Development of Flexible Gas Fuel Engine System*

Yudai YAMASAKI**, Yukihiro NISHIZAWA** and Shigehiko KANEKO**

**The University of Tokyo, Dept. of Mechanical Engineering,

7-3-1 Hongo, Bunkyo-ku, Tokyo, 113-8656 Japan

E-mail: yudai_y@fiv.t.u-tokyo.ac.jp

Abstract

In this study, the authors aim to develop a small gas engine system for biomass gas by modifying the control system of a conventional spark ignition engine. Before developing a control algorithm, combustion experiments with various component fuels assumed to have the components of real biomass gases, such as fermentation and pyrolysis gases, were carried out. It was clarified that the relationship between dimensionless combustion duration and equivalence ratio can be expressed by a first-order linear function regardless of fuel composition. Indicated thermal efficiency can also be expressed by combustion duration and volumetric efficiency when the coefficient of variation of indicated mean effective pressure is lower than 5% and pumping loss decreases against volumetric efficiency linearly, and the relationship between combustion duration and MBT can also be expressed by a first-order function. By using these relationships, a gas engine control algorithm, which can define the target values of the equivalence ratio of a premixture and ignition timing that realize a high thermal efficiency for fuel compositions automatically by analyzing in-cylinder gas pressure data in real time, is developed in order to use gaseous fuels produced from biomass resources effectively. Thus, a biomass gas fuel engine system is developed by applying the algorithm to an automobile gasoline engine, hardware modifications of which are only the fuel supply system and flywheel. The engine system is connected to a gasification plant using wood chips and an operation test is carried out. As a result, the engine system can set an optimum premixture condition and an ignition timing, which realizes a stable and high-thermal-efficiency operation automatically.

Key words: Internal Combustion Engine, Gaseous Fuel, Biomass Energy, Controller, Efficiency

1. Introduction

Fuels for internal combustion engines will vary in the future, because an efficient utilization of a new energy is necessary for energy security and overcoming environmental problems. Biomass resources are one of the new energies widely distributed, particularly in Japan. Their application to the local production of local consumption systems, such as distributed energy systems, but not to the production of large-scale concentrated power-generating systems, is an appropriate method because of their lean energy density.

When biomass resources are converted to gas fuels (biomass gas), the average composition of biomass gas changes with its resources and gasification method. Gas composition changes with time as gasification conditions change ⁽¹⁾. In addition, the low heating value (LHV) of biomass gas ranges from about 6 to 20MJ/m³, which is lower than that of city gas 13A (LHV 40.4MJ/m³), because biomass gas contains many

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incombustible components, such as CO_2 and N_2 . When biomass gas is used in reciprocating engines, engine systems require some response for various fuel compositions. Thus, the adjustment of operation parameters of engine controllers based on the results of the pre-composition analysis of biomass gas realizes a suitable engine operation for fuel composition changes caused by different resources. Moreover, the time fluctuation of fuel compositions caused by gasification condition changes makes the realization of a stable engine operation difficult. When small engines use biomass gas, gas engine systems are required to respond to various fuel compositions, as well as to the time fluctuation of the fuel compositions and the demanded output change with time. When the composition changes in a large-scale gas engine, the change can be reduced since the engine and the fuel supply piping system act as buffers. Stable ignition and combustion can also be achieved using a subchamber to overcome the fuel composition change. Diesel fuel injection or a jet from the prechamber can also be employed; in such a case, a premixture of biomass gas and air in the main chamber is ignited. The energy of jet and pilot fuel ignition is higher than that of a spark plug⁽²⁾. However, for a small engine, the installation of the prechamber and other parts is difficult because of the limited space and increased cost. Therefore, such a small and highly thermally efficient engine for biomass gas has not been developed.

