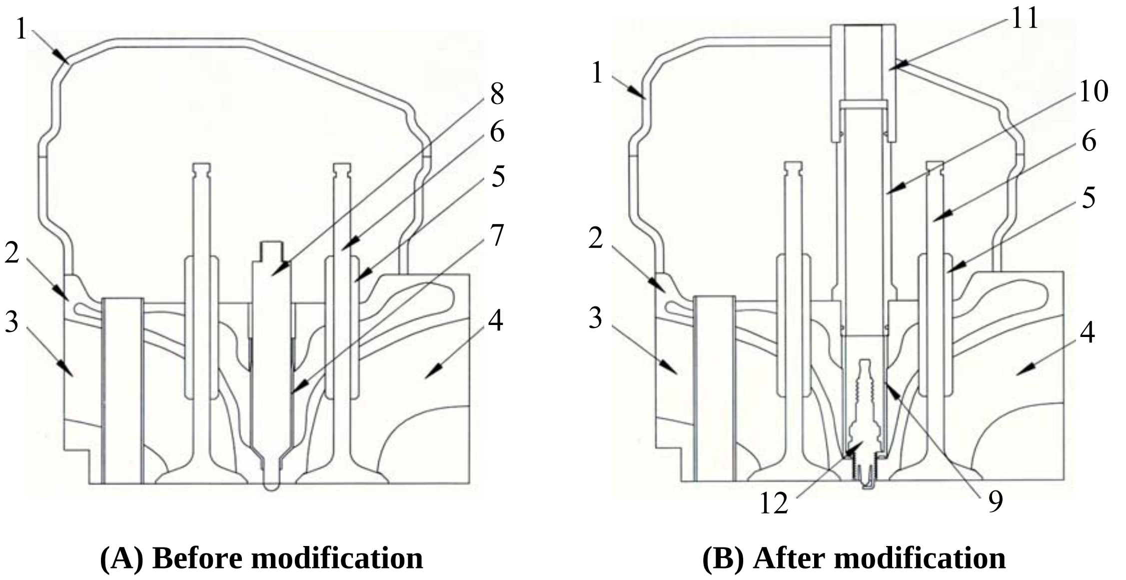

Figure 2. Cutaway view of the modified cylinder head: (1) Valve cover, (2) Cylinder head, (3) Intake port, (4) Exhaust port, (5) Valve guide, (6) Exhaust valve, (7) Diesel injection nozzle guide, (8) Diesel injection nozzle, (9) Spark plug guide, (10) Middle spark plug rod guide, (11) Upper spark plug rod guide, and (12) Spark plug

Compression ratio

For proper biogas operation, the compression ratio needs to be brought down from the original 16:1 to some lower value to avoid engine knock. Literature suggests that the compression ratio should be between 10:1 and 12:1 for biogas operation [2]. Since a turbocharger would be used, the compression ratio was further lowered to 8:1. To achieve this, a steel spacer 10 mm thick was fabricated for this purpose.

Turbocharger boost pressure

The turbocharger pressure is another parameter to be optimised. Too low pressure will cause loss of efficiency while too much boost will cause knocking, which can damage the engine in the long run. The boost pressure in the experimental engine was varied by modifying the waste gate spring housing so it could be adjusted to vary the turbocharger pressure. After the modification, the turbocharger boost could be adjusted between 40 kPa and 80 kPa. Details of the modification are in Figure 3.

