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Effects of Turbine Inlet Temperature on Performance of Regenerative Gas Turbine System with Afterfogging

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Abstract

Afterfogging of the regenerative gas turbine system has an advantage over inlet fogging in that the high outlet temperature of air compressor makes the injection of more water and the recuperation of more exhaust heat possible. This study investigates the effects of turbine inlet temperature (TIT) on the performance of regenerative gas turbine system with afterfogging through a thermodynamic analysis model. For the standard ambient conditions and the water injection ratios up to 5%, the variation of system performance including the thermal efficiency is numerically analyzed with respect to the variations of TIT and pressure ratio. It is also analyzed how the maximum thermal efficiency, net specific work, and pressure ratio itself change with TIT at the peak points of thermal efficiency curve. All of these are found to increase almost linearly with the increases of both TIT and water injection ratio.

Key words: Afterfogging, Gas turbine, Regeneration, Turbine inlet temperature (TIT), Water injection

Nomenclature

c_p : specific heat [kJ/kg]
f_w : water injection ratio [kg/kg dry air]
h : enthalpy [kJ/kg dry air]
h_f^0 : enthalpy of formation [kJ/kg]
P : pressure [kPa]
r_p : pressure ratio
R_m : gas constant of gas mixture [kJ/kgK]
RH : relative humidity [%]
s^0 : entropy function [kJ/kgK]
T : temperature [K, ℃]
TIT : turbine inlet temperature [℃]
v : specific volume of gas [m^3/kg]
w_{net} : net specific work [kJ/kg]
w_{ac} : specific work of air compressor [kJ/kg]
w_{fc} : specific work of fuel compressor [kJ/kg]
w_t : specific work of turbine [kJ/kg]

Greeks

α : mass fraction [kg/kg]
ε_r : effectiveness of regenerator
ε_t : temperature efficiency of regenerator
ε_{th} : thermal efficiency
η_c : polytropic efficiency of compressor
η_t : polytropic efficiency of turbine

Superscripts

c : colder fluid side of regenerator
h : hotter fluid side of regenerator
i : i th-component gas
p : products of combustion
r : reactants of combustion

Subscripts

1 : ambient air
in : inlet
m : gas mixture
out : outlet

1. Introduction

In power generation system it is very important to improve efficiency of power generation and specific power. Reduction of the amount of pollutant dis-

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charge and installation or operation cost is also important. The gas turbine system meets such requirements well since it features a relatively high efficiency of power generation with low unit cost, production of high power per unit facility weight, and a small amount of pollutant discharge. This has resulted in the global trend that more than half of power plants constructed all over the world since the late 1990s adopted the gas turbine system using natural gas for fuel[1].

The wet gas turbine system that sprays water or steam can offer a high efficiency of power generation and a high specific power with a relatively low cost compared to high efficiency combined-cycle gas turbine system. For the wet gas turbine system, there are various methods in use including the inlet fogging method of spraying water to inlet of air compressor to cool air[2], the wet compression method of spraying water inside compressor [3-6], the STIG (Steam Injection Gas Turbine) method of generating steam by the recuperation of exhaust heat and spraying it to combustor, and the evaporation cooling (EvGT: Evaporative Gas Turbine or HAT: Humid Air Turbine) method of using spray tower[7-8].

The drawback of the water spraying method at a compressor inlet is that the amount of water spray is limited to 1~2% of air mass so that there is a limitation to improving power output. This is because corrosion or other problems may be caused if too much water is sprayed and the unevaporated water droplets flow into the compressor. Since temperature is high at the compressor outlet, the water spraying method at the compressor outlet can allow spraying of more water than the method of spraying water at the compressor inlet so that a relatively higher evaporation effect can be expected. In regard to the regenerative gas turbine system with afterfogging, Nishida et al.[8] investigated the effects of pressure ratio on thermal efficiency for the specific values of water injection ratio and ambient temperature. Kim et al.[9] carried out the research on the regenerative gas turbine system with afterfogging to analyze a fundamental system performance when the TIT (Turbine Inlet Temperature) was 1200℃.

