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IMPACT OF STEAM INJECTION AND TURBINE EXHAUST GAS BYPASS IN THE RECUPERATIVE CYCLE GAS TURBINE CHP SYSTEM

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ABSTRACT

The capability of modulating power and heat productions by steam injection in a recuperative cycle gas turbine was investigated. A combined heat and power system using a current state-of-the-art recuperative cycle gas turbine was modeled. Variations in engine performance characteristics due to steam injection were examined. A full off-design analysis was carried out to investigate not only the performance change but also the variation in engine operation caused by the steam injection. Impact of injecting steam at different locations (recuperator and combustor) was investigated. A special attention was given to the change in the compressor surge margin, and a couple of operations that secures a minimum surge margin were comparatively analyzed. Bypass of turbine exhaust gas around the recuperator to increase steam generation was simulated and its usefulness in controlling heat to power ratio was demonstrated. Variations in electric power and thermal energy productions in response to the modulations of injection ratio and gas bypass were presented for a wide ambient temperature range.

NOMENCLATURE

A	area [m^2]
c_p	constant pressure specific heat [kJ/kg·K]
CHP	combined heat and power
FSI-C	full steam injection in the combustor
FSI-R	full steam injection at the recuperator inlet
HRSG	heat recovery steam generator
h	heat transfer coefficient [kJ/s·m^2·K]
LHV	lower heating value [kJ/kg]
$\dot{m}$	mass flow rate [kg/s]
NTU	number of transfer units
P	pressure [kPa]
ΔP	pressure loss [kPa]
PSI-C	partial steam injection in the combustor
PSI-R	partial steam injection at the recuperator inlet
PR	pressure ratio
$\dot{Q}$	heat transfer rate [kW]
R	gas constant [kJ/kg·K]
T	temperature [K]
TIT	turbine inlet temperature [K]
U	overall heat transfer coefficient [kJ/s·m^2·K]
UF	under-firing
ε	effectiveness
γ	specific heat ratio
κ	constant

Subscript

c	cold side
cl	coolant
d	design
h	hot side
in	inlet

INTRODUCTION

Small gas turbines in the power range of hundreds of kWs to several MWs are suitable for distributed power generation including community energy systems (CES). In many of such applications, thermal energy demand is as important as electric power demand. In the small power range, simple cycle gas turbines are not as efficient as those for large power stations in terms of electric power generation efficiency. To overcome the shortcoming, various cycle modifications are possible, but the heat recuperation is the simplest way. Micro gas turbines under hundreds of kWs are good examples of recuperative cycle gas turbines. Efforts have also been made to develop MW-class gas turbines with efficiencies comparable to those of large frame engines. With such a high efficiency, recuperative cycle gas turbines would be competitive in small scale combined heat and power (CHP) and CES systems.

In many CHP systems, demands for electric power and thermal energy fluctuate hourly as well as seasonally. In particular, seasonal variation in the heat to power demand ratio is usually large. In gas turbine CHP systems, steam or water injection to modulate heat to power ratio is a way to cope with such a variation in the demand. Humidification of gas turbines has been researched constantly in last decades [1]. Steam injection is the simplest and most cost-effective method among various humidification schemes suggested for performance upgrade of gas turbines. One of the major targets to adopt steam injection is to mitigate a reduction of power output in hot seasons [2]. Commercial engines with steam injection are available, some of which are possible to follow the variation in heat and power demands [3]. Historically, diverse fundamental studies on the characteristics of steam injected cycles have been performed including comparisons with other cycle schemes such as the combined gas/steam turbine cycle and further improvement to steam injected cycles [4-6]. Until now, steam injection studies have been focused on simple cycle gas turbines. However, with the recent advent of recuperative cycle gas turbines in the market, basic researches have also been initiated regarding their performance enhancement by steam and water injection. Both theoretical [7,8] and experimental [9] studies on the influence of steam injection on the performance of recuperative cycle micro gas turbines have been published recently. A general parametric study on the recuperative cycle including the effect of injection location has been presented as well [10].

Since it is not very economically beneficial to design a new steam injected gas turbine which entails a full revision of many components, the injection of steam to an existing engine with minimum hardware modification is more practical. Even though results of several researches on the steam injection in the recuperative cycle have been published, most of them are pure theoretical ones focusing on design mode calculations. Injection of steam in an existing engine changes its operating condition, and some critical factors such as compressor surge may be an obstacle to achieving the theoretical performance enhancement predicted by design mode calculations. Therefore, an exact off-design analysis taking into account practical operating issues is quite important in understanding the practically achievable performance enhancement. There have been published only a few off-design (operating mode) analyses. A recent study [8] presented simulation results on the steam and water injection effect in a recuperative cycle micro gas turbine on the basis of experimentally observed engine operation data. The major research outcome was performance and operating condition changes depending on injection medium (water or steam) and injection location.

This study aimed to investigate the impact of steam injection in a recuperative cycle engine taking into account practical operating restrictions. A combined heat and power system using a current state-of-the-art recuperative cycle gas turbine was modeled, and a full off-design analysis was carried out. Injection of steam at different locations (recuperator and combustor) was investigated. A special attention was given to the change in the compressor surge margin, and a couple of operations that secures a minimum surge margin were comparatively analyzed. Bypass of turbine exhaust gas around the recuperator was simulated and its usefulness in controlling heat to power ratio was presented. Variations in electric power and thermal energy productions in response to the modulations of injecting ratio and gas bypass were presented for a wide ambient temperature range.

