A Computer Simulation of the Effect of the Inert Gas Volume Fraction in Low-Caloric Biogas on the Performance of an Engine

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Abstract—A computer simulation of a gas engine was performed to investigate the effects of the inert gas volume fraction in biogas on engine performance, specifically the engine torque and the brake-specific fuel consumption (BSFC) using GT-Power®. The engine speeds used in the simulation were 900 and 1800 rpm, while the simulated engine loads were 25, 50, 75 and 100%. The volume fraction of the inert gas N_2 in the biogas was varied from 20 to 80% with an interval of 10%. In a simulation of a naturally aspirated gas engine which is operated with an 80% volume fraction of N_2 in biogas, the optimal air-fuel ratio in terms of the fuel economy and brake power generation was 3.5. In a simulation of a turbo intercooler gas engine operated with an 80% volume fraction of N_2 in biogas, the optimal air-fuel ratios with regard to the fuel economy and brake power generation were 5.0 and 3.5, respectively.

Keyword - biogas, engine performance, inert gas volume fraction, computer simulation, air-fuel ratio

I. INTRODUCTION

Biogas can be obtained from either biomass or biogenic materials through an anaerobic digestion process [1]. In general, as biogas is a low-caloric gas and its reaction rate is slow comparing to high-caloric LPG and CNG, many problems can arise during the combustion process [2]-[3]. In spite of the shortcomings of biogas, it has garnered attention as it is carbon-neutral through its recycling of carbon dioxide, thus lowering the amounts of fossil fuels used [5]. Biogas is mainly used in medium and small-scale air-conditioning systems. Recently however, the applicability of biogas to small electric power generator engines has also been studied [6].

The combustible constituents in biogas are methane, hydrogen, and carbon monoxide, and the non-combustible constituents are nitrogen and carbon dioxide [2]. General biogas consists of 40~70% combustibles, 30~60% non-combustibles and a small amount of hydrogen sulfide.

One obstacle which prevents the wider application of biogas to gas engines is the instability of the combustion process due to the high volume fraction of the non-combustible constituents in biogas. This high volume fraction lowers the combustion gas temperature, which results in misfiring of the engine cylinders. Porpatham et al. [7] investigated the effects of the carbon dioxide concentration on the exhaust emissions from an engine. Huanga and Crookes [8] investigated the effects of both the carbon dioxide concentration and the engine compression ratio on the exhaust emissions and the BMEP (brake-mean-effective pressure). Lee et al. [5] investigated the effect of the inert N_2 concentration in biogas on engine exhaust emissions.

The previous work in this area [5, 7, 8] mainly investigated the effects of the inert gas concentration on the exhaust emissions from engines experimentally. The effects of the inert gas concentration on engine performance parameters such as the engine torque, brake power, BSFC and volumetric efficiency have not been studied sufficiently in computer simulations. In this study, the effects of the inert gas concentration on these engine performance parameters are studied. Also, the effects of the air-fuel ratio on the engine performance parameters are also investigated.

II. ENGINE PERFORMANCE SIMULATION

The engine used in the simulation is a six-cylinder engine with a displacement volume of 11.6 ℓ. Details of the engine specifications are summarized in Table 1. An engine part-load test is a fundamental test pertaining to engine development conducted by automakers. The engine part-load test can be performed by mounting the engine to be tested onto an engine dynamometer. After the tested engine is controlled to rotate at a specific speed in the dynamometer, the engine torque is increased at a specified interval by controlling the throttle valve position, as shown in Fig. 1. At a specified torque and engine speed, the fuel consumption, exhaust gas emissions and signals from several sensors are measured. The measured data sets at an engine speed of N with the torque increasing at the aforementioned interval are referred to as the part-load performance at engine speed N. Fig. 1 shows a typical part-load test map resulting from this method when it is used to measure the part-load performance.