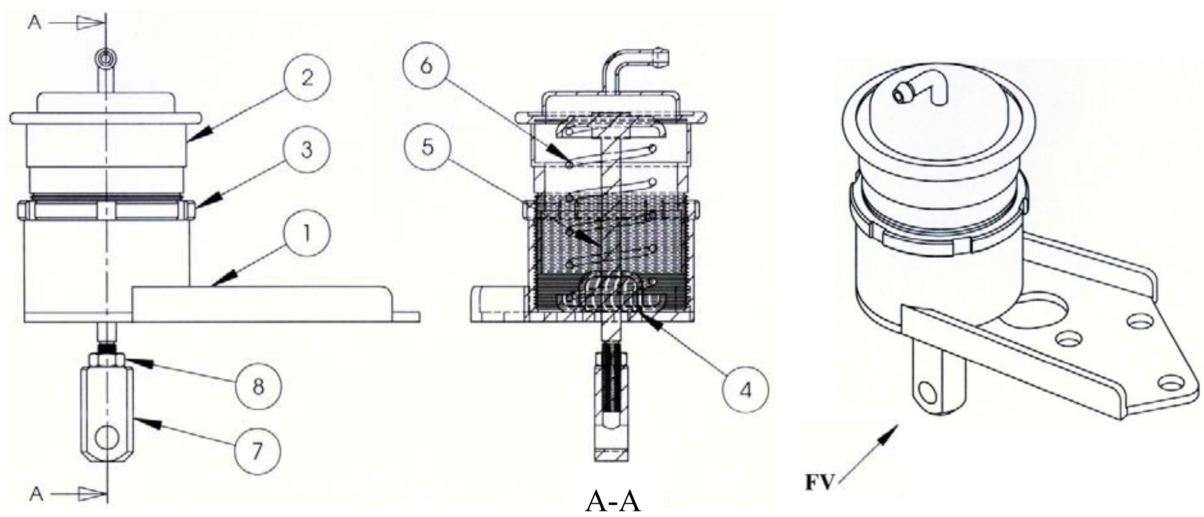

Figure 3. Modified waste gate from standard waste gate for adjustable turbocharger pressure: (1) Waste gate base, (2) Waste gate spring housing, (3) Waste gate adjustment lock nut, (4) Push rod bushing, (5) Waste gate push rod, (6) Return spring, (7) Adjustable push rod joint, and (8) Lock nut

Test Rig Description

The purpose of this experiment is to find an optimal tuning condition when a bus diesel engine is modified and converted to a spark ignition engine which will be fueled by biogas to drive an electrical generator to produce electricity. Inlet air and biogas were monitored so air/fuel ratio could be calculated. Exhaust gas was also checked for carbon monoxide and oxide of nitrogen emission at various engine settings. Each component of the test rig is briefly described in the following section. The overall block diagram of the test rig is shown in Figure 4.

Test engine

The test engine was a Hino 13,000cc diesel engine with 4 valves per cylinder, modified for biogas fuel as described above. The engine model was K-13CTI and was normally installed in buses in Thailand. This engine was chosen because of used engine availability and commonly available spare parts. It was overhauled to new engine specification before testing and is currently undergoing endurance test at 4T farm, Chiang Mai, Thailand.

Electrical generator unit

The generator used was a 3-phase 4-pole 132-kW generator with an induction motor made by HASCON, model number 200L2-2. The generator was directly coupled by a clutch mechanism to the test engine and was synchronised to the power grid when in operation. The nominal operating speed was 1,500 rpm.

