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A Study of Performance on Advanced Humid Air Turbine Systems

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ABSTRACT

Comparative studies are presented to determine the effect of the turbine blade cooling method on the Advanced Humid Air Turbine (AHAT) system performance. In AHAT systems, two kinds of fluids can be used as a coolant: air extracted from the compressor and humid air. Three types of turbine blade cooling systems are evaluated using the in-house heat cycle simulator. As a result, the thermal efficiency is improved 1.7 percentage points by using the humid air cooling method.

The possibility of remodeling a typical mid-size simple cycle gas turbine for use in AHAT systems is examined. A remodeled AHAT system, which can be realized by adjusting the compressor intake flow rate, has 1.3 percentage points higher thermal efficiency than the combined cycle system using the same core gas turbine.

NOMENCLATURE

*n*N, *n*R	the *n*-th stage turbine nozzle, rotor blade
AHAT	advanced humid air turbine
G	mass flow rate
LHV	lower heating value
P	pressure
PR	pressure ratio
T	temperature
TET	turbine entry temperature (ISO)
WAC	water atomization cooling
W_{sp}	specific work
η	efficiency or effectiveness

Subscripts

c	cooling or coolant
g	mainstream gas
th	thermal

INTRODUCTION

Gas turbines play a big role in power generation; they may range from small-scale private power generation sources to large-scale combined cycle plants for electric utilities. In combined cycle plants, which are the most efficient of current technologies for power generation, the turbine inlet temperature is increased to enhance the thermal efficiency. However, gas turbines with a turbine inlet temperature of over 1,500°C are difficult to develop because of extensive increases in coolant consumption and NOx emissions.

This situation has led to promoting improvement of the heat cycle in recent years. Heat cycle using humid air is one change that can be made. Various novel cycles such as Humid Air Turbine (HAT) (Rao, 1991), Cascaded Humidified Advanced Turbine (CHAT) (Nakhamkin, 1995), REVAP (Ruyck et al., 1996), and TOPHAT (van Liere, 1998) have been proposed for highly effective operation by combining the regenerative cycle with humid air.

The concept of the gas turbine system using high humidity air (HAT system) was reported two decades ago. It is thought to be suitable for distributed power supply to a combined cycle because of high efficiency, simplicity of the system, and applicability to co-generation. Many theoretical and practical feasibility studies were made, and a small pilot plant, EvGT in Sweden, has been put tests operation recently (Agren et al., 2000). It seems that developing a new gas turbine with an intercooler would give the highest performance for the HAT cycle.

The AHAT system is a system which uses existing mutual technologies and aims at high performance equal to that of the HAT system (Hatamiya et al., 2000). It is suitable for a typical gas turbine, which is widely used for industrial small-size to large-size machines.

This paper presents comparative calculation results on the AHAT system performance and especially focuses on the influence of different turbine blade cooling systems. All results are evaluated by the in-house heat cycle simulator, which performs mass and energy balancing based on enthalpy calculations. All blade cooling systems are based on data for a middle-size gas turbine. The basic AHAT system has a conventional turbine blade cooling system which uses compressed air extracted from the compressor as coolant. This system is compared with two other AHAT systems. One uses high humidity air as coolant, which is humidified in the saturator, for all turbine blades, and the other uses high humidity air for turbine nozzle blades and compressed air for turbine rotor blades.

Moreover, this paper discusses a remodeling method, which converts a typical gas turbine to the AHAT system. Comparison of performance for typical gas turbines (simple cycle and combined cycle) and the AHAT system are also described.

SYSTEM DESCRIPTION

Figure 1 shows the conceptual plant scheme of AHAT system. The AHAT system is suitable for a typical gas turbine system, which is widely used from industrial small-size machines to large-size machines for power generation. The compressor does not need to be divided for intercooling. The water atomization cooling (WAC) system is set in the air intake duct of the compressor, and fine droplets of water are sprayed there. Some of the water droplets evaporate at the compressor entrance and cool the intake air. The remainder evaporate while being compressed in the compressor, and they suppress the rise of the air temperature. The compressed air comes in contact directly with hot water in the saturator and it becomes humid air with 100% relative humidity. As a result, the

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flow rate and the specific heat of the working fluid increase, and the turbine generation power increases. High humidity air flows into the combustion chamber after it is pre-heated by the turbine exhaust gas with the recuperator. The high temperature combustion gas is discharged, after the turbine is driven and the recuperator and economizer collect the exhaust heat. The exhaust gas that exits the economizer contacts directly with cold water in the water recovery system to be cooled and the moisture is condensed. The exhaust gas that exited the water recovery system is discharged into the atmosphere from the stack after it is heated with the exhaust gas heater.

On the other hand, the air cooler and economizer collect heat from the hot gas and supply warm water. A part of the warm water evaporates by coming in contact directly with compressed air of the saturator. The temperature of the remaining warm water has dropped because it is deprived of the latent heat in the evaporation process when it was taken out of the bottom of the saturator. Finally the water is sent to the air cooler and economizer again, and used as a heat recovery medium by the high temperature gas.

