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The Suction Air Cooling Gas Turbine with Vapor Compression Refrigerator*

(2nd Report, The Optimum Point of Cycle and the Effect of Combination)

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The thermodynamic optimum point of the suction air cooling gas turbine cycle with a vapor compression refrigerator is investigated. The maximum thermal efficiency and the maximum specific power increase monotonously with a rise of the temperature ratio by the suction air cooling, but the rate of increase of the thermal efficiency has a tendency to give the optimum value and that of the specific power has a tendency to decrease monotonously, compared with the conventional cycle. The effect of the reversed cycle (refrigerator) on the normal cycle (heat engine) is investigated. The effect on the thermal efficiency is larger than that on the specific power in the region of high temperature ratios.

1. Introduction

Great efforts have been made to improve the performance of a gas turbine in various fields of technology. Among possible measures for such performance improvement, including a method for improving the performances of various equipment and machinery constituting a gas turbine cycle, there are possibilities consisting in lowering the temperature of working fluid at the inlet of compressor and in raising the temperature at the inlet of gas turbine. Between these two possibilities, the latter, that is, cycle performance improvement with a gas turbine inlet temperature established at a higher level has been widely applied particularly by means of developing effective heat resisting materials, adopting turbine blade cooling etc. If raising of the inlet temperature of gas turbine is compared with lowering of that of compressor by an amount of the same temperature difference, the latter is found, as a means to improve the performance of the cycle, more advantageous in general. The authors believe, consequently, that it is necessary to intensify further, as one of the means to improve the performance of gas turbine, the effort to lower the temperature of working fluid at the inlet of compressor, in other words, that of suction air.

Based on the above-mentioned discussion, the authors have made an attempt to improve the performance of gas turbine cycle by means of suction air cooling with a vapor compression refrigerator and in the 1st report, carried out a performance analysis with the performances of various equipment and machinery constituting a gas turbine cycle taken into consideration in order to disclose various possibilities of the cycle and at the same time, though within a limited range, to show its general characteristics (1) . In the present report, the authors calculate the thermodynamic optimum point of this cycle and at the same time show a thermodynamic limit of this combination cycle. The authors also carry out analysis from the standpoint of a combination of normal and reversed cycles, calculate the equivalent atmospheric and maximum temperatures derived from a definition that a performance improvement due to a combination of cycles corresponds to that due to normal cycle only, and propose a temperature criterion for the performance evaluation of the Brayton cycle to demonstrate effects of the combination.

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Nomenclature

- ε : coefficient of performance Eq.(3)
- η : thermal efficiency
- η_{EX} : temperature-efficiency of regenerator $=(T_5-T_2)/(T_4-T_2)$
- Θ_0: rate of equivalent atmospheric temperature drop Eq.(13)
- Θ_3: rate of equivalent maximum temperature rise Eq.(16)
- θ : ratio of suction air temperature drop $=T_1/T_0$
- κ : ratio of specific heats at constant pressure and volume
- λ : specific power
- τ : ratio of maximum and atmospheric temperatures $=T_3/T_0$.

ϕ : pressure ratio $=P_2/P_1$
Φ : pressure ratio $=\phi^{(\kappa-1)/\kappa}$
χ : rate of increase in thermal efficiency or in specific power Eq.(11)
P : pressure, kg/cm^2 abs
t : temperature, $^{\circ}$C
T : temperature, $^{\circ}$K
s : specific entropy, Kcal/Kg$^{\circ}$K
c_p : specific heat at constant pressure, Kcal/Kg$^{\circ}$K
G_a : flow rate of air, Kg/s
L : power, Kg m/s
A : thermal equivaleut of work, Kcal/Kg m

Subscripts

0 : optimum value, maximum value
SA : with suction air cooling (suction air cooling cycle)
CON : without suction air cooling (conventional cycle)
η : thermal efficiency
λ : specific power
τ : ratio of maximum and atmospheric temperatures
eq : equivalent value

2. An analytical cycle and constitutions

The authors here utilize a cycle arrangement, similar to that in the 1st report. Since it is one of the aims in this 2nd report to search a limit which the performance value of this cycle can attain, it is assumed that the compression and expantion processes at the side of Brayton cycle (1/C/E) (normal cycle) are, as shown in Fig. 1, adiabatic, provided that for the reason to be later described the quantity of heat exchange in a regenerator will be prescribed with its temperature-efficiency of regenerator η_{EX} . As vapor compression refrigeration cycle (reversed cycle), a reversed Carnot cycle is utilized, the condensation and evaporation temperatures being equal to those of the atmosphere T_0 and at the compressor inlet on the Brayton cycle side T_1 , respectively. It is also assumed that the values of physical properties are constant for both the normal and reversed cycles neglecting the losses of flow rate and heat. In Fig. 1, the number of each part refers to the following :