SYSTEM AND MODELING

System configuration

Fig. 1 shows the schematic layout of the CHP system using a recuperative cycle gas turbine. In the simple CHP mode operation, the steam generated at the HRSG is used solely for thermal energy. In the steam injected operation, the generated steam is injected into the gas turbine. The design specification of the gas turbine was set up on the basis of a commercially available recuperative cycle gas turbine. The net power output and efficiency of the engine are 4.6MW and 38.5% [11,12]. In addition to the injection in the combustor (SI-C), the injection at the recuperator inlet (SI-R) was also investigated. Both the full and partial injections of the generated steam were simulated. Also, the influence of partial bypass of the turbine outlet gas on the system performance and operation was examined. The simulations of all of the gas turbine CHP systems were performed using GateCycle [13].

Gas turbine

This section summarizes the modeling of the gas turbine. Firstly, design point performance was simulated on the basis of published design specifications [11,12]: power, efficiency, exhaust gas flow, and turbine inlet and exhaust gas

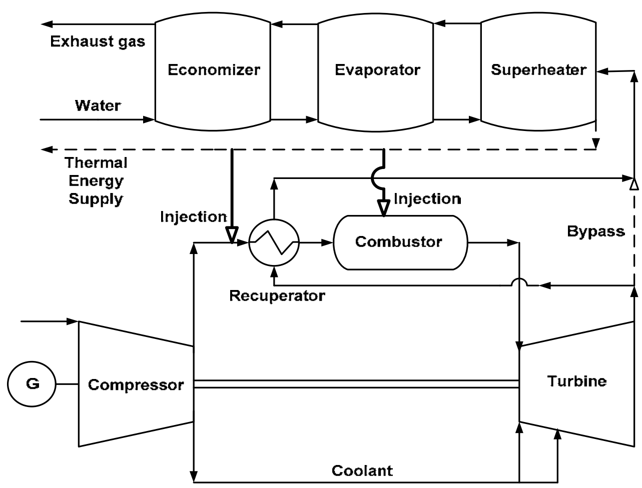

Fig.1 Schematic diagram of the system

Table 1 Design parameters and specifications of the gas turbine

Inlet	Air temperature(K)	288.1
	Air pressure (kPa)	101.3
	Pressure loss (%)	0.5
Compressor	Pressure ratio	9.9
	Number of stages	10
	isentropic efficiency (%)	86.6
Combustor	Fuel flow rate (kg/s)	0.2424
	Fuel lower heating value (kJ/kg)	49300
	Pressure loss (%)	3.0
Recuperator	Effectiveness (%)	91.0
	Hot side Pressure loss (%)	1.5
	Cold side Pressure loss (%)	1.0
	Energy loss (%)	2.0
Turbine	Turbine inlet temperature (K)	1466.5
	Exhaust temperature (K)	657.1
	Number of stages	2
	Stage efficiency (%)	80.6
	Total coolant flow relative to compressor inlet air flow (%)	10.0
Exhaust gas	Pressure (kPa)	101.3
	Gas flow rate (kg/s)	17.73
	Pressure loss (%)	0.5
Performance	Gearbox efficiency (%)	98.1
	Generator efficiency (%)	98.1
	Power (kW)	4600
	Thermal efficiency (%)	38.5

temperatures. Component design information such as numbers of compressor and turbine stages, coolant bleed positions were also referred to the literatures. Minor pressure and energy losses were assumed reasonably. Component characteristic parameters such as compressor and turbine stage efficiencies (assumed identical for all stages), recuperator effectiveness and turbine coolant flow rates, were tuned such that the calculated design specifications were in good agreements with published data. Table 1 lists the simulated major design parameters of the gas turbine. The gas turbine efficiency was defined as follows:

$$\eta_{GT} = \frac{\dot{W}_{GT}}{(\dot{m} \cdot LHV)_{fuel}} \tag{1}$$

Once the design simulation was confirmed, the off-design operation of the gas turbine was modeled. The off-design simulation was required to predict the performance variation of the engine due to ambient temperature change and steam injection. Off-design models of the compressor and the turbine and their matching are the primary requirement for the simulation of engine operation. Since the manufacturer's compressor performance map was not available, several general maps [13] were tried and the one that made the predicted engine performance close enough to the manufacturer's data for a wide ambient temperature range was selected. The compressor map is shown in Fig. 2. The surge margin, defined in the following equation, is 20% at the design point.

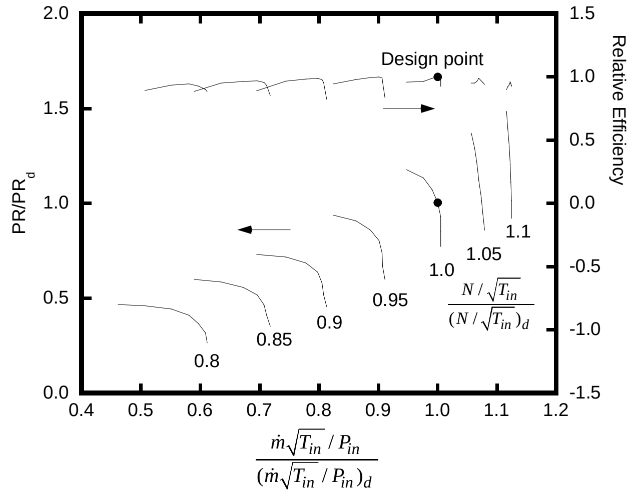

Fig. 2 Compressor performance map

$$\text{Surge margin} = \frac{PR_{surge} - PR_{operation}}{PR_{operation}} \tag{2}$$

The turbine operation was modeled by the following choking equation.

$$\left.\frac{\dot{m}_{in}\sqrt{T_{in}}}{\kappa A_{in} P_{in}}\right)_T = \text{constant}, \quad \text{where } \kappa = \sqrt{\frac{\gamma}{R}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}} \tag{3}$$