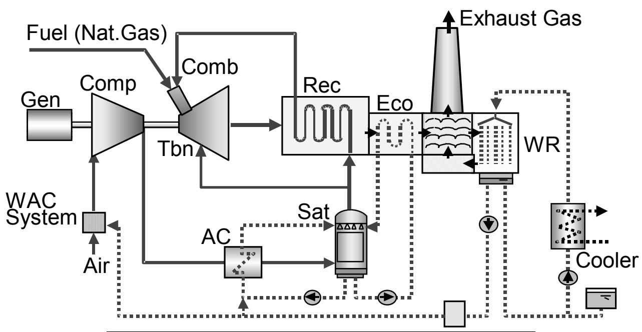
Figure 2 shows the concept of the WAC system. The spray nozzle which provides atomization of the water (Sauter Mean Diameter: 10 microns or less) at ambient temperature is arranged in the air intake duct that is located in front of the compressor inlet. Some of the droplets evaporate in the air intake duct, and the air becomes humidified air of 100% relative humidity before it reaches the compressor inlet. This process and the density of air increase provide humidified cooling and the compressor intake air flow rate increases. Fine water droplets in the air evaporate in the compressor. As a result, the compression work decreases, shifting from adiabatic compression to isothermal compression. This shift is promoted as the atomization flow rate increases.

In the AHAT system two fluids are considered as coolant for gas turbine hot parts. One is air extracted from the compressor which generates the compressed air the same as in the typical gas turbine system for combustion, and the other is high humidity air after the saturator. Three types of turbine blade cooling systems, which are comparatively studied here, are illustrated in Fig.3: (a) extracted air cooling, (b) hybrid cooling and (c) humid air cooling.

Figure 3(a) shows the extracted air cooling system. All turbine blades are cooled by compressed air extracted from the compressor. The coolant piping is the same as in the typical gas turbine. That is, coolant for the nozzle blades is extracted from slits facing the outer wall of the compressor gas path and it is fed to nozzle blades through external piping linked to the turbine casing. Coolant for the rotor blades is extracted from the slits facing the inner wall of the compressor gas path and it is fed to rotor blades through the path inside the rotor.

Figure 3(b) shows the hybrid cooling system. The turbine rotor blades are cooled by compressed air extracted from the compressor the same as in Fig. 3(a). However, the turbine nozzle blades are cooled by high humidity air which is humidified in the saturator. This system can be realizable by remodeling the coolant piping for the nozzle blades.

Figure 3(c) shows the humid air cooling system. All turbine blades are cooled by high humidity air which is humidified in the saturator. In this system since the coolant for rotor blades is also humid air it is necessary to provide a supply means from a still system to a rotation system. Thus, the rotor structure differs from that of Fig. 3(a) or Fig. 3(b).



Symbols			
Comp	Compressor	AC	Air Cooler
Gen	Generator	Eco	Economizer
Tbn	Turbine	Rec	Recuperator
Comb	Combustor	Sat	Saturator
		WR	Water Recovery

Fig.1 Conceptual Plant Scheme of AHAT System

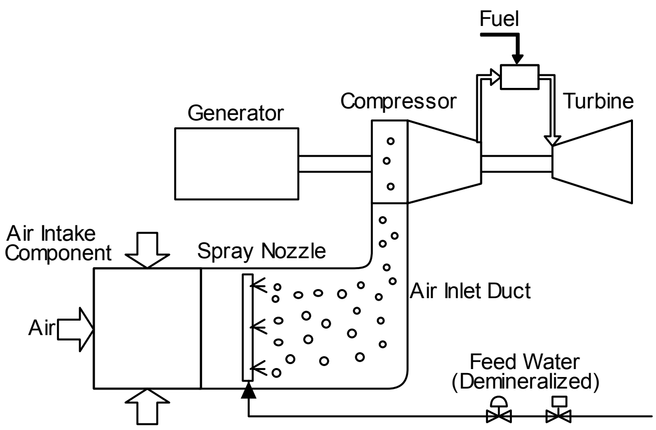
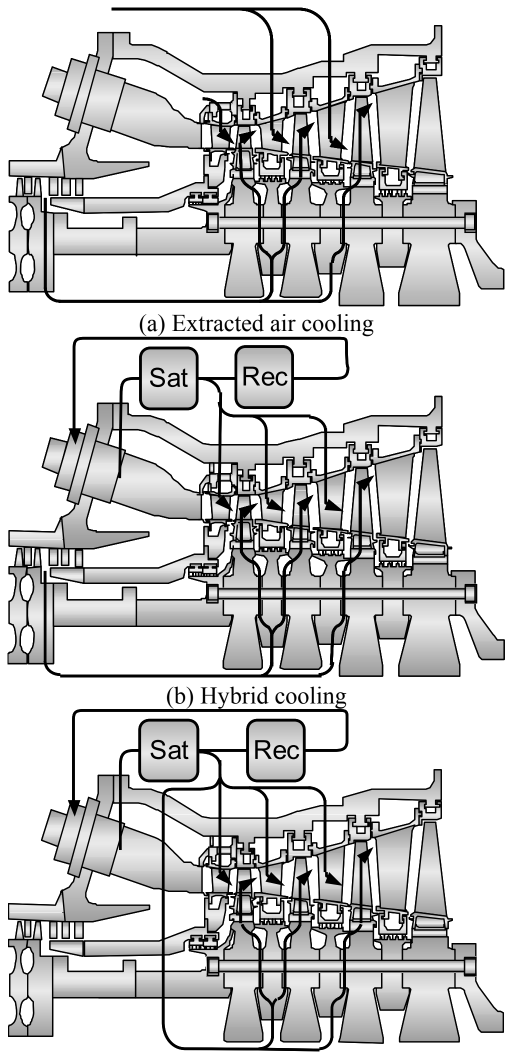