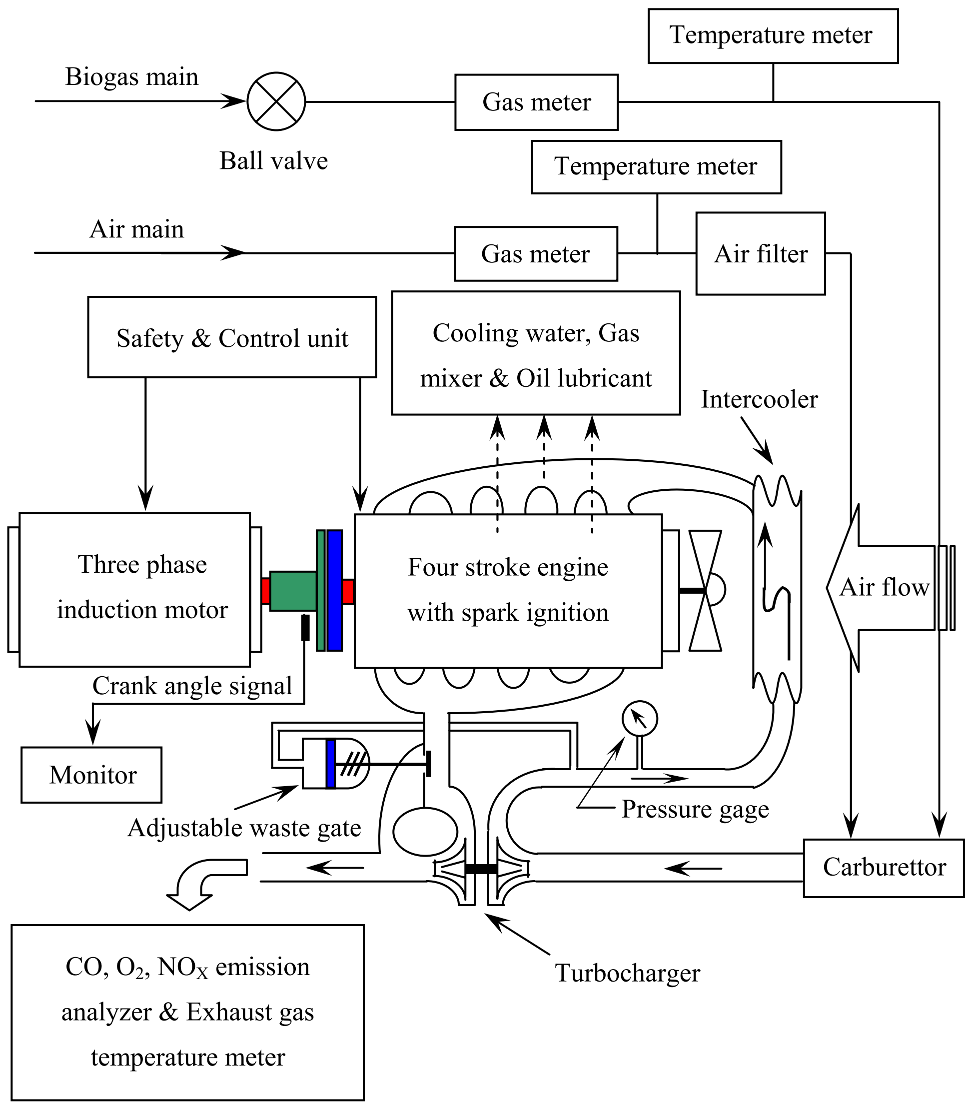

Figure 4. Block diagram of the test rig

Control unit

Biogas Advisory Unit, Chiang Mai University, designed the control unit and the blueprint was available to the public at no cost. It was used to start the test engine, synchronise the generator with the power grid, as well as monitor safety function in the case of gas supply shortage, short circuit, overloading in power grid, engine overheating, loss of oil pressure in the engine, or power grid failure.

Intake air measurement

Intake air flow rate was measured using a Kobold gas flow meter model GVPA-303-GDR.

Intake biogas measurement

The composition of inlet biogas varies with the type of animal waste used to produce the gas as well as the ambient temperature in which the fermentation occurs. In general, biogas contains carbon dioxide, methane and one percent of other trace gases. The methane content of biogas can be determined by finding the amount of carbon dioxide, adding 1 percent to the amount and subtracting this from 100, thus yielding the percentage by volume of methane. The amount of carbon dioxide in the biogas was determined by using the Brigon IND 60 gas analyser. The volume flow rate of biogas was measured using a Kobold gas flow meter model GVPA-303-GDR.

Exhaust gas measurement

A flue gas analyser (Testo 300XL-1) was used to monitor the quantity of O_2, CO, and NO_X in the exhaust gas. The probe attached could also measure the exhaust gas temperature as well.

Ignition timing measurement

Ignition timing was measured by a timing light (Sincro model DG86) with trigger signal from the high tension wire from the distributor to cylinder number one spark plug.

Power output measurement

The power output from the generator was monitored by Curcutor model AR5 supply network analyser.

Experimental Procedure

The experiment was carried out using the following steps to collect data for analysis.

Turbocharger pressure adjustment

The engine was run on biogas with RON of about 130. To get the most benefit from the higher RON, higher compression ratio was used for higher thermal efficiency [15-18]. The proper compression ratio for the spark ignition engine to be run on biogas fuel was between 10 to 12:1 [2].