The TIT has a significant influence on performance of general gas turbine system. In this study, we will use the thermodynamic analysis model of regenerative gas turbine system that was developed in the previous study [9] to examine the effects of the TIT on performance of the regenerative gas turbine system with afterfogging. First, under the standard ambient conditions, we will examine the relation between pressure ratio and system performance such as thermal efficiency and fuel consumption ratio when up to 5% of water is sprayed. Next, we will examine the change in pressure ratio that makes thermal efficiency the highest with respect to the TIT and water injection ratio. Under such circumstance, we will also examine the changes in thermal efficiency and specific power.

2. System modeling

Fig. 1 shows the schematic diagram of the system that will be analyzed in this study. The air for combustion coming from the atmosphere is pressurized in the compressor (1→2), mixed with water in the mixing chamber (2→3) and goes into the regenerator (3 →4, 6→7). The moisture air, heated by exchanging heat with exhaust gas, enters the combustor (4+9→5) along with (gas) fuel pressurized in the fuel compressor (8→9) before being burned, which is later provided to the turbine (5→6) at high temperature and pressure. After expansion and generating power in the turbine, the gas mixture passes through the regenerator and are finally released to the atmosphere.

The main assumptions used in this study are as follows.

(1) All gases are ideal gases. One mole of dry air at a compressor inlet is composed of 0.2095 mole of O_2, 0.7902 mole of N_2 and 0.0003 mole of CO_2.

(2) There is no turbine cooling. And pressure loss of the system is neglected.

(3) Combustion takes place in an adiabatic complete combustion process. And methane (CH_4) is used for fuel.

(4) The polytropic efficiencies of compressor and turbine are constant.

(5) The effectiveness of regenerative heat exchanger is constant.

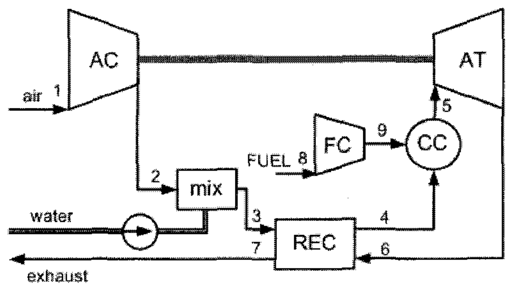

Fig. 1. Schematic diagram of the system.

Five kinds of gases such as O_2, N_2, CO_2, CH_4, and $H_2O(g)$, and water in liquid phase are used as a working fluid for gas turbine cycle considered in this study. These will be referred to by the subscripts from 1 to 6 in order of appearance. In the analysis of the system, all properties are based on 1 kg of dry air at the air compressor inlet. Enthalpy h and entropy function s^0 can be calculated as follows.

$$h(T,\alpha)=\sum_{i=1}^{6}\alpha^i h^i(T) \tag{1}$$

$$s^0(T,\alpha)=\sum_{i=1}^{6}\alpha^i\int_{T_0}^{T}\frac{c_p^i(T,\alpha^i)}{T}dT \tag{2}$$

In the equations above, subscript i indicates the aforementioned six components. α^i, h^i and c_p^i are mass fraction, enthalpy and specific heat at constant pressure for each component per kg of dry air.

Air enters the compressor under the conditions of temperature, pressure and relative humidity of T_1, P_1 and RH_1, respectively. If the pressure ratio of the compressor is r_p, the pressure at the compressor outlet is $r_p P_1$. Performance of the compressors can be reflected in the system analysis[3-4] by using the polytropic efficiency η_c defined by

$$dh=vdP/\eta_c \tag{3}$$

Since the conditions at the compressor inlet are known, temperature at the outlet can be determined implicitly from the following equation[6].

$$s_{out}^0-s_{in}^0=\alpha_m R_m \ln(r_p)/\eta_c \tag{4}$$

Here, α_m and R_m are mass fraction and gas constant of gas mixture. The required specific works of air compressor, w_{ac}, and fuel compressor, w_{fc}, can be calculated as follows.