Fig.2 Water Atomization Cooling (WAC) System



(a) Extracted air cooling

(b) Hybrid cooling

(c) Humid air cooling

Fig.3 Turbine Blade Cooling Systems

ANALYTIC METHOD

The in-house heat cycle simulator evaluates the performance of each system. It performs mass and energy balancing based on enthalpy calculations. The thermophysical properties of fluids are obtained by JSME (1983, 1999). The steam tables are based on IAPWS-IF97.

In many cycle analyses, turbine efficiency and coolant consumption were held constant for simplicity, but in fact the thermal efficiency of the gas turbine strongly influences the overall plant thermal efficiency. Thus in this study, the gas turbine performance is calculated in detail; the row-by-row turbine design code considering aerodynamic analysis for the turbine performance, thermal analysis for the consumption of coolant flow and reliability analysis for stress.

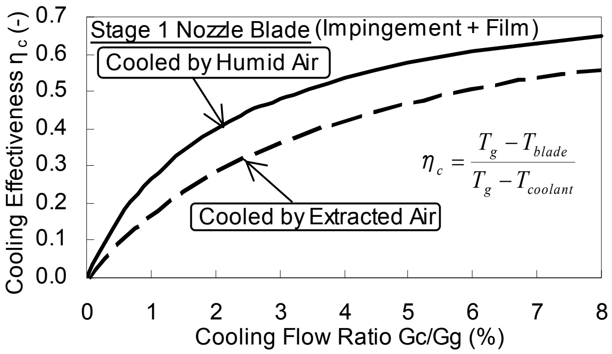
Turbine aerodynamic analysis

The aerodynamic performance of turbine cascades is calculated using the Ainley/ Mathieson/ Dunham/ Came and Kacker/ Okapuu (AMDC+KO) loss prediction method (Kacker and Okapuu, 1981). Additional total pressure and dilution losses due to mixing of the coolant flow with the mainstream are obtained from one-dimensional analyses for conservation of mass, momentum, and energy for each cooling and sealing flow ejection.

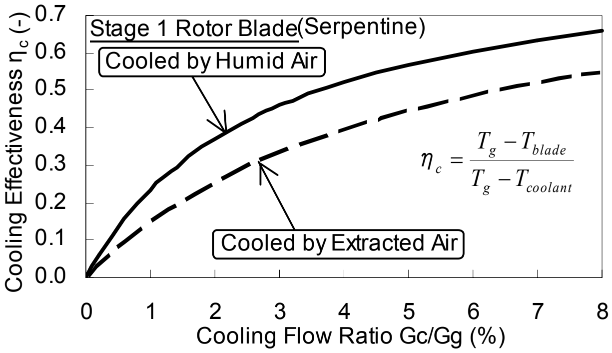
Turbine thermal analysis

The consumption of the coolant influences the gas turbine performance. Thermal analysis predicts the required coolant flow rate. The coolant flow rate is calculated from the given peak gas temperature and the maximum blade temperature, assuming the cooling effectiveness curve for the cooled blade, which is defined based on in-house experimental results. The peak gas temperature for the blades is calculated from the combustor traverse number which gives the temperature difference between the maximum and average gas temperatures at the combustor exit.

The cooling effectiveness depends on the cooling scheme, the states of the flow field and the properties of both the mainstream gas and the coolant. The blade cooling effectiveness curves shown in Fig. 4 are defined using two assumptions: advanced cooling technologies such as V-shaped staggered turbulence promoter ribs and serpentine cooling passages are used; and cooling effectiveness for the high humidity air is obtained from the preliminary cooling design with the condition that it is about 20% higher than that for the compressed air extracted from the compressor due to the gas properties such as specific heat, gas constant, and thermal conductivity. Figure 4(a) shows the cooling effectiveness curves for the first stage nozzle blade, which is made of single crystal superalloy and is cooled by impingement cooling in combination with film cooling system. Figure 4(b) shows the cooling effectiveness curves for the first stage rotor blade. It is also made of single crystal superalloy and cooled by the serpentine passage cooling system employing V-shaped staggered turbulence promoter ribs. The solid lines in these graphs show the effectiveness curves for high humidity air as the coolant and the dashed lines show them for compressed air. The effectiveness curves for the other blades are also defined in the same way.



(a) Nozzle Blade



(b) Rotor Blade

Fig.4 Cooling Effectiveness Curves

Compressor work analysis

To predict compressor work, the evaporation profile within the compressor gas path has to be calculated. This profile is calculated by using the CFD code. Details of the technique were reported by Utamura et al. (1999). Figure 5 shows the evaporation profile within the compressor calculated. The simulator employs this profile to predict the compressor performance and the gas conditions.