Thus, the authors have been studying a small gas engine system fueled by low-calorific and fuels of various compositions aiming to realize a stable and high thermal efficiency operation. It was clarified that appropriate premixture conditions can realize stable ignition and combustion even in low-calorific fuels⁽³⁾. An automatic engine operation algorithm for biomass gas was also constructed. This control system can evaluate the fuel composition change by measuring in-cylinder gas pressure, analyze it thermodynamically in real time, and control engine rotational speed, amount of the supplied fuel, and ignition timing. By adopting this control system in a traditional spark ignition engine, stable engine speed and output were realized when fuel composition was changed from methane to a mock biomass gas of 1/6 LHV of city gas⁽⁴⁾.

In this study, the original engine operation algorithm⁽⁴⁾ was improved to realize a high thermal efficient operation by adjusting the most suitable premixture conditions and ignition timing automatically. In addition, the gas engine system, on which the original operation algorithm would be installed, was constructed and an operation test was carried out by connecting it to a gasification plant.

2. Nomenclature

Symbols

m_{air} : Mass flow of air [kg/s]
 m_{fuel} : Mass flow of fuel [kg/s]
 m_{pre} : Mass flow of pre-mixture [kg/s]
 M_{pre} : Molecular weight of pre-mixture [kg/mol]
 n_{cyl} : Number of Cylinders
 N_e : Engine speed [rpm]
 P_{cyl} : In-cylinder pressure [MPa]
 P_{im} : Intake manifold pressure [kPa]
 Q_{comb} : Heat quantity of combustion [J/(cycle×cylinder)]
 Q_{LHV} : Low calorific heat value [kJ/mol]
 r_{st} : Theoretical air fuel ratio [-]
 V_{cyl} : Cylinder volume [m³]
 V_d : Displacement [m³]
 $W_{c,i}$: Indicated output [J]

Greek Symbols

ϕ : Equivalence ratio
 ϕ_d : Target Equivalence ratio
 ϕ_w : Cooling loss (= 0.35) [-]
 η_{comb} : Combustion efficiency (= 0.965) [-]
 η_i : Indicated thermal efficiency [-]
 η_{vol} : Volumetric efficiency [-]
 κ : Heat specific ratio of pre-mixture [-]
 θ : Crank angle [degree]
 θ_{ign} : Ignition timing [degree]
 $\theta_{ign,MBT}$: Ignition timing of MBT [degree]
 θ_{ivc} : Intake valve close timing(= 135) [degree BTDC]
 θ_{evo} : Exhaust valve open timing(= 135) [degree ATDC]
 ρ_{fuel} : Density of fuel [kg/Nl]
 τ_{comb} : Combustion duration [degree]
 $\tau_{comb,N}$: Characteristic value of combustion duration [degree]

Subscripts

$comb$: Combustion
 cyl : Cylinder
 d : Displacement
 e : Engine
 evo : Exhaust valve open
 ign : Ignition
 im : Intake manifold
 ivc : Intake valve close
 N : Normalized
 st : Stoichiometric
 vol : Volume
 w : Water

Abbreviations

ATDC : After top dead center
BTDC : Before top dead center
LHV : Low calorific heat value
MBT : Most advance best torque

3. Experimental Setup

3.1 Engine System

In this study, an original automobile gasoline engine was diverted. Table 1 shows the engine specifications and Figure 1 shows the engine system. The engine had 6 cylinders, a displacement of 3.5L, and a compression ratio of 10.8. The engine was used with some modifications. The main modifications made were the fuel supply system and flywheel. The fuel supply system was equipped with a butterfly valve to control fuel flow, which is the same as an airflow control valve; this butterfly valve was designed with a gas pressure of almost 0.1MPa at the outlet port of the gasification pit. Its diameter was 42mm, which is the same as that of the airflow control valve. The difference between fuel and air flow rates is small in this engine system, because the theoretical air and fuel ratio for an assumed average fuel composition ($H_2:CO:CH_4:N_2:CO_2 = 30:17:3:30:20$, $LHV=6.5MJ/Nm^3$) is 1.48. The fuel composition was assumed in the pyrolysis gas of wood, which was produced using both a rotary kiln and a downdraft pit⁽⁵⁾. Two absolute pressure sensors were placed at the upstream point of the butterfly valve for fuel and at the mixture point of air and fuel, which was downstream of the fuel and air flow valves, in order to estimate fuel and air flow rates. The inertia moment of the original flywheel was designed to be as large as possible ($330 \times 10^3 kg \cdot mm^2$), because this engine system was used for power generation, in which a constant engine speed operation is preferable. A rotary encoder was attached to the crankshaft, and signals from the encoder were used to control the engine and as a trigger clock of data acquisition. A piezo-type pressure sensor was set on one cylinder to determine in-cylinder gas pressure. Using an inverter motor attached to the crankshaft, start up and generation operations were carried out. An engine controller was developed

