CONTROL STRATEGIES FOR VARIETIES IN GASEOUS PETROL COMPOSITION IN AIR–FUEL CONTROLLED PETROLEUM GAS MOTORS

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ABSTRACT:
In the current examination, the countermeasures are proposed to limit the issue on force and power yield execution and fumes gas emanations of gaseous petrol motor with utilization of lower calorific gas. An examination was led to recognize warm productivity and destructive fumes gas outflow qualities under fractional burden conditions to improve productive fuel use in motors influenced by the presentation of low calorific gases. A countermeasure for adapting to emanation gas guidelines also, forestalling warm productivity decay under evaluated power working conditions was at that point introduced. A 11 L six-chamber super charged motor for city transports agreeable with the EURO 6 guideline was utilized in the examination, and the outcomes got utilizing the reference petroleum gas fuel were contrasted and those acquired utilizing recreated low calorific gases. Unadulterated methane (CH4) was additionally used to research the impacts of gas sythes is changes on warm productivity and fumes gas outflows.

At the point when N2 is added or unadulterated CH4 is utilized under incomplete burden working conditions, the burning rate diminishes; therefore, the ideal start timing is also exceptional comparative with that acquired when the reference gaseous petrol fuel is utilized. On the off chance that the N2 blending proportion is expanded to a base of 4.7% under appraised power working conditions, ignition gets unsteady. Stable activity can be gotten by expanding the set base fuel sum, 2.35% and 9.41% for unadulterated CH4 and 8% N2, individually; notwithstanding, force diminishes with respect to the ignition speed of the gas fuel. The methodology of lift pressure control for the force pay can limit the lessening in warm proficiency.

Keywords: Natural gas, Fuel composition, Internal combustion engine, Ignition timing, Countermeasures, Torque compensation

1. Introduction

Natural gas exhibits more advantages than petroleum in terms of reserves, distribution of production areas, and price, and hence, is easier to supply in a stable manner. Natural gas is also a very advantageous automobile fuel with regard to supply. Because natural gas has a high octane value of approximately 130, it can be used without knocking at a high compression ratio (Kubesh et al., 1992). Furthermore, owing to the high thermal efficiency and power output, as well as broad flammability limit of natural gas, lean combustion can be realized, and thus, low fuel consumption and nitrogen oxide production (NOx) can be achieved (Li et al., 2019). Consequently, the emission of particulate matter is very low. As gas combustion devices, natural gas engines are used in compressed natural gas (CNG) vehicles for transportation, cogeneration systems, and gas heat pumps. Their use is increasing annually owing to energy diversification policies and distributed power generation (Turrio-Baldassarri et al., 2006).

As the availability of high-caloric natural gas is decreasing worldwide and that of low-caloric natural gas, such as pipeline natural gas (PNG), is increasing, the domestic natural gas supply in Korea is expected to become low in calories; therefore, it is necessary to develop countermeasures for efficient use as well as minimize the limitations of domestic industrial gas combustion devices that use natural gas as fuel. In particular, for natural gas engines in vehicles, the use of low-calorie gas may adversely affect engine performance and production of exhaust gases because the operation control variable of the engine is determined based on the existing calorific value. In fact, based on the results of previous studies on variations in calorific value and applicability, the emission of carbon monoxide (CO), carbon dioxide (CO2), and NOx emissions decrease with the calorific value, and the power output fluctuates under constant ignition timing and air– fuel ratio conditions (Ha et al., 2011; Sakai and Kuroda, 1996; Kuroda and Sakai, 1996; Kuroda et al., 1997). In addition, with an increasing methane number, torque output decreases and the emission concentration of hydrocarbons increases (Feist et al., 2010).

Table 1
Compositions and properties of test gases.

	Natural gas (100% NG)	Low calorific gas I (4% N_2)	Low calorific gas II (8% N_2)	Pure methane (CH_4)
CH_4	92.93	89.29	85.64	100
C_2H_6	5.34	5.13	4.92	0
C_3H_8	1.07	1.03	0.98	0
i-C_4H_{10}	0.22	0.21	0.20	0
n-C_4H_{10}	0.28	0.27	0.26	0
i-C_5H_{12}	0.01	0.01	0.01	0
n-C_5H_{12}	0	0	0	0
N_2	0.15	4.06	7.98	0
Specific gravity (–)	0.599	0.613	0.628	0.557
Higher heating value (MJ/kg)	49.49	46.43	43.50	50.05
Wobbe index (MJ/kg)	63.94	59.30	54.90	67.06

Since the mid-2000s, the higher heating values of liquid natural gases (LNGs) introduced in Korea have considerably been lowered. A study was conducted to address the issues regarding the reduction of the calorific value of natural gas in the mid- to long-term; to this end, shale LNG and PNG were introduced. CNG engine experiments were conducted at the stoichiometric equivalence ratio in a small spark ignition engine because of the huge experimental gas manufacturing cost problem. The experiment employed a 2500cc engine. The experimental results demonstrated that the power output decreased to less than 3% as the higher heating value was varied to 41 MJ/Nm3 (9800 kcal/Nm3, the minimum higher heating value according to the supply regulation at the time). No problems regarding thermal efficiency and exhaust gas were observed; further, the thermal efficiency slightly increased as the caloric value of the gas decreased (Kim et al., 2007).