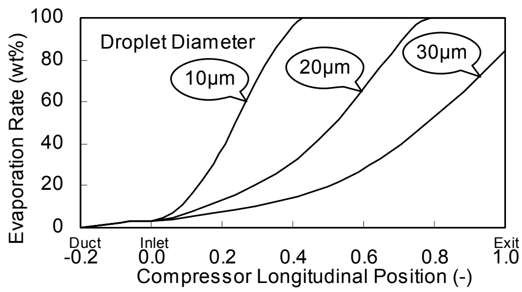


Fig. 5 Accumulated Evaporation Profile within Compressor

RESULTS AND DISCUSSION

Performance of the optimally designed AHAT systems

All results are calculated under the conditions for a mid-size gas turbine shown in Table 1. High grade supperalloy is used for performance in this study. Figure 6 shows carpet diagrams of the thermal efficiency for AHAT systems with different types of turbine blade cooling. Each blade cooling system has a different turbine entry temperature (TET based on ISO) and pressure ratio, the components are optimally designed for performance. That is, the turbine cascades and the compressor cascades are designed to be suitable for each TET, pressure ratio, mass flow rate and etc. The thermal efficiency and specific output increase with TET in all turbine blade cooling systems. On the other hand, specific output increases as the pressure ratio increases. The optimal pressure ratio at which the thermal efficiency is the highest exists in the systems with the same TET. It becomes higher as the TET increases.

It is clear that the humid air cooling system has the highest thermal efficiency and specific output among the three systems and that the gain in thermal efficiency for TET is the highest. The difference in coolant consumption causes these results. Figure 7 shows coolant consumption for each system with TET of 1,350°C and pressure ratio of 20. The consumption of the extracted air cooling system is nearly as twice that of the humid air cooling system. The consumption of the hybrid cooling system is between these two. Extracted air from the compressor has nearly as the same thermophysical property as dry air because the compressor intake air is humidified with only 3.5wt% water by the WAC system.

Table 1 Design Parameters of Main Components

Gas Turbine
Type
・Single-shaft, mid-size industrial
Inlet duct
・Ambient condition = ISO standard condition
・Pressure loss $\Delta P/P = 0.009$
Compressor
・Inlet mass flow (excludes WAC) = 100 kg/s
・Adiabatic efficiency = calculated
・Pressure ratio = 18 - 24 (surveyed)
Combustor
・Fuel = natural gas
・Combustion efficiency = 0.999
・Pressure loss $\Delta P/P = 0.04$
Turbine
・7280 rpm, 4-stage
・TET (ISO) = 1,300 - 1,450°C (surveyed)
・1N = single crystal (SC) 2N, 3N & 4N = conventionally cast (CC)
・1R & 2R = single crystal (SC), unshrouded 3R & 4R = directionally solidified (DS), shrouded
・Adiabatic efficiency = calculated
・Consumption of coolant flow = calculated
・Diffuser pressure recovery factor = 0.75
・Exhaust gas temperature = calculated
Exhaust duct connecting to the recuperator
・Pressure loss $\Delta P/P = 0.003$
Utilities
WAC system
・Flow rate = 3.5 wt% of compressor inlet mass flow
・Droplet diameter = 10μm
Saturator
・$T_{inlet,\ gas} = 105°C$
Recuperator
・Temperature effectiveness = 0.95
・$T_{inlet,\ high} - T_{exit,\ low} = 20°C$
・Pressure loss $(\Delta P/P)_g = 0.022$, $(\Delta P/P)_{exhuast} = 0.035$
Economizer
・Texit, water = 160°C
・Texit, gas = 110°C
・Pressure loss $(\Delta P/P)_{exhaust} = 0.003$

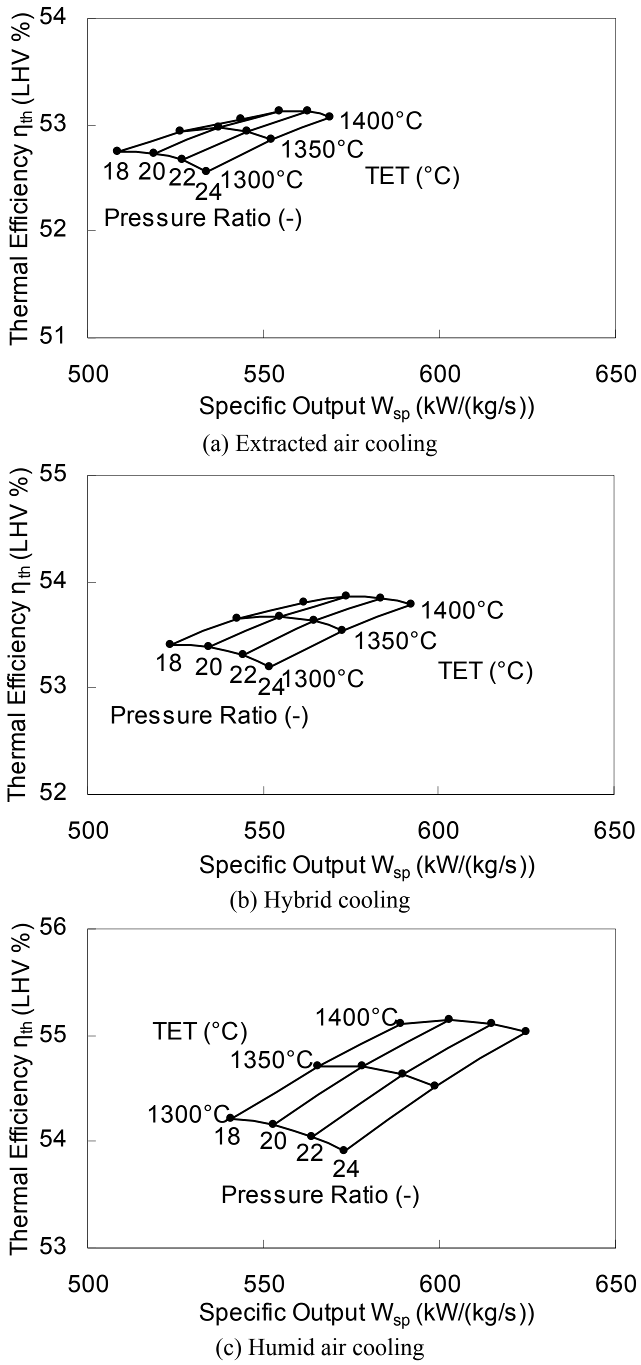
However, in any systems the mainstream gas is totally humidified with about 20wt% water by the WAC system and the saturator. Compared with the humid air cooling method, the extracted air cooling method requires a larger coolant flow rate because extracted air has lower heat transfer coefficient than humid air and the temperature of the extracted air is higher than that of the humid air. This fact means that the gain in thermal efficiency in the extracted air cooling system tends to level off at high TET. Figure 8 shows mass and heat balance diagram.