While a naturally aspirated spark ignition engine may have sufficient margin of safety relative to knock to allow modest inlet-air boost, any substantial air compression prior to cylinder entry will require changes in engine design and/or operating variables to offset the negative impact on knock. The variables which were adjusted to control knock in a turbocharged SI engine were: compression ratio, spark retard from optimum, charge air temperature, and air/fuel equivalence ratio [18]. Therefore the cylinder head was installed with a spacer to increase the combustion chamber volume and produce lower compression ratio. The experiment would start at the compression ratio of 8:1 and the turbocharger pressure was adjusted by the waste gate at 40 kPa and increased at an increment of 4 kPa until engine efficiency began to decrease.

Initial air/fuel mixture adjustment

The engine was started, the ignition timing was set at about mid range (55^o BTDC) and the air/fuel mixture screw (metering adjustment screw in the carburetor) was adjusted to the position that the engine just ran smoothly (lean mixture).

Initial ignition timing adjustment

The ignition timing was set at 50^o BTDC at the beginning of a data collection.

Data collection

The recording of the following set of data was performed: air temperature, biogas temperature before and after boost by turbocharger and after cooling by intercooler, engine coolant temperature, lubricating oil temperature, exhaust gas temperature, air and biogas consumption rate, ignition timing, generator power output, oxygen remaining after combustion, carbon monoxide and oxide of nitrogen in the exhaust gas.

Ignition timing increment

The ignition timing was advanced 2^o and the data above were again collected. The process was repeated until the ignition timing reached 60^o BTDC or excessive pre-ignition was observed.

Air/fuel mixture increment

After a set of data was collected, the air/fuel mixture screw was turned half a revolution in the rich direction. Another set of data was collected and the process was repeated until the mixture was too rich for the engine to run smoothly.

Results

The collected data were processed into excess air ratio (λ), engine efficiency, and then the system efficiency (overall efficiency), power output, exhaust gas temperature, specific biogas consumption, oxygen quantity in exhaust gas, carbon monoxide and oxide of nitrogen emission were plotted against the ignition timing, excess air ratio, and turbocharger pressure. The resulting graphs are shown in Figures 5-7.