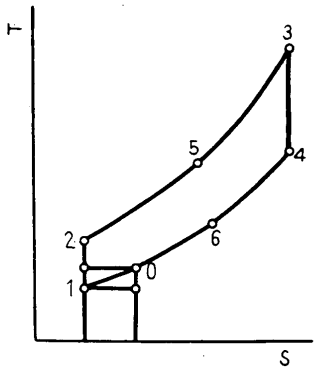

Fig.1 *T-S* **diagram for the gas turbine cycle and the refrigeration cycle**

0: the atmospheric condition, inside the refrigerant condenser
1: inlet of the air compressor, inside the refrigerant evaporator
2: inlet on the air side of the regenerator
3: inlet of the turbine
4: inlet on the gas side of the regenerator
5: inlet of the combustion chamber
6: outlet on the gas side of the regenerator

3. Fundamental equations of the cycle

If the values of various efficiencies except η_{EX} are assumed to be equal to unity in Eqs.(17) and (16) of the preceding 1st report, then the thermal efficiency η and the specific power λ can be written as follows :

$$\eta=\frac{\tau(1-1/\phi^{(\kappa-1)/\kappa})-\{\theta(\phi^{(\kappa-1)/\kappa}-1)+(1-\theta)/\varepsilon\}}{\tau-(1-\eta_{EX})\theta\phi^{(\kappa-1)/\kappa}-\tau\eta_{EX}/\phi^{(\kappa-1)/\kappa}} \quad \cdots\cdots(1)$$

$$\lambda=\frac{AL}{c_pT_0G_a}=\tau(1-1/\phi^{(\kappa-1)/\kappa})-\{\theta(\phi^{(\kappa-1)/\kappa}-1)+(1-\theta)/\varepsilon\} \quad \cdots\cdots(2)$$

The coefficient of performance ε in the refrigeration cycle can be expressed by the following equation on the basis of the conditions established :

$$\varepsilon=\theta/(1-\theta) \quad \cdots\cdots(3)$$

If $\eta_{EX}=1$ in Eq.(1), η is indefinite for $\phi=1$ and this does not suit the purpose of this report, so that η_{EX} will be here taken as an established value. Since under the present conditions establishted, η and λ are monotonous functions of τ, the ratio of the maximum and the atmospheric temperatures, let us consider also τ to be another established value. When τ and η_{EX} are constant, consequently, η and λ are two-variable functions of the pressure ratio Φ and the ratio of suction air temperature drop θ, so that the pressure ratio $\Phi_{0\eta}$ (the optimum pressure ratio) and the ratio of suction air temperature drop $\theta_{0\eta}$ (the optimum ratio of suction air temperature drop) which gives η_0 , the maximum value of η (the maximum thermal efficiency), can be obtained as a solution of the following simultaneous equations (4) and (5), similarly $\Phi_{0\lambda}$ and $\theta_{0\lambda}$ which give λ_0 , the maximum value of λ (the maximum specific power), can be obtained as a solution of the following equations (6) and (7), that is, in the form of Eqs. (8) and (9) :

From $\partial\eta/\partial\phi=0$,

$$\begin{aligned}&\{(2\theta-1)(1-\eta_{EX})-\tau\theta\eta_{EX}\}\Phi^2\\&+2\theta\tau(2\eta_{EX}-1)\Phi+\tau[\tau(1-\eta_{EX})\\&-\{\theta-(1-\theta)^2/\theta\}\eta_{EX}]=0 \quad \cdots\cdots(4)\end{aligned}$$

From $\partial\eta/\partial\theta=0$,

$$\begin{aligned}&[\tau\{(2-\Phi)\eta_{EX}-1\}+2\Phi(1-\eta_{EX})]\theta^2\\&-2(1-\eta_{EX})\Phi\theta-\tau(\eta_{EX}/\Phi-1)=0 \quad \cdots\cdots(5)\end{aligned}$$

