Thermodynamic Performance of an Internal Reheat Gas Turbine (IRGT) with Hydrogen Combustion*

By

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ABSTRACT

Combustion of hydrogen gas fuel between blade rows realizes the internal reheat gas turbine (IRGT), which can be extended to a multi stage reheat gas turbine. Thermodynamic analysis was performed on three kinds of multi-stage reheat gas turbine cycles with regenerator. The first one is a regenerative reheat gas turbine system with a hydrogen gas preheater and a steam generator. The second one is the first one with a steam injection at the exit of compressor. The third one is the first one combined with a steam turbine. The cycle calculations were performed for both steam cooled and non-cooled IRGT. The combined cycle gives the best thermal efficiency, that is, 63% for non-cooled IRGT and 58% for blade cooled IRGT. The thermal efficiency of the combined cycle with a regenerative reheat gas turbine is better than the one of the combined cycle with a simple cycle gas turbine.

INTRODUCTION

Hydrogen gas is not only reproducible fuel but also cleaner than fossil ones in regard to combustion emission. Although it requires energy sources to be produced, it will be able to become a substitute of oil provided that a conversion of inexhaustible energy such as solar energy is facilitated. An advanced reheat gas turbine concept with combustion of hydrogen gas has already been proposed by the authors, which is named the Internal Reheat Gas Turbine (IRGT) [1] . Hydrogen gas has a couple of advantages as a fuel, which are its high flame velocity, wide flammable limit and high heating value. Furthermore, its high specific heat and high thermal conductivity can be useful to cool a turbine blade. In the IRGT, hydrogen gas is used for the turbine blade cooling for the first step, then introduced into the hot main gas stream from the trailing edges of blades and it burns up in the main gas stream. This is the way of the internal reheat for the gas turbine, that is, the reheat between turbine blade rows.

It has already been shown that the combustion of a hydrogen gas is realized in the downstream of a blade row by the experiments of a two-

* Received on January 12, 1987

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dimensional blade row and a nozzle sector designed for an experimental gas turbine [1]. It is now in process to make the realization of the IRGT concept using the experimental gas turbine.

If the reheat inside a gas turbine becomes possible, it can be extended to a multi-stage reheat gas turbine. This system leads to a high thermal efficiency. If a gas turbine cycle with isothermal compression and expansion processes and with a complete regeneration were realized, the cycle could have as high a thermal efficiency as the one of the Carnot cycle which is the Ericsson cycle. The isothermal expansion process would be achieved by an infinite-stage reheat. The multi-stage reheat is an approximation of this process.

The combustion of hydrogen gas between blade rows needs 1.5 to 2.0 times blade chord length in the axial distance to complete the combustion itself [1] . This means an increase in the surface of hot gas passage, that is, an increase in the surface to be cooled. Gas turbines in the field adopt compressed air for a blade cooling now, while much consumption of it for cooling comes to a reduction in the thermal efficiency. Due to the internal reheat, however, much more coolant is required, so it is necessary to use other cooling substance than compressed air. In the present report, steam generated with an exhaust gas is considered as the coolant.

A heat-exchanger must be used as a regenerator in case of reheat gas turbine to recover the heat of the high temperature exhaust gas, then an improvement of the thermal efficiency may be achieved as well as an increment of specific output power. In a multi-stage reheat gas turbine, the exhaust gas temperature will become higher, so it is necessary to effectively recover the high temperature waste heat. In this report, the following items are taken into account in addition to the regenerator as the effective recovery means of waste heat:

i) to generate steam with the exhaust gas for the cooling of the turbine blades and casing,

ii) to inject a part of the steam generated with the exhaust gas into the main air flow at the exit of compressor,

iii) to construct a combined cycle system with a steam turbine,

iv) to preheat the fuel hydrogen gas with the exhaust gas.

In order to estimate the performance of the IRGT, cycle calculations were carried out for three kinds of gas turbine systems, which are supposed to be used for marine propulsion. The first one is named the GT system which is a regenerative multi-stage internal reheat gas turbine system with a hydrogen gas preheater and a steam generator. The second one is named the SI system which is the GT system with a steam injection at the exit of compressor. The last one is named the CC system which is the GT system combined with a steam turbine. The cycle calculations were performed for both of steam cooled and non-cooled IRGT. The non-cooled IRGT is supposed to make use of new ceramic materials which are under active researches and developments and are expected to become usable for a gas turbine in the future.

NOMENCLATURE

C_p=specific heat at constant pressure
C_v=specific heat at constant volume
f=fraction of exhaust gas flow
H_u=lower heating value
i=enthalpy
L=output power, work done, blade height
l=blade chord length
N=net specific output power
P=pressure
ΔP=pressure difference
q=heat flux
R=gas constant
S_g=blade perimeter
S_t=Stanton number
T=temperature
ΔT=temperature difference
t=blade pitch
U=velocity at inlet of blade row
W=mass flow rate
α=heat transfer coefficient
γ=density of gas
ε=heat exchanger effectiveness
η=efficiency, effectiveness
$\kappa=C_p/C_v$

Subscript

a=air
B=waste heat recovery boiler
b=combustion, blade
C=compressor
Ch=hydrogen compressor
c=coolant

ex = exit
f = fuel
fr = fuel used for reheat
fw = feed water
g = main gas flow, gas side
H = hydrogen gas
h = preheater, high pressure
in = inlet
j = j th stage of turbine
l = low pressure
m = mixture
N = nitrogen gas
O = oxygen gas
p = pinch point
R = regenerator
r = reheat
ST = steam turbine
s = steam, isentropic
sh = super-heated
sa = saturated
T = gas turbine
0-8,61 = state of gas