The following equation [14] was adopted to simulate the coolant flow variation due to operating condition change:

$$\dot{m}_{cl} = \dot{m}_{cl,d}\left(\frac{P_{cl}}{P_{cl,d}}\right)\left(\frac{T_{cl,d}}{T_{cl}}\right)^{0.5} \tag{4}$$

The pressure drops at flow elements, such as ducts and the recuperator, were corrected using the following equation [15] that considers the changes in flow rate and flow properties:

$$\frac{\Delta P}{\Delta P_d} = \left(\frac{\dot{m}}{\dot{m}_d}\right)^{1.84}\left(\frac{T}{T_d}\right)^{1}\left(\frac{P}{P_d}\right)^{-1} \tag{5}$$

The recuperator is a primary surface type heat exchanger consisting of hot and cold flow streams. The heat exchanger model can be described by the following equation.

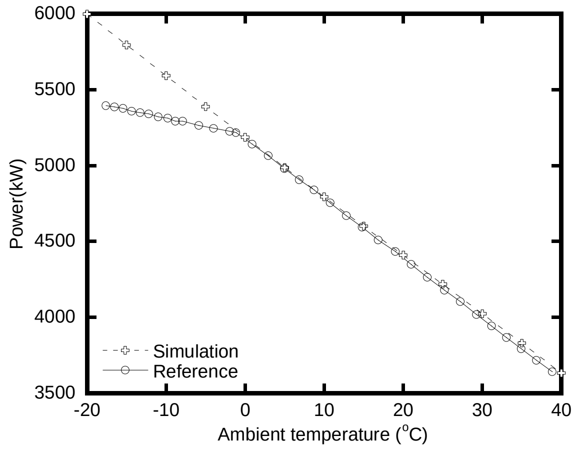

Fig. 3 Variation in power output of the gas turbine with ambient temperature

$$NTU \equiv \frac{UA}{\left(\dot{m}c_p\right)_{\min}} = f(\varepsilon) \quad \text{where,} \quad \varepsilon = \frac{\dot{Q}_{actual}}{\dot{Q}_{\max}} \tag{6}$$

A counter flow heat exchanger [13] was selected. Once the thermodynamic properties, such as cold and hot stream temperatures at both ends of the recuperator and their flow rates, were known from the design calculation, the number of transfer unit (NTU) was determined by the NTU-effectiveness relation. Then, the thermal size of the recuperator at the design point, $(UA)_d$, was determined. The hot and cold side areas were assumed to be identical and similar *hA* values were assigned to both sides at the design point to satisfy the given $(UA)_d$ value according to the following equation.

$$\frac{1}{UA} = \frac{1}{(hA)_h} + \frac{1}{(hA)_c} \tag{7}$$

At off-design operations, the effectiveness varies according to the change in heat transfer characteristics. The major contribution is due to the change in heat transfer coefficients which is primarily caused by the mass flow change. To account for the variation, the following model was used.

$$hA = (hA)_d \left(\frac{\dot{m}}{\dot{m}_d}\right)^{0.3} \tag{8}$$

The exponent was set at 0.3 with a reference to a literature [16]. The separate treatment of hot and cold sides enabled the simulation of bypassing some of the hot gas flow around the recuperator. For the off-design simulation, the same principle described by Eq. (6) was used but the calculation sequence was reversed. Given the heat transfer coefficients at both the hot and cold sides calculated by Eq. (8), the *UA* value was calculated by Eq. (7). Then, NTU was calculated using its definition and the *UA* value (Eq. (6)). Finally, the NTU-effectiveness relation produced the off-design effectiveness value.

Fig. 3 illustrates the simulated full load (design TIT) power output of the gas turbine together with manufacturer's data. The overall agreements are fairy good considering that the engine's real component maps were not able to use. The manufacturer's data showed an abnormal power variation in the low ambient temperature range. This is believed to be a result of partial load operation, i.e., intentional power reduction, due to some practical reasons such as generator and/or bearing capacities. Our simulation result is based on full load operation.

Table 2 Design parameters of HRSG

Inlet water temperature (℃)	15
Inlet water pressure (kPa)	2000
Pinch temperature (℃)	10
Economizer exit sub-cooling (℃)	10
Gas-steam approach temperature difference (℃)	30
Water & steam pressure drop at each section (%)	3.0
Pressure drop at gas side (%)	3.0
Heat loss at each section (%)	1.0

Table 3 Design performance of the CHP system

Gas turbine power output (kW)	4501
Gas turbine efficiency (%)	37.94
HRSG inlet gas temperature (K)	658.1
HRSG outlet gas temperature (K)	435.1
Heat recovery (kW)	4212
CHP efficiency (%)	73.45
Generated steam flow (kg/s)	1.364

HRSG and CHP system

The HRSG consists of three heat exchanger sections: economizer, evaporator and superheater. The steam pressure was set sufficiently high to be injected into the high pressure side of the gas turbine. The design parameters of the HRSG, determined with a reference to a literature [4], are summarized in Table 2. The heat transfer performance of each section was corrected for the off-design simulation. The correction procedure of the heat transfer rate of the HRSG is similar to that of the recuperator. The only difference is that the overall heat transfer coefficient of the HRSG can be corrected by the changes in the gas side properties only because the heat transfer coefficient of the water/steam side is sufficiently higher than that of the gas side. The adopted correction [13,17] is as follows.

$$UA = (UA)_d \left(\frac{\dot{m}}{\dot{m}_d}\right)^{0.8} \tag{9}$$

Table 4 Summary of various operation modes

Operating mode	description	Steam injection	TIT
CHP	Normal CHP operation	None	Design value
FSI-C	Full steam injection in the combustor	Full	Design value
FSI-R	Full steam injection at the recuperator inlet	Full	Design value
FSI-C-UF	Full steam injection in the combustor with under-firing	Full	Reduced
FSI-R-UF	Full steam injection at the recuperator inlet with under-firing	Full	Reduced
PSI-C	Partial steam injection in the combustor to meet 10% surge margin	Partial	Design value
PSI-R	Partial steam injection at the recuperator inlet to meet 10% surge margin	partial	Design value

Once the *UA* was corrected from the design value, the calculation sequence is quite similar to that of the recuperator calculation: the estimation of NTU and calculation of effectiveness according to the NTU-effectiveness relation followed by the calculation of transferred heat.