$$w_{ac}=h_2-h_1 \tag{5}$$
$$w_{fc}=h_9-h_8 \tag{6}$$

When the polytropic efficiency of the turbine, η_t, defined by

$$dh=\eta_t vdP \tag{7}$$

is contant and the conditions at the turbine inlet are known, turbine outlet temperature can be calculated such that the following equation is satisfied[9].

$$s_5^0-s_6^0=\eta_t\alpha_m R_m\ln(r_p) \tag{8}$$

The specific work of turbine w_t and the net specific work of system w_{net} can be calculated as

$$w_t=h_5-h_6 \tag{9}$$
$$w_{net}=w_t-(w_{ac}+w_{fc}) \tag{10}$$

The combustion of methane in the combustion camber takes place as a complete combustion process with the following equation of reaction.

$$CH_4+2O_2\rightarrow CO_2+2H_2O \tag{11}$$

The assumption of adiabatic complete combustion process and the consideration of the enthalpy of formation make possible the determination of fuel consumption ratio which gives a combustor outlet temperature equal to desired TIT. The equation to be satisfied is[10]

$$\sum_{i=1}^{5}\alpha^{r,i}\left[h^i(T^r)-h^i(T_{ref})+h_f^{0,i}\right]$$
$$=\sum_{i=1}^{5}\alpha^{p,i}\left[h^i(T^p)-h^i(T_{ref})+h_f^{0,i}\right] \tag{12}$$

where $h_f^{0,i}$ is the enthalpy of formation for the five kinds of gases already introduced. Reference temperature T_{ref} is 298.15 K. Heat provided to the system by fuel, q_{in}, and the thermal efficiency of the system, ε_{th}, can be calculated as follows.

$$q_{in}=h_5-h_4-h_9 \tag{13}$$
$$\varepsilon_{th}=w_{net}/q_{in} \tag{14}$$

In the mixing chamber, f_w kg of water per kg of dry air is sprayed to and mixed with moisture air which leaves the air compressor. The energy balance equation for the mixing process is

$$h_2+h_w=h_3 \tag{15}$$

where h_w is the specific enthalpy of water being sprayed. If the water injection ratio f_w is high, the partial pressure of water vapor reaches to saturation pressure so that unevaporated water droplets along with gas mixture may flow into the regenerator.

The energy balance in the regenerator is written as follows.

$$h_4 - h_3 = h_6 - h_7 \tag{16}$$

The effectiveness of the regenerator is the ratio of heat exchange amount to the maximum heat transfer when heat exchange area increases without limit. Therefore it may be defined as[11]

$$\varepsilon_r = \frac{q}{q_{max}} = \frac{q}{\min\left\{\begin{array}{l} h(T_{in}^h, \alpha^h) - h(T_{in}^c, \alpha^h), \\ h(T_{in}^h, \alpha^c) - h(T_{in}^c, \alpha^c) \end{array}\right\}} \tag{17}$$

In the analysis of regenerative gas turbine system, it is often the case that instead of effectiveness of heat exchanger, its temperature efficiency ε_t which is defined by

$$\varepsilon_t = (T_{in}^h - T_{out}^h)/(T_{in}^h - T_{in}^c) \tag{18}$$

is assumed to be constant[6]. However, when the mass flow rate of water spray is getting higher or in particular, when water droplets flow into the heat exchanger, the difference between temperature efficiency and effectiveness becomes larger. For this reason a model that the effectiveness of the regenerator is kept constant is used in this study.

3. Results and discussions

Table 1 shows the values of the main system parameters used in this study. Excluding the TIT and the water injection ratio (f_w), all of the conditions are assumed to be constant as shown.

After solving the system of simultaneous equations that were derived in the previous section, all

Table 1. Calculation conditions for the system.