Table 1. Engine specifications used in the simulation

Bore [mm]	119.0
Stroke [mm]	175.0
Connecting Rod Length [mm]	300.0
Piston Pin Offset [mm]	0.00
Displacement/Cylinder [liter]	1.946
Total Displacement [liter]	11.678
Number of Cylinders	6
Compression Ratio	13.00
Bore/Stroke	0.680
IVC [CA]	-118
EVO [CA]	100
IVO [CA]	314
EVC [CA]	400

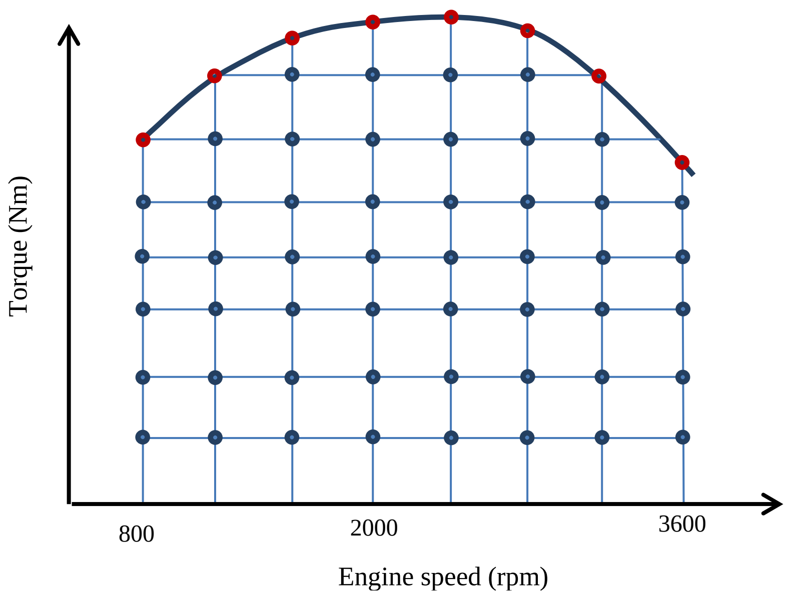

Fig. 1. A typical engine part-load measurement point

In this study, a computer simulation of several specified part-load test points is performed using GT-Power®. The simulation engine speed is either 900 or 1800 rpm, at which the generator can produce AC electric power of 60 Hz. The simulated part-load points at each engine speed are 25, 50, 75, and 100%. The simulated engine performance parameters are the engine torque, brake power, BSFC and volume metric efficiency. The simulated volume fraction of N_2 gas in biogas was varied from 20 to 80% with an interval of 10%. The simulated air-fuel ratios are 2.5, 3.5, 5.0, and 14.5. The performance levels of both a naturally aspirated engine and a turbo-intercooler engine are simulated based on the stated simulation conditions. The biogas constituent in the simulations is determined using combinations of methane and nitrogen. The combinations of methane and nitrogen in terms of the volume fraction percent are (80, 20), (70, 30), (60, 40), (50, 50), (40, 60), (30, 70), and (20, 80).

III. Results and discussions

In order to investigate the effects of the volume fraction of the inert gas N_2 in biogas on the engine performance, a computer simulation was performed.

Fig. 2 shows the simulation results of the brake torque when the N_2 volume fraction was increased from 20 to 80% at an interval of 10%. The simulated engine speed and air-fuel ratio were 1800 rpm and 14.5, respectively. Various simulated engine loads of 25, 50, 75 and 100% were assessed. The simulation result for a typical CNG gas mixture (methane 88%, ethane 6%, and propane 6%) was also compared to the results with biogas. For an N_2 volume fraction 20%, the brake torque curve with an increase in the engine load is nearly identical to that with the CNG gas mixture. As the N_2 volume fraction increases, the brake torque decreases. Moreover, it decreases sharply when the N_2 volume fraction exceeds 70%. An increase of the N_2 volume fraction in biogas reduces the combustible constituents in biogas, resulting in unstable combustion in the engine cylinder.