Meanwhile, research has been conducted for the development of an improved engine considering variable factors that affect various operations, such that the natural gas engines can be operated in an optimal state. Mendis et al. (1993) performed numerical calculations and experiments on natural gas engines in parallel and performed overall mapping work by closely experimenting with variations in power output corresponding to variations in ignition timing and air–fuel ratio. Matthews et al. (1996a,b) investigated the composition of the natural gas supplied to several cities, such as Houston, through pipelines from natural gas production centers in Texas, USA. Based on this survey, they analyzed the effects of the composition of natural gas fuel on exhaust gas, fuel economy, and drivability, using gas fuels with the highest and lowest Wobbe index (WI) values, natural gas used for North American certification, and pure methane (CH4). Variations in exhaust gas and fuel economy were investigated, and for hydrocarbon exhaust gas, it was shown that the smaller is the ratio of CH4, the more were the non-methane hydrocarbons generated. Poulsen and Wallace (1994) conducted a study on engine variables that affect hydrocarbon-based exhaust components in natural gas engines. Compared with the change in the composition of the fuel, it was found that the change in the amount of exhaust emission caused by the change in speed, air–fuel ratio, and ignition timing during engine operation was very small except for engines operated with a richer mixture than the stoichiometric.

Previous research results (Park et al., 2015; Gupta and Mittal, 2019) confirmed that in a natural gas engine, in which the air– fuel ratio was appropriately controlled by varying the fuel supply amount based on the current natural gas calorific value variations in the calorific value of the gas, can affect the engine torque output and efficiency. However, CNG engines using stoichiometric combustion are not representative of large lean-burn engines used for natural gas buses and cogeneration. Therefore, complementary experiments were required, as it was suggested that a trend different from that obtained for the nitrogen (N2) dilution method, previously conducted by this research institute, could be observed (Park et al., 2015, 2019; Lee et al., 2021; Park et al., 2021). Therefore, in this study, by considering the characteristics of these CNG engines, a large turbo lean-burn engine was selected to minimize the problems of natural gas engines in general and prepare for the expected variations in the composition of natural gas as the calorific value gradually decreases. This study aimed to suggest countermeasures to reduce harmful exhaust gas emissions from engines including control variables and to prevent reduction in torque and power output performance and efficiency.

2. Natural gas composition

The gas fuel used in this study was selected based on its calorific value. The composition of natural gas varies depending on the production area, and thus, its WI value also varies; further, WI is used as important variables for evaluating the characteristics of natural gases, such that they can be categorized, and for determining their interchangeability. Table 1 summarizes the composition and physical properties of the test natural gas fuel used in this study. To examine the effects of low-calorie gas based on the natural gas (100% NG) currently supplied by KOGAS in Korea, low-calorie gas (4% N2, 8% N2) was mixed with standard natural gas fuel and simulated. When a low calorific value was realized by mixing N2, hydrocarbon-based components, such as CH4, ethane, and propane, which are the main constituents of natural gas apart from N2, were constant. In such cases, to immediately examine the effects of lowering the calorific value owing to variation in natural gas components, pure CH4 was used as a fuel for comparison of gas fuel composition to low-calorie gas by mixing N2.

WI is mainly used to determine the interchangeability of general gas combustion devices, such as gas burners and heat treatment furnaces. WI is an index representing the flow of energy through an orifice or valve for a given pressure drop. When WI is large, a large amount of heat is being released from the orifice at a given time.

3. Experimental procedure

3.1. Experimental setup

In this study, an 11 L six-cylinder supercharged engine used in city buses with a lean combustion in compliance with EURO 6 regulations and test equipment was installed for identifying any problems that low-calorie gas causes for gas engines; countermeasure technologies were then developed. The specifications of the engine are listed in Table 2.

Table 2
Specifications of test engine.

Type	Description
Number of cylinders	6
Bore (mm)	123
Stroke (mm)	155
Displacement volume (cc)	11,051
Compression ratio	10.5
Max. Power	222 kW / 2,100 rpm
Max. Torque	1,150 Nm / 1,300 rpm

Fig. 1 shows a schematic of the experimental equipment employed, entailing the engine, control system, fuel supply system, and analysis equipment used in this test. To control engine speed and load, an EC dynamometer (WT470, HORIBA ATS) was connected and used. The ignition timing and fuel injection amount were controlled using a universal engine control unit (ECU), which can change the engine combustion control variables, and the equivalence ratio and closed-loop control status were monitored. Combustion pressure data were measured using a spark-plug type pressure sensor (6118BFD35, Kistler) and combustion analyzer (DEWE800, DEWETRON Co.). In addition, an exhaust analyzer (AMA i60, AVL) was used to measure CO, CO2, total hydrocarbons (THC), and NOx. The measurement equipment and their accuracy and uncertainty are summarized in Table 3.