From $\partial\lambda/\partial\phi=0$,

$$\Phi-\sqrt{\tau/\theta}=0 \quad \cdots\cdots(6)$$

From $\partial\lambda/\partial\theta=0$,

$$\theta=1/\sqrt{\Phi} \quad \cdots\cdots(7)$$

$$\Phi_{0,\lambda}=\tau^{2/3} \quad \cdots\cdots(8)$$

$$\theta_{0,\lambda}=\tau^{-1/3} \quad \cdots\cdots(9)$$

Where Φ denotes $\phi^{(\kappa-1)/\kappa}$. The solution $\Phi_{0,\eta}$ and $\theta_{0,\eta}$ of the simultaneous equations (4) and (5) was obtained by an iterative method and considered to suit the purpose of the problem when the relative error of each value between two subsequent iterations is less than 10^{-6}. If Φ_0 and θ_0 obtained from the above-mentioned procedure are substituted into Eqs.(1) and (2), the maximum thermal efficiency $\eta_{0,SA}(\tau,\eta_{EX})$ and the maximum specific power $\lambda_{0,SA}(\tau)$ can be obtained. By assuming $\theta=1$ in Eqs.(1),(2),(4) and (6), it is possible to determine these values $\eta_{0,CON}(\tau,\eta_{EX})$ and $\lambda_{0,CON}(\tau)$, which correspond to a conventional cycle without suction air cooling.

Since the Brayton cycle is used in the form of 1/C/E, it is necessary that a relation $T_4>T_2$ holds. If $T_4=T_2$, that is $\theta\Phi=\tau/\Phi$ in Eq.(4), then $\eta_{EX}=0.5$ and the above-mentioned relation $T_4>T_2$ holds within a range $\eta_{EX}>0.5$. On the other hand, Eq.(6) shows that $T_4=T_2$ and no regenerator is necessary in the cycle which exhibits a maximum value of the specific power. Since it appears however that between the values of pressure ratio giving the maxima of the thermal efficiency and specific power, the latter is larger as described in the previous report and to be described later and the pressure ratio in cycle design is in general lower than that corresponding to the specific power, and from the purpose of the present report that a limit value of the cycle performance is to be known, the authors adopt as solution the values obtained from Eqs.(4) and (5) as well as from Eqs.(8) and (9). It is possible to apply these relations to the cases where the gas turbine cycle is operated without suction air cooling.

4. Results and discussion

4.1 Investigation of the optimum point

Figures 2 and 3 show the maximum thermal efficiency η_0 and the maximum specific power λ_0 and the optimum pressure ratio $\Phi_{0,\eta}$ and $\Phi_{0,\lambda}$, respectively, in comparison between the cases with and without the suction air cooling, with the two parameters i.e., the ratio of the maximum and the atmospheric temperatures τ and the temperature-efficiency of regenerator η_{EX}. Figure 4 shows the ratio of optimum suction air temperature drop $\theta_{0,\eta}$ and $\theta_{0,\lambda}$ under suction air cooling. If θ_0 shown in this diagram is substituted into Eq.(3), it is possible to obtain the coefficient of performance ε. It is necessary to cool with the atmoshpere the working fluid (which performs isobaric changes from point 6 to point 0) as shown in Fig. 1 and when the suction air temperature T_1 drops, T_6 also does. Consequently, Fig. 5 is a graphical representation of $\theta_0\Phi_0$ which introduces the following relation while utilizing $T_6>T_2$ in order to confirm $T_6>T_0$:

$$T_6/T_0>T_2/T_0=\theta\Phi>1 \quad \cdots\cdots(10)$$

To illustrate the effect of suction air cooling, Fig. 6 shows the rates of increase in thermal efficiency χ_η and in specific power χ_λ, defined in the following :

$$\left.\begin{aligned}\chi_\eta(\tau,\eta_{EX})&=[(\eta_{0,SA}-\eta_{0,CON})/\eta_{0,CON}]_{\tau,\eta_{EX}}\\ \chi_\lambda(\tau)&=[(\lambda_{0,SA}-\lambda_{0,CON})/\lambda_{0,CON}]_\tau\end{aligned}\right\}\cdots(11)$$

While the rate of increase $\chi_{0,\eta}$ in the maximum thermal efficiency η_0 tends to have a

