Research and Development of Mixed and Standard Type Biomass Gas Turbine System with Enhanced Fuel Applicability

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Niigata University and TERI collaborated in developing a biomass gas turbine system with an enhanced fuel applicability for rural villages in south Asia. This paper describes a mixed and standard type gas turbine system for this purpose. The mixed type gas turbine employs a regenerated Brayton cycle with a secondary combustor to burn various fuels. The thermodynamic analysis revealed that the mixed type gas turbine system has a capability to achieve the thermal efficiency between a regenerated Brayton cycle and a complete external fired system. An experimental test was done using mixtures of bio diesel fuel and kerosene. It demonstrated that the engine performance changed only slightly with regard to the mixture ratio. In the experiment, a prototype biomass gas turbine was able to achieve the 600 W power generation. The mixed fuel with the mixture ratio of bio fuel up to 60% was confirmed to be applicable to the current gas turbine system.

Key Words

Biomass fuel, Compact gas turbine system, Thermodynamic analysis, Prototype experiment

1. Introduction

In south Asia, almost four billion people live in remote villages not connected to any power system. The utilization of decentralized system based on renewable energy is expected to prevail in such villages. Focusing on the development of the decentralized system is in line with the recent world-wide effort on preventing global warming by suppressing the rising consumption of fossil fuel due to modernization. Recently, researchers have been developing biomass liquefiers, biomass gas producers and reciprocate engine biomass power systems for rural villages. However, such reciprocate engines can only be used with purified bio-

fue...
is made for the mixed type gas turbine system. Secondly, the experiment is made for the standard gas turbine using the bio-diesel fuel (BDF).

2. Field Research in India

The field research has been conducted in Haryana and Tamil Nadu states of India since 2009. Figs 1 and 2 are the photographs taken in Mulana-Ambala of Yamuna Nagar, Haryana. Yamuna Nagar, 280 km north of New Delhi, has a population of 380 thousands. Its major industry is lumber. The central town is modernized; however, most areas have kept their pre-modern appearances. The people in the rural area continue to use wood, agriculture waste and dried cow dung as fuel. The Chander Pul Works, Yamuna Nagar, is an engineering company producing steel-industry equipment and biomass gasification reactor. Its down-draft fixed-bed reactors have been installed into Indian remote villages to be built in decentralized power systems.

Based on the field work and existing literatures, Indian decentralized electrified area can be conceptualized in Fig. 3. Since it is remote from major cities, the energy system needs to be sustained by recovering natural energy from local facilities. The circulation of energy and material is expected to progress into the network for promoting environmental preservation and the industrial advancement as shown in Fig. 4.

3. Analysis of Mixed-type Gas Turbine

Fig. 5 shows the mixed type gas turbine cycle. This is a mixed system of the standard and the externally fired (EF) gas turbine. This cycle corresponds to a regenerated Brayton cycle equipped with a secondary combustor. Here, primary combustor is installed before the turbine as the standard Brayton cycle. A secondary combustor is incorporated after the turbine to burn various fuels. The exhaust gas from a secondary combustor does not pass through the turbine. The secondary combustor thus can...
burn various fuels including the crude biomass. The gas
turbine system with a secondary combustor is an externally
fired system.

This technology is expected to overcome the
deficiency of the biomass gas engines with internal
combustion, which are likely to suffer from dirty exhaust
gas. The benefit of utilizing this new technology is
promising in remote villages where the conventional fuels
are scarcely provided but there is an abundant supply of
biomass. This can also be fueled by heat sources such as
the exhaust heat from a factory or the concentrated solar
heat.

Fig. 6 is a photograph of the 2 kW prototype of
mixed type gas turbine system. This is configured by an
automobile turbo charger, two combustors, a stainless-steel
plate type heat exchanger and a generator. Two combustors
are for liquid fuels with different purification. The system
is equipped with the flow controllers for regulating flow rates
of fuels. This test system is currently being adjusted for
stable operation. This paper describes the thermodynamic
analysis of the mixed type system.