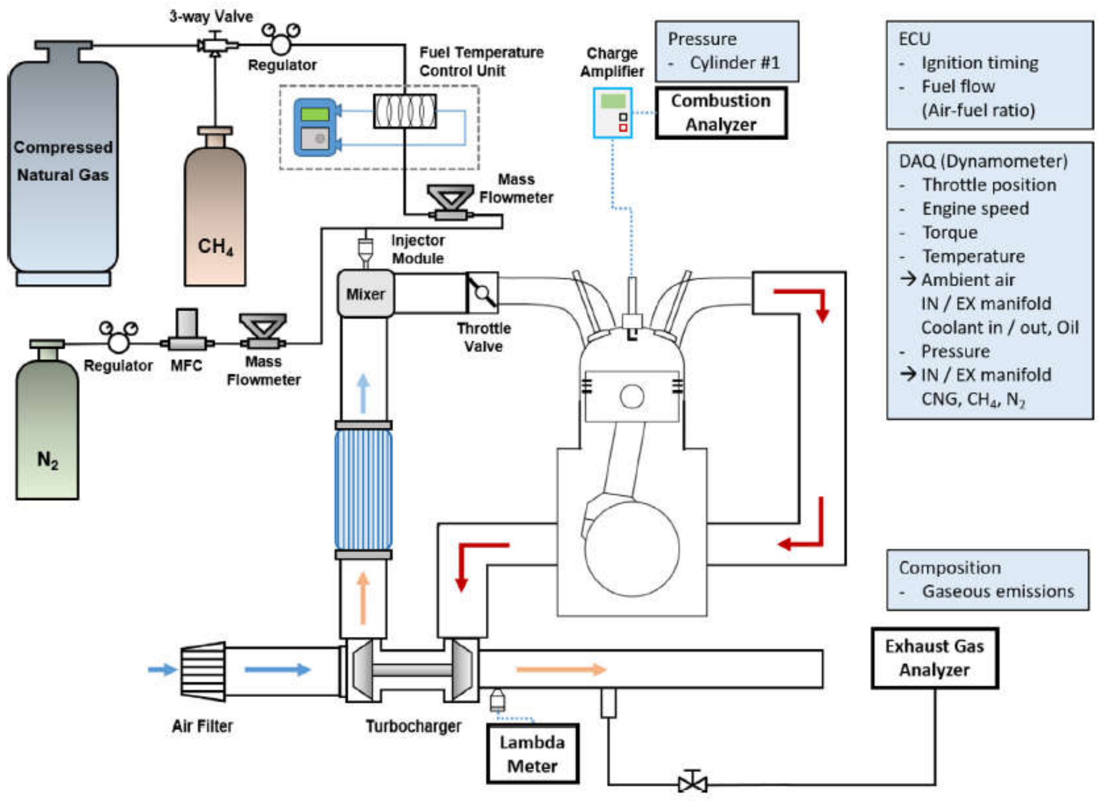

Fig. 1. Schematic of experimental setup.

The test engine originally equipped with an oxidation catalyst and selective catalytic reduction system. However, all the exhaust gas results of the present study are engine-out emissions, measured from the upstream of after treatment system.

To ensure a stable supply of natural gas and a constant calorific value, the pressure of the fuel was reduced to 0.8 MPa in the regulator of a high-pressure fuel container filled with a standard natural gas fuel with a pressure of approximately 20 MPa; the former fuel was then supplied to the fuel metering valve. To supply and control high-pressure N2 and pure CH4, two high-pressure N2 gas containers and 10 high-pressure CH4 gas containers were first connected in parallel; then, for N2 addition. When pure CH4 was used, the standard natural gas fuel, 100% NG, could be replaced by switching the three-way valve.

To prevent the cooling of the measuring devices owing to the expansion of the decompressed gas fuel in the regulator, a closed recirculating heater (RW-3025G, JEIO TECH) and heat exchanger were installed. This helped realize a continuously supply of hot water at 40 ◦C, which helped maintain the high temperature of the measuring devices. The fuel flow rate was controlled using a fuel metering valve comprising six injectors in the existing engine, and the flow rate was measured using a Coriolis-type mass flowmeter (CFM025M, Micro Motion). The experiment was conducted by varying the nitrogen flow rate for each operating condition using a mass flow controller (M3400V, LINE TECH), such that the flow rate of the supplied fuel could be measured in real time and the calorific value of the gas fuel could be controlled.

Table 3
Specifications of all measurement equipment.

Parameter and analyzer	Measuring range	Accuracy	Uncertainty
Load cell for torque	0–3000 Nm	±0.2% of full scale	±4 Nm (about 95% confidence level, k = 2)
High-performance pressure transducer for boost air pressure	0–0.3 MPa (absolute)	±0.25% of full scale	±0.0004 MPa (about 95% confidence level, k = 2)
Wide band oxygen sensor for air–fuel ratio	0.125–1.25	±4.2% of measured value	±0.024 (about 95% confidence level, k = 2)
FID for THC and CH_4	0–10,000 ppm	±0.5% of full scale	±29 (about 95% confidence level, k = 2)
NDIR for CO	0–10,000 ppm	±1% of full scale	±58 (about 95% confidence level, k = 2)
NDIR for CO_2	0%–25%	±1% of full scale	±0.29 (about 95% confidence level, k = 2)
CLD for NO_x	0–10,000 ppm	±1% of full scale	±58 (about 95% confidence level, k = 2)
Mass flowmeter for CNG flowrate	0–240 kg/h	±0.1% of measured value	±0.02 kg/h (±95% confidence level, k = 2)
High-performance pressure transducer for in-cylinder pressure	0-20 MPa	±0.5% of full scale	±0.06 MPa (±95% confidence level, k = 2)
Thermocouple for exhaust gas temperature	−200–1250 °C	±0.75% of full scale	±2.2 °C (±95% confidence level, k = 2)

3.2. Experimental conditions

To examine the thermal efficiency and exhaust gas characteristics when using low-calorie gas fuel and to derive measures for optimizing performance and reducing harmful exhaust gas emissions, the experiment was conducted at a torque of 550 Nm and an engine speed of 1400 rpm, which are typical partial load operation conditions. The dynamometer was controlled at a speed/throttle mode. The ignition timing was varied for each gas fuel to identify and optimize the thermal efficiency and exhaust gas characteristics. To confirm the effect of improving the rated power output based on the fuel quantity control variable, an experiment was conducted under the rated performance conditions, which are an engine speed of 2100 rpm and full load conditions, in which the throttle was fully opened. Wastegate turbocharger manipulated by the three-way control valve with compressed air controls the intake manifold pressure, correspond to power output at a certain equivalence ratio condition. For each operating condition, the engine was operated by maintaining the air–fuel ratio of the mixture and the boost pressure was set in the engine, and the experiment was conducted with the engine coolant heated sufficiently.