<u>Remodeling the simple cycle gas turbine to the AHAT system</u>

Consideration is given to whether the gas turbine, which was designed for a typical simple cycle system, can be remodeled to the gas turbine for the AHAT systems. If that can be realized, cost and time of development for the gas turbine suited to the AHAT systems are saved. The performance and reliability need to be compared for the optimally designed systems.

Table 2 shows the remodeling method which converts a typical gas turbine to one for the AHAT system. The balance of mass flow rate between the turbine and the compressor in the AHAT systems differs from that in a typical gas turbine due to the humidification, which is amount of about 20wt% of the compressor intake flow rate. Thus either the turbine or the compressor has to be modified for the mass flow rate. When manufacturing cost is taken into account, in general, the turbine is more expensive than the compressor. From the viewpoint of cost as much of the present turbine as possible should be utilized.

Since the turbine cascades have already been decided, the pressure ratio will be decided by the mainstream gas flow rate based on the turbine characteristics (nondimensional mass flow rate



(a) Extracted air cooling

(b) Hybrid cooling

(c) Humid air cooling

Fig.6 Performance of Optimally Designed AHAT Systems

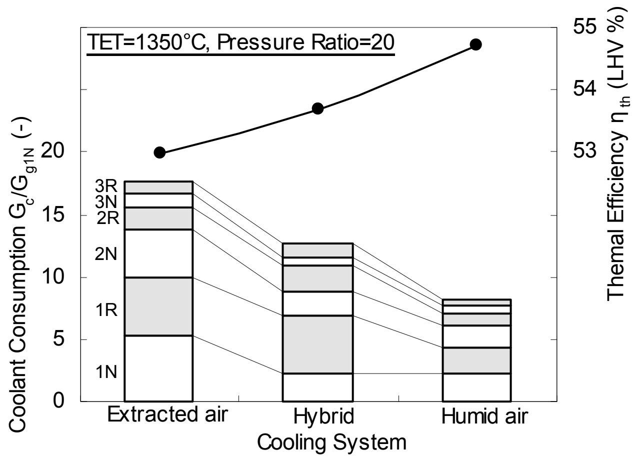


Fig. 7 Coolant Consumption

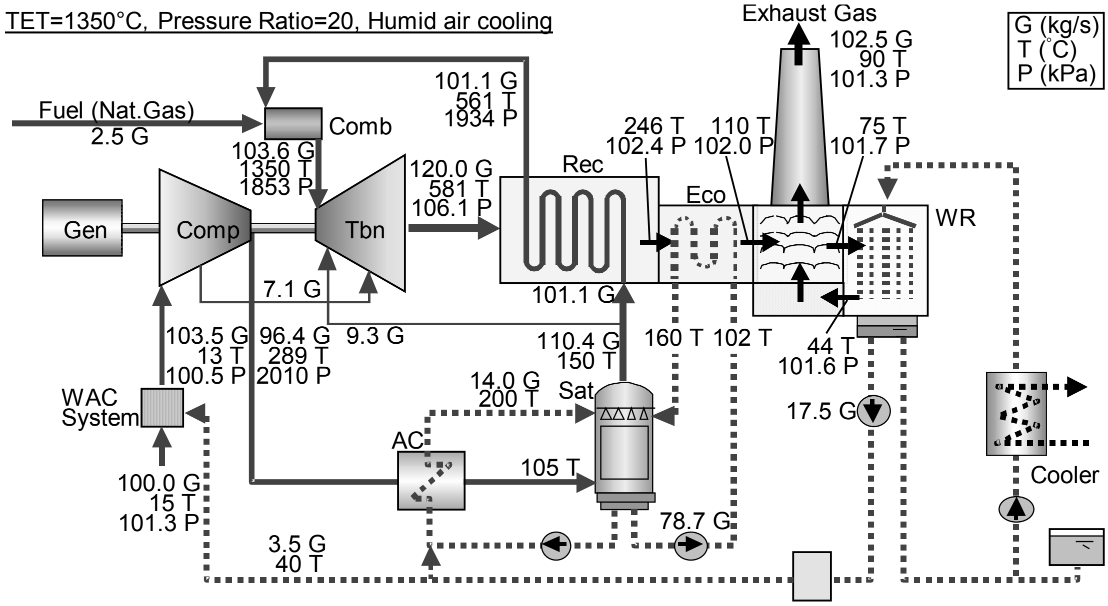


Fig.8 Mass and Heat Balance

Table 2 Remodeling Method

Compressor
- Add WAC system. - Reduce intake air flow rate to adjust pressure ratio to the same value as the base gas turbine.
Combustor
- Modify the flow path to connect the utilities located outside of the gas turbine. - Combust fuel with high humidity air.
Turbine
- Employ the blade cooling method. - Tune up coolant flow rate.

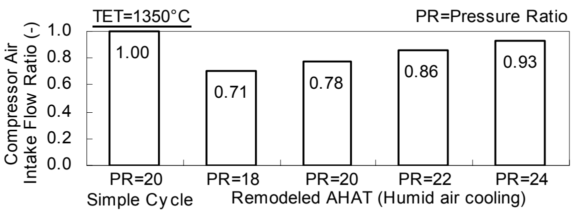


Fig.9 Comparison of Compressor Intake Flow Rate