Table1 Engine specification

Type name	TOYOTA 2GR-FE
Type	V-type, 6cylinders
Displacement cc	3456
Bore and stroke mm	94 x 83
Compression ratio	10.8

Fig.1 Engine system

using general-purpose FPGA devices. In addition, the experimental data of a 3-cylinder engine (bore x stroke: 88 x 90; compression ratio: 9.8)⁽³⁾ were used in the development phase of the control algorithm of premixture conditions.

3.2 Biomass Gas Supply System

Both mock and actual biomass gases were used in our experiments. Mock biomass gases were mainly used in order to obtain basic combustion data. Mock biomass gases consist of hydrogen (H_2), methane (CH_4), carbon monoxide (CO), nitrogen (N_2), and carbon dioxide (CO_2), which are the main components of actual biomass gas. These components were supplied from high-pressure gas cylinders to the gas engine, and mass flow controllers adjusted their flow rates to produce any components. Moreover, actual biomass gas was used in a verification test. Actual biomass gas was generated from woodchips using the rotary kiln and the downdraft bed⁽⁵⁾.

4. Engine Operation Algorithm

An algorithm of automatic engine control was developed to realize a stable and high thermal efficient operation for varying fuel components. The control items in this system are the equivalence ratio of the premixture, ignition timing, and engine speed. Figure 2 shows a block diagram of this system. The control part consists of three parts, namely, the engine speed control part, the equivalence ratio control part and the ignition timing control part. The actuators of this system are a throttle valve for air and a throttle valve for fuel and spark plugs; each actuator is controlled by a feedback loop with a PID controller. First, the data on intake manifold pressure, fuel manifold pressure, engine speed, the position of the throttle valve for fuel, the position of the throttle valve for air and in-cylinder gas pressure are obtained using the sensors. Then, these data are fed to the combustion analysis block, and the theoretical air and fuel ratio, volumetric efficiency, equivalence ratio, combustion duration and the indicated work are calculated. Next, at the high-efficiency operational condition prediction block, appropriate equivalence ratio and ignition timing are obtained for the fuel composition to realize a highthermal efficiency. The calculation processes in the combustion analysis and high-efficiency operational condition prediction blocks are as follows.

4.1 Combustion Analysis Block

The combustion analysis block estimates airflow rate, fuel flow rate, heat release rate,

Fig.2 Block diagram of developed engine controller

LHV, theoretical air and fuel ratio and equivalence ratio of the fuel.

4.1.1 Estimations of Air Flow and Fuel Flow

The mass flow rates of air and fuel are estimated from the positions of the throttle valves, the pressures of intake manifold and the temperature.

4.1.2 Heat Release

First, the heat release rate is calculated for each cycle using Eq.(1). Next, the amount of combustion heat of one cycle is calculated through the integration of the heat release rate (Eq.(2)). When the heat release rate is assumed to be an isosceles triangle and the area is appropriate for the combustion heat, combustion duration is defined as base length that is twice the amount of combustion heat divided by the peak value of heat release rate (Eq.(3)). Indicated work is also calculated using Eq.(4).