OUTLINE OF CYCLES

The three gas turbine systems are illustrated in Fig.1 to Fig.3. Fig.4 indicates a T-S (temperature and entropy) diagram of a multi-stage internal reheat gas turbine. In the GT system as shown in Fig.1, the ambient air, which is 15°C, 1.033kg/cm^2, enters the compressor at the state 1 and is

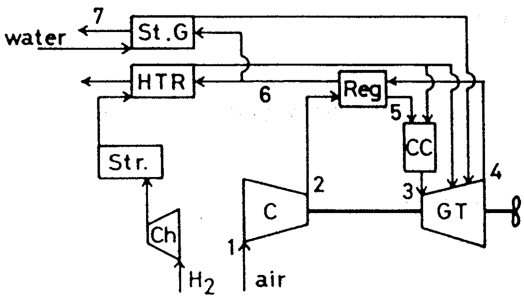

Fig.1 Outline of GT system

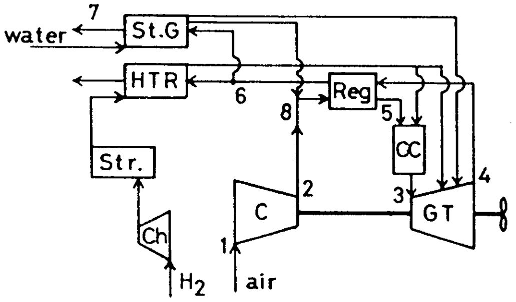

Fig.2 Outline of SI system

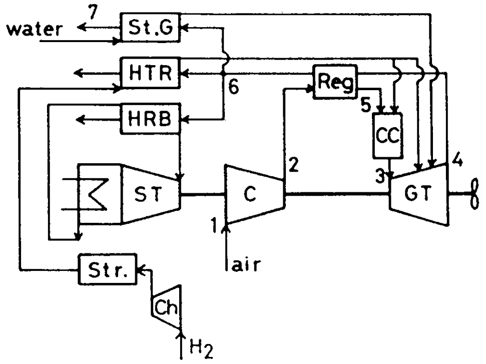

Fig.3 Outline of CC system

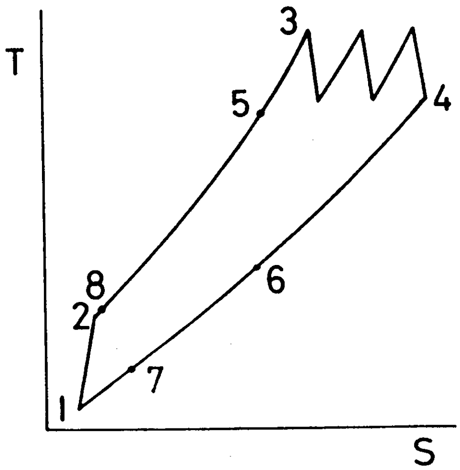

Fig.4 T-S diagram of multi-stage reheat gas turbine

compressed to the state 2. The compressed air is heated through the regenerator(Reg) and introduced to the combustion chamber(CC) where hydrogen gas burns. The combustion gas runs into the turbine(GT) at the state 3 and expands to the state 4. During the expansion, the internal reheat is executed. The exhaust gas from the turbine heats the air in the regenerator for the first time and then preheats the fuel hydrogen gas in the preheater(HTR) and also generates steam for cooling in the steam generator (St.G) before it is discarded into the ambient air through a stack.

For the fuel hydrogen gas, boil-off gas of liquified hydrogen is compressed by a hydrogen compressor(Ch) and warmed up to the atmospheric temperature 15°C with sea water(Str). Before the compressed hydrogen gas is supplied to both the main combustor and the turbine for the reheat, it is preheated by the preheater(HTR) up to the temperature level below 400°C.

The exhaust gas has the ability to generate more steam than the one necessary to cooling of the hot components of the turbine. In the SI system (Fig.2), the rest of the steam is heated up to a temperature level below 500°C and then is injected at the exit of the compressor. In the CC system(Fig.3), a heat recovery boiler(HRB) makes use of the rest of the capacity of steam generation and generates steam for a steam turbine(ST). Both the output powers of steam and gas turbine are used for the propulsion.

CYCLE CALCULATION METHOD

Here are primary assumptions and definitions described. Detailed calculation method is presented in the appendix. The working gas properties used in the cycle analysis are taken as an ideal gas mixture of nitrogen, oxygen, hydrogen and steam. The thermal efficiency η_{th}, the specific output power N and the heat exchanger effectiveness η_R are defined as follows:

$$N = L_T + L_{ST} - L_C - L_{Ch} - L_{fw} \tag{1}$$

$$\eta_{th} = N/(W_f + W_{fr})H_u \tag{2}$$

$$\eta_R = (T_4 - T_6)/(T_4 - T_8) \tag{3}$$

The number of turbine stages is set to be RHT+1, where RHT is the reheat time. The expansion ratio is held constant for each stage. The reheat inside the turbine is supposed to be done at constant pressure after the expansion of each stage (except the last stage) and the working gas is to be heated up to the turbine inlet temperature T_3 again.

Blades and annulous walls of the turbine are cooled with steam when the temperature at the entry of a turbine stage is over 800°C. All the steam used for cooling is assumed to be mixed up with the main working gas right after the expansion of the stage. Pressure losses during the mixing are ignored.