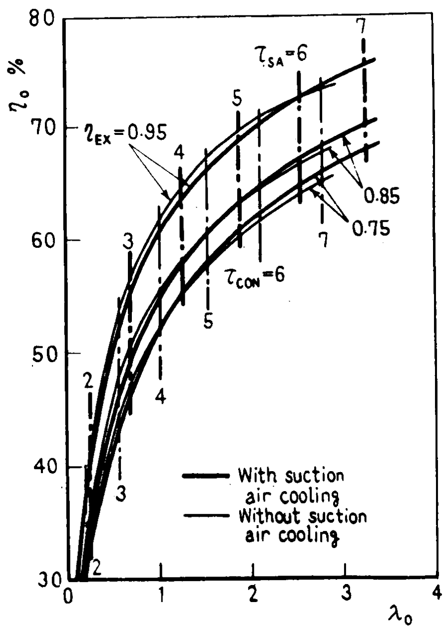

Fig.2 The maximum thermal efficiency and the maximum specific power

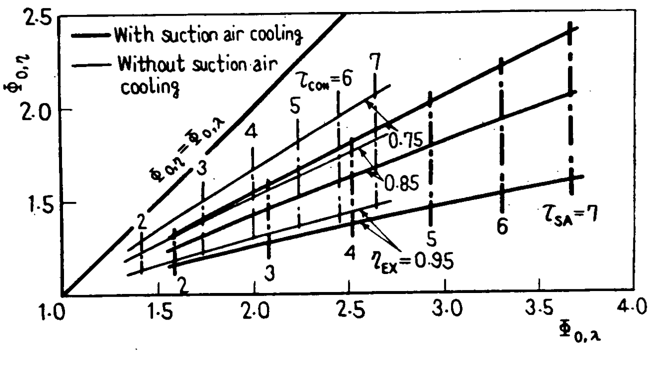

Fig.3 The optimum pressure ratio

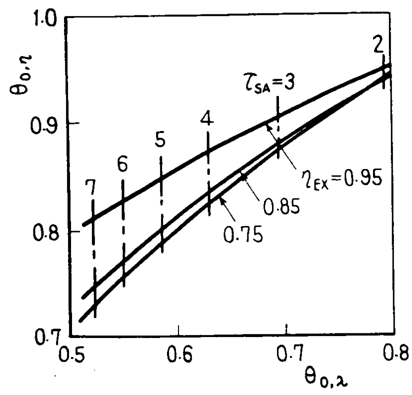

Fig.4 The rate of optimum suction air temperature drop

maximum value as a function of the ratio of the maximum and the atmospheric temperatures τ_{SA}, the rate of increase $\chi_{0,\lambda}$ in the maximum specific power λ_0 has a tendency, as shown in Fig. 6, of monotonous decrease, and the absolute values of increase in thermal efficiency and specific power have, as shown in Fig. 2, similar tendencies. The value of $\chi_{0,\eta}$ increases when the temperature-efficiency of the regenerator η_{EX} decreases. In addition, the value of $\chi_{0,\lambda}$ is high, in other words, the suction air cooling remarkably increases the specific power and it appears that the disadvantage of equipment being complicated as a result of cooling cycle combination can be considerably compensated.

The values Φ_0 are the optimum pressure ratios obtained with and without suction air cooling shown in Fig. 3, and it can be seen that the values of the optimum pressure ratio $\Phi_{0,\eta}$ and $\Phi_{0,\lambda}$ giving the maximum thermal efficiency and specific power are close to each other in a conventional cycle without suction air cooling and that suction air cooling remarkably increases the value of $\Phi_{0,\lambda}$. It can be also seen from Fig. 4 that the ratios of the optimum suction air temperature drop $\theta_{0,\eta}$ and $\theta_{0,\lambda}$ giving the maximum thermal efficiency and specific power approach each other when τ_{SA} decreases and on the contrary the difference between them becomes larger when η_{EX} increases.

Within the range of this calculation, all the values of $\theta_0\Phi_0$ shown in Fig. 5 satisfy the condition of Eq.(10). Without suction air cooling, $T_2/T_0=\Phi$ and its optimum value $\Phi_{0,CON}$ shown in Fig. 3 is larger than $\theta_0\Phi_0$ shown in Fig. 5, so that it can be seen that at the same values of τ and η_{EX}, the exhaust gas temperature T_6 is higher in a cycle without suction air cooling.