Table 3 summarizes the design performance of the entire CHP system. The power output and efficiency of the gas turbine in the CHP system is slightly lower than the nominal values in Table 1 because of the additional pressure drop at the HRSG. The design heat recovery rate is a bit less than the electric power.

Steam injection

A part or all of the generated steam from the HRSG was injected to the gas turbine. The gas turbine calculation needed the injected steam as an input value. However, the exact steam generation rate can be decided by the HRSG calculation. Therefore, an iterative calculation inside the entire system was required: we assumed the injected steam rate and checked it as a calculation outcome, and iterated until the two values agreed.

The location of steam injection in the recuperative cycle affects the steam generation rates and thus the performance and operation of the gas turbine [8]. Diverse operating strategies were simulated as summarized in Table 4. 'CHP' means the reference operation without steam injection. 'FSI' refers to full steam injections where all of the steam generated from the HRSG is injected to the gas turbine. 'C' and 'R' denote injections into the combustor and at the recuperator inlet, respectively. In general, the increased turbine gas flow due to steam injection causes the turbine inlet pressure, i.e. the compressor discharge pressure, to rise according to the compressor and turbine matching, thereby reducing the surge margin. Therefore, the full steam injection may cause an unacceptably low surge margin. Such an operation was predicted in this study as will be shown in the next section. Two different approaches to avoid too low a surge margin were simulated and compared: 'UF' and 'PSI' in Table 4 denote the under-firing and the partial steam injection, respectively. In the under-firing, TIT was reduced to meet 10% surge margin. In the partial injection, TIT is maintained at the design value but only part of the generated steam was injected to meet the same minimum surge margin.

Hot gas bypass

Injecting steam into the gas turbine enhances electric power output but reduces the capacity of thermal energy supply. If the thermal energy demand is quite small compared with the increased electric power demand (i.e. if the site-required heat to power ratio is quite small), such a steam injected operation could be a good solution. However, if the thermal energy demand is still pretty high, the electric power augmentation needs to be limited to satisfy the thermal energy demand. In the recuperative cycle, the HRSG heat recovery rate depends much on the heat transfer rate at the recuperator. Therefore, if the recuperator heat transfer could be controlled, the HRSG heat recovery, i.e. the steam generation, could be modulated to some degree. The hot gas bypass around the recuperator enables such an operation (See Fig. 1). The bypass would be beneficial in both the normal CHP operation and the steam injected operation because the capacity of thermal energy supply can be increased by the bypass while electric power is kept at a target level. As the fraction of bypassed hot gas increases, the HRSG inlet gas temperature rises, which in turn generates more steam. Then, for a required electric power augmentation which can be achieved by a certain amount of steam injection, the available steam for thermal energy supply increases.

RESULTS AND DISCUSSION

Full steam injection

First, the full injection at the two different locations was

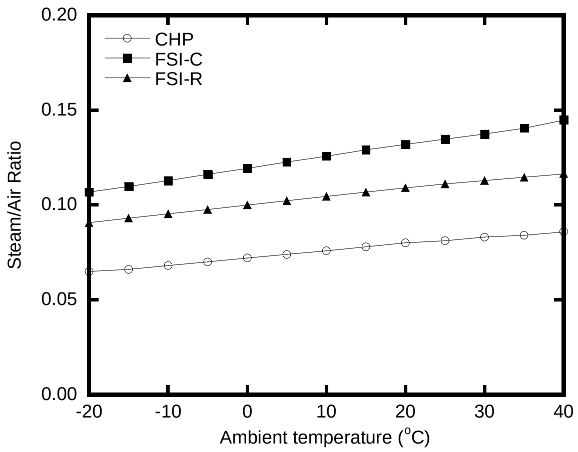

Fig. 4 Steam to air ratio in the operations with full steam injection

simulated for a wide ambient temperature range. Fig. 4 shows the ratio between the injected steam flow, which is the same as the generated steam flow in the full injection, and the compressor inlet air flow. The result for the simple CHP operation is also drawn for comparison. In the simple CHP operation at 15^oC ambient temperature, the steam flow amounts to 7.8% of the inlet air flow. The steam/air ratio increases as the ambient temperature rises because a higher ambient temperature leads to a higher HRSG inlet gas temperature. In the steam injected operations, the generated steam increases considerably compared to the simple CHP operation. This is due to the increases in both the mass flow and the specific heat of the hot gas stream at the recuperator. A distinct difference in the steam flow rates is observed between the injections at the two different locations. The injection at the recuperator inlet (FSI-R) requires more heat exchange at the recuperator compared to the injection in the combustor (FSI-C) because the flow rate of the cold side air (more exactly, air/steam mixture) is larger, which lowers the HRSG inlet gas temperature. Thus, FSI-R generates less steam at the HRSG.