Fig. 7 shows T-s diagram of the mixed type gas
turbine system. The compression of air 1-2 and the
expansion of gas 3-4 are illustrated isentropic ideally.
However, the analysis considers the irreversibility
generating entropy in those processes. The adiabatic
efficiencies of the compressor and that of the turbine are
defined as
\[
\eta_C = \frac{h_2 - h_1}{h_2' - h_1}
\]
\[
\eta_T = \frac{h_3 - h_4}{h_3' - h_4}
\]
where \(h_2 [J/kg]\) and \(h_1 [J/kg]\) are the real enthalpies after
irreversible processes. The EF combustor after the turbine
can burn crude fuel producing hot gas at 6. The gas heats
the compressed air at 2 via the regeneration heat exchanger.

The temperature efficiency of the heat exchanger is
\[
\eta_R = \frac{h_7 - h_2'}{h_6' - h_2'}
\]

Heat per unit mass working gas and net work by that are
respectively written as
\[
q_H = (h_3 - h_7) + (h_6' - h_4') (J/kg)
\]
\[
l = (h_3 - h_4') - (h_2' - h_1) (J/kg)
\]
The thermal efficiency of the cycle is
\[
\eta_\text{th} = \frac{l}{q_H}
\]
The heat and the net work can be rewritten by non-
dimensional parameters as
\[
\frac{q_H}{c_pT_1} = \tau \left\{ \left( 1 - \eta_R \right) + \eta_R \left( 1 - \varphi^{(\tau - 1)} \right) \right\}
\]
\[- \left( 1 - \eta_R \right) \left\{ 1 + \left( 1 - \varphi^{(\tau - 1)} \right) / \eta_C \right\} \right\}
\]
\[
\frac{l}{c_pT_1} = \eta_T \left( 1 - \varphi^{(\tau - 1)} \right) - \left( \varphi^{(\tau - 1)} \right) \eta_C
\]

These equations use the pressure ratio, the temperature
ratio and the EF temperature ratio defined, respectively, as
\[
\varphi = \frac{p_2}{p_1} = \frac{p_3}{p_4}
\]
\[
\tau = \frac{T_s}{T_1}
\]
\[
\tau_{EF} = \frac{T_s}{T_1}
\]

The EF temperature, \(T_s\), must be higher than the
turbine exit temperature \(T_4\) and lower than limit as
\[
\frac{T_s}{T_1} \leq \tau_{EF} \leq \tau_{EF, \text{max}}
\]
The upper limit of the EF temperature ratio, \(\tau_{EF, \text{max}}\),
is decided so that the heat exchanger exit temperature, \(T_5\),
reaches the turbine inlet temperature, \(T_1\). This condition
leads to
\[
\tau_{EF, \text{max}} = \frac{\tau}{\eta_R} + \left\{ 1 - \frac{1}{\eta_R} \right\} \left\{ 1 + \left( \frac{\varphi^{(\tau - 1)} - 1}{\eta_C} \right) \right\}
\]
When \(\tau_{EF} = \tau_{EF, \text{max}}\), the cycle corresponds to the external
fired system without the assist combustor.

It is notable that equation (7) is a generalized form
of heat by regeneration cycle. When the EF temperature,
$T_e$, is the same as the turbine exit temperature, $T_e$, the mixed cycle corresponds to the regenerated cycle without EF combustor. Namely, equation (7) yields the heat of regeneration cycle when replacing $\tau_{EF}$ with $\tau$.

Fig. 8 shows the net work by the mixed gas turbine system for the condition listed in Table 1. The net work of the mixed system is identical to the standard system without the regenerator nor the EF combustor, or to the regeneration system without the EF combustor. The net work depends on the temperature ratio $\tau$, but is irrelevant to the EF temperature ratio, $\tau_{EF}$. In the figure, the net work is maximized around the pressure ratio of 4.0.

The thermal efficiency of the mixed system for the condition of Table 1 is shown in Fig. 9. The figure presents the standard gas turbine system without the regenerator nor the EF burner, the regeneration system ($\tau_{EF} = T_{4'}/T_1$) and the EF system ($\tau_{EF} = \tau_{EF,\text{max}}$) for the condition shown in Table 1. The thermal efficiency of the mixed, the regeneration and the EF system is larger than the standard cycle when the pressure ratio is less than 5.6. This means that the exhaust heat can be recovered only when the turbine exit temperature at 4 is higher than the compressor exit temperature at 2 for relatively low values of pressure ratio as suggested by T-S diagram of Fig. 7.