The mass flow of blade cooling steam is determined by Eq.(4) which is derived from the following assumption: turbine blades are treated as heat exchangers operating at constant metal temperature and the coolant exit temperature is simply expressed as a function of a blade heat exchanger effectiveness ε (see Eq.21 in the Appendix). The coolant mass flow for the annulous walls is assumed to be proportional to the area ratio of the annulous walls to the blade surface.

$$\frac{W_c}{W_g}=\frac{S_t}{\varepsilon}\frac{C_{pg}}{C_{pc}}\frac{S_g}{t}\frac{T_g-T_b}{T_b-T_{cin}} \tag{4}$$

The following values for the variables are used in this analysis.
$\varepsilon=0.8$, $T_b=800°C$, $S_t=0.005$, $S_g/l=2.4$, $t/l=0.8$, $L/t=2$.

The heat recovery boiler-system and the steam turbine-system are outlined in Fig.5. The selected pressures are 40ata for the high pressure turbine, 12ata for the middle pressure turbine, 2.5ata for the low pressure turbine and 0.05ata for the condenser. The temperature of super-heated steam is limited to 500°C and is set lower than the gas temperature T_6 by 50°C. The reheated steam temperature is supposed to be the same as that of the high pressure steam.

The primary parameters used in this analysis are listed in Table 1.

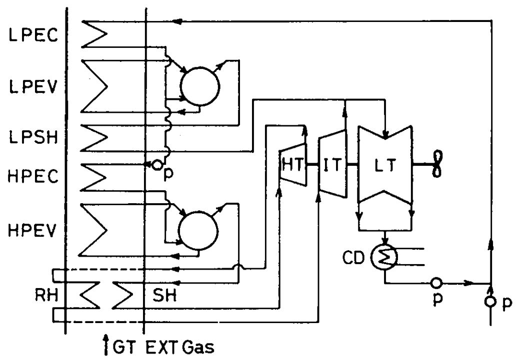

Fig.5 Heat recovery boiler system

RESULTS AND DISCUSSION

Figs.6 and 7 show the obtained results of the thermal efficiencies η_{th} of the three systems, namely, GT system, SI system and CC system, against various pressure ratios Pr. In the case of no blade cooling (see Fig.6), the more the reheat times increase, the better thermal efficiencies of the three systems become. Fig.6 also shows that the thermal efficiencies of GT system are improved by SI system and much more improved by CC system.

Table 1 Primary parameters used in the analysis

Turbine inlet temperature	T_3	1000°C～1300°C
Pressure ratio	$Pr=P_2/P_1$	3～20
Compressor efficiency	η_C	0.88
Turbine efficiency	η_T	0.9
Steam turbine efficiency	η_{ST}	0.85
Heat exchanger effectiveness	η_R	0.6～0.8
Combustion efficiency	η_b	1.0
Boiler efficiency	η_B	0.95
Pressure loss in combustor	ΔP_c	5%
Air pressure loss in heat exchanger	ΔP_a	3%
Gas pressure loss in heat exchanger	ΔP_g	4%
Gas pressure loss in steam generator	ΔP_s	4%
Hydrogen compressor efficiency	η_{Ch}	0.50
Water pump efficiency	η_P	0.7

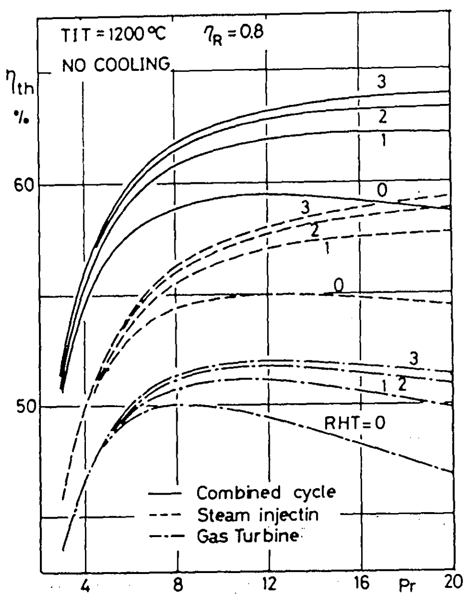

Fig.6 Thermal efficiency of GT, SI and CC system for no blade cooling

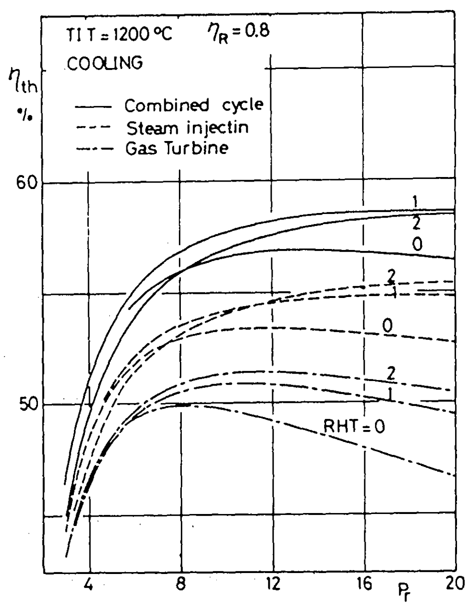

Fig.7 Thermal efficiency of GT, SI and CC system for blade cooling

In the case of turbine blade cooling(see Fig.7), SI and CC system decrease their thermal efficiencies η_{th} a lot, as being compared to the ones for no blade cooling. The fact is attributed to that the blade cooling reduces the steam to be used for steam injection or steam turbine. On the other hand, the thermal efficiencies of GT system are almost in the same level as the ones for no blade cooling, because the blade cooling steam raised in a waste heat recovery boiler does some expansion work in the gas turbine and compensates the gas temperature reduction due to blade cooling.