$$\frac{dQ_{comb}}{d\theta_{CA}} = \frac{1}{\kappa - 1} (\kappa P_{cyl} dV_{cyl} + V_{cyl} dP_{cyl}) \quad (1)$$

$$Q_{comb} = \int_{\theta_{rc}}^{\theta_{ev}} dQ_{comb} = \frac{1}{\kappa - 1} \int_{\theta_{rc}}^{\theta_{ev}} (\kappa P_{cyl} dV_{cyl} + V_{cyl} dP_{cyl}) d\theta_{CA} \quad (2)$$

$$\tau_{comb} = 2Q_{comb} / \left(\frac{dQ_{comb}}{d\theta_{CA}} \right)_{peak} \quad (3)$$

$$W_{c,i} = \oint P_{cyl} dV_{cyl} \quad (4)$$

4.1.3 Estimations of LHV, Theoretical Air and Fuel Ratio and Equivalence Ratio

By considering the combustion heat calculated from Eq.(2), the LHV of the fuel can be expressed by Eq.(5). The relationships between the LHVs and theoretical air and fuel ratios of H₂, CH₄ and CO, which are the main combustible components of biomass gas, and gas fuels diluted appropriately with N₂ and CO₂ are linear, as expressed by Eq.(6) ⁽⁴⁾. Thus, calculating in-cylinder gas pressure from Eqs.(1),(2),(5) and (6) in real time provides the theoretical air and fuel ratio, even if fuel components are unknown. In addition, equivalence ratio is also expressed by Eq.(7) in real time with the air and fuel flow rates obtained previously. In this study, combustion efficiency and cooling loss were treated as constants; these are the general values ⁽⁶⁾ shown in the nomenclature.

$$Q_{LHV} = \frac{293 \cdot 22.4}{273 \cdot m_{fuel} / \rho_{fuel}} \frac{n_{cyl} \cdot N_e}{120} \frac{Q_{comb}}{\eta_{comb} (1 - \phi_w)} \times 10^{-3} \quad (5)$$

$$r_{st} = -0.42 + 0.0124 Q_{LHV} \quad (6)$$

$$\phi = \frac{r_{st}}{m_{air} / m_{fuel}} \quad (7)$$

4.2 High-Efficiency Operational Condition Prediction Block

This section shows how to select target values for realizing a high-thermal-efficiency operation by analyzing combustion properties. Combustion start timing, combustion duration and pumping loss are affected by the combustion characteristics of fuels, equivalence ratio, ignition timing and operation conditions. Among the three control variables, namely, the equivalence ratio, ignition timing and engine speed of this system, equivalence ratio and ignition timing are controlled to realize a high thermal efficiency, because engine speed is assumed to be constant for power generation. The following sections show how to select the target values of equivalence ratio and ignition timing automatically when the fuel component varies.

4.2.1 Target Equivalence Ratio

A real-time control strategy of equivalence ratio along with the fuel combustion properties is considered. It is not clear which premixture condition, lean or stoichiometric, is more appropriate for realizing a high thermal efficiency, when the fuel component varies. Lean premixture operation shows a lower pumping loss than stoichiometric premixture operation, but its combustion duration is generally longer. On the other hand, stoichiometric premixture operation realizes a higher degree of constant volume but a higher pumping loss than lean premixture operation. Thus, first, the relationships between combustion duration, volumetric efficiency and indicated work at different equivalence ratios for various fuels under steady engine operation conditions were analyzed. Figure 3 shows the relationship between volumetric efficiency and indicated thermal efficiency for each combustion duration condition. The fuel components of the plots are as follow: 9-100% CH₄, 0-30% H₂, 10-50% CO₂ and 10-60% N₂. Ignition timings ranged from -50 to -30deg ATDC, and equivalence ratios, from 0.8 to 1.0. In addition, these components were assumed to be actual biomass gas components: fermentation gas (CH₄:CO₂ = 60:40) and pyrolysis gas (CH₄:H₂:CO:N₂:CO₂ = 0-15:10-40:10-30:45-70). Indicated thermal efficiency linearly increased with an increase in volumetric efficiency. A short combustion duration increased the indicated thermal efficiency at the same volumetric efficiency. In addition, pumping loss was confirmed to decrease linearly with an increase in volumetric efficiency. Thus, as shown in Figure 3, the relationship between indicated thermal efficiency and volumetric efficiency can be approximated in a first-order linear function regardless of fuel composition for each combustion duration condition. When the