Symbol	Parameter	Value
T_1	ambient temperature	15℃
P_1	ambient pressure	101.3kPa
RH_1	relative humidity	60%
η_c	polytropic efficiency of compressors	80%
η_t	polytropic efficiency of turbine	80%
ε_r	effectiveness of regenerator	83%
	fuel	CH_4

the properties can be determined at each position of the gas turbine system. Table 2 shows a numerical calculation result for a certain case, where major operation conditions are also specified. Mass fractions of O_2, N_2, CO_2, and CH_4 change only at the combustor. The mass fraction of water vapor starts with 1.64%, which corresponds to RH_1 = 60%, and increases to 6.64% in the mixing chamber due to water spray (f_w =5%). The jump of temperature in the compressors, drop of temperature in the turbine, cooling due to water spray, and the changes of temperature in the colder and hotter sides of the regenerator can be easily recognized from the table. It is also possible to figure out the fuel consumption ratio that is necessary to obtain the preset value of the TIT. Since the pressure loss of the system is neglected, the pressure ratio of the high pressure part to the low pressure part is kept constant at 5.

If a large amount of water is sprayed at the compressor outlet, water does not flow along with air. Instead, it is condensed and flows down the wall of duct or in the regenerator, which causes corrosion. As a result, it is a common practice to spray water within

Table 2. Representative properties of the system at each position.

position	O_2 (%)	N_2 (%)	CO_2 (%)	CH_4 (%)	$H_2O(g)$ (%)	$H_2O(l)$ (%)	T (°C)	P (kPa)	h (kJ/kg)	s (kJ/kgK)
1	23.23	76.72	0.05	0.00	1.64	0.00	15.00	101.33	294.07	6.87
2	23.23	76.72	0.05	0.00	1.64	0.00	235.04	506.63	524.69	6.99
3	23.23	76.72	0.05	0.00	6.64	0.00	109.05	506.63	427.60	7.20
4	23.23	76.72	0.05	0.00	6.64	0.00	574.72	506.63	987.39	8.16
5	18.47	76.72	3.32	0.00	6.64	0.00	1000.00	506.63	1597.44	8.87
6	18.47	76.72	3.32	0.00	9.32	0.00	665.02	101.33	1137.65	8.98
7	18.47	76.72	3.32	0.00	9.32	0.00	224.36	101.33	577.86	8.18
8	0.00	0.00	0.00	1.19	0.00	0.00	15.00	101.33	5.88	0.14
9	0.00	0.00	0.00	1.19	0.00	0.00	168.89	506.63	10.38	0.14

(T_1 = 15℃, RH_1 = 60%, $\eta_c = \eta_t = 0.80$, $\varepsilon_r = 0.83$, TIT = 1000℃, $r_p = 5$, $f_w = 5\%$)

an amount that does not cause water to be condensed. In this study, the water injection ratio was considered to be up to 5% for the performance analysis.

Figs. 2-4 show changes in thermal efficiency, fuel consumption ratio and specific power of the system according to pressure ratio (r_p), TIT, and water injection ratio (f_w). The ambient temperature and relative humidity were fixed at 15 and 60%, respectively. As shown in Fig. 2, the thermal efficiency can be raised by the spray of water at the compressor outlet (afterfogging). The reason is that the sprayed water leads to a decrease in the regenerator inlet temperature, enhancing heat exchange due to the increased temperature difference in the regenerator. Although the water spray causes an increase of mass flow rate and the accompanying increase in the fuel consumption ratio (energy input), the increased mass flow rate can make much more turbine work be generated at the same time. The increase in thermal efficiency is possible when the latter effect is dominant over the former. The numerical results for TIT=1200℃ are in good agreement with those of Nishida et al[8].