Figs.8-A to 8-F indicate the maximum thermal efficiencies η_{thmax} with respect to the turbine inlet temperatures(TIT), where the heat exchanger effectiveness and the reheat time are kept constant. Figs.8-A to 8-C for no blade cooling show that the themal efficiencies η_{th} are improved as the turbine inlet temperatures(TIT) become higher. As the heat exchanger effectiveness η_R increases, the thermal efficiencies of all the three systems are improved and the thermal efficiency improvement by the reheat becomes larger. Fig.9 explains the fact mentioned above, which is redrawn for the relation between η_{thmax} and η_R at TIT=1200°C.

Figs.8-D to 8-F for blade cooling present the followings. A higher heat exchanger effectiveness restricts the reheat time. The reason is that the heat capacity of the exhaust gas from the turbine becomes larger than that of the compressed air due to the cooling steam injected into the working gas and it becomes too large in the case of a high TIT and a large number of reheat time to keep a high heat exchanger effectiveness. Although there is no restriction in the reheat time at a lower heat exchanger effectiveness, multi-stage reheat cannot contribute to the thermal efficiency improvement because of insufficient regeneration. Fig.10 explains the fact mentioned above with a relation of η_{thmax} with respect to η_R at 1200°C.

Fig.11 shows the effect of supply temperature of the fuel hydrogen on the thermal efficiencies. The preheated hydrogen fuel improves both the thermal efficiencies of GT system and CC system. In all the calculations in this report, the hydrogen fuel is assumed to be heated up to 400°C just below the temperature of its ignition 530°C

So far, the discussion has been based on the calculation range of pressure ratios between 3 and 20. It is not realistic, however, in development of a regenerative gas turbine to consider gas turbine cycles with too high pressure ratios, because the seal of rotating part of heat exchanger becomes one constraint. Here the case with the compression ratio of 10 is chosen as an example. Fig.12 for no blade cooling depicts the relations between thermal efficiencies and specific output powers. It shows that the increment in the TIT and the reheat time improves the thermal efficiencies and the specific

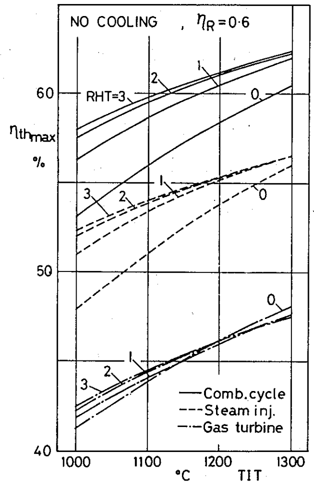

Fig.8-A

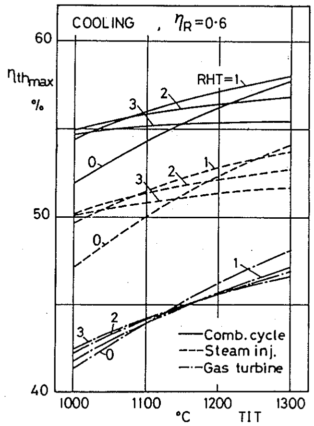

Fig.8-D

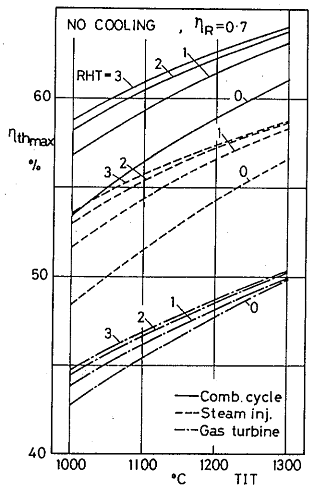

Fig.8-B

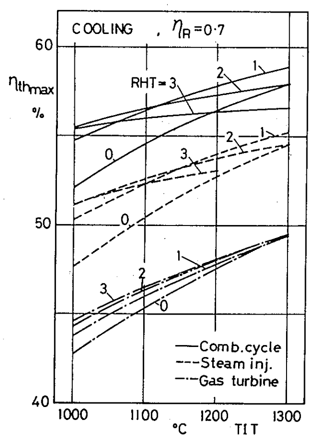

Fig.8-E

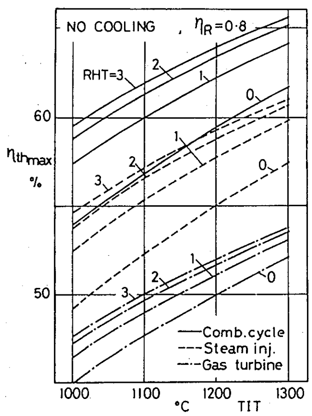

Fig.8-C

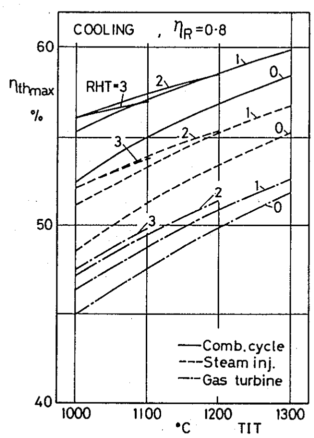

Fig.8-F

Fig.8 Maximum thermal efficiencies with respect to turbine inlet temperature

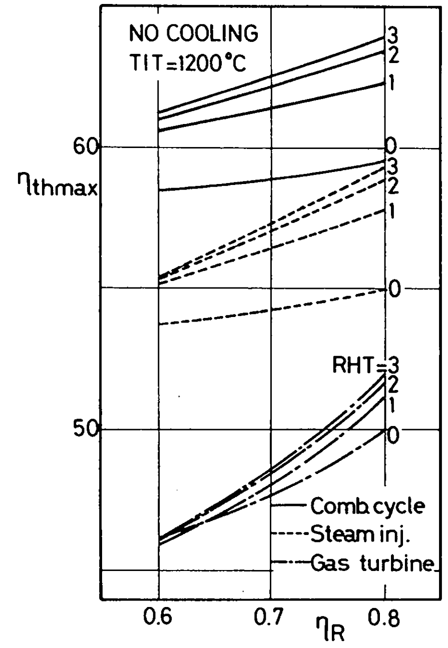

Fig.9 Maximum thermal efficiencies with respect to heat exchanger effectiveness for no blade cooling