Table 4
Combustion related parameters.

	Natural gas (100% NG) [CAD, ATDC]	Low calorific gas I (4% N_2) [CAD, ATDC]	Low calorific gas II (8% N_2) [CAD, ATDC]	Pure methane (CH_4) [CAD, ATDC]
CA10 (burned mass fraction 10% timing)	−3.55	−3.95	−4.78	−5.64
CA50 (burned mass fraction 50% timing)	8.55	8.24	7.73	8.11
CA90 (burned mass fraction 90% timing)	40.99	41.61	42.19	42.42
Main burn duration (CA10–CA90)	44.54	45.56	46.97	48.06

4. Results

4.1. Optimization of part load performance

To cope with the low calorific value of natural gas, engine problems due to the introduction of low calorific gas should be understood. Measures for efficient use must then be developed and the impact of the problem on the engine performance must be minimized. Fig. 2 shows the manifold air pressure (MAP) and thermal efficiency corresponding to the variations in the ignition timing under torque conditions of 1400 rpm and 550 Nm, which are the partial load operation conditions. First, the MAPs corresponding to the variations in the calorific value of the fuel due to addition of N2 and variations in fuel composition due to the use of pure CH4 and the standard natural gas fuel (100% NG) were obtained. The results demonstrate that 100% NG exhibited the lowest MAP. The lower was the heating value, the higher was the MAP. For pure CH4, the calorific value was higher than that of the simulated low-calorie gas mixed with 8% N2; however, pure CH4 exhibited the highest MAP trend, which can be considered to be a consequence of thermal efficiency deterioration.

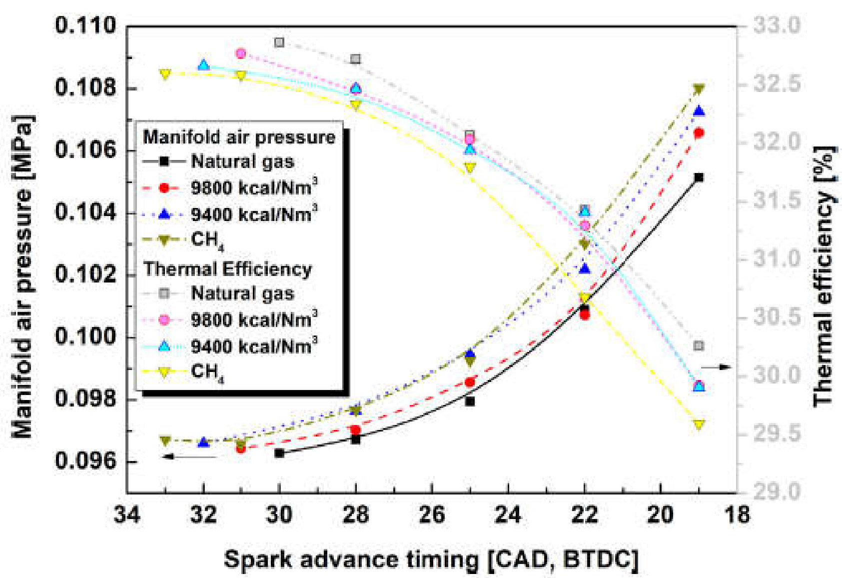

Fig. 2. Manifold air pressure and thermal efficiency variations for each gas composition with ignition timing changes under part load operating conditions.

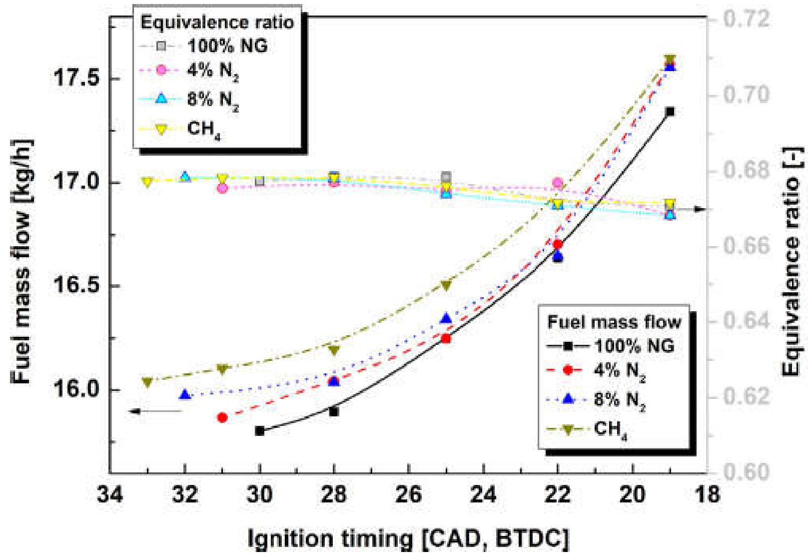

Fig. 3. Fuel mass flow and equivalence ratio variations for each gas composition with ignition timing changes under part load operating conditions.