Fig.3 Relationship between volumetric efficiency and indicated thermal efficiency

Fig.4 Gradient and intercept of approximate linear function on volumetric efficiency and indicated thermal efficiency

gradients and intercepts of these approximated lines are assumed to be functions of combustion duration, they are also expressed as linear functions for combustion duration, as shown in Figure 4. The horizontal axis of Figure 4 indicates is dimensionless combustion duration, which is divided by 55deg of combustion duration as the characteristic value. The gradients of functions in Figure 3 decreased and the intercepts increased with an increase in dimensionless combustion duration. Thus, the relationship between combustion duration, volumetric efficiency and indicated thermal efficiency can be expressed by Eq.(8).

$$\eta_i = (-0.538 \frac{\tau_{comb}}{\tau_{comb,N}} + 0.862) \eta_{vol} + 0.0814 \frac{\tau_{comb}}{\tau_{comb,N}} + 0.102 \quad (8)$$

Next, the relationship between equivalence ratio and combustion duration was discussed (Figure (5)). A decrease in equivalence ratio made combustion duration longer regardless of fuel composition. The vertical axis of Figure 6 indicates the transformed vertical axis of Figure 5 divided by combustion duration at an equivalence ratio of 1.0. Dimensionless combustion duration showed a first-order dependence on equivalence ratio, as shown in Figure 6, regardless of fuel composition. Thus, a change in combustion duration was approximated using Eq.(9), when equivalence ratio changed from ϕ to ϕ_d .

$$\frac{\tau_{comb}(\phi_d)}{\tau_{comb}(\phi)} = \frac{-0.9\phi_d + 1.9}{-0.9\phi + 1.9} \quad (9)$$

In addition, the relationship between equivalence ratio and volumetric efficiency was discussed. Generally, the relationship between volumetric efficiency and the mass of the premixture trapped in a combustion chamber is expressed by Eq.(10).

$$m_{pre} = \eta_{vol} \times \frac{N_e}{120} \times V_d \times \frac{M_{pre}}{22.4} \quad (10)$$

By assuming only airflow changes, when equivalence ratio changes, Eq.(11) is derived.

$$\eta_{vol}(\phi_d) - \eta_{vol}(\phi) = \frac{120 \times 22.4}{N_e V_d M_{pre}} (m_{air}(\phi_d) - m_{air}(\phi)) \quad (11)$$

Equation (12) is derived from Eqs.(7) and (11), and transformed to Eq.(13).

$$\begin{aligned} \eta_{vol}(\phi_d) - \eta_{vol}(\phi) &= \frac{120 \times 22.4}{N_e V_d M_{pre}} r_{st} m_{fuel} \left(\frac{1}{\phi_d} - \frac{1}{\phi} \right) \\ &= \frac{120 \times 22.4}{N_e V_d M_{pre}} m_{pre} \frac{r_{st} \phi}{r_{st} + \phi} \left(\frac{1}{\phi_d} - \frac{1}{\phi} \right) \\ &= \eta_{vol}(\phi) \frac{r_{st} \phi}{r_{st} + \phi} \left(\frac{1}{\phi_d} - \frac{1}{\phi} \right) \end{aligned} \quad (12)$$

$$\eta_{vol}(\phi_d) = \eta_{vol}(\phi) \left(1 + \frac{r_{st} \phi}{r_{st} + \phi} \left(\frac{1}{\phi_d} - \frac{1}{\phi} \right) \right) \quad (13)$$

From Eqs.(8), (9) and (13), indicated thermal efficiency is expressed by Eq.(14) using the function of equivalence ratio and combustion duration.

$$\eta_i = (-0.54 \frac{\tau_{comb}(\phi) - 0.9\phi_d + 1.9}{\tau_{comb,N} - 0.9\phi + 1.9} + 0.86) \times \eta_{vol}(\phi) \left(1 + \frac{r_{st}\phi}{r_{st} + \phi} \left(\frac{1}{\phi_d} - \frac{1}{\phi} \right) \right) + 0.081 \frac{\tau_{comb}(\phi) - 0.9\phi_d + 1.9}{\tau_{comb,N} - 0.9\phi + 1.9} + 0.1 \quad (14)$$