is constant). In AHAT systems, the humid air humidified with about 20wt% water of the compressor intake air flow rate will flow into the turbine. It is necessary to reduce the flow rate equivalent to the amount of humidification to adjust the pressure ratio to the same value as the base gas turbine shown in Fig.9. In general, controlling the flow rate by the compressor inlet guide vane (IGV) decreases compressor efficiency due to the mismatch of the velocity triangles. Thus, the compressor of the base gas turbine cannot be used as it is. One of the methods to reduce the compressor intake air flow rate while maintaing high efficiency is to reduce the area of the gas path by cutting off the blade length.

Since mainstream gas and coolant conditions differ from those of the base gas turbine, the coolant flow rate must be reset up to prevent blade metal temperature from going over the limit.

Performance of the remodeled AHAT systems

Table 3 shows specifications of the base gas turbine. Figure 10 shows the performance of the remodeled system. The remodeled system is almost the same as the optimally designed system for the conditions of TET of 1,350°C and pressure ratio of 20. The difference in pressure ratio causes a mismatch between the blade angle and the gas flow angle and the drop in the turbine performance. The depression of the aerodynamic performance is a main factor affecting performance.

Table 3 Base Gas Turbine Specifications

Gas Turbine
Type ・Single-shaft, mid-size industrial
Inlet duct ・Ambient condition = ISO standard condition ・Pressure loss $\Delta P/P = 0.009$
Compressor ・Inlet mass flow = around 120 kg/s (This value is suited for 100kg/s-AHAT system) ・Adiabatic efficiency = calculated ・Pressure ratio = 20
Combustor ・Fuel = natural gas ・Combustion efficiency = 0.999 ・Pressure loss $\Delta P/P = 0.04$
Turbine ・7280 rpm, 4-stage ・TET (ISO) = 1,350°C ・1N = single crystal (SC) 2N, 3N & 4N = conventionally cast (CC) ・1R & 2R = single crystal (SC), unshrouded 3R & 4R = directionally solidified (DS), shrouded ・Adiabatic efficiency = calculated ・Consumption of coolant flow = calculated ・Diffuser pressure recovery factor = 0.75 ・Exhaust gas temperature = calculated
Exhaust duct ・Pressure loss $\Delta P/P = 0.006$ (for Simple Cycle)