In Fig. 9, the thermal efficiency of the mixed cycle and the EF cycle are lower than the regeneration cycle for the whole range of pressure ratio. However, the mixed and the EF cycles enhance the thermal efficiency to 0.26 or 0.27 from 0.2 of the standard cycle at the optimized pressure ratio. The mixed and the EF cycles are obviously advantageous since they can burn wider range of fuels than the standard or the regeneration cycles.

4. Experimental Test of Standard Gas Turbine Power Generation

An effort has been made also for experimental consideration of the bio-oil fueled gas turbine by present authors. The electricity generation was made by the power-turbine-equipped standard gas turbine shown in Figs 10 and 11. The engine part is JetCat P60. Experiments are made in Ikarashi campus of Niigata University (Ikarashi 2-nocho, Niigata, Japan). Options for the joint development and testing of the integrated system with TERI’s patented technologies, namely biomass pyrolysis oil and gas from biomass gasifiers, are considered for the advanced future developments in TERI Gram (Gual Pahari, Gurgaon, India) from 2012 through 2013 (Fig. 12).

The experimental tests were conducted using

| Table 1 Condition for mixed turbine calculation |
|-----------------------------------------------|
| $\eta_T$ | 0.85 |
| $\eta_C$ | 0.85 |
| $\eta_R$ | 0.85 |
| $\kappa$ | 14 |
| $\tau$ | 30 |
| $\tau_{EF}$ | 30 |
kerosene fuel. The gas turbine system was capable of generating maximum of 600 W electricity. Although the power was successfully generated in the experiments, it was suggested that the fundamental examination was expected to be made by a simpler method. This is because there was a significant loss in the energy transfer from the compression turbine to the power turbine. The present study thus focuses on the gas turbine experiment with the jet nozzle to reveal the applicability of the bio-oil-additive fuels to the combustion and compression turbine of the standard gas turbine cycle.

5. Experimental Test of Gas Turbine with Jet Nozzle

5.1 Experimental apparatus

This study uses the Germany-made compact gas turbine engine, P60, for the test of bio-oil mixed fuel. The specification of the engine is the same as the engine for power generation. The power turbine of this engine was replaced by the jet nozzle for load. The total length is 225 mm. The diameter of the jet nozzle is 54.6 mm at the inlet and 40 mm at the exit. The schematic diagram of the gas turbine with nozzle is shown in Fig. 13, and the photograph in Fig. 14. The measurement system is shown in Fig. 15. The gas turbine is fixed on the linear guide. The load cell transducer is used to measure thrust by the jet stream.

5.2 Bio-oil-mixed fuel

This study uses the bio diesel fuel (BDF) as a bio oil for the mixture fuel. The BDF prepared by the Ecology Project Niigata Company is mixed with kerosene to make the mixture fuel. The BDF used in the experiments was made through methyl esterification of waste cooking oil, with addition and removal of components. The vegetable oils have high viscosity and low cetane numbers. The production process adjusts the properties in useable level. The photograph of the BDF and other oils are shown in Fig. 16.

In the present study, the measurements of the kinematic viscosity of the BDF-mixed kerosene are taken by an Ubbelohde viscometer. Fig. 17 shows the kinematic viscosities for various mixture ratios at 40 °C. The figure also shows the values for the mixture oils with turbine oil and the mixture oils with Jatropha oil. The Jatropha oil is a crude oil made from Jatropha seeds without esterification. It shows the viscosity at 30 times as high as that of the kerosene. Although the mixture of kerosene reduces the viscosity of Jatropha oil, the mixed oil still shows very high viscosity at the mixture ratio of 60%. The BDF has...
the viscosity four times as high as that of the kerosene. However, the viscosity of the BDF is reduced to two times of kerosene at the mixture ratio of 60%. Therefore, the esterified BDF is less difficult in handling for burning.