Fig. 3 shows the fuel consumption ratio with and without water spray, as a function of pressure ratio and TIT. As shown in the figure, the fuel consumption ratio has a monotone increase as the pressure ratio increases. The reason is that as the pressure ratio goes up, the temperature at the compressor outlet increases, the turbine outlet temperature decreases, and the temperature difference in the heat exchanger diminishes. For the fixed temperature at the turbine inlet, the temperature of air flowing into the combustor goes down as the pressure ratio increases, which requires more fuel to be provided. The fuel consumption ratio increases with the TIT. This seems natural because more fuel is required to increase the equilibrium temperature at the combustor outlet. In the meantime, as water spray increases, fuel consumption is also on the increase. This is attributable mainly to the increase in mass flow rate due to water spray. However, in the low pressure ratio zone in the figure, if the water injection ratio is too high, the percentage of water that remains in the liquid state becomes too high at the regenerative inlet where water and air are mixed. As a result, it is possible that the heat capacity at the colder side of the regenerator could be higher than that of the hotter side or exhaust gas side. If this happens, temperature change in the colder side is smaller than that of the hotter side, and the dependence of thermal efficiency on pressure ratio shows a different tendency.

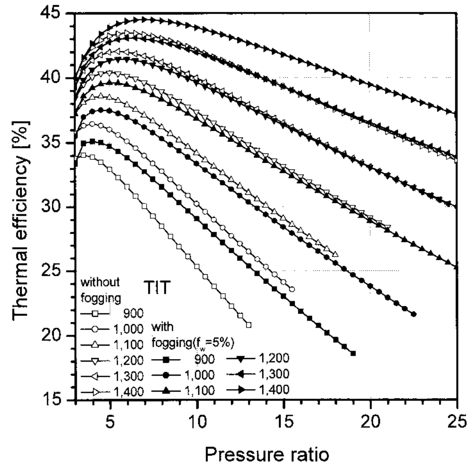

Fig. 2. Thermal efficiency as a function of pressure ratio for various TIT and water injection ratios.

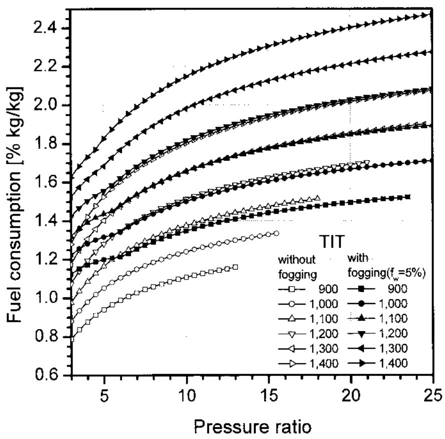

Fig. 3. Fuel consumption as a function of pressure ratio for various TIT and water injection ratios.

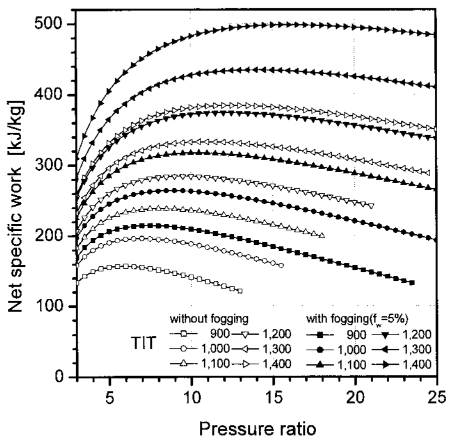

Fig. 4. Net specific work as a function of pressure ratio for various TIT and water injection ratios.

The dependence of net specific work (w_{net}), that is, the net power per unit mass flow rate of dry air on the pressure ratio and the TIT in the cases of with and without water spray is shown in Fig. 4. Regardless of the TIT and water injection ratio, the net specific work increases as the pressure ratio is on the increase. And it reaches a maximum value before decreasing. The pressure ratio at which the net specific work becomes maximum increases with the increases of the TIT and f_w just as the case with thermal efficiency. The pressure ratio giving the peak value of net specific work is higher than that of thermal efficiency. When compared with the thermal efficiency curve in Fig. 2, the slope of the net specific work curve is less steep. This is because the amount of fuel consumption also increases when the pressure ratio goes up. The spray of water leads to an improvement of net specific work. As discussed above this trend is attributable to the increase of flow rate in the turbine due to water spray. For example, when the TIT is 1000, the maximum value of the net specific work increases by approximately 21%. It is 196 kJ/kg when f_w is 0%, while it is 238 kJ/kg when f_w is 5%.