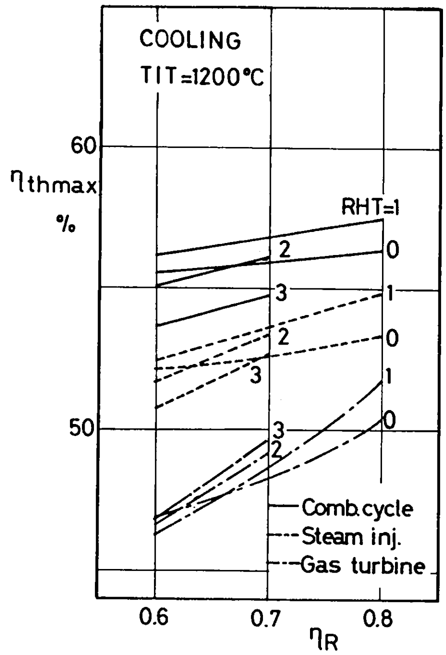

Fig.10 Maximum thermal efficiencies with respect to heat exchanger effectiveness for blade cooling

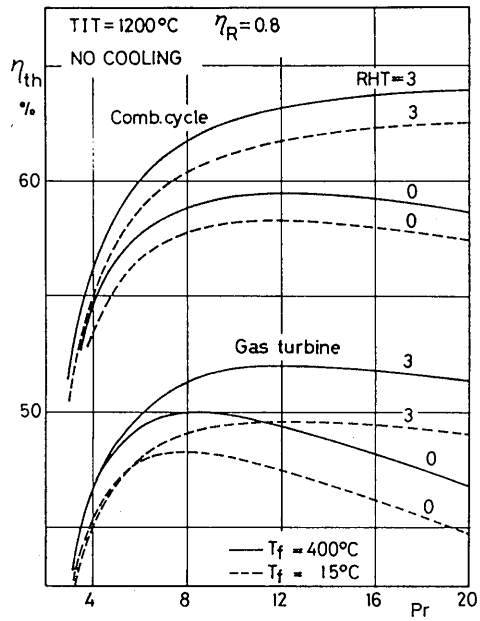

Fig.11 Effect of supply temperature of hydrogen gas fuel on the thermal efficiencies

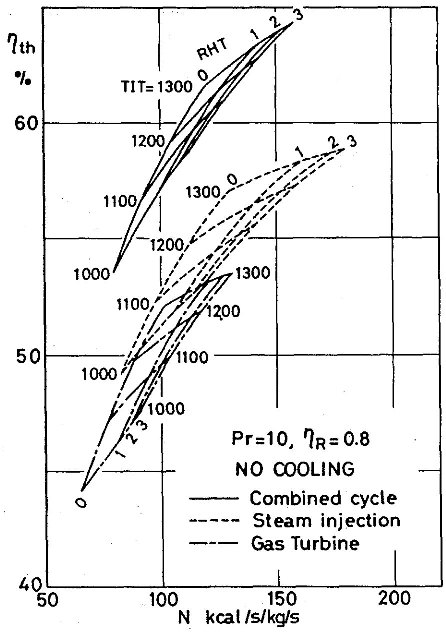

Fig.12 Relations between thermal efficiencies and specific output powers for no blade cooling

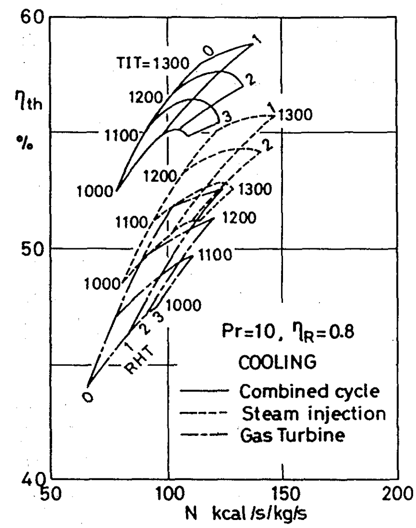

Fig.13 Relations between thermal efficiencies and specific output powers for blade cooling

output powers. At the TIT of 1300°C the thermal efficiency for the CC system is the highest of the three systems and it is above 63%. 58% and 53% are for the SI system and for the GT system respectively. The specific output powers for the SI system are the highest of the three and they are more than 150 kcal/s/kg/s. Fig.13 is for the case of blade cooling. At the TIT of 1300°C and the reheat time of 1, the thermal efficiency for the CC system is above 58%, and 55% for the SI system and 52% for the GT system. Cooling of blades and annulous walls reduces the advantage of the multi-stage reheat and limits the reheat time. Adoption of film cooling will reduce the coolant flow rate and also loosen the constraints of the multi-stage reheat, although only the internal cooling was considered in this analysis.

Regenerative gas turbine cycles were discussed so far, although they have some drawbacks of complexity and constraint of seal. Here a simple cycle gas turbine without regenerator is to be compared with the regenerative gas turbine. Figs.14 and 15 show that the thermal efficiencies of the CC system with the heat exchanger effectiveness of 0.8 are higher than the ones of a combined cycle with a simple cycle gas turbine, of which heat exchanger effectivness is 0.