Control variables that affect the combustion performance of an engine include ignition timing, equivalence ratio, intake, and exhaust manifold pressure. These variables are set based on 100% NG, which is the reference gas fuel on the control variable map of the ECU, which controls the engine. Even if lean combustion is conducted using standard natural gas, the ignition timing is generally retarded compared with optimum ignition timing, which shows maximum efficiency to satisfy the NOx limit of the enforced exhaust gas regulation. Owing to the variations in thermal efficiency corresponding to the variations in the ignition timing, the efficiency increases as the ignition timing advances from before top dead center (BTDC) 19 in the crank angle degree (CAD), which is the ignition timing set in the ECU based on the reference gas fuel. As a result, the MAP decreases because the amount of air and fuel decreases to satisfy the 550 Nm torque set as the driving condition.

For 100% NG, which is the reference gas fuel, the ignition timing that represents the maximum efficiency corresponds to BTDC 30 CAD, which should be advanced by 11 CAD compared with the ignition timing set in the ECU. In general, the combustion rate decreases when the mixture of air and fuel is diluted by adding N2. Even under the operating conditions that satisfy the torque of 550 Nm for the simulated low calorific gases, the higher is the N2 mixing ratio, the lower are the combustion rate and thermal efficiency. The optimum ignition timing, which represents the maximum thermal efficiency, is BTDC 31 CAD and BTDC 32 CAD for each simulated low calorific gas with 4% N2 and 8% N2, respectively. These results demonstrate that the optimum ignition timings should be advanced by 1 CAD and 2 CAD, respectively, compared with that of the reference gas fuel. Because the volumetric ratio of air to natural gas is high, the dilution effect of mixing 4% N2 and 8% N2 in the fuel does not appear large, and thus, the difference in optimal ignition timing is not large.

For pure CH4, the optimum ignition timing was the most advanced timing as BTDC 33 CAD, and the combustion rate was the slowest of all four fuel gases used in this study. As shown in Table 4, CA50s, burned mass fraction 50% timing, of combustion analysis data for each gas have similar values of crank angle degrees when those are compared for optimum ignition timing. However, the main burn duration, defined as the duration from CA10 (burned mass fraction 10% timing) to CA90 (burned mass fraction 90% timing) shows the maximum for pure CH4, consistent with the result of optimum ignition timing, due to the slowest combustion rate. The thermal efficiency was the lowest owing to the low combustion speed, and the MAP was highest because the mixer the required air–fuel mixture to satisfy the torque performance of 550 Nm corresponding to the operating condition under a constant ignition timing. Even if the ignition timings were advanced to represent a maximum efficiency performance considering the decrease in combustion rate, the maximum thermal efficiency was the highest for the reference gas fuel and the lowest for pure CH4, which has the slowest combustion rate. As shown in Fig. 3, which shows variations in fuel mass flow rate and measured equivalence ratios corresponding to the variations in ignition time, the operating conditions were constant even though the ignition time varied; thus, a constant equivalence ratio was maintained irrespective of the type of gas. However, the flow rate of gas fuel required to satisfy the set operating condition of 550 Nm decreased as the ignition time increased, and the flow rate of gas fuel consumed increased as the combustion rate decreased because of the characteristics of the gas fuel.

Fig. 4 shows the emission results of THC and unburned CH4 according to the optimization of the ignition timing under partial load operation conditions. As the ignition timing advances, the exhaust temperature decreases, reducing the effects of the reduction of unburned hydrocarbons and methane due to post combustion. The emission of THC and unburned CH4 increased because the unburned mixture was generated in the crevice volume during the expansion stroke of the piston due to the relatively early stage combustion termination compared to the condition where the ignition timing was retarded. For the simulated low calorific value gas mixed with N2, THC slightly increased relative to the reference gas fuel; however, the width was not large, and the emission of unburned CH4 was similar to that of the reference gas fuel. However, when using pure CH4, unburned CH4 increased significantly compared with when other gaseous fuels were used, and this affected the trend of increasing THC emissions.

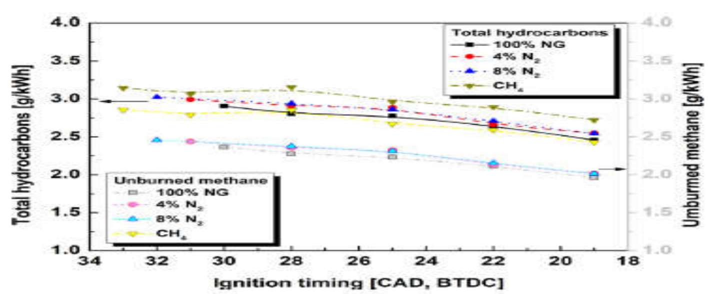

Fig. 4. Total hydrocarbons and unburned methane variations for each gas composition with ignition timing changes under part load operating conditions.

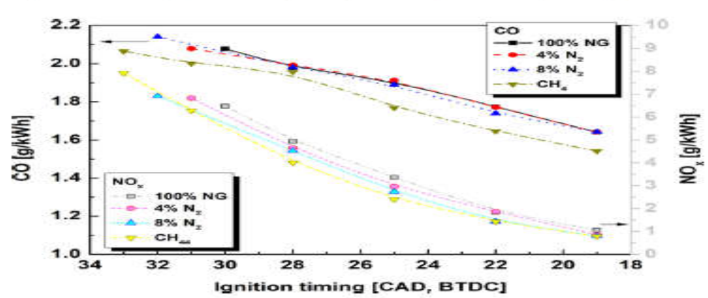

Fig. 5. Carbon monoxide and nitrogen oxide variations for each gas composition with ignition timing changes under part load operating conditions.