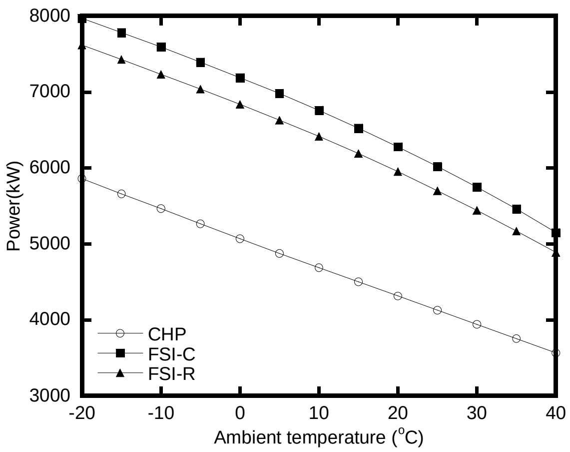

Fig. 5 Gas turbine power output in the operations with full steam injection

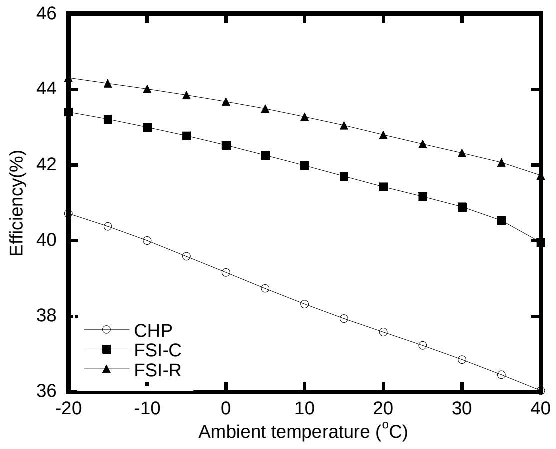

Fig. 6 Gas turbine efficiency in the operations with full steam injection

Figs. 5 and 6 show the power output and efficiency of the gas turbine. At ambient temperatures above 15^oC, FSI-C and FSI-R can produce roughly 45% and 38% larger power output than the simple CHP operation. The efficiency gain over the simple CHP operation is 10.0% and 13.5% for the FSI-C and FSI-R, respectively, at 15^oC ambient temperature. The injection at the recuperator inlet requires less fuel supply at the combustor because the air/steam mixture is pre-heated at the recuperator. In summary, FSI-C and FSI-R are superior to each other in terms of power and efficiency, respectively. The qualitative results are similar to those obtained from a simulation of a much smaller and less efficient recuperative cycle micro gas turbine [8]. But, the relative performance augmentation is greater in the present case: the power and efficiency augmentations were 24% and 3.8% for the full injection in the combustor, and 20% and 8.6% for the full injection at the recuperator in the case of a micro gas turbine of 30kW class [8]. Thus, it can be said that the advantage of steam injection becomes greater as the design performance of the gas turbine becomes higher. A more extensive study can justify this tentative conclusion. Returning to the present results, the relative efficiency gain increases as the ambient temperature rises, reaching 10.9% and 15.8% at 40^oC for FSI-C and FSI-R, respectively.

Operation with surge margin control

Despite the remarkable performance enhancement, the full steam injection is accompanied by a critical problem. The increased turbine mass flow causes the compressor discharge pressure to rise, approaching the surge point. Fig. 7 shows the surge margin for the three operations. In the simple operation without steam injection, the surge margin remains greater than 10% even at 40^oC ambient condition. However, with the full steam injection the surge margin reduces drastically. Of course, FSI-C exhibits a smaller surge margin because of the relatively steam injection. In almost all ambient conditions, the surge margin of the fully injected operations falls below 10%. The

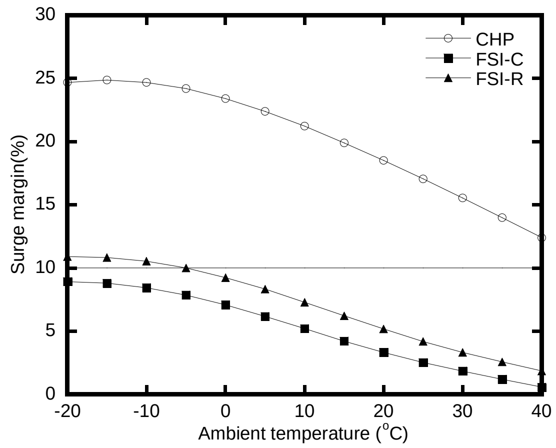

Fig. 7 Surge margin in the operations with full steam injection

surge margin approaches zero as the temperature rises over typical summer conditions. It is not allowable in the view point of engine safety.

The allowable minimum surge margin may depend on various factors, which can best be suggested by engine manufacturers. In this work, a case study was tried to present the effect of surge margin control. 10% surge margin was used as a guideline. As described in Table 4, two methods were introduced to recover the surge margin to 10%: under-firing (FSI-C-UF and FSI-R-UF) and partial injection (PSI-C and PSI-R). Fig. 8 shows the ratio between the injected steam flow and the compressor inlet air flow, and Fig. 9 shows the turbine inlet temperature. In the under-firing operations, the generated steam is fully injected while the turbine inlet temperature is reduced to achieve 10% surge margin. The TIT reduction needs to be intensified as the ambient temperate rises. The difference in the injected (i.e. generated) steam flow rates between the under-firing (Fig. 8) and the full-firing (Fig. 4) is due to the difference in the HRSG inlet gas temperatures: the case with under-firing generates less steam due to the lower HRSG inlet gas temperature. The injection in the combustor allows a larger steam injection but needs a greater TIT reduction to meet the same surge margin. In the partially injected operations, the steam generation rates are similar to those of the under-firing operations. However, only part of the generated steam is allowed to be injected to meet the 10% surge margin while the turbine inlet temperature is kept at the design value. The portion of injected steam decreases as the ambient temperature rises. Because the injection amount is the major parameter that determines the compressor operating pressure, the injected steam flow is effectively the same regardless of the injection location as long as the same 10% surge margin is satisfied in the partial injections (PSI-C and PSI-R).