As already examined in Fig. 2 there exists a pressure ratio that makes thermal efficiency of gas turbine system the highest, which is dependent on the water injection ratio and the TIT. The remaining part of this study will be focused on the examination of the system characteristics for the pressure ratios where thermal efficiency reaches its local maximum value. The results are given in Figs. 5-8. The features of pressure ratio that makes thermal efficiency the highest, maximum thermal efficiency, and net specific work for maximum thermal efficiency are represented as functions of the TIT in Figs. 5-7, respectively, while Fig. 8 represents the system characteristic curve of net specific work vs. maximum thermal efficiency.

Fig. 5 shows the change in pressure ratio for maximum thermal efficiency according to the change in the TIT, for the values of f_w from 0 to 5%. The pressure ratio increases as the TIT and f_w increase. With respect to the same value of f_w, as the TIT increases by 100℃, the pressure ratio for maximum thermal efficiency goes up by about 0.56. With respect to the same value of the TIT, as the value of f_w increases by 1%, the pressure ratio for maximum thermal efficiency goes up by about 0.15.

Fig. 6 shows the dependence of maximum thermal efficiency on the TIT and f_w. As can be seen in the figure, the maximum thermal efficiency has an almost

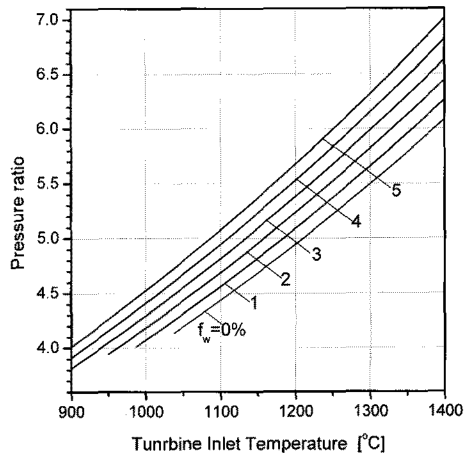

Fig. 5. Pressure ratio for maximum thermal efficiency as a function of TIT for various water injection ratios.

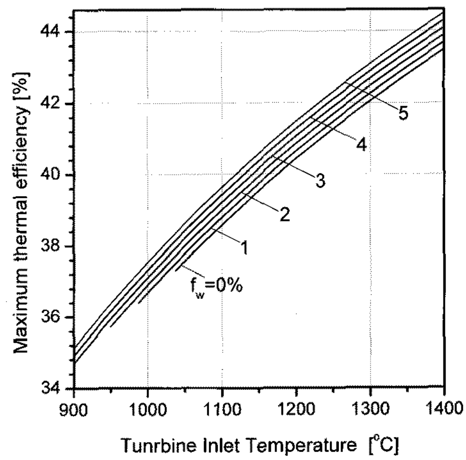

Fig. 6. Maximum thermal efficiency as a function of TIT for various water injection ratios.

linear increase with respect to the TIT and f_w. The thermal efficiency increases by 1.9% as the TIT is raised by 100℃ and rises by about 0.2% as the value of f_w goes up by 1%.

Fig. 7 shows change in net specific work fo the system according to changes in the TIT and f_w when it is operated at the pressure ratio for maximum thermal efficiency. In the case of pressure ratio that ensures maximum thermal efficiency, the net specific work also shows an almost linear increase against the TIT. With respect to the same value of f_w, as the TIT goes up by 100℃, the net specific work increases by around 45 kJ/kg. With respect to the same TIT, as the value of f_w goes up by 1%, the net specific work increases by around 10.5 kJ/kg.