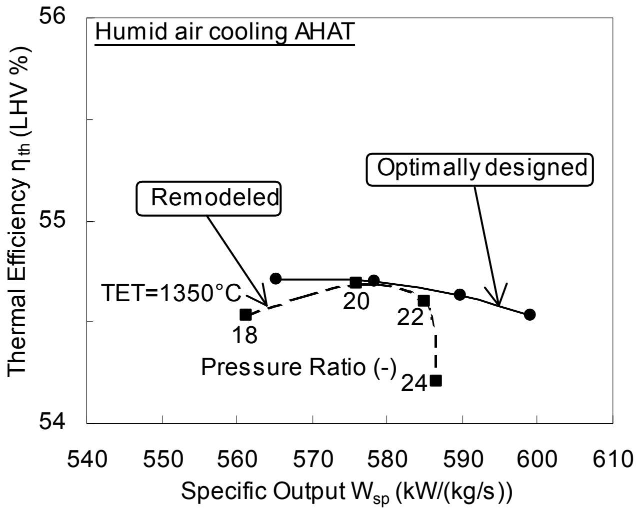


Fig.10 Performance of Remodeled AHAT System

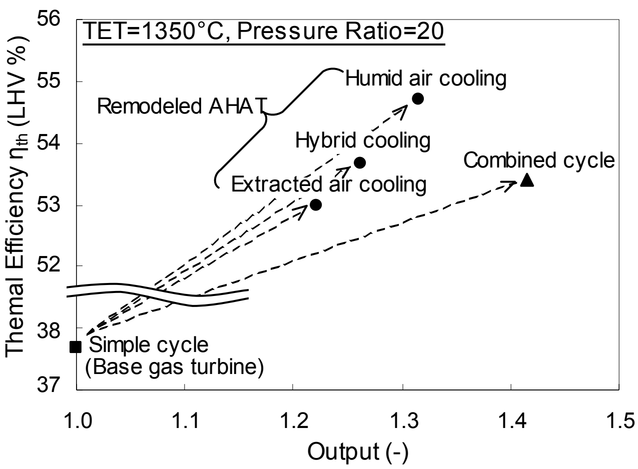


Fig.11 Comparison of Performance

Table 4 Bottoming Cycle Specifications

Bottoming Cycle
HRSG ・Non-reheat-double pressure ・Natural circulation Steam turbines ・Steam conditions : HPT = 5.0MPa, 505°C LPT = 0.6MPa, 165°C ・Exhaust pressure = 4.9 kPa

Figure 11 compares performance. The humid air cooling AHAT system has 1.3 times larger output than the simple cycle system and output of the combined cycle system is 1.4 times larger. Table 4 shows the combined cycle specifications. Triple-pressure reheat HRSG is not used because the system is mid-size. By remodeling the base gas turbine, which is designed for a simple cycle system, to a gas turbine for the AHAT system, raises thermal efficiency from 37.7% to 54.7%. On the other hand, combining the base gas turbine and the bottoming cycle gives a thermal efficiency of 53.4%. The humid air cooling AHAT system has 1.3 percentage points higher thermal efficiency. Of course, as the thermal efficiency of a base gas turbine become higher, those of an AHAT system and a combined cycle system become higher.

Reliability of the remodeled AHAT systems

Figure 12 compares the turbine and the compressor work for the base gas turbine and the three remodeled AHAT systems. Compared with the base gas turbine system, the turbine work of the AHAT systems is almost the same, however the compressor work is less than half. The compressor work is reduced by the reduction of the compressor intake air flow rate and the effect of the WAC system, which decreases the temperature of the intake air. The difference between the turbine work and the compressor work is the output of the system. Of course, the loss and the work of utilities have to be taken into account. The difference between the turbine work and the compressor work in the AHAT systems is much larger than that in a typical simple cycle system and this characteristic makes the output of the AHAT systems larger. In general the stress which acts on parts is proportional to the work. For example, the stress on the turbine blade is proportional to the turbine work. Thus, when the turbine work becomes larger, the stress on the blade becomes larger. The increase in the work may make the stress over the design limit. However, the turbine work of the remodeled AHAT systems never exceeds that of the base simple cycle system. Similarly the compressor work does not exceed that of the base. It follows from this that the typical simple cycle gas turbine can be remodeled to AHAT systems from the viewpoint of reliability. Moreover it seems that the remodeled AHAT system can use utilities for the combined cycle system because output is similar shown in Fig. 11.

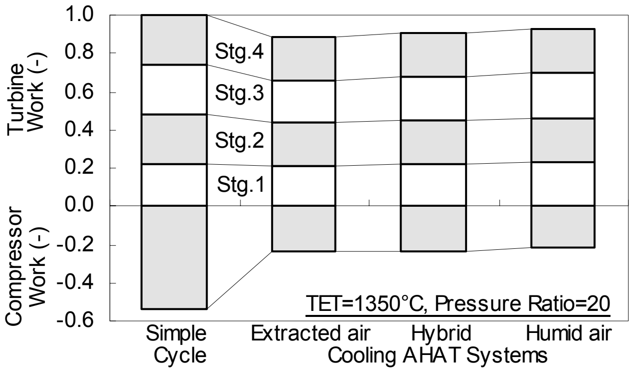


Fig.12 Comparison of Turbine and Compressor Work

CONCLUSIONS

Three types of AHAT systems with different turbine blade cooling systems were simulated. The first type cools all turbine blades with air extracted from the compressor, the second cools all turbine blades with high humidity air and the third cools the turbine nozzle blades with high humidity air and the turbine rotor blades with extracted air. Moreover the applicability of the typical mid-size simple cycle gas turbine to the AHAT system was examined. Main results are as follows.

(1) The humid air cooling system had 1.7 percentage points higher thermal efficiency than extracted air cooling system. The advantage of the humid air cooling system tended to become larger as the turbine entry temperature became higher.

(2) It was found preferable to adjust the compressor intake air flow rate to get the same pressure ratio when the simple cycle gas turbine is remodeled to the AHAT systems.

(3) The AHAT system has 1.3 percentage points higher thermal efficiency than a combined cycle system using the same core gas turbine.