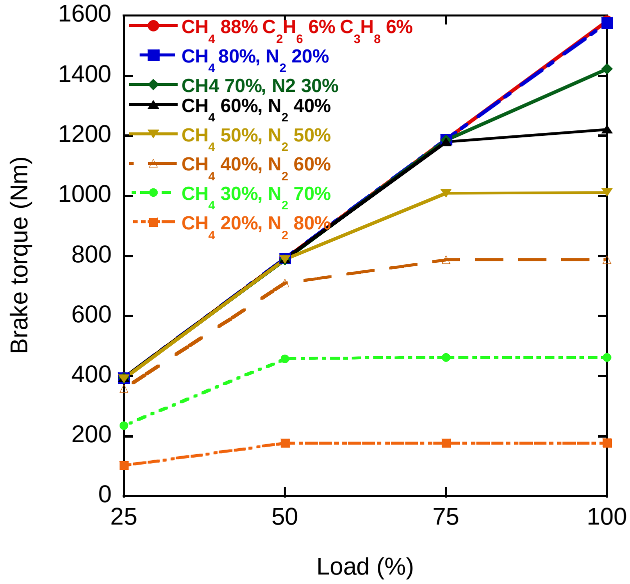

Fig. 2. Simulation results of the brake torque when the N_2 volume fraction increases from 20 to 80% with a 10% interval

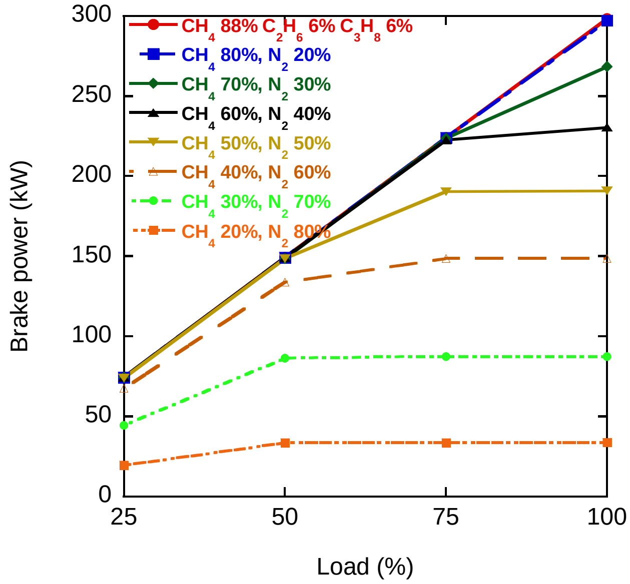

Fig. 3. Simulation results of the brake power when the N_2 volume fraction increases from 20 to 80% with a 10% interval

Fig. 3 shows the brake power simulated under the simulation conditions identical to those in Fig. 2. As the brake power is obtained by multiplying the brake torque by the engine speed, it shows the same trend as the brake torque.

Fig. 4 shows the simulated BSFC under the same simulation conditions used in Fig. 2. For an N_2 volume fraction of 20% in biogas, the BSFC is nearly identical to that of the CNG gas mixture. As the N_2 volume fraction is increased from 30 to 60%, the BSFC increases with a small gradient. For N_2 volume fractions of 70 and 80%, the BSFC increases sharply.

The volume fraction of inert gas in a typical form of biogas exceeds 60%. If biogas which includes inert gas at a rate exceeding 60% in terms of the volume fraction is used in a gas engine, the combustion stability will be degraded, resulting in an increase in the BSFC. In order to mitigate combustion instability, the air-fuel ratio is reduced from the current value of 14.5 to the range 2.5-5.0. When the N_2 volume fraction in the biogas is 80%, the air-fuel ratio for the combustible mixture reach a stoichiometric condition is 3.5. By reducing the air-fuel ratio from 14.5 until it is close to 3.5, the combustion stability can be enhanced and the power generated by the engine will increase.