4.2 The effect of the combination of normal and reversed cycles

In general, the cycle of a heat engine (normal cycle) under consideration acts between the atmospheric temperature (the lowest temperature of surrounding) and the established maximum temperatures. However, such a normal cycle supported by a reversed cycle as in the present research does not always act within the above-mentioned temperature range and on the normal cycle (Brayton cycle) side the temperature at the inlet of compressor is in general lower than the atmospheric temperature. When a design of the Brayton cycle presupposes the support of a reversed cycle, it is not always necessary to limit the lowest temperature of the cycle to the atmospheric temperature, and then a value, lower than the atmospheric temperature can be established. Consequently, it is possible to represent the properties of normal-reversed cycles with an apparent lowest temperature of cycle as a whole, besides with the thermal efficiency and specific power, used in the previous report and the section 4.1 of this report, and it appears that this value can be useful for the performance evaluation of a cylce in general and the Brayton cycle in particular. Among the following normal-reversed combined cycles classified according to the objects, consequently, (i) " a normal cycle supported by the reversed ", taken up for the purpose of the present research will be considered and the effect of combination corresponding to the performance improvement on the normal cycle side will be discussed with calculated results of an equivalent atmospheric temperature (an apparent lowest temperature) to be defined in the following and an equivalent maximum temperauture (an apparent maximum temperature) introduced under a notion, similar to the above-mentioned :

(i) Cycle for improving the performance of a normal cycle supported by reversed cycle.
(ii) Cycle for improving the performance of the reversed cycle supported by the normal cycle[2].
(iii) Cycle for improving the performance of both the normal and reversed cycles supported mutually by either of them.

4.2.1 Equivalent atmospheric temperature $T_{0,eq}$

For the purpose of determining the equivalent atmospheric temperature, first let τ_{eq} denote the ratio of the maximum and the atmospheric temperatures τ_{CON} of a conventional cycle without suction air cooling, which has the same values as $\eta_{0,SA}$ and $\lambda_{0,SA}$, in comparison with the ratio of the maximum and the atmospheric temperatures τ_{SA} of a cycle with suction air cooling. That is

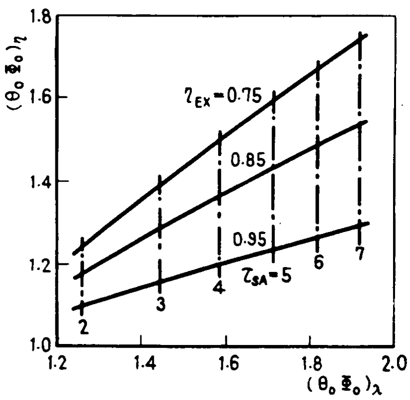

Fig.5 $\theta_0\Phi_0$ giving the maximum thermal efficiency and specific power

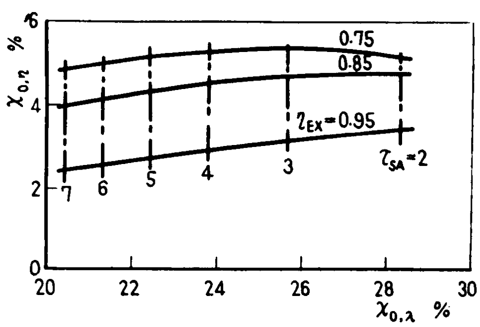

Fig.6 The rates of increase in thermal efficiency and in specific power with the suction air cooling

$$\left.\begin{aligned}&[\eta_{0,SA}(\tau_{SA,\eta})-\eta_{0,CON}(\tau_{eq,\eta})]_{\eta_{EX}=\text{const.}}=0\\&\lambda_{0,SA}(\tau_{SA,\lambda})-\lambda_{0,CON}(\tau_{eq,\lambda})=0\end{aligned}\right\}\quad\cdots\cdots(12)$$

Using τ_{eq} and arbitrary values of the atmosphric temperature T_0 and the temperature-efficiency of regenerator η_{EX}, the rate of equivalent atmospheric temperature dorp Θ_0 and the drop of equivalent atmospheric temperature ΔT_0 are defined as follows :

$$\left.\begin{aligned}\Theta_{0,\eta}&=(T_0-T_{0,eq,\eta})/T_0=\Delta T_{0,\eta}/T_0\\&=(\tau_{eq,\eta}-\tau_{SA})/\tau_{eq,\eta}\end{aligned}\right\}\quad\cdots\cdots(13)$$

$$\left.\begin{aligned}\Theta_{0,\lambda}&=(T_0-T_{0,eq,\lambda})/T_0=\Delta T_{0,\lambda}/T_0\\&=(\tau_{eq,\lambda}-\tau_{SA})/\tau_{eq,\lambda}\end{aligned}\right\}\quad\cdots\cdots(14)$$

where $T_{0,eq}$ is the equivalent atmospheric temperature, which is determined as follows :

$$T_{0,eq,\eta}=T_0-\Delta T_{0,\eta},\quad T_{0,eq,\lambda}=T_0-\Delta T_{0,\lambda}\cdots(15)$$

Figure 7 shows Θ_0 and Fig. 8 shows ΔT_0 at the atmospheric temperature, for example, t_0= 15°C.