The power output and efficiency of the gas turbine in the surge margin controlled operation is shown in Figs. 10 and 11. Both the under-firing and partial injection reduce the power and efficiency in comparison to the full steam injection with full firing (see Figs. 5 and 6). The partial injection provides relatively larger power output and higher efficiency than the under-firing. The difference becomes larger as the ambient temperature rises. Since PSI-C and PSI-R provide almost the

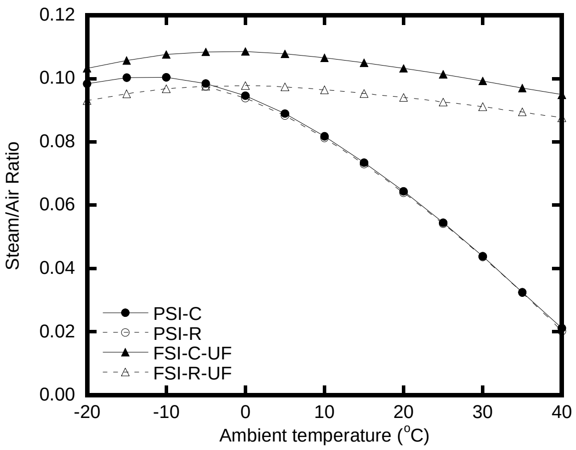

Fig. 8 Injected steam to air ratio in operations with 10% surge margin

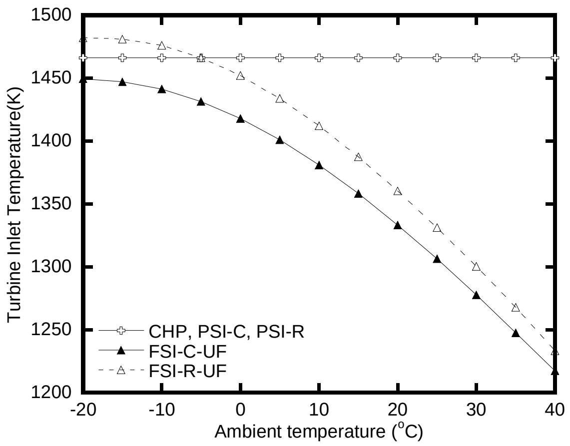

Fig. 9 Turbine inlet temperature for operations with 10% surge margin

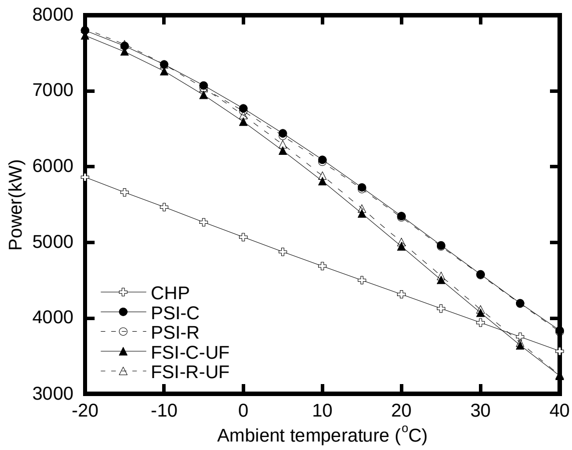

Fig. 10 Gas turbine power output in operations with 10% surge margin

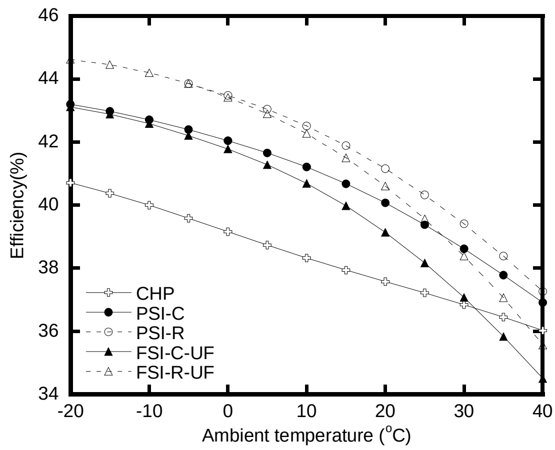

Fig. 11 Gas turbine efficiency in operations with 10% surge margin

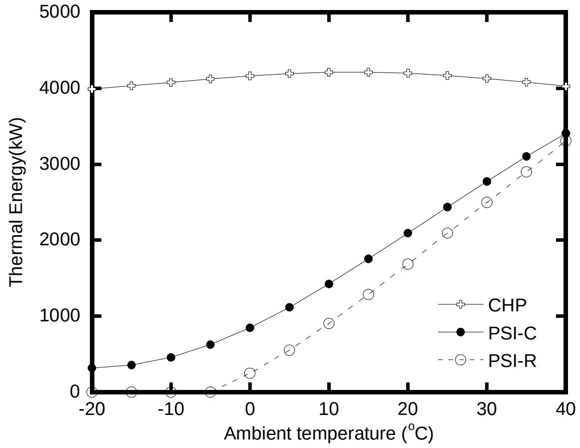

Fig. 12 Capacity of thermal energy supply in operations with 10% surge margin

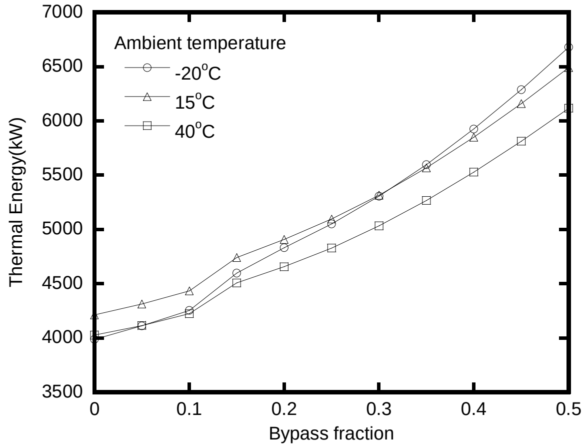

Fig. 13 The influence of hot gas bypass on the capacity of thermal energy supply in the simple CHP operation