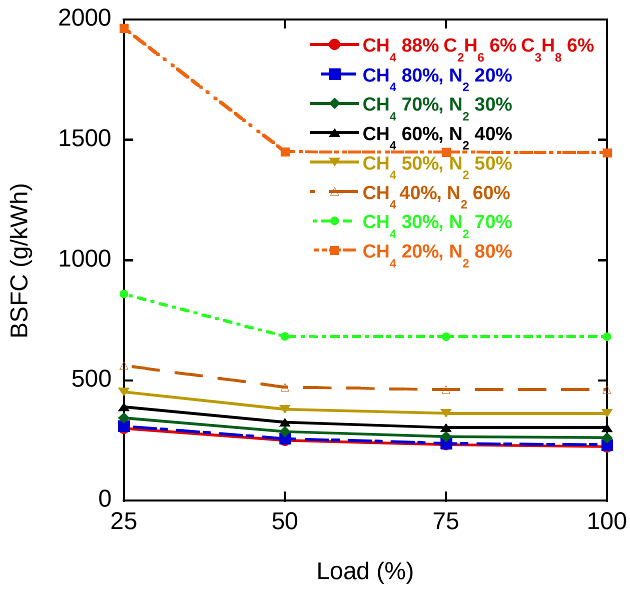

Fig. 4. Simulation results of the BSFC when the N_2 volume fraction increases from 20 to 80% with a 10% interval

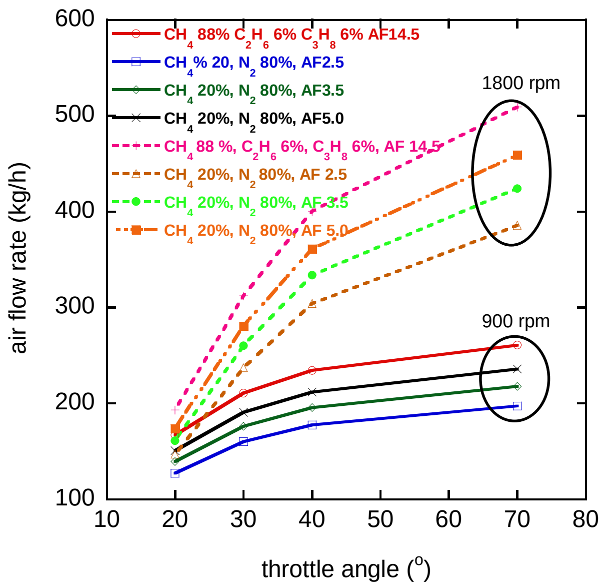

Fig. 5. NA engine simulation results of the air flow rate with an increase in the throttle angle at engine speeds of 900 and 1800 rpm and air-fuel ratios of 2.5, 3.5, and 5.0

The performance of a naturally aspirated (NA) engine is simulated with a volume fraction of the inert gas of 80% and engine speeds of 900 or 1800 rpm. The air-fuel ratios used in the simulation are 2.5, 3.5 and 5.0. Load control in the NA engine simulation is accomplished by changing the throttle angle. The lambda values corresponding to the air-fuel ratios of 2.5, 3.5, and 5.0 are 0.73, 1.0, and 1.46, respectively. Fig. 5 shows the simulation results of the intake air mass flow rate of the NA engine. The air mass flow rates increases with an increase in the throttle angle as the flow restriction through the throttle valve decreases. The mass flow rate at an engine speed of 1800 rpm increases more sharply than that at 900 rpm, as the flow velocity passing through the throttle valve at 900 rpm is lower.

Fig. 6 shows the simulated brake power with the same simulation conditions used in Fig. 5. The brake power simulation trends in Fig. 6 are similar to those of the intake air flow rate shown in Fig. 5. The brake power is determined by the brake torque if the engine speed is constant, while the brake torque is mainly dependent on the intake air mass flow rate in an ignition spark engine.