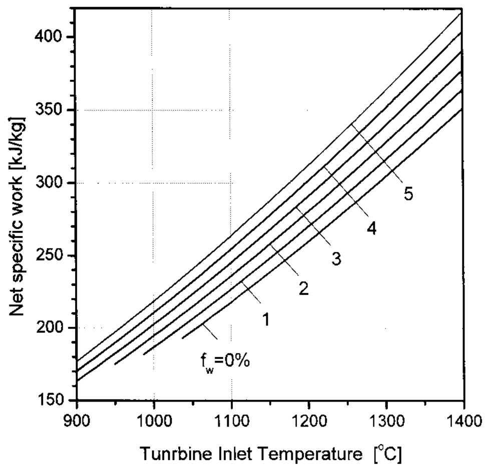

Fig. 7. Net specific work for maximum thermal efficiency as a function of TIT for various water injection ratios.

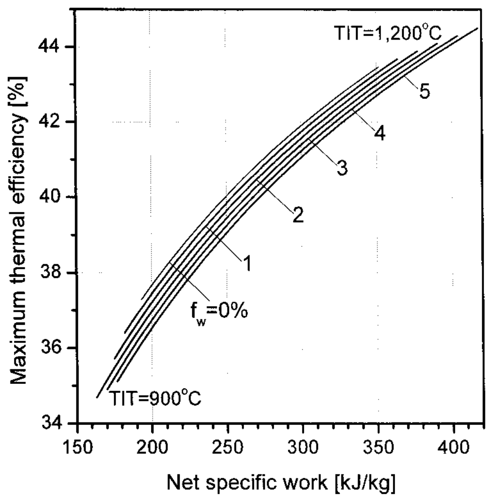

Fig. 8. Maximum thermal efficiency as a function of net specific work for various TIT and water injection ratios.

Fig. 8 is the characteristic curve of net specific work vs. thermal efficiency when the gas turbine system has the maximum values of thermal efficiency. As shown in the figure, the thermal efficiency and net specific work of the system increase as the TIT goes up, if other operation conditions are not changed. And as the water injection ratio goes up, the net specific work increases with the same efficiency. In other words, when the TIT or the mass flow rate of water spray increases, the curve moves to the right and upward. This visually shows that as the turbine inlet temperature or the water injection ratio at the compressor outlet is increased, the enhancement of the net specific work and thermal efficiency of the gas turbine system is expected.

In this study, we analyzed the effects of the turbine inlet temperature and the water injection ratio at the compressor outlet on the performance of the regenerative gas turbine system under the typical operation conditions and without turbine cooling. Micro-turbine is usually operated at the relative low TIT's of 1000 or lower. When the TIT is low, turbine cooling is not required. But if the TIT increases to a higher temperature, the effect of turbine cooling might be important and should be included in the analysis of gas turbine system[12].

4. Conclusions

In this study, the effects of the turbine inlet temperature on the performance of regenerative gas turbine system with afterfogging are investigated. And the pressure ratio at which the thermal efficiency of the system reaches maximum value is also investigated.

The main results of the investigation for the typical operation conditions may be summarized as follows:

(1) Thermal efficiency and net specific work increase with respect to the increase of turbine inlet temperature or water injection ratio and have the maximum values at some pressure ratios. The pressure ratio giving the peak value of net specific work is higher than that of thermal efficiency.

(2) Fuel consumption ratio increases with the increase of the turbine inlet temperature, water injection ratio, and pressure ratio.

(3) Pressure ratio for the maximum thermal efficiency increases by about 0.56 as the TIT goes up by 100℃. And it increases by about 0.15 as the value of f_w goes up by 1%. The maximum thermal efficiency increases by about 1.9% according to the 100℃ increase of the TIT. And it increases by around 0.2% according to the 1% increase of the f_w value.

(4) Under the conditions of pressure ratio that make thermal efficiency the highest, net specific work increases by about 45 kJ/kg as the TIT goes up by 100℃. And it increases by about 10.5 kJ/kg as the value of f_w goes up by 1%.

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