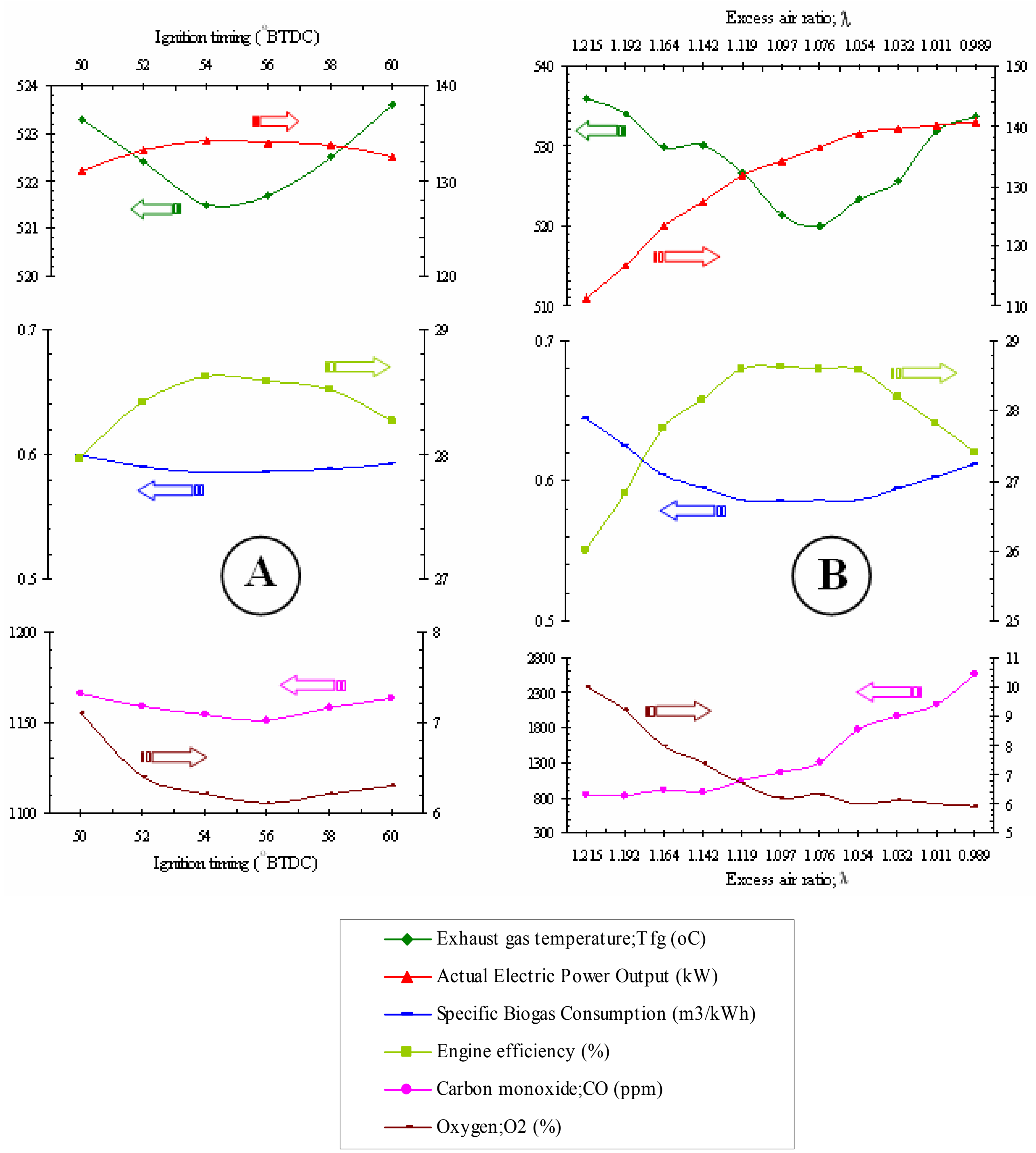

Figure 5. Exhaust gas temperature, electric power output, specific biogas consumption, engine efficiency, carbon monoxide, and oxygen for compression ratio of 8:1 and turbocharger pressure setting at 56 kPa plotted against: (A) Ignition timing, and (B) Excess air ratio.

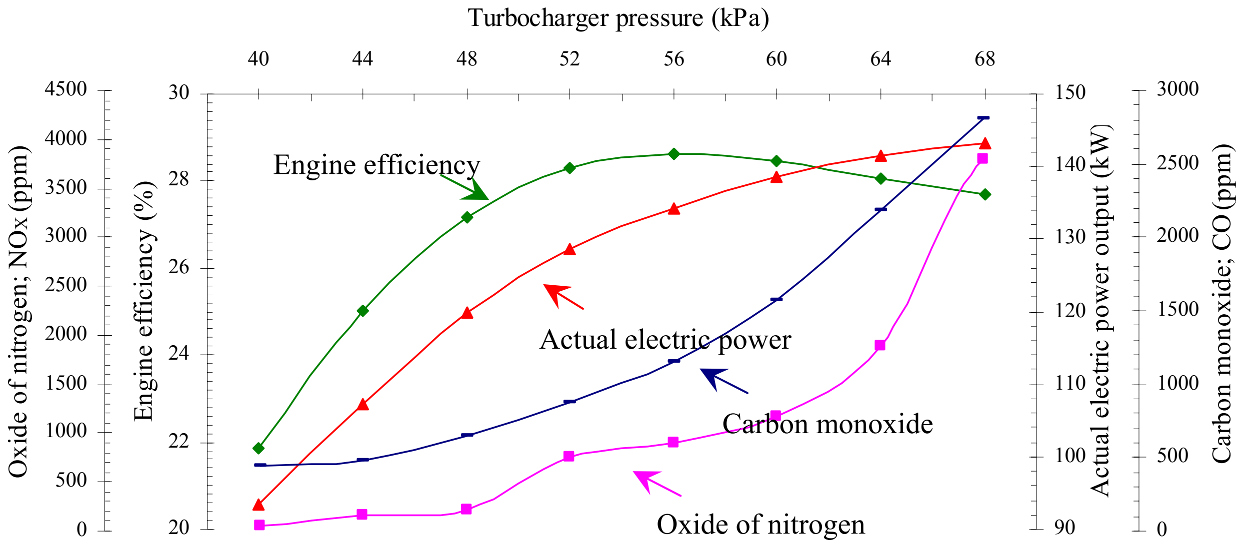

Figure 6. Output power, engine efficiency, carbon monoxide and oxide of nitrogen plotted against turbocharger boost pressure.