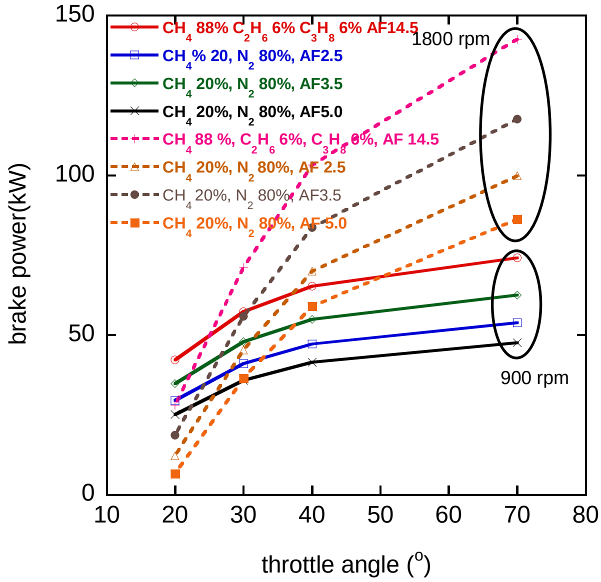

Fig. 6. NA engine simulation results of the brake power with an increase in the throttle angle at engine speeds of 900 and 1800 rpm and air-fuel ratios of 2.5, 3.5, and 5.0

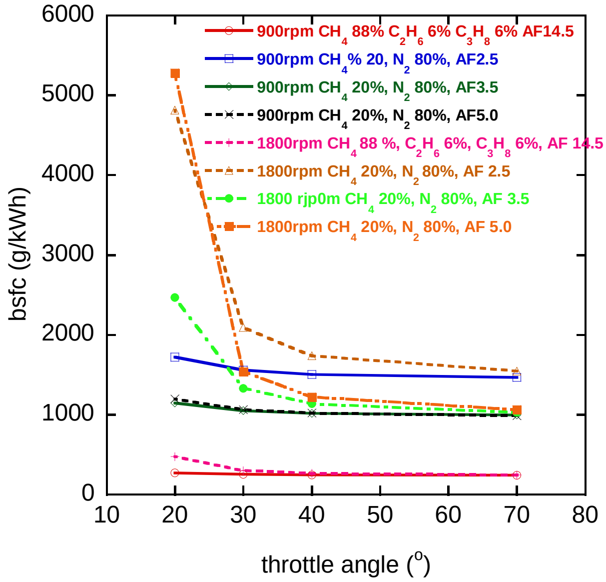

Fig. 7. NA engine simulation results of the BSFC with an increase in the throttle angle at engine speeds of 900 and 1800 rpm and air-fuel ratios of 2.5, 3.5, and 5.0

Fig. 7 shows the simulated bsfc with the same conditions used in Fig. 5. The bsfc is nearly constant with an increase in the throttle angle except at a speed of 1800 rpm. Throughout the range of throttle angles, the bsfc at 900 rpm is lower than that at 1800rpm. In comparison to the bsfc at 900 rpm, the bsfc at 1800 rpm increases severely below a throttle angle of 30° because the flow restriction at 1800 rpm increases as the throttle angle decreases. At both 900 and 1800 rpm, the bsfc is lowest when CNG gas is used in the simulation. With regard to the effect of the air-fuel ratio, the bsfc at an engine speed 900 rpm shows lower values in the order of 3.5, 5.0, and 2.5. The bsfc at the engine speed 1800 rpm shows lower values in the order of 3.5, 5.0, and 2.5.

The performance of a turbo intercooler (TI) engine is simulated with a volume fraction of inert gas of 80% and an engine speed of 1800 rpm. The air-fuel ratios used in this simulation are 2.5, 3.5 and 5.0. Load control in the TI engine simulation is done by adjusting the fuel flow rate. Fig. 8 shows the simulated brake power of the TI engine. The simulated brake power with an air-fuel ratio of 3.5 is nearly identical to that of 5.0. Also, the brake power range at the air fuel ratio of 3.5 is twice as wide as that at 5.0. At an air-fuel ratio of 2.5, the brake power shows a very low value compared to those at the air-fuel ratios of 3.5 and 5.0 if the brake power levels are compared at the same fuel flow rate. Fig. 9 shows the simulated bsfc of the TI engine with the same simulation conditions used in Fig. 2. The bsfc at the same fuel flow rate shows lower values in the order of 5.0, 3.5, and 2.5. The BSFC at an air-fuel ratio of 2.5 is nearly doubled compared to the bsfc values at air-fuel fuel ratios of 3.5 and 5.0.