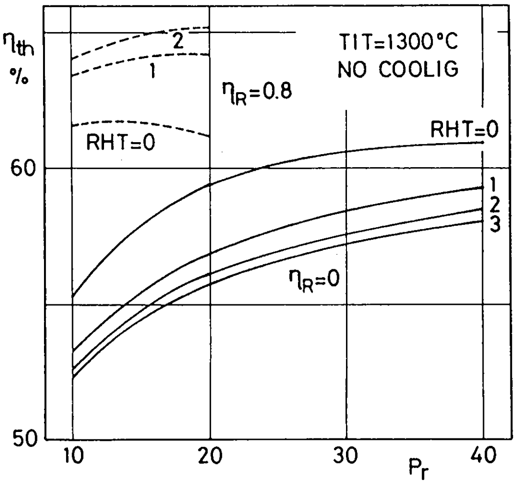

Fig.14 Thermal efficiencies of the combined cycle with the regenerative reheat gas turbine and with a simple gas turbine for no blade cooling

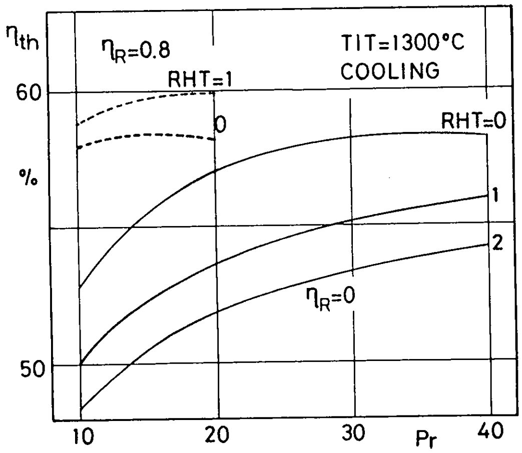

Fig.15 Thermal efficiencies of the combined cycle with the regenerative reheat gas turbine and with a simple gas turbine for blade cooling

CONCLUSIONS

A thermodynamic analysis of an internal reheat gas turbine (IRGT) leads to the following conclusions.

(1) The internal reheat may realize a multi-stage reheat gas turbine and the multi-stage reheat can increase the thermal efficiency and specific output power of the GT system as long as high heat exchanger effectiveness is kept.

(2) The thermal efficiency of the GT system is considerably improved with the SI and CC systems. Especially, the CC system is the best of all.

(3) In the case of blade cooling, multi-stage reheat dose not always lead to high thermal efficiency at a high turbine inlet temperature.

(4) Preheating of fuel hydrogen is effective on improvement of thermal efficiencies of the three systems.

(5) The three systems have the following thermal efficiencies at TIT=1300°C. They are 63% (CC), 58% (SI) and 53% (GT) for no blade cooling, and 58% (CC), 55% (SI) and 52% (GT) for blade cooling.

(6) The thermal efficiencies of the combined cycle with the regenerative reheat gas turbine is better than the ones of a combined cycle with a simple cycle gas turbine.

(7) Detailed analysis is necessary to be done with respect to blade cooling and heat exchanger for more exact performance evaluation.

REFERENCE

1) S. Kan, T. Morishita, K. Hiraoka; "Reheat Gas Turbine with Hydrogen Combustion between Blade Rows", 1983 Tokyo International Gas Turbine Congress, 83-Tokyo-IGTC-27

APPENDIX

DETAIL OF CALCULATION METHOD

1. Gas property

The values of specific heats of the ideal gases are cited from the reference [A1]. The enthalpies of the gases are given by a numerical integration of the specific heats, where 0°C is taken as the reference temperature. All the properties for water and steam between 0°C and 800°C are given by the SRI (Ship Rerearch Institute) steam table library [A2]. The specific heat and enthalpy of steam above 900°C are given from the ideal gas data. The specific heat and enthalpy of steam between 800°C and 900°C are given by an interpolation between the values at the two temperature levels.

The specific heat used in the calculation of boil-off hydrogen gas compression is given in Table A1. According to the reference [A3], the

Table A1 Assumed specific heats of boil-off hydrogen gas

T (K)	21	80	160	220	300
Cp (kcal/kgK)	2.8	2.6	3.0	3.35	3.4

specific heat changes considerably with respect to the pressure in the lower temperature region. However, for simplification of the analysis, the specific heat data between the pressure 1ata and 10ata [A3] are taken as reprerentatives.

The specific heats C_{pm}, C_{vm}, the ratio of specific heats $\varkappa_m$, the enthalpy for the gas mixture i_m and the partial pressure of steam P_s are calculated from the following equations.

$$W_g = W_N + W_O + W_H + W_S \tag{1}$$

$$i_m = (W_N i_N + W_O i_O + W_H i_H + W_S i_S)/W_g \tag{2}$$

$$C_{pm} = (W_N C_{pN} + W_O C_{pO} + W_H C_{pH} + W_S C_{pS})/W_g \tag{3}$$

$$C_{Vm} = (W_N C_{vN} + W_O C_{vO} + W_H C_{vH} + W_S C_{vS})/W_g \tag{4}$$

$$\varkappa_m = C_{pm}/C_{vm} \tag{5}$$

$$P_S/P_m = W_S R_S/(W_N R_N + W_O R_O + W_H R_H + W_S R_S) \tag{6}$$

2. Compressor

A unit mass flow 1kg/sec of air with 15°C, 1.033kg/cm², is assumed to enter the compressor C. For a given inlet temperature T_1, inlet pressure P_1 and exit pressure P_2, an isentropic compression process gives an exit temperature T_{2s} by the following relation,

$$T_{2S} = T_1(P_2/P_1)^{(\kappa-1)/\kappa} \tag{7}$$

The value κ is taken as a mean value between the values at T_1 and T_{2s}. Providing the compressor efficiency η_C, and the enthalpies i_1 and i_{2s} at T_1 and T_{2s} respectively, the exit enthalpy of the compressed air i_2 is given by

$$i_2 = i_1 + (i_{2S} - i_1)/\eta_C \tag{8}$$

Here the work required to drive the compressor is given by

$$L_C = i_2 - i_1 \tag{9}$$

The exit temperature T_2 is calculated from the relation between T and i.