Also measured in the present study is the density of the bio-oil-mixed kerosene. The result is shown in Fig. 18. The BDF has slightly lower density than Jatropha oil. The density is shown to change in proportion with the mixture ratio. The lower heating values (LHV) are calculated using the measured density, and they are shown in Fig. 19.

The experiment of gas turbine is made by using kerosene or its mixture with the BDF. In the case of kerosene without the BDF, the oil is blended by 4% turbine oil for lubrication for the experiment. This paper deals with the mixture ratio of BDF-mixed kerosene changing mixture ratio from 10% to 60%. The lower heating value of the BDF is 80-90% of kerosene due to the chemically bound oxygen as suggested in Fig. 19.

5.3 Experimental result
5.3.1 Thrust and exhaust gas temperature

The gas turbine was able to operate using the mixed fuel for the mixture ratio up to 60%. White smoke was observed for the cases of higher mixture ratio. However, the operation and measurement were successful for the tested range of the mixture ratio.

The fuel consumption by the gas turbine is shown in Fig. 20. The horizontal axis is the engine revolution per minute. The gas turbine consumes the fuel at 0.5 g/s to 3.0 g/s. The consumption increases according to the increase of the engine revolution. Despite of the considerable changes in properties of bio-oil-additive fuel, the fuel consumption for the case of mixed fuel almost follows that for kerosene with no bio oil. The slight reduction occurs at the highest revolution. This means the possibility of bio oil for reducing the fuel consumption. However, this reduction is too small for quantitative evaluation as the advantage of mixed fuel.

Fig. 21 shows the thrust by the gas turbine. Addition of the bio oil makes almost no essential difference to the thrust curves. The gas turbine produces the thrust of 3 N to 35N according to the engine revolution. Although the fuel consumption changes nearly linearly, the thrust increases exponentially. The gas turbine thus produces the thrust more efficiently at higher revolutions.

The gas temperature at the nozzle exit is shown in Fig. 22. The temperature decreases slightly and increases with increasing revolution. These changes imply the change of air fuel ratio discussed later. In the mixed oil cases, there
is some temperature rise around revolution of 80,000 rpm. Since this is not consistent with the gas analysis in later discussion, this temperature rise is thought to come from the after burning of the unburnt bio oil.

5.3.2 Gas analysis

The exhaust gases from the jet nozzle are analyzed by the testo gas analyzer. The concentration of oxygen is shown in Fig. 23. The oxygen concentration generally decreases with increase of the engine revolution. This is consistent with the decrease of air fuel ratio hinted by the temperature rise in Fig. 22. The air intake per fuel is thus suggested to decrease with increase of engine revolution. As to the bio oil effects, the oxygen concentration generally increases over the whole tested range of the revolution. This increase does not occur by decreasing the air fuel ratio. This is because the temperature averts such trend of the air fuel ratio. The increase of oxygen concentration for the mixed oil cases can occur from the increase of the chemically bound oxygen in bio molecules.

Fig. 24 shows concentration of carbon monoxide. There is an increase of carbon monoxide for high revolution over 150,000 rpm regardless of the bio oil mixing ratio. This increase matches the decrease of the air fuel ratio suggested by the oxygen concentration and the temperature. Although the bio oil effects do not appear in the high revolution regime, there is a remarkable increase of carbon monoxide at revolutions around 90,000 rpm. This increase do not occur by the change of air fuel ratio since the oxygen and the temperature trends contradict the decrease of the air fuel ratio. The incomplete combustion is thus considered to occur by increasing the mixing ratio of bio oil to rise up the concentration of carbon monoxide.

5.3.3 Analysis of engine performance

The engine performance is examined by analyzing the experimental data. The thrust and the power of exhaust gas can be calculated by

\[ F = \dot{m}(v_e - v_i) \quad \text{(N)} \]  
\[ W = \frac{1}{2} \dot{m}(v_e^2 - v_i^2) \quad \text{(W)} \]  

where \( \dot{m} [\text{kg/s}] \) is mass flow rate of gas, \( v_i [\text{m/s}] \) and \( v_e [\text{m/s}] \) are velocity at inlet and exit, respectively. Since only the engine thrust was obtained by the engine test, quantities in the right hand side of equation (15) needed to be estimated from the measured values.