Thus, the target equivalence ratio is selected by incorporating the estimated instantaneous equivalence ratio for the fuel and the combustion duration analyzed from in-cylinder gas pressure into Eq.(14). However, emissions are not discussed for selecting the target equivalence ratio, because biomass gas emits lower THC and NOx than methane, which is the main component of city gas 13A, at the same equivalence ratio and ignition timing⁽⁷⁾. In addition, when the engine system is placed in a biomass-generating unit with a gasification pit, exhaust gas can be fed to the pit, and various after-treatment methods are applied easily. In addition, for the target equivalence ratio defined by the automatic controller, from the industrial viewpoint, the upper and lower limits preventing high levels of emissions can be set easily by changing the parameters of the control software.

4.2.2 Target Ignition Timing

In gasoline engines, ignition timing is generally advanced until knock. This ignition control method cannot be applied to the biomass gas fuel engine. No knock may occur even in advanced ignition timing in this engine, because the specifications of the original engine, inlet valve timing, exhaust valve timing and compression ratio are not modified in order to realize low costs. Thus, an ignition control algorithm, which is based on

Fig.5 Relationship between equivalence ratio and combustion duration

Fig.6 Relationship between equivalence ratio and non-dimensional combustion duration

Fig.7 Relationship between combustion duration and MBT

in-cylinder gas pressure, was developed.

Figure 7 shows the combustion duration and ignition timing of minimum advance for best torque (MBT) for loads. In this experiment, actual biomass gases were used and their components were follows: $\text{CH}_4:\text{H}_2:\text{CO}:\text{N}_2:\text{CO}_2 = 3\text{-}4\text{:}12\text{-}17\text{:}17\text{-}21\text{:}43\text{-}49\text{:}8\text{-}14$. Engine speed was 1800rpm and equivalence ratio ranged from 0.7 to 1.0. The ignition timing of MBT increased linearly with an increase in combustion duration for each load condition. This relationship is universal. In a spark ignition engine with gasoline, the crank angle, at a maximum in-cylinder gas pressure, is constant when ignition timing is adjusted to MBT. The relationship between combustion duration and such a crank angle is first-order linear at the conditions that combustion duration is shorter than 60deg⁽⁸⁾. Thus, the gradients and intercepts of functions, which indicate the relationship between combustion duration and ignition timing, are expressed as functions of indicated work, and Eq.(15) is derived.

$$\theta_{\text{ign,MBT}} = (-0.403 \frac{W_{c,i}}{W_{c,i,N}} + 2.50) \tau_{\text{comb}} - 2.57 \frac{W_{c,i}}{W_{c,i,N}} - 11.7 \quad (15)$$

The target ignition timing is selected by incorporating the indicated work and combustion duration analyzed from in-cylinder gas pressure into Eq.(15). Here, $W_{c,i,N}$ is characteristic indicated work 100[J/cycle].

5. Operation Test

An engine operation test was carried out using the engine control unit installed with the developed control algorithm. After examining the operations of the algorithm, a generating test was carried out.

5.1 Examination of Operation of Control System

5.1.1 Estimation of LHV

The control system was examined to estimate LHV in real time. Figure 8 shows the result of the operation test obtained when the fuel was supplied by mass flow controllers, the target engine speed was 1500rpm and the load was 1kw. In this experiment, the target equivalence ratio and ignition timing were imposed without performing high-efficiency operational condition prediction block. Their values were 0.75 and -75deg ATDC, respectively. The flow set signals of the mass flow controllers were changed from $\text{CH}_4:\text{H}_2:\text{CO}:\text{CO}_2:\text{N}_2 = 3\text{:}30\text{:}17\text{:}20\text{:}30$ to $\text{CH}_4:\text{H}_2:\text{CO}:\text{CO}_2:\text{N}_2 = 3\text{:}20\text{:}17\text{:}20\text{:}4$ at 50s, which was changed back to the initial composition at 150s, and it was also changed from the initial composition to $\text{CH}_4:\text{H}_2:\text{CO}:\text{CO}_2:\text{N}_2 = 10\text{:}30\text{:}10\text{:}20\text{:}30$ at 230s, which was changed back to

Fig. 8 LHV estimation test

Fig. 9 Operation test with equivalence ratio control

the initial condition at 330s again. The estimated LHV changed with the actual fuel LHV change, and the engine speed was kept at the target value of 1500rpm. In addition, the phase difference between the estimated LHV and the actual fuel LHV was responsible for the length of the fuel supply pipe from the place of the mass flow controllers, which was about 15m from the engine.