Discussion

Effect of Ignition Timing on Engine Performance

From Figure 5(A) it can be seen that the maximum engine efficiency and maximum output power can be achieved at 54^{o} BTDC ignition timing. When the timing is retarded from the optimal setting, the combustion process is completed after the bottom dead center position of the crankshaft and the thermal energy transferred to shaft power is reduced. The result is higher exhaust gas temperature as shown in the graph. If the ignition timing is advanced beyond the optimal point, knocking occurs, resulting in excessive combustion temperature and increase in NO_X and CO emission.

Effect of Air/Fuel Mixture on Engine Performance

Figure 5(B) shows that the optimal excess air ratio for this engine is 1.097. Leaner mixture results in lower flame speed and incomplete combustion, which leads to high exhaust gas temperature. Rich mixture can lead to engine knock and results in higher NO_X and CO emission as well as high exhaust temperature.

Effect of Turbocharger Boost Pressure on Engine Performance

Figures 6 and 7 show that the increase in turbocharger boost pressure can increase the engine output even though the engine efficiency has dropped off at higher boost pressure. Increasing the boost pressure beyond the maximum efficiency point results in high CO and NO_X emission as well as excessive engine vibration and it appears that the increased engine output is not worth the increased pollution and shortened engine life due to vibration.

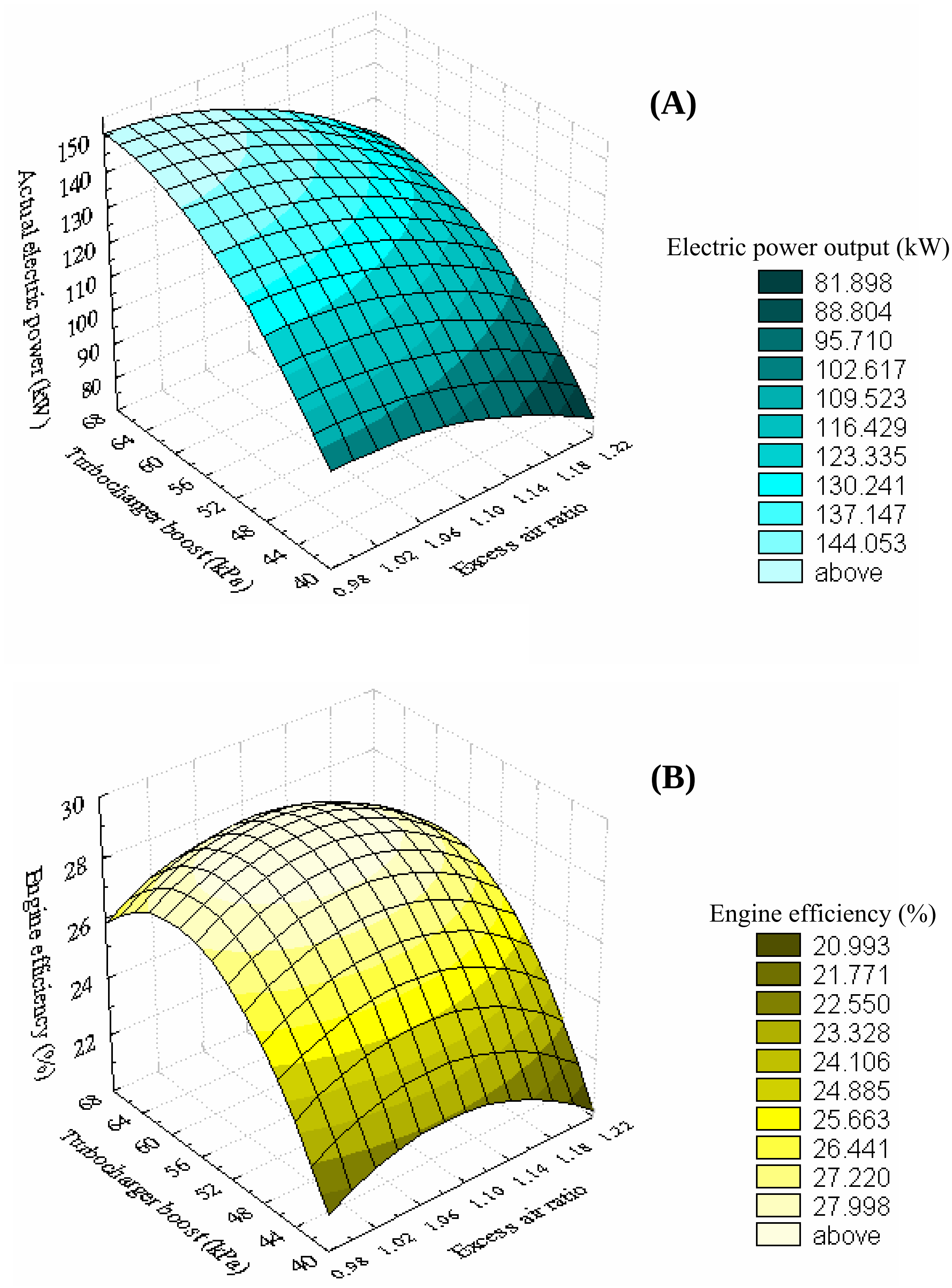

Figure 7. Power output (A) and engine efficiency (B) plotted against excess air ratio and turbocharger pressure at compression ratio of 8:1

Three categories of tuning optimisation were done. The categories are lowest CO emission, highest engine efficiency, and highest power output. The tuning parameters and collected data for the three categories are presented in Table 1.

With the presented data, it can be shown that a large diesel engine can be economically adapted for power generation in a farm. The payback period is less than one year and when properly tuned, the engine passes the internationally accepted pollution control standard of carbon monoxide of a vehicle engine [19, 20].

Table 1. Summarised data of fine tuning of biogas engine for optimum operation when compression ratio is 8:1 and turbocharger boost pressure is 56 kPa