The mass flow rate of gas is calculated from the fuel consumption and the oxygen concentration in the exhaust gas. Since the incomplete combustion is low, which is suggested by the carbon monoxide level, the calculation assumes complete combustion with approximation of fuel:
tetradecane for kerosene and oleic acid for the BDF. This estimation of mass flow rate of gas enables to obtain the velocity at the nozzle exit by $\dot{m} = \rho v A_e$, and the velocity at the inlet by equation (14). The quantities of right hand side of equation (15) are accordingly known for evaluation of the power of exhaust gas.

The power of exhaust gas thus evaluated is shown in Fig. 25. The power increases exponentially to 8.0 to 10.0 kW as the engine revolution increases. The power differs little by the change of the BDF mixture ratio at the lower revolutions than 120,000 rpm. However, the power is enhanced in the cases of the BDF mixed fuel at the higher revolutions than 120,000 rpm. This enhancement comes from the higher exhaust gas temperature. The LHV of the BDF is lower than kerosene and the fuel consumption is increased in the case of mixed fuel to maintain the same revolution as in the case of kerosene. This increased consumption rate leads to the increase of gas flow rate and exhaust gas power.

The efficiency of the gas turbine is calculated from the exhaust gas power and the input energy as shown in Fig. 26. The latter is obtained from the gas flow rate and the LHV. The thermal efficiency is also enhanced in the case of the BDF-mixed fuel. The trends of the thermal efficiency is similar to those of the exhaust gas power. The enhancement occurs for the engine revolution higher than 120,000 rpm. The enhancement of the efficiency means that the input energy is maintained although the fuel consumption is increased for the BDF-mixed fuel. This essentially comes from the specialty of the bio fuel which include chemically bound oxygen within their molecules. The stoichiometric air fuel ratio is 14.9 for tetradecane and 12.9 for oleic acid. The complete combustion of bio oil needs less air than that of fossil fuel. This can lead to specific changes in higher engine revolution. However, the advantages of the bio oil cannot be judged by the current experiment alone. Further investigation on bio oil for wider conditions is needed.

6. Conclusion

The present paper describes the research work for developing the gas turbine system with enhanced fuel applicability. The conclusions are summarized as follow:

(1) The field research was conducted in an Indian remote village to know the energy usage in Asian rural areas. The bio fuels such as wood and farm wastes are continuously used in Indian villages. There is an increasing number of bio-gas reciprocate engine generator in rural villages. Therefore, there are much chances to introduce the biomass gas turbine in Indian villages.

(2) The thermodynamic analysis is made for the mixed type gas turbine cycle. The non-dimensional expression for the net work and the heat are deduced to make their diagrams against pressure ratio. The mixed type gas turbine shows the medium thermal efficiency between the external fired (EF) and the regenerated cycle. The mixed cycle is inferior to the regenerated cycles in the thermal efficiency. However, the mixed cycle is superior to the EF cycle and the standard cycle without regeneration heat exchanger.

(3) The gas turbine with jet nozzle is used for the experimental test on bio-oil mixed kerosene. The kinematic viscosity and the density of the BDF and Jatropha mixed kerosene are measured. The LHV of them are calculated from the density. The addition of kerosene to the BDF or Jatropha reduces the kinematic viscosity exponentially.

(4) The experimental test on the gas turbine with jet nozzle revealed that the gas turbine engine can work using the mixed fuel with the mixture ratio up to 60%. The carbon monoxide level is increased by raising the mixture ratio. However, the engine thrust is maintained.
at normal values for the case of the mixed fuel.

The analysis is made for the exhaust gas power of the gas turbine experiment. The complete combustion is assumed to calculate the gas flow rate for the evaluation of the gas power. It was shown that the power increases 8.0 kW to 10.0 kW with escalating engine revolution. There is growing effects by the addition of bio-oil in the high revolution. However, the power is not affected by the bio-oil addition at least in as low engine revolution as 120,000 rpm.

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