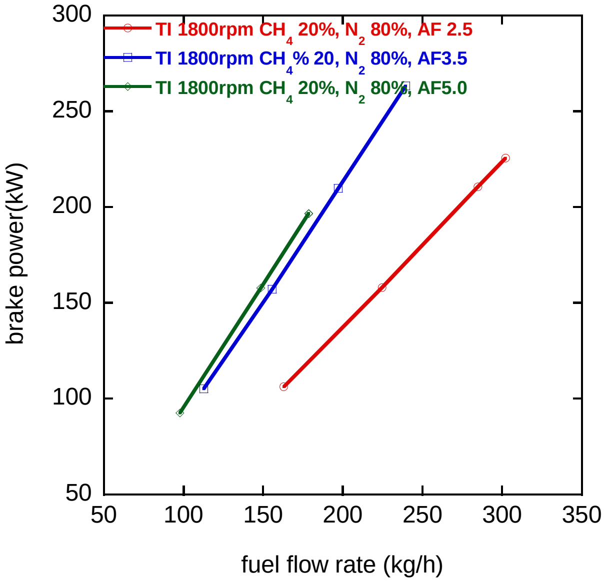

Fig. 8. TI engine simulation results of the BSFC with an increase in the throttle angle at an engine speed of 1800 rpm and air-fuel ratios of 2.5, 3.5, and 5.0

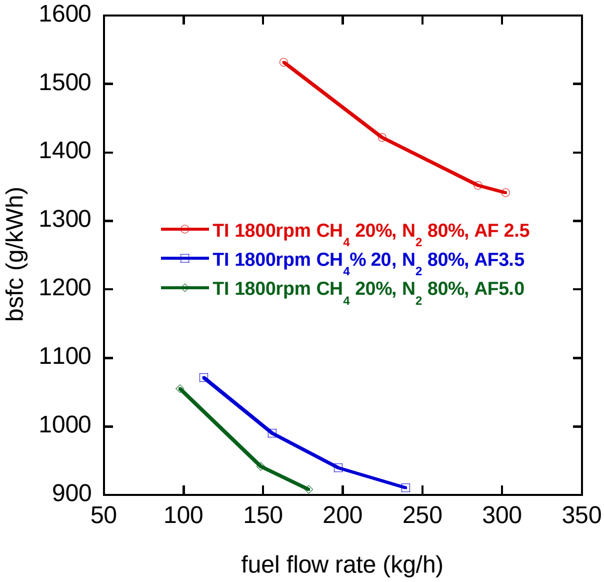

Fig. 9. TI engine simulation results of the brake power with an increase in the throttle angle at an engine speed of 1800 rpm and air-fuel ratios of 2.5, 3.5, and 5.0

IV. Conclusion

The effects of the inert gas concentration in biogas on engine performance levels are investigated in a computer simulation. The performance levels of a NA engine and a TI engine are simulated with a volume fraction of inert gas of 80% and engine speeds of 900 or 1800 rpm, at which a typical generator can produce 60Hz of electric power. The bsfc values of the NA engine at engine speeds of 900 and 1800 rpm are lower in the order of 3.5, 5.0, and 2.5. The bsfc values of the TI engine at an engine speed of 1800 rpm are lower in the order of 5.0, 3.5, and 2.5. The optimal air-fuel ratio for the NA engine is 3.5, at which the lambda value is 1.0 considering the brake power and the BSFC. In the simulation of the turbo intercooler gas engine, the optimal air-fuel ratios considering the fuel economy and the brake power were 5.0 and 3.5.

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