As shown in Fig. 5, the emission results of CO and NOx for the variations in ignition timing demonstrate that the CO emission increased as the ignition timing advanced. These results were similar to those obtained for the THC emissions; however, the extent of the increase for CO and NOx was not as large as for THC. For the simulated low calorific gases, with N2 was added into the reference gas fuel, the generation of CO, which is mainly generated in a high-temperature combustion atmosphere, decreased. For pure CH4, which exhibits the slowest combustion rate and does not contain hydrocarbon-based components with a large number of carbon atoms, the generation of CO was considerably reduced and the lowest emission result was achieved. For NOx, the combustion temperature increased, resulting in an increase in ignition time, as is the general trend for spark ignition engines. As the N2 mixing ratio increased, the emission of NOx for simulated low calorific gases decreased compared to the reference gas fuel, 100% NG, due to the decrease in combustion temperature by the dilution effect; however, the reduction was small because the dilution effect was not large. Under certain conditions of ignition timing, NOx emission was at the lowest when pure CH4 was used; however, because NOx emission is predominantly influenced by ignition timing rather than composition and physical properties of the fuel, the more advanced is the ignition period, the higher is the NOx emission under the maximum thermal efficiency condition. When pure CH4 with the most advanced ignition timing was employed, NOx emission was at the highest. Similarly, because a reduction in NOx emission is predominantly influenced by retarded ignition timing, a lower emission of NOx was obtained for pure CH4 than for the reference gas fuel under the retarded ignition timing conditions to BTDC 19 CAD. Further, the ignition timing set in the ECU, irrespective of the type of gas.

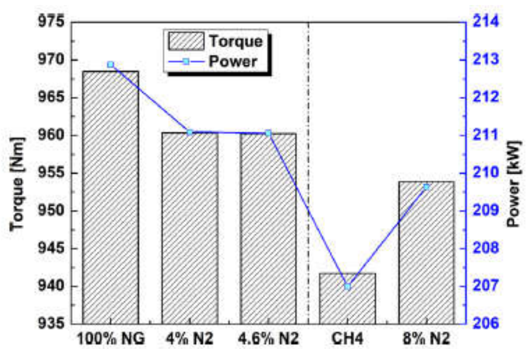

Fig. 6. Torque and power output variations for each gas composition under rated power operating conditions.

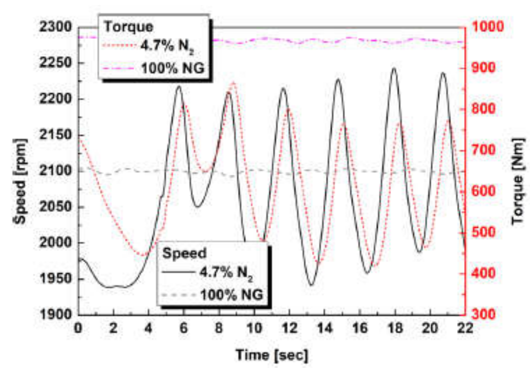

Fig. 7. Engine speed and torque variations over time for the reference natural gas, 100% NG and 4.7% of N_2 mixing into 100% NG, 4.7% N_2 under rated power operating conditions.

4.2. Optimization of rated power performance

When other conditions are constant, the change in power output is generally proportional to the calorific value of the fuel. Therefore, changes in power output are an important index for determining the interchangeability of a fuel. In general, for gas combustion devices, WI has been mainly used as a criterion for determining the interchangeability of gases related to calorific value. WI is a value proposed to determine interchangeability for general combustion devices with a fixed nozzle size; it is not suitable for use here as this concerns an engine that continuously changes the amount of fuel for controlling the air–fuel ratio. As a result of the partial load performance optimization described above, pure CH4 showed the lowest thermal efficiency owing to its slow combustion rate, although the heating value was higher than that for gas with a N2 mixing ratio of 8%. Therefore, because an engine's power output is not proportional to the calorific value, as combustion characteristics and control variables are important factors, we compared the torque and power output performance to that of existing state-of-the-art methods by performing an experiment under rated power operating conditions, taking measures to minimize any adverse effects.

Fig. 6 shows the results of the maximum torque and maximum power output for each gas fuel under the 2100 rpm full-load operating condition, which is the rated power operating condition. As expected, the maximum torque and power output decreased as N2 was mixed with the reference gas fuel, 100% NG. It is shown that N2 mixing ratio was 4.6%, and stable and steady operation was possible under the rated operation conditions. The above results are consistent with those reported by Sakai et al. (Sakai and Kuroda, 1996; Kuroda and Sakai, 1996; Kuroda et al., 1997). However, an unusual phenomenon is observed for the results with N2 mixing ratio of 4.7% and any previous studies did not discuss about the unstable operation due to the gas composition at the rated power operating condition. If the N2 mixing ratio was increased to higher than this quantity, the engine speed and torque fluctuated greatly, as shown in the results for an N2 mixing ratio of 4.7% in Fig. 7. This can be regarded as a phenomenon caused due to the effect of the changes in the composition and calorific value of the gas fuel on the control parameters of the engine. As N2 is mixed, the equivalence ratio measured in the exhaust gas does not satisfy the equivalence ratio set in the ECU. Therefore, the closed loop correction factor, an index that corrects the amount of fuel, is increased until the measured equivalence ratio reaches the set equivalence ratio value in the ECU, and the supplied fuel flow rate is increased. However, because the equivalence ratio set for the rated power operating conditions is lower than that at lower engine speed and load conditions, the process of correcting the fuel amount instantaneously results in an excessively lean condition, thereby resulting in a phenomenon in which the engine torque output decreases rapidly. The sudden decrease in power also affects the engine speed change because the absorption energy is constant by the dynamometer at steady state operation of speed/throttle mode; hence, the ECU determines that this is a transient operation rather than a steady state, and continuously changes the fuel amount and boosts pressure. As a result, stable combustion in a steady state is not possible, and engine speed and torque fluctuate repeatedly.