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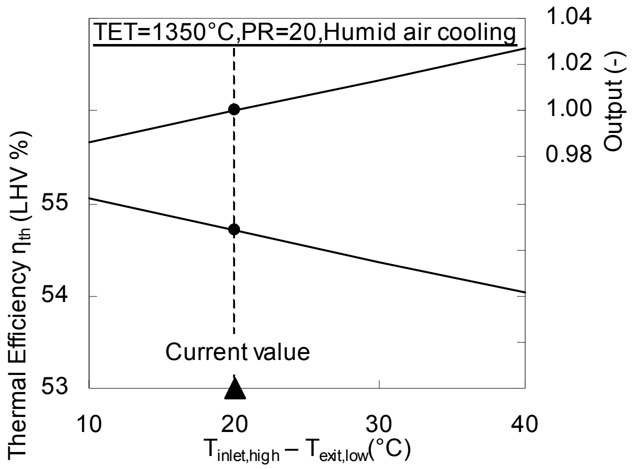
APPENDIX: Impacts of component efficiencies

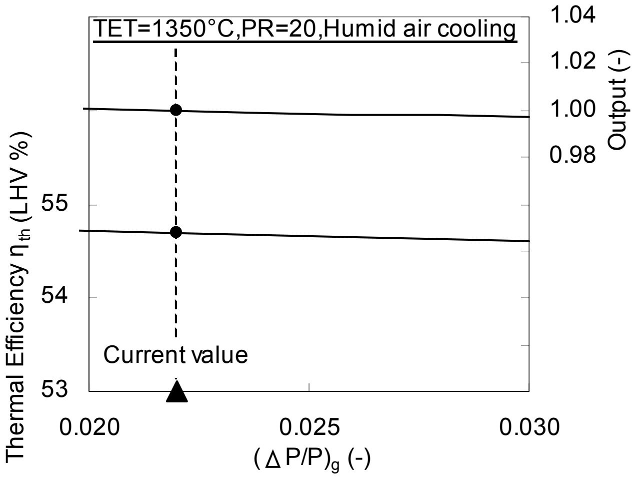
The results have been attained by employing the in-house simulator and by proper numerical assumptions (See Table 1). In this section the impacts of component efficiencies on the overall performance are shown.

Recuperator (Fig.A-1)

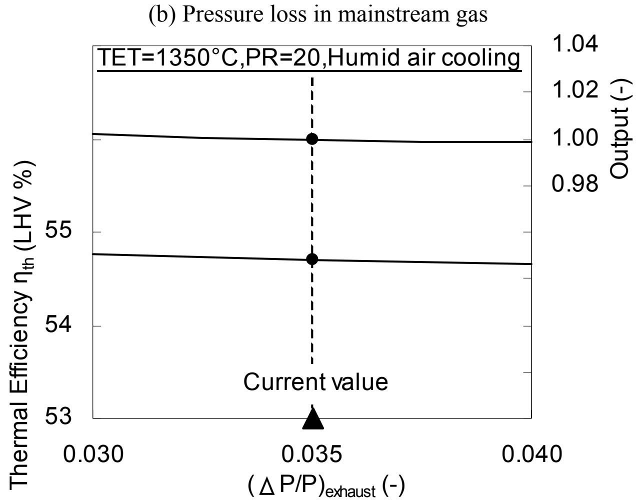
In AHAT systems high efficiency recuperator is one of the most important components for performance. Current value of temperature difference may seem to be too low (that is, efficiency is too high), however, it is selected in consideration of initial and running cost of plants.



(a) Temperature difference



(b) Pressure loss in mainstream gas

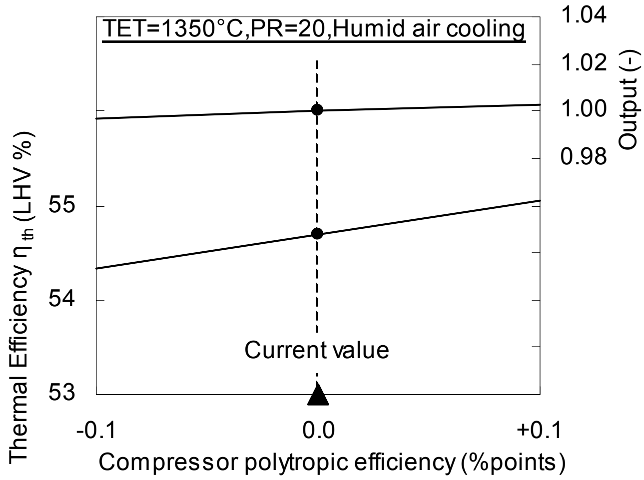


(c) Pressure loss in exhaust gas

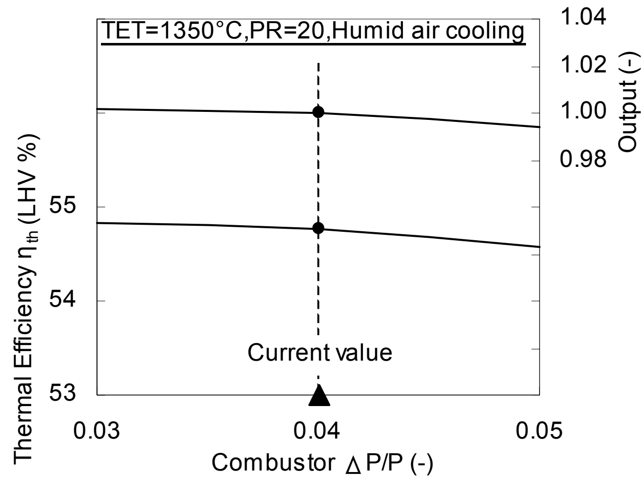
Fig.A-1 Impacts of recuperator parameters

Gas turbine (Fig.A-2)

The compressor efficiency has a large impact on the overall thermal efficiency.



(a) Compressor efficiency



(b) Pressure loss in combustor

Fig.A-2 Impacts of gas turbine parameters