The compression work of boil-off hydrogen gas is calculated in the similar way by using the data of low temperature region. The compression ratio is assumed to be 1.5 times of the pressure ratio Pr.

3. Steam injection

The super heated steam is injected into the main compressed air flow at the exit of compressor. No pressure loss is assumed in this process. Then, the enthalpy of the mixture i_8 is given by the following energy conservation relation.

$$(W_a + W_s)i_8 = W_a i_2 + W_s i_s \qquad (10)$$

Then the temperature T_8 is calculated from the enthalpy i_8.

4. Regenerator

In the regenerator, the gas from the turbine discharge heats the compressed air, and Eq.(11) is given for energy conservation using the notation of Fig.2.

$$W_g(i_4 - i_6) = W_a(i_5 - i_8) \qquad (11)$$

where i_5 and i_6 are unknown. Regenerators are usually evaluated in the form of temperature effectiveness, that is,

$$\eta_{Rin} = (T_5 - T_8)/(T_4 - T_8) \qquad (12)$$

or

$$\eta_{Rex} = (T_4 - T_6)/(T_4 - T_8) \qquad (13)$$

In this cycle calculation η_{Rex} is specified because the heat capacity of the exhaust gas is larger than that of the air due to the turbine cooling steam. Eq.(13) gives T_6, then i_6 is calculated from the relation between temperature and enthalpy. Now the enthalpy i_5 at inlet to the combustor can be determined from Eq.(11) and then T_5 is given. Pressure losses in the passages of the regenerator are taken into account and the pressure loss rates $\Delta P_a/P_8$ for air side and $\Delta P_g/P_6$ for gas side are assumed to be constant.

5. Combustor

In the combustor the air at T_5 is heated up to the specified turbine inlet temperature T_3. If the hydrogen fuel is supplied at T_f, the fuel flow rate W_f can be calculated from the following energy conservation.

$$\begin{aligned}\eta_b W_f H_u &+ W_f(i_{Hf} - i_{H0}) + W_o(i_{o5} - i_{o0}) + W_N(i_{N5} - i_{N0}) + W_s(i_{s5} - i_{s0}) \\ &= W_N(i_{N3} - i_{N5}) + (W_0 - 8\eta_b W_f)(i_{o3} - i_{o5}) + 9\eta_b W_f(i_{S3} - i_{S0}) \\ &\quad + (1 - \eta_b) W_f(i_{H3} - i_{Hf})\end{aligned} \qquad (14)$$

6. Turbine

For a single stage of expansion ratio Pr_j, an isentropic expansion process gives an exit temperature T_{4sj}.

$$T_{4sj} = T_{3j}(Pr_j)^{(1-\kappa)/\kappa} \qquad (15)$$

Since the enthalpies i_{3j} and i_{4sj} are calculated with T_{3j} and T_{4sj}, a turbine output power L_{Tj} for a single stage with turbine efficiency η_T is given by

$$L_{Tj}=\eta_T(i_{3j}-i_{4sj})W_{gj} \tag{16}$$

Then the turbine output power L_T through the entire expansion process is given by

$$L_T=\sum_j^n L_{Tj} \tag{17}$$

7. Turbine cooling

The heat flux from the main gas flow to a turbine blade is given by

$$\begin{aligned} q_g&=S_gL\alpha_g(T_g-T_b)\\ &=\frac{\alpha_g}{C_{pg}\gamma U}\gamma tLUC_{pg}\frac{S_g}{t}(T_g-T_b) \end{aligned} \tag{18}$$

which is reduced to

$$q_g=S_tC_{pg}W_g\left(\frac{S_g}{t}\right)(T_g-T_b) \tag{19}$$

The heat flux from the blade to cooling steam is given by

$$q_c=C_{pc}W_c\varepsilon(T_b-T_{cin}) \tag{20}$$

where

$$\varepsilon=(T_{cex}-T_{cin})/(T_b-T_{cin}) \tag{21}$$

The mass flow ratio of blade cooling steam to main flow gas is given by the energy balance between q_g and q_c, as shown in the following relation,

$$\frac{W_c}{W_g}=\frac{S_t}{\varepsilon}\frac{C_{pg}}{C_{pc}}\frac{S_g}{t}\frac{T_g-T_b}{T_b-T_{cin}} \tag{22}$$

It is assumed that the mass flow of coolant for annulous walls is proportional to the factor, which is defined by the ratio of the surface area of annulous walls to blades. The annulous walls area A_a between two blades is about 2tl, where t is blade spacing and l is blade chord. Since the IRGT requires more space between turbine stages than conventional ones, it is assumed that the surface area to be cooled in one stage is five times of A_a.

8. Preheater of hydrogen gas

The fuel hydrogen gas is preheated in the preheater from the atmospheric temperature T_l to a fuel feeding temperature T_f. T_f is set at a temperature lower than the regenerator exit temperature T_{g6} by 50°C and is limlited to 400°C below its ignition temperature 530°C. The exit gas is led to stack at the temperature T_{g7} of 100°C. The fraction f_h of the exhaust gas introduced into the preheater is given by the next heat balance.

$$W_f(i_{Hf}-i_{H1})=f_h\eta_B W_g(i_{g6}-i_{g7}) \tag{23}$$

9. Steam generator

The steam for cooling or steam injection is assumed to be generated in the steam generator on the following conditions.