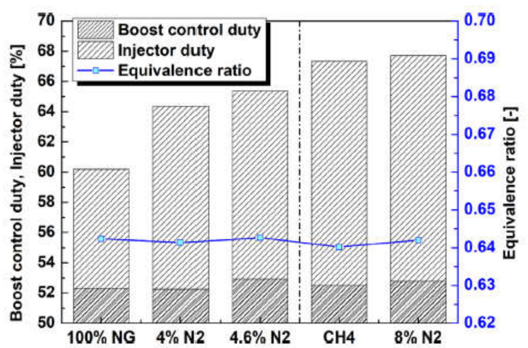

Fig. 9. Boost control duty, duty compared with the maximum possible pulse width of the fuel injector, and measured equivalence ratio variations for each gas composition under rated power operating conditions.

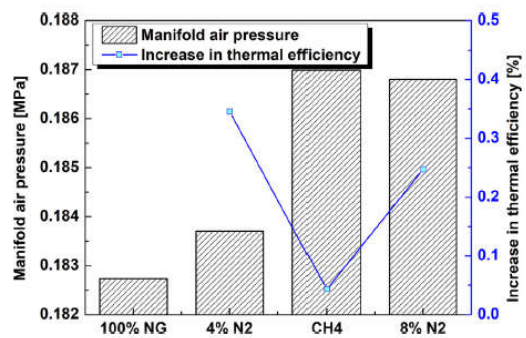

Fig. 10. Manifold air pressure and increase in thermal efficiency variations that can obtain the torque level of the reference natural gas fuel by increasing boost pressure for each gas fuel under rated power operating conditions.

When using pure CH4, the torque is lower than the reference fuel, 100% NG, because the equivalence ratio is lower than that set in the ECU. Despite this, a stable engine speed is maintained. This is shown in Fig. 8, which also depicts the changes in engine speed and torque over time, when N2 is mixed with the reference fuel. The closed-loop correction factor remained stable at the level of 1.07 for a certain period of time, but when the closed loop correction factor exceeded the set maintenance period limit, the closed loop correction factor was restored to 1.0, and the engine speed and torque rapidly decreased because the fuel amount was not corrected. After that, the engine speed and torque gradually recovered by fuel amount correction, and then, the phenomenon of maintaining a stable operation, followed by a rapid decrease, was repeated. This is related to the closed loop correction factor, and this phenomenon does not appear when learning about the properties of a fuel is completed by adaptive learning of the ECU. However, in this study, adaptive learning was excluded to examine the rated torque and power output characteristics of the composition and physical properties of fuel without considering adaptive learning. As shown in Fig. 7, it is practically difficult to proceed with adaptive learning when an instantaneous rapid variation is observed. Therefore, in this study, torque and power output were compared under the condition of maintaining stable combustion by increasing the basic fuel amount set in the ECU map based on 100% NG, to ensure that the closed-loop correction factor did not increase significantly or exceed the limit of the maintenance period. During stable operation, the results of torque and power output decreased with increasing N2 mixing ratio for simulated low calorific gas, as in the previous partial load operation condition, and the lowest value was exhibited for pure CH4. For reference, stable operation was possible when the rate of increase of the basic fuel amount was 2.35% and 9.41% for pure CH4 and 8% N2, respectively. Even if the basic fuel amount set in the ECU map is increased, this corresponds to a control variable strategy for stable operation, and the final fuel amount supplied is corrected such that a flow rate corresponding to the set equivalence ratio is supplied.

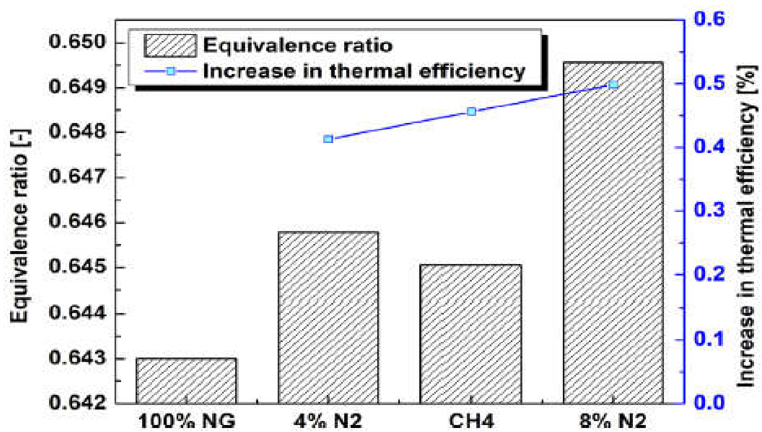

Fig. 11. Equivalence ratio and increase in thermal efficiency variations that can obtain the torque level of the reference natural gas fuel by increasing equivalence ratio for each gas fuel under rated power operating conditions.