Item	Lowest CO emission	Highest engine efficiency	Highest power output
1. Optimum ignition timing (^{o} BTDC)	57	54	52
2. Excess air ratio; λ	1.215	1.097	0.989
3. Fuel consumption; f_C (m^3/hr)	71.80	78.60	86.10
4. Specific fuel consumption; sfc (m^3/kWh)	0.64	0.59	0.61
5. Actual electric power output; P_{EL} (kW)	111.40	134.20	140.70
6. Engine efficiency; η_{eng} (%)	26.01	28.63	27.40
7. Overall efficiency; η_{tot} (%)	23.41	25.76	24.66
8. Oxide of nitrogen emission; NO_X (ppm)	511	896	843
9. Carbon monoxide emission; CO (ppm)	854	1,154	2,576
10. Oxygen in exhaust gas; O_2 (%)	10.00	6.20	5.90
11. Engine coolant temperature; T_{eng} (^{o}C)	88.2	88.7	89.0
12. Lubricating oil temperature; T_{oil} (^{o}C)	87.8	87.2	87.6
13. Exhaust gas temperature; T_{fg} (^{o}C)	536.0	521.5	533.8
14. Payback period (year)	0.78	0.52	0.49

Conclusion

In this study, engine performance and pollution figures are recorded for a range of engine tuning parameters, viz. turbocharger boost, ignition timing and excess air ratio. The compression ratio was set at 8:1 and the turbocharger boost was varied by adjusting the waste gate. When the system was operating at 1,500 rpm, the range of engine setting was as follows: excess air ratio between 0.9 to 1.2, ignition timing between 50^o to 60^o before top dead center, and turbocharger pressure setting between 40 to 68 kPa. Under these operating conditions, the engine efficiency increased as the boost was increased from 40 to 56 kPa and there was a slight increase of NO_X and CO as the boost went up. As the boost was increased from 56 kPa to 68 kPa, the engine efficiency began to decrease and the amount of pollution was increased. Increase in engine vibration was also noted in this turbocharger boost range. Oxide of nitrogen (NO_X) was high when high pressure and temperature occurred in the combustion. A test showed that more power can be generated if the engine is operated with rich excess air ratio and high turbocharger boost. Higher engine output will yield shorter payback period for the investment but increasing the boost pressure beyond 56 kPa will cause excessive pollution emission and engine vibration, which will probably shorten the engine life. It can be concluded that the boost pressure of 56 kPa yields the highest efficiency with acceptable pollution level for this engine.

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