In Fig. 7, the effect due to the support of a reversed cycle is more remarkable in specific power than in thermal efficiency below a straight line $\Theta_{0,\eta}=\Theta_{0,\lambda}$, resulting in a condition, where the lowest temperature of cycle is further lowered. When τ_{SA}, that is, the maximum temperature rises, $\Theta_{0,\eta}$ becomes larger than $\Theta_{0,\lambda}$, the effect due to suction air cooling appears, if estimated from the working temperature width of cycle, more remarkably in thermal efficiency, thus the features of a cycle under suction air cooling, different from the representation with η and λ in Fig. 6, can be observed and based on the above-mentioned it can be seen that the support of reversed cycle becomes more important as the high-temperature technology advances and the other methods of improving the cycle are developed. In addition to the representation of performance with the thermal efficiency and the specific power, that with the working temperature of cycle by means of such Θ_0 as this demonstrates the possibility of suction air cooling through vapor compression refrigerator [3] in addition to the cycle investigated in the present research, considers comprehensively the cycles to facilitate thier performance comparison, gives the evaluation of cycle performance a temperature reference etc. and in this way it appears that this is a method of representation, which is useful for showing the effect of combined normal-reversed cycles. From Fig. 8 where Θ_0 is represented by temperature, it is possible to know concretely suction air cooling, that is, the apparent lowest temperature of normal cycle supported by the reversed one and it can be seen that $T_{0,eq}$ is considerably lowered with an increaseing τ_{SA} .

4.2.2 Equivalent maximum temperature $T_{3,eq}$

Using τ_{eq} determined from Eq.(12), according to the preceding paragraph, define the following rate of equivalent maximum temperature rise Θ_3 and the rise of equivalent maximum temperature ΔT_3 for arbitrary values of the maximum temperature T_3 and the temperature-efficiency of regenerator η_{EX}:

$$\left.\begin{aligned}\Theta_{3,\eta}&=(T_{3,eq,\eta}-T_3)/T_3=\Delta T_{3,\eta}/T_3\\&=(\tau_{eq,\eta}-\tau_{SA})/\tau_{SA}\\\Theta_{3,\lambda}&=(T_{3,eq,\lambda}-T_3)/T_3=\Delta T_{3,\lambda}/T_3\\&=(\tau_{eq,\lambda}-\tau_{SA})/\tau_{SA}\end{aligned}\right\}\quad\cdots\cdots(16)$$

where $T_{3,eq}$ is the equivalent maximum temperature, which can be determined as follows :

$$T_{3,eq,\eta}=T_3+\Delta T_{3,\eta},\quad T_{3,eq,\lambda}=T_3+\Delta T_{3,\lambda}\quad\cdots\cdots(17)$$