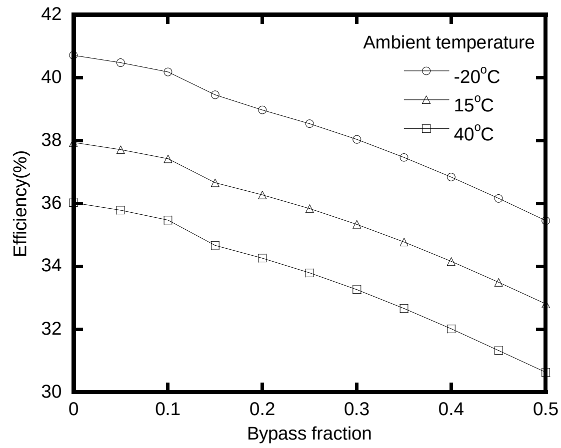

Fig. 14 The influence of hot gas bypass on the gas turbine efficiency in the simple CHP operation

same injection rates, they provide nearly equivalent power outputs. However, PSI-R requires less fuel supply to the combustor owing to the preheating of the injected steam at the recuperator, resulting in relatively higher efficiency. With under-firing, the two counteracting effects of the larger turbine flow and the lower turbine inlet temperature and vice versa results in nearly the same power output between FSI-C-UF and FSI-R-UF. Of course, FSI-R-UF provides higher efficiency than FSI-C-UF due to the preheating of the injected steam before its entry into the combustor. The performance penalty increases dramatically as the ambient temperature rises. In particular, the under-firing exhibits lower power output and efficiency in the high temperature range, which means that the TIT reduction (more than 200K for 40°C ambient temperature) penalizes the performance too much. Fig. 12 shows the variation in the capacity of thermal energy supply, i.e. the energy of the remaining steam. In the simple CHP operation, the exhaust gas flow, i.e. HRSG gas flow, decreases with increasing ambient temperature. However, the gas turbine exhaust temperature, i.e. HRSG inlet gas temperature, increases with increasing ambient temperature. These two counteracting factors resulted in the relatively flat variation (slight maximum around 10°) in the thermal energy. Of course, no steam, i.e. no thermal energy, is available in the case of under-firing. On the contrary, the partial injection can provide thermal energy, which increases as the ambient temperature rises due to the reduction in the portion of injected steam. With the recuperator inlet injection, over 10% surge margin is possible in the ambient temperature range below -5°C even with the full injection (see Fig. 7). As a result, in that region, full injection is still applied for PSI-R in Fig. 12, and thus thermal energy is not available (zero).

In conclusion, the partial injection is superior to the under-firing in all thermodynamic aspects: gas turbine power and efficiency, and thermal energy supply. However, even in the partial injection, the enhancement of gas turbine performance appears to be marginal in the high ambient temperature range due to the limitation in the injection amount. A relaxation (reduction) of the minimum surge margin would increase the benefit of the steam injection. For example, an additional simulation for the operation with 5% surge margin showed that the relative power augmentation with PSI-C at 40°C ambient temperature would increase from 7.3% in the case of 10% surge margin to 24.6% in the case of 5% surge margin.

Hot gas bypass

The influence of hot gas bypass on the system performance was investigated for the simple CHP operation first, and then for the steam injected operation with surge margin control. If some of the hot gas from the turbine exit is bypassed around the recuperator, the steam generation rate increases due to the increased HRSG inlet gas temperature, but the combustor fuel supply increases (i.e. efficiency decreases) because the preheating of the air at the recuperator becomes weaker. The results are presented as a function of bypass fraction for three ambient temperatures in Figs. 13 and 14. The bypass fraction in

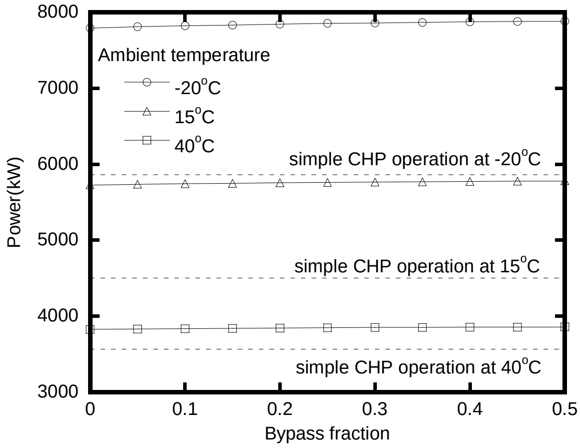

Fig. 15 The influence of hot gas bypass on the gas turbine power output in PSI-C

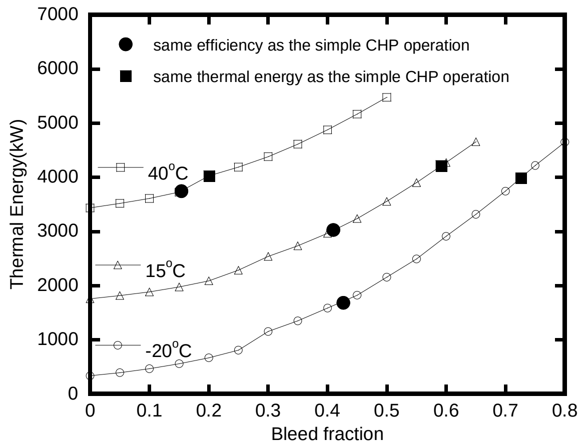

Fig. 16 The influence of hot gas bypass on the capacity of thermal energy supply in PSI-C