5.1.2 Control of Equivalence Ratio

The tracking performance of equivalence ratio was tested. Figure 9 shows the operation

test result obtained when the target equivalence ratio was manually changed at the target engine speed of 1800rpm and at the constant load of 10kW. Ignition timing was adjusted using the automatic engine controller. Equivalence ratio could follow the target equivalence ratio. The throttle valve for airflow was operated by adjusting equivalence ratio, and the opening ratio of the throttle valve for airflow increased with a decrease in target equivalence ratio. Engine speed was also kept constant, and the controller could follow the target equivalence ratio change adequately.

5.1.3 Control of Ignition timing

The control performance of ignition timing was tested. Figure 10 shows the operation

Fig. 10 Operation test with ignition timing control

Fig.11 Effect of control of targets of equivalence ratio and ignition timing

test result with actual biomass gas obtained when the target equivalence ratio was manually changed at the target engine speed of 1800rpm and at the constant load of 10kW. The decreased target equivalence ratio yielded a long combustion duration, and the increased target equivalence ratio yielded a short combustion duration. Ignition timing was adjusted automatically with the combustion duration change. Since a long combustion duration required an advanced ignition timing, the ignition-timing controller was confirmed to function appropriately.

5.2 Generating Test

An operation test was carried out using the automatic engine controller and the actual biomass gas supplied from a gasification pit. Figure 11 shows operation test results. In this figure, there are two operation test results obtained under different sets of conditions; one operation test result was obtained by adjusting target equivalence ratio and target ignition timing automatically, and the other operation test result was manually obtained at the equivalence ratio constant of 1.0 and at the ignition timing constant of -70deg ATDC. In addition, the fuel components in the tests were follows; $\text{CH}_4:\text{H}_2:\text{CO}:\text{N}_2:\text{CO}_2:\text{Others} = 4.6:17:16.9:46.4:9.5:5.6$ in the operation adjusting target values, and $\text{CH}_4:\text{H}_2:\text{CO}:\text{N}_2:\text{CO}_2:\text{Others} = 3.5:14.2:16.6:48.7:12.1:4.9$ in the operation with constant target values. The difference of fuel components was caused from the operation condition of the gasification pit.

In this operation condition, lower equivalence ratio was appropriate for realizing a higher thermal efficiency, the automatic engine controller decreased equivalence ratio and advanced the ignition timing. The target equivalence ratio, however, increased once at approximately 300s. This was caused by another safety algorithm installed in order to prevent the induction of the equivalence ratio expressed by Eq.(14), which exceeded the lean condition. The indicated thermal efficiency of the automatic control system was higher than the target constant condition. The advantage of the automatic control system was verified.

6. Conclusion

A small gas engine system used to realize a stable and high-thermal-efficiency combustion for various gas fuel components was developed in order to use biomass resources effectively. It was clarified that the relationship between equivalence ratio and combustion duration can be expressed by a first-order linear function, and that the ignition timing of MBT depends on combustion linearly. An automatic control algorithm, which defines the target equivalence ratio and ignition timing in order to realize a high thermal efficiency for varying fuel components using in-cylinder gas pressure in real time, was developed. By installing the control algorithm to an engine, which is originally used as a gasoline engine for automobiles, a generating test was carried out. The developed automatic control algorithm could define the target equivalence ratio and ignition timing for varying fuel components automatically and realize a high-thermal-efficiency operation. The engine system with the developed automatic control system was verified to use biomass gas efficiently.

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