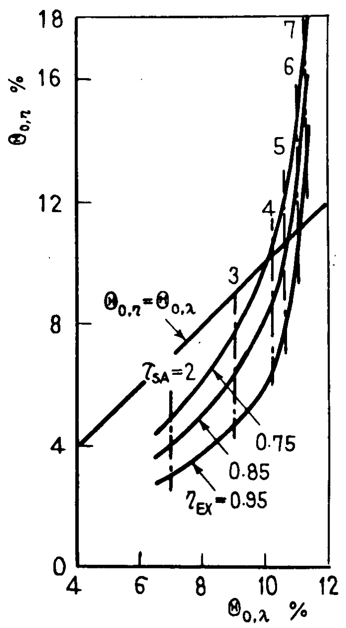

Fig.7 The rate of equivalent atmospheric temperature drop

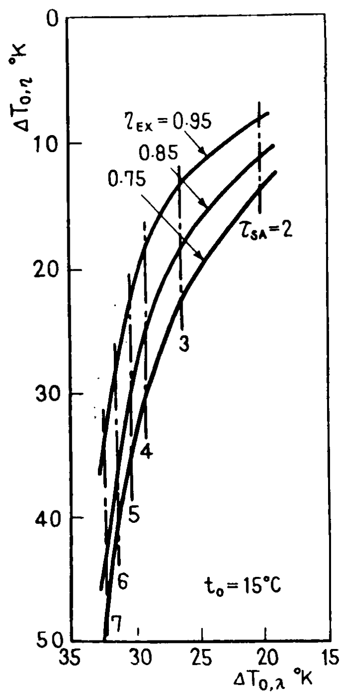

Fig.8 The drop of equivalent atmospheric temperature

Figures 9 and 10 show respective values of Θ_3 and ΔT_3. Θ_3 shown in Fig. 9 has a tendency, similar to Θ_0 shown in Fig. 7. $\Delta T_{3,\eta}$ in Fig. 10 remarkably rises with an increasing τ_{SA} and this fact suggests that it is necessary not only to more improve the high-temperature technology of gas turbine blade but also to introduce actively a support by reversed cycle, for example, suction air cooling with a vapor compression refrigerator.

5. Conclusions

The authors investigated the thermodynamic optimum point in a suction air cooling gas turbine cycle with a vapor compression refrigerator and determined a thermodynamic limit, which this cycle can attain, concerning the ratio of the maximum and the atmospheric temperatures as well as the temperature-efficiency of regenerator. From the standpoint of normal-reversed combination cycle, they also investigated the effect of the reversed cycle on the normal and obtained the following conclusions besides those shown in the 1st report :

(1) With suction air cooling, the maximum thermal efficiency and the maximum specific power monotonously increase with a rise in the ratio of the maximum and the atmospheric temperatures and as for their rates of increase in comparison with those of a cycle without suction air cooling, the rate of increase in thermal efficiency tends to have a maximum, whereas that in specific power has a tendency of nomotonous decrease.

(2) The optimum pressure ratio and the optimum ratio of suction air temperature drop, which give the maximum thermal efficiency and specific power, do not coincide with each other except in some special cases. The optimum pressure ratio is considerably higher than that in a cycle without suction air cooling and the difference of the optimum pressure ratio concerning thermal efficiency from that concerning specific power is larger than the corresponding value in a conventional cycle.

(3) Equivalent atmospheric and maximum temperatures are proposed and the effect of the reversed cycle in comparison with that of the normal is clearly demonstrated. When estimated from the working temperature width of cycle, it can be confirmed that when the ratio of the maximum and the atmospheric temperatures is higher, the effect of the support of reversed cycle appears more remarkably in thermal efficiency and the usefulness of combination of a normal cycle with a reversed one increases with the progress of the high-temperature technology of gas turbine.

Finally, the authors express their sincere gratitude to Prof. T. Saito, Toyohashi University of Technology and Prof. H. Taniguchi, Hokkaido University for their cordial instructions and advices on the present study.

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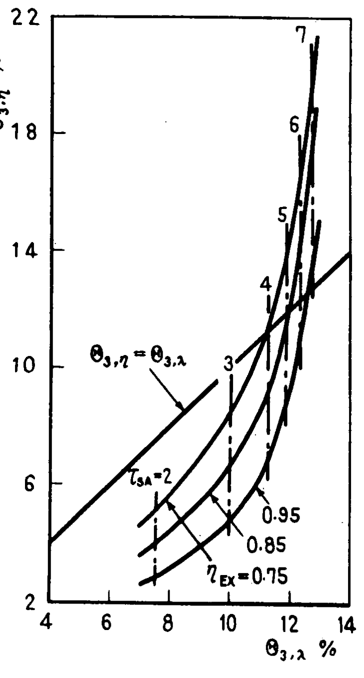

Fig.9 The rate of equivalent maximum temperature rise

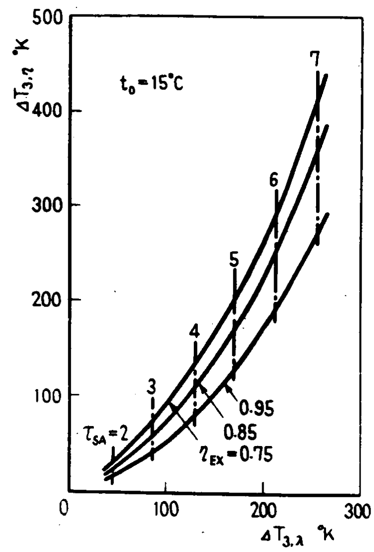

Fig.10 The rise of equivalent maximum temperature