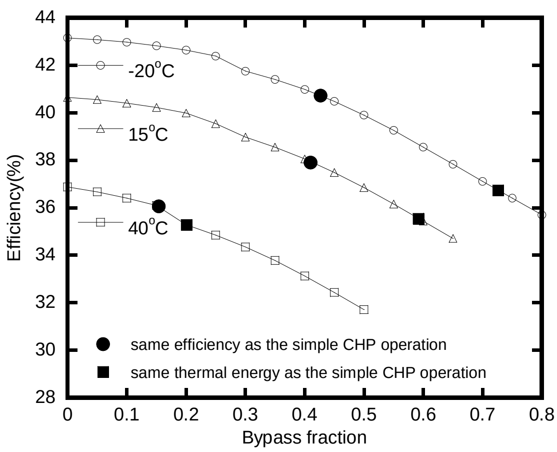

Fig. 17 The influence of hot gas bypass on the gas turbine efficiency in PSI-C

the figure denotes the ratio of the bypassed gas flow to the total turbine exit gas flow. Bypassing a half of the hot gas increases the capacity of thermal energy supply (i.e. the generated steam) by more than 50%. The penalty of power generation efficiency at the same condition is more than 5 percent points. The gas turbine power output slightly increases with increasing bypass fraction due to a reduction in the pressure loss at the hot side of the recuperator. However, since the effect is very marginal, the power output remains effectively constant. The hot gas bypass can be a solution to manage the varying heat to power demand ratio. However, the efficiency penalty must be a disadvantage.

Impact of hot gas bypass in the steam injected operation is illustrated for the partial injection in the combustor with 10% surge margin (PSI-C). The influences of hot gas bypass on power output, thermal energy and efficiency are presented for three ambient temperatures in Figs. 15 to 17. The data for 0% bypass are those for the PSI-C in Figs. 10 to 12. Gas turbine power output is insensitive to the bypass fraction. The bypass considerably enhances the capacity of thermal energy supply as in the simple CHP operation. The power generation efficiency decreases with increasing bypass fraction, but the degree of efficiency penalty is less than that of the simple CHP operation (compare results of Figs. 14 and 16). In summary, the capacity of thermal energy supply can be modulated by controlling the bypass fraction, while maintaining the same level of power augmentation. Therefore, a quite flexible operation depending on the heat to power demand ratio, especially on the variation in thermal energy demand, is possible. Another advantage of such an operation is that the power generation efficiency can be higher than that of the simple CHP operation. For example, at -20°C ambient temperature, the operation with the bypass fraction up to 42.7% guarantees power generation efficiency over the efficiency in the simple operation. The bypass fractions that exhibit the same power generation efficiencies as the simple CHP operations are marked by the dark circles in the figures. The operation flexibility also allows for further increase of the capacity of thermal energy supply. The bypass fractions that correspond to the same capacity of thermal energy supply as the simple CHP operation are marked by dark rectangles in the figures. The performance of various operations is compared in Table 5.

CONCLUSION

The influence of steam injection and hot gas bypass on the performance and operation of a CHP system using a recuperative cycle gas turbine was simulated. The results are summarized as follows.

(1) With steam injection, much more steam can be generated due to the increases in both the flow rate and the specific heat of the exhaust gas. The steam injection in the combustor provides greater power augmentation compared to the injection at the recuperator, while the latter yields better power generation efficiency. The full injection is accompanied by considerably surge margin reduction, leading to unacceptably small surge margins in some operating environments.

Table 5 Comparison of various PSI-C operations with simple CHP operation

Ambient temperature	PSI-C operation with the same efficiency as the simple CHP operation				PSI-C operation with the same thermal energy as the simple CHP operation				PSI-C operation with zero bypass			Simple CHP operation		
	Bypass fraction (%)	GT eff. (%)	GT power (kW)	Thermal energy (kW)	Bypass fraction (%)	GT eff. (%)	GT power (kW)	Thermal energy (kW)	GT eff. (%)	GT power (kW)	Thermal energy (kW)	GT eff. (%)	GT power (kW)	Thermal energy (kW)
-20°C	42.7	40.7	7878	1686	72.6	36.74	7901	3988	43.2	7794	334	40.7	5859	3988
15°C	41.0	37.9	5766	3026	59.2	35.55	5787	4212	40.7	5727	1758	37.9	4501	4212
40°C	15.4	36.0	3839	3741	20.1	35.28	3843	4025	36.9	3826	3437	36.0	3564	4025

(2) Two strategies to achieve a target (minimum) surge margin were compared: full injection with under-firing and partial injection with full firing. The partial injection provides relatively greater power output and higher efficiency. Its superiority becomes marked as the required degree of surge margin recovery becomes greater.

(3) The bypass of some of the turbine exhaust gas flow around the recuperator provides a high flexibility of varying the capacity of thermal energy supply in both the simple CHP operation and the steam injected operation. In particular, in the partially injected operation, the capacity of thermal energy supply can be modulated by controlling the bypass fraction, while producing greater power output than the simple CHP system. Moreover, the power generation efficiency is higher than that of the simple CHP operation for a wide bypass fraction range.

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