(1) The temperature of super-heated steam T_{sh} is limited to 500°C and is set lower than the gas temperature T_{g6} by 50°C. For steam cooling, T_{sh} is fixed at a temperature above a saturation temperature T_{sa} by 10°C.

(2) The temperature difference ΔT_p between gas and water at the pinch point is to be more than 10°C.

(3) The exit gas temperature T_{g7} is to be above 100°C.

(4) The water is fed at a temperature T_{fw}=15°C and is pumped up to 1.3 times a pressure of compression ratio Pr.

(5) The boiler effiency η_B is assumed to be 0.95 due to gas leakage and radiation loss.

The fraction f_c of the exhaust gas led to the steam generator for generating cooling steam is given by the following heat balance.

$$W_c(i_{shc}-i_{fw})=f_c\eta_B W_g(i_{g6}-i_{g7}) \tag{24}$$

Then the fraction f_s of the exhaust gas for steam injection or steam turbine is given by

$$f_s=1-f_h-f_c \tag{25}$$

10. Steam turbine

The heat recovery boiler is assumed to be composed of two pressure stages in order to recover the waste heat as much as possible. In this analysis the high pressure steam temperature varies mainly in the region of 350°C to 450°C. The steam turbine pressure level for the steam temperature 400°C usually drops on the region between 30 and 50ata, therefore, the high pressure tubine is assumed to be driven at 40ata.

A T-S diagram for the steam turbine system is indicated in Fig.A1. Fig.A2 shows the gas and steam temperatures during the heat recovery process.

An enthalpy of a state after an isentropic expansion can be determined by a given inlet super-heated steam temperature and pressure and a given discharge pressure. Then the enthalpy drop for a unit mass flow of steam is given by

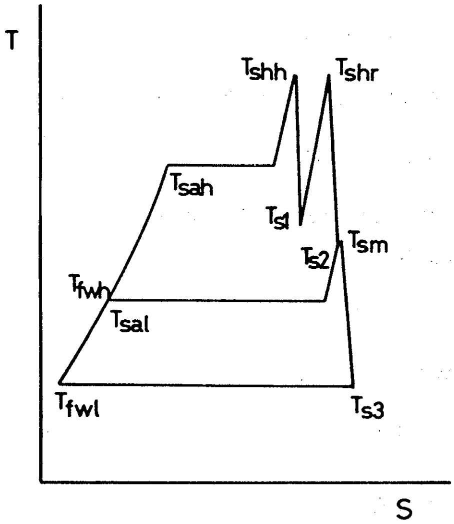

Fig.A1 T-S diagram of steam turbine

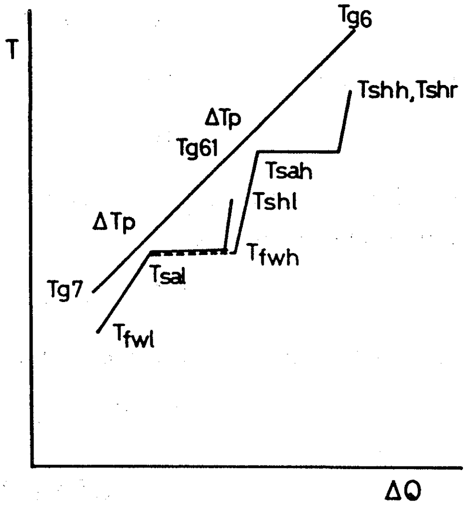

Fig.A2 Relation between gas temperatures and steam temperatures

$$\Delta i_{sh} = \eta_{ST}(i_{shh} - i_{s1s}) \tag{26}$$

in the high pressure turbine.
And the enthalpy of steam discharged from the high pressure turbine is

$$i_{s1} = i_{shh} - \Delta i_{sh} \tag{27}$$

Similarly, the enthalpy drops and the enthalpies of steam discharged from the turbines are

$$\Delta i_{sr} = \eta_{ST}(i_{shr} - i_{s2s}) \tag{28}$$
$$i_{s2} = i_{shr} - \Delta i_{sr} \tag{29}$$
$$\Delta i_{sl} = \eta_{ST}(i_{sm} - i_{s3s}) \tag{30}$$
$$i_{s3} = i_{sm} - \Delta i_{sl} \tag{31}$$

for the middle and low pressure turbine.
The steam flow rate W_{sh} generated for the high pressure turbine is given by the following heat balance between the inlet of the high pressure super heater and the pinch point.

$$W_{sh}[(i_{shh} - i_{sah}) + (i_{ihr} - i_{s1})] = \eta_B(i_{g6} - i_{g6p}) \tag{32}$$

And the enthalpy of the gas at the exit of high pressure economizer is represented by

$$\eta_B(i_{g6p}-i_{g61})=W_{sh}(i_{sah}-i_{fwh}) \tag{33}$$

The heat balance for generation of the low pressure steam is expressed by

$$W_{sl}(i_{shl}-i_{sal})=\eta_B(i_{g61}-i_{g61p}) \tag{34}$$

The mixture of the low pressure steam and the steam discharged from the middle pressure turbine gives the following heat balance.

$$(W_{sh}+W_{sl})i_{sm}=W_{sh}i_{s2}+W_{sl}i_{shl} \tag{35}$$

Finaly, the total steam turbine output power is given by

$$L_{ST}=f_s W_g[W_{sh}(\Delta i_{sh}+\Delta i_{sr})+(W_{sh}+W_{sl})\Delta i_{sl}] \tag{36}$$

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