Fig. 9 shows the results of the boost pressure control duty for each gas fuel, the duty compared with the maximum possible pulse width of the fuel injector, and the equivalence ratio measured during engine operation. The engine control variable set in the ECU was based on 100% NG, which is the reference gas fuel, and because the composition and properties of the gas fuel do not affect the set boost pressure, boost pressure control duty was maintained at a level of 52% regardless of the type of gas. Regarding fuel injector duty, the amount of fuel supplied to meet the equivalence ratio set in the ECU was continuously corrected, and the fuel injector duty increased as the ratio of N2 mixed into the reference natural gas increased under stable operating conditions. Note that, the engine power output and thermal efficiency exhibited the lowest results for pure CH4, which has the slowest combustion rate, but the fuel injector was higher than that of the 4.6% N2 mixture and lower than that of the 8% N2 mixture, because the calorific value of the fuel is higher than that of the 8% N2 mixture. As a result, the duty of the fuel injector was inversely proportional to the calorific value of the fuel, and the equivalence ratio measured by the fuel amount correction was similar to the equivalence ratio set in the ECU regardless of the type of gas.

As shown in Fig. 6, even when the supplied fuel quantity was corrected and the equivalence ratio set on the ECU was satisfied, the torque and power output decreased owing to the characteristics of the fuel, which affects the interchangeability of gas fuel. Therefore, to secure the same torque and power output that can be obtained from the reference gas fuel, it is necessary to select a method to control parameters, such as ignition timing, in highly efficient operating conditions by changing the ignition timing. However, in general, when the ignition timing is basically retarded compared with the optimum ignition timing, the exhaust gas temperature and the exhaust energy decrease. As a result, because the effect of advancing the ignition timing is canceled owing to the decrease in the boost pressure, this method is not considered for general turbo-charged engines, and was excluded from this study. Other methods include the additional supply of fuel by changing the set equivalence ratio, or an increase in the amount of the mixture in the engine combustion chamber by increasing the boost pressure.

Fig. 10 shows the results of MAP and the increase in thermal efficiency that can obtain the torque level of the reference natural gas fuel when adopting the method of boost pressure increase for each gas fuel. Because the desired equivalence ratio is constant, the amount of fuel required increases as the MAP increases. To obtain energy for boosting, the exhaust manifold pressure has to be increased; hence, pumping loss, which results in thermal efficiency deterioration, increases. As a result, the extent of the increase in thermal efficiency by torque compensation as compared with that at the MAP for reference natural gas fuel is not large. 33.76% of the thermal efficiency for 100% NG as a base point is measured and the value corresponds to the general that of heavy-duty spark ignition natural gas engine at a rated power operation point. An increase of only 0.04% in thermal efficiency was recorded for pure CH_4, while the MAP had to be increased to 0.187 MPa to satisfy the torque level. In contrast, Fig. 11 shows the equivalence ratio and increase in thermal efficiency that results from the torque level of the reference natural gas fuel by increasing the equivalence ratio for each gas fuel. Although the amount of fuel should be increased, the pumping loss does not increase under a constant boost pressure condition for the turbocharger, and thermal efficiencies generally increase with relatively richer mixtures under wide-open throttle conditions. Therefore, the extent of the increase in thermal efficiency is 0.41–0.5%, which is much larger than when increasing boost pressure, and thermal efficiencies are compensated to levels similar to those for reference natural gas fuel. The natural gas engine applied in this study is a lean combustion engine for city buses corresponding to EURO-6, and is expected to be considered about the application of PNG, represented by low-calorie gas fuel, if PNG is introduced before 2030. Since efficiency and power reduction will be the biggest issues for users in the field, appropriate measures need to be taken to solve them. The reduction in output due to the decrease in heating value is inevitable, and in order to compensate for this, it is necessary to improve the S/W or H/W of the turbocharger so that more fuel can be supplied. In the long run, it is most important to restore deteriorated efficiency by developing logic that can optimize the ignition timing according to the fuel composition.

5. Conclusions

This study aimed to identify thermal efficiency and harmful exhaust gas emission characteristics under partial load conditions, and to present a countermeasure for coping with emission gas regulations and preventing thermal efficiency deterioration. The conclusions were obtained by applying a strategy in accordance with the power reduction in the rated operation and a partial load test for each ignition timing using an 11 L six-cylinder turbo-charged engine for city buses compliant with the EURO 6 regulation.

1. As a result of the decrease in the combustion rate as N2 is mixed under the partial load operating condition, the optimum ignition timing, which represents the maximum efficiency for simulated low calorific gas mixed with 4% and 8% N2, is an additional 1 CAD and 2 CAD advancement, respectively, compared with the reference gas fuel. When pure CH4 with the slowest combustion rate is used, it is advanced by an additional 3 CAD.

2. The exhaust gas, including THC, unburned CH4, CO, and NOx decreases as the ignition timing is retarded, and as the reduction in NOx emissions is predominantly affected by the ignition timing retard, NOx is emitted at the level of the reference gas fuel under the condition that it is retarded until the ignition timing set in the ECU, regardless of the type of gas.

3. If the N2 mixing ratio is increased to 4.7% or more, under the rated power operating conditions, combustion becomes unstable during the fuel amount correction process used for controlling the equivalence ratio of the engine, causing the engine speed and torque to change rapidly.

4. Even if the N2 mixing ratio increases above a certain level, or pure CH4 is used under the rated power operating conditions, stable operation can be secured by increasing the set base fuel amount in the ECU map, but the torque and power output decrease in proportion to the combustion speed owing to the characteristics of the fuel.

5. To secure the same torque and power output that can be obtained from the reference natural gas fuel, the exhaust temperature has to increase rapidly when the equivalence ratio increases, but when the boost pressure is increased, the desired level of torque can be obtained without a rapid increase in exhaust temperature.

6. When the natural gas fuel composition changes under different operating conditions, the results of the ignition timing optimization for compensating the efficiency and securing the exhaust gas level should be